# ANALYSIS OF VIBRATION IN CAM FOLLOWER SYSTEM

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We certify that the project entitled "*Analysis of Vibration in Cam Follower System*" is written by *Mohd Hanaffi Bin Othman*. We have examined the final copy of this project and in our opinion; it is fully adequate in terms of scope and quality for the award of the degree of Bachelor of Engineering. We herewith recommend that it be accepted in partial fulfillment of the requirements for the degree of Bachelor of Mechanical Engineering with Manufacturing Engineering.

.....

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### ANALYSIS OF VIBRATION IN CAM FOLLOWER SYSTEM

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A report submitted in partial fulfillment of The requirements for the award of the degree of Bachelor of Mechanical Engineering With Manufacturing Engineering

Faculty of Mechanical Engineering UNIVERSITI MALAYSIA PAHANG

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iv

### STUDENT'S DECLARATION

I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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To my beloved parents

### SEMEK@HASMAH BINTI MAMAT

To my supervisor Madam Mas Ayu binti Hassan

### To my Academic Advisor Mr. Muhamad Zuhairi Sulaiman

To all FKM's staffs and lecturers

To all my classmates

And To my Special friend out there Siti Norsyahinas Binti Che Man

Thank you for your supporting and teaching.

Thank you for everything that you gave during studies and the knowledge that we shared.

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### ABSTRACT

Cam is a mechanical component that translates movement from circular to reciprocating by using mating component, called the follower. The principal aim of this work is to study and analysis the vibration factor on vertical position cam using heart shape cam. Then, verifies the data using DYNACAM simulated and analysis the effect of impact force on vibration in cam follower system. The parameters such as displacement, velocity and acceleration are involved in finding the optimum force to the system of vibration cam in high speed rotation per minute (RPM). The analysis process will be done using DYNACAM simulation software that able to verify the results from obtained the experiment whether the data are correct or not for principal of vibration in cam follower system. The analysis process also focus on the jump force of follower, that effect of vibration factor in cam mechanism system. During analysis data will be process, high speed will be select from the types of speed of cam in follower system. The speeds that will be used are 300RPM and it is suitable for analyze the vibration factor and more accurate. When high speeds system pumping chemically processed air can have gradually applied loads.

#### ABSTRAK

Sesondol merupakan komponen mekanikal yang menukarkan pergerakan daripada pusingan kepada pergerakan timbal balik dengan menggunakan pasangan komponen yang dikenali sebagai penurut. Matlamat utama kajian ini adalah untuk mempelajari dan menganalisis faktor getaran pada kedudukan tegak menggunakan sesondol berbentuk hati. Kemudian kajian pada seseondol akan mengunakan simulasi DYNACAM dan menganalisis kesan daya impak getaran pada sistem penurut. Parameter seperti jarak, halaju dan pecutan terlibat dalam mencari daya yang sesuai kepada sistem getaran sesondol dengan menggunakan kelajuan pusingan sesondol pada kadar yang tinggi. Parameter yang dipilih untuk eksperimen ini ialah sudut pusingan dan kelajuan sesondol. Setelah menjalankan eksperimen, analisis akan tercapai dengan mengesahkan menggunakan simulasi DYNACAM sama ada keputusan eksperiment adalah betul atau salah dengan bersandarkan prinsip getaran pada sistem sesondol penurut. Proses analisis juga melibatkan kesan daya impak pada penurut, dimana juga memberi kesan kepada kesan getaran sistem sesondol. Pada proses analisis kelajuan sesondol yang digunakan adalah pada kelajuan 300RPM, dimana kesan getaran lebih sesuai untuk dikaji pada kelajuan yang tinggi.

# **TABLE OF CONTENTS**

ii		
iii		
iv		
v		

STUDENT'S DECLARATION	iv
DEDICATION	V
ACKNOWLEDGEMENTS	vi
ABSTRACT	vii
ABSTRAK	viii
TABLE OF CONTENTS	ix
LIST OF TABLES	xiii
LIST OF FIGURES	xiv
LIST OF EQUATIONS	xvii
LIST OF SYMBOLS	xviii

### CHAPTER 1 INTRODUCTION

**EXAMINER'S DECLARATION** 

SUPERVISOR'S DECLARATION

1.1	Project Background	1
1.2	Problem Statement	3
1.3	Objectives	3
1.4	Scope	3

Page

# CHAPTER 2 LITERATURE REVIEW

2.1	Introd	uction	4
2.2	Criteri	Criteria of Cam Follower	
2.3	Туре с	of Disk Cam	9
	2.3.1	Pear Shaped Cam	9
	2.3.2	Circular Shaped Cam	10
	2.3.3	Heart Shaped Cam	11
2.4	Types	of Follower	12
	2.4.1	The Knife Edge Follower	12
	2.4.2	The Flat-faced Follower	13
	2.4.3	The Roller Follower	13
	2.4.4	Spherical Faced Follower	14
	2.4.5	Additional Follower	14
2.5	Cam N	Material	16
	2.5.1	Cast iron	16
	2.5.2	Gray cast iron	16
	2.5.3	Forged steel	17
	2.5.4	Hot Rolled Steel	17
	2.5.5	Cold-rolled Steel	17
2.6	Dynan	nic Modelling	18
	2.6.1	Single degree- of- freedom model (SDOF)	19
	2.	6.1.1 One - mass dynamics models	19
	2.6.2	Constant Velocity Motion	23
	2.6.3	Constant Acceleration Motion	24
	2.6.4	Harmonic Motion	25
2.7	Impac	t Modelling	25

# CHAPTER 3 METHODOLOGY

3.1	Introduction	27
3.2	Flow Chart for Methodology	28
3.3	Dynamic Model	29
3.4	Analysis Parameter	34
	3.4.1 DEWESOFT Software	35
3.5	DYNACAM Software	36
3.6	Energy methods for impact modelling	37

# CHAPTER 4 RESULTS AND DISCUSSION

4.1	Introdu	action	39
4.2	Experi	mental Results	40
	4.2.1	Results for 300RPM	40
	4.2.2	Constant Acceleration Motion	41
4.3	Analys	sis of the experimental result	42
	4.3.1	Displacement Graph.	42
	4.3.2	Velocity Graph	43
	4.3.3	Acceleration Graph.	45
4.4	Graph	Simulated from DYNACAM	47
	4.4.1	Displacement Diagram.	47
	4.42	Velocity Diagram.	49
	4.4.3	Acceleration Diagram.	51

4.5	Comparison Experimental Graph with DYNACAM Graph	54
	4.5.1 Displacement	55
	4.5.2 Velocity	56
	4.5.3 Acceleration	57
4.6	Analysis of Impact Force	58
	4.6.1 Ball Drop Experiment	58
4.7	Comparison Data of Experiment Ball Drop.	63
4.8	Discussion	66

# CHAPTER 5 CONCLUSION AND RECOMMENDATIONS

5.1	Conclusions	67
5.3	Recommendations	68

# REFERENCES

69

APPENDIX		71
A	Result and Data From Experiment Cam Follower	71
B	Table Result From DYNACAM Simulated	75

# LIST OF TABLES

Table No.		Page
3.1	Experiment Data for 300RPM	33
3.2	Types of Sensor	35
4.1	Result from Dewesoft Software at 300RPM	40
4.2	Equation of Constant Acceleration Motion	41
4.3	Ball Drop Experiment Data.	60
4.4	Data Comparison of Experiment Ball Drop	63
4.5	Percent error of D&C and Common Velocity	64

# LIST OF FIGURES

Figure No.		Page
2.1	Automobile Cam-Driven Overhead Valve Train Linkage	5
2.2	Function of Generator	7
2.3	Path Generator cam	7
2.4	Motion-generator cam	8
2.5	Pear Shaped Cam	9
2.6	Circular Shaped Cam	10
2.7	Heart Shaped Cam.	11
2.8	Knife Edge Follower	12
2.9	Flat-face Follower	13
2.10	Roller Follower	13
2.11	Spherical Faced Follower	14
2.12	Radial Cam	15
2.13	Translating Cam	15
2.14	Overhead Valve Linkage	20
2.15	Simplified Valve Train One-Mass Model	21
2.16	Motion Events	23
3.1	Summary of Research Methodology	28
3.2	Equivalent Spring-Mass System	28

xiv

3.3	The Dynamic Model Of One Single Degree Of Freedom	30
3.4	Vertical Cam	38
3.5	Heart Shape Cam	38
3.6	Screen Display from DYNACAM Software	36
3.7	Force vs Deflection	37
4.1	Displacement Diagram	43
4.2	Velocity Diagram	44
4.3	Acceleration Diagram	46
4.4	Displacement diagram from 0°-90	47
4.5	Displacement diagram from 90°-180°.	47
4.6	Displacement diagram from 180°-270°.	48
4.7	Displacement diagram from 270°-360°.	48
4.8	Velocity diagram from 0°-90°.	49
4.9	Velocity diagram from 90°-180°.	49
4.10	Velocity diagram from 180°-270°.	50
4.11	Velocity diagram from 270°-360°.	50
4.12	Acceleration diagram from 0°-90°.	51
4.13	Acceleration diagram from 90°-180°.	52
4.14	Acceleration diagram from 180°-270°.	52
4.15	Acceleration diagram from 270°-360°.	53
4.16	DYNACAM Single dwell Translating of Cam follower System.	54
4.17	Experiment vs DYNACAM Comparison Displacement.	55

4.18	Experiment vs DYNACAM Comparison Velocity	56
4.19	Experiment vs DYNACAM Comparison Acceleration	57
4.20	Ball Drop Experiment Setup	59
4.21	Impact Force Comparison	64
4.22	Percent Error Comparison	65

# LIST OF EQUATIONS

Eq. No.		Page
2.1	Equation of Motion One mass Model	21
2.2	Equation of Motion One mass Model	21
2.3	Equation of Motion One mass Model	21
2.4	Equation of Motion One mass Model	22
3.1	Newton Second Law Motion	31
3.2	Newton Second Law Motion	31
3.3	Newton Second Law Motion	31
3.4	Model Single Degree of Freedom	31
3.5	Potential Energy	37
3.6	Force vs Deflection	37
3.7	Model Single Degree of Freedom	38
4.1	Impact Velocity	59
4.2	Impact Velocity	59
4.3	Deflection and Correction Factor	61
4.4	Deflection and Correction Factor	61

xvii

### LIST OF SYMBOLS

Cam Angle Rotation

Angle of Rotation

Follower Displacement

Instantaneous Follower Velocity

Instantaneous Follower Acceleration

 $\Delta \mathbf{R}$  Instantaneous Follower Displacement At Time

Speed of the Cam

- K Spring Stiffness Deflection
- **F** Force

Mass of Follower

Damping Coefficient

Natural frequency

### **CHAPTER 1**

#### INTRODUCTION

#### 1.1 **Project Background**

Cam is an element of the cam-follower mechanical system that compels the movement of the follower by direct contact (Horald A.Rothbart ,2004). Cam is a mechanical component that translates movement from circular to reciprocating by using mating follower. A cam can be defined as a device that having a curved outline or a curved groove that usually called as cam profile. A common example is the camshaft of an automobile, which takes the rotary motion of the engine and translates it into the reciprocating motion necessary to operate the intake and exhaust valves of the cylinders. The cam mechanism may be modeled as a three mass system (leading element, cam and follower) with three degrees of freedom (displacement of the leading element, cam and follower). Due to complexity of such a model, usually, the mechanism is divided into two systems: leading element-camshaft-cam system and cam-follower system which are considered as one-degree-of-freedom systems.

The motion of the follower is the result of the program. Just as a computer program, so is a cam. Thus, the system can be thought of as a mechanical information device. Accordingly, the goals of the designer is to build a program, establish the locus of the contact point between the cam and follower, produce the cam profile coordinate system, and fabricate the cam within an acceptable accuracy. After all the parts are assembled the performance of the cam-follower system is observe.(Horald A.Rothbart ,2004).

There are three types of cam followers, and each type of the follower influences the profile of the cam. The three types are the knife-edge, the roller follower and the flat face follower. The follower restraint to the cam is positive-driven by the use of rollers in the cam groove or multiple conjugate cams, is spring-loaded, or occurs by gravity.

In the cam-type transfer unit, transferring motion, which consists of feeding, lifting and clamping, is actuated by the feed cam, lift cam, and clamp cam, respectively. Cam is mechanism that can reliably perform a repetitive and complex motion at high velocity. The feed cam, lift cam and clamp cam are equipped to one shaft, which is synchronized with the pressing motion of the press machine. Therefore, it is ensure that the transportation system taken outside than die path before the tools are closed.

The most of the journal consider the vibrations of the cam-follower system which is assumed as an oscillator with a mass and a spring. The return system of the follower contains a spring and a damper. The oscillator is excited with the function of shape of the cam which depends on the angle of rotation of the cam. This basic model is extended including the Columb friction at the rocker arm pivot and the Hertzian contact between the follower and cam. To reduce the sensitivity of the follower motion on the parameter variations the optimal design methods for cam curve are developed. Unfortunately, the results are obtained by omitting the influence of the camshaft and leading element.

#### **1.2 Problem statement**

One of the many potential problems with unwanted vibrations in high-speed machinery is the possible introduction of follower jump in a cam-follower mechanism. Jump is a situation where the cam and follower physically separate. When they come back together the impact introduces large forces and thus large stresses, which can cause both vibrations and early failure of the mechanism. Many companies are now conducting indepth vibration analyses on their existing machines and redesigning many stations to reduce the overall vibrations in the machine.

#### 1.3 Objective

- **1.3.1** Study and analysis the vibration factor and verify using DYNACAM 1998 Simulation
- **1.3.2** Analysis the effect of impact force on vibration in cam follower system

### 1.4 Scope

A study was conducted to find the optimum equation of motion for the system from the dynamic model. The equation relates the cam displacement, velocity and acceleration to analyze and prove the vibration system of cam follower. The velocity, acceleration and displacement also involved to investigate the optimum force to the cam system on vibration in high speed rotation per minute (RPM).

#### **CHAPTER 2**

#### LITERATURE REVIEW

#### 2.1 Introduction

Cam is a versatile, specially shaped part of a machine that is always in contact with a member a called the follower. The name cam should not be confused with the common abbreviation cam for camera and camcorder, both used in the fields of photography and video, nor with the acronym CAM applied to computer applied to computer-aided manufacturing, which utilizes computational facilities for machinery fabrication of all kinds.

Many different types of cam profile are designed and manufactured depending on a machine's requirements (P.W Jensen, 1987). Cam is a part of a rotating wheel or shaft that strikes a lever at one or more points on its circular path. The cam is in most cases merely a flat piece of metal that has had an unusual shaped or profile machined onto it.

Many studies on the cam mechanisms concern the problem of vibrations. As machine speed increases the problem of vibrations of the cam mechanism has the more significant importance. The vibration level has the influence on the wear rate, noise level and service life of the cam actuated machines and devices and also to the precision operation of machines. Because of that it is important to understand the cause of vibrations and provide means to control or to minimize unwanted vibrations so that desirable system response characteristics may be predicted and obtained. Cam follower mechanism are found in almost all mechanical device and machine for example in agriculture, transportation equipment, textiles, packaging, machine tools, printing press, automobile internal combustion engines, food processing machines, switches, ejection molds, and control systems, and more recently in micro machines such as micro electromechanical system[MEMS]. Figure 2.1 showed the automobile cam-driven overhead valve train linkage



Figure 2.1 Automobile cam-driven overhead valve train linkage Source: Harold a.Rothbart, Cam Design HandBook, 2003.

The motion of the follower as a function of cam curve and parameters of the cammechanism are determined. The special attention is given to analysis of the cam velocity, damping properties of the camshaft and mass ratio of the follower and cam. As an example the vibrations of the cam mechanism with polynomial cam-curve are investigated.

The mechanism consists of a leading element, an elastic cam-shaft, a heavy cam and an elastic follower. The leading force and the force of the follower act. If the shaft which connects the leading system and the cam is rigid, the model is a system with one degree of freedom. The generalized coordinate is the displacement of the cam. The cam has a profile which causes the follower to move in certain manner. The differential equation of motion is the second order, non-linear and with time variable coefficients one. For some parameters of the cam mechanism the differential equation of motion is with small parameters. Then the approximate analytic solution of the equation is obtained.

For the small velocity of cam motion in comparison the other parameters (approximately 0.5) the strong non-linear differential equation with slowly varying parameters is solved using the elliptic-Krylov-Bogolubov method. Analyzing the obtained solution it is obvious that the amplitude of vibrations depend on the velocity of cam motion and the relation between the masses of the follower and the cam. For higher velocity of cam motion the accuracy of follower motion is smaller. It is recommended the mass of the cam to be decreased in comparison to the mass of the follower. The amplitude variation depends on the cam profile, too.

The vibration properties of the system depend on the parameters of the mechanism. The damping properties of the cam-shaft have a significant influence on the vibrations of the system. For higher values of the damping coefficient of the connecting shaft the vibrations of the mechanism are smaller and the motion of the follower differs only a bit from the projected theoretical one. Namely, for higher damping coefficient the motion of the follower is much more accurate than for smaller damping coefficient.

#### 2.2 Criteria of Cam Follower

The cam-follower system may be designed for path, motion, or function generation. This book treats the cam and the follower almost totally as a function generator in which the output of the follower is a function of the cam input illustrate (Erdman and Sandor, 1997) the three types of cam design function in Figure 2.2 until 2.4.

Figure 2.2 shows a function of generator in which the cam drives a four bar linkage to a type slug bar of an electric type writer. Figure 2.3 shows a path generator cam that uses a double cam to produce a line that is desired. (Erdman and Sandor. 1997).



**Figure 2.2:** Function of Generator Source: Erdman and Sandor. 1997.



**Figure 2.3:** Path Generator cam Source: Erdman and Sandor. 1997.



**Figure 2.4:** Motion-generator cam Source: Erdman and Sandor. 1997.

Figure 2.4 showed a motion-generator cam in which a drift meter operates to define the aircraft direction of motion relative to ground. The sight wire is aligned to follow an object on the earth that passes through the center. The instrument rotates about point 0 by two fixed guiding in a circular arc-shaped cam slot about its center. Without this design a physical pivot needed of point 0 would hinder the vision. (Erdman and Sandor. 1997).

#### 2.3 Type of Disk Cam

#### 2.3.1 Pear Shaped Cam

This type cams as shown in Figure 2.5 is often used for controlling valves. For example, they are used on motor car camshafts to operate the engine valves. A follower controlled by a pear-shaped cam remains motionless for about half a revolution of the cam. During the time that the follower is stationary, the cam is in a dwell period. During the other half revolution of the cam, the follower rises and then falls. As the pear-shaped cam is symmetrical, the rise motion is the same as the fall motion.



**Figure 2.5:** Pear Shaped Cam Source: Yi Zhang with Susan Finger and Stephanie Behrens

#### 2.3.2 Circular Shaped Cam

This type cams are shown in Figure 2.6 is sometimes called eccentric cams. The cam profile is a circle. The center of rotation of the cam is often from the geometric center of the circle. The circular cam produces a smooth form of motion called a simple harmonic motion. These cams are often used to produce motion in pumps. Circular cams are often used to operate steam engine valves. As the cam is symmetrical, the rise and fall motions are the same.



**Figure 2.6:** Circular Shaped Cam Source: Yi Zhang with Susan Finger and Stephanie Behrens

#### 2.3.3 Heart Shaped Cam

Figure 2.7 showed the cam motion is effect the follower to move with a uniform velocity. Heart-shaped cams are essential when the follower motion needs to be uniform or steady as, for example, in the mechanism that winds thread evenly on the bobbin of a sewing machine. They are used in systems that require this uniform movement, such as the mechanism that winds the tread evenly onto a bobbin or winds wire onto a solenoid coil.



**Figure 2.7:** Heart Shaped Cam Source: Yi Zhang with Susan Finger and Stephanie Behrens

#### 2.4 Types of Follower

The cam follower system can be classified by referring to the follower or the cam both. The follower movement is translation, oscillation or indexing. The follower restraint to the cam is positive-driven by the use of roller in cam groove or multiple conjugate cams, is spring-loaded, or occurs or gravity.

Also, the translating follower line of motion with reference to the cam center may be radial or offset. The follower shape can be categories in four, which is knife-edge follower, flat follower, roller follower and spherical faced follower. (Horald A.Rothbart, 2004).

### 2.4.1 The Knife Edge Follower

A knife edge follower as the name implies in Figure 2.8 is formed to a pint and drags the edges of cam. This is simplest type, is not often used due to the rapid rate of wear. It is employed in design as the center of the roller follower.



**Figure 2.8:** Knife Edge Follower Source: Harold a.Rothbart, Cam Design HandBook, 2003.

#### 2.4.2 The Flat-faced Follower

Figure 2.9 show a flat-faced follower that consists of a follower that is formed with large, flat surface to contact the cam. This type of follower can be used with steep cam motion and do not jam. Usually, this cam is used when quick motions are required.



**Figure 2.9:** Flat-face Follower Source: Harold a.Rothbart, Cam Design HandBook.2003.

### 2.4.3 The Roller Follower

A roller follower consists of follower that has a separate part, a roller that is pinned to the follower stem as shown in Figure 2.10. This is most commonly used follower because the friction and contact stress are lower than knife edge follower. However, it can possibly jam during steep cam displacement.



**Figure 2.10:** Roller Follower Source: Harold a.Rothbart, Cam Design HandBook.2003

#### 2.4.4 Spherical Faced Follower

Figure 2.11 showed spherical faced follower consists of a follower formed with a radius face that contact the cam and this follower can be used with steep cam motion without jamming. However, the frictional force of this follower greater than roller follower.



**Figure 2.11:** Spherical Faced Follower Source: Harold a.Rothbart, Cam Design HandBook.2003.

#### 2.4.5 Additional Follower

The offset follower is one in which the axis of the follower movement is displaced from the cam center of rotation. Offsetting often improve action by reducing forces, stress and also the cam's size. The eccentricity should be in direction that improves force components tending to jam the translating follower in its bearing guide. Figure 2.12 showed a follower on a radial cam with an offset shown. Figure 2.13 showed the same relationships for a translating cam. Both of cam, the follower path is not the profile displacement of the cam.



Figure 2.12: Radial Cam Source: V, Ryan, 2003.



**Figure 2.13:** Translating Cam Source: V, Ryan, 2003.
# 2.5 Cam Material

Cam are usually made from a strong and hard materials because to avoid wear. Basically, the four kinds of wear in cam follower mechanism are: adhesive wear, abrasive wear, corrosive wear, and surface fatigue wear (Horald A.Rothbart ,2004). The most commonly used cam materials are cast iron and steel. (Robert L.Norton, 2002)

#### 2.5.1 Cast iron

Cast iron constituted a whole family of material. Their main advantages are relatively low cost and ease of fabrication. Some are weak in tension compared to steel but like most cast material, have good compressive strength their densities are slightly lower than steel at about 6920 kg/m<sup>3</sup>. Most cast iron not exhibit a linear stress-strain below the elastic limit and do not obey Hooke's law. (Robert L.Norton, 2002)

#### 2.5.2 Gray cast iron

This iron is most commonly used to form of cast design. Its graphite flakes gives its gray appearance and name. The ASTM grades gray cast iron into seven classes based on the minimum tensile strength in kpsi. Class 20 has a minimum tensile strength of 20 kpsi(138Mpa). The class number 20,25,30,35,40,50 and 60 then represent the tensile strength in kpsi. This alloy is easy to pour , easy to machine and offer good acoustical damping. (Robert L.Norton, 2002)

# 2.5.3 Forged steel

Large cams or complex shapes such as IC engine camshaft are often from by hot forging a steel billet to an approximate shape for later machining. If sufficient quantity is required to offset the cost of forging dies significant saving of machining time can be realized over starting each cam with billet. Also, the strength of forged part especially against fatigue loading can be superior to that of cam made from billet. (Robert L.Norton, 2002)

# 2.5.4 Hot Rolled Steel

This hot rolled steel is produced by forcing hot billets of steel through set of roller or dies that progressively changed their shaped into I-beam, channel section, angle iron, etc. The surface finish of this material is rough due to oxidation at the elevated temperature. The mechanical properties are also relatively low because the material ends up in annealed or normalized state, unless deliberately heat-treated later. (Robert L.Norton, 2002)

# 2.5.5 Cold-rolled Steel

This steel is produce from a billet, the shape of cold rolled steel are brought to final form and size by rolling between hardened steel roller or drawing through dies at room temperatures. The result is a material with good surface finish and accurate dimension compared to hot rolled material. Its strength and hardness are increase at the expense of significant built in strain which can later be release during machining, welding or heat treatment. (Robert L.Norton, 2002)

# 2.6 Dynamic Modeling

Dynamic modeling is a mathematical tool that is used to describe the behavior of physical systems. These systems may be represented by single or multiple differential equations and may be a mechanical, electrical, thermal, or any other time-varying system. In this particular case, only dynamic models for mechanical systems are considered.

Every real mechanical system has infinite degrees of freedom. The higher the degree of freedom in the model, the more accurate the simulation will be, at the price of model complexity and computation time. In order to have a reasonable computation time and acceptable results, the model needs to be simplified. This simplification may be done by reducing the degrees of freedom by combining masses, stiffness constants, and damping coefficients. The simplest dynamic model is a single degree of freedom model with one mass, one spring, and one damper. More complex models have multiple degrees of freedom with multiple masses, springs, and dampers. Simplifications of complex models to simple models are shown in the following sections.

The application of dynamic modeling to cam-follower systems was first seen in the automotive industry in 1953 when a single-degree-of-freedom dynamic model was created with good correlation between experimental and simulated data [Barkan 1953]. Superior correlation was obtained when a twenty-one degree-of-freedom dynamic model was created for the valve-train system [Seidlitz 1989]. The disadvantage of the latter model was a longer modeling and computational time. Other applications included modeling of a robotic arm with impact (Ferretti et al 1998) and modeling of industrial cam-follower systems (Livija Cveticanin, 2006)

By creating a dynamic model, the designer is able to determine the behavior of a system prior to expensive manufacture, assembly, and testing. If the requirements are not met, appropriate fundamental changes may be made early on in the product cycle to obtain acceptable behavior.

#### 2.6.1 Single degree- of- freedom model (SDOF)

A single degree of freedom (SDOF) model is the simplest dynamic model. An SDOF model can have one or two lumped masses and is typically used as a quick approximation of the dynamic behavior of a system prior to increasing the complexity of the model for a more accurate analysis. The advantages and disadvantages of one-mass and two-mass SDOF models are discussed at the end of this subsection. (Livija Cveticanin, 2006)

#### 2.6.1.1 One - mass dynamics models

One-mass SDOF model is a simplified model used to predict the dynamic behavior of the motion of a system. The application and derivation of one-mass SDOF model was explicitly shown in 1953 by Barkan. Prior to Barkan's work, there were limited uses of dynamic models in the simulation of mechanical systems in the automotive industry. A dynamic model was developed for the high-speed motion of a cam-actuated engine valve and overhead valve linkage shown in Figure 2.14



Figure 2.14: Overhead valve linkage Source: (Barkan 1953)

To simplify the system shown in Figure 2.14, Barkan divided the valve-train into several concentrated masses, and then relocated the masses to the valve head's axis of translation using the appropriate lever ratios to create one lumped mass. Once the lumped parameters were obtained, equations of motion were developed. To create the equations of motion, the forces acting on the system were identified.

These comprised the spring force, inertia force, linkage compression force, friction force, and gas force. Barkan resolved the spring force into valve spring compression force, valve spring preload force, and the force produced due to the vibration of the springs. Three types of friction were taken into account for the damping, namely coulomb friction, viscous friction proportional to relative velocity, and viscous friction proportional to absolute velocity. The most complex portion of the equation was determined to be the gas force, which occurred when there was a difference in pressures. Barkan excluded spring vibration and gas force from the equations of motion because the spring surge had been determined to be insignificant (Oliver and Mills 1945). Other authors disagreed with the elimination of the spring surge and stated that unacceptable errors may occur (Philips, Schamel, and Meyer 1989). The gas force was very complex to model and would have required experimental data which was not readily available, therefore it was neglected. The equations of motion were created for the simplified one mass model show in Figure 2.15.



Figure 2.15: Simplified Valve Train One-Mass Model Source: (Barkan 1953)

Depending on the cam and follower contact condition, one of the following equations would be calculated using the notation in Figure 2.15.

$$"+2$$
  $'+$   $2 = - ()$  (2.1)

$$"+2$$
  $'+$   $2 = - ()$  (2.2)

$$"+2$$
  $'+$   $2 = - ()$  (2.3)

Where is displacement, 'is velocity, "acceleration, and  $\zeta$ i are the critical damping factors.

= ---, = --- (2.4)

Equations 2.1 2.2, 2.3 and 2.4 can be applied when  $\cdot > 0$  (valve opening),  $\cdot < 0$  (valve closing), and x < 0 (valve jumps), respectively. The variable x represents the displacement of mass M with respect to equivalent cam. Barkan's work proved the validity of a mathematical model when he compared the simulated results with the experimental result. Other works that followed tried to improve upon different aspects of the modeling in attempt to increase accuracy or obtain a better understanding of the problems. Most of the work tried to increase accuracy by increasing the degree of freedom of the spring model (Pisano et al. 1983, Seidlitz 1989).

In 1995, Barkan and Dresner determined that the single degree of freedom model is satisfactory as long as it meets two conditions:

- 1. The excitation amplitudes near the first mode frequency are significantly greater than those at the second mode frequency.
- 2. The higher mode vibrations are not able to build up over time to high magnitudes.

Condition 1 was proven by Barkan in 1953, while condition 2 is true for most camfollower systems where the follower rests on the cam through a large portion of the cycle or the excitations were low and internal damping was enough to damp the excitations. While many researchers have utilized the one-mass SDOF model to perform their analysis (Barkan 1953, Mendez-Adriani 1983, Matsuda 1990), others venture into the two-mass SDOF model and multiple degrees of freedom (MDOF) models (Barkan and Dresner 1995, Matsuda 1990). The advantage of utilizing a one-mass SDOF model is simplicity. However, a onemass model does not predict the valve jump accurately. An extreme case of inaccuracy of a one-mass SDOF model was presented in 1983 by Mendez-Adriani. They found that after optimizing the system, it was possible for a one-mass SDOF model to operate at any speed without occurrences of jumps, which was not possible in practice. Therefore, a multi-mass model was created to eliminate this possibility.



**Figure 2.16**: Motion events Source: Yi Zhang with Susan Finger and Stephanie Behrens

**Constant Velocity Motion** 

2.6.2

If the motion of the follower were a straight line, Figure 2.16 (a) (b) (c), it would have equal displacements in equal units of time, for example uniform velocity from the beginning to the end of the stroke, as shown in b. The acceleration, except at the end of the stroke would be zero, as shown in c. The diagrams show abrupt changes of velocity, which result in large forces at the beginning and the end of the stroke. These forces are undesirable, especially when the cam rotates at high velocity. The constant velocity motion is therefore only of theoretical interest

# 2.6.3 Constant Acceleration Motion

Constant acceleration motion is shown in Figure 2.16 (d) (e) (f). As indicated in e, the velocity increases at a uniform rate during the first half of the motion and decreases at a uniform rate during the second half of the motion. The acceleration is constant and positive throughout the first half of the motion, as shown in f, and is constant and negative throughout the second half. This type of motion gives the follower the smallest value of maximum acceleration along the path of motion. In high-speed machinery this is particularly important because of the forces that are required to produce the accelerations.

When  $0 \le \phi \le \frac{\beta}{2}$ 

$$S(\phi) = 2\hbar \frac{\phi^2}{\beta^2}$$
$$V(\phi) = \frac{4\hbar}{\beta^2} \phi$$
$$A(\phi) = \frac{4\hbar}{\beta^2}$$



$$S(\phi) = \hbar - \frac{2\hbar}{\beta^2} (\beta - \phi)^2$$
$$V(\phi) = \frac{4\hbar}{\beta} \left(1 - \frac{\phi}{\beta}\right)$$
$$A(\phi) = \frac{4\hbar}{\beta^2}$$

# 2.6.4 Harmonic Motion

A cam mechanism with the basic curve like g in Figure 2.16 (g) will impart simple harmonic motion to the follower. The velocity diagram at h indicates smooth action. The acceleration, as shown at i, is maximum at the initial position, zero at the mid-position, and negative maximum at the final position.

# 2.7 Impact Modeling

Impact modeling is the modeling of impact force that occurs due to a collision of a rigid and elastic, or of two elastic bodies. The first time that impact was considered in the valve-train model was in 1953 by Barkan. Although Barkan knew that the valve head would impact the valve seat, he ignored the impact events and assumed the first impact event as the end of the analysis. He deemed the impact due to valve seating as irrelevant to the motion of the valve head during the period of interest. The valve-seat condition was also excluded by Matsuda et al. (1990) in their analysis of the five mass MDOF model. In 1995, Dresner and Barkan also neglected the seat flexibility from their analysis because they felt that the seat deflection contributes only during lift off and seating.

They considered valve seating as the end of their analysis, which was a justification for the exclusion. Even though they excluded the valve seating impact, they included the cam jump with a coefficient of restitution of zero. This allowed researchers to determine the jump conditions but not the effects of impact, which may be appropriate depending on the motivation of the analysis. Pisano and Freudenstein, 1983, included valve-seat flexibility in their model but their most important contribution was the distributed parameters of the valve spring, which gave a more accurate result. The impact modeling technique shown in the published literature used the Heaviside step function, which was easily implemented and gave accurate results. The valve-seat was represented by a stiffness constant and viscous damping in the dynamic model. The calculations of how these values were obtained were also excluded from the literature, which made it hard to follow. Seidlitz continued the inclusion of the valve-seating as well as increasing the complexity of the valve spring.

# **CHAPTER 3**

#### METHODOLOGY

#### 3.1 Introduction

The methodology used in this study is to obtain the optimum result for experimental process on disk cam profile by using *DEWESOFT* Software. There are two main procedures attempted in this project, calculate the three factors of vibrations, displacement, velocity and acceleration and after that verify the calculation from experiment using DYNACAM Software to simulate at 300RPM. Step one was to using the constant acceleration of motion. Using dynamic modeling techniques, a cam follower system was design one mass single degree of freedom (SDOF)

A complete dynamic analysis of the system was then carried out, consisting of creating the dynamic model and deriving the equations of motion. Using the parameter of dynamic modeling, mass, stiffness and damping, one mass model the equations of motion were taken directly from "The Cam Design and Manufacturing Handbook" by R.L. Norton. The differential equations then needed to be solved and the output motions calculated. Using the contact force equation, the jump phenomenon was predicted the impact force of cam follower system.

# 3.2 Flow Chart for Methodology



Figure 3.1: Summary of Research Methodology

From Figure 3.1 showed the summary of research methodology for this project. From get the title, start with studying and researching the background of this project. Before searching the source and references for guide and as a input for making the project, the project scope and objective of the project must be clear. After that, learn the manual book DEWESOFT Software for setup the experiment and consider the parameter that want be used. Then, make the experiment and get the data from the experiment focus on the parameter what want to be analysis. Analyze the data using calculation and if having a error during calculation turn back to run the experiment. After finished analysis the data, verify the data using DYNACAM Software to check the error during calculation and the same step like before, turn back to calculate again the data if the result analysis having a error. After finish this step, make a conclusion for the result that will be analyze and give more recommendation for improve the project.

#### **3.3 Dynamic Model**

In this study, two experiments will be conduct to complete the study for stability analysis for the cam follower system. The first experiment only defined the parameters of displacement, velocity and acceleration. The effect of stiffness and damping for the equation of motion single degree of freedom spring-mass system are taken into consideration. The dynamic model is preferred for the analysis to computes the parameters. Figure 3.2 showed the model of the system.



Figure 3.2: Equivalent spring-mass system

Source: Singiresu S.Rao (2005)

Using constant acceleration of motion, will consider the displacement, velocity and acceleration. The calculation for three of parameters are consider depends on the data that collect from DEWESOFT. The procedure need to consider is select suitable coordinate to describe the linear motion of a point mass or the centroid of a rigid body, an angular coordinate to describe the angular motion of a rigid body.

The static equilibrium configuration will determine of the system and measure the displacement of the mass or rigid body from static equilibrium position. Newton's second laws applying in this free body diagram. (Singiresu S.Rao, 2005)

Newton's second laws of motion can be stated as follows:

$$\vec{()} = - \frac{\vec{()}}{()} \tag{3.1}$$

Thus, if mass is displaced a distance x(t) when acted upon by a resultant force F(t) in the same direction. If mass m is constant, this equation reduces to

$$\vec{()} = -\vec{()} = \vec{()}$$
(3.2)

Where

$$= \frac{1}{2}$$
(3.3)

The acceleration of the mass. equation above can be stated in word as:

Resultant force on the mass = mass x acceleration

The equation of model single degree of freedom is

$$\div$$
 + +  $\doteq$  () (3.4)

Where m is mass,  $c_f$  is stiffness and b is damping coefficient of the follower. The return spring preload to ensure that the cam and follower are always in contact at a given speed. As the cam rotates it displace the follower by an amount  $y_A(t)$ . As a result the follower is displaced by an amount q(t). (Livija Cveticanin, 2006).

The simplest dynamic model is a single degree of freedom model with one mass, one spring, and one damper. More complex models have multiple degrees of freedom with multiple masses, springs, and dampers. Simplifications of complex models to simple models are shown in the following sections. From Figure 3.3 showed the model of one single Degree of Freedom where  $\alpha$  is pressure angle, p is the pitch curve, w is cam rotational speed, rad/sec and  $N_I$  is normal component of velocity.



Figure 3.3: The Dynamic Model Of One Single Degree Of Freedom

Source: Vasin Paradorn (2007)

Speed of cam follower system is select for suitable analysis of vibration. For analysis, speed at 300RPM are select because in my literature review, say that, when high speed of cam are choose, the vibration cam of follower are more accurate rather than at low speed of cam. When run the experiment, three parameter are select for analysis. Table 3.1 showed the incremental angle  $=30^{\circ}$  for rotation cam, displacement, velocity and acceleration of cam follower system are take for considerable in analysis process.

**Table 3.1**: Experiment Data for 300RPM

Angle(°)	Displacement (mm)	Velocity (mm/s)	Acceleration $(mm/s^2)$
0			
30			
60			
90			
120			
150			
180			
210			
240			
270			
300			
330			
360			

# 3.4 Analysis Parameter

Figure 3.4 showed the test fixture being used in this project. The data will be collect from the experiment to verify the equation of motion that gets from the dynamic model using mathematical model. This experiment used Cam Mechanism Analysis System as the machine to run the disc cam in vertical position and DEWESOFT Software are used as software to collect the data and analysis of disk cam profile. The results will be showed in graph type such as displacement graph, torque graph and vibration graph. Figure 3.4 showed the vertical cam that will be used in experiment and Figure 3.5 showed the heart shape cam are used because the types of cam is suitable for analyze the vibration factor .



Figure 3.4: Vertical Cam

Figure 3.5: Heart Shape Cam

To simplify the experiments and to more accurately create the model, the test fixture was modified. The data from the experiment will be defined the vibration factor of cam follower system, and also focus on the force jumped by the follower.

#### 3.4.1 DEWESOFT Software

The Dewesoft software is software that used to perform a series of standard procedure test, however the system are flexible where so any kind of additional test can be defined to archive the requirement that needed to verify the stability.

The system run a standard software package which allow user to customized input setup, acquisition, analysis, exporting and print out data. The input data is usually a sensor that connected to the A/D board. Table 3.2 showed there are few types of sensor that been used to measured the disk cam analysis and below are the specification for each sensor. For sensor of accelerometer, the type of measure is vibration in range +/- 500g, and the sensitivity about 10.75 m/g. Sensor draw wire displacement transducers can measure the displacement at range 1000mm.

TYPE OF SENSOR	TYPE OF MEASUREMENT	RANGE	SENSITIVITY
ACCELEROMETER	VIBRATION	+/-500g	10.75mV/g
DRAW WIRE			
DISPLACEMENT	DISPLACEMENT	1000mm	
TRANSDUCERS			
VELOCITY	Air Velocity	0 to 10 m/s	
TRANSDUCERS			
INCREMENTAL	RPM (Speed)	1000 Pulses/	
ROTARY ENCODER		revolution	
FGP LOAD CELL	Force	10kN	-177.36mV

Table 3.2: Types of Sensor

# 3.5 DYNACAM Software

In analysis of three parameter from experimental, the method to verify the data are correct or not by using DYNACAM Software Simulated. From this approach method, we can see the graph from DYNACAM must same or having a little different, but the shape or the rotation of cam must always same to prove the factor of vibrations by three elements. Using DYNACAM, we must set the speed of RPM (Revolution per Minute) of cam follower system. Besides that, input the data of angle and No of Segment in box that display in Figure 3.6.

After put the data in box, press the 'calculate' box to calculate the input, and we can see the output data about the three parameters, displacement, velocity and graph. For another of motion, also have a jerk, but in this experiment only three parameter are consider and suitable for one mass degree of freedom that only have one mass and spring. The figure below, showed the appearance of screen DYNACAM.

Cam Data	Segme	ent Data i	0		_			0t		Destiller	4-5
Cam Omeda		Data	Angles	<b>F</b> 1				Lam Contour		Position	(m)
(rpm)	Seg	Beta	Start	End		Motio		Program		Start	End
200	1	90	0	90.000		Rise	•	MS - Mod Sine	-	0	25
300	2	90	90.000	180.000		Dwell	•	DW - Dwell	-	25	25
No. Segments	3	90	180.000	270.000		Fall	•	MT - Mod Trap	•	25	
4	4	90	270.000	360.000		Dwell	•	DW - Dwell	-	0	0
Delta Theta	5					·	_	·			
(deg)	2										
0.125 💌	6										
Follower	7										
Translating	8										
O Oscillating	9										
Starting Angle	10										
0 deg	11										
	12										

Figure 3.6: Screen Display from DYNACAM Software

Source: DYNACAM Software

# 3.6 Energy methods for impact modeling



Figure 3.7: Force vs Deflection

Source: Ginsberg, Jerry H. (2001)

The energy method states that the sum of the kinetic energy of the colliding masses prior to impact equals the sum of the potential energy of the collided masses. Once the masses reach the maximum deformation, all the energy has been stored in a form of potential energy and the masses are moving at the same velocity. All of the potential energy is released and converted into kinetic energy. In the case of a linear elastic force versus deflection curve. Figure 3.7 the potential energy is simply:

$$. = \int = ---- (3.5)$$

Where  $F_{max}$  is the maximum impact force and  $\delta_{max}$  is the maximum deflection. Since the relationship of force versus deflection is known:

= (3.6)

It is possible to obtain a simpler potential energy equation:

where k is the elasticity of the impacted system. From all the equation above, the deflection and correction approach can obtain. For detail calculation, can be referred in the Chapter 4; Result and Discussion.

# **CHAPTER 4**

# **RESULTS AND DISCUSSION**

# 4.1 Introduction

In this chapter will all showed the results obtained from this study. Table of results, graphs, and figures will be included. Details explanation on graphs and figures are also will be provided. The data is collected starting with the experimental result from cam follower in vertical position. The type of cam that used for this experiment is heart shape cam. The result focused on the vibration of motion cam follower system that considers three parameter, displacement, velocity and acceleration. The analysis method is used are DYNACAM software to comparing the data on the vibration factor. After analysis the result, the impact force also analysis to correlations of acceleration and force described in this project, a successful experimental investigation and modeling of impact in an over-travel mechanism was performed.

#### 4.2 Experimental Results

#### 4.2.1 Results for 300RPM

In order to obtain the result of displacement, velocity and acceleration, a Dewesoft software is used for input 300RPM. The speed used because suitable for analysis the vibration factor on the motion cam follower system. From my literature review, when speed is high the analysis for vibration more effective rather low speed. This equipment equipped with combine with Oscilloscope to record the data. Microsoft Excel was used to format the data and plot the results of the tests. From the data, the displacement, velocity, and acceleration are constructing based on the data and using constant acceleration motion. Table 4.1 below showed the data for speed 300RPM that consist the Degree of rotational cam, tine, torque, force, displacement, and vibration.

Degree [°]	Time [s]	Torque [ Nm]	Force [kN]	Displacement - f [mm]	Vibration [g]	Tacho - [rpm]
0	0	24.837494	0.0086208	0	-0.2154633	106.20117
30	0.041	10.696411	0.0891747	7.3314681	-0.2177636	108.30688
60	0.083	15.487671	0.1537846	13.559437	0.14492016	111.60278
90	0.125	18.489838	0.2272007	21.221084	0.65559119	117.55371
120	0.167	24.169922	0.2881954	22.561682	-0.2215975	122.13135
150	0.209	24.147034	0.3532688	36.009121	-0.0874122	126.52588
180	0.2425	15.922546	0.4119461	41.930237	0.07514378	126.80054
210	0.284	-19.298553	0.3462238	35.877541	-5.5698414	123.59619
240	0.3265	-19.248962	0.2769791	29.528788	-1.0727159	128.17383
270	0.364	-14.648438	0.217931	22.127392	-0.3174442	121.76514
300	0.4065	-9.967804	0.1477592	16.502333	-0.3320128	118.7439
330	0.449	-7.3280334	0.0745285	7.850924	0.09738021	111.51123
360	0.489	- 0.47302246	0.0006489	-0.20837259	-0.0452396	106.75049

Table 4.1: Result from Dewesoft Software at 300RPM

# 4.2.2 Constant Acceleration Motion

From the data at Table 4.1, the displacement, velocity and acceleration diagram are constructed by using the constant acceleration motion. The motion at am cam follower must consider rotate on position RDFD (Rise, Dwell, Fall, Dwell). The graph will be plot versus the cam angle rotation (), the incremental angle = ° is choose because is show smoothness line when the graph is plotted. The equations of constant acceleration as shown in Table 4.2.

 Table 4.2: Equation of constant acceleration motion

For 0< <0.5	For 0.5 < <
$\frac{\underline{\text{Rise}}}{\Delta \mathbf{R} = 2}  ( \ / \ )^2$ $= \ / \ ^2$ $= \ ( \ / \ )^2$	$\frac{\underline{\text{Rise}}}{\Delta \mathbf{R}} = -(-/)^{2}$ $= (/)(-/)$ $= -(/)^{2}$
$\frac{Fall}{\Delta \mathbf{R}} = - ( / )^{2}$ $- = / ^{2}$ $= - ( / )^{2}$	$\frac{Fall}{\Delta \mathbf{R}} = -( - / )^{2}$ = -( / )(1- / ) = ( / )

Where:

- $\Delta R$  = Instantaneous follower displacement at time t or cam angle  $\beta$ .
- v = Instantaneous follower velocity.
- a= Instantaneous follower acceleration
- H = Total follower displacement
- $\beta$  = Rotational angle of cam during the rise or fall interval
- $\phi$ = Angle into rise or fall interval
- $\omega$ = Speed of the cam

# 4.3 Analysis of the Experimental Result

For the analysis, the calculation for three parameters is made by using constant acceleration motion. The graph kinematics' motion for displacement, velocity and acceleration is constructed.

# 4.3.1 Displacement Graph

The graph for displacement at 300RPM plot by using Microsoft Word EXCEL. The graph showed in Figure 4.1 is the result at every angel of motion cam follower system. The cams turn through one cycle of 360°, in calculation the incremental  $\emptyset = 30°$  is used to plot the graph. The result show that when cam rotates from zero angle until 90° the motion of cam is rise, this is because the cam rotate at constant speed. From 90° until 180 is constant or Dwell, for that time cam is zero acceleration, from 180° until 270° is fall because the cam is moving on zero velocity and for the last cycle from 270° until 360° is motion is dwell or constant. The maximum value at 180° is 21.22 mm. Besides that, the graph shows the result of moving stability at 300° to 360°. See Appendix A for Table A1 of this graph. The example of calculation for displacement is show below:

Example calculation at rotation 180°

 $\Delta R = 2 \quad ( \ / \ )^2$ = 180° x ---= 3.142 rad H= 42 mm = 360° x---= 6.283 rad  $\Delta R = 2(42)(3.142/6.83)^2$ = 21.22 mm



Figure 4.1: Displacement diagram

#### 4.3.2 Velocity Graph

The motion of velocity of cam follower system is constructed more depends on the angular velocity,  $\omega$  and the displacement of cam. When high speeds apply on machine, the velocity is also increase proportionally and the displacement also directs proportionally effect of velocity. From the graph shown in Figure 4.2, the velocity of cam follower is rise start on 30° until 60° because the displacement are increase, then the graph show from 60° the velocity is fall until 90°. The velocity has zero position from 90° until 180°. For this situation, the speed are constant, From 180° until 240° the velocity is decrease that's means the motion is fall maybe at this point the displacement starts dwell and the effect of velocity of motion. Then start at 270° the velocity is zero degree. At this section, the displacement also zero that happen the velocity is zero. Based on this graph, the maximum of velocity is 564.23mm/s and the lowest velocity is -700.34mm/s. See Appendix A for Table A2 of this graph.

From the calculation the graph will be plot. The example of calculation for velocity is show below:

Example calculation at rotation 210°

$$- = 4 / {}^{2}$$

$$H= 35.88 \text{ mm}$$

$$= 300 \text{ x} - -$$

$$= 31.42 \text{ rad}$$

$$= 210 \text{ x} - -$$

$$= 3.7 \text{ rad}$$

$$= 360 \text{ x} - -$$

$$= 6.283 \text{ rad}$$

$$- = 4 / {}^{2}$$

$$= 4(35.88)(31.42)(3.7)/6.283^{2}$$

= -<u>422.77 mm/s</u>



Figure 4.2: Velocity Diagram

# 4.3.3 Acceleration Graph

The acceleration graph will present the combining of displacement, speed, spring thinness and another factor will effect of acceleration curve. Based on the literature review, the maximum positive acceleration larger than the negative maximum acceleration about ration 3:1, that would be good choice for spring-loaded high speed cams. From the graph below, the maximum of acceleration of cam follower is 88767mm/s<sup>2</sup> at the 240°, and the lowest or minimum of acceleration is -46374 at 60°. When the displacement is maximum, the velocity will be zero because this is the position at where its direction is reverses. The motion of acceleration in Figure 4.3 presents at zero degree and will until 30°. At this moment the motion will be rise. However it will fall again until 60°, and then the acceleration is dwell start from 90° until 180°. See Appendix A for Table A3 of this graph. The example calculation of acceleration as showed below;

Example calculation at rotation 210°

 $= -4 ( / )^{2}$ H = 35.88 mm = 300 x --= 31.42 rad = 360 x ---= 6.283 rad = -4 ( / )^{2} = -38566 mm/s^{2}



Figure 4.3: Acceleration Diagram

# 4.4 Graph Simulated from DYNACAM

#### 4.4.1 Displacement Diagram.

From the simulated DYNACAM Software, the graph for displacement for cam motion is display after key in the same data of experiment. The graph is show the cycle of cam rotating at 360° angle of cam.

#### <u>From 0°-90°</u>

From the Figure 4.4, the cam is on position rise that the cam is rotate anti-clock wise and the displacement is increase when angle of cam is increase. The maximum of displacement are 25.00 mm. The rotation of cam is full filled the criteria of simple harmonic motion with dwells.



Figure 4.4: Displacement diagram from 0°-90°.

#### From 90°-180°

From the Figure 4.5, the graph showed the cam is on position dwell from angle  $90^{\circ}$  into  $180^{\circ}$ . The displacement is constant at 25.00mm; this is because the cam is on the zero velocity and acceleration of cam rotation.



Figure 4.5: Displacement diagram from 90°-180°.

#### From 180°-270°

From the Figure 4.6, the cam is on position fall that the rotate anti-clock wise and the displacement is increase because the acceleration of cam for that time the acceleration is increase and the velocity is low. The torque for that is also effect the displacement of cam, the torque is negative.



Figure 4.6: Displacement diagram from 180°-270°.

# From 270°-360°

From the Figure 4.6, the graph showed the cam is on position dwell again at zero displacement from angle 270° into 360°. The displacement is constant at zero displacement, this is because the cam is turning back on the cycle of rotation and the acceleration is zero.



Figure 4.7: Displacement diagram from 270°-360°.

# 4.4.2 Velocity Diagram.

Many machine design situation demand some particular combination of displacement, velocity and acceleration constraints over one or more segments of motion. The diagrams from Figure 4.8 until Figure 4.11 showed the velocity diagram for simple harmonic motion with dwell.

# <u>From 0°-90°</u>

From the Figure 4.8, the cam rotation at zero degree until 90° is rise until 45° and fall again until 90°. The situations show that the placement of these interior knots has a significant effect on the shape of the curves. The velocity overshoots to 880 mm/s for high value of velocity diagram.



Figure 4.8: Velocity diagram from 0°-90°.

# From 90°-180°

From the Figure 4.9, the cam rotation at 90° until 180° is dwell at zero position. The splitting is constant velocity portions in rotation in the hope that the lower order polynomials and not overshoot the target. Beside that the velocity is zero when the displacement is constant and the acceleration must be zero at this situation.



Figure 4.9: Velocity diagram from 90°-180°.

#### From 180°-270°

For the next diagram of velocity shown in Figure 4.10, the cam rotation at 180° until 270°, the cam is rise at lower position of overshoots and rise again from 225° until 270°. The value of velocity at this time is negative because the displacement of cam is also decrease and for that, the velocity is must negative. It also has the smallest cam for a given rise and provides a long stroke action.



Figure 4.10: Velocity diagram from 180°-270°.

#### From 270°-360°

The last cycle of cam rotation for velocity diagram is between  $270^{\circ}$  until  $360^{\circ}$ . Figure 4.11 showed the cam velocity is at zero velocity or dwell position. This is because, the displacement of cam is also zero and based on theoretical when zero displacement, the velocity must zero.



Figure 4.11: Velocity diagram from 270°-360°.

# 4.4.3 Acceleration Diagram

Acceleration diagram is also considered based more on the displacement and velocity situation. This is because, acceleration is the rate of change of velocity and velocity is related with the displacement of cam follower system. Figure 4.12 until 4.15 describe the diagram position at four cycle start zero degree.

# From 0°-90°

From the Figure 4.12, the cam rotation at zero degree until 90°, the cam is rise until  $11.25^{\circ}$  and fall again until 45°. The situations show that the placement of these interior knots has a significant effect on the shape of the curves. The acceleration overshoots to 55280 mm/s<sup>2</sup> for high value of velocity diagram. After that, the cam is on negative when position from 45° until 90°. The cam diagram is fall at low value -55280 mm/s<sup>2</sup> at 78.75° and turn to zero at position at 90°.



**Figure 4.12:** Acceleration diagram from 0°-90°.
### From 90°-180°

From the Figure 4.13, the cam rotation at 90° degree until 180°, the cam is zero acceleration because the velocity of cam system is zero so the rate of change for velocity not occurs.



Figure 4.13: Acceleration diagram from 90°-180°.

#### From 180°-270°

For the next diagram of acceleration in Figure 4.14, the cam rotation at  $180^{\circ}$  until 270°, the cam is fall at lower position and dwell from  $191.25^{\circ}$  until 213.75° than is zero at 225°. Then the cam is overshoots and rise again from 225° until 258.75° at 977762 mm/s<sup>2</sup>. The value of acceleration at this time is rise again because the velocity of cam is also rise and after that the acceleration is zero at 270°.



Figure 4.14: Acceleration diagram from 180°-270°.

### From 270°-360°

The last cycle of cam rotation for velocity diagram is between 270° until 360°. The Figure 4.15 showed the cam acceleration is at zero velocity or dwell position. This is because, the displacement of cam is also zero and velocity is also zero and based on theoretical when zero displacement and velocity, the acceleration must zero. This create a more ragged or rough jerk function which will increase vibration problem, that means the surface cam are not overshoot the target.



Figure 4.15: Acceleration diagram from 270°-360°.

### 4.5 Comparison Experimental Graph with DYNACAM Graph

A comparison graph between experimental and DYNACAM graph its because to verify that the data will get from experiment are validate or not. Besides that, the method will used because the DYNACAM Software are usually used in dynamic analysis of cam. Once the calculation was completed, the data was exported into a text file which could be opened in spreadsheet software for comparison purposes. The appearance of screen DYNACAM Software showed Figure 4.16. The simulated data obtained were inserted into Excel and are shown in Appendix B, Table B1 until Table B3



Figure 4.16: DYNACAM Single dwell Translating of Cam follower System.

### 4.5.1 Displacement

The displacement was compare from zero degree until 360°. The speed of am system is setting 300RPM and the value for no of segment is 4. Data that collect has a 17 of point that can present of DYNACAM graph. From the data from calculation and simulated, the graph was plotted using Excel to compare between experimental and simulated DYNACAM. From the Figure 4.17 the graph showed that shows the displacement versus cam angle comparison of the simulated data obtained from DYNACAM and experiment, which are shown in red and blue colour, respectively. Unfortunately, the displacement difference is apparent during the upper dwell, cam angle 90 to 180 degrees. This was due to the significant figure cut-off on experiment part, but this was not determined to be important because the objective of this comparison is to verify the data from experiment are correct or not. The graph from experiment have little different between DYNACAM graph maybe during experiment have human error and temperature environment effect the data of experiment.



Figure 4.17: Experiment vs DYNACAM comparison Displacement.

### 4.5.2 Velocity

The simulated of DYNACAM already run at the same parameter for displacement experiment. The point of graph is 17 teen, starts from zero degree until 360°. The velocity versus cam angle comparison at Figure 4.18 showed very little difference between the experiment and DYNACAM model. The differences can be seen during the positive and negative peaks at 45 degrees, respectively. The blue line is experiment graph and red line is DYNACAM simulated. Even these locations, the difference were determined to be a minimal and ignored. One would expect the vibration to be more pronounce in the acceleration result than the velocity result. However, the vibrations due to splits in the cam were more pronounce in the velocity plot. While the displacement and velocity showed limited differences, the most sensitive component would be the acceleration simulated result, which is shown in Figure 4.18 at comparison acceleration.



Figure 4.18: Experiment vs DYNACAM Comparison Velocity

### 4.5.3 Acceleration

For the last comparison is acceleration comparison graph between Experiment and DYNACAM simulated. From the Figure 4.19, the graph Acceleration vs Angle (°) was plot to compare between experiment and simulated DYNACAM. The acceleration versus cam angle comparison once again shows almost are little difference between the DYNACAM and Experiment results. The maximum and minimum values of accelerations at Figure 4.19 are practically the same and the frequencies of these two data match very well. From the above comparisons of the displacement, velocity, and acceleration between DYNACAM and experiment, it would be reasonable to assume that the data from experiment use very well and DYNACAM's result very well. From the above comparisons, it was determined that vibration effect on the two type of model is reasonable to compare, and the fact about vibration in three factors are correct.



Figure 4.19: Experiment vs DYNACAM Acceleration Comparison

### 4.6 Analysis of Impact Force

Vibrations are generally caused by forces whose magnitudes, directions, and/or point of application vary with time. These forces produce variations in elastic deformation. These vibrations in turn produce stress and force that are superimposed on the inertia and other forces in the follower system. The magnitudes of the vibratory stresses and force are influenced by the acceleration characteristics, as well as the rigidity and the damping of the follower mechanism.

#### 4.6.1 Ball Drop Experiment

There are two main methods of approximating impact force, which are the energy and wave methods. Although the wave method gives a more accurate approximation, the complexities exceed the accuracy advantage. The energy method was considered and researched. The two energy methods found to be applicable were, as described in Mechanical Analysis and Design (Burr 1982) and Impact Forces in Mechanisms (Johnson 1958), deflection and correction factor approach and relative velocity approaches, respectively.

To determine the best impact force approximation, a simple impact experiment was created. This is a striking impact that occurs when a spherical ball strikes a flat surface. The repeatability and multiple impact velocities were the major concerns in this experiment. To obtain a repeatable experiment, the conditions of the impact must be consistent while obtaining multiple impact velocities requires multiple drop heights.



Figure 4.20: Ball Drop Experiment Setup

Figure 4.20 shows the ball drop experiment setup with the stopper in place. As seen in this picture, the two dimensions D1 and D2 are known. Therefore, the drop height is D2 - D1. From the known relative height, it is possible to obtain the impact velocity from the following equation:

$$\boldsymbol{v}_{\boldsymbol{l}}^2 = \boldsymbol{v}_{\boldsymbol{0}}^2 + \boldsymbol{2}\boldsymbol{g}\boldsymbol{h} \tag{4.1}$$

- $v_t$  = Impact Velocity  $v_0$  = Initial Velocity g = Gravitational Acceleration
- h =Relative Height

Because the initial velocity is zero, the final equation for impact velocity is:

$$v_i = \sqrt{2gh} \tag{4.2}$$

The impact velocity is needed in order to obtain the impact force approximation through common velocity approach while only the relative height is needed for the deflection and correction factor approach.

The heights at which the ball was dropped from were 25.4mm, 31.75mm, 44.75mm, and 57.15mm. Only the first impact data was saved in the oscilloscope by setting a trigger delay. The data of that experiment is below:

Height(mm)	Voltage(V)	Force(N)
25.4	2.8	231.074
31.75	3	249.357
44.45	3.45	286.76
57.15	4	332.461

 Table 4.3: Ball Drop Experiment Data.

Table 4.3 showed the different heights at which the ball was dropped and their respective voltage and force outputs. Even though the default output unit of the oscilloscope is in Volts, it was easily converted into pound force with the known sensitivity of 53.9 mV per pound force. These data will be compared to the theoretical data computed through deflection and correction factor and common velocity approaches.

The deflection and correction factor was the first method that was derived for this comparison. The original form is not suitable for this particular experiment. Therefore a new equation must be derived from the known properties. The original deflection and correction factor approach has the following form:

$$F_{i_0 def} = \left[1 + \sqrt{1 + \frac{2h}{\delta_{st}}}\right] \acute{W}$$
(4.3)

As the name applies, the equation contains both deflection factor,  $\mathfrak{G}_{st}$ , and correction factor,  $\eta$ . For this experiment, the correction factor was substituted by one because of the large driven mass to driving mass. A new equation was formed from the mentioned simplification and is shown below.

$$F_{i_0 def} = \left[1 + \sqrt{1 + \frac{2h}{\delta}}\right] \acute{W}$$

$$(4.4)$$

To calculate the above equation, three values must be determined: the drop height, h, static deflection due to the weight of driving mass,  $\delta$ , and the weight of the driving mass, W'. The drop height and the weight of the driving mass could easily be obtained and calculated whereas the static deflection required more calculations. The heights were easily obtained from the experimental heights.

The weight of the driving mass is simply:

W' = 
$$\frac{4}{3}\pi \left(\frac{Dia}{2}\right)^3 x \rho x g$$

For example of calculation:

Dia = diameter of spherical ball  $\rho$  = Mass density of the ball g = Gravitational Acceleration Dia = 0.00953 m  $\rho$  = 0.002 g/cc G = 9.81 m/s<sup>2</sup>  $k_{hs}$  = 10647.66 KN/m  $\frac{4}{3}\pi \left(\frac{0.00953}{2}\right)^3 x 0.002 x 9.81$ W'= 3.487x10<sup>-3</sup>

$$F_{i_0def} = \left[1 + \sqrt{1 + \frac{2h}{\delta_{st}}}\right] \tilde{W}$$
$$= \left[1 + \sqrt{1 + \frac{2(0.00254)}{3.274x10^{h} - 11}}\right] \times 3.487 \times 10^{-3}$$
$$= \underline{137.35N}$$

### 4.7 Comparison Data of Experiment Ball Drop.

From the calculation above, the result is will be obtained from the experiment setup. The data is show below:

-	FORCE (N)		
Height( mm)	Experiment	D&C Factor	Common Velocity
25.4	231.074	137.35	256.47
31.75	249.357	153.59	293.26
44.45	286.76	181.7	358.82
57.15	332.461	206.03	417.22

**Table 4.4**: Data Comparison of Experiment Ball Drop.

Table 4.4 compares the calculated and experimental impact forces obtained through experiment, the deflection and correction factor energy method, and the common velocity approach. Figure 4.21 and Table 4.5 showed the graph impact force comparison and the percent error of three method applied which deflection and correction factor, common velocity and experiment.



Figure 4.21: Impact Force Comparison

Data below show the percent of error between experiment with deflection and correction factor and common velocity. Figure 4.21 showed that deflection and correction factor method. The result is underestimates the actual impact force whereas the common velocity approach predicts higher forces for every drop height. The graph from Figure 4.21 showed higher impact from common velocity, then the second graph line is using deflection and correction factor and the last graph line is the low impact from experiment ball drop.

 Table 4.5: Percent error of D&C and Common Velocity

HEIGHT(mm)	PERCENT OF ERROR D&C%	PERCENT OF ERROR COMMON VELOCITY%
25.4	-41	11
31.75	-38.41	17.6
44.45	-36.64	25.13
57.15	-38.03	25.5

From the data at Table 4.5, the graph percent of error are plot. The detail about this graph is shown at Figure 4.22:



Figure 4.22: Percent Error Comparison

Figure 4.22 showed the percent error relative to the experiment between the deflection and correction factor and the common velocity approach. As seen in the Figure 4.22 the deflection and correction factor method was between 36% and 40% lower, but the common velocity overestimated the impact forces by 10% to 26%. This overestimation automatically gives the approximation a minimum factor of safety of 1.1. From the above result, it would seem appropriate to use the common velocity to approximate impact forces for the modeling of cam-follower system with impact loading.

#### 4.8 Discussion

Based on the experiment, the displacement that is calculated looking of a vibrating object as a measure of its vibration amplitude. The displacement is simply the distance from a reference position, or equilibrium point. In addition to varying displacement, a vibrating object will experience a varying velocity and a varying acceleration. Velocity is defined as the rate of change of displacement, and in the English system is usually measured in units of inches per second. Acceleration is defined as the rate of change of velocity, and in the SI unit system, is usually measured in units of mm/s<sup>2</sup>, or the average acceleration due to gravity at the earth's surface.

The displacement of a body undergoing simple harmonic motion is a sine wave as we have seen. It also turns out, that the velocity of the motion is sinusoidal. When the displacement is at a maximum, the velocity will be zero because that is the position at which its direction of motion reverses. When the displacement is zero (the equilibrium point), the velocity will be at a maximum. This means that the phase of velocity waveform will be displaced to the left by 90 degrees compared to the displacement waveform. In other words, the velocity is said to lead the displacement by a 90-degree phase angle.

Remembering that acceleration is the rate of change of velocity, it can be shown that the acceleration waveform of an object undergoing simple harmonic motion is also sinusoidal, and also that when the velocity is at a maximum, the acceleration is zero. In other words, the velocity is not changing at this instant. Then, when the velocity is zero, the acceleration is at a maximum 88767 mm/s<sup>2</sup>, the velocity is changing the fastest at this instant. The sine curve of acceleration versus time is thus seen to be 90 degrees phase shifted to the left of the velocity curve, and therefore acceleration leads velocity by 90 degrees.

#### **CHAPTER 5**

### **CONCLUSION AND RECOMMENDATION**

### 5.1 CONCLUSION

As the conclusion of this project, learning, researching and problems solving process based on design of model SDOF (Single Degree of Freedom System). From the analysis the vibration of cam follower are studying and the graph from DYNACAM show the motion of three element of vibration (displacement, velocity, and acceleration). From that graph, the relationship between displacement, velocity and acceleration is show on experiment below. From this analysis the acceleration waveform of an object undergoing simple harmonic motion is also sinusoidal, and also that when the velocity is at a maximum, the acceleration is zero Note here that the acceleration is 180 degrees out of phase with the displacement. This means the acceleration of a vibrating object is always in the opposite direction to the displacement.

Based on the correlations of acceleration and force described in this project, a successful experimental investigation and modeling of impact in an over-travel mechanism was performed. Impact and over-travel mechanisms were designed and manufactured to create measurable forces when impact occurred. DOF dynamic model was created and simulations run to predict the dynamic behavior of the assembly machine and associated forces. The model, consisting of three second-order differential equations, was driven by the cam's input function. This model included impact and over-travel forces which were not included in other industrial cam-follower system models found during an extensive literature search.

Use of this impact model for an industrial cam-follower system can guide mechanical engineers through the designing stage of assembly machines with impact and vibration of cam follower This model will allow the elimination of expensive and time-consuming full modeling methods, which will reduce machine development costs as well as development time. From the last result on the comparison, it would seem appropriate to use the common velocity to approximate impact forces for the modeling of cam-follower system with impact loading.

### 5.2 **RECOMMENDATION**

In order to achieve the good result during the experiment, the precaution step should be follow and all the input setup for sensor should be in the range of sensitivity. In future research, more compatible study the second mass in the SDOF model allowed one to determine the contact force between the cam and follower and obtain a more accurate result. In addition, the two-mass models predict jump more accurately than the one-mass models.

The two mass model have more accurately predicted jump, and allow calculation of contact force. Besides that, give a more accurate comparison to the experimental data. On my literature review, almost researcher study more than one of mass of single degree of freedom cam follower system. Because, from their experiment they can find more about the cam such as the stability, the motion control of cam, investigate the dynamic effects of incipient separation of industrial cam-follower systems and etc.

The other potential area that should be improved is the optimization of the am speed. There are lot types of cam and each cam has the maximum cam speed that they can endure. So they analysis should be carried out to know the maximum speed of cam can endure and whether the cam is suitable for high or low speed application.

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### APPENDIX A

# **Result Experiment for All Speed of Cam Follower**

## 100 RPM

Degree [°]	Time [s]	Torque [ Nm]	Force [kN]	Displacement - f [mm]	Vibration [g]	Tacho - [rpm]
0	0	-5.3825378	0.0367081	0	0.02913739	0
30	0.138	6.2255859	0.0531155	5.3180022	0.06977637	40.557861
60	0.276	13.09967	0.1366356	13.804935	-0.0460064	87.70752
90	0.414	19.107819	0.2156135	20.844484	-0.01303515	106.38428
120	0.552	21.640778	0.2914398	25.627472	0.05214059	96.679688
150	0.69	32.333374	0.3655975	31.061764	0.10658149	69.122314
180	0.829	-29.262543	0.3876594	27.601284	-0.20166138	35.15625
210	0.967	-19.622803	0.2928303	30.055109	-0.02760384	11.993408
240	1.105	-14.015198	0.2021725	20.647114	-0.00690096	0
270	1.174	-11.390686	0.1523941	15.548374	-0.057508	0
300	1.289	-6.7481995	0.080183	6.5022254	0.04140576	0
330	1.358	-5.3672791	0.0279018	3.3442969	-0.02607029	
360	1.391	-0.17929077	-0.001576	0.021893017	0.09891376	5.1269531

### 150RPM

	Time	Torque [		Displacement - f		Tacho -
Degree(°)	[S]	Nm]	Force [kN]	[mm]	Vibration [g]	[rpm]
0	0.9695	7.8964233	0.0088062	0	1.011374	13.73291
30	1.0595	10.185242	0.0928825	6.3640976	-0.00383387	43.85376
60	1.1495	13.492584	0.1566582	12.877325	0.03527157	72.601318
90	1.2395	19.90509	0.2227512	18.686609	0.021469653	91.278076
120	1.3295	26.031494	0.2873612	24.134047	-0.05520768	101.16577
150	1.4195	20.713806	0.3465019	30.088058	0.16102239	94.573975
180	1.5015	16.25824	0.4111118	36.930237	0.023003198	84.777832
210	1 5915	- 16 265869	0 3418671	30.088058	-0 30977643	67 474365
2.0	1.0010	-			0.00011010	
240	1.6815	16.963959	0.2662262	27.456396	0.094313115	48.339844
270	1.7715	- 13.122559	0.1973522	20.351057	0.068242826	26.367188
	4 00 4 5	-		40.470000	0.044470054	45 50005
300	1.8615	9.1438293	0.1280148	13.179928	0.044472851	15.563965
330	1.9515	- 4.2304993	0.048666	5.0548415	0.65329087	4.5776367
360	2.0415	8.0108643	0.0246574	2.4890246	0.087412156	20.507813

### **200RPM**

	Time	Torque [		Displacement		Tacho -
Degree(°)	[s]	Nm]	Force [kN]	- f [mm]	Vibration [g]	[rpm]
0	0	9.727478	0.0990005	0	-0.013801919	62.347412
30	0.053	14.030457	0.1468323	5.120738	-0.062108636	71.411133
60	0.107	18.062592	0.2045826	10.910275	-0.085111834	83.221436
90	0.161	21.434784	0.2631672	16.798494	0.16178916	92.74292
120	0.215	27.832031	0.3184147	22.357765	-0.20549524	98.419189
150	0.269	37.757874	0.3707886	27.555187	-0.24536745	101.25732
		-				
180	0.3205	0.52642822	0.408609	34.897339	0.32281157	99.42627
210	0.377	-17.311096	0.3363052	32.495945	-4.3061991	95.214844
240	0.431	-15.842438	0.2763302	28.410355	-0.036805119	83.953857
270	0.485	-14.202118	0.2163551	22.390553	-0.13571887	77.087402
300	0.539	-11.417389	0.1589756	16.831284	0.026837066	69.213867
330	0.593	-7.5302124	0.0972393	9.7588387	0.13801919	57.678223
360	0.647	-5.355835	0.0334637	3.5087724	0.003067093	49.438477

### 250RPM

	Time	Torque [		Displacement - f		Tacho -
Degree	[s]	Nm]	Force [kN]	[mm]	Vibration [g]	[rpm]
			-			
0	0.195	1.3237	0.004078674	0	-0.30670932	77.728271
30	0.2475	9.5481873	0.078329086	6.6535535	0.29367417	81.207275
					-	
60	0.3	13.183594	0.1424755	124.693102	0.030670932	88.256836
90	0.3525	17.814636	0.20912474	18.502386	-0.14492016	97.137451
120	0.405	22.30072	0.27688637	24.982718	0.056741223	109.40552
					-	
150	0.4575	20.736694	0.34613112	30.048561	0.015335466	108.03223
					-	
180	0.51	13.771057	0.41380003	37.930237	0.019936105	111.69434
210	0 5625	- 19 428253	0.33964232	34 92358	- 0 074377008	110 4126
210	0.0020	10.120200	0.0000 1202	01.02000	-	110.1120
240	0.615	-17.63916	0.27011946	27.916927	0.052140586	106.56738
		-			-	
270	0.6675	13.046265	0.19846457	20.383953	0.081277966	99.609375
		-				
300	0.72	9.3383789	0.129776	13.245718	0.084345065	90.911865
220	0 7705	-	0.055000070	E 7450004	0.00004500	00.044004
330	0.7725	5.2146912	0.055803679	5.7456384	-0.08664538	80.841064
360	0.825	23.265839	0.008250046	1.0745358	-0.11884986	74.066162

Angle(°)	Displacement(mm)
0	0
30	7.33
60	13.56
90	21.22
120	21.22
150	21.22
180	21.22
210	13.56
240	7.33
270	0
300	0
330	0
360	0

### TABLE A1 DISPLACEMENT FOR 300RPM

### TABLE A2 VELOCITY FOR 300RPM

Angle(°)	Velocity(mm/s)
0	0
30	448.22
60	564.23
90	0
120	0
150	0
180	0
210	-512.23
240	-700.34
270	0
300	0
330	0
360	0

Angle	Acceleration(mm/s <sup>2)</sup>
0	0
30	45672
60	-46734
90	0
120	0
150	0
180	0
210	-38566
240	88767
270	0
300	0
330	0
360	0

### TABLE A3 ACCELERATION FOR 300 RPM

### **APPENDIX B**

# TABLE RESULT FROM DYNACAM SIMULATED

### TABLE B1 DISPLACEMENT FROM DYNACAM

Angle(°)	DISPLACEMENT(mm)
0	0
22.5	6.25
45	12.75
67.5	18.75
90	25
112.5	25
135	25
157.5	25
180	25
202.5	18.75
225	12.5
247.5	6.25
270	0
292.5	0
315	0
337.5	0
360	0

### TABLE B2 VELOCITY FROM DYNACAM

Angle(°)	Velocity (mm/s)
0	0
22.5	440
45	880
67.5	440
90	0
112.5	0
135	0
157.5	0
180	0
202.5	-500
225	-1000
247.5	-500
270	0
292.5	0
315	0
337.5	0
360	0

Angle(°)	Acceleration (mm/s <sup>2</sup> )
0	0
11.25	55280
33.75	27640
45	0
56.25	-18426.667
67.5	-55280
78.75	-36853.334
90	0
101.25	0
112.5	0
123.75	0
135	0
146.25	0
157.5	0
168.75	0
180	0
191.25	-48881
202.5	-48881
213.75	-48881
225	0
236.25	97762
247.5	97762
258.75	97762
270	0
281.25	0
292.5	0
303.75	0
315	0
326.25	0
337.5	0
348.75	0
360	0

### TABLE B3 ACCELERATION FROM DYNACAM