Transilvania University of Brasov

INTERDISCIPLINARY DOCTORAL SCHOOL

Faculty of Mechanical Engineering

Ing. Karim SHALABY

Kinematics and Dynamics of Inverted Tooth Chain Drives based on Analogous Huygens Pendulum Model

Cinematica și dinamica transmisiilor cu lanț dințat pe baza modelului pendulului analogic al lui Huygens

ABSTRACT/ REZUMAT

Scientific supervisor

Prof.dr.ing. Simona LACHE

BRAȘOV, 2018

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To Mrs./ Mr. ....

Review Board of the Doctorat Thesis

Enforced by the Decision of the Rector of the Transilvania University of Brașov
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Date, time and place of the public defence of the doctoral thesis:
12th of October 2018, at 12.00, room UII3.

Any appreciations or comments on the content of the doctoral thesis will be sent in time, by email, to the following address: karim.shalaby@unitbv.ro .

At the same time, you are kindly invited to take part in the public session of the thesis' defense.

Thank you.

CONTENT (English)

CUPRINS (lb. română)

Foreword

This doctoral thesis named *Kinematics and Dynamics of inverted Tooth Chain Drives based on Analogues Huygens Pendulum* represented for me an interesting, challenging and fulfilling educational journey.

The thesis was inspired by the work of Prof. Dr. Eng. Șerban Bobancu from Transilvania University of Brasov as he first introduced me to the Analogues Huygens Pendulum. He raised my curiosity and passion to discover chains behavior with unconventional methods. I believe that trying to reduce the level of vibrations would lead to a better understanding of Inverted Tooth Chain characteristics, together with enhanced performance of this mechanical system.

My special thanks are addressed to my scientific coordinator Prof. Dr. Eng. Simona Lache who believed in me and guided me along the journey in every way and with a lot of patience. I have greatly benefitted from her scientific expertise and exigency.

I wish to thank Dr. Eng. Radu Plamadeala from Schaeffler Romania S.R.L. who inspired me through our professional discussions and encouraged me all the way.

During my PhD work I had the opportunity to conduct experiments on the Analogues Huygens Pendulum by the curtesy of Schaeffler Romania S.R.L. I would like to thank my colleagues from Schaeffler who gave me the support I needed in conducting my research.

Many thanks to the members of my guiding scientific commission, Prof. Dr. Eng. Gheorghe Mogan, Prof. Dr. Eng. Sorin Vlase and Assistant Prof. Dr. Eng. Marian Velea, who thoroughly advised and discussed my work and had a great contribution to my professional development. I am also grateful to Prof. Dr. Eng. Doina Pîslă, Prof. Dr. Eng. Ioan Doroftei and Prof. Dr. Eng. Sorin Vlase for accepting to review the thesis and for passing their valuable feedback.

I would also like to thank my mother and her dear friends for their steady support during the completion of the work.

Finally, I want to dedicate this thesis to my late grandmother, Prof. Letitia Bolbocianu, who believed in me.

Acknowledgements

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1. Introduction

Mechanical drive systems were used since the beginning of times. Man, in an effort to ease his life, has always had the tendency to invent machines. Primitive mechanical drive systems helped in building ancient civilization's buildings and also helped as means of transportation. Man's own power wasn't enough to lift heavy objects for short periods of time nor for long distances, thus mechanical drive systems came as a way of overcoming man's natural limitations.

Chain drive systems are mechanical systems that can transform rotational motion to linear motion. Chain drive systems have many advantages such as relative low prices compared to other drive systems. They are easy to maintain, have a long lifetime until wear and can be reliable for heavy duty applications such as: industrial conveyors, military applications, pharamaceutical industry and automotive industry.

Man would use his shear power to move mechanical drives, yet, it wouldn't help his demands or the capability to use drive systems for a long period of time. Since the industrial revolution mechanical drive systems, like any other mechanical systems, were then powered by various energy sources such as coal, petrol or natural gases. The burning of petrol or natural gases lead to polution, another drawback of these sources being also that they are non-renewable resources. In modern days, electrical energy is used to power the mechanical drive systems.

Chain drive systems have some drawbacks as well, due to the vibrations and noise caused by the contacts of the chain links with the sprockets on one hand and the contact of the chain links with the tensioners guiding the chain motion, on the other hand.

Vibrations and noise are indications that a form of energy is transformed into another energy that is undesirable for a certain application. In this specific case the kinetic energy is transformed to sound energy or heat energy according to the conservation of energy theory. Vibrations and noises are good diagnostics that a dynamic system in motion isn't functioning at its best capacity. This can be caused by contacts of the parts of the system with each other, or by the source of energy keeping a system in motion that might have some irregularities. Therefore, for a steady motion of a dynamic system, it is important that the signal produced by the source of energy used in transmission should be clean. The transformation of energy can change the state of a material used in building a drive system, producing harmful vibrations and noise. They can also be an indication that more energy is needed to give the amount of motion desired for the application used. As fuel in general is a limited resource, there is a global inclination to preserve it so as not to waste energy. Also it is encouraged the use of renewable sources of energy in order to avoid discomfort and health issues in our daily lives.

Inverted Tooth Chain (I.T.C.) drive systems are reliable and are characterized as having slightly less vibrations compared to other chain drive systems. One of I.T.C. drive systems' main assets is that they have great potential over the other chain type drives, as they are much more silent. They lose less energy during motion compared to other types of chains, thus improving the overall quality of the mechanical motion transmitted in a chain drive system. Therefore, they are are used in high speed automotive industrial conveyors and medical conveyors where stability is needed.

However, still the vibrations and noises of I.T.C.s are relatively high due to the contacts of the chain drive components during motion. When looking at a linear continuous time where a system moves uninterruptedly it becomes hard to discretisize and understand the causes of vibrations and noise.

Many researchers studied chain drive systems whether in a fast dynamic motion or in a static form trying to find the forces induced during motion. They also attempted to minimize the irregularites caused during motion. In an effort of reducing vibration and noise a curiosity to discover how to minimize the irregularities caused during the motion of the I.T.C. about the sprockets in a chain drive system arouse. As a consequence, in the present work the study of an I.T.C. drive system by an unconventional method such as using the Analogous Huygens Pendulum (A.H.P.) is performed. The A.H.P. is characterized by its slow motions which would help in studying the kinematics and dynamics of an I.T.C. in various angular positions during engagement and disengagement of the chain plates of the I.T.C. with the sprocket.

A brief overview of the thesis guidelines is given below:

In chapter two a short history of chain drive systems and their development is discussed, followed by a critical review of the state of the art. At the end of this chapter the objectives of the thesis are established.

Chapter three discusses the kinematic analysis of the I.T.C. in an A.H.P. according to the multibody approach. The model is firstly designed using a C.A.D. software. A block diagram is created in an effort to understand the mathematical relation of the kinematics of a chain in the A.H.P. in general. The model is exported to a M.B.D. software. A multitude of experiments are conducted in analyzing the kinematics of an I.T.C. on the A.H.P. in order to be statistically relevant. The M.B.D. approach is used to track the displacement of the marked links thus determining the overall oscillation of the I.T.C. about the sprocket in the A.H.P. The numerical results are then correlated with the experimental data in order to validate the correctness of the numerical model. The experimental data is also used as a guideline in improving the numerical model.

In chapter four a study of the contact forces in an I.T.C. of an A.H.P. using the rigid body approach is realized. Contact forces are subdivided into normal contact forces and tangential contact forces. The normal contact forces are numerically calculated using the impact forces based on modified Hertzian law for contact forces. The tangential contact forces are also calculated based on the Coulomb's law for frictional forces.

In chapter five a dynamic analysis of the I.T.C. of an A.H.P. using flexible body approach is performed, where the rigid plates of an I.T.C. are meshed into finite elements and nodes, so as to transform the rigid bodies to deformable bodies. Flexible bodies or deformable bodies use the modal displacements of the nodes which indicate the effect of the environment on a flexible plate. The dynamic analysis of the I.T.C. in the A.H.P. using the M.B.D. flexible body approach helps in evaluating the displacement of the nodes on a plate due to the magnitude of the contact forces between the flexible plates and the sprocket.

In chapter six a modal analysis of the I.T.C. on the A.H.P is performed. After determining the "hot spots" of the plate due to the contact forces applied on it and after determining the eigenfrequencies of the plate by the means of a F.E.A. software, a correlation between the maximum principle stresses and the frequencies in a transient response is also established. The last part of the chapter tackles the determined damping characteristics of the I.T.C. plates. This is done by knowing the angular displacements and their derivatives from the kinematic analysis of the I.T.C. in the A.H.P., as well as the rigidity of the plate of the I.T.C., which later helps in correlating the damping ratios to the velocity of the plate.

In chapter seven the general conclusions of the thesis are formulated and some suggestions for future studies are introduced.

2.

Critical Review of the State of the Art on Chain Drive Systems

2.1 General Considerations

A chain drive is a mechanism for transferring mechanical power between two places and is a common means of locomotion in bicycles, motorcycles and automotives. It is also a motive source for many different types of machinery.

Typically, a chain drive works by having a power source, usually a motor or pedal system. The power source rotates a toothed wheel known as a sprocket around which a specially designed chain is looped. As the sprocket spins, its teeth catch slots in the chain drive causing it to rotate around the sprocket. At the other end is a second gear that transforms the mechanical energy delivered by the drive chain into the desired force (Green, 1996).

The main advantages of using a chain transmission drive systems are: 1- they are much cheaper than the gear transmission drive systems (although relatively more expensive than the belt transmission drive systems); 2– they can operate on higher loads.

In this chapter, a brief history of chain drives systems is presented, showing their evolution and why it is important to improve them. Further, a critical review of the state-of-the-art is performed, in order to identify the research problem of this doctoral thesis.

2.2 Brief History of Chains

Chains have been used for centuries to make machines work and move materials on conveyors and up elevators. In 225 B.C. Philo described a chain driven water lift that had the form of a bucket elevator (American Chain Association, 2006). The first continuous power-transmitting chain drive was depicted in the written horological treatise of the Song Dynasty (960–1279) by the Chinese engineer Su Song (1020-1101 AD). He used it to operate the armillary sphere of his astronomical clock tower as well as the clock jack figurines presenting the time of day by mechanically banging gongs and drums (Eedham, 1986). The chain drive itself was given power via the hydraulic works of Su's water clock tank and waterwheel, the latter acting as a large gear (Eedham, 1986).

2.2.1 Cog Chains

The cog chain (American Chain Association, 2006) was developed in the early 1800s to transmit power or motion between the shafts of treadmills to water elevators, weaving looms, and harvesting machinery.

2.2.2 Cast Detachable Chains

The cast detachable chain (American Chain Association, 2006) was introduced in 1873 and overcame many of the problems of the cog chain.

2.2.3 Cast Pintle Chains

The pintle chain is the direct ancestor of both standard roller and engineering steel chain. It was described as a “closed barrel” design, heavier link sections and steel pins or pointless pintle chain.

2.2.4 Precision Roller Chains

A few years after the cast detachable chain was introduced, a chain made of all steel parts was launched for driving bicycles. A patent for roller chain was issued in 1880 marking the beginning of the roller chain industry.

2.2.5 Engineering Steel Chains

Engineering steel chains were first introduced in the 1880s (American Chain Association, 2006). They were developed for greater strength, speed and shock resistance, and for a better dimensional control than that which could be obtained from the cast pintle chains.

2.2.6 Silent Chains

Early designs of silent chains may be found in Leonardo da Vinci’s sketches from the 1500s. However, the first recorded commercial application of silent chains occurred in 1843 with the launch of the SS Great Britain. Sir Isambard Brunel supervised the building of the Great Britain, and it was a revolutionary ship design in many ways.

Throughout the 20th century, the industry improved material quality, processing technology, and chain designs to increase the load and speed capacity of the silent chain. That led to its being used in many demanding industrial and automotive applications, particularly those requiring a compact, high-speed, silent drive.

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2.3 Critical Review of The State of The Art

Silent chain drive systems, also known in the literature as Inverted Tooth Chain (I.T.C.) drive systems, consist of two inner plates, a middle plate and two outer plates. The inner and middle plates are known of having inverted toothed profile. The outer plates are shaped like the roller or bush chains. The I.T.C. drive systems do not depend on the contacts of bushes or rollers against the sprocket. Due to the logarithmic profiles of the I.T.C. plates and the logarithmic profile of the tooth of the sprocket it is hard to detect the position of impacts, as they are surface to surface contacts.

I.T.C. drive systems still encounter many problems due to the contacts of the plates with the sprockets, which cause vibration and noise. Moreover, there is the concern of discontinuity of motion of the I.T.C. system which occurs due to the noticed hexagonal effect during the transmission. This is detected in the automotive industry as well as in conveyor systems.

There have been many attempts to understand the behavior of the I.T.C. system in general in order to eliminate these problems.

The I.T.C. system has a lot of breakdowns due to spin losses in the presence of open clutches and the losses of the hydraulic pump as clarified by Norman K. Bucknar (Bucknar, 2004). Norman K. Bucknar predicted the potential performance of the Travelling Chain Transmission (T.C.T.). A multibody dynamic model of a T.C.T. was used. This model was then validated by a dynamometer test-data.

The simulated transmission is a four speed automatic transmission converted to a five speed transmission by replacing the fixed speed ratio drop chain with the dual speed ratio T.C.T. This is in order to simulate and predict the resulting torques and speed transients from an up shift of the 4^{th} to the 5^{th} gear. It was observed that the automatic transmissions have two primary sources of energy loss excluding the torque converter: a) spin losses due to presence of open clutches; b) losses from the hydraulic pump.

Ishihama Masao and Watanabe Hiroyuki came close to suppressing the noise and vibration of an I.T.C. using induced multibody dynamics simulation (Ishihama & Watanabe, 2010). The study showed the difference between the generating mechanism of an I.T.C. and a roller chain. The roller chain's noise is generated by a polygon motion of its links on a sprocket. However, the vibration of I.T.C. system was observed to be generated at the engagement and disengagement process of the I.T.C. with the teeth sprocket and then being transmitted as waves on the I.T.C. This approach provides a better understanding of the vibrations generated by the mechanism of the I.T.C. system. It was followed by a design of a chain guide that suppresses the vibrations in the I.T.C. systems.

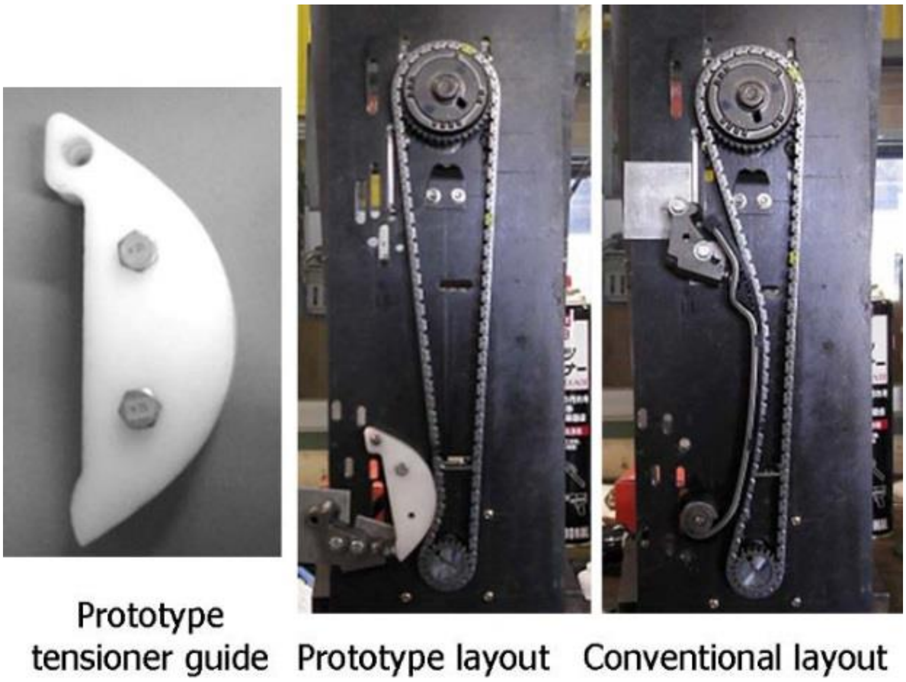


Fig. 2.21 A prototype of the new tensioner designed for low noise generation (Ishihama & Watanabe, 2010)

From Ishihama and Watanabe's paper one can deduce that the new I.T.C. guide design helps in reducing the noise in an I.T.C. system. The new I.T.C. guide also helps in minimizing the tension of the chain. The vibration insulation of the chain guide on the tension side is effective in reducing noise.

Veikos (Veikos & Preudenstein, 1992) developed a more realistic approach for the analytical procedures of the dynamic analysis of roller chain drives. Many test cases were performed using that approach with various types of chains operating at variable velocities, sprocket ratios and loads.

Wang (Wang, 1992) developed a model to study the system's global vibrations and the dynamic behavior of a complete chain transmission. Wang used two axially moving strings where each of their ends was fixed by two rigid sprockets to describe the model. Liu (Liu et al., 1999) analyzed the discrete nature of the chain/sprocket dynamic system. Liu modeled each link as a series of lumped masses connected together by massless springs and dampers.

Troedsson (Troedsson & Vedmar, 1999; Troedsson & Vedmar, 2001) was the first to use the complete geometry with a dynamic model for determining the load distributions along the sprockets and chains. Troedsson divided the chain drive system into four separate parts, each of them being separately analyzed: a- the tight span; b- the slack span; c- driving sprocket; d- driven sprocket.

Mihai T. L., Radu P. (Mihai & Radu, 2016) identified the lubrication contact problem between the chain links and the tensioner guide. Therefore, they attempted to study the dynamic behavior, wear and lifetime caused by the friction contact forces. The authors presented a Finite Element Analysis using a specific software. They calculated the area of contact of different chains and the resultant forces of the tensioner guide applied on the chain spam. Their observations coincide with the results of Gavrilă C.C. (Gavrilă, 2014a; Gavrilă, 2014b), Hyakuta T. et al. (Hyakutake et al., 2001) and Velicu R. (Velicu, 2012) who concluded that close to 25% of the friction is produced by the contact between the chain and the tensioner guide. Mihai T. L. and Radu P. (Mihai & Radu, 2016) arrived at the conclusion that the friction phenomenon in guide chain contacts is influenced by the type of lubricants used and the gap between the tensioner guide and the chain span. Also a steady flow of the lubricant during the motion of the chain span towards the guide should be maintained.

2.4 Conclusions of The State -of-The-Art Review

Chain drives kinetics transfer mechanical motion from the cam shaft to the crank shaft. The sprocket from the cam shaft is the driving sprocket while the sprocket from the crank shaft is the driven sprocket. The mechanical motion in a chain drive system generates vibrations and noises.

From the literature review, it can be observed that many researches were conducted in studying roller and bush chain drive systems. The roller and bush chains have bushes connecting between the inner and outer plates of the chain. The bushes are cylindrical forms that are mostly subjected to impacts when moving towards the sprocket. These types of impacts have cylindrical type contacts that could be simplified to line to line contacts. The top surfaces of the plates are subjected to contacts with tensioner guides. These types of chain drive systems have contacts that are easier tracked, described and analyzed.

Although some research has been conducted on I.T.C. drive systems, there are still a lot of questions to be answered when it comes to improving the level of noise and vibration. The I.T.C.s have logarithmic profiled plates connected with pin joints. The sprockets also have logarithmic profiled teeth causing difficulties in predicting the locations of impacts with the plates and therefore have harder traceability of the plates' trajectories. There are still a lot of drawbacks when talking about I.T.C. systems in general. There are still a lot of experiments needed to better understand the I.T.C. behavior in different applications and conditions such as identifying the response of the system at different applied shocks and over tensioning. Some researchers proved that over tensioning the I.T.C. system won't actually help in decreasing the vibrations and noise, and designed a smaller tensioner blade to decrease the noise (Ishihama & Watanabe, 2010).

Given the critical review of the state of the art in the I.T.C. drive systems and the problems identified as still representing challenges for industry, the main goal of this doctoral research has been set, together with its research objectives, as presented in the following paragraph.

2.5 Research Goals and Objectives

This doctoral thesis addresses one of the research problems related to the I.T.C. drive systems, aiming at developing a study for reducing the level of vibration caused by this type of chain drives. In this respect, the I.T.C. investigation aims to contribute to a better understanding of the behavior of the contacts between plates and sprocket, and how to predict the contact phenomena.

I.T.C. drive systems have great potentials as they are much more silent than other drives. They lose less energy during motion compared to other types of chains and thus improving the mechanical motion transmitted in a chain drive system in general. At the same time, I.T.C. drive systems have complex geometrical shapes which make the task of understanding and predicting the contact phenomena more difficult.

The original idea that underlies this work is to use the Analogous Huygens Pendulum (A.H.P.) created by Prof. Dr. Eng. Șerban Bobancu from the Transilvania University of Brașov, as study model for investigating the kinematic and dynamic behavior of the I.T.C. The A.H.P. is an unconventional system used to differentiate the performance of different types of chains while moving in a slow motion; the plates can be introduced by local markers and their motion can be predicted with the help of a camera. At the same time, the Multi-Body Dynamics (M.B.D.) and Finite Element (F.E.) theories help to develop reliable models and perform numerical simulations that may offer a better perspective of predicting the I.T.C.s' behavior in various scenarios.

In this respect, the following research objectives of the thesis have been set:

1. **The kinematic analysis of the I.T.C. in an A.H.P. according to the Multi-Body approach**
   - ✓ The model development of an A.H.P.
   - ✓ The kinematic analysis of an A.H.P. using the numerical approach;
   - ✓ The kinematic analysis of an A.H. P. based on experimental data.
   - ✓ Correlation of the numerical results with the experimental data.
2. **The study of contact forces in I.T.C. of an A.H.P. using the rigid body approach**
   - ✓ The determination of contact forces by the numerical approach (Multi-Body Analysis).
3. **Dynamic analysis of the I.T.C. of an A.H.P. using flexible body approach**
   - ✓ The numerical modelling of the A.H.P. with flexible bodies.
   - ✓ The dynamic analysis and the determination of contact forces.
4. **Modal analysis and damping characteristics of the I.T.C. plates**
   - ✓ Performing the modal analysis in order to identify the resonance frequencies of the I.T.C. plates.
   - ✓ Identification of damping characteristics of an I.T.C. plate.

3.

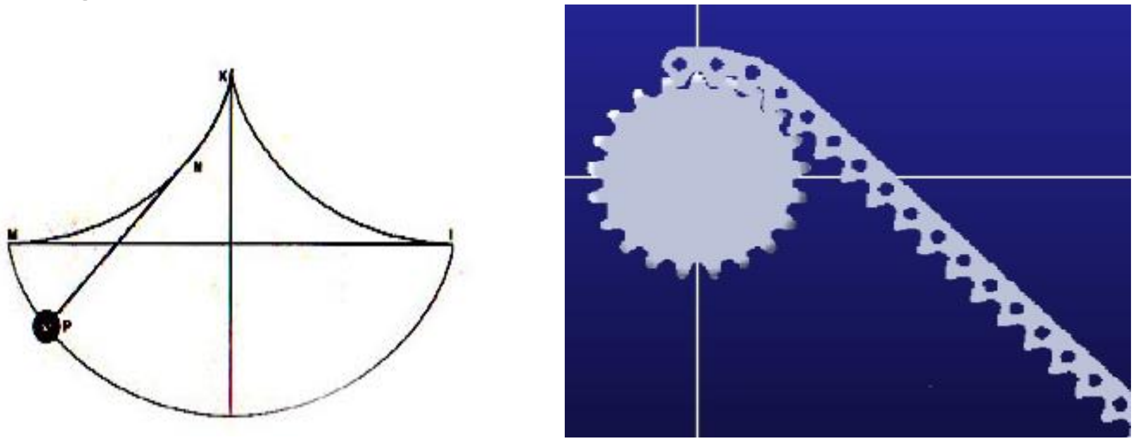
Kinematic Analysis of the Inverted Tooth Chain System of an Analogous Huygens Pendulum

3.1 General Considerations on Inverted Tooth Chains

When designing mechanical systems, it is of utmost importance to understand how the chain drives influence the system performance. As there are different types of chain drives, it is important to question what type behave better in a given mechanical system and why.

This chapter tackles the kinematics of the Inverted Tooth Chain (I.T.C.) drive, considering it is included in an Analogous Huygens Pendulum.

The research conducted in this thesis, in order to better understand the kinematics between the plates of the I.T.C. and the sprocket, uses the multibody approach, assuming that all the bodies (chain elements and sprockets) are rigid. The general purpose is to understand the behavior in different situations of an I.T. chain by applying different conditions. The study is based on the Analogue Huygens Pendulum effect, Figure 3.4, caused by a fixed centroid around which the chain oscillates against the sprocket.



a) b)

Fig. 3.4 a) - Huygens Pendulum (Emmerson, 2006); b) - the model of the studied Analogue Huygens Pendulum

In this way a mono-involute motion is created, where each link during its engagement with the teeth of the sprocket becomes a center of rotation according to the link's position. There is a dead weight at the end of the chain of a fixed mass. It is clear that the free fall of the chain during oscillation determines the speed. The free fall of the chain will also depend on the quality of the chain tested according to the friction between the links, the contact forces and some other criteria. The oscillation of a chain has a low speed as there are no external forces influencing it. Usually, a low speed could give the opportunity to observe the behavior of a chain in a better way.

3.2 The Theoretical Approach

There are considered *N* links in a chain that are connected together with revolute joints. A revolute joint gives one degree of freedom and blocks the rest of the motions. Therefore, one may assume that the motion is taking place only in $\widehat{XZ}$ plane and there is no vibration emitted on the $\widehat{YZ}$ plane. The rest of the chain parts that are not in contact with the sprocket could be considered as a semi rigid body due to an enforced force that would strain the chain.

3.3 The Analogous Huygens Pendulum Model Development

3.3.1 Model Development of the Analogous Huygens Pendulum in MapleSim

In general, in order to understand the kinematic behavior of a chain installed on an A.H.P., a mathematical model for an A.H.P is needed. In this respect, the block diagram approach has been used for developing the model and performing the analysis.

In order to create the block diagram for the A.H.P., three stages are considered: a) creating the links and the pendulum, b) connecting the links with a suitable joint and c) defining the contacts between the links and the pendulum.

- **Creating the Links and the Pendulum**

Any link or body, in general, should contain a rigid body and a rigid body frame, or a flexible body frame. In this case a rigid body frame was used.

In order to describe a link of a chain applied on the A.H.P. in MapleSim, it is important to understand that each link has a mass with a center of mass represented by the rigid body and a diameter. The coordinates of a link and its length are represented by two rigid body frames. The end of the two rigid body frames can then be connected with each other, illustrating the displacement of the whole entire link from one frame to another in the chain during oscillation on the A.H.P. Figure 3.8 represents the construction of a link as a sub block.

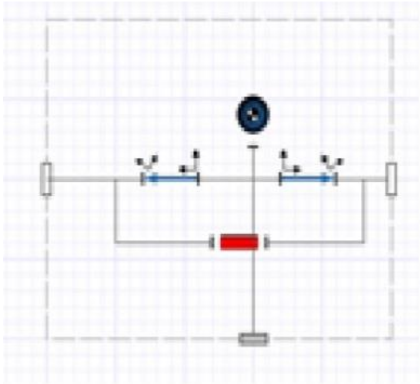


Fig. 3.8 Representation of a link in a sub block,in MapleSim

MapleSim gives the opportunity to represent a link according to the geometry needed, for example the geometric figure could be a cylinder or it could be an image imported from a CAD model.

- **Kinematic Joints**

The main joints used to express the kinematic motion of the links are revolute joints in the A.H.P. system. The revolute joints simply express the rotation of a single axis of one link in respect to the axis of the previous link, thus blocking the translational displacement and the other two axii of rotations.

- **Creating Contacts**

The possibility of creating contacts is flexible in MapleSim by using open Modelica language. The block diagram in Figure 3.10 describes the use of displacement sensor between a link's position towards that of the pendulum represented as a cylinder. The position is then normalized before entering the equation contact.

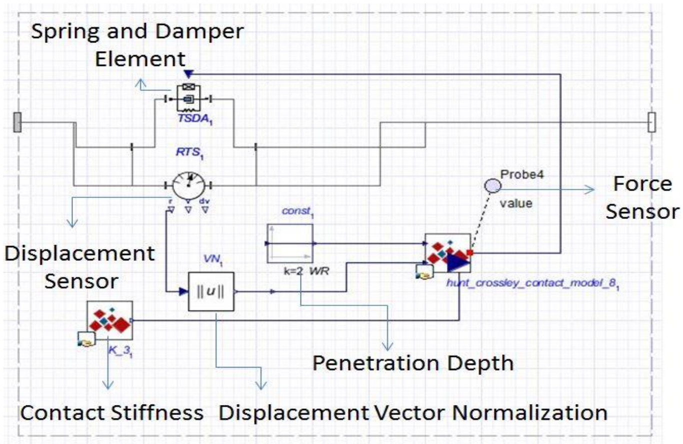


Fig. 3.10 Hunt and Crossley's contact block diagram in MapleSim

3.3.2 Block Diagram of an Analogous Huygens Pendulum

There are several links in a chain that are installed on the sprocket or pendulum in this case. The first link can be considered as a ground connected as it is mounted tightly on the pendulum. The rest of the links are set up to be positioned according to their initial positions. The links sub block diagrams are connected together by the means of the revolute joints. Each joint is interconnected with a spring-damper element to create the tangential contact forces between the links and the joints. Each link except the first link which is grounded are then connected with the Normal contact force sub block diagram. The Figure 3.11 represents the block diagram created in MapleSim.

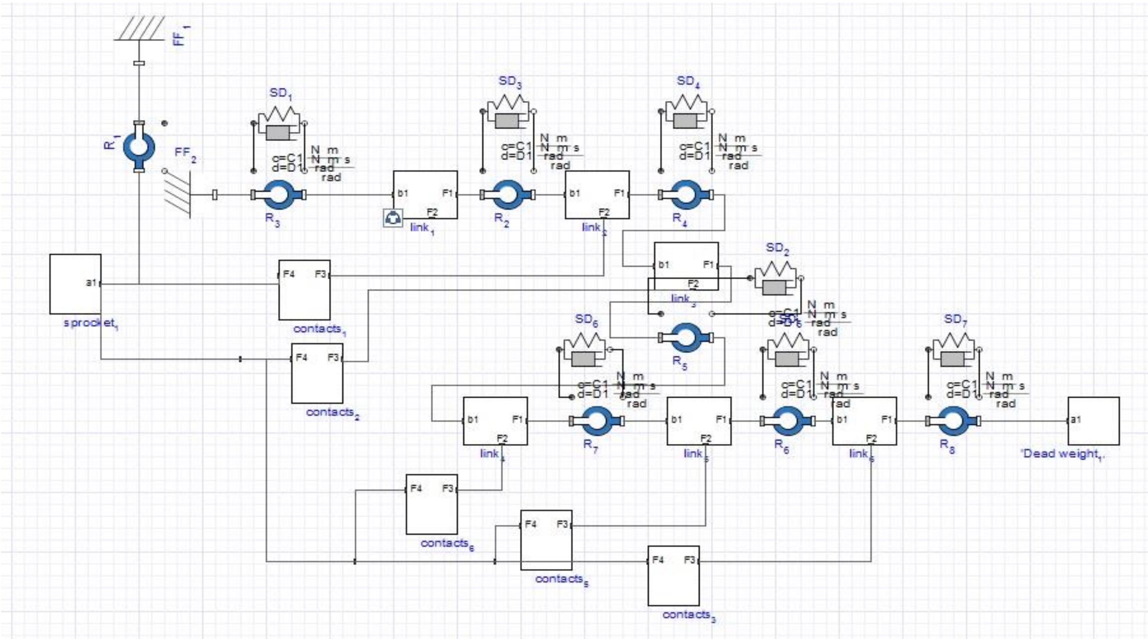


Fig. 3.11 Block diagram of the Analogous Huygens Pendulum in MapleSim

A simplified equation of acceleration is represented below. By simple/ double integration, the velocity/ displacement of a link is determined for its initial position, according to the time frame. In the case below the initial time is zero.

$$-m_{link}\Bigg[\left(\frac{d^2 link}{dt^2}y(t)\right)\sin\alpha(t)\sin\beta(t)\sin\gamma(t) - \left(\frac{d^2 link}{dt^2}z(t)\right)\sin\alpha(t)\sin\beta(t)\cos\gamma(t) + 9.81(\sin\alpha(t)\sin\beta(t)\sin\gamma(t)) - \left(\frac{d^2 link}{dt^2}y(t)\right)\cos\alpha(t)\cos\gamma(t) - \left(\frac{d^2 link}{dt^2}z(t)\right)\cos\alpha(t)\sin\gamma(t) + \left(\frac{d^2 link}{dt^2}x(t)\right)\sin\alpha(t)\cos\beta(t) - 9.81\cdot(\cos\alpha(t)\cos\gamma(t))\Bigg] = 0, \tag{3.22}$$

where:

$x(t)$, $y(t)$ and $z(t)$ define the position of the link with respect to the translation axes;

$\alpha(t)$, $\beta(t)$ and $\gamma(t)$ represent the angles with respect to the rotational axes about oX, oY and oZ respectively;

g = 9.81 $\frac{\mathrm{m}}{\mathrm{s}^2}$ is the gravitational acceleration constant.

The equation helps foreseeing how a link could be positioned according to the time frame of its initial position and the mass of the link. This could significantly reduce the time of simulations just by creating various experiments of all different type of scenarios using different constants. Its use is that it can display a rough calculation or a first impression of how the chain could behave in an A.H.P.

3.4 The Kinematic Analysis of an Analogous Huygens Pendulum Using the Numerical Approach

The kinematic analysis of a certain assembly can be performed by using the analytical (mathematical), numerical and experimental approach. Given the advantages of simulations, widely known and acknowledged, in order to study the kinematics of the A.H.P. a numerical model was developed. As the CAD model allows only seeing how the motion might look like, without the possibility to add the boundary conditions and the connected environment that could affect the results, a multibody software was used, i.e. MSC Adams. It gives the possibility to develop a reliable model, considering all the boundary conditions and the surrounding environment, and further calculates the kinematic parameters and shows, with high accuracy, how the assembly behaves.

3.4.1 Developing the Model in MSC Adams

- **Building the CAD Model**

In order to develop the numerical model in MSC Adams, some initial steps need to be taken. To create the complex geometry of an assembly, parts are built individually aiding C.A.D. tools and then are assembled on the accurate initial positions, using constraints. Figure 3.12 illustrates the C.A.D. model of an assembly of an I.T.C mounted on the A.H.P.

- **Importing the CAD Model in MSC Adams**

One of the main advantages in MSC Adams is that it can import any assembly from any C.A.D. software with various types of extensions such as (*.iges, *. Step, *. Stl,*.x-t, etc.). There is a huge importance to accurately make any assembly according to the needs. During exportation a part can be exported as a part with a defined solid or a part can be exploded into a solid and shells. In this case the model was exported as (*.x-t) to maintain all parts as solids. This is also to reduce all calculations during simulations, as there are many parts in a chain and in the assembly, in general.

- **Setting the Material Types**

The type of materials used to describe the mass properties of the parts used is determined either by using the MSC Adams library or by adding the density required. This helps in finding the moment of inertia of each part used. Also, the center of mass is calculated and can further be used as a marker.

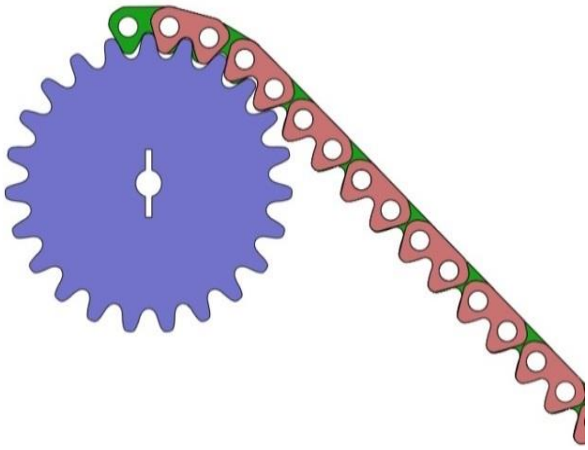


Fig. 3. 12 The CAD model of an I.T. chain mounted on A.H.P.

- **Creating Markers**

In any multybody analysis, markers are vital for constructing the model. MSC Adams gives the possibility of creating markers in a part as an extended geometry which then could be placed at the exact coordinates that are required.

- **Types of Joints Used**

MSC Adams has three main family types of joints (primitive joints, joints and couplers). Any joint has the same main principles: a) selecting the parts required to be joined, b) selecting the markers and the axis which the two parts are joined, c) setting the direction of motion of the joint. In the present model the sprocket is considered as a grounded body, annihilating any motion due to inertias influencing the body and thus creating the centroid center of the A.H.P. The first plate on the sprocket (pendulum) is translationary constrained by the $\widehat{XZ}$ and the $\widehat{YZ}$ planes. The rest are constrained by revolute joints. This would create the boundary conditions of the $\widehat{YZ}$ plane eliminating the $\widehat{YZ}$ displacement of the plates and leading to a calculation reduction during simulation.

3.4.2 Description of the Kinematic Analysis

The kinematic analysis is performed to determine displacements, velocities and accelerations of the I.T. chain during oscillation on the sprocket in the A.H.P., which will give important information on the behavior of the I.T. chain. The results obtained from the numerical simulations will be compared with the experiments, as it is illustrated further in this chapter.

For this analysis the outer plates have been removed. The outer plates give stiffness for the chain so as not to be easily bent. They also act like outer coverings to prevent the chain from slipping during motion. The pins were also removed. The revolute joints were used for the previous reasons as assuming that no slippage would occur.

3.4.3 Results and Discussion

- **Angular Displacements Graphs**

A general aspect to be taken into consideration is that the sprocket has global coordinates, while all the rest of the bodies have their own local coordinates that change in time due to the forces affecting the system at different intervals of time. The angular displacement graphs help to define the position of a certain body *j* in time, as illustrated in Figure 3.16.

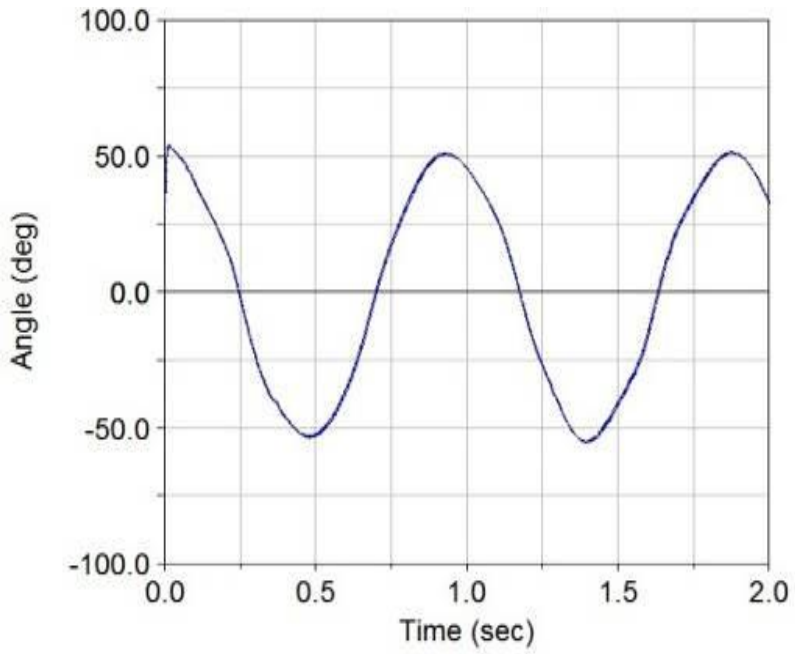


Fig. 3.16 The angular displacement between body_20 and the sprocket (Shalaby and Lache, 2015)

Figure 3.16 represents the angular displacement of the plate *x* just after the last body in contact. The first body has a complete oscillation with the sprocket. The bodies (plates) oscillate with respect to the global coordinates positioned at the center of mass of the sprocket. It can also be observed the double pendulum effect of a body in the chain, further from the impacted bodies with the sprocket.

- **Angular Velocity Graphs**

The angular velocity effect of each body in the chain is represented to its respected order from the angular displacement graphs. Also the Figure 3.20 illustrates the clear first derivation of the angular displacements with respect to time.

It can be observed that as the links depart from the ground link or from where the chain is mounted velocities increase and can be seen as the velocities increase from the magnitude velocities in plates 10-16. This can be caused to higher vibration subjection due to the higher displacements of the plates towards contacting with the sprocket. The plates from 20 to 26 have a slight increase in velocity, yet, there are less vibrations and it is normally due to the speed of the plates at free oscillations about the sprocket.

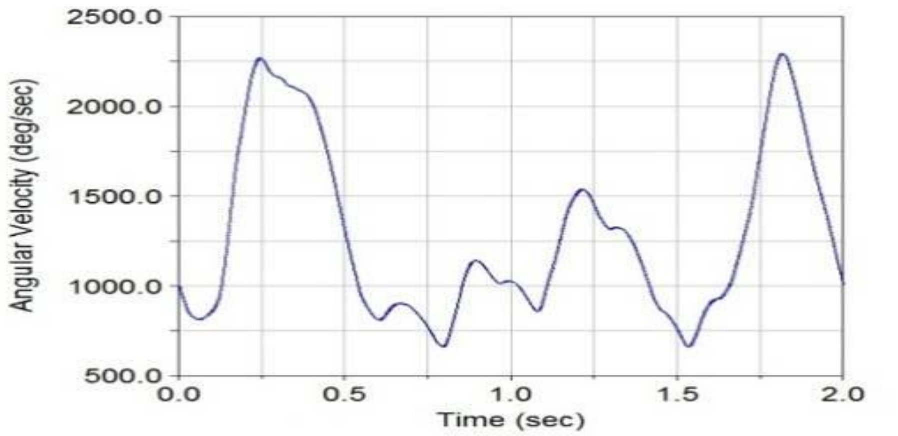


Fig. 3.20 The magnitude of the angular velocity of the body_ 20 (Shalaby and Lache, 2015)

- **Angular Acceleration Graphs**

The Figure 3.28 clearly illustrates the second derivation of the displacement according to time and the vibrational noises caused by the impact of the bodies' "plates" with the sprocket.

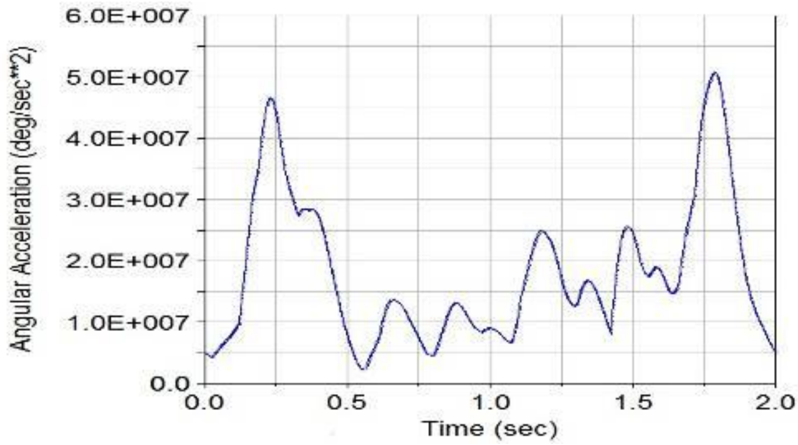


Fig. 3.28 The magnitude of the angular acceleration of body_ 20 (Shalaby and Lache, 2015)

By analyzing the angular acceleration and velocity graphs, one may formulate the following observations:

- The angular velocities and accelerations of bodies subjected to impacts react against the movement of the rest of the chain or the pendulum system as they have higher displacements from the sprocket during contacts causing vibrations. Yet, bodies further away from impact situations tend to respect the total velocity and acceleration of the pendulum system itself.
- Each link of the chain tends to have its own properties. This is because the Analgous Huygens Pendulum does not give a uniform harmonic oscillation, due to the generated centrifugal,

centripetal and centroid forces generated by the impacting bodies, which tend to tilt and un-align the pendulum system.

3.5 Kinematic Analysis of the Analogous Huygens Pendulum Based on Experimental Data

The I.T.C. installed on A.H.P. shows explicitly how the kinematics of an I.T.C. acts. The importance of studying the kinematics is to understand how a plate in an I.T.C. would contact a sprocket in an A.H.P. This would also give a broader view of the kinematics of an I.T.C. plate behavior in a chain drive system. This could prove of having a great importance in determining the characteristics of an I.T.C.

3.5.1 The Experimental Settings

The A.H.P. is used to conduct the experiments. Setting up the A.H.P. might be a relatively challenging process. The adequate sprocket must be installed which would be a proper match for the I.T. chain used in testing. The sprocket must be completely fixed and installed at the correct positioning so that it would allow the proper engagement of the chain installed on it. The sprocket can be considered as the ground. The sprocket's center is the origin of the global coordinates of the A.H.P. The first link of the chain is then fixed to the ground in assurance of no slippage or of any transversal displacement that might occur. The first link of the chain can be considered as the second ground in the A.H.P. The rest of the chain is installed at a precise position taking into consideration the initial positioning. Controllers and motors are used to position the chain at its correct initial positioning.

Fig. 3.33 The A.H.P. – partial installment

Figure 3.33 represents a partial installment of the A.H.P. It can be observed that the sprocket and the chain has been painted so as to get a matt color, not to reflect light and induce the camera to error. The white painting can act as image recognition in the image process of the camera. The white dots on the chain assure the correct alignment of the chain at the initial position. They also help in creating a vector straight line to know the position of the chain during oscillation in respect to the sprocket.

3.5.2 Experimental Results and Discussions

The first experiments resulted in rough data and a significant noise occurred. Three main factors could lead to this: a) - excess amount of light and unwanted reflections, b) - the amount of data input due to the speed of capturing data [F.P.S.] of the camera, c) - the data loss during transfer between the camera and the PC, depending on the interfaces.

In order to increase the accuracy of the processed data, a filtering process was applied and only the maximum peaks of the time intervals taken by very small fixed steps were considered. The filtered data is illustrated in Figure 3.35.

For the reasons mentioned above, tens of experiments were conducted to assure that the correct data of the oscillation was processed. A mean result of the measurements is deduced according to the number of experiments conducted. As shown in Figure 3.35, the experiments have a good repeatability assuring the high accuracy of the results.

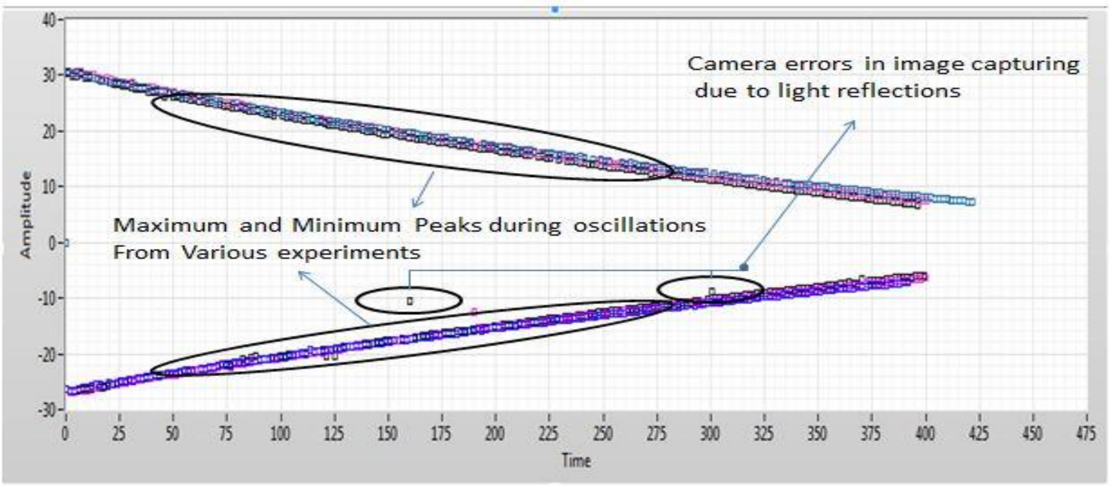


Fig. 3.35 Filtered data of the chain oscillating on the A.H.P.

3.6 Correlation of the numerical results with the experimental data

The experimental results obtained by the A.H.P. are compared to the numerical modeling from the point of view of the I.T.C. kinematics. This section also represents a small benchmark of two different types of I.T.C.s from the lacing and geometrical aspects as illustrated in Figures 3.36 to 3.38. Both the simulation and experimental approach have been considered to illustrate the effect of different geometries and lacings. This section demonstrates which type of lacing and geometry of chains all together will lose less energy and will have less friction. Yet, both chains have fixed pitches assuring the same type of category used on a specific application.

3.6.1 Experimental Results and Discussions

The experiments show differences between the two models. The chart from Figure 3.39 illustrates the difference between two chains of types A and B during oscillations. It is observed the slight difference between the two types during contacts or, in other words, on the oscillation of the chain plates towards the sprocket. The difference between the two chains gapes wider during returning to their initial position. This clearly shows the loss of energy of each type of chain after a contact between the I.T.C. plates and the teeth of the sprocket. Figure 3.39 presents the damping of the oscillations and illustrates the dissipation of energy during contacts of the chain plates and sprocket, as well as the links of the chain plates and their joints.

The upper part of Figure 3.39 shows the oscillation of the chain away from the sprocket, while the lower part illustrates the oscillation of the chain towards the sprocket.

One can notice the difference between the two chains: the difference or the deflection is 2.709% during oscillation away from the sprocket, meaning that chain type B is subjected to a greater loss of energy. Yet, during oscillation towards the sprocket one can notice that there is a really slight difference between them, of 0.945%.

Fig. 3.36 Lacing Type(A) [by the curtesy of Schaeffler Romania SRL]

Fig. 3.37 Lacing Type (B) [by the curtesy of Schaeffler Romania SRL]

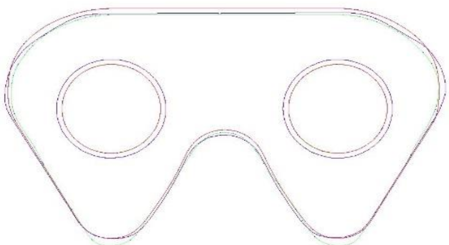


Fig. 3.38 Different geometry contours on chain plates of types A and B respectively [by the curtesy of Schaeffler Romania SRL]

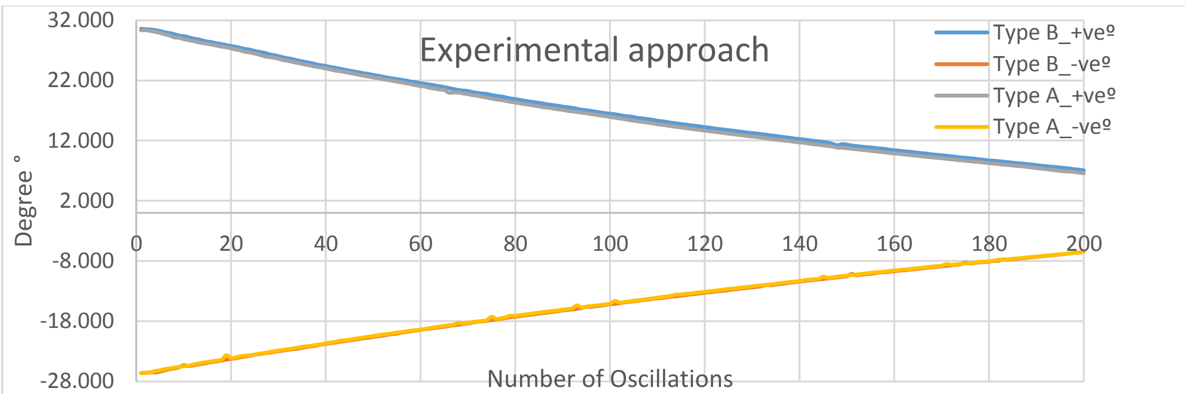


Fig. 3.39 The difference of oscillations between the two types of chains (A) and (B) (Shalaby et al., 2016a)

3.6.2 Simulation Results and Discussions

Two numerical models have been developed. Within the first model, Figure 3.42, each two coinciding plates are reduced to a single plate of double thickness. This is done to reduce the number of joints needed and the number of bodies, in general. In the second model, Figure 3.43, the plates do not exactly coincide with each other, so there is no need to reduce the number of joints. In both cases the outer plates have been removed as they are not subjected to direct contacts with the sprockets. The friction between the plates and their joints is neglected.

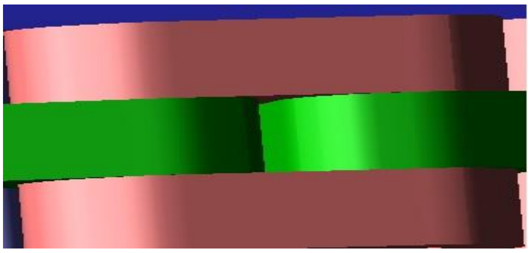


Fig. 3.42 Modified Lacing Type (A) (Shalaby et al., 2016a)

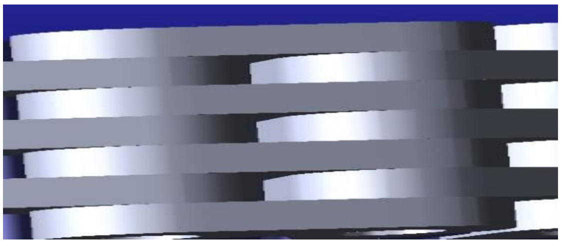


Fig. 3.43 Lacing Type (B) (Shalaby et al., 2016a)

The simulations were conducted for thirty seconds, in order to enable a differentiation between the two chains. The analysis performed on the numerical model allows to compare the results between the two types of lacings and slightly different geometries; at the same time, it offers a better understanding of the difference between each setting (e.g. different thickness, different type of joint uses for modelling). As illustrated in Figure 3.44, the first two oscillations can be neglected as the simulation then stabilizes and the error decreases.

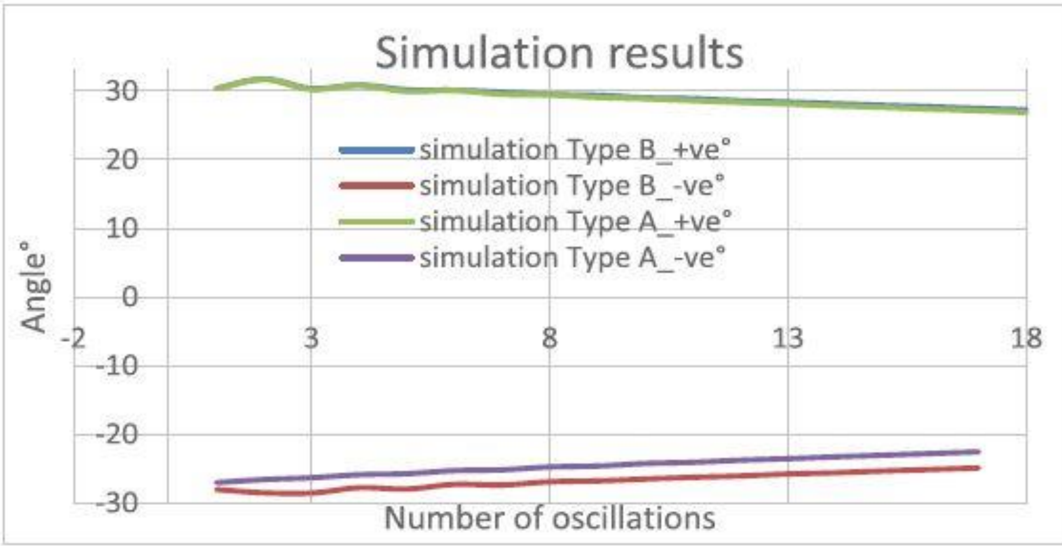


Fig. 3.44. Difference between the two types of chains, illustrated by the numerical simulations (Shalaby et al., 2016a)

3.6.3 Correlation of the Numerical Simulations with the Experiments

There is a slight difference between the experimental approach and the numerical model, as shown in Figure 3.47. A normalization has been used to assure the same starting point for the simulated model and the experimental approach. By comparing the experimental results with the ones resulted from numerical simulations, during the oscillations of the chain away from the sprocket it is observed a loss of potential energy in both cases. However, the differences may be structured in two directions: a) the contact damping factor coefficients are not accurately estimated, b) the tangential forces are highly estimated during contacts of the chains towards their respective sprockets.

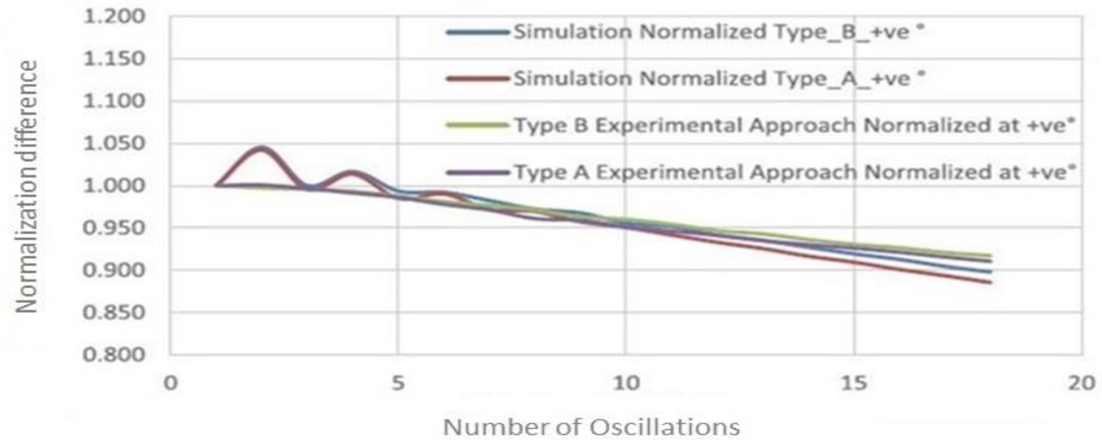


Fig. 3.47 The correlation between the two chains in experimental and numerical approaches (Shalaby et al., 2016a)

Although, the divergence between the results obtained by numerical modeling and by experiments is of 6%, this would still be considered a huge error as the simulations only cover 30 seconds of the entire experimental data. Yet, the results obtained during simulation show very close correlation to the experimental approach. The overall damping curves, resulted from both experimental and numerical approaches, being so similar prove the precision of the multibody dynamic analysis of a system in general. The results also show that type A loses more energy during oscillating back to the initial position. The loss of energy is due to the contacts of the I.T.C. plates with the sprocket. As the geometries are quite similar it is safe to state that the importance of the lacing is imminent in conserving or losing potential energy. More experiments were conducted on trying to find suitable contact damping factors due to the reason previously mentioned in point (a) of the same section.

3.6.4 Contact Damping Factors and Reconstruction of Oscillations

The Contact Damping Factor (C.D.F.) values are important for anticipating the correct kinematic and contact forces behavior. More numerical simulations were conducted to find the proper values of the C.D.F. These factors were calculated according to the energy lost from oscillation between the chain and the sprocket. There is a small range of force damping factors that can be applied reaching nearly the experimental values. This can assure the quality of the chain and what would be expected from the chain's performances and characteristics. There is a huge difference between the damping factor values due to the speed of the system. The speed changes from the retreat of the chain links of the sprocket and the engagement of the chain links towards the sprocket during contact. This leads into dividing or separating the simulation into two main C.D.F. values: a) moving away from the sprocket, b) moving towards the sprocket.

In Figure 3.50 an assumed linear trend of the simulation is shown compared to the entire experimental data. It shows the difference between the experimental data and various C.D.F. values with a maximum error of approximately 2° and 1° at the positive and negative C.D.F. values, respectively. Therefore, the positive 4.95 C.D.F. value from the simulation was taken and combined with the negative 2.05 C.D.F. value in reconstructing the chain oscillation.

There is a noticeable difference between simulations and experimental data of ~ 1° to 2°, which is due to three main reasons.

From Figure 3.50 it can also be observed that there is a change of the kinematic behavior of the chain, due to the speed change during oscillation. Therefore, it can be divided into two main intervals: a) - number of cycles between [0-120], b) - number of cycles between [120-200].

The simulations allow reconstructing the oscillations in respect to the experimental data using the positive C.D.F. values. In this case the positive value 4.95 of the C.D.F. and the negative value 2.05 of the C.D.F. were used.

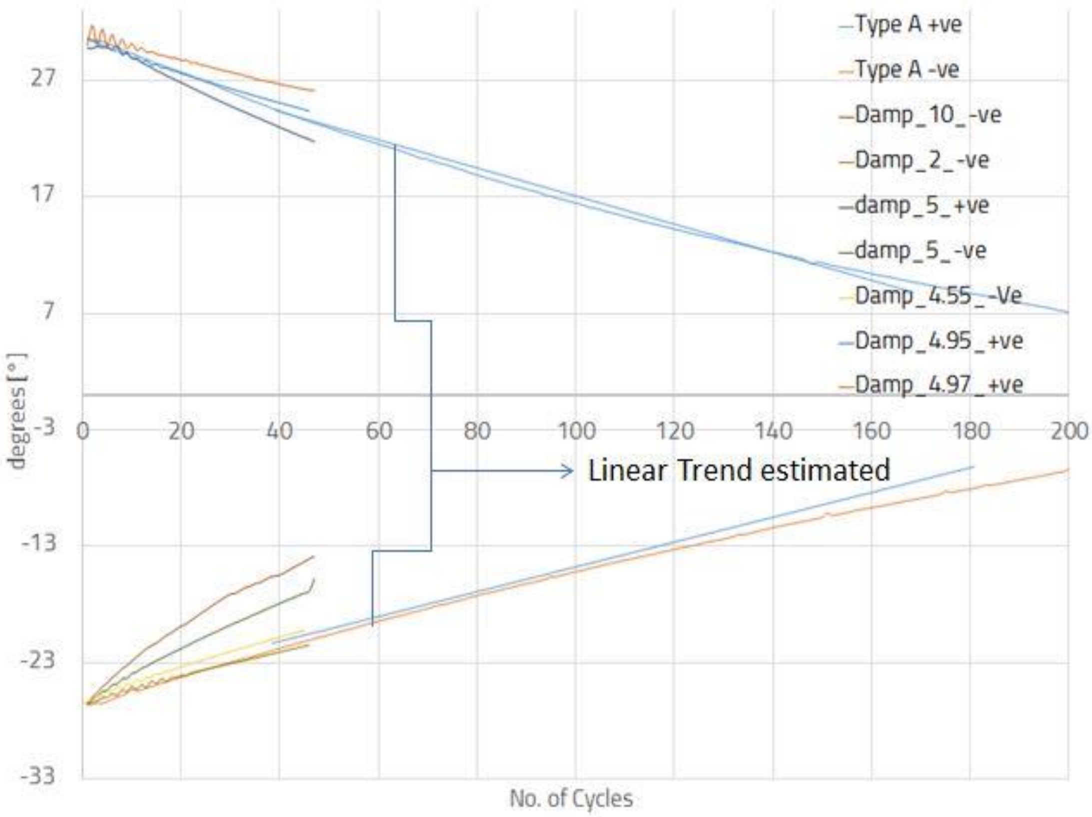


Fig. 3.50. C.D.F. Values in comparing the trend between numerical and experimental datas

3.7 Conclusions

This chapter has tackled the kinematic studies of an I.T. chain in the A.H.P. The chapter discusses the causes of ununiformity of the angular displacements and their derivations (velocities and accelerations) of the chain plates during oscillations along the A.H.P. The effect of double pendulum is clearly observed. It is surely obvious that the system depends on the rigidity of the chain. When the rigidity is higher the effect of the double pendulum tends to disappear. The vibration of the links that are located exactly after the last link subjected to contact or impact with the sprocket teeth dissipates slowly along the chain.

A mathematical modeling of the A.H.P. has been done to create a better background of the kinematics of any chain along the A.H.P., illustrating the behavior of the oscillation in a mathematical equation.

The results obtained during simulation show very close correlation to the experimental approach. The overall damping curves, resulted from both experimental and numerical approaches, being so similar prove the precision of the multibody dynamic analysis of a system in general. The chapter also tackles the effect of different types of chain lacings and different contour geometries of different types of I.T. chains. The differences show that type A chain loses more energy during oscillating back to the initial position. The loss of energy is due to the contacts of the I.T.C. plates with the sprocket. As the geometries are quite similar it is safe to state that the importance of the lacing is imminent in conserving kinematic energy during oscillations.

It is of great importance to determine correctly the Contact Damping Factors (C.D.F.) when solving the kinematics of an I.T. chain in the A.H.P. It can be concluded that the I.T. chain has two different behaviors that occur along the timeline of the oscillation. This behavior could be categorized as large angular displacements and small angular displacements. It could also be concluded that the I.T. chain has different behaviors when oscillating towards the A.H.P. or the sprocket as well as when the I.T. chain oscillates away from the A.H.P. or the sprocket, on the condition that the upper links of the chain are in permanent contacts with the sprocket. Both conclusions demonstrate that the I.T chain has a non-linear behavior and should be divided into four different analyses that lead into finding different C.D.F. coefficients in solving the I.T. chain behavior.

4.

The Study of Contact Forces in Inverted Tooth Chains on the Analogous Huygens Pendulum Using the Rigid Body Approach

In this chapter, the research results presented in Chapter 3 are used in determining the contact forces.

As contact forces can be the reason for the vibrations and noise induced in an I.T.C. drive system, their study is needed and it is supposed to lead to a better understanding of the dynamic behavior of an I.T.C. in the A.H.P.

4.1 Study of Contact Forces in Inverted Tooth Chains. The Theoretical Background

Studying contact forces in mechanical systems has always been a topic of interest, as they provide useful information on the dispersion of energy in the system. Contact forces appear as a consequence of dynamic phenomena, so they vary in time. Contacts are associated to plastic, like in the cases of crash situations, or elastic, like in the cases of an elastic ball hitting the ground which could be also called impact.

The velocities of colliding bodies change rapidly and their reactions to contacts are impulsive. According to the Young's modulus E, stiffness K and damping coefficients C of each body, contacts can be classified into elastic contacts, plastic contacts and elasto-plastic contacts.

The contact forces are divided into two main forces, the normal contact force and the tangential contact force. The normal force depends on the external forces that increase the speed of a moving body towards another body, the Contact Damping Factor (C.D.F.), stiffness contact coefficient and the penetration depth. While, the tangential force depends on the friction coefficient and the velocity during contact meaning if the speed is higher than static friction velocity or lower.

4.2 The Determination of Contact Forces by the Numerical Approach

There are two types of numerical analyses that help in determining the contact forces:

a) theFinite Element Analysis (F.E.A.),

b) the Multibody Dynamics Analysis (M.B.D.).

4.2.1 Setting Up the Model in MSC Adams

In creating the contacts between the I.T.C. plates and the sprocket in the A.H.P., the initial setting was explained in chapter three, section 3.4.1.

The outward normal definition, which is the calculation of the outward normal of a geometric graphic by defining the type of material, determines the direction of the contact normal forces. For three dimensional solids, which is the case in the present model, the geometric graphics are closed by definition. The outward normal is included in the geometric description and there is no open interpretation in its definition.

There are many types of contacts used in MSC Adams such as (solid to solid, curve to curve, point to curve, curve to plain, flexible body to solid, flexible body to flexible body, etc.). In this case solid to solid type of contact was used, were the solids of the moving plates were considered as solid I and the fixed solid of the sprocket was considered as solid J at all times.

The normal contact force calculation is then defined. The software presents two methods for calculating the normal force: the Impact Force Method and the Restitution Force Method.

The tangential contact force used in the software is determined by calculating the friction force. The software also uses a relatively simple velocity based friction model for contacts using Coulomb's laws that was applied in calculating the tangential contact at the A.H.P.

4.2.2 Theory Based Calculation of Contact Forces in MSC Adams

The contact statements in MSC Adams/SOLVER software give the possibility of defining two or three dimensional contacts between a pair of geometric objects.

The software models the contact force as a unilateral constraint, in other words, the penetration depth between the plate of the I.T.C. and the sprocket. This means that a force has zero values when no penetration exists between the specified geometries. However, the force obtains a positive value when subjected to penetration between the two specified geometries.

The contact statements in the software support the multiple contacts, the dynamic frictions and the contacts between three-dimensional solid geometries (Troedsson & Vedmar, 2001).

4.2.3 Results and Discussions

The results of the simulations lead to the contact forces generated from to the impact between the I.T.C. plate and the corresponding sprocket in the A.H.P. The results also highlight the effect the angular displacement and its derivations have on the contact forces, as discussed in Chapter 3.

By considering the first set of pairs of the I.T. chain plates totally constrained to the sprocket, the possibility of creating contact forces or friction forces between them is eliminated. As the next set of pairs of the I.T. chain plates is not constrained to the sprocket, but rather constrained with the rest of the links of the chain, one will notice the contacts between the links and the sprocket as shown in Figure 4.4.

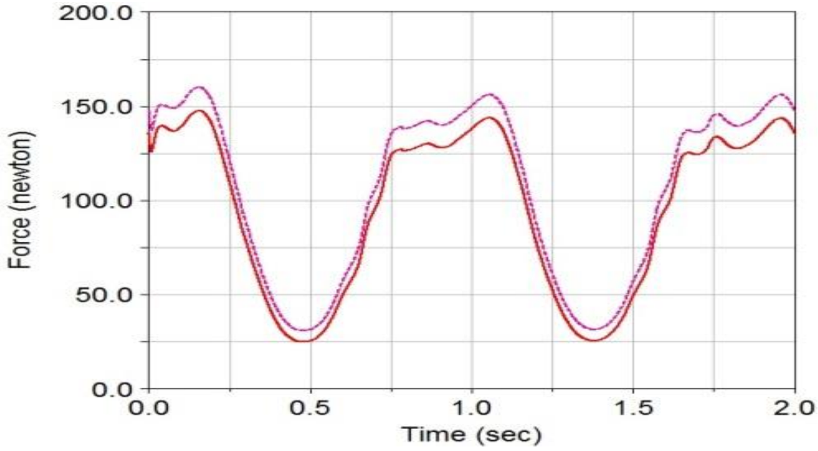


Fig. 4.4 Contact forces of the second set of pairs (Shalaby et al., 2016b)

The distance between the plates and the sprocket is too small, thus forcing the plates and the sprocket to be always in contact. This creates friction forces which are permanent and relatively high during the period of oscillation. In general, when the distance between the plates and the sprocket increases, the contact forces of the plates against the sprocket increase whereas the friction forces decrease.

In order to understand the behavior of the distribution of forces during contact, one can distinctly divide the contacts of a plate with the sprocket into four main zones as shown in Figure 4.5:

a) the engagement zone where the plate has a first main contact with the sprocket;

b) the second zone which clearly shows the friction of a plate with the sprocket;

c) the disengagement zone where the plate spontaneously disengages the sprocket;

d) the contactless forces zone which marks the end of one oscillating cycle of the plate toward the sprocket. (Emmerson, 2006) (Shabana, 2013).

By studying the contact forces along the next links on the chain, where the distance of the plates of the links grows further apart from the sprocket, it can be noticed that each set of links has slightly different behaviors. It would have been expected during contact that the engagement force would always be greater than the disengagement force. However, with the decrease of the engagement period of time and the increase of the disengagement period of time while the total contact period of time decreases, the disengagement force tends to be greater than the engagement force.

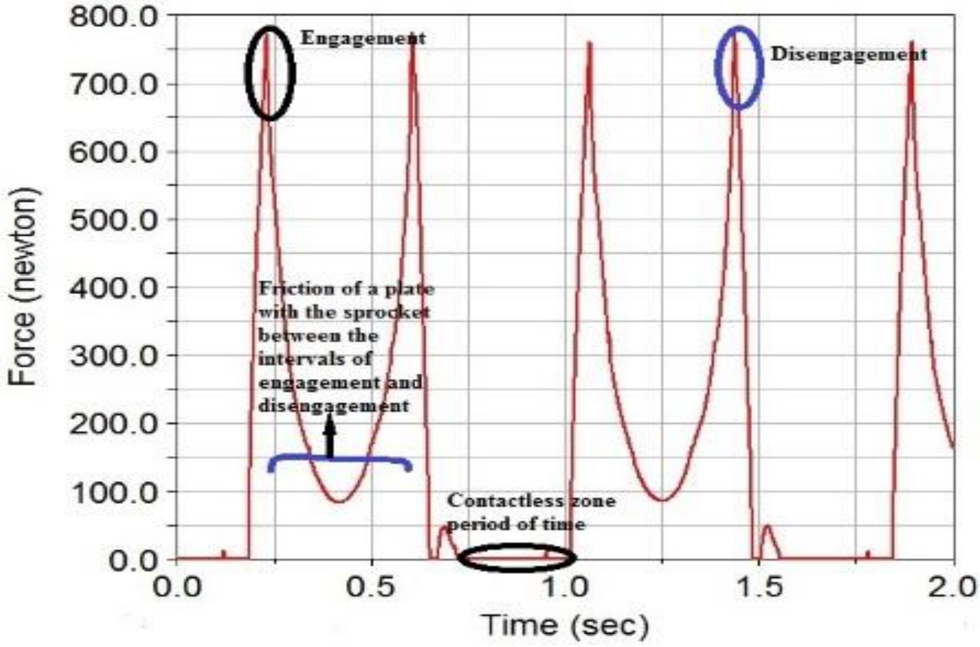


Fig. 4.5 Illustration of each zone (Shalaby et al., 2016b)

Throughout large contact periods of time the engagement forces between the plates and the sprocket are higher than the disengagement forces, as illustrated in Figure 4.6.

In the Analogue Huygens Pendulum system, the contact forces can be divided into four categories (Shalaby et al., 2016a):

- The impulse impacts are shortly lived and the friction forces mainly prevail due to an insignificant distance between the plates and the sprocket.
- The first intermediate distance shows that the impulses of engagement are higher than the forces of disengagement. Yet, the overall forces would slightly dissipate in time due to the damping coefficient of the materials.
- The second intermediate distance, slightly bigger than the first intermediate distance, illustrates that the forces of engagement are smaller than the forces of disengagement as the overall period of impact decreases.
- The distance becomes very large and the impact forces start to fade. The impact is very shortly lived. One can anticipate a reduction in friction forces, therefore the disengagement forces nearly disappear whereas the shocks of previous links of the chain are nearly insignificant, yet still exist.

In Chapter 3 section 3.6 shows various kinematic simulations were conducted to determine the correct C.D.F. values, in order to reconstruct more realistic results of the simulations so as to resemble them to the experimental results. The reconstructed oscillation of the plate located in the third set of pairs was calculated using the numerical method discussed in the previous section of this chapter.

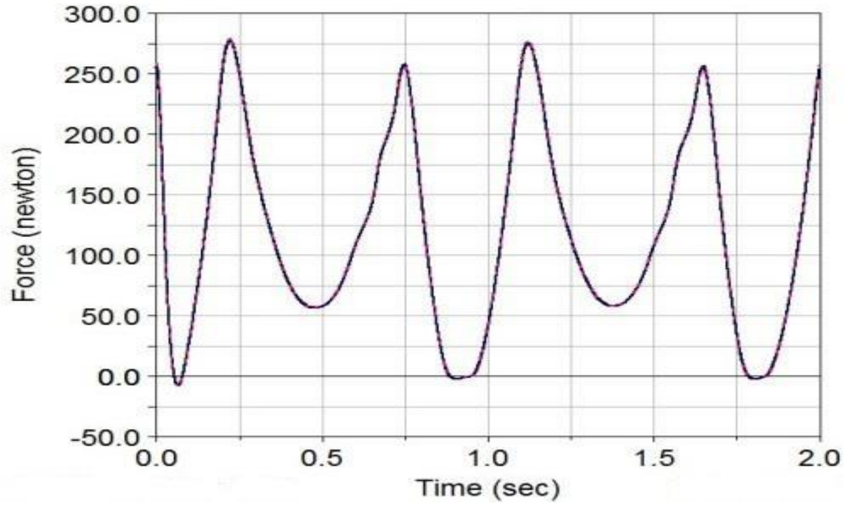


Fig. 4.6 Contact forces of the third set of pairs (Shalaby et al., 2016b)

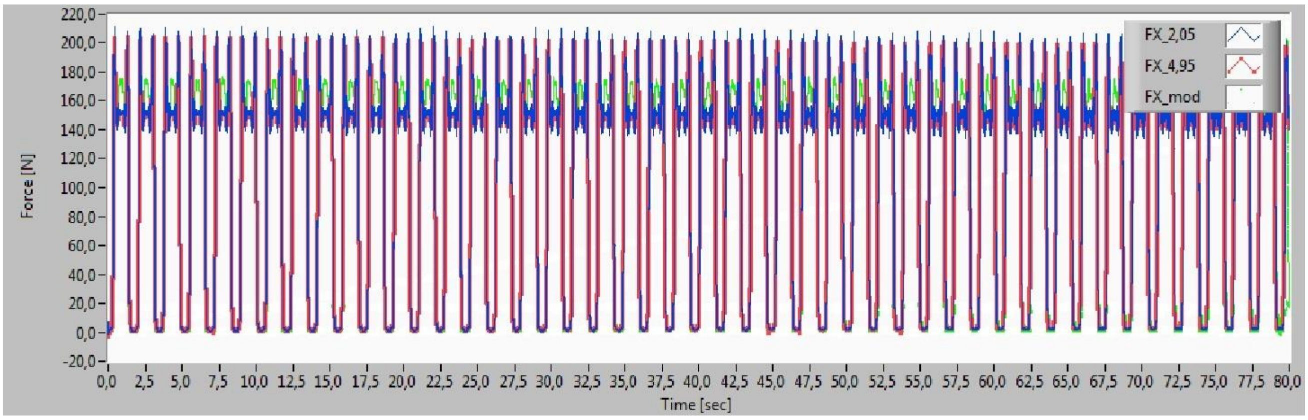


Fig.4.12 Contact forces of the third set of pairs

There is a slight difference between the simulations and the reconstructed contact forces as the length of contact on the reconstructed contact forces slightly exceed the simulation results shown in Figure 4.13. From Figure 4.13 one can observe that the tangential forces have higher values in the reconstructed contact forces than the simulation results. These observations can be concluded from the slight change in the angular displacements of the plates between the different C.D.F. values and the reconstructed angular displacement.

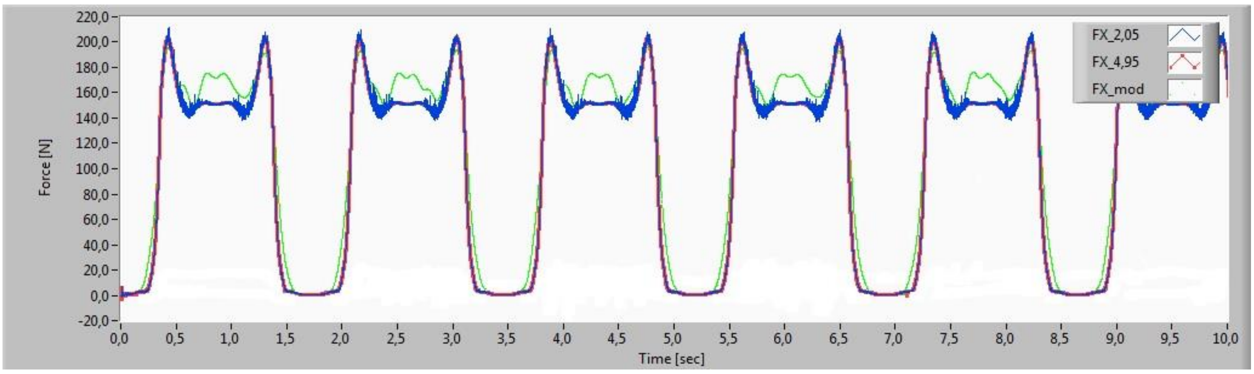


Fig.4.13 A magnified view of the contact forces of the third set of pairs

4.3 Conclusions

This chapter demonstrates the effect of the contact forces during the oscillation of the I.T.C. towards the sprocket in the A.H.P., in the sense on why the angular displacement magnitude diminishes.

The correlation between the angular displacement and its derivatives in anticipating the magnitude of the contact forces has been discussed

It has been put into evidence the importance of the I.T.C. stiffness, which is dependent on the type of material used and the geometric form, in finding the contact force and the Contact Damping coefficient.

The contact forces are considered as two main categories:

- the normal contact force - calculated using modified Hertzian laws of contact, that is influenced by the stiffness of the bodies in contact, C.D.F. values, the penetration depth between the marker points of the two bodies in contact, and the force exponent, known also as the exponential restitution coefficient.
- The tangential forces - calculated using the friction forces into consideration. The friction forces take into account the Coulomb's law of friction and rely on the static friction coefficient, dynamic friction coefficient and the transition velocities.

The discussion on contact forces of the chain plates on the sprocket in the A.H.P. is clearly represented within the numerical approach presented in this chapter. The chain in the Analogue Huygens Pendulum behaves nearly like the system of an automobile or any industrial application, from the cyclic point of view. In the present system, the plate takes one cycle to oscillate along the sprocket, whereas in the automobile a plate has to complete a revolution along two or more sprockets.

After studying the I.T.C. system in the Analogue Huygens Pendulum, whether kinematically or from the contact forces point of view, resulted a better understanding of the form of contact forces, the different velocities, accelerations and kinetic energy. One can observe that there are different types of contacts in different forms.

5

Dynamic Analysis of the Inverted Tooth Chain System of an Analogous Huygens Pendulum Using Flexible Body Approach

5.1 General Considerations

In the previous chapters, the kinematics and contact forces were discussed between the plates of the I.T.C. and the sprocket in the A.H.P. Yet, the studies conducted tackle the I.T.C. in the A.H.P. as being rigid bodies and do not examine the phenomena occurring due to the contacts of the I.T.C. towards the sprocket in the A.H.P. This chapter studies in more depth the effect the contact forces have upon a plate in the I.T.C. using the flexible body method (M.B.D. analysis).

Geneally, contact forces affect the performance of a mechanical system mostly in a negative way, causing the loss of energy and generating vibrations and noise. The purpose of this study is to understand the behavior of an I.T. chain in different situations like the different angular contacts and velocities between the plates of an I.T.C. and its corresponding sprocket. This can be obtained by looking at the nodal displacements of the plates in contact, placed at different initial positions of the I.T.C. in the A.H.P. It becomes easier through flexible body analysis to know in general the maximum stresses and deformations of a plate in the I.T.C. during contact with the sprocket by calculating the general mass, general stiffness and the critical frequencies of the system.

5.2 Theoretical Background

In order to understand the flexible multibody method used for the dynamic analysis of the I.T.C. of an A.H.P., it is important to define the flexible bodies. There are many methods of defining flexible bodies and maybe one of the most important definitions from the point of view of multibody dynamics is that of Shabana (Shabana, 1997), and presented in the following section.

5.2.1 Flexible Body Definition

Flexible bodies were defined by Shabana (Shabana, 1997), using the moving frame approach within the setting provided by the reference point coordinates in 1989. The natural coordinates of the body unequivocally are used to define the large overall rigid bodies motion to which the elastic deformations variables are referred. Overall one can conclude that flexible bodies can be defined as the rigid bodies displacement multiplied by the coefficient of the elasticity of the body, to express the deformation caused on the body (Shabana, 1997).

The natural coordinates of a body do not include relative translations or rotations and are subjected to the corresponding rigid body constraints. The constraints in flexible bodies are different than those of rigid bodies, as points and vectors cannot be shared at the joints, since the elastic deformations should be included.

Floating frame reference formulation was suggested by Shabana (Shabana, 2013).

5.2.2 Theoretical Approach of Contact Forces

Regarding contacts there are many theories about how to obtain reasonable results of contacting bodies. This is mainly because more complicated systems have a huge number of D.O.F.s due to many bodies that can fall in contact with one another and are in motion relatively with each other. Literature has assumed three main types of contacts according to the rigidity or flexibility of the parts in contacts: rigid-rigid bodies, flexible-rigid bodies and flexible-flexible bodies. One main and important reason to study contacts is to see the propagation of stresses and strains affecting the parts during impact or contacts, which can only be done when one of the bodies is considered flexible or deformable.

In order to simulate their behavior, all flexible bodies are represented using the finite element approach, thus the number of D.O.F.s, from infinite becomes finite. Each part has its own local reference frame (local coordinate system) that is defined by a position vector of a global reference frame.

5.3 Numerical Modeling of the A.H.P.

5.3.1 Flexible Bodies in MSC Adams

MSC Adams Flex-Bodies considers a small linear deformation at the local form reference frame (Flores & Machado, 2001). The small linear deformations can be approximated as superposition of a number of shape vectors. The shape vectors can be determined with a modal frequency analysis called modal superposition. This can be done by MSC Adams/FLEX or any F.E.A. software that can be used in performing the modal analysis (MSC Adams, 2013). The analysis calculates how many mode shapes are needed as well as their corresponding natural frequencies of the modes. The results of this analysis are stored as binary files or modal neutral files (*.mnf) that MSC Adams can import and represent the flexible bodies.

5.3.2 Theoretical Background of MSC Adams Software

The first step for analyzing flexible bodies is to find the natural frequencies and the eigenmodes of an undamped system. This means that it is necessary to create an equation of motion with free vibration on condition that no external forces are to be applied on the parts.

5.3.3 The Modal Superposition Theory

The location of a node *P* is defined by the vector from the ground origin to the origin of the local body reference frame *B*. This is quite similar to that of Shabana's Floating Frame Theory, wher*e* $\vec{S}_p$ is the position vector from the load body reference frame of *B* to the un-deformed position of point *P* and $\vec{U}_p$ is the translational deformation vector of point *P*.

This conveys an understanding of the kinematics created in MSC Adams when equations for flexible bodies are to be resolved. The importance of such data is that it illustrates how the chain plate structure is affected during contacts. This leads to the next topic which is how does MSC Adams calculate contacts.

5.3.4 Developing the Numerical Model of the A.H.P. in MSC Adams

The A.H.P. model has originally been setup as discussed in the previous chapters so as to assure the correct coordinates of each part in the A.H.P. The plates are then converted from rigid bodies to flexible bodies where a (*.mnf) file is created in MSC Adams/FLEX. In creating a (*.mnf) file one must take into consideration some steps such as meshing the plates, setting up the generalized damping and the positioning of the flexible bodies.

5.3.5 Contacts in MSC Adams

MSC Adams doesn't totally base the calculations on the Hertz theory for calculating contact forces and stiffness. The geometrical shape of the plate and the geometrical shape of the sprocket are also important in order to give MSC Adams the ability to detect the bodies entering contacts. The stiffness of the bodies that are subjected to contacts is also important.

5.3.6 Results and Discussions

The importance of studying the model from the flexible multibody dynamics point of view is to see the effect of the plates colliding with the sprocket from the perspective of kinetic energy, deformation and frequency response. As mentioned before, for reducing the magnitude of simulation, only the plates are considered flexible bodies. It can be said that the motion of the dead body weight in time represent the total amount of kinetic energy loss during motion, as shown in Figure 5.1.

The kinetic energy loss is due to the contact forces and the friction forces between the plates and the sprocket, as well as the friction between the plates and the pins (which are represented in the model as revolute joints). According to the facts mentioned above, it is safe to say that the I.T.C. loses energy in an exponential form and that the I.T.C. has a good damping behavior in its natural form and also to multiple shocks without adding external excitation forces.

The exponential damping of the system gives the opportunity to understand why I.T.C.s are also called "silent chains".

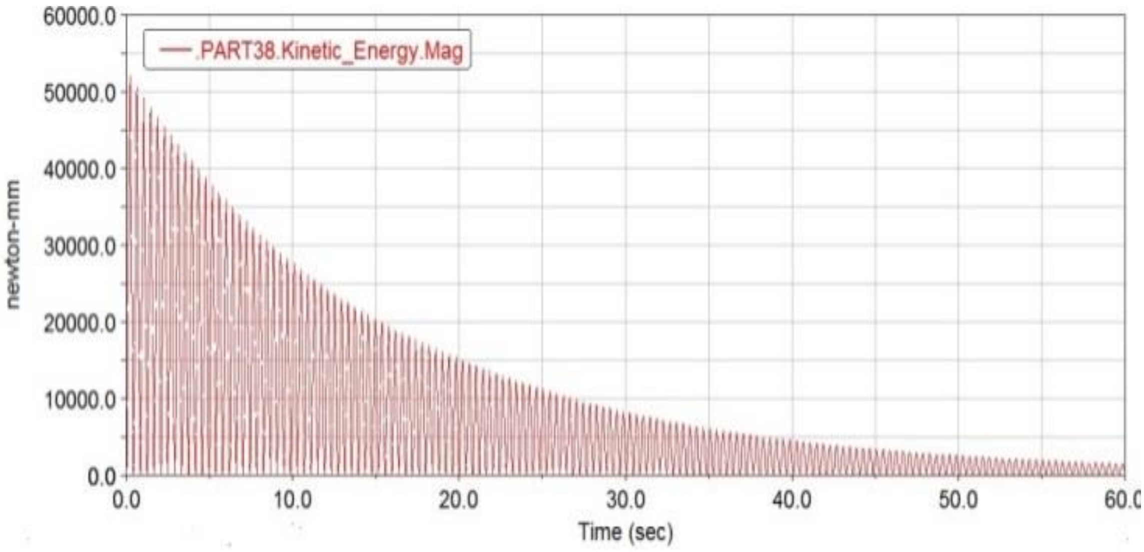


Fig. 5.1 The kinetic energy of the deadweight (Shalaby et al., 2015)

One can easily observe how the clean accelerations help in understanding the increase or decrease of the amount of vibrations induced due to the contacts of the moving plates with the fixed sprocket. This can be seen in Figure 5.3 by following the development of a body's acceleration and its corresponding frequency respons.

At the same time, it is possible to observe the increase of vibrations when the first moving body has the highest vibration in contacts, as shown in Figure 5.3.

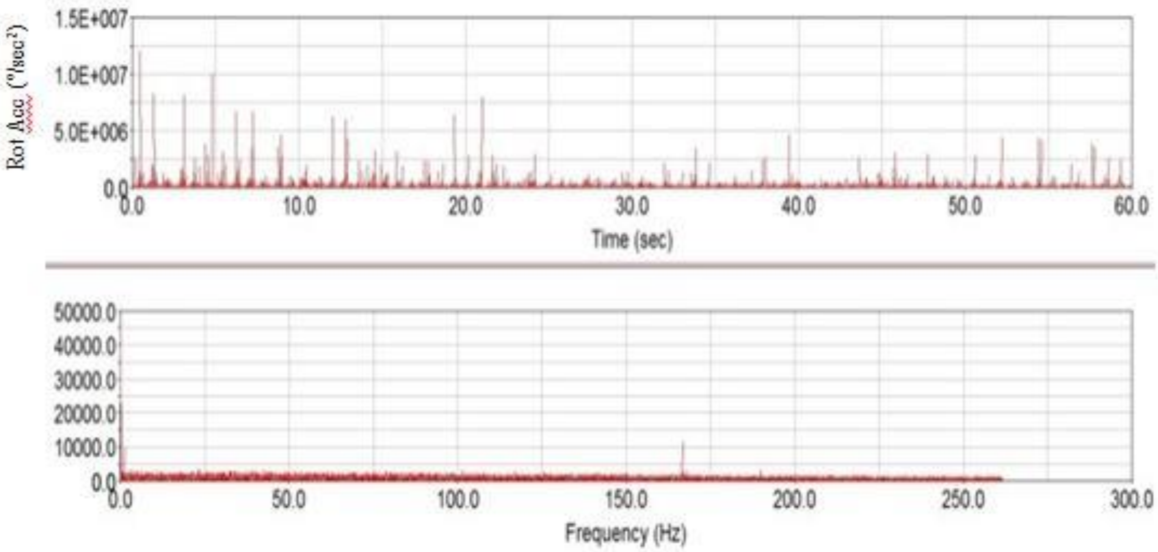


Fig.5.3 The acceleration of the first body in contact, in time and frequency domains (Shalaby et al., 2015)

The highest contacts of pair plates of the chain can be simply subdivided into main intervals. One can notice the engagement, friction and disengagement impacts to form a total contact of the body, as shown in Figure 5.5.

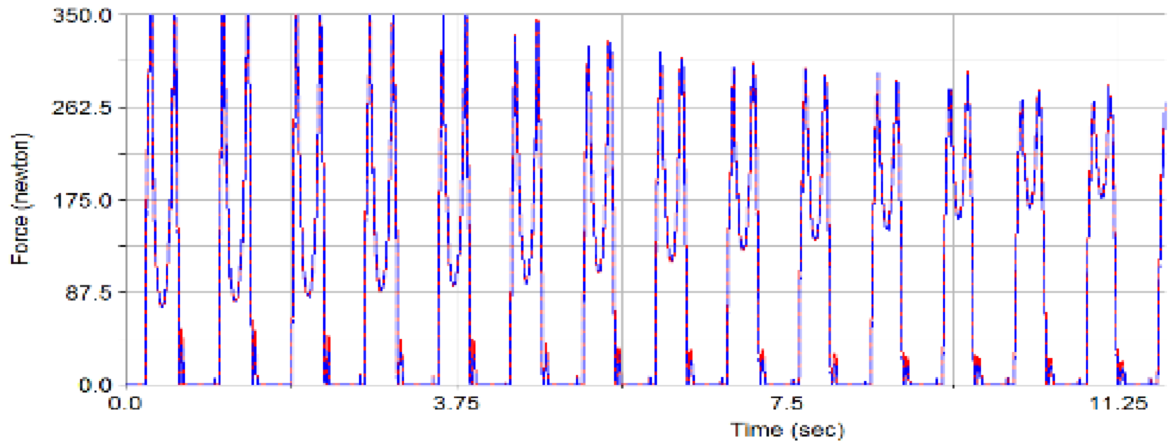


Fig. 5.5 Contact forces during the first interval (Shalaby et al., 2015)

The second interval illustrates the creation of the impulsive impacts due to the total loss of the kinetic energy in the chain during motion causing the chain to reach the rest position. The contact will simply dissipate and will take the form of the last body in contact with the sprocket in the form of the second interval of a normal plate position in contact, meaning the impulsive contacts, as shown in Figure 5.6.

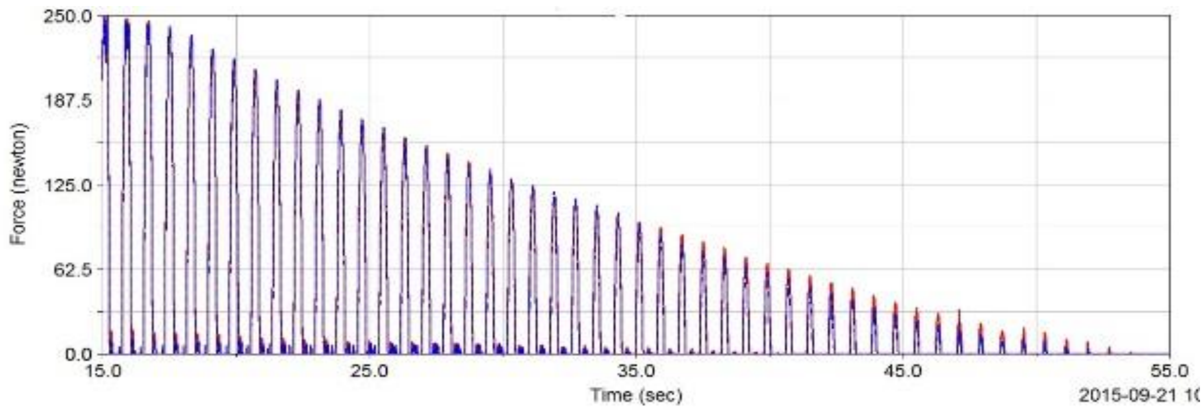


Fig. 5.6 Contact forces dissipating till reaching impulsive contacts or the second interval of contacts (Shalaby et al., 2015)

Figure 5.9 indicates the impulsive contacts with very low bandwidth and the magnitude of the contact forces that simply disappears till no impulsive contact is observed, meaning that the oscillation doesn't have the sufficient power to raise the plate to contact the sprocket.

One can observe how the motion of the deadweight simply decelerates from the time and frequency perspectives, due to the contacts of the chain plates with the sprocket and the friction of the joints connecting the plates of the chain with each other.

One can approximately indicate the period and location of the first contact of the plate with the sprocket and the small spikes of pre total contacts. The impulsive spike contact decreases the magnitude force of the total contact of the plate. Figure 5.11 and Figure 5.12 indicate approximately the location of the first impulsive contacts.

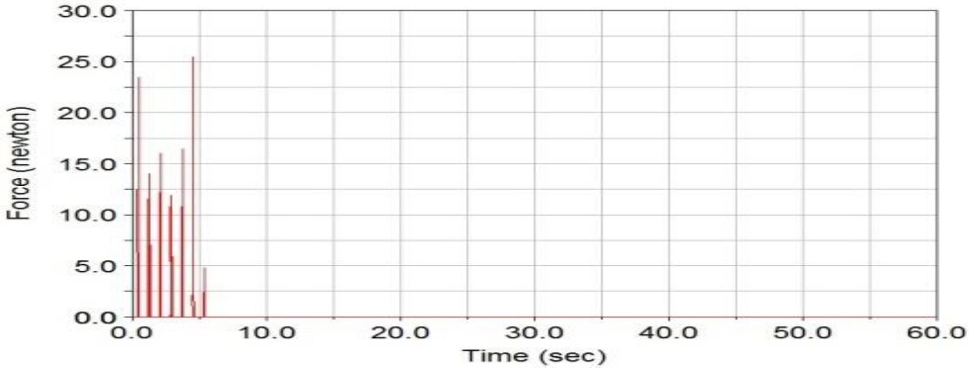


Fig. 5.9 Impulsive contact forces (Shalaby et al., 2015)

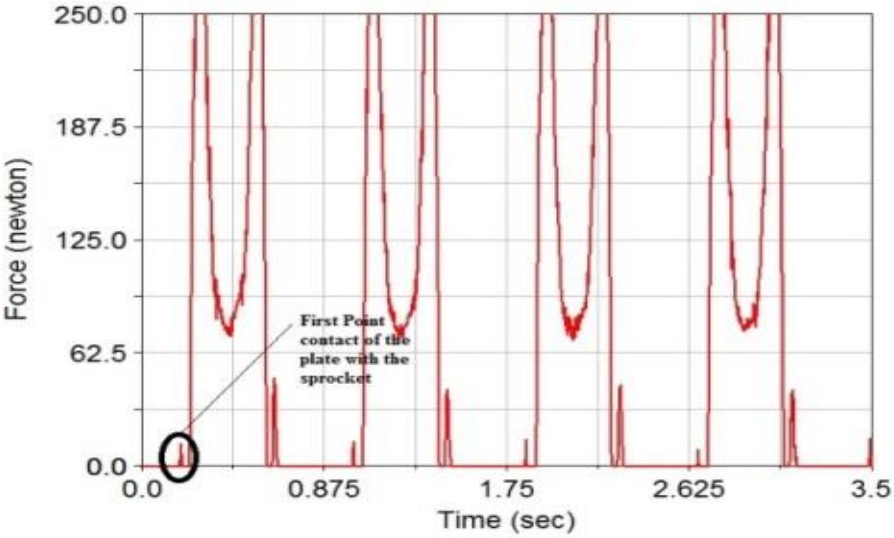


Fig. 5.11 First contact location of a plate with the sprocket (Shalaby et al., 2015)

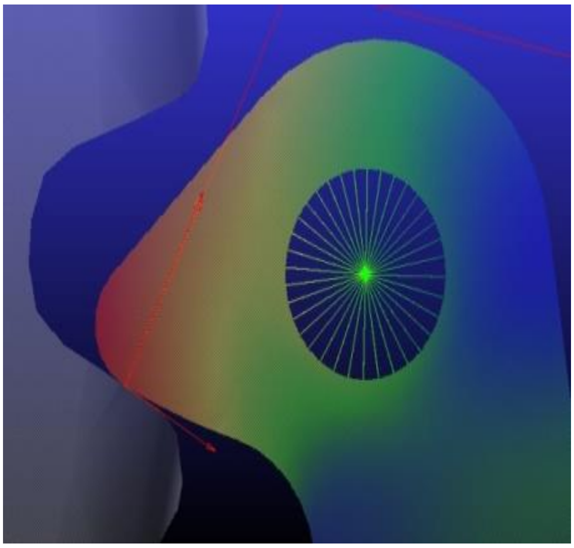


Fig. 5.12 First contact partial view with deformation spectra due to contact (Shalaby et al., 2015)

The results illustrate the displacements of the nodes during contacts which could lead to more stresses at the center of the plate and at the pitches which would be discussed in the next chapter.

5.4 Conclusions

In this chapter are presented the results of the analysis of the dynamic behavior of the I.T.C. on a A.H.P using the Flexible Multi-Body approach. This type of analysis gives a better perception on the Analogous Huygens Pendulum motion time domain and frequency domain. It also gives the prospect, in an exponential form, of how does the system decelerates.

Along all the results of the simulations, the following conclusions can be formulated (Shalaby et al., 2015):

1. As the angular gap between the plates and the sprocket is larger, the vibration produced becomes smaller giving the impression that the length is in inverse proportion to the frequency.
2. The single connecting plate suffers much more contact than the two plates in a link, as the surface area is much bigger.
3. A first contact on the plate with the sprocket is detected. This gives a reduction of the contact force acting like a first aid damping for the impact, even if the values are small. The first point of contact simply gives a sudden deceleration in order to not have a full contact at a single step.
4. One can realize if there is no first contact at the inner side of the plates during oscillation and contacts. The magnitude of the contacts is much higher and gives a higher distortion or vibration of the system. To obtain a first contact depends also on the angle of the plates coming in contact with the sprocket. The double contact of the I.T.C. gives an exponential deceleration as shown clearly at the displacements of the system and the deceleration curve.
5. The first contact assures a better positioning and assures a better engagement in order to reduce the total deformation of the chain plates during contacts.
6. The system in itself suffers during low frequencies and as the frequency increases the system gives a more stable behavior.

6.

Modal Analysis and Damping Characteristics of the I.T. Chain Plates

6.1 General Considerations

This chapter analyses further the effect of vibrations on the I.T. plate, as well as the frequencies that affect the plate. The purpose is to determine and avoid the critical frequencies or prevent the plates to reach their natural frequencies during oscillations (and, thus, to avoid resonance).

At the same time, finding the damping ratio of an I.T.C. plate, by solving the equation of motion of a plate during oscillation is important in order to understand the characteristics of a plate. Damping ratios also, help in identifying the cause of kinetic energy loss during motion.

Most of the studies conducted in the chain drive system were about the contacts between the top surfaces of the chain plates and the tensioner blades. However, the question of "how can one diminish stress increase over certain limits in contacts?" is still present. Attempts have been done for a better understanding of the contact phenomena but there are still unsolved issues related to the behavior of chain plates towards sprockets, as discussed in previous chapters.

The Analogous Huygens Pendulum is an efficient tool for such a research as it can lead to a better understanding of an I.T.C. Also it helps in finding common behaviours of an I.T.C. in a chain drive system.

6.2 The Numerical Modeling and Analysis

6.2.1 Modeling and Analysis Using the Multibody Approach

The chain under study has a 2-2-1 lacing, meaning that there are two outer plates, two inner plates and one middle plate. The movement of the elements is reversed, that means the sprocket is fixed and the chain is movable. The model setup in MSC Adams has been discussed in the previous chapters. Only one flexible body is used for modelling the chain; the initial position of the flexible body is shown in Figure 6.1.

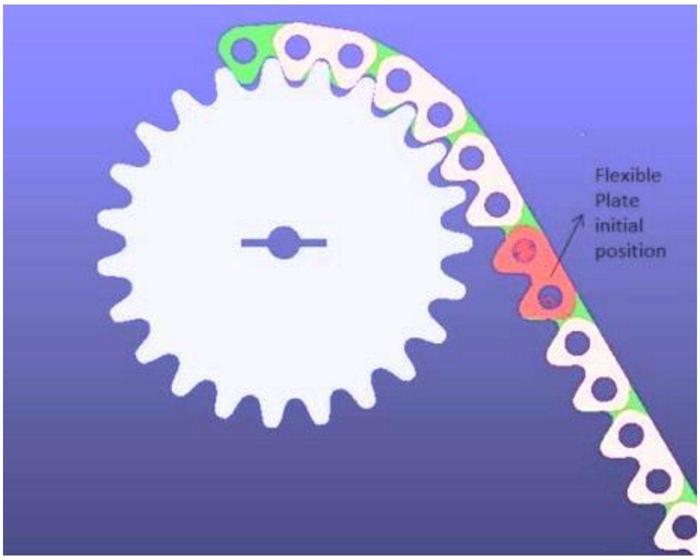


Fig 6.1 Sprocket and the I.T.C with the flexible body highlighted (Shalaby et al., 2017)

The model was set to run for ten seconds. This gave the possibility of calculating the maximum and minimum principle stresses of the flexible body thus indicating the "hotspot", the highest node element affected at the plate, indicating the most affected zones caused by the contact between the flexible body plates of the I.T.C. towards the sprocket, as illustrated in Figure 6.4.

According to the calculations performed, the maximum principal stress was determined as 628.1 MPa at $t = 0.674$ s, in node 4089.

The analysis gives the possibility to measure the contact forces between the plates of interest and the sprocket, and the forces at the joints. The outer plates are not contacting the sprocket in question during contacts, the forces in the joints being divided by three, as three bodies hold that joint. The magnitudes of the forces at the joints which were considered at the same moment in time are listed below, in Table 6.1.

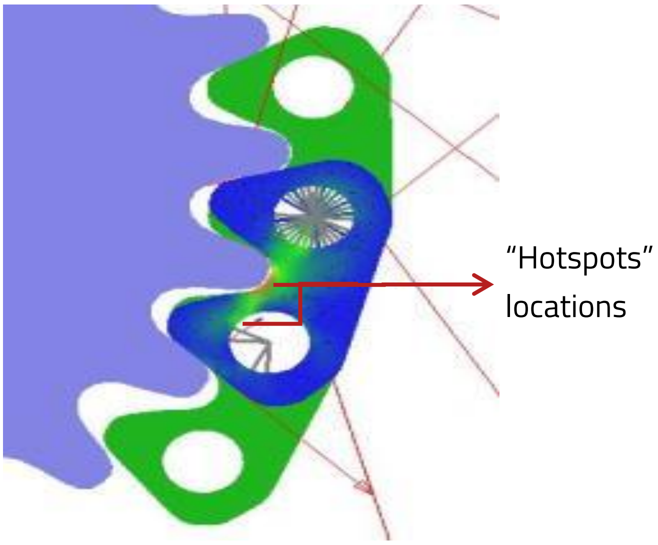


Fig. 6.4 The most affected stress zones due to contacts (Shalaby et al., 2017)

Table 6.1 The magnitudes of the forces at the joints (Shalaby et al., 2017)

Joint	Magnitude of force at the joint [N]
Plate 12 - Plate 13	192.79259
Plate 13 - Plate 15	241.66594

6.2.2 Modelling and Analysis Using the Finite Element Approach

The exact position of each plate has been calculated at that exact position in the frame. These results have been further used for the finite element analysis, performed with ABAQUS software. For developing the finite element model, the precise position of the model at the specific frame time was set in order to consider the forces as static forces. The F.E. model developed for plate 13 has been investigated under the boundary conditions, frame by frame. In the next step, the joints between the plate holes have been added, by means of introducing kinematic couplers, the couplers being totally constrained. The couplers, representing the pins joining the chain, are considered as not deformable.

Assumptions on pins being rigid and having low value of surface roughness have been applied, for simplifying the calculations. Consequently, it has been assumed that the friction forces between the joints were low enough. The upper hole at plate 12 and the lower hole at plate 15 have been fully constrained, yet leaving them to rotate freely around the Z-axis, representing the reactions of the plates due to the imposed forces.

The outside left and outside right surfaces of plates 13 and 14, illustrated in Figure 6.5, are translational constrained on the Y-axis in the $\widehat{YZ}$ plane. They replicate the teeth of the sprocket reactions on the chain plates. Preloaded forces have been added on the holes of plates 13 and 14, as they had been previously calculated in the MSC Adams analysis, at the specific moment. Hexa-form elements have been used for meshing.

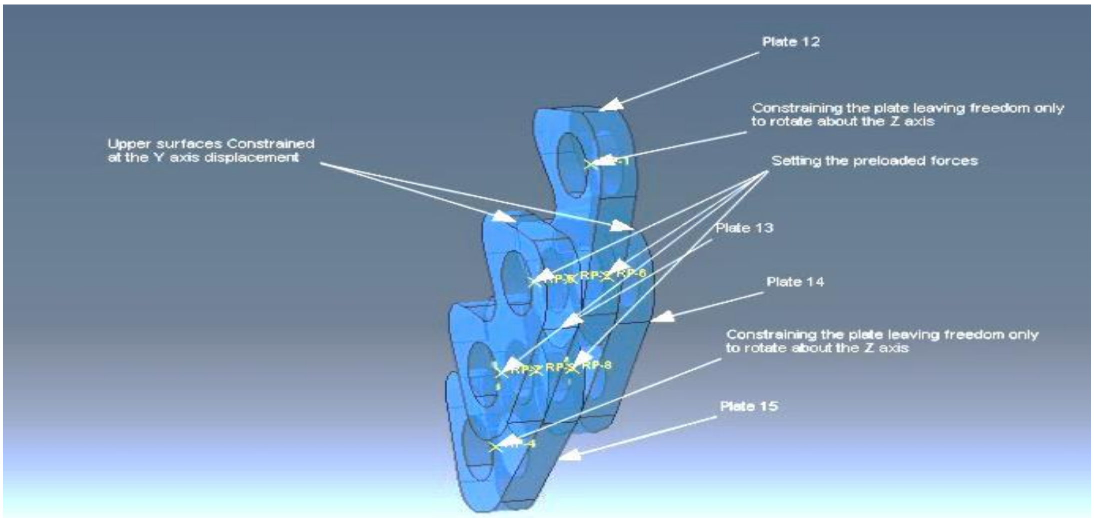


Fig. 6.5 The setup of the model in Abaqus (Shalaby et al., 2017)

The main aim of the dynamic analysis performed is to calculate the eigenfrequencies for which a steady state dynamics of the system is to be obtained.

6.2.3 Results and Discussions

The first six eigenfrequencies and the corresponding eigenmodes (excluding the eigenfrequencies and eigenmodes of the rigid body motion, which result in 0 values) resulted from the dynamic analysis performed using the finite element method.The frames from the transient response were used for their identification.

Random nodes of plate 13 were considered for further interpretation of results. These nodes are mostly affected by the stresses imposed on them. The nodes presented in Figure 6.6 are distributed between the two teeth and the external left plane of the plate. They have been selected in order to see the evolution of stresses in terms of frequencies applied to the plate.

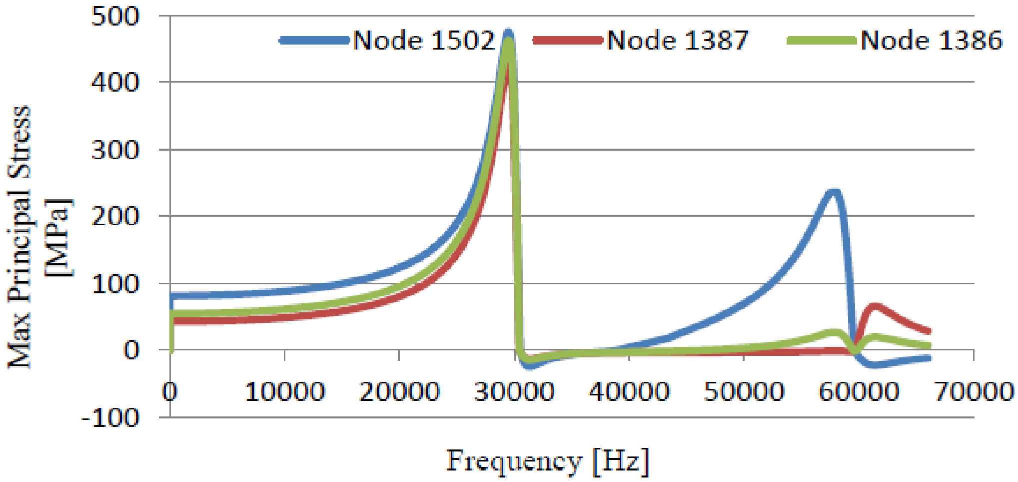


Fig. 6.6 The magnitude of maximum principal stress and the corresponding frequencies (Shalaby et al., 2017)

At the same time, as the frequencies increase, the maximum principal stresses and displacement of nodes decrease, as seen in Figure 6.7 to Figure 6.12.

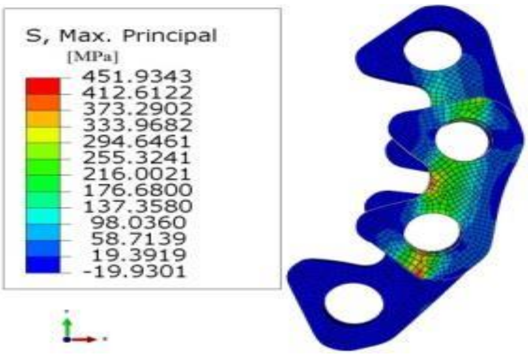


Fig. 6.7 Distribution of the maximum principal stress for mode 2 (f = 27955 Hz) (Shalaby et al., 2017)

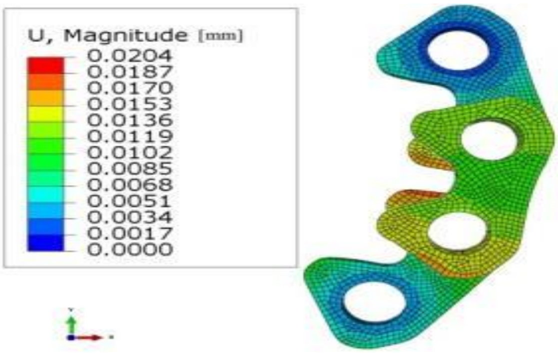


Fig. 6.8 Distribution of displacement field for mode 2 (f = 27955 Hz) (Shalaby et al., 2017)

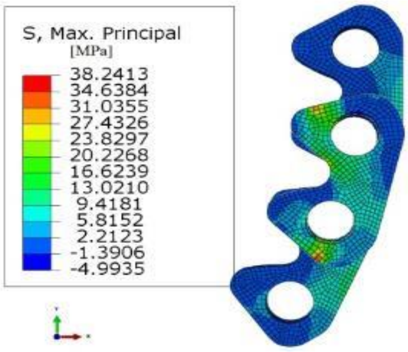


Fig. 6.9 Distribution of the maximum principal stress for mode 3 (f = 30430 Hz) (Shalaby et al., 2017)

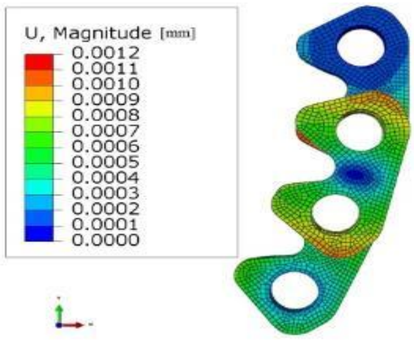


Fig. 6.10 Distribution of displacement field for mode 3 (f = 30430 Hz) (Shalaby et al., 2017)

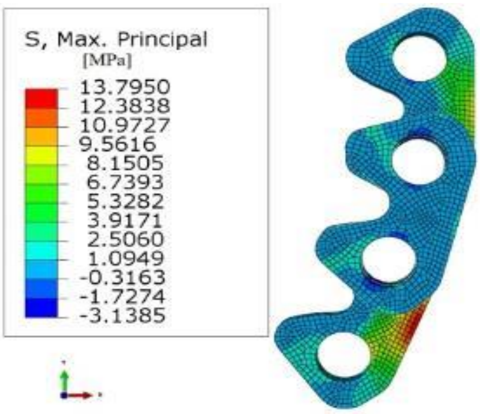


Fig. 6.11 Distribution of the maximum principal stress for mode 5 (f = 59635 Hz) (Shalaby et al., 2017)

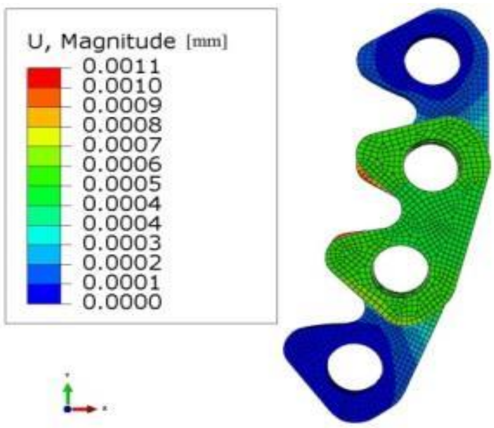


Fig. 6.12 Distribution of displacement field for mode 5 (f = 59635 Hz) (Shalaby et al., 2017)

From the Figures 6.7 to 6.12, it can be observed that the most effective frequencies range could be for modes 2, 3 and 5. It is important to notice the distribution of the maximum principal stresses and the displacement of the nodes on the plates at the specified modes.

6.3 Damping Characteristics of an I.T.C. Plate

From chapter three section 3.4.3, it was possible to obtain angular displacements, velocities and accelerations. The results give the possibility to obtain the damping ratios of a plate according to various velocities using a one Degree of Freedom (D.O.F.) equation of motion.

The chain plate has been constrained by the means of adding rigid shells at the holes (pitches) of the plate. The first shell at origin has been totally constrained and on the other shell various forces have been applied on the X direction of the plate.

Figure 6.14 presents the maximum displacements of the nodes in an I.T.C. plate due to the forces exerted on the plate in the X-axis direction, thus giving the possibility to calculate the linear stiffness of a plate. The linear stiffness will then be used in a one D.O.F. equation of motion.

The angular displacements, velocities and accelerations calculated in section 3.4.3 and 3.6.4 are then substituted in the general equation of motion (Francis, 1994) expressed below. The equation helps in determining the damping coefficient that would lead in calculating the damping ratio, assuming that no external forces are applied because the reaction forces have already been expressed by the displacements, velocities and accelerations coordinates.

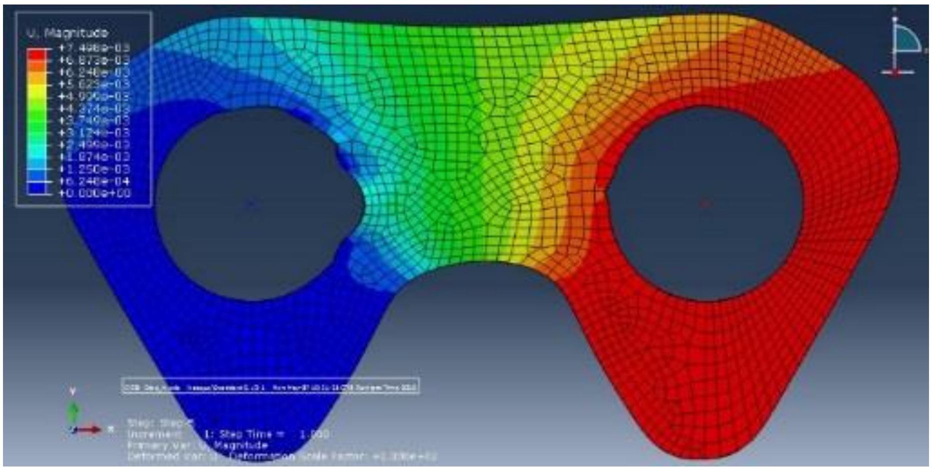


Fig. 6.14 Displacement of nodes of an I.T.chain plate

$$I \cdot \{\ddot{\theta}\} + C \cdot \{\dot{\theta}\} + K \cdot \{\theta\} = 0 \,, \tag{6.6}$$

where:

K–is the linear stiffness calculated by the means of Hook's law and ABAQUS;

I– is the inertia magnitude of the sum of moment of inertias of a plate in the I.T.C.;

$\ddot{\theta}$ - is the angular acceleration of a plate in the I.T.C.;

$\dot{\theta}$ – is the angular velocity of a plate in the I.T.C.;

θ – is the angular displacement of a plate in the I.T.C.;

C– is the damping coefficient of a plate in the I.T.C.

Where ω_n is the angular velocity depending on the natural frequency of the flexible plate (Francis, 1994)

$$\omega_n = \frac{1}{2 \cdot \pi} \cdot \sqrt{\frac{K}{I}} \quad . \tag{6.7}$$

The damping ratio ζ is then calculated (Francis, 1994)

$$\zeta = \frac{\omega}{\omega_n}, \tag{6.8}$$

where ω is the angular velocity depending on the induced frequency applied on the flexible plate .

The Figure 6.15 indicates the exponential decay of the damping ratio towards velocity. Since there are many complete oscillations, calculating the numerical Equation (6.6), results in estimating more than one damping ratio for a single velocity. Therefore, a curve fitting the data from the results of the Equation (6.6) was calculated in effort of finding the correlation between the damping ratio and the velocity in a plate. The correlation can be expressed as:

$$\zeta(\%) = 62.92 \cdot exp\left(-0.6591 \cdot \dot{\theta}\right) + 6.385 \cdot exp(-0.05017 \cdot \dot{\theta}) \tag{6.9}$$

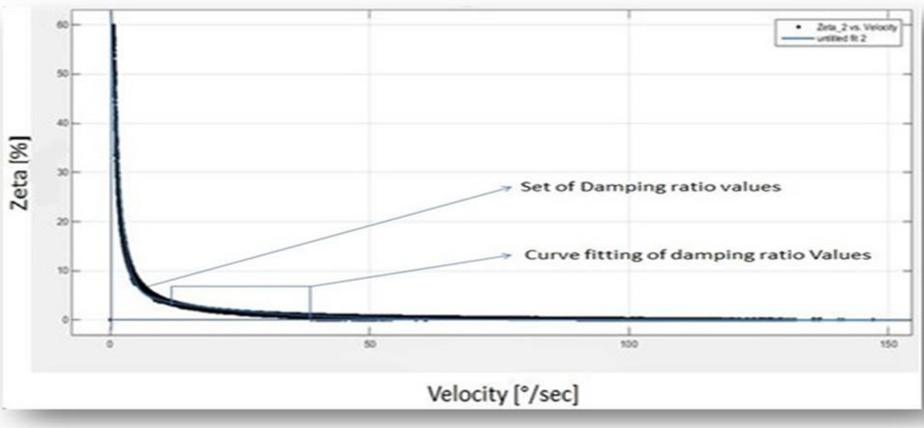


Fig. 6.15 The relation of the damping ratio to velocity

The correlation between the damping ratios and the velocities of a plate during motion in the A.H.P. illustrates the causes of kinetic energy loss.

6.4 Conclusions

A modal analysis of an I.T.C. plate towards a sprocket has been performed, aiming to identify its behavior in the dynamic transient state by defining the poles of a plate in an I.T.C. at the A.H.P.

The results also allow identifying the eigenfrequencies, which should be avoided in order not to reach resonance so as the system's dynamic behavior wouldn't reach a critical state. The research method presented in this chapter also shows the zones of maximum stresses and the displacement of nodes due to the contact between chain plates and sprocket in various frequencies.

Various damping ratio points were obtained at very low speeds. The damping ratios indicate the quality of a chain at low speeds. As velocity increases, the damping ratio decreases, as discussed in section 6.3. This could indicate the energy loss when the system decelerates due to inertia. This methodology would certainly help understanding the characteristics of an I.T.C. in general.

The damping ratios obtained by studying the motion induced during oscillation of the I.T.C. about the sprocket in the A.H.P. can then be used in finding the influence of an I.T. plate in motion at any drive system.

7. General Conclusions and Original Contributions

This doctoral thesis addresses one of the research problems related to the I.T.C. drive systems, aiming at developing a study for reducing the level of vibration caused by this type of chain drives. In this respect, the I.T.C. investigation aims to contribute to a better understanding of the behavior of the contacts between plates and sprocket, and how to predict the contact phenomena. This means that the I.T.C. would be less constrained as less external forces would be applied on it. The behavior that characterizes an I.T.C. depends on the contact forces caused by the oscillation of the I.T.C. plate towards the sprocket in the A.H.P.

The scientific objectives were initially formulated and were successfully conducted by following the proposed research steps. The research results have led to the conclusions presented below:

- The causes of ununiformity of the angular displacements and their derivations (velocities and accelerations) of the chain plates during oscillations along the A.H.P. were determined. The effect of double pendulum during oscillation is clearly observed. It is surely obvious that the system depends on the rigidity of the chain. When the rigidity is higher, the effect of the double pendulum tends to disappear. The vibration of the links that are located exactly after the last link subjected to contact or impact with the sprocket teeth dissipates slowly along the chain.
- A mathematical modeling of the A.H.P. has been developed to create a better background of the kinematics of any chain along the A.H.P., illustrating the behavior of the oscillation in a simplified equation of acceleration which helps foreseeing how a link could be positioned according to the time frame of its initial position. The mass of the link was also established. This could significantly reduce the time of simulations.
- The results obtained during the kinematic simulations of the I.T.C. in the A.H.P. show very close correlation to the experimental approach. The overall damping curves resulted from both experimental and numerical approaches being so similar prove the precision of the multibody dynamic analysis of a system in general. The effect of different types of chain lacings and different contour geometries of different types of I.T. chains were highlighted. The differences show that a particular chain type loses more energy during its oscillation back to the initial position. The loss of energy is due to the contacts of the I.T.C. plates with the sprocket. As the geometries are quite similar it is safe to state that the importance of the chain lacing is crucial in conserving kinematic energy during oscillations.
- It is also of great importance to determine correctly the Contact Damping Factors (C.D.F.) when solving the kinematics of an I.T. chain. It can be concluded that the I.T. chain has two different behaviors that occur along the timeline of the oscillation. These behaviors could be categorized as

large angular displacements and small angular displacements. It could also be concluded that the I.T. chain has different behaviors when it moves towards the A.H.P. or the sprocket as well as when the I.T. chain moves away from the A.H.P. or the sprocket. This is on the condition that the upper links of the chain are in permanent contacts with the sprocket. Both conclusions demonstrate that the I.T. chain has a non-linear behavior and should be divided into four different analyses that lead into finding different C.D.F. coefficients in solving the I.T. chain behavior.

- The close correlation of the experimental results and the numerical simulations validates the numerical model.
- The studies of contact forces between the plates of an I.T.C. and the sprocket in an A.H.P. were conducted taking into consideration the characteristics of the material used in constructing the I.T.C. plates and sprocket. The kinematic results helped as well in determining the contact forces. The conclusions are the following:
  - As the angular gap between the plates and the sprocket is larger, the vibration produced becomes smaller giving the impression that the length is in inverse proportion to the frequency.
  - As lower the number of the plates in contact, as higher the stress on those plates is, because of the smaller contact area. The single connecting plate suffers much more contact than the two plates in a link, as the surface area is much bigger
  - The contact of the plate teeth with the sprocket is a two stages event. There is a first contact on the plate with the sprocket which simply gives a reduction of the main contact force, acting like a first aid damping for the impact, even if the values are small. The first point of contact simply gives a sudden deceleration so that there is no full contact at a single step.
  - One can realize that if there is no first impact at the inner side of the plates during oscillation and contacts then the magnitude of the contacts is much higher and gives a higher distortion or vibration of the system. To obtain a first contact depends also on the angle of the plates coming in contact with the sprocket. The double contact of the I.T.C. gives an exponential deceleration as shown clearly at the displacements of the system and the deceleration curve.
  - The first contact assures a better positioning and assures a better engagement of the plates' teeth and sprocket teeth, in order to reduce the total deformation of the chain plates during contacts.
  - The system in itself suffers during low frequencies and as the frequency increases the system gives a more stable behavior.
- A dynamic analysis of the I.T.C. on the A.H.P. using the flexible M.B.D. approach was performed to determine the moment of the highest contact forces and also to determine the location of a plate during that contact force applied.
- The results also allow identifying the eigenfrequencies, which should be avoided in order not to reach resonance so as the system’s dynamic behavior wouldn’t reach a critical state. The research method presented also shows the zones of maximum stresses and the displacement of nodes due to the contact between chain plates and sprocket in various frequencies.
- Various damping ratio points were obtained at very low speeds. The damping ratios are a good indicator of the quality of a chain at low speeds. As velocity increases, the damping ratio

decreases, as discussed in section 6.3. This indicates the energy loss when the system decelerates due to inertia. This methodology would certainly help understanding the characteristics of an I.T.C. in general.

- The damping ratios obtained by studying the motion induced during oscillation of the I.T.C. about the sprocket in the A.H.P. can then be used in finding the influence of an I.T. plate in motion at any drive system.

The main advantages and future work of using the A.H.P.

- Most of the research is based on the dynamics of fast chain drive systems. The A.H.P. has low speeds where the kinematic changes of a plate in an I.T.C. which occur during motion have better traceability.
- The conduction of experiments doesn't take a long time and is a cheaper method in correlating experimental data with numerical analysis.
- From the data obtained in the experimental approach the damping characteristics of a chain plate can be obtained, thus understanding the dynamics of a chain during motion.
- Acoustic studies can be performed for better understanding the causes of energy losses during motion. These studies can point to how much kinetic energy is transformed to sound energy due to frictions between the plates and the joints or due to the contact between plates and the sprocket.
- Various contact theories can be applied in finding the correlation between them due to the kinematics occurred during oscillation between the I.T.C. and the sprocket in the A.H.P.

Personal original contributions in the present thesis are:

- The numerical modeling of an I.T.C. using the A.H.P. was performed. The kinematics of the I.T.C. in the A.H.P. was studied using the numerical approach in MSC Adams based on the rigid bodies M.B.D. theory. Creating a block diagram of the A.H.P. experiments were conducted in extracting and analyzing the kinematic results of the I.T.C. in the A.H.P. system. A correlation between the experimental results and the numerical approach was realized.
- The studies of contact forces between the plates of an I.T.C. and the sprocket in an A.H.P. were conducted taking into consideration the characteristics of the material used in constructing the I.T.C. plates and sprocket. Also, the contact forces have been determined.
- Dynamic analysis of the I.T.C. on the A.H.P. using the flexible M.B.D. approach was performed for determining the moment of the highest contact forces and also determining the location of a plate during that contact force applied.
- A modal analysis was performed on the I.T. plates in contact with the sprocket in the A.H.P. to find the correlation between different frequencies and the stresses on the plates due to contacts affecting the plates. Also, the damping characteristics of an I.T. plate were obtained in trying to understand the causes of energy losses during oscillations.

The results of the research were disseminated in four scientific papers, all presented in international conferences and published in conference proceeding/ journals and one article in the Scientific Bulletin of Transilvania University of Brasov, as presented below**:**

List of Publications

1. **Shalaby, K.**, Lache, S., Plămădeală, R., Dynamic Analysis of the Inverted Tooth Chain Plates Moving Towards Sprocket, in: Herisanu N., Marinca V. (eds.) Acoustics and Vibration of Mechanical Structures - AVMS-2017. Springer Proceedings in Physics, vol. 198. Springer, Cham, pp 341-348, DOI: https://doi.org/10.1007/978-3-319-69823-6_40, Print ISBN: 978-3-319-69822-9, Online ISBN: 978-3-319-69823-6 (indexed BD SCOPUS and Web of Science). Paper presented at the International Conference Acoustics and Vibration of Mechanical Structures (AVMS 2017), 25-26 May 2017, Timisoara, Romania.
2. **Shalaby, K.**, Corciova, F., Lache, S., Validation of Kinematic Simulation of Sprocket Contacts of Chain Links by Experiments, **in** "CONAT 2016 International Congress of Automotive and Transport Engineering", pp. 144-151, Ed. Springer 2016 (indexed BD Web of Science – conference proceedings). Paper presented at the 12th International Congress of Automotive and Transport Engineering (CONAT 2016), 26-29 October 2016, Brasov, Romania.
3. **Shalaby, K.**, Lache, S., Corciova, F., Contact Forces Analysis of an Analogous Huygens Pendulum Using Inverted Tooth Chain, International Journal of Materials, Mechanics and Manufacturing, Vol. 4, No. 3, August 2016, ISSN 1793-8198, DOI: 10.7763/ IJMMM.2016. V4. 259, pp. 195-199. Paper presented at the International Conference on Advances in Engineering Materials (ICAEM 2015), 27-30 June 2015, Constanta, Romania.
4. **Shalaby, K.**, Lache, S., Radu, F., The Analysis of an Analogous Huygens Pendulum Connected with I.T.C. Using Flexible Multibody Dynamic Simulations, the 6th International Conference Computational Mechanics and Virtual Engineering (COMEC 2015), 15-16 October 2015, Braşov, Romania, pp. 65-71, ISSN 2457-8541.
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Summary

Inverted Tooth Chain (I.T.C). drive systems are reliable and are characterized by their silence as they produce slightly less vibrations compared to other chain drive systems. They are used in high speed automobiles and conveyers for that matter. One of I.T.C. drive systems' main asset is that they have great potential over the other chain type drives, as they are much more silent. They lose less energy during motion compared to other types of chains, thus improving the overall quality of the mechanical motion transmitted in a chain drive system. At the same time I.T.C. drive systems have complex geometrical shapes which make the task of understanding and predicting the contact phenomena more difficult. The main objective of the doctoral thesis is to address one of the research problems related to the I.T.C. drive systems, aiming at developing a study for reducing the level of vibration caused by this type of chain drives. In this respect, the I.T. chains investigation proposes to contribute to a better understanding of the contacts between plates and sprocket, and their kinematics. The use of an unconventional Analogous Huygens Pendulum (A.H.P.) test rig created and developed at Schaeffler Romania SRL in collaboration with Transilvania University of Brasov gave the opportunity of studying the kinematics and the contact forces of an I.T.C. in a more profound manner, using the Multibody approach. Moreover, the Analogous Huygens Pendulum model made it possible to determine the damping characteristics of an I.T.C. by finding the damping ratio of a plate in the chain. The main advantage of studying an I.T.C. using the A.H.P. is the slow motion of the system giving greater chances of detecting anomalies occurred due to the oscillation of an I.T.C. towards the sprocket.

Sistemele de transmisie cu lanț dințat (I.T.C – Inverted Tooth Chain) sunt fiabile și se caracterizează prin silențiozitate, deoarece nivelul de vibrații produse este mai mic în comparație cu alte sisteme de transmisie cu lanț. Prin urmare, ele sunt utilizate în constucția automobilelor de mare viteză sau a benzior transportoare. Ascendentul principal al acestor sisteme față de celelalte sisteme de acționare îl reprezintă silențiozitatea. Ei pierd mai puțină energie în timpul mișcării în comparație cu alte tipuri de lanțuri, îmbunătățind astfel calitatea generală a mișcării mecanice transmise. Pe de altă parte, sistemele I.T.C. au forme geometrice complexe care fac dificilă înțelegerea și prezicerea fenomenelor de contact. Obiectivul principal al tezei de doctorat îl constituie abordarea uneia dintre problemele de cercetare legate de sistemele I.T.C. cu scopul de a elabora un studiu pentru reducerea nivelului de vibrații cauzate de acest tip de lanțuri. În acest sens, cercetările privind lanțurile dințate sunt menite să contribuie la o mai bună înțelegere atât a fenomenelor de contact dintre zale și pinion cât și a cinematicii ansamblului. Utilizarea unei platforme de testare neconventionale având la bază pendulul lui Huygens (A.H.P. - Analogous Pendulum Huygens), creată și dezvoltată la Schaeffler România SRL în colaborare cu Universitatea Transilvania din Brașov, a oferit posibilitatea studierii cinematicii și a forțelor de contact ale unui sistem de transmisie cu lanț dințat într-o manieră mai elaborată, folosind abordarea sistemelor multicorp. Mai mult, modelul pendulului analogic al lui Huygens a făcut posibilă determinarea caracteristicilor de amortizare a sistemului I.T.C. prin identificarea raportului de amortizare a unei zale în cadrul lanțului. Principalul avantaj al studierii transmisiilor cu lanț dințat folosind metoda A.H.P. îl reprezintă posibilitatea unei mișcări lente a sistemului, oferind astfel mai multe șanse de detectare a anomaliilor apărute datorită oscilației lanțului către pinion.

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- 03/2014-11/2017: Mechanical engineer, SCHAEFFLER Romania S.R.L., Brasov, Romania, Benchmark Department, measurements and analysis of different products
- 10/2011-03/2014: Mechanical engineer technician, SCHAEFFLER Romania S.R.L., Brasov, Romania, Measurement Laboratory Department, geometric and tribologic measurements of different products

EDUCATION:

- 2009-2011: Masters Degree in Mechatronics, Transilvania University of Brasov, Faculty of Mechanical Engineering.
- 2001-2008: Bachelor Degree in Mechatronics, H.T.I., Faculty of Mechanics, Cairo, Egypt
- 1998-2001: A.B. High School, Cairo, Egypt, Scientific section – Mathematics profile

PUBLICATIONS:

Published articles: 2

Conference papers: 3

LANGUAGES:

Foreign Languages: Romanian, English, French