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Study and Development of Cam Based Infinitely Variable Transmission

A Thesis Submitted to the Department of Mechanical Engineering / University of Kerbala in Partial Fulfillment of the Requirements for the Degree of Master of Science in Mechanical Engineering.

(Applied Mechanics)

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1442هـ 2021

بسم الله الرحمن الرحيم

هُوَ الَّذِي بَعَثَ فِي الْأُمِّيِّينَ رَسُولًا مِّنْهُمْ يَتْلُو عَلَيْهِمْ آيَاتِهِ وَيُزَكِّيهِمْ
وَيُعَلِّمُهُمُ الْكِتَابَ وَالْحِكْمَةَ وَإِن كَانُوا مِن قَبْلُ لَفِي ضَلَالٍ مُّبِينٍ

صدق الله العلي العظيم

سورة الجمعة - الآية 2

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We certify that this thesis entitled "***Study and development of cam based infinitely variable transmission***"

"Which is Prepared by "Muhaiman Faleh Hamad" under our supervision at the Mechanical Engineering Department, University of Karbala, as partial fulfilment of the requirements for the degree of master of science in mechanical engineering / applied mechanics.

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LINGUISTIC CERTIFICATE

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Dedication

I would like to dedicate my thesis to

My mother who supported me in all my life ….

Muhaiman faleh Hamad

2021

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First of all I thank God, who gave me the strength to finish this work.

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Muhaiman F. Hamad

2021

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Nomenclature and Symbols

Symbol	Description	Units
Acc	Acceleration Cam Follower	mm /S^2
Ci & Co	input & output Coefficients	-
Dim	Dimeter	-
Dis	Displacement Cam Follower	mm
DUD	Down Up Down	-
FGW	Follower Grooved Wheel	-
H	Start displacement Cam Follower =24.29	mm
Hz	Hertz	1/S
IVT	Infinitely Variable Transmission	
P	Constant value =62.84	-
Ploy	Polynomial	-
r. p. m	Revelation per minute	-
r_b	Cam Base circle (40)	mm
S.L	Slotted Link	-
SW & Th	Solidworks simulation and Theoretical	-
U1 & U2	Unit 1 & Unit2 of device	
Vel	Velocity Cam Follower	mm /S
Wi	Angular velocity input shaft (52.35)	Rad/S
Wo	Angular velocity output shaft	Rad/S
X_0	Displacement Grooved wheel Follower	mm
Xi	Displacement Cam Follower	mm
Y	length if the slotted link, equal 110 mm	mm
Yi	Length arm at side input for slotted link	mm
Yo	Length arm at side output for slotted link	mm
θ_{i}	Angular displacement input Shaft	rad
θ_o	Angular displacement output Shaft	rad

Abstract

Infinitely variable transmission (IVT) is the system that transmits rotational motion between two rotating elements with the ability to produce continuous (stepless) transmission ratios, including zero value. In this thesis, a cam based (IVT) system is studied, geometrically designed and assembled, where this IVT is recently developed. The system consists of two identical units; each unit contains cam, cam-follower, slotted link, follower grooved wheel, grooved wheel and ratchet. A theoretical kinematic analysis of the system is presented. In addition, a simulation model using Solidworks software is constructed to simulate the system in practice.

The cam profile selection and design is the main task in the designing process in this work. A uniform velocity and fifth-degree polynomial profiles were selected for the outward and return stroke, respectively.

The dynamic theoretical analysis in conjunction with the simulation model are applied to the whole system and for each part individually. A comparison is made between both types of analysis for each element in the system. The importance of having the simulation model is to ensure that the theoretical analysis is correct. On the other hand, the importance of the theoretical analysis is to achieve a conceptual appreciation of the dynamics of the system.

The analysis and results were carried out under two conditions; fixed transmission ratio and variable transmission ratio. For the first condition, typical values of the vertical distance Yi are considered (45,55 & 65) mm. There was a high agreement between the results obtained from both theoretical study and simulation. For the Second condition, a variable Yi was taken during the operation, and excellent agreements between results were achieved with a small amount of error of about 2%.

In addition to the kinematics study, which applied using both analyses, a force analysis is achieved for the system using the simulation approach. This is important to determine the contact forces.

1 Chapter one: Introduction

1.1 General

Power and torque generated from power sources (wind turbines & automobile engines, or any other power source) cannot be transferred directly for applications. Therefore, transmission is integrated. For example, in a wind turbine, variable input (wind speed) must be converted to a constant rotational speed in the exit axle in order to generate the desired features of the electric power. In vehicles, transmission (Manual or Automatic) must provide the following features: movement flexibility (front, rear & neutral), efficiency, low fuel consumption, and comfortable for the driver and passengers.

The researchers found that the above traditional transmission (Manual and Automatic) are complex, heavy-weight, rather expensive. For these reasons, it is preferable to adopt a more dependable one. Accordingly, continuously variable transmission (CVT) is suggested to be applied instead of the traditional transmissions. This will be explained in the next Sections.

1.2 Continuously Variable Transmission (CVT)

A continuously variable transmission (CVT) is a device that permits the user to change the transmission ratio between rotating input and output gradually from one positive value to another**[1]**. Dissimilar to the traditional transmissions, the selection of gears is not limited to a finite number of ratios. An infinitely variable transmission (IVT) is kind of (CVT) that can provide zero transmission ratio in the range of variation [2].

Regarding the above discussion, car manufacturers and car companies aim to obtain a power transmission with specifications (uncomplicated, inexpensive, and its efficiency is higher than the traditional power transmission). For this reason, the CVT was invented, which depends on variable diameter pulley and the power is transported by special belts. This type of CVT is somewhat similar to the processes of power transmission (Milton Reeves) produced at the beginning of the twentieth century with the advantage of having multiple gear ratios. The V-belt and other types of CVT are presented in the next Sections.

The use of CVT in equipment and medium-sized vehicles are to reduce fuel consumption by 3% of vehicles (traditional automatic transmissions & manual) [2].

1.3 Continuously Variable Transmission types

Most CVT transmissions are generally classified into the following two types:

1.3.1 Traction Continuously Variable Transmission

1.3.1.1 Variable-Diameter Pulley (VDP)

This type of CVT is the most common in use. The speed ratio is variated by changing the diameter of the pulleys, as shown in figure (1.1). It is most commonly due to the small number of its components, the possibility of obtaining many rotational speed ratios without the need for complex parts. The following issues should be taken into account: Maintaining the distance between the axles of the pulleys and Preserving the tension value in the belt conveyor capacity. The main parts of this CVT are:

1- Power transmission belt (metal figure 1.2 or rubber figure (1.3))
2- Variable diameter entry shaft pulley.
3- Variable diameter exit shaft pulley.
4- Simple control system.

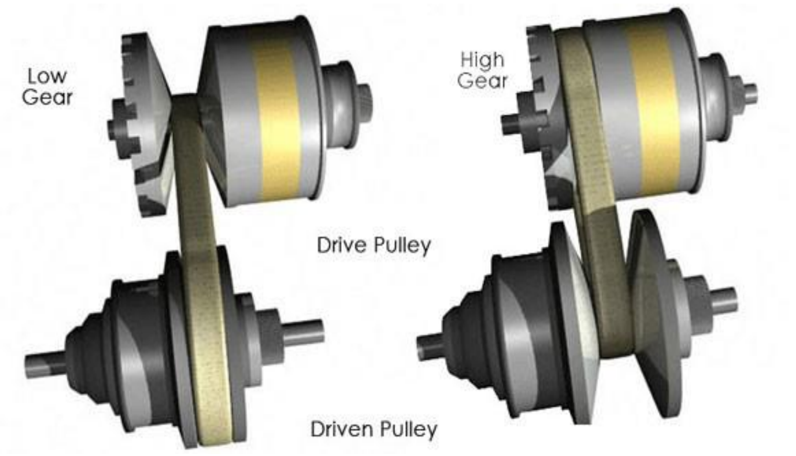

Figure 1-1:Variable-Diameter pulley CV [3]

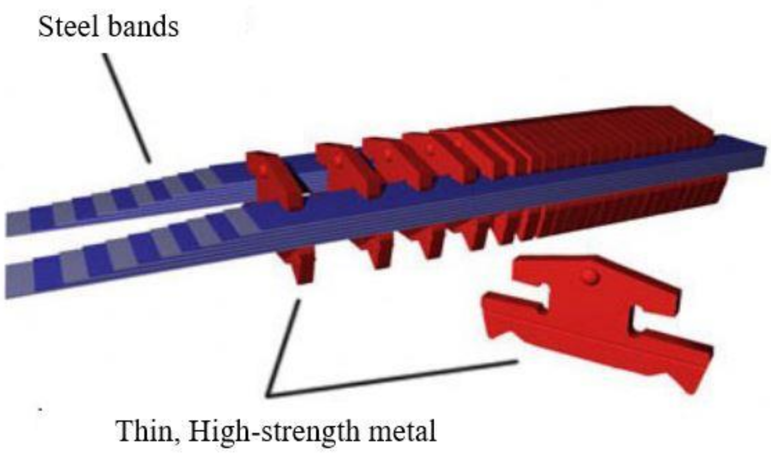

Figure 1-2: Metal belt [3]

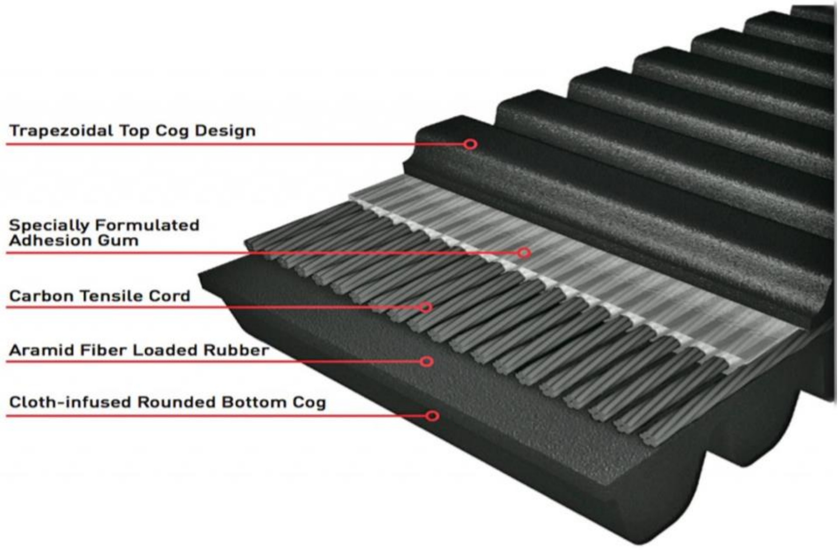

Figure 1-3: Rubber belt [4]

1.3.1.2 Toroidal CVTs

This type is used with the medium torque equipment and mechanisms. The V-belt type also involves uncomplicated components, as shown in figure (1.4).

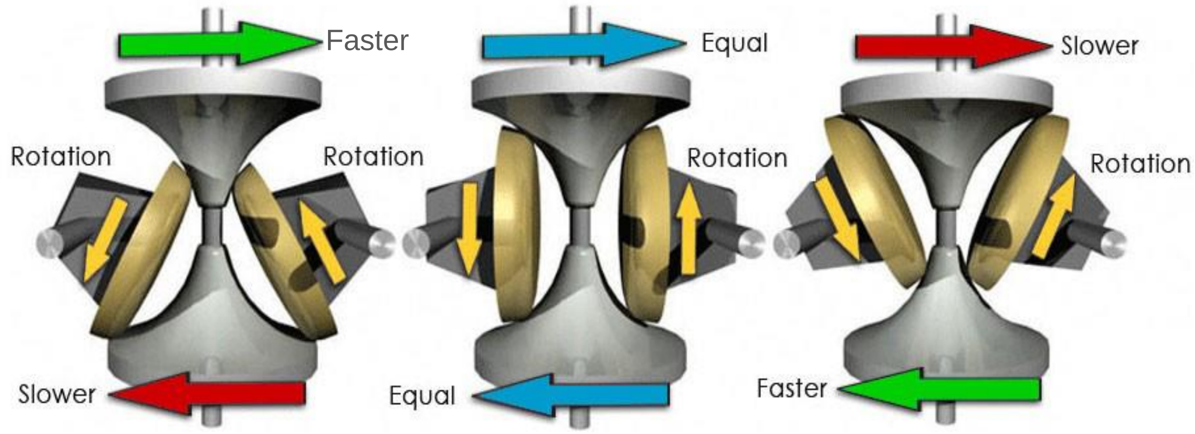

Figure 1-4: Basic component of toroidal CVT [3]

The operation of this mechanism can be described as follows: one of the discs is installed on the power source while the Second disc is considered an output. For obtaining variable speed ratios, the roller position is controlled, which is the transmission medium. The power transfer mechanism does not depend on the friction between the surfaces; it relies on the shear property of the oils that are used for lubricating the transmission. This type is classified into two basic types**.**

1. Full toroidal.
2. Half toroidal.

The Second type is distinguished from the first as it has higher efficiency, the possibility of conveying a higher capacity, and takes up less space figure (1.5) **[5].**

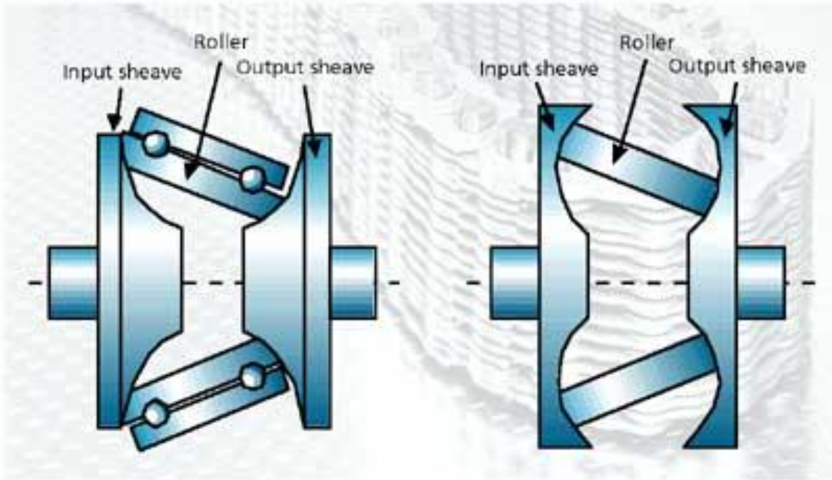

Figure 1-5 : Full and half toroidal CVT [6]

1.3.2 Non-traction Continuously Variable Transmission

1.3.2.1 Link Infinitely Variable Transmission (IVT).

The basic configuration of this mechanism is shown in figure (1.6). The main feature of the non-traction types is that they do not depend on friction in power transmission, which means higher power can be transferred. This mechanism is also having the features of simplicity in installation and high reliability. The operation of the mechanism is as follows; The rotating speed is transformed into an oscillating velocity in (follower) by (free shaft) which is self-regulating by spring, as shown in figure (1.7). The reciprocating movement is transmitted to the (planetary gear set) by (one-way clutch) to convert the intermittent reciprocating speed into the continuous rotation. This device is best suited for use in a wind turbine for the following reasons (simplicity of installation, lightweight, ease of manufacture and maintenance). Mode A represents loaded phase and mode B represents unloaded phase.

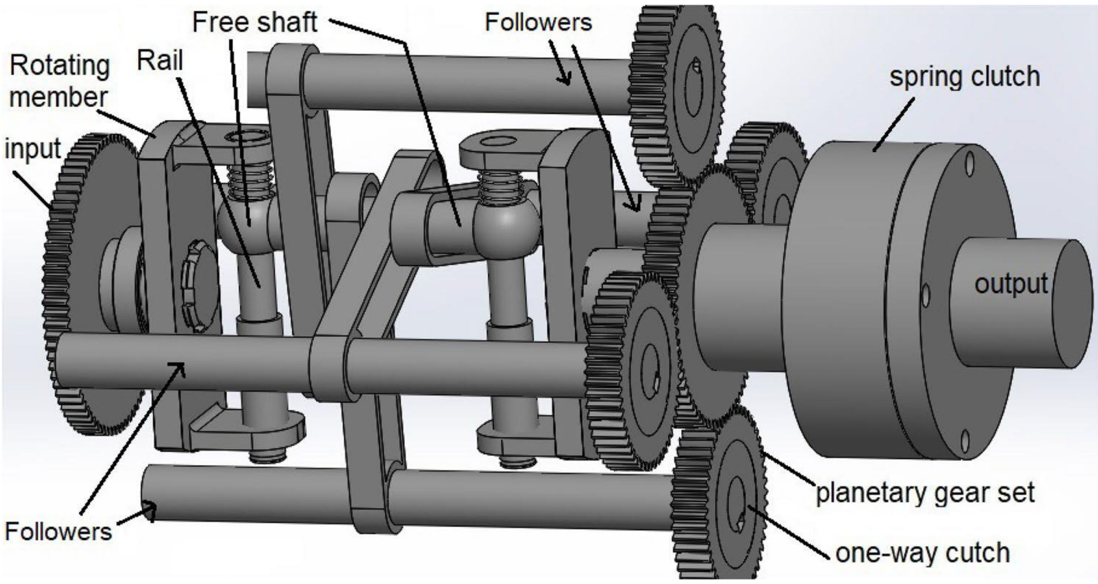

Figure 1-6: the main parts of the link Infinitely Variable Transmission [7]

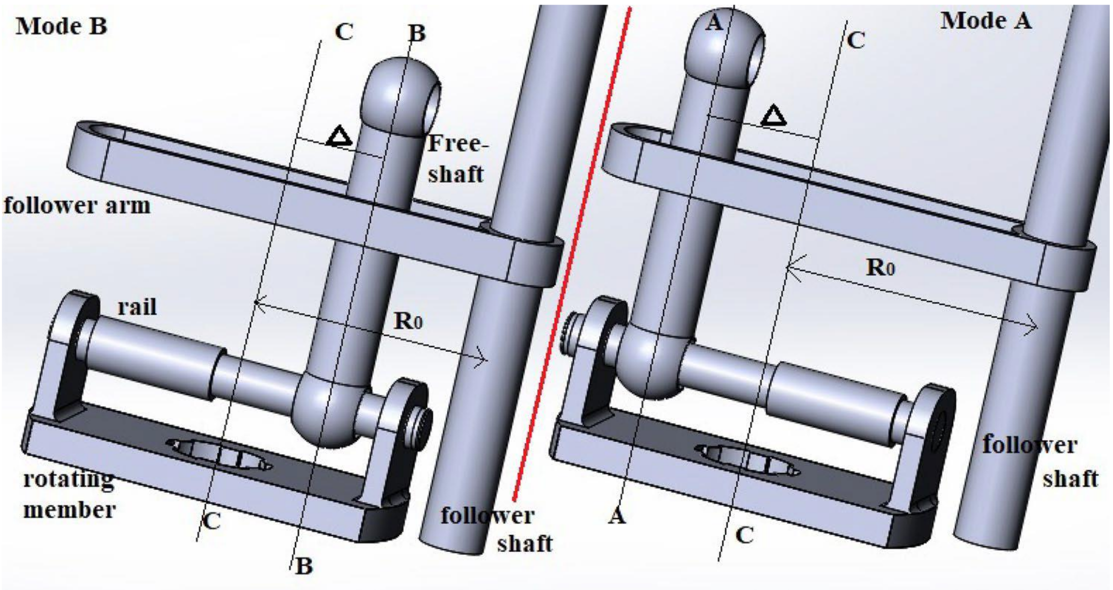

Figure 1-7: The mechanism of moving follower of the Link Infinitely Variable Transmission[7].

1.3.2.2 Cam-based Infinitely Variable Transmission

This type of Cam-based Infinitely Variable is somewhat similar in parts to the previous IVT type, but the difference is that the followers are driven by a 3d cam, as shown in figure (1.8). For obtaining variable speed ratios, the follower position is changed by a special external mechanism, unlike the previous device, which automatically organizes itself.

The main components of this device are 3D-cam, followers, one-way clutch, planet gear, and sun gear. With regards to the mechanism movement: the rotational motion is organized to the cam, which in turn drives the follower a rotational frequency movement that depends on the shape of the cam and its dimensions. The reciprocating rotational movement is transmitted to (one-way clutch) used to obtain a positive rotational speed. The rotational motion is transferred to planet gear and to (sun gear), which is considered the output shaft. The follower is moved to a position on an asymmetrical (cam) surface around the axis of rotation to obtain zero speed. The main feature in this type is the possibility of obtaining infinite speed ratios. The power transmission is limited due to the presence of stress in the contact surfaces between the cam and the followers) **[2]**

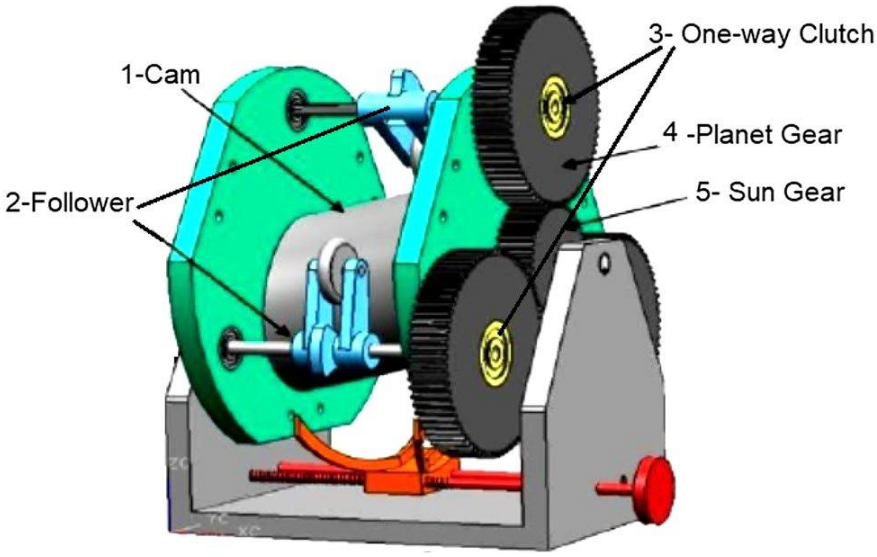

Figure 1-8: The main parts Cam-based Infinitely Variable Transmission [2].

1.3.2.3 Infinitely Variable Drive (IVD).

The basic components of this type of CVT, which is called (IVD) are shown in Figure (1-9). These components are input shaft, tilting plate, bevel gear set, one-way clutch, push link, output shaft and frame one - way clutch. The operation can be explained according to the following steps:

1- Energy is supplied from the source to the input shaft, which rotates at a constant rotational speed.

2- The constant rotational velocity is transformed into frequency by using a tilting plate. The angle of inclination of the tilting plate can be changed as needed, i.e., in the case of obtaining a zero velocity in the output shaft, the angle is zero. For increasing the velocity, the angle value must be increased.

3. The reciprocating movement is transmitted from the tilting plate to the push link by a spherical joint. This joint is used for freedom of movement in three axes (x, y, z). This joint is on both ends of the push link.

4. The reciprocating movement is transmitted from push link to one-way clutch: the function of this part is to transmit power or angular velocity in the positive part only. For the device to be more stable in angular velocity, use six one-way clutches.

The speed or frequency displacement is transferred to the bevel gear set, which is a bevel gear 3, driven gear, where it is connected to the output shaft to be used as required. The planetary gear set is used to obtain the reverse direction to the input speed **[1].**

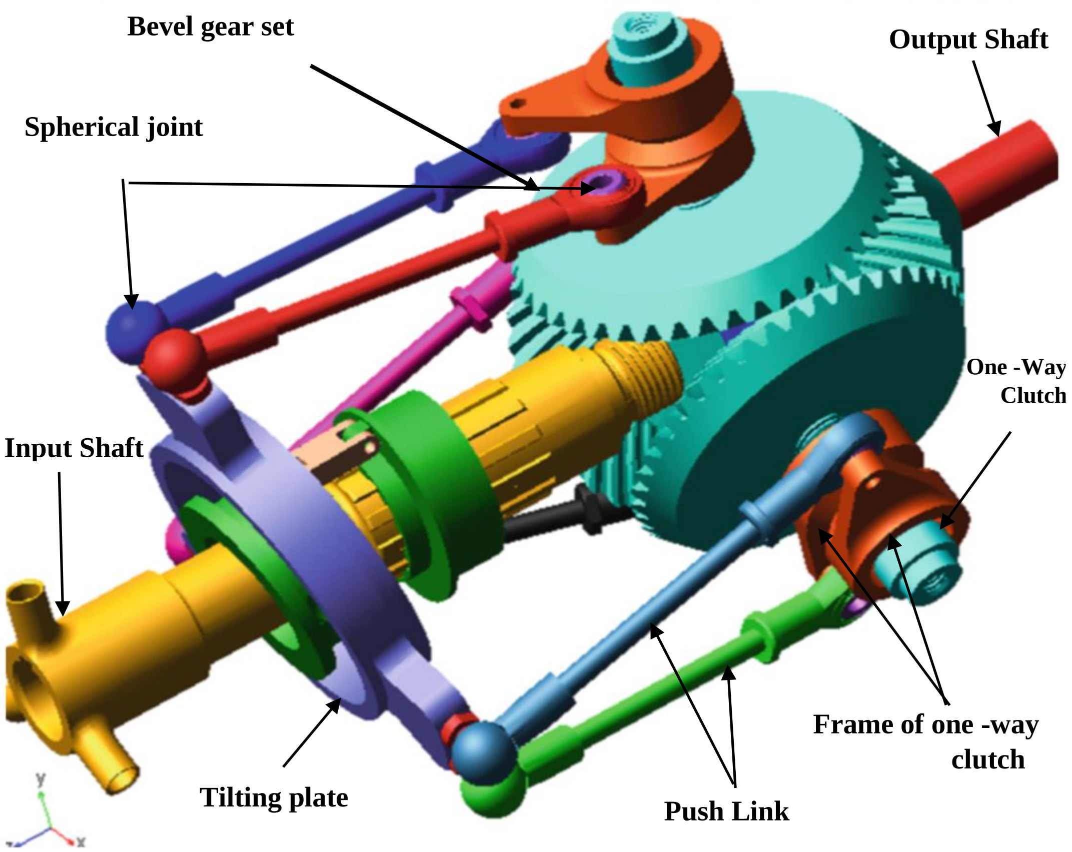

Figure 1-9: The Infinitely Variable Drive IVD [1]

1.3.3 Hydrostatic Continuously Variable Transmission.

Transmission of power by HYDROSTATIC CVTS is an ideal choice regarding efficiency and low mechanical losses such as friction. A typical hydrostatic CVT is shown in figure (1-10). The direction of movement can also be reversed easily. The small space it occupies provides the appropriate weight and place to other devices (suspension systems), especially in vehicles and construction equipment. The most important disadvantage of this type is the high cost. As for the principle of work: a high-pressure liquid is supplied from a special pump to a hydrostatic cvts device, which can change the speed ratios between the inlets and outlet axis by the same mechanism mentioned above. **[8]**

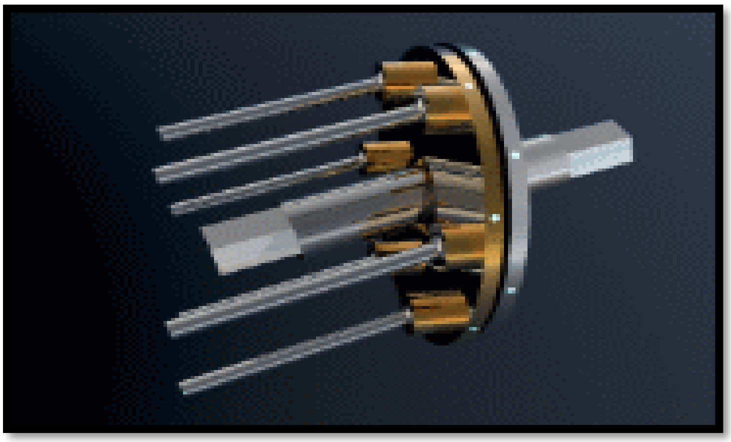

Figure 1-10: Typical Hydrostatic CVTS [9].

1.4 Advantage and disadvantage of Continuously Variable Transmission

1.4.1 Advantage:

1.4.1.1 Built-in Design

The CVT is less complex in terms of design and number of parts because most of the design does not contain (planetary gears) and this feature leads to lightweight, less maintenance and manufacturing costs compared to the traditional gearbox [9].

1.4.1.2 Efficiency and Emissions.

The CVT transmission is considered economical in fuel economy, and these results in environmentally-friendly equipment, were when used as an alternative to conventional or automatic power transmissions, it saves fuel consumed by about (11%-12%).

1.4.1.3 Losses.

Lower interior power losses (e.g. friction and wear losses)

1.4.2 Disadvantage:

1.4.2.1 Oil used in lubrication

It needs special oils that are used in lubrication. These specifications in oils are not available most of the time.

1.4.2.2 Speed of rotation

When using (CVT) in vehicles, we need to increase the speed or accelerate in some cases. In this case, the motor rotates at a high speed at a certain level, and this mode reduces the operating life of the motor parts.

1.4.2.3 Parts replacement and repair

Most types (CVT) when one of its parts is damaged. It is replaced completely (i.e., only one part can be replaced to connect the parts in an integrated way)

1.5 Applications of CVT in Power Plants

1. **Wind Turbines**

   In wind turbine applications, the energy is achieved from the wind via the turbine rotor, where the amount of this energy depends on the wind speed. There are many electrical base technologies to control the frequency of electricity for variable wind speed.

   In addition to the electrical technologies in generators, mechanical technologies use traditional gearbox or Continuously Variable Transmission (CVT) to be variable speed according to the wind speed on one side and fixed speed on the other side of the generator. The advantages of this technology can provide a constant speed for the generator (fixed frequency) utilizing a special control system without the need for expensive electronic devices **[10].**

2. Gas Turbine

The gas turbine is characterized by operating at constant rotational speeds to meet the requirements of the generator or electrical grid. But in some special circumstances, reducing the speed of the turbine requires the role of influencing the output of the generator and the requirements of the electrical network. Frequency and voltages are directly related to the number of revolutions of the turbine shaft. One of these requirements is the decrease in the pressure of the gas delivered to operate the turbine during maintenance work or keeping the turbine running. When electric power is needed at peak times, but with less capacity.

One of the goals of this study is to investigate the possibility of replacing the traditional reduction gearbox in Al-Hilla power plant with the IVT under consideration in this study. In other words, replacing the gearbox with the IVT provides the possibility of operating the gas turbine at different gas pressures without the need for complex systems, simple monitoring on operating the unit. Figure (1-10) shows a schematic diagram of al-Hillah power plants, including the traditional gearbox.

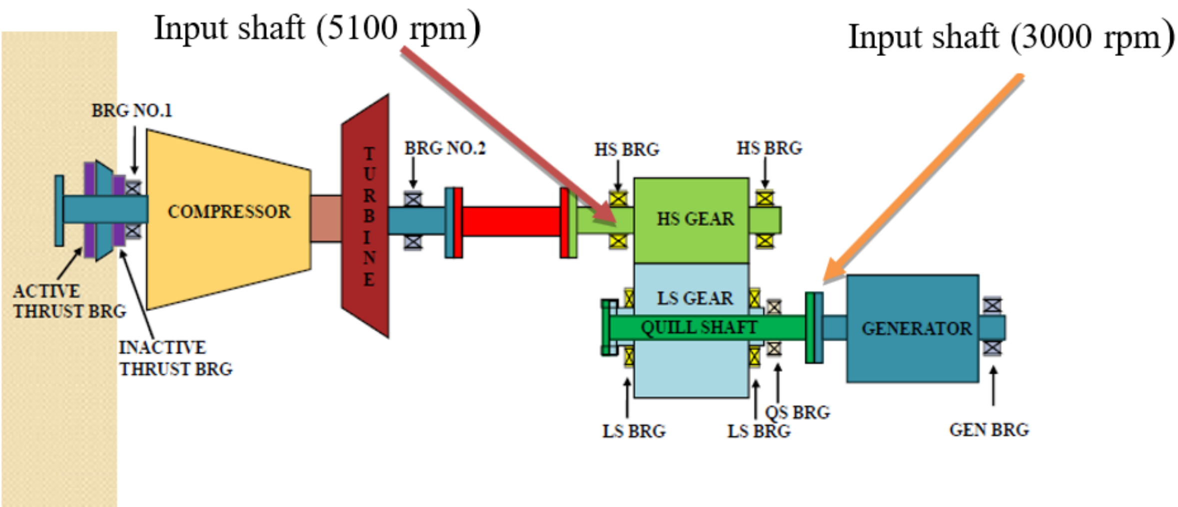

Figure 1-11: A Schematic diagram of Al-Hillah-1 power plants.

1.6 Thesis layout

Five detailed chapters with supplementary appendix make up this thesis as follows.

Chapter One: Introduction.

Chapter Two: Deals with the literature review on the main topics of the thesis.

Chapter Three: Theoretical analysis

Chapter Four: Simulation of the IVT by SOLIDWORKS

Chapter Five: lists the results obtained from theoretical and simulation along with their discussion.

Chapter six: provides the conclusions summarized from the work and gives some recommendations for future work.

2 Chapter Two: Literature Review

2.1 Introduction

The CVT is a relatively modern device. However, previous ideas were touched upon, and the first designer to discover it was the Italian designer Leonardo DaVinci. Car companies continued, for example, with GE starting research or application in 1924, followed by the English company Austin in 1934.

In general, the first to use the CVT is the car companies. The reason behind this is to satisfy the passenger comfort and the driver's lack of effort. As mentioned earlier, some types of CVT do not require a clutch in addition to the simplicity of installation and maintenance. Concerning wind or water turbines, the input energy is not uniform. The CVT can provide the advantage of constant speed in the outlet axis. This is the most important feature for obtaining a constant frequency. In this chapter, a historical review of CVT is presented based on their most common types **[2].**

2.2 Traction Continuously Variable Transmission (CVT)

2.2.1 Variable-Diameter Pulley (VDP)

Mangialardi and Mantriotn (1992) [10] discussed the possibility of using a (CVT) device between a wind turbine and a generator employing mathematical analysis. As for the indications for using (CVT), it is known that generating electrical energy requires a mechanical power source with constant rotation and torque, and the wind turbine is subjected to permanently changing working conditions. Previously, this problem was solved in two ways; the first is to change the step angle of the wind turbine blades, and the Second is to use electronic devices, but the previous two solutions are considered expensive devices. After analysis, it was found that adding the CVT to a wind turbine, raises its efficiency and gives stability in frequency and voltage. In general, it increases the efficiency of the turbine.

Mangialardi and Mantriota (1994)[11] used a mathematical model to analyze the results obtained from wind turbines that use a CVT-type transmission and compare the results with a wind turbine that uses a conventional power transmission (gearbox). The data was taken from working conditions known in advance. The researcher concluded: (CVT) the proposed use in this research is simple in components and can be trusted in power transmission and thus is economically inexpensive. As for efficiency, it was found that the use of wind turbines (CVTs), especially at high wind speeds, gives an increase in the production of electrical power.

Mangialardi and Mantriota (1994) [12] suggested the use of (CVT) in wind turbines on which water pumps are installed. This is because the efficiency of the system drops dramatically at high wind speeds(3m/s). It is known that these systems operate in places far from energy sources. For this reason, it was suggested that the use of CVT be (Low cost (automatic adjustment according to the comfort rate), high utility, high reliability. The researcher extracted from the numerical simulations results in an excellent fluid flow rate, especially in a place with a high wind speed. When CVT is used, it reduces the starting torque required for turbine rotation (which is because CVT provides multiple transmission ratios). A flywheel is required to regulate the pump operation. The CVT proposed to be used provides a combination of simplicity of installation, which gives low cost, and high reliability, leading to not requiring high skill in operating and maintenance procedures.

Chen et al. (1998) [13] conducted a practical study to find out the losses in torque, speed, and power in the type of V-BELT transmission (CVT) operating under variable working conditions. The research's focused on the following conditions under which the test is performed: speed, change ratios, rotational speed, and the tension in the belt conveyor for power, which affected external load, and pulley dimensions. Through practical experience, it was found that the power transmission of the type (V-Belt CVT) differs regarding the usual behavior found in the transmission belts. These differences are represented in the presence of (variable speed ratio, presence of axial force, and variable tension according to external load), where all these previous things have greatly affected the efficiency of the CVT V-Belt transmission. The finding through the test showed that the highest efficiency at the ratio of speed between the reels of the driving and driven was equal to one.

Carbone et al. (2001) [14] Theoretically suggested the use of v-belt metal type. This is due to its high ability to transmit moments. However, the calculations he made are dynamic calculations under the influence of (thrust pivotal, torque, the tension in the case of tight, slack). concluded: the amount of friction is very high, especially when the change in velocity ratios between the inlets and outlet axes, and for this reason, the angle of pulleys must be calculated. The theoretical calculations must be compared with a practical model.

Gibbs (2009) [15] suggested using an electromechanical or hydraulic control system to be practically applied instead of mechanical control systems in lightweight vehicles to obtain: control accuracy thus leading to an increase in efficiency and a weight reduction. In any case, he concluded that using the electromechanical or hydraulic control system increases the efficiency of (CVT) by 17% compared to the power used to operate the controller 0.027 HP.

Laxman And Scholar (2019) [16] compared two models: (parts of a conventional rotating shift gearbox), (CVT). Where it is suggested to use the two models in vehicles. In addition to the parts drawing and static analysis using the (Creo 2.0) program, dynamic stresses were simulated by (ANSYS 14.5) program. Likewise, more than one material (Carbon Fiber, E Glass Epoxy, and Steel) was chosen in the simulations. The researcher concluded the following:

1. Deformations, stresses, and strains are less when using CVT compared to a conventional gearbox.

2. The stresses are low in using steel, but it is heavier than the other materials suggested for use in the analysis (Carbon Fiber & E Glass Epoxy). The shear strain when using a carbon fiber material is less compared with (Steel) in the case of random vibration test, and it is less value when using (E Glass Epoxy).

5. It is better to use CVT because it is less complicated regarding mechanical parts.

6. It is preferable to use composite materials (Carbon Fiber & E Glass Epoxy) because it can control its mechanical properties and is lighter.

Culebro and Villar (2019) [17] introduced a numerical model to calculate and analyze torque and angular velocity calculations using MATLAB software, as well as created a test model to compare the results. Regarding installation, the device consists of two parts. The first part is somewhat similar to the type (CVT) — (V-Belt), but in this device, there is the possibility of getting movement in the opposite direction and a neutral movement without the need for a clutch, and this saves the cost and the volume that the device occupies. It is a control system that gives infinite velocity change rates, and for this reason, this power vector is classified as (IVT). The above features allow the possibility of using (IVT) in the following applications: wind turbines and vehicles. The operation results have been shown, which have been compared with the theoretical results. Matching very close in addition to the work of the device in the test model very quietly and its high efficiency.

2.2.2 Toroidal CVTs

Tanak et al. (1995) [18] discussed the following features on a double-cavity CVT power transmission device, a radius of 40 mm, the distance between the two toroid centers. The tested data was 7000 r.p.m, torque of 340 N.m, oil pressure between surfaces 3.2 kHz. It was concluded that the fatigue reached the highest value when the oil pressure in the contact area reaches 2.7 Hz. The highest stress limit is at the oil pressure value of 3 kHz. Ranging from (0.44-2.28), the slow rotation of the parts of the device does not generate high heat, and as a result, it leads to transmission speed and higher efficiency.

Belfiore et al. (2003) [19] created a new model of the annular CVT, in which the mechanical parts of the device was briefly described. They also performed a mechanical analysis of the structure of the device, which is unique. The mechanical analysis was performed at two angles to prevent slipping to some extent in the contact areas. The results of the theoretical analysis proved that the device could provide infinitely fast ratios. As for the control mechanism, it depends on the value of the two angles (theta and alpha fig) that achieve the movement of the movement without sliding. The highest transferable torque was found through theoretical analysis of static forces, which is 30 N.m.

Tatsuhiko (2006) [20] suggested using (CVT) between uniaxial gas turbine and (AC generator, pump fluid). Inferred when the loads or turbine speed change, the CVT has a high response in the constant speed rotation of the generator so that the outputs are stable. Its number of revolutions may be reduced (to reduce emissions, more operating life of turbine parts) as well as give (CVT) rotational speed that makes the generator have outputs according to the design (Hz, volt). As pumps, the turbine at start-up is unable to supply fluids according to the design conditions sometimes required (pressure, flow rate), and for this reason, it is preferred to use (CVT) instead of hydraulic clutches.

Miyata and Liu (2007) [21] Studied the stress analysis of half-toroidal CVT under loading or power transmission. The aim is to find (plastic deformation, tolerances). The analysis and results were obtained by using

MATLAB (Simulink). This is because all previous studies were not considered when analyzing the conditions to which the half-toroidal CVT is exposed, which is (permittivity and plastic deformation), in addition to, finding the behavior of (CVT). At the same time, it is under load conditions to determine its efficiency. concluded that the ready-made with relatively high efficiency under the load of up to 97%, obtaining change rates in speed smoothly and quietly.

Nashed (2009) [22] suggested using CVT in a gas turbine which uses the principle of propeller propulsion, as this type is used in aircraft. As for the location of (CVT), it shall be before the turbine compressor and on the same axis of the turbine to equip the turbine fan with variable speed according to need with the stability of the rotation the speed of the turbine to keep the capacity stable. He concludes that the power transmission (CVT) suggested being used is of the (toroidal) type, and it is possible to use the rest of the types considering the work conditions represented (stress, transfer rates, temperature, and vibration).

Richard and Nozari (2012) [23] suggested utilizing (CVT) in commercial aircraft turbines to equip the auxiliary devices with power, which are (hydraulic and pneumatic devices in addition to electric generators. As the speed of the turbine is high. It can reach 10,000 r.p.m. In some cases, the rotational speed may be variable according to the need and keep up with this change.

Yang et al. (2013) [24] experimentally and theoretically studied the use of a Continuously Variable Planetary System (CVP) instead of the electronic devices used in wind turbines, which are used as systems that control the output of the generator (frequency & voltage axis). They had experimented in practice on a test device of the type (Viryd 8kW horizontal wind turbine). It is a simulator of a horizontal axis wind turbine. As for the results, they were given to the MATLAB / Simulink® environment program. They concluded that the use of Continuously Variable Planetary (CVP) is useful as a control in the output of the generator on the one hand, resulting in an increase in the rate of energy produced annually, on the other hand. In addition to reducing the total cost of the wind turbine, it did not require electronic devices. Continuously Variable Planetary (CVP) used gives great benefit in improving the performance and operation of the wind turbine by maintaining the rotation speed of the generator shaft to be constant under changing conditions.

Wang et al. (2014) [25] Investigated the possibility of using a new power transmission whose principle is very close to the type (Toroidal-CVT), but the mechanism for transmitting power or torque to the outlet axis has done by (Sphere install on shaft have universal joint). It was concluded that the device could provide change ratios, simple structure, simple operation idea, ineffective in transmitting torque, compared to (Toroidal-CVT), which is considered one of the (Toroidal) types.

Tyreas and Nikolakopoulos (2015) [5] Investigate the possibility of using (CVT Half -Toroidal) in a (2 kW) wind turbine by making a model using the CATIA program and the (ADAMS MSC 2013) software. It was found that the angular velocity and the value of the stress resulted from the moment and forces of the contact surfaces in (CVT). They concluded that the presence of high friction generates high heat between the contact surfaces in the power transfer of (CVT). To overcome this heat and to prevent this power loss, it is recommended to paint the surfaces with an alloy (Chromium Nitrite Coating or Nickel Coating, which tailor's friction properties). During the experiment, it was found that (CVT Half — Toroidal) works with variable air speeds and gives high efficiency and stability in the required output (angular velocity and angular torque).

Da Costa et al. (2015). [26] experimentally studied a model of (CVT) installed on a sports car prototype intended for testing. The results were also compared with a gearbox installed on the same test model (230 kg vehicle weight, Honda CBR 600rr engine type). This model is a test model and has not been considered the moment of inertia of parts (CVT). It was that conclude the efficiency of (CVT) is low, and it is considered a problem in its use.

Verbelen et al. (2018) [6] compared theoretically between full and a half (CVT) under dynamic loading. The efficiency of CVT, which operates by running range, is determined in these circumstances. The results found that the (CVT half) during loading is more efficient than (CVT full). In other words, the device is very efficient even when the ratios change suddenly, and the (CVT half) can transmit high torque at high and low speeds.

Milazzo et al. (2020) [27] studied the possibility of using a self-controlled CVT with a working principle similar to (CVT-full-toroidal) but with less complexity. The number of parts was reduced to be available for use in devices with lower cost and personal use or somewhat low power, for example, wind turbines, motorcycles, and bike. The researcher made a simulation model built in the CATIA program and according to external conditions, wind speed and temperatures in the conditions of Italy. It was found that the model is ideal and does not require additional forces to overcome the inertia torque forces.

Marathe and Wakchaure (2020) [28] performed an analysis of Single Ball Traction Drive for Continuously Variable Transmission. They presented a practical model that could prepare different rates of change. The device's working mechanism was innovative (CVT-toroidal), but it was less complicated and low in cost. Through practical experience and the data obtained, they concluded the following:

1. The possibility of changing the speed ratios between the axis's of entry and exit was (step-less).

2. Compact size and low cost.

3. The device has high efficiency of up to 75%.

4. For quiet operation (no shock when changing proportions), jerk (single ball) should be used.

5. It needs a special mechanism when used in mechanical applications to change transfer ratios.

6. The inverse relationship between efficiency and ability.

2.2.3 Non-traction Continuously Variable Transmission (CVT)

Benitez et al. (2004). [29] studied experimentally, numerically and analytically a device of the type (IVT), which belongs to the category of ratcheting, and contains a gear group. The movement mechanism of the device parts was described, and then numerical simulations were made for dynamic analysis by the program (ADAMS). The researcher concluded that the resultant speed was oscillatory and was caused by the use of (ratcheting). To overcome this problem, the researcher suggested using one of the following two methods: the first is by using a flywheel that links at the exit axis. The Second method makes the power transmission frequency the same as the ICE frequency. With these two methods, IVT can be used in vehicles.

Lahr and Hong (2006) [30] introduced a new power and transmission (IVT) ratchet. In addition, a test model was made of aluminum and carbon fiber. The principle of this type (IVT) is based on: planetary gear set, follower cam, carrier, overrunning clutches. The practical test was at the following conversion ratios (1: 1, 1: 1.3, and1: 1.6.). The researcher concluded that (IVT) ratchet is the best regarding regular rotational speed in the exit axis among all types. For this reason, it is considered the best among all types of (IVT) in practical application, the possibility of reversing the direction of movement and converting speed ratios.

Lahr (2009) [31] presented a practical study of developing and analyzing an IVT Cam-based model. This type can provide speed (forward, reverse, neutral), and these options are not available in the rest of the CVT models. In general, its work is somewhat similar to the work of planetary gears, which provide speed reversal. The equations of planetary gear have been adopted in finding the kinematic equations. Cam provides infinite velocity change and neutral velocity. Its shape is designed for constant velocity (zero acceleration) and trapezoidal curve to obtain an acceptable jerk. The stress between (follower) and the cam surface was reduced. This application is suitable for use in vehicles and generators. The inlet hub can be used as an outlet axis and vice versa. It was recommended to replace the planetary gears with a planetary belt gear regulator to increase the transmitted torque and reduce stress. The efficiency of the device is up to 93%.

Abood (2010) [32] presented a theoretical proposal with a simulation model for a cam based (IVT). The system consisted of two units to obtain a smooth change in the output. The theoretical part was described through a mathematical model that describes the ratios of (displacement, speed and acceleration. It was concluded that the system produces constant outputs for constant inputs. The presence of fluctuation in the case of changing the speed ratios between the inputs and the outputs can be reduced by using more than two units.

Morales and Benitez (2014) [33] conducted analytical and numerical studies to find out the most usable designs. They concluded that overrunning clutches are ideal when used with power carriers in vehicles, but the resonance rate of its parts must be calculated to avoid failure. This is because internal combusection engines are a source of vibration. Rectifiers can be used, whose principle of operation is passing the positive part and cutting the negative part to the output axis. Therefore, the speed was converted into high torque, which is preferred with dynamic CVTs in heavy equipment.

Hernandez et al. (2015) [34] proposed an experimental model of the type (CVT) consisted of a combination of gears and conventional clutches. They concluded that this transmission could be installed on wind turbines with a capacity of 8 Mw. The ability of this type to obtain a constant frequency of approximately 60 Hz. The possibility of transferring the torque from the turbine to the generator with soft torque is due to the use of clutches. This CVT has high operational life because it eliminates fatigue in addition to the possibility of operating the turbines at high wind speeds. The cost of turbine blades was decreased due to the reduction in the dimensions and in the cost invested in energy production.

Nummelin (2017) [3] studied experimentally the influence of reducing complexity on the improvement of the performance of IVT used in vehicles. Two models were proposed. The researcher concluded that the proposed mechanism in the two manufactured models was practically distinguished by the following factors:

1. Net torque without outlet waviness.
2. Reducing the complexity of the previously used device by suggesting simple mechanisms.
3. The possibility of obtaining speed (neutral, opposite, forward) and infinity values.

Bhusal (2017) [35] studied practically by a test model in addition to simulation-based (ADAMS MSC) program the performance of IVD device. To simplify the analysis of the per simulation, a set of planetary gears that are intrinsic, as well as with respect to one-way clutches, have been studied. The results obtained from the simulation and the modified model were compared in practice during the test. It was concluded that this device provides angular velocity (forward, neutral, and opposite). This feature is provided by the planetary gear set. Moreover, the experimental findings agreed well with the simulated results.

Al-Hamood et al. (2018) [36] presented a theoretical study of a novel cam based IVT. The speed ratio is changed by changing the position of the slotted rod (the vertical distance between a grooved wheel and a slotted rod). The larger ratio produces greater speed and acceleration in the exit axis. The researcher concluded that the system produces a fluctuation in the angular velocity and acceleration at the output shaft. This oscillation occurs when the speed of the slotted rod is not equal to zero, but the oscillation value is considered small. Therefore, a flywheel is recommended to be mounted on the outlet axis. This proposed IVT has the feature of to be transmitting high torques and obtaining the opposite direction of rotation of the outlet axis. This feature is possible for use in a one-way clutch.

Bhusal et al. (2018)[1] studied completely new (CVT). It consists of (pushing rod, tilting plate, ratchet planetary gear set. To find out the behavior of the device, the mechanical simulations of the aforementioned parts were done first by the MSC ADAMS program). Secondly, the dynamics of the device was fully simulated, as this device is considered one of the fully mechanical types in transmitting torque and velocity (i.e. not dependent on friction). The purpose of this proposed device was to obtain zero velocity with a velocity in the inlet axis. The device is highly efficient because it works according to a purely mechanical principle in transmitting torque and power, its ability to produce zero output speed.

Bhusal et al. (2018) [37] proposed an experimental and mathematical model for a CVT-type transmission. The researcher concluded that the theoretical calculations are very close to the practical calculations. This leads to an increase in confidence and durability in the proposed conveyor by reducing the number of clutches, which are the weakest part of the device. The conveyor has the ability to transmit a high amount of torque and power. The power transmission shaft of the type (torsional shaft) is very efficient. This feature reduces the forces and loads on the clutches (leading to transmitting relatively high loads.

Olyaei et al. (2019) [7] presented a completely new model of power and motion vector (IVT). He concluded that this device provides a uniform speed with little stress compared to the transmission that adopts cam. For this reason, the advantage of this device is the transmission of high torque and speed, making it available for use in turbines. In addition, this device is cheap regarding manufacturing, maintenance, and installation.

Al-Hamood et al. (2020) [38] studied theoretically the angular output (displacement, velocity, acceleration) of the angular mechanism focusing on the one-way clutch. The clutches work by cutting the positive or negative part, or vice versa, to obtain the regular output. It was concluded that when increasing the transmission ratio in (IVT), sliding in the clutches is increased. In addition, the wheels require additional torque to accomplish the return stroke, and rotational springs may be used for this purpose.

Shinde and Khan (2015) [39] Improved the design of the IVT devices using a practical test device. The modification was made considering the allowable cost and the available raw materials. A 3D model was made using CATIA V5 software, and stress analysis and kinematics calculations were made by ANSYS. They concluded that the possibility of obtaining infinite speed without needing a clutch (low cost, size and total weight), the mechanical output smooth of the test device was excellent and very close to the theoretical results.

2.3 Concluding Remarks:

The amount of torque transmitted and the rate of change of velocity between the axes of rotation and the exit in the transmission mechanisms were in the research and development stage for previous years. As evidenced by the aforementioned business in the current years, this importance has come from today's applications that require variable speeds and torque as needed. In their work, the researchers used experimental or theoretical patterns or both in research (CVT-IVT). The CVT is one of the most interesting types of power transmission devices, especially variable capacitance. To find out the possibility of CVT, most researchers compared the experimental results with the theory. However, despite many previous studies of CVT systems, they need more development and study to raise efficiency. These studies will provide further clarification of the CVT- IVT principle of action. Therefore, the current work calculates the kinematic analysis at different positions to obtain different speed ratios in the shaft exit with constant angular velocity in the entry shaft. For this purpose, a three-dimensional model was created for the above purpose and to know the stability of the device upon loading.

2.4 Research objectives

An objective of this study was to investigate the following points
1- Finding special diagrams (displacement, velocity and acceleration) for each of the cam follower and the groove follower and a comparison (displacement, velocity and acceleration) of rotation between the input and output shaft, was performed considering the following point:

A. At the following dimensions (at the center = 195, u = 205 and D = 185) mm from the main base of the device

B. The purpose of this investigation is to explore the relationship between constant speed and composite profile (constant speed and 1-5 polynomial).

2 Studying displacement, velocity and acceleration diagrams in case the slotted rod changes in the following order (DUD, DU and UD)

3- Studying the mechanism of slotted rod positioning by means of a linear motor.

4- Finding the reaction force between the parts that have direct contact.

3 Chapter Three: Theoretical Analysis

3.1 Introduction

This chapter contains the theoretical parts of the IVT system under consideration. It illustrates the derivations & assumptions that have been used to analyze and study the (IVT) system. The methods used to solve the formula, which is the governing equation for motion, are illustrated. The output axle rotational speed and acceleration were then evaluated by using kinematic analysis equations. This chapter also discusses the construction of different kinds of cam profiles. **[36]**

3.2 Introductions to CVTs and IVTs

CVTs are the system that allows for stepless change in the speed ratio between two rotating elements. On the other hand, discrete shifting is used in conventional power transmission, such as gears boxes, to achieve variable (discrete) transmission ratios. A typical type of CVTs is the transmission's IVT technology (CVT system) which is highly variable can utilize a zero-transmission ratio. Engineers use CVT in wind and water turbines as an alternative to the conventional transmission techniques to achieve the fixed rotational speed of the electric generators against the variable speed of the power source (wind or water). As a consequence, the necessary electric power voltage and frequency are met.

In the automotive industry, CVT gearboxes are started to be used instead of conventional gearboxes. The CVT system ensures that the engine and the wheels still in sync even during changing the transmission ratio. This resulting in quiet engine loading and a relaxed ride. Furthermore, engine power loss is reduced during gear shifting, resulting in lower fuel consumption and emissions.

CVT systems are being considered for use in a variety of driving devices to support power output and speed at the input and output components more generally. A considerable amount previous literature has been applied. There are various types of continuously variable transmission systems that have been developed in the Section. However, as will be discussed later, Because of practical considerations, their application is currently restricted.

Another common type of CVT system is the ratcheting type. A ratcheting IVT system, with two epicyclical gear systems and one-way clutches, was introduced by Benities et al. [14]. Dennis et al. [9] showed a (one-way clutch) IVT device that included a 3-D cam, multiple followers cam, clutches one-way and the epicyclic train gear. A cam-based one-way clutch IVT mechanism was also described. A 3-D cam, follower cam, and wheel grooved are employed in this system, which is made up of a number of similar units [27].

A recent clutches one-way kind (cam-based IVT mechanism) is studied in detail in this thesis. As described earlier in chapter two, this IVT system is presented by Figure 3.1, showing in detail the IVT system under consideration after a detailed design by the author of this thesis.

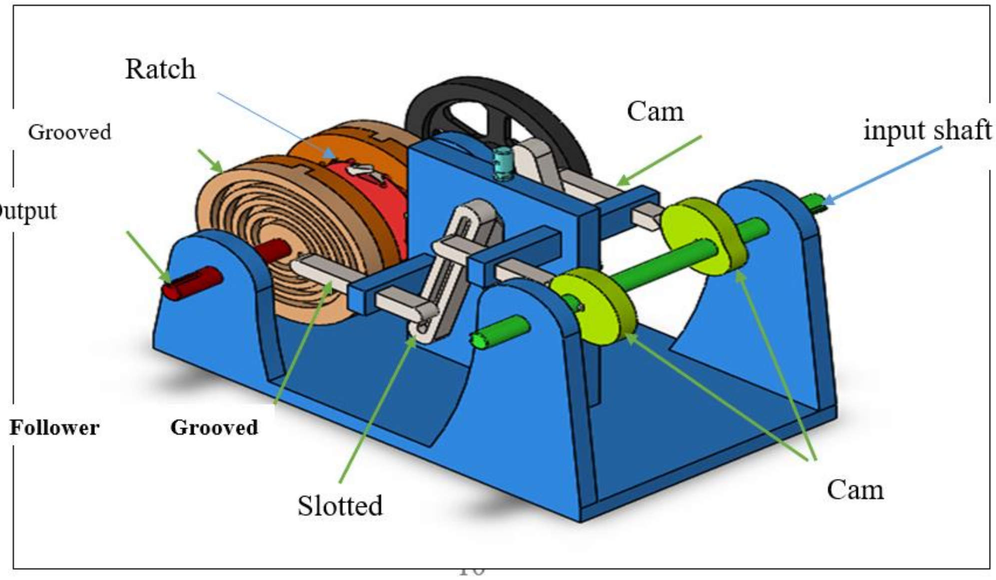

Figure 3-1: Main parts of the system.

3.3 Configuration and elements of the system

As shown in Fig. (3.1), the proposed IVT system is made up of two equivalent units, each of which contains the following components:

(a)A cam attached to the input shaft.

(b) A cam follower.

(c) A slotted linkage hinged with regard to the spindle set to the adjustable rotation speed (power screw);

(d) A follower of a grooved wheel.

(e) A grooving wheel attached to the outer rod by the one-way clutch.

The above parts can be seen in detail in Figures (3.2) and (3.3).

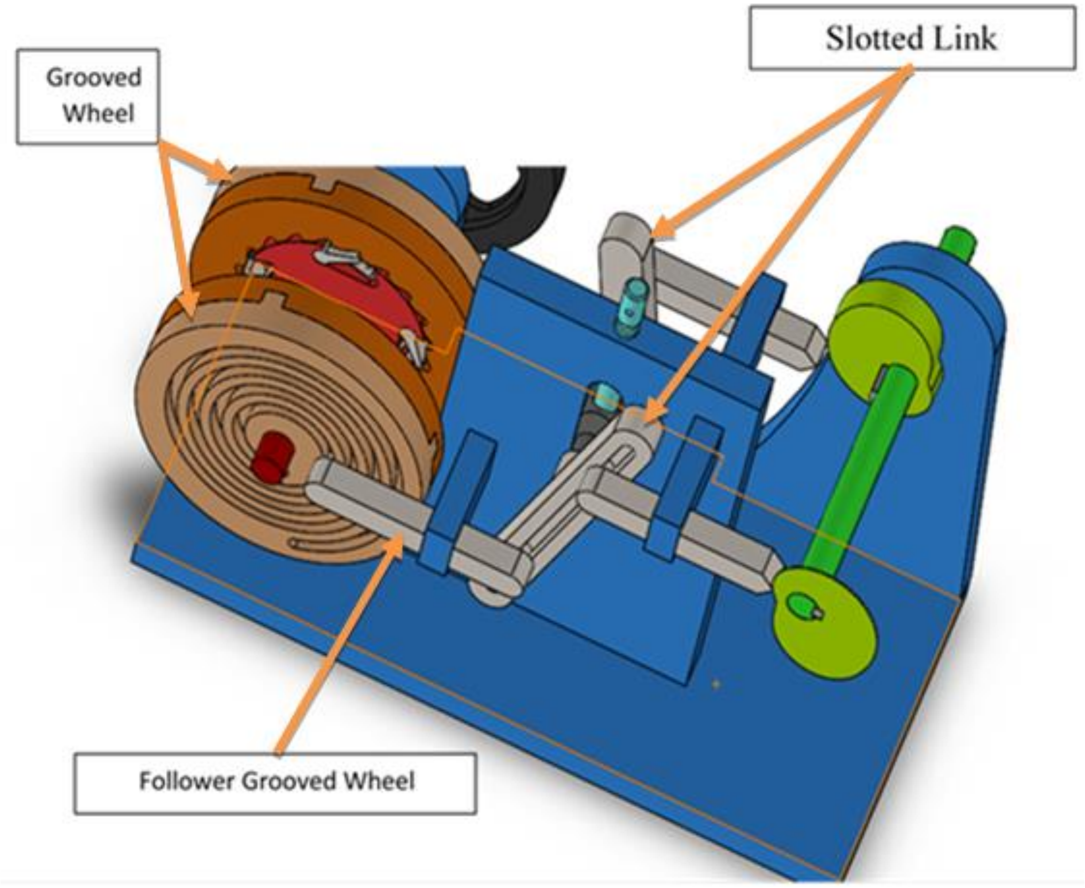

Figure 3-2: Connection between (Slotted Link, Grooved Wheel) and Follower Grooved Wheel.

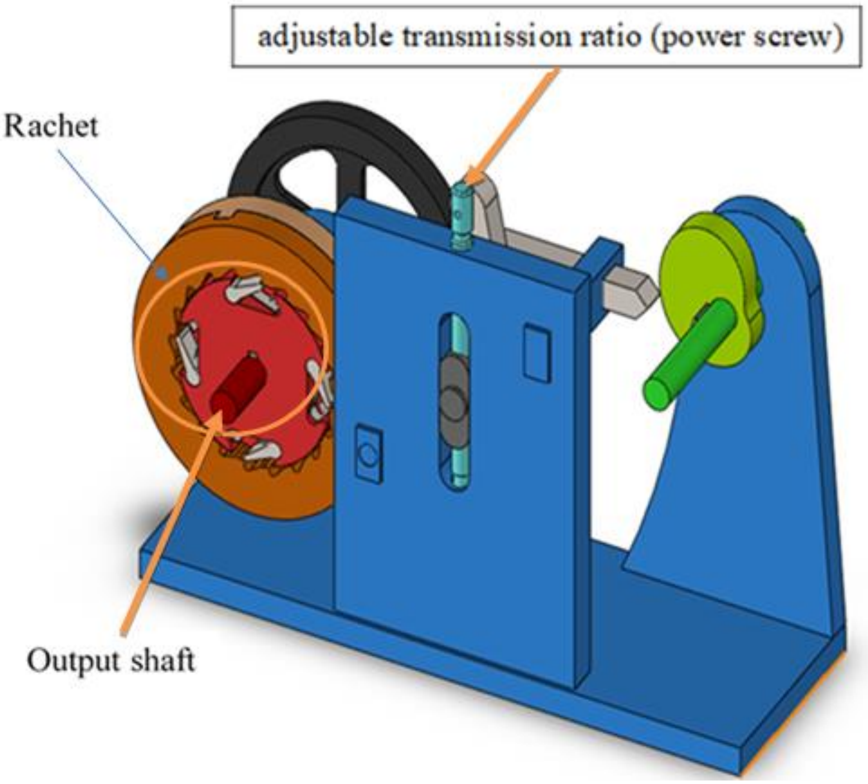

Figure 3-3: Ratchet and adjustable transmission ratio (power screw)

3.4 Operation of the system:

The steps of operating the examined IVT system are as follows:

(a) The cam follower mechanism converts the circular motion of the input shaft into reciprocating motion.

(b) At the slotted link, the repeating follower motion is transformed to an oscillating rotational motion. The magnitude of this oscillating angular motion is determined by the vertical direction of the slotting link pivot, which is controlled by a power screw or servo motor.

(c) The oscillating angular movement of the slotted rod is rectified by the grooved wheel in a reciprocal movement.

(d) The follower grooved wheel motion is transformed from an oscillation rotational motion into one-directional movement at the outer shaft's grooved wheel by (ratchet). The angular amplitude of this motion may vary from that of the cam, depending on the vertical position of the slotted link pivot. Accordingly, the

grooved wheel rotates at an angle greater than one revolution for high transmission ratios. This may require using a groove that covers an angle greater than one revolution. This is why a groove is used instead of a cam on a wheel.
The movement of the follower cam during the forward stroke is depicted in the above description. The mechanism returns to its original role on the return stroke. This is accomplished by installing a torsional spring at the slotted link to hold the cam and the follower in continuous connection through the process. The grooved wheel and the output shaft are locked through the forward stroke due to the influence of the (clutch-one way), but they slide down relative to each other through the backward stroke. It is ensured that the (clutch-one way) holds the output shaft rotating in a single direction given oscillatory angular motion. As a result, if one unit is used in the system, there will be a discontinuity in power transfer during the return stroke. The current IVT mechanism requirement has at minimum dual units geared along with the axles' outputs and inputs in order to provide continuous transmitted torque (see Fig.3.1).

A stepless adjustment (shifting) of the center of the revolution of the slotted links is used to change the ratio of the transmission. As shown in Fig.3.2, a mechanical ram can be used for this aim. The magnitude of the slotted link's curved motion will change as a result of this change. In other words, the continuous (stepless) adjustable location along the vertical axis adjusts the ratio of input to output angular displacements، As a result of this feature, the suggested system is a continuously variable transmission system. The machine's kinematics analysis, detailed in the following Section, shows that the system can also be called an IVT.

3.5 Constant-constant Cam profile

The kinematic of the proposed system for the constant-constant Cam profile is derived in this Section for a one unit. Figure (3.4) is a schematic diagram that shows the kinematics of the system.

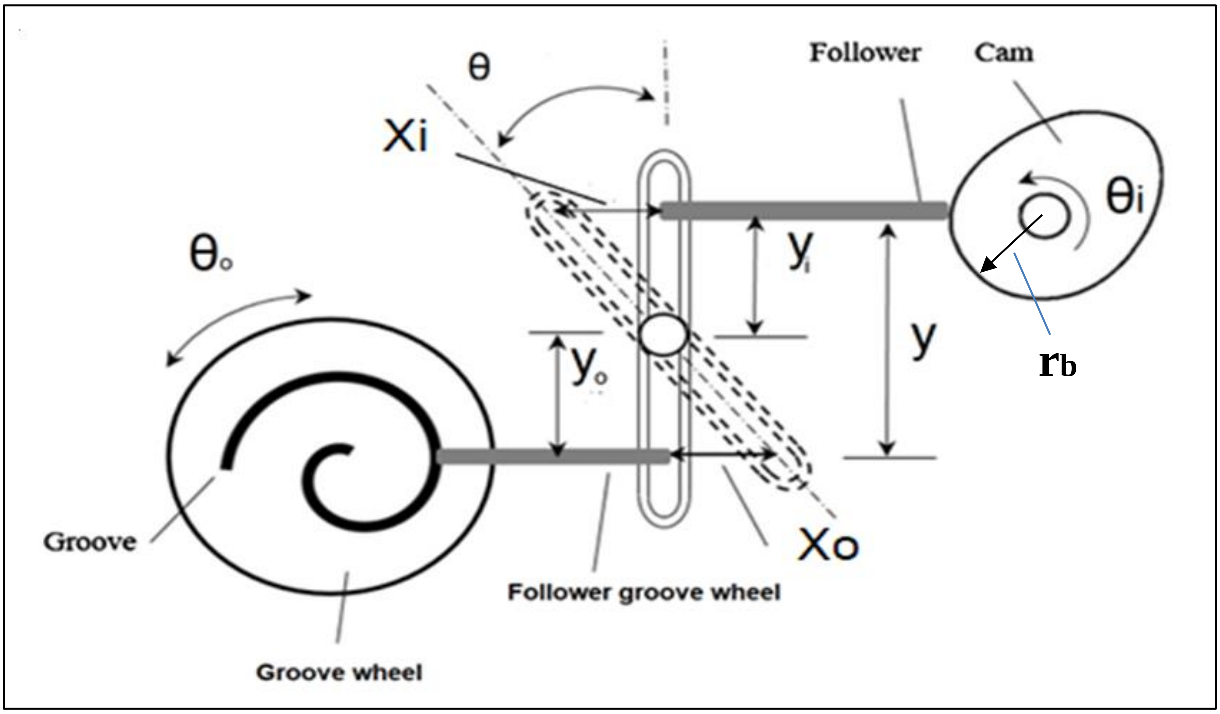

Figure 3-4: Operating of a single group in the IVT system.

The displacement equation of the follower during Cam rotation can be written as.

$$x_i = r_b + c_i \theta_i \tag{3.1}$$

Where

r_b: base circle radius of the cam.

c_i : constant of proportional.

θ_i: angular displacement of the cam.

Now differentiate equation (3.1) with respect to time gives.

$$\frac{dx_i}{dt} = c_i \frac{d\theta_i}{dt}$$

Or

$$v_i = c_i \omega_i \tag{3.2}$$

Where

v_i : velocity of the cam-follower.

ω_i : angular velocity of the cam.

To find the acceleration, again differentiate equation (3.2) with respect to time gives.

$$a_i = c_i \frac{d\omega_i}{dt} = 0 \tag{3.3}$$

Note that in this work, constant input angular velocity is considered. For the return stroke, the constant velocity profile analysis is similar to the backward stroke. Figure (3.5) shows the displacement, velocity and acceleration.

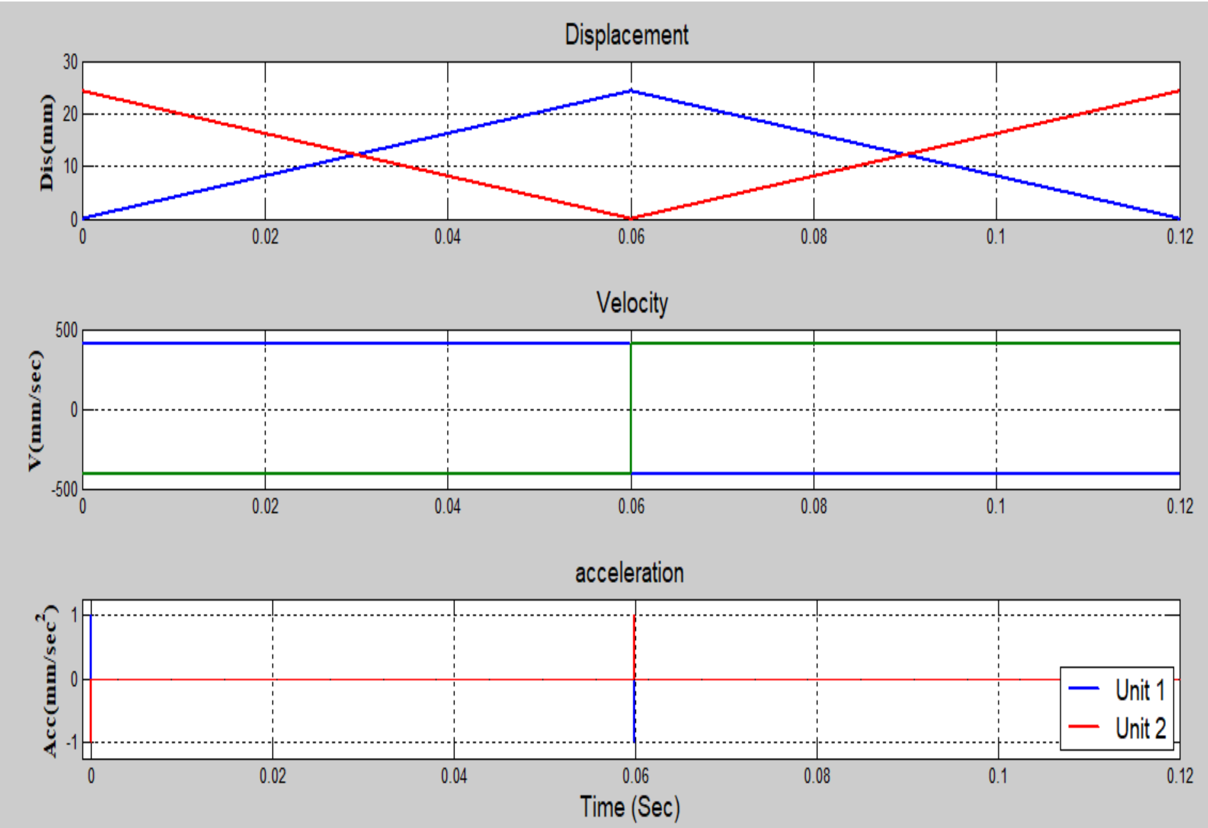

Figure 3-5: Constant Velocity cam profile (displacement, velocity and acceleration).

It can be noted that the speed of the follower has a minus value in the range ($\pi \leq \theta 1 \leq 2\pi$), representing the back-Follower's Stroke. This issue can easily be solved by adding a Second unit to the system.

While the first unit is in the returning stroke, the Second unit works at an active stroke.

It can be seen that during the sudden change in the velocity when θi= (0 and π) when, the acceleration goes toward (+∞,-∞). An immeasurable amount of strength is needed at the point of connection between the cam and the follower, which can lead to parts failure. This issue can be sorted by designing a polynomial (1-5) cam profile for the return stroke. When the constant velocity and poly (1-5) curve profiles are combined, the follower's displacement, velocity, and acceleration have a regular and soft behavior. The equation describing the profiles at all intervals is presented in detail in the next Sections.

3.6 Constant -polynomial cam profile.

In order to overcome the problems associated with the constant-constant profile, a fifth-degree polynomial cam profile is suggested for backward stroke

The general equation of the fifth-degree polynomial equation is.

$$x_i = a_o + a_1\theta_i + a_2\theta_i^2 + a_3\theta_i^3 + a_4\theta_i^4 + a_5\theta_i^5 \qquad (3.4)$$

Where

a_o-a_5 are constants.

x_i : follower displacement

θ_i : angular displacement of the cam.

The velocity and acceleration relations are presented by differentiating equation (3.4) with respect to time, this gives:

$$v_i = \frac{dx_i}{dt} = a_1 + 2a_2\theta_{i1} + 3a_3\theta_{i1}^2 + 4a_4\theta_{i1}^3 + 5a_5\theta_{i1}^4 \qquad (3.5)$$

$$a_i = \frac{dv_i}{dt} = 2a_2 + 6a_3\theta_{i1} + 12a_4\theta_{i1}^2 + 20a_5\theta_{i1}^3 \qquad (3.6)$$

These constants are determined by applying the boundary conditions, which are explained in the next Section.

3.6.1 Boundary Conditions

In order to determine the coefficients of a polynomial function, the following boundary conditions are considered. For a continuous cam profile, the connection between the two profiles should be smooth. So that from the constant velocity profile, the first boundary condition is:

At $\theta = \pi$; x_i=$c_i\pi$; $v = c_i\omega_i$; a=0

Applying these values into equations (3.4, 3.5, and 3.6)

$$c_i\pi = a_o + a_1\pi + a_2\pi^2 + a_3\pi^3 + a_4\pi^4 + a_5\pi^5 \quad (3.7)$$

$$c_i\omega_i = a_1 + 2a_2\pi + 3a_3\pi^2 + 4a_4\pi^3 + 5a_5\pi^4 \quad (3.8)$$

$$0 = 2a_2 + 6a_3\pi + 12a_4\pi^2 + 20a_5\pi^3 \quad (3.9)$$

The Second boundary condition is:

At $\theta = 2\pi$; x_i=0 ; $v = c_i\omega_i$; a=0

Applying these values into equations (3.4, 3.5, and 3.6) gives:

$$0 = a_o + 2a_1\pi + 4a_2\pi^2 + 8a_3\pi^3 + 16a_4\pi^4 + 32a_5\pi^5 \quad (3.10)$$

$$c_i\omega_i = a_1 + 4a_2\pi + 12a_3\pi^2 + 32a_4\pi^3 + 80a_5\pi^4 \quad (3.11)$$

$$= 2a_2 + 12a_3\pi + 48a_4\pi^2 + 160a_5\pi^3 \quad (3.12)$$

By using MATLAB &Applying the above equation into matrix form gives:

$$\begin{bmatrix} 1 & \pi & \pi^2 & \pi^3 & \pi^4 & \pi^5 \\ 0 & 1 & 2\pi & 3\pi^2 & 4\pi^3 & 5\pi^4 \\ 0 & 0 & 2 & 6\pi & 12\pi^2 & 20\pi^3 \\ 1 & \pi & \pi^2 & \pi^3 & \pi^4 & \pi^5 \\ 0 & 1 & 2\pi & 3\pi^2 & 4\pi^3 & 5\pi^4 \\ 0 & 0 & 2 & 6\pi & 12\pi^2 & 20\pi^3 \end{bmatrix} \begin{bmatrix} a_o \\ a_1 \\ a_2 \\ a_3 \\ a_4 \\ a_5 \end{bmatrix} = \begin{bmatrix} c_i\pi \\ c_i\omega_i \\ 0 \\ 0 \\ c_i\omega_i \\ 0 \end{bmatrix}$$

Solution of the above system of equation simultaneously gives the values of the coefficients a_o-a_5. Figure (3,6) shows the displacement, velocity and acceleration for the constant-polynomial 1-5 cam profile.

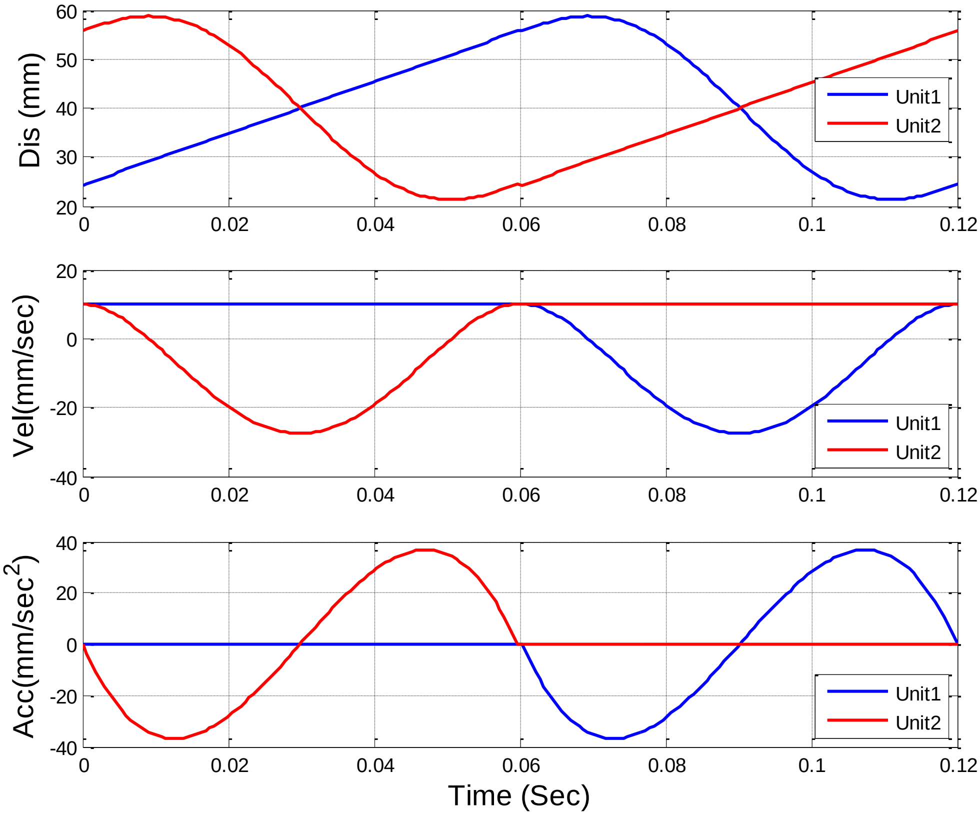

Figure 3-6: Outward and return stroke displacement, speed, and acceleration for a cam with a (constant velocity & poly (1-5)) profile.

The equations representing the whole cam profile can be listed in the table below:

Table 3-1: The kinematic equations of the whole cam profile, which is a combination of constant velocity and 5th-degree polynomial, with H as the follower stroke.

Angle of cam rotation (θ)	Type of profile	Displacement	Velocity	Acceleration
[0- π]	Constant velocity	$x_{i=r_b+c_i\,\theta_i}$	$v_i = c_i\,\omega_i$	0
[π- 2π]	5th degree polynomial	$a_o + a_1\theta_2 + a_2\theta_2^2 + a_3\theta_2^3 + a_4\theta_2^4 + a_5\theta_2^5$	$1a_1 + 2a_2\theta_2 + 3a_3\theta_2^2 + 4a_4\theta_2^3 + 5a_5\theta_2^4$	$2a_2 + 6a_3\theta_2 + 12a_4\theta_2^2 + 20a_5\theta_2^3$

3.7 Slotted Link

In order to find the kinematics of the slotted link, the geometrical relation shown in figure (3.4) should be presented. The equation which represents the angular displacement of the slotted link can be written as follows:

$$\tan\theta = \frac{X_i}{Y_i}$$

or $$\theta = tan^{-1}\frac{X_i}{Y_i} \qquad (3.13)$$

Where

θ : Angular displacement of the slotted link.

X_i : Liner displacement of the cam follower.

Y_i : Vertical displacement between axis of rotation of the slotted link and the line of motion of the cam follower.

For finding the angular velocity of the slotted link, equation 3.7 is differentiated with respect to time, this gives:

$$\frac{d}{dt}tan^{-1}\frac{X}{Y} = \frac{\frac{1}{y}*x^{\cdot}}{1+\left(\frac{x}{y}\right)^2}$$

$$\dot{\theta} = \frac{Y_i\,X_i - \dot{Y}_o X_i}{Y^2+X^2} \qquad (3.14)$$

Where

$\dot{\theta}$: Angular velocity for slotted link

$\dot{Y}_o$: Vertical velocity of the slotted link.

To find the angular acceleration of the slotted link equation 3.8 is differentiated with respect to time, this gives:

$$\ddot{\theta} = \frac{(Y^2+X^2)[(Y_i\dot{X}_i+X_i\dot{Y}_o)-(\dot{Y}_o\dot{X}_i)]-[(Y_iX_i-X_i\dot{Y}_o)(2Y_i\dot{Y}_o+2X\dot{Y}_o)]}{(Y^2+X^2)^2} \qquad (3.15)$$

Where

$\ddot{\theta}$: the angular acceleration of the slotted link.

$\dot{X}_i$: the velocity of the cam follower.

3.8 Follower Grooved Wheel

The follower grooved wheel is the element that is analyzed in this Section. From figure (3.4), the following displacement relation can be written:

$$X_o = \frac{Y_o X_i}{Y_i} \qquad (3.16)$$

Where

X_o : linear displacement of the follower grooved wheel.

Y_o : normal distance between the line of motion follower grooved wheel and axis of rotation of the slotted link.

Now differentiate equation (3.16) with respect to time to find the velocity of the grooved wheel follower.

$$\dot{X}_o = \frac{\dot{X}_i + X_i * \dot{Y}_o - X_i * \dot{Y}_o}{Y_i^2}$$

$$\dot{X}_o = \frac{\dot{X}_i}{Y_i^2} \qquad (3.17)$$

Where

$\dot{X}_o$: linear velocity of the follower grooved wheel.

X_i: linear displacement of the cam follower.

The equation (3.11) is differentiated to find acceleration follower grooved wheel:

$$\ddot{X} = \frac{Y_i^2 \ddot{X}_i - 2\dot{X}_i Y_i}{Y_i^4} \tag{3.18}$$

Where

$\dot{X}_i$: linear velocity of the cam follower.

3.9 Grooved wheel and output shaft

The equation of motion for the grooved wheel follower during cam rotation can be written as in equation 3.18.

$$X_o = \frac{Y_o X_i}{Y_i}$$

The groove is planned to have a constant velocity profile as:

$$X_o = C_o \theta_o \tag{3.19}$$

where

X_o: the displacement of the grooved wheel follower.

C_o : constant = 1.

θ_o : the grooved wheel's angle of rotation.

Combining equations (3.16) and (3.19) provides the following relationship between the input and output shafts:

$$\frac{X_i}{Y_i} = \frac{X_o}{Y_o} \tag{3.20}$$

Eq. (3.14) can be restructured using the relation $Y_i + Y_o$=Y, where Y is the perpendicular distance between the cam follower and the followe r grooved wheel, as shown in figure (3.4).

$$\frac{X_i}{Y-Y_o} = \frac{X_o}{Y_o} \tag{3.21}$$

Rearrangement the above equation

$$\theta_o = \frac{X_i Y_o}{X_o} \tag{3.22}$$

Where

θ_o : equation angular displacement for the grooved wheel.

Now differentiate equation (3.18) with respect to time to find the angular velocity:

$$\frac{d\theta_o}{dt} = \frac{C_i}{C_o} * \frac{d}{dt}\frac{\theta_i Y_o}{(Y - Y_o)}$$

$$\omega_o = \frac{1}{C_o} * \left[\frac{X_i * \dot{Y}_o + \dot{X}_i - (X_i * \dot{X}_i)}{(Yi)^2}\right] \tag{3.23}$$

Where

ω_o: $\frac{d\theta_o}{dt}$ = angular velocity of the grooved wheel and output shaft.

Now to find angular acceleration differentiate equation (3.17) with respect to time:

$$\frac{d\omega_o}{dt} = \frac{1}{C_o}\frac{d}{dt}\left[\frac{(Yi)^2 * \left((X_i\ddot{Y}_o + \dot{Y}_o\dot{X}_i) + \ddot{X}_i - (X_i\ddot{X}_i + 2\dot{X}_i)\right) - \left(2Yi\left(X_i * \dot{Y}_o + \dot{X}_i - (X_i * \dot{X}_i)\right)\right)}{(Yi)^4}\right]$$

$$\alpha_o = \frac{1}{C_o} * \left[\frac{(Yi)^2 * \left((X_i\ddot{Y}_o + \dot{Y}_o\dot{X}_i) + \ddot{X}_i - (X_i\ddot{X}_i + 2\dot{X}_i)\right) - \left(2Yi\left(X_i * \dot{Y}_o + \dot{X}_i - (X_i * \dot{X}_i)\right)\right)}{(Yi)^4}\right] \tag{3.24}$$

Where

α_o : $\frac{d\omega_o}{dt}$ $=$ angular acceleration grooved wheel.

3.10 Ratchet

As mentioned previously, the function of the ratchet is to lock the grooved wheel to the output shaft in one direction and disconnect the in the reverse direction. Accordingly, it has a high rubbing (or sliding) angular velocity during the return stroke of each unit. This rubbing velocity depends on the value of the transmission ratio selected in the system. The kinematic equations governing the motion of the ratchet are the same as the grooved wheel derived above. The linear sliding velocity depends on the working radius of the ratchet. The feature of the ratchet used in this study can be seen in the appendix.

4 Chapter Four: Simulation of The IVT by SOLIDWORKS

4.1 Simulation of system

4.1.1 SolidWorks Premium 2018

It is a program used in the design of 3D mechanical parts (CAD). This program works under the Microsoft environment, where it was developed by the Dassault Systems Solidworks Corp Company, which is one of the Dassault Systems groups in France. Three of the German universities have approved (Solidworks) as a scheduled program to train their students, and that is why they bought 500 educational copies of it. It is also a design program approved by many international companies and research agencies such as (NASA).

4.1.2 Consideration of the study

The limitations of the present study include the following

1. Theoretical study

2. Dimensions in (mm)

3. Study two type cam profile

4. Study (kinematic & kinetic) for system

5. Use aluminum as a material for some part don’t have contact force

6. Use steel (St 37) as material for some parts that have contact force

4.2 Mechanism and operation of the system

4.2.1 Mechanism

The planned (IVT) system is a structure of two similar units gathered in a single system illustrated in Fig. (4-1). Each unit including the next elements:

(a) Cam joined on the entry shaft of the structure;

(b) Follower cam;

(c) a Slotted linkage hinged with respect to the shaft fixed to the adjustable transmission ratio (power screw);

(d) Follower a grooved wheel, this bar having a cylinder-shaped crossway tip-off on both finales in order to attach with the slotted link at one margin and with the grooved wheel at the other margin (see Fig.4.2);

(e) Grooved wheel installed to the output shaft by (ratchet).

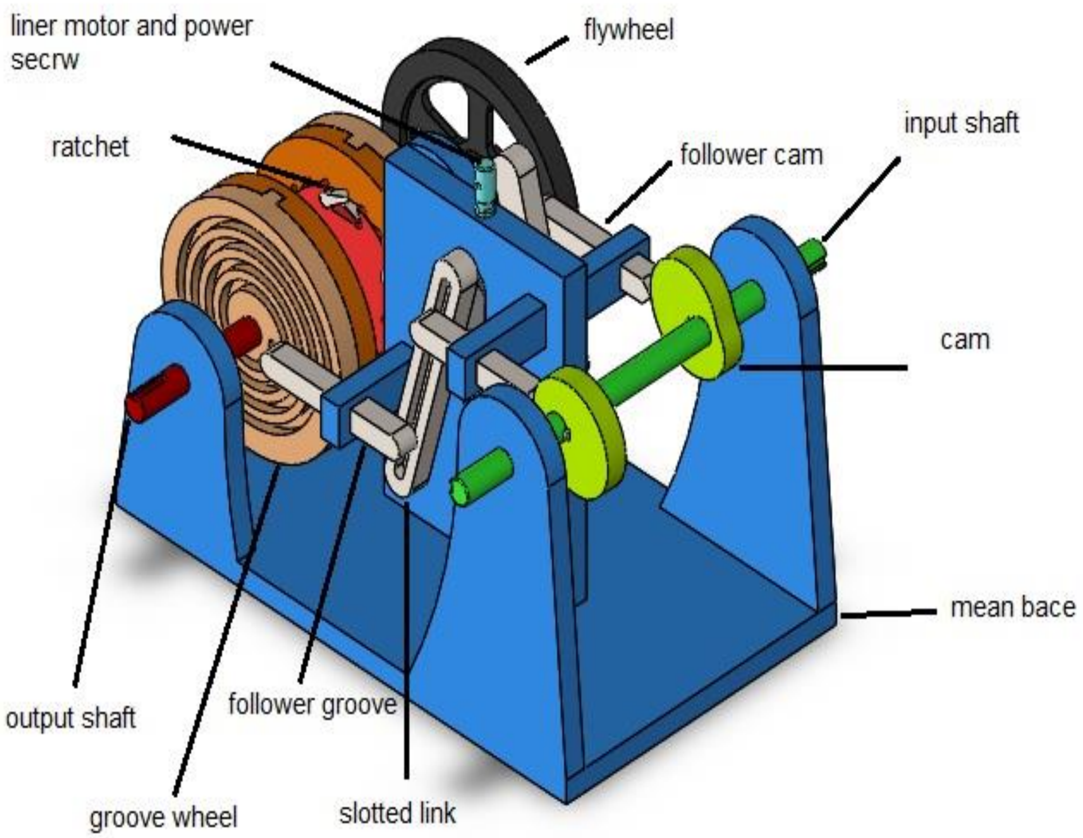

Figure 4-1: The planned (IVT) system.

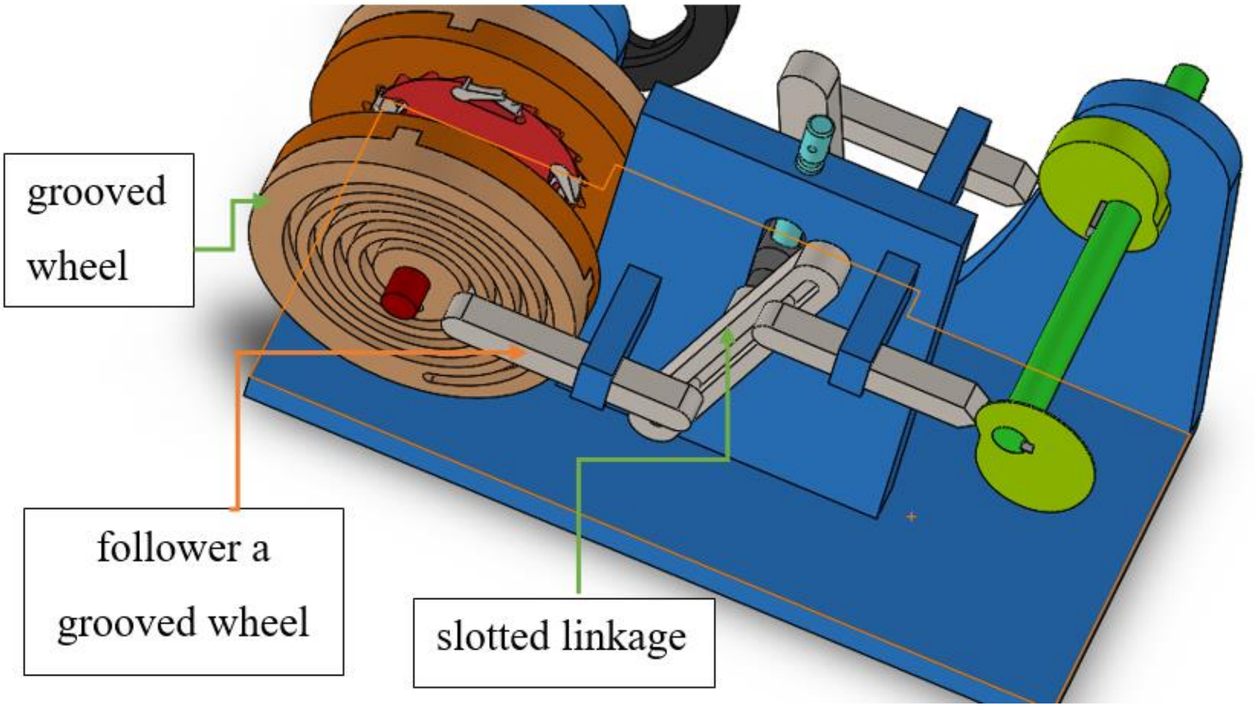

Figure 4-2: The connection follower grooved wheel.

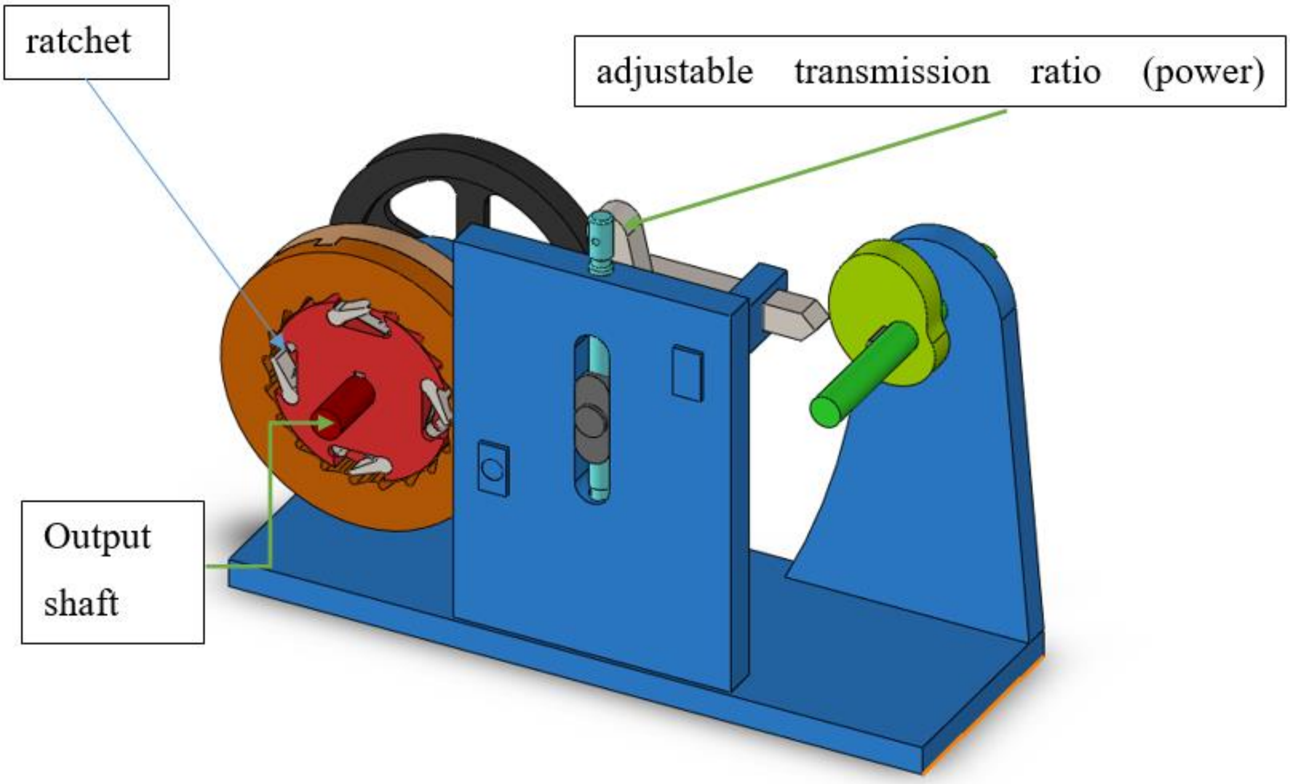

Figure 4-3: Rachet and power screw output shaft.

4.2.2 Operation of the system

The idea of operation of the planned (IVT) device can be explained in terms of stages representative of the form of a motion at each element as follows.

(a) The angular velocity of the cam at the (input-shaft) is changed into a reciprocating velocity by the follower cam.

(b) The reciprocating follower velocity is changed into an oscillatory rotational velocity at the slotted linkage. The amount of this oscillatory angular velocity depends on the selected vertical location of the pivot of revolution of the slotted linkage, which is regulated by the adjustable transmission ratio (power screw).

(c) The oscillatory angular velocity of the slotted linkage is repaired again into a reciprocating linear velocity by the follower of a grooved wheel.

(d)The velocity follower of the grooved wheel is changed into an oscillatory rotational velocity at the grooved wheel on the output shaft. The angular amount of this velocity, depending on the linear velocity of the (follower a grooved wheel), may vary from that at the cam for the same cycle of time. For high transmission ratios(4 time), the grooved wheel rotates with an angle greater than one cycle, i.e. there is a need for a cam contour covering an angle more than 2π. This is the aim behind using a grooved wheel rather than a cam.

(e) By using (ratchet), a selected one-way direction of the angular oscillatory velocity of the grooved wheel will activate the output shaft. This product is a one-directional rotational velocity of the output shaft.

The explanation of the procedure above shows the motion through the outer stroke of the follower cam. For the revert stroke, the mechanism trips to its initial location. This can be accomplished via torsional springs supplied to the slotted linkages that maintain the cam and the follower in connection constantly through the operation. Through the outward stroke, via the consequence of the ratchet, the grooved wheel and the output shaft are joined while they slide relative to each other through the return stroke. This means that the ratchet continues an elected unidirectional rotating motion of the output shaft from oscillatory angular velocity. Accordingly, through the return stroke, there will be a cutoff in the transmission of the power if one unit is utilized in the system. In order to accomplish a constant transmitted torque, the device must have at smallest two units oriented along the (in and out) shafts (see Fig.4-1).

The change in the transmission ratio is accomplished via a stepless movement (shifting) of the axis of revolution of the slotted linkages via an adjustable transmission ratio. A (power screw linear motor) can be utilized for this drive, as showed in Fig. (4.3). This shifting will vary the amount of the angular oscillatory velocity of the slotted linkage and then the angular displacement of the output shaft. In other words, the relation between the (in and out) angular displacements are regulated via the adjustable transmission ratio location, which can be changed continuously (stepless) lengthwise the vertical axis. This mechanism permits the planned system to be considered as a (CVT) system. The system also can be considered as an IVT, as it will be demonstrated by the kinematic analysis of the system that is presented in detail in the next Section.

4.3 (IVT) Simulation Model

The principle of the device's work, as shown in the previous paragraph, was modelled by (SOLIDWORKS 2018). A three-dimensional (IVT) model was built from scratch. The specifications of the materials used in the parts of the device were selected from the program library (Solidworks 2018). More than once, the shape of the cam was tested and according to what will be explained in the following paragraphs. In the simulation, the friction was neglected, the bearings have not used the length of the helical track in (Grooved wheel) were obtained experimentally to suit the simulation requirements, more than one a type (ratchet) was used, where the type that provides more stability in the angular velocity of the exit axis was used.

4.3.1 Considerations of the Simulation Model

In the simulation of the device, parts very close to a practical reality (implementable dimensions) were used. At the same time, the parts of the device were very simple, and this facilitates the implementation process. This feature makes the simulation process easy and uncomplicated. For facilitating the simulation and an analysis process more, the parts of the device are considered rigid bodies. The main base is not done. The simulation considers that the machine does not contain bearings, (a power screw, the linear motor) which is used to relocate (slotted link). The simulation process of the system was carried out at the following locations (45, 55, 65) mm, where these distances represent the center of rotation (slotted link) from the horizontal path of the cam follower.

4.3.2 Component of the simulation model

The simulated parts are explained as follows:

1.Two Profile (constant speed & (constant speed & poly (1-5))
2. Input shaft.

3. Two units of cams followers.

4. Slotted joint two units.

5. Groove wheel follower of two units.

6. Groove wheel two units.

7. Take out the shaft with the ratchet.

The 3-D representation of the main components of can be described in Figure 4-4.

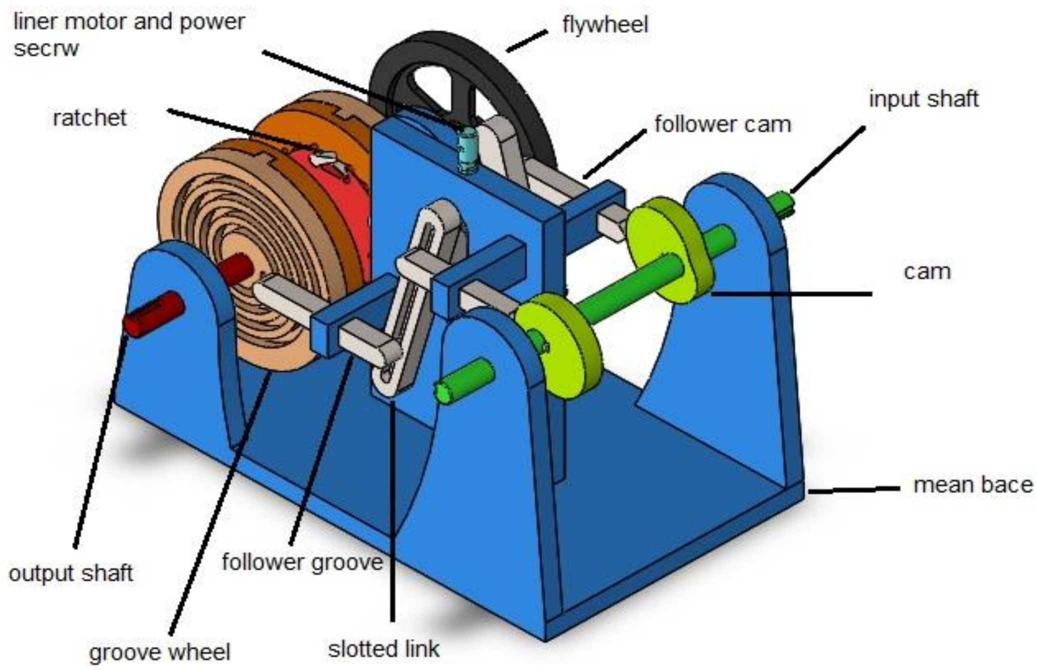

Figure 4-4: Parts of (IVT) system used at simulation.

4.3.2.1 Cam profile

In this system, we use two types of cam profiles as below.

1. Constant velocity for range (0- π).

2. Constant velocity range (0-π) & poly (1-5) range (π-2π). The dimensions for the two cams are listed in Table 4-1.

Table 4-1: Properties & dimensions of the two-cam profile.

<table>
<tr><td>Type of cam</td><td>0-π</td><td>π -2π</td><td>H (mm)</td><td>Base Circle (mm)</td><td>Hight (mm)</td><td>Dim Shaft (mm)</td></tr>
<tr><td>Mode stroke</td><td>forward</td><td>backward</td><td rowspan="3">24.29</td><td rowspan="3">40</td><td rowspan="3">20</td><td rowspan="3">22</td></tr>
<tr><td rowspan="2">Type of profile cam</td><td>Constant velocity</td><td>Constant velocity</td></tr>
<tr><td>Constant velocity</td><td>Poly (1-5)</td></tr>
</table>

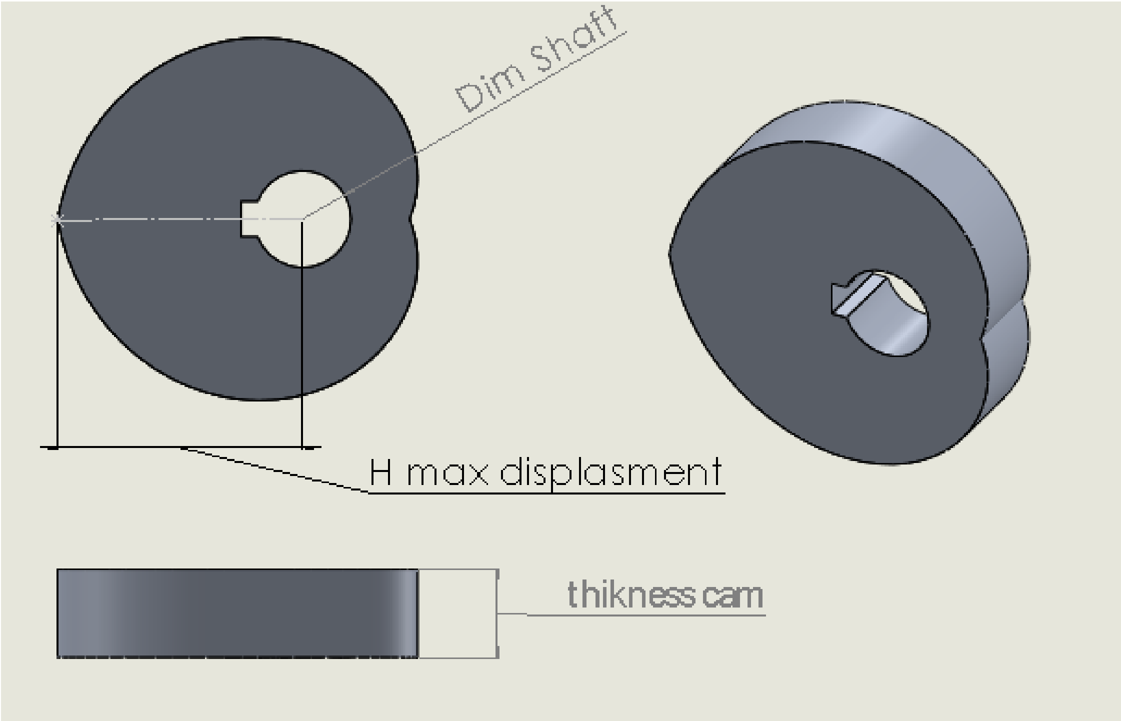

Figure 4-5: Cam profile constant velocity (2D&3D)

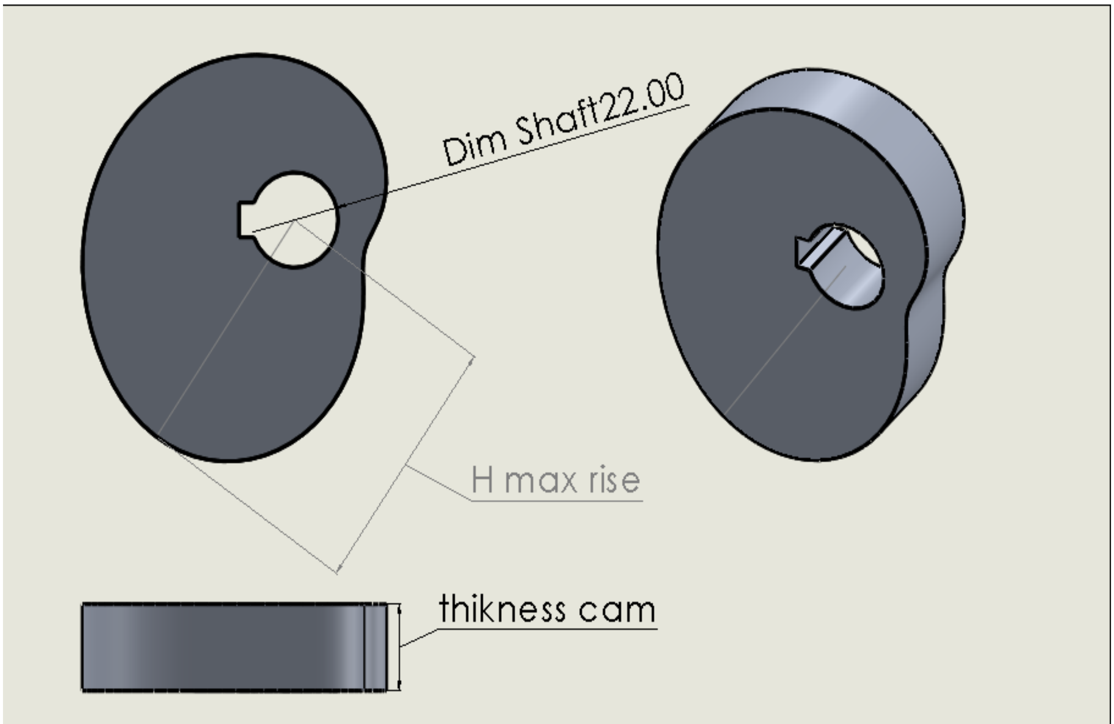

Figure 4-6: Cam profile (constant velocity & poly (1-5)) (2D&3D).

4.3.2.2 Follower cam

In this part, the dimensions were chosen according to the requirements of the work and the performance of the device. It is designed to be with an end (knife) from the contact side with the cam because it contains narrow places that the rest of the types cannot reach and therefore does not give accurate movement. In addition, it contains a cantilever pin. The frequency movement is transferred to a slotted link. According to the dimensions, they are executable dimensions if used as a test model in the future. This part is used for both assemblies (constant velocity cam profile constant velocity & poly (1-5)) as shown below.

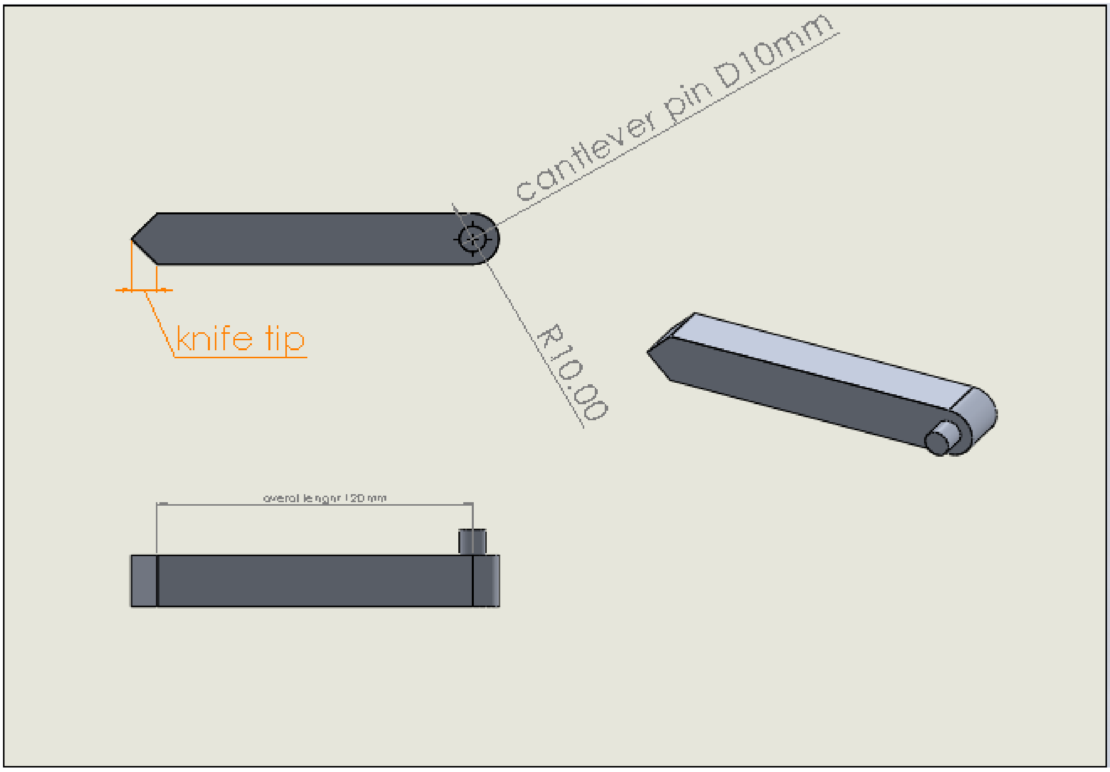

Figure 4-7: 2D & 3D follower cam.

4.3.2.3 Grooved wheel

The grooved wheel is the part of the device responsible for transmitting the linear reciprocating movement in (Grooved wheel follower) and converting it into a rotational frequency movement prepared into (output shaft). Also, the dimensions of this part were chosen to be suitable for switching between (input shaft & output shaft). The reason is that the length of the spiral fissure reaches (8π). It is fabricated as a constant velocity profile, as the grooved wheel is connected from the side (output shaft) by a mechanism (ratchet).

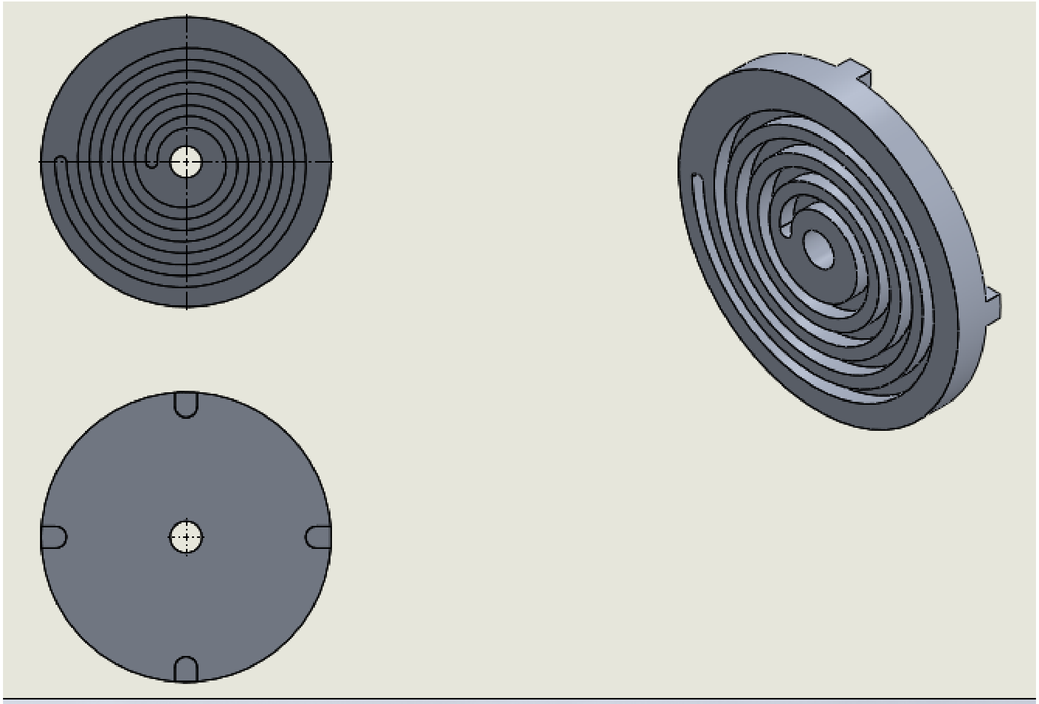

Figure 4-8: Grooved wheel.

4.3.2.4 Input and output shaft

The dimensions of input and output shaft were chosen to have the ability to transmit momentum and loads on the one hand and consider their presence in the local markets if the device is implemented liberally from another side, and for the above reasons, their radii (22 mm) was chosen. It will be explained in the next paragraph.

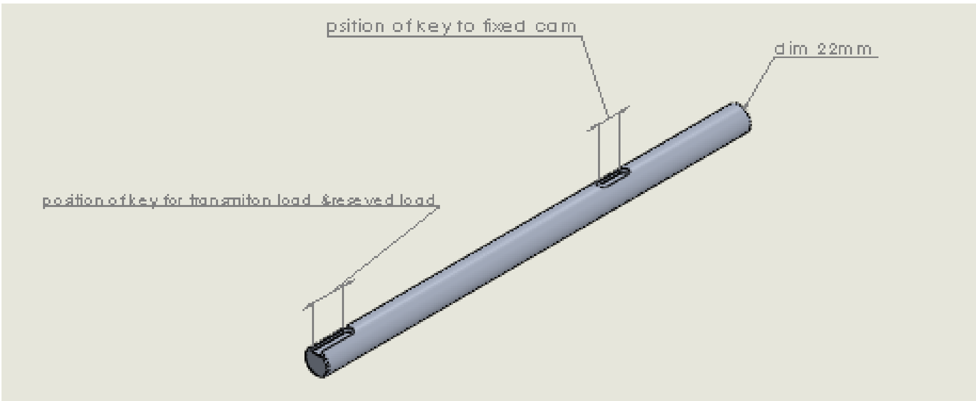

Figure 4-9: Input/output shaft.

Hint: keys haven't carried out in simulation.

4.3.2.5 The raw materials used in the machine

The materials were chosen to be suitable to withstand stress and were included in the library of the SOLIDWORKS program. In addition to their availability in the market, for this reason, the following materials were chosen (ST 37 & AL 3060).

4.3.2.6 Mate & constrain

To ensure SOLIDWORKS works properly, a suitable mate must be chosen to have an impact on the results obtained from the device. Used mate should approve the input, output, and main base. Above is important also for option MBD.

4.3.2.7 Input & output boundary

The input shaft is provided with a fixed rotating speed (500 R.P.M). The rotational speed in the output shaft is read when the slotted link position changes in the following locations (65, 55, 45) mm, as well as up and down between these values using a linear speed motor Fixed (6.25,12.5,18.75&25) mm / S. Loads were added to the output shaft.

4.3.2.8 Ratchet or (one-way clutch)

This mechanism is important because it converts the reciprocating rotational movement resulting from the grooved wheel into a rotational motion in one direction only, in other words (engagement & disengagement). For this reason, two types of ratchets were chosen. Stable rotation.

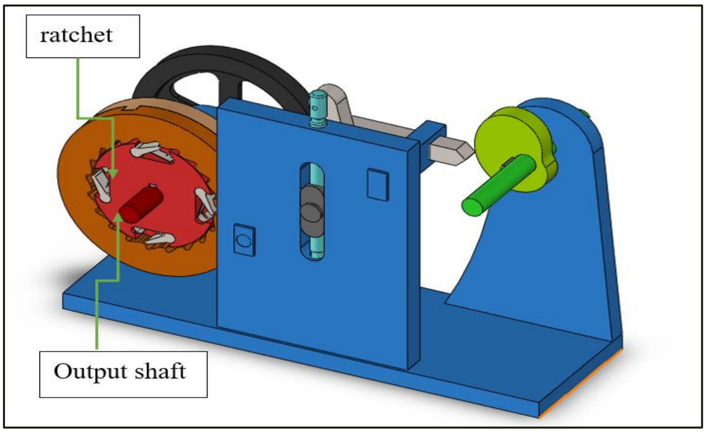

Figure 4-10: Ratchet connection with the output shaft.

4.3.2.9 Power Screw

A power screw is used in this system according to dimensions suitable with the dimensions of the device, where the rotational motion provided for it by (electric motor or stepper motor) is converted to a linear velocity of 6.8 mm / S. This part is considered one of the basic parts. It changes shift ratios by changing the position of the spindle (slotted link). Bearings are not used between it and the main base.

4.3.2.10 Main Base & Middle Base

The main base, the dimensions and measurements of this basic part were chosen to suit the device's work requirements (300, 500, 400) mm and thickness 20mm. A middle base was chosen to ensure that the parts (the camera follower and the follower, a grooved wheel & power screw) are in the specified location. . All parts are explained in Figure.

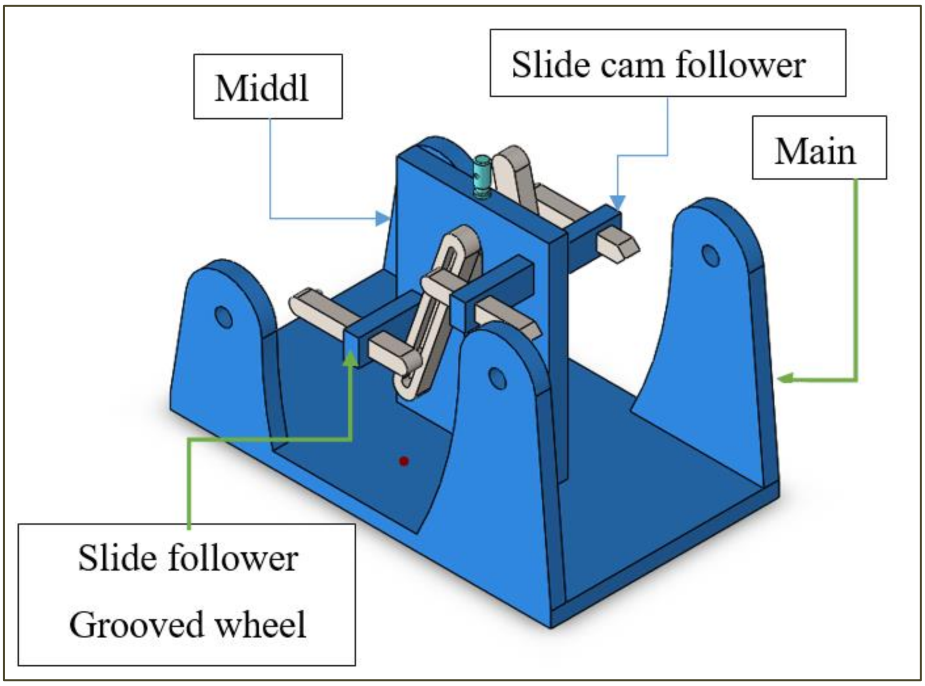

Figure 4-11: Main & middle base

5 Chapter Five: Results and Discussion

5.1 Introduction

In this chapter, the results of the theoretical analysis and simulation of the IVT system under consideration are presented and discussed in detail. The theoretical analysis presented earlier in chapter three gives full kinematic analysis for the whole IVT system as well as for every single element. The simulation study presented in chapter four was based on modelling the system using SOLIDWORKS software. Building the system model using this software is considered a considerable achievement. This is due to the high functionality features embedded in this software. The IVT system was designed in detail and operated similarly to that followed in the prototyping process. The simulation scheme is not limited to that, and it offers a full dynamic result of the system, which can be used for the sake of examining and comparison with the theoretical analysis. The next Sections in this chapter present the results obtained from both theoretical analysis and simulation.

5.2 Result of simulation & theoretical analyses for constant - constant velocity cam profile

In this Section, the results obtained from the simulation of SOLIDWORKS model & theoretical analysis are presented.

5.2.1 Single Unit

In figure 5.1, the displacement, velocity & acceleration are shown for a single unit during one revolution of the input shaft, which takes about 0.12 Seconds.

From 5.1 c, it can be noted that the value of the acceleration goes toward infinity at the beginning of both outworks and return strokes at times 0 & 0.06 Seconds, which is corresponding to 0 & 180 degrees of cam rotation. This behavior is caused by the sudden change of velocity with no time (theoretically). This is the reason behind selecting another type of cam profile for the return stroke, and this will be seen next Section. Another issue that can be noticed is that during the return stroke, the velocity of the follower is in the reverse direction. This was discussed previously in chapter three. Accordingly, multi-units (at least two) are needed for continuous power transmission, and this can be seen in the next Section.

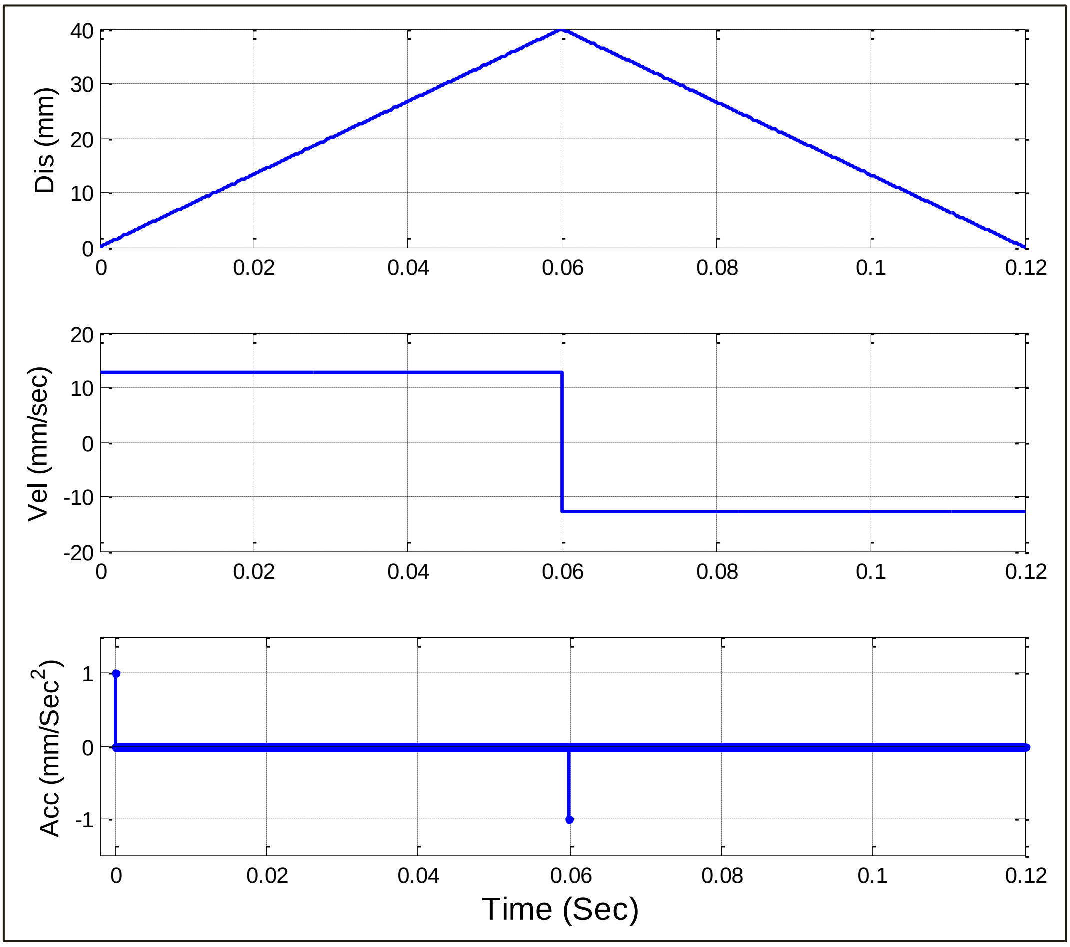

Figure 5-1: Cam follower for constant - constant velocity cam profile (single unit). Displacement, Velocity & Acceleration respectively.

5.2.2 Two Units

Figure 5.2 shows the results of simulation and the theoretical analysis for the displacement, velocity & acceleration when using two units oriented with a 180-degree lag. The result was obtained at a rotational speed of the input shaft of 500 r.p.m.

The value is of the linear velocity that was resulting from the simulation is higher than that from the theoretical analysis at the time of 0.06 Seconds for both units. The reason behind this is probably using a knife-edge follower type.

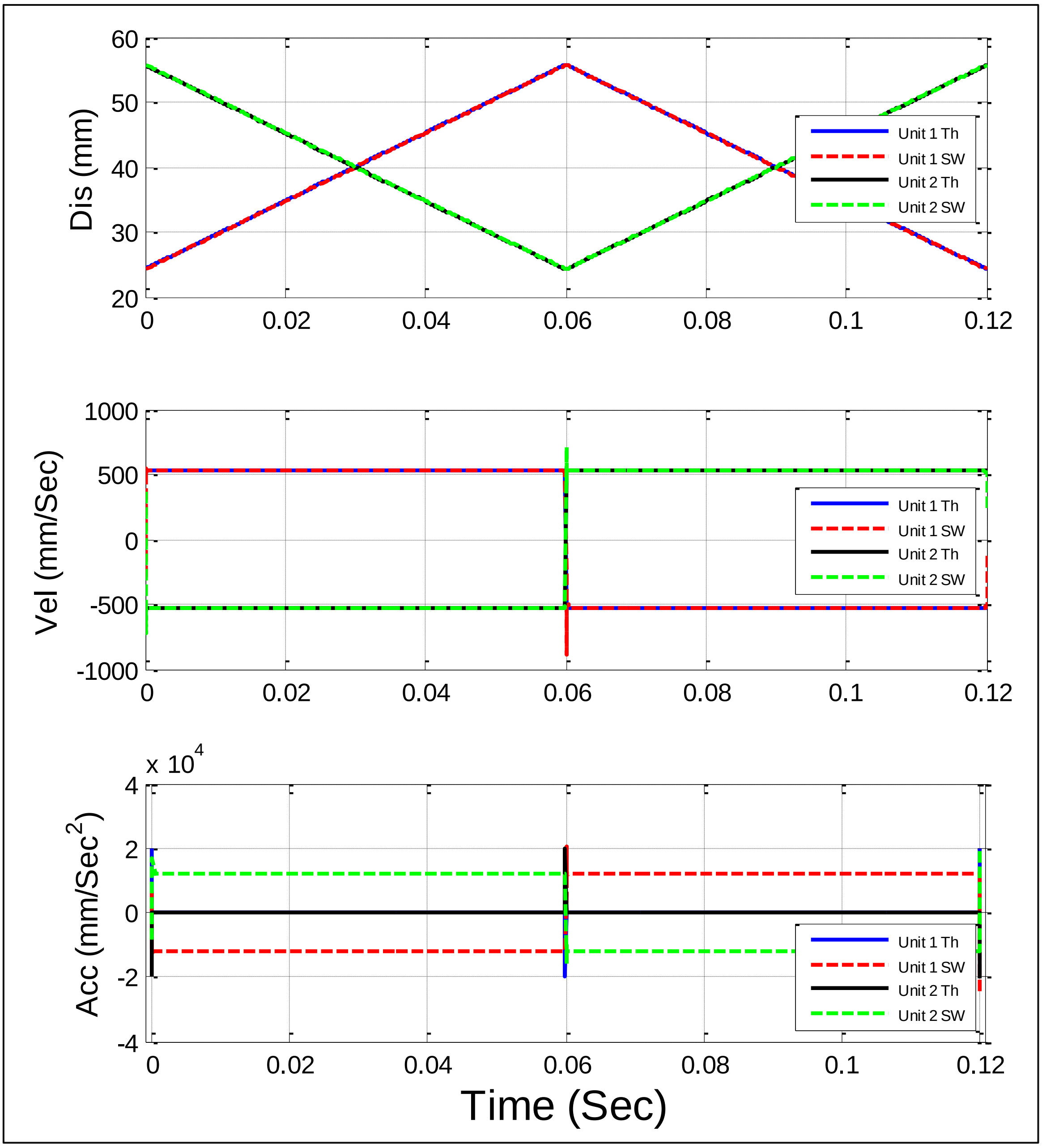

Figure 5-2: Cam follower for constant -constant velocity cam profile (two units). Displacement, velocity & acceleration Respectively.

5.2.3 The slotted link

Figure (5.3) shows the theoretical and simulation angular velocity of the slotted link for the two units of the system. The results were taken at 500 r.p.m, and the slotted link rotation axis is fixed (i.e. there is no falling or rising speed & Yi=55mm).

It can be noted that there is a difference between the simulation and the theoretical results of about 1.95%. This may be caused by the use of a flywheel in the simulation model, while on the theoretical work, it was not used. Another expected reason for that behavior is the use of knife-edge cam followers.

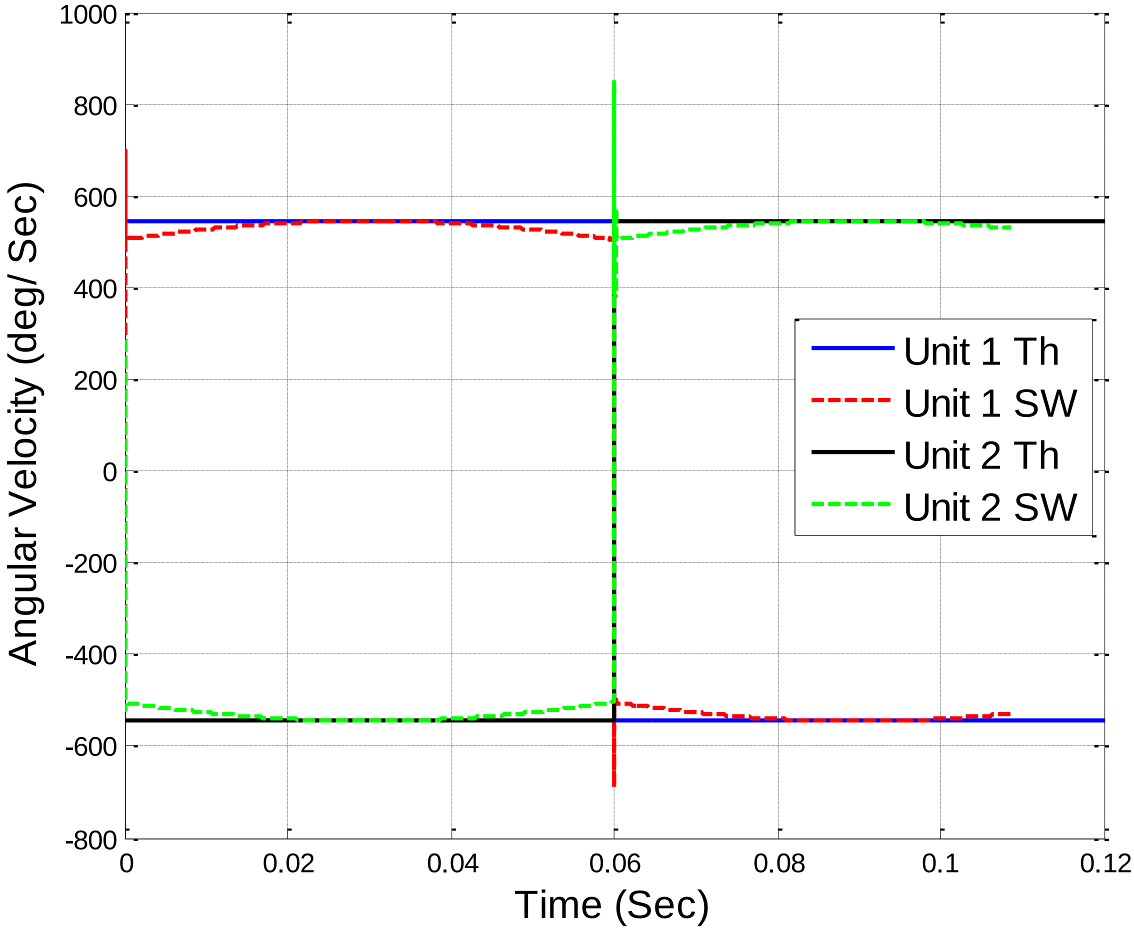

Figure 5-3: The angular velocity of the slotted Link for Two Units at Constant velocity Cam profile Th & SW (at Yi fixed at 55 mm)

5.2.4 Theoretical Slotted link angular velocity at different values of Yi

In figure (5.4), the effect of the Yi on the slotted link angular velocity has been represented theoretically using the values of 45, 55 & 65 mm for one cycle of the system. The figure shows that as the value of Yi increased, the slotted link velocity is decreased during both forward and backward periods.

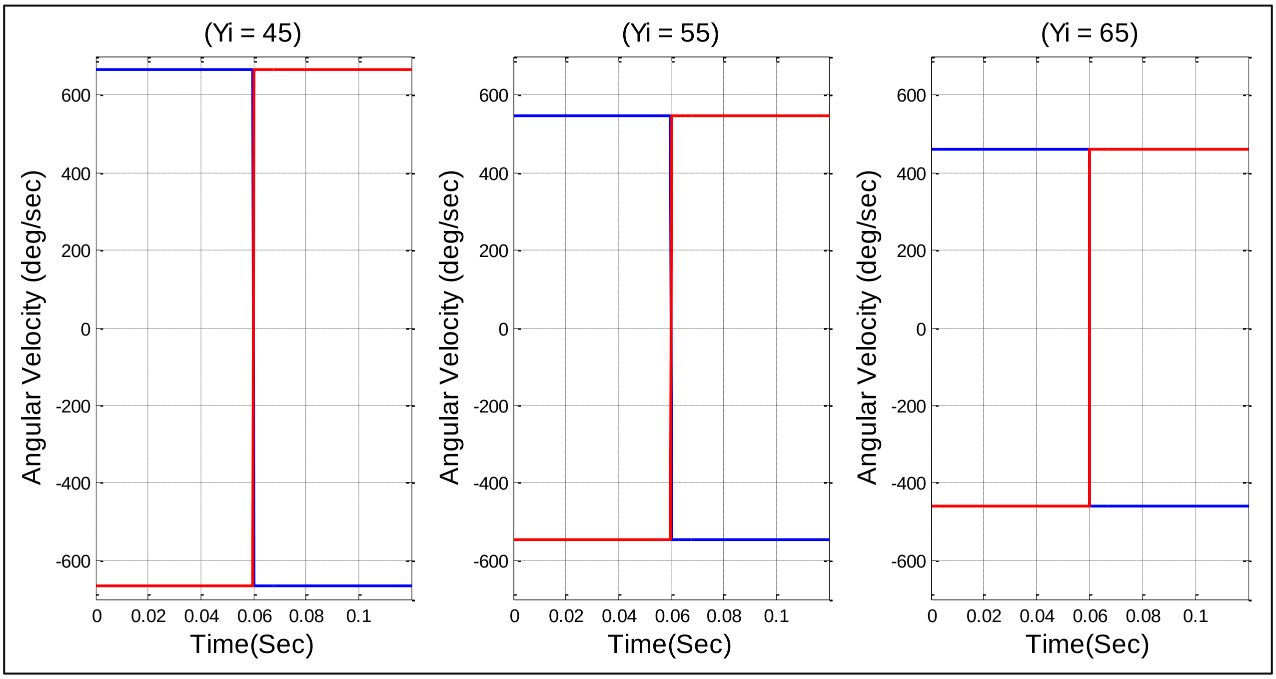

Figure 5-4: Theoretical Angular velocity of the Slotted Link for different values of Yi; 45, 55 and 65mm. Blue line: U1 & Red line: U2.

5.2.5 Linear velocity of the Grooved Wheel Follower

Figure (5.5) shows the results obtained from the simulation & theoretical analysis of the two units. At time intervals of (0-0.12) S, Yi = 55mm & an angular velocity of 500 r.p.m for the input shaft.

In general, the agreement between the simulation and the theoretical results approaches 100%, but at the following angular values (0,180 & 360, which corresponds to (0, 0.06 & 0.12 Seconds on the diagram), there are speed values higher than the theoretical results. The reason behind this again, probably using knife-edge cam follower. In this figure, it can be noted that the value of the linear velocity of the grooved wheel follower is equal to the value of the Cam Follower because the value of Yi = 55 mm, which represents the middle distance between the followers.

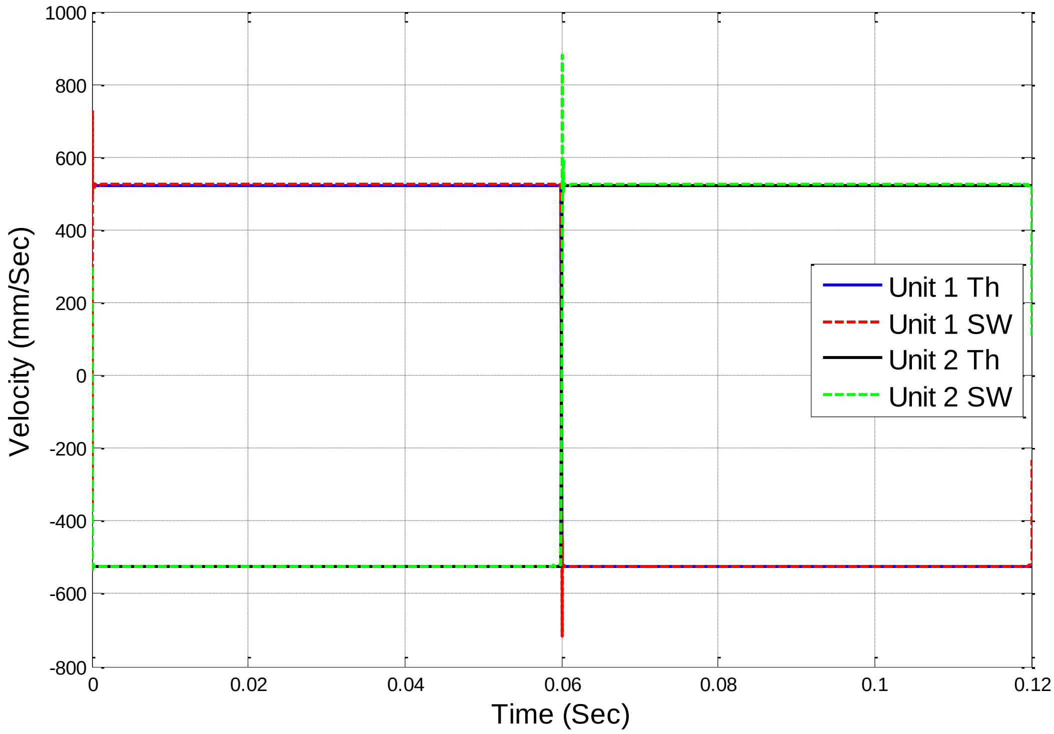

Figure 5-5: Linear velocity Follower Grooved Wheel Two unit & SOLIDWORKS &Theoretical at Yi=55 mm.

5.2.6 Angular Velocity of Output Shaft

Figure (5.6) shows the angular velocity of the Grooved Wheel & the Output Shaft. The simulation was for one cycle, which is equivalent to 0.12 Seconds. The value of the vertical distance is Yi = 55mm & the angular speed of the input shaft is 500 r.p.m. Theoretically, the value of the rotational speed of the spindle of output is 12000 r.p.m, which is four times the value of the angular velocity in the input shaft. This is due to the value of the profile coefficient of the Grooved Wheel, which was chosen to suit the working conditions of the device.

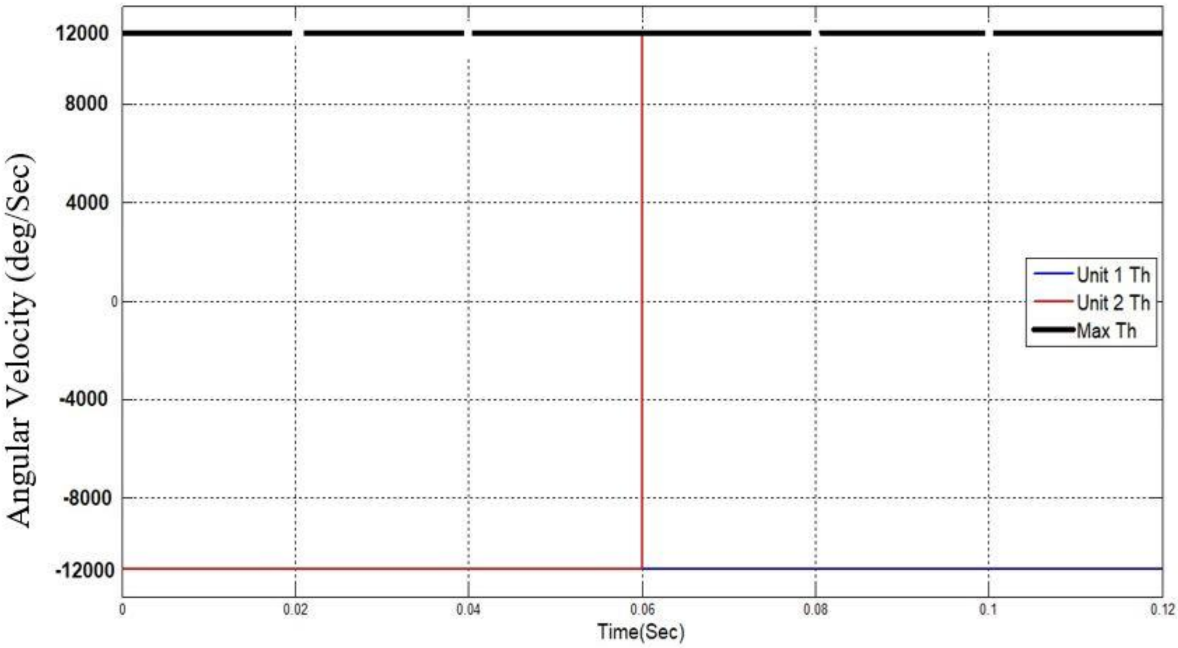

Figure 5-6: The rotational speed of the Grooved Wheel & the spindle of output

5.2.7 Angular velocity output shaft as compared to the input shaft

Figure (5.7) illustrates the value of the angular velocity of the input shaft with different velocities of the output shaft. Based on different values of Yi, the time interval was used is 0.12 S (the time sufficient to complete one cycle). The values of the angular velocity of the output shaft were taken based on the value of the vertical distance between the cam follower and the slotted link pivot, Yi, with the value of (45, 55, and 65) mm as shown in figure (5-7), with different colors.

Theoretically, the smooth output is observed in the value of the rotational speed at the spindle of output for all cases. The angular velocity is inversely proportional to the value of Yi. The values of the rotational speed of the spindle of output is 8200 deg /s at Yi = 65mm, 12000 deg/S when Yi = 55mm & 17500 deg/S at Yi=45mm.) In other words, the angular velocity of the output shaft is inversely proportional to the value of Yi.

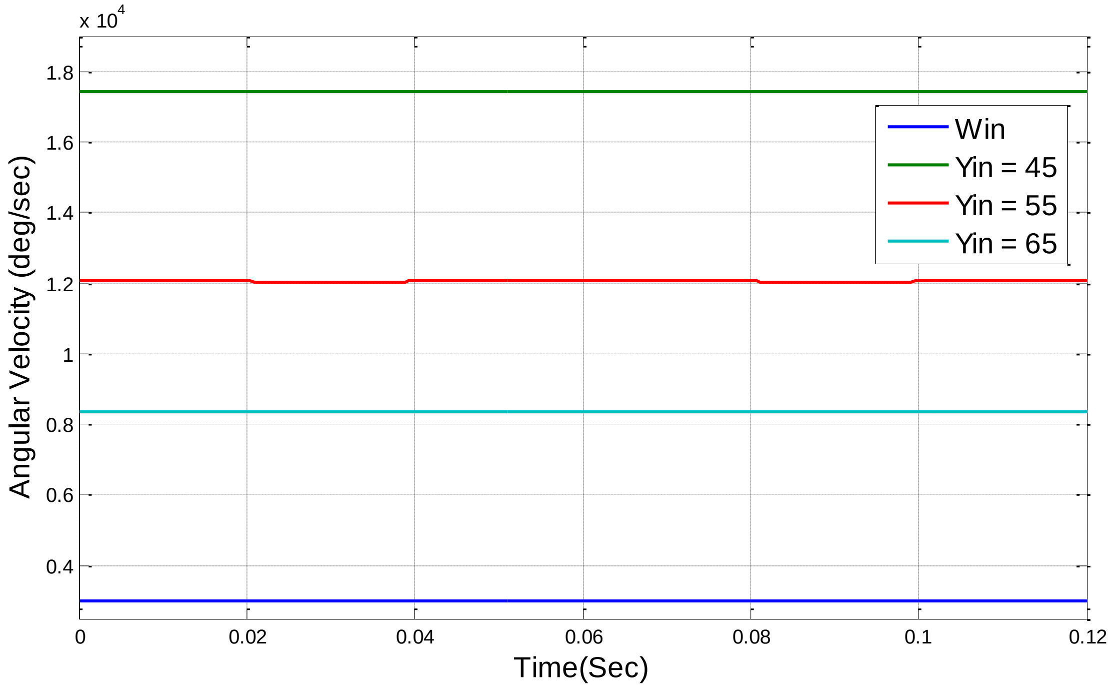

Figure 5-7: The value of the angular velocity of the input and output shafts with different for different values of Yi.

5.3 Results for constant &the poly (1-5) cam profile

In this part, the results of the combination of cam profile of constant & poly (1-5) will be presented and discussed in detail in the next Sections.

5.3.1 Cam Follower

Figure (5.8) shows theoretical &simulation result for two units.

In this figure, an error of 2% between theoretical & simulation results was observed. This error may result from the used spline at cam profile in the SOLIDWORK model. The error in figure 5.8 b is about 1%. The largest error was observed at the acceleration diagram (figure 5.8 c) of 15% for the same reason described above (used spline at cam profile).

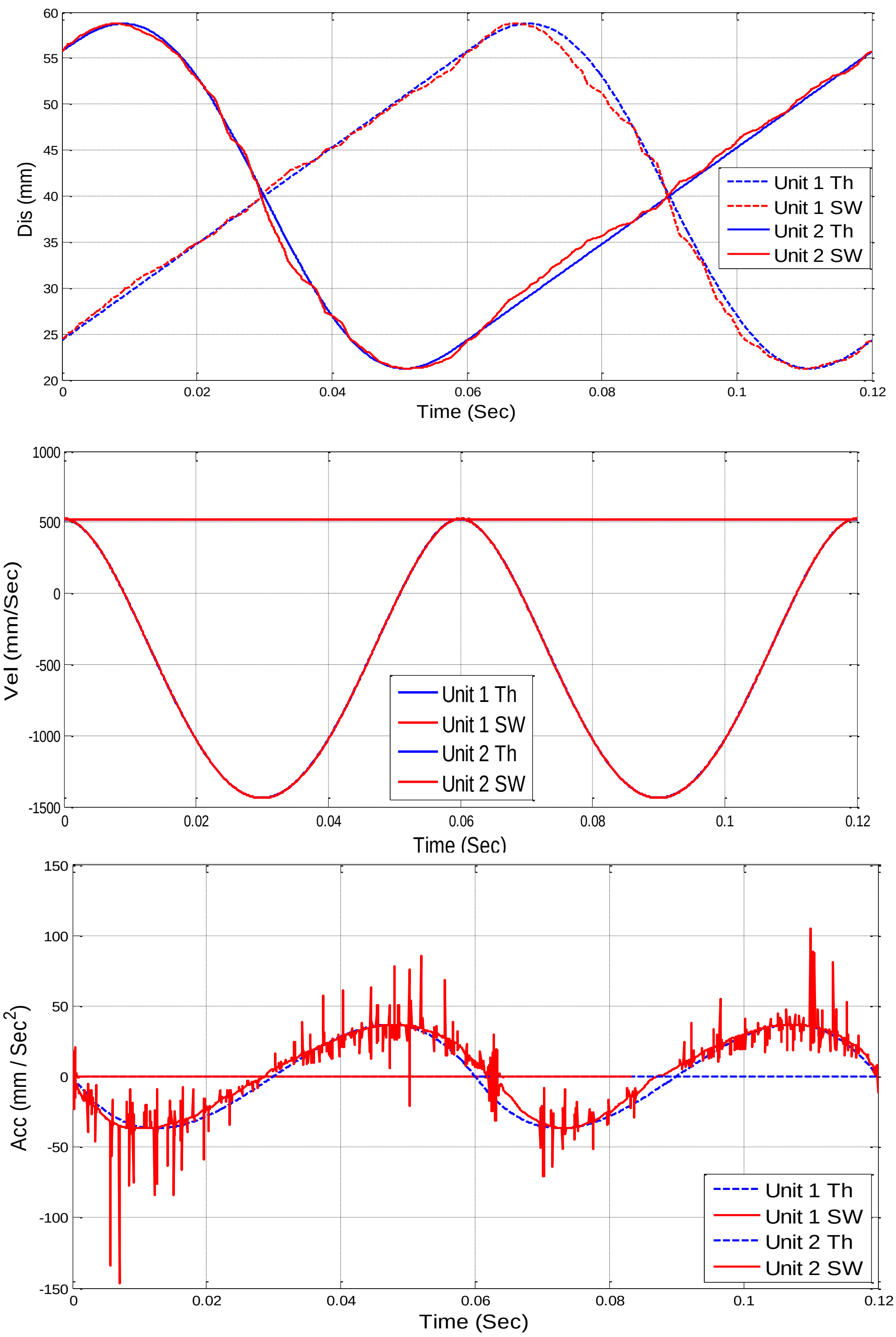

Figure 5-8: Displacement, velocity & acceleration respectively for constant & poly (1-5) cam profile based on theoretical & simulation for two units

5.3.2 Angular velocity slotted link

Figure (5.9) shows theoretical & simulation results for the angular velocity of the slotted link for two units equals to 55mm. The time interval is 0.12 S, and Yi is 55 mm.

An error of 0.29% was observed between theoretical and simulation, which can be caused by the use of a flywheel installed on the output shaft. At some parts of the time interval, the error is less than the amount mentioned above.

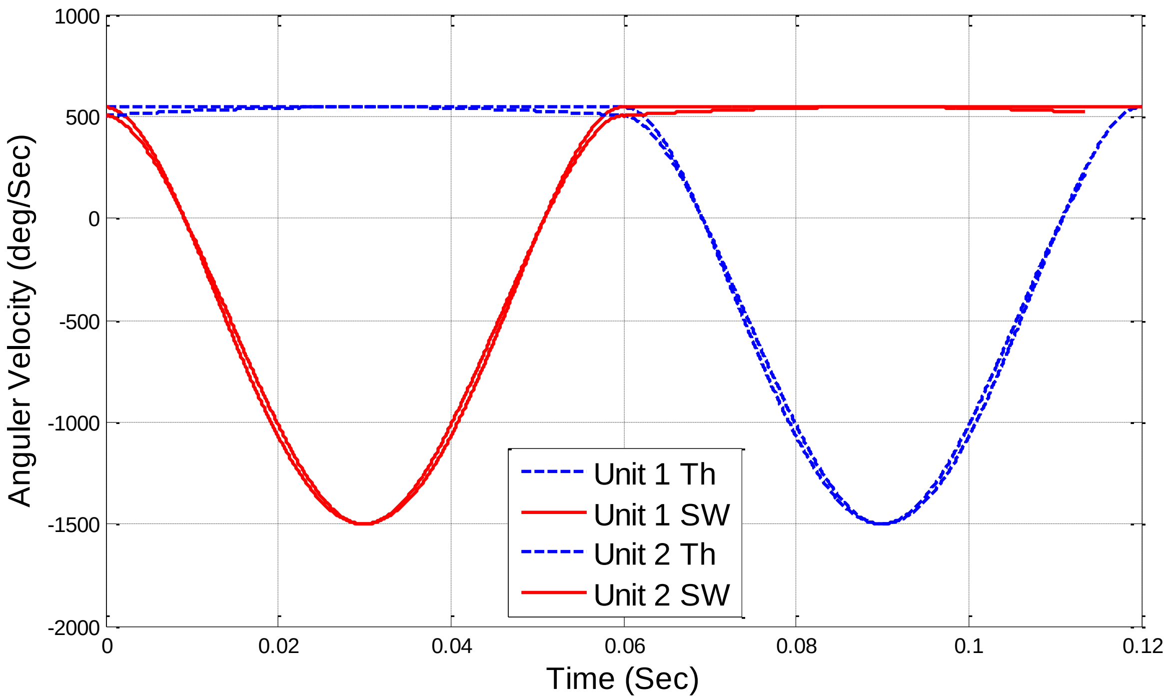

Figure 5-9: Angular velocity for the slotted link for two-unit with constant velocity &poly (1-5)) Yi=55mm.

5.3.3 Linear velocity of the Grooved Wheel Follower

Figure (5.10) shows the linear velocity for the theoretical & simulation analyses for two units. The normal distance Yi is 55mm. An error of approximately 0.01% was observed between the two types of analysis (theoretical & simulation).

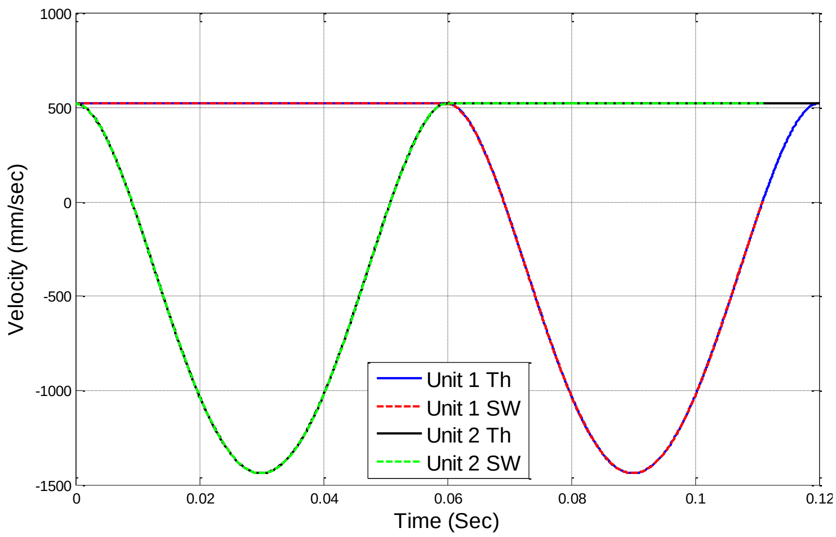

Figure 5-10: Linear velocity Follower Grooved Wheel at (constant velocity & Poly (1-5) cam profile) (simulation &Theoretical) Two unit & Yi=55mm.

5.3.4 Angular velocity of the Grooved wheel

Figure (5.11) shows the theoretical &simulation results of the angular velocity for two units. A perfect match between theoretical and simulation results was observed.

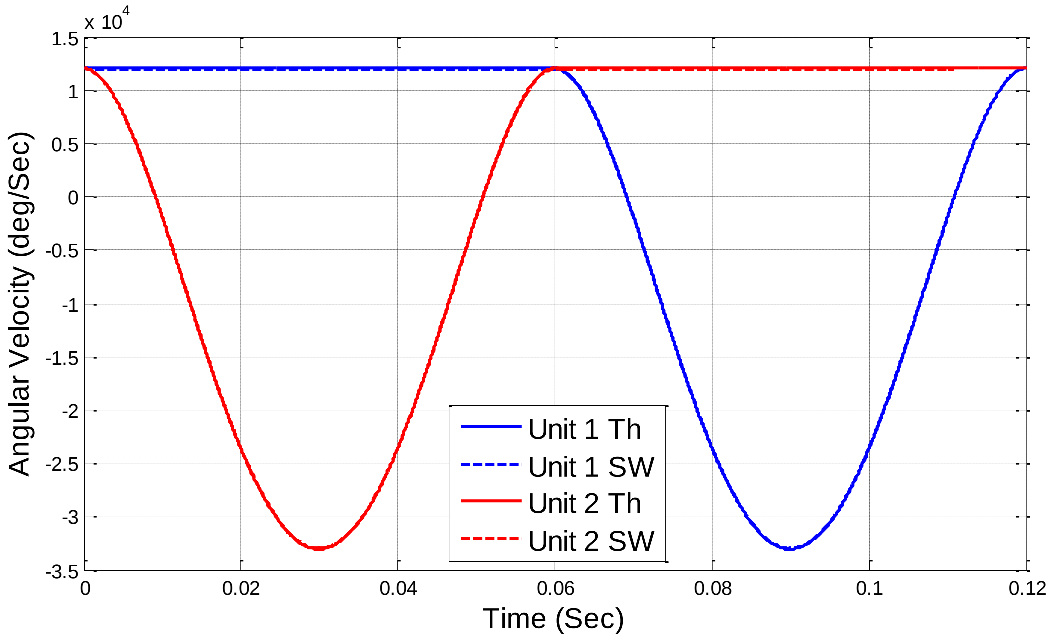

Figure 5-11: The theoretical &simulation results of the angular velocity for two units.

5.3.5 Angular velocity the slotted links

Figure (5.12) illustration the theoretical angular velocity of the slotted links for the two units) at Yi=45, 55& 65mm, and time interval of 0.12S.

The results show smooth curves in all cases. This will be reflected in the acceleration value at the connected part of the system. The angular velocity of the slotted link increased when the value of Yi decreased and vice versa. All angular velocity at portion poly (1-5) meets at zero, which is a start point to go the negative zone or backwards stroke at times 0.01, 0.05, 0.07& 0.11 S. Therefore, use two units to give positive value only. In other words, the constant velocity portion stays at work at all times of operation.

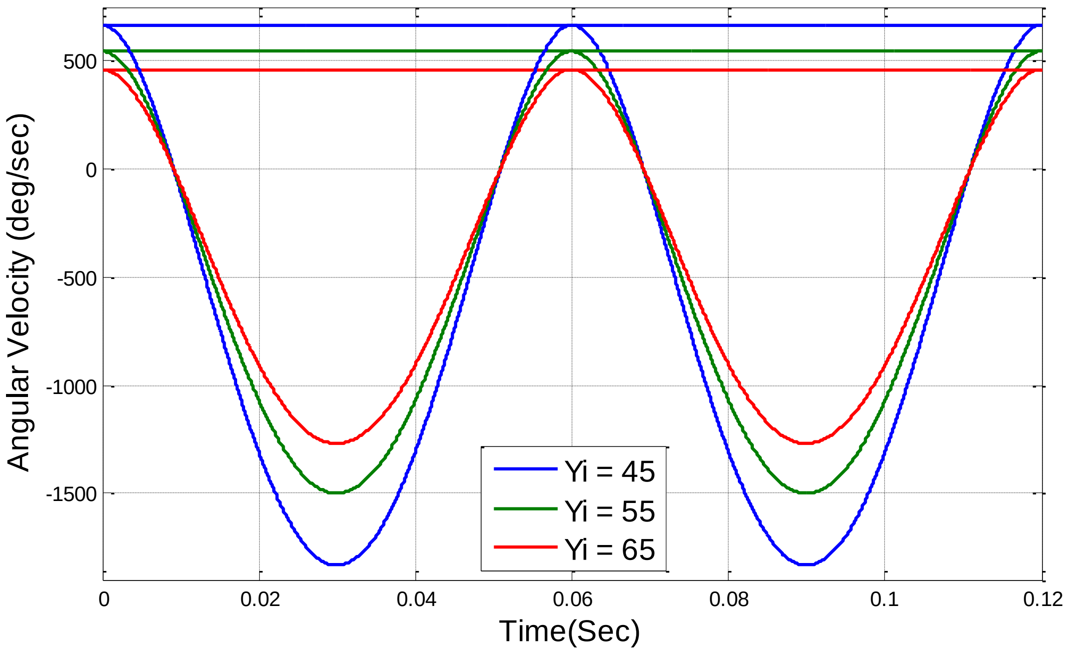

Figure 5-12: Theoretical angular velocity of the slotted link at (Constant poly (1-5) Cam profile for Yi = (45, 55, and 65) mm.

5.3.6 liner velocity Follower Grooved Wheel theoretical

Figure (5.13) illustrates the theoretical results of the linear velocity for the Grooved Wheel Follower for the two units at different Yi values. This result was obtained using a cam profile of constant velocity & poly (1-5). The used values of Yi are 45, 55 & 65mm. As usual, the time interval was taken equal to 0.12

Seconds, which the time is required to complete one revolution of the input shaft when it rotates at 500 r.p.m.

Figure (5.13) shows the inverse proportionality between the value of the follower linear velocity and the value of the vertical position Yi. It can be noticed that all curves met at zero velocity at the time (0.01, 005, 0.07 & 0.11).

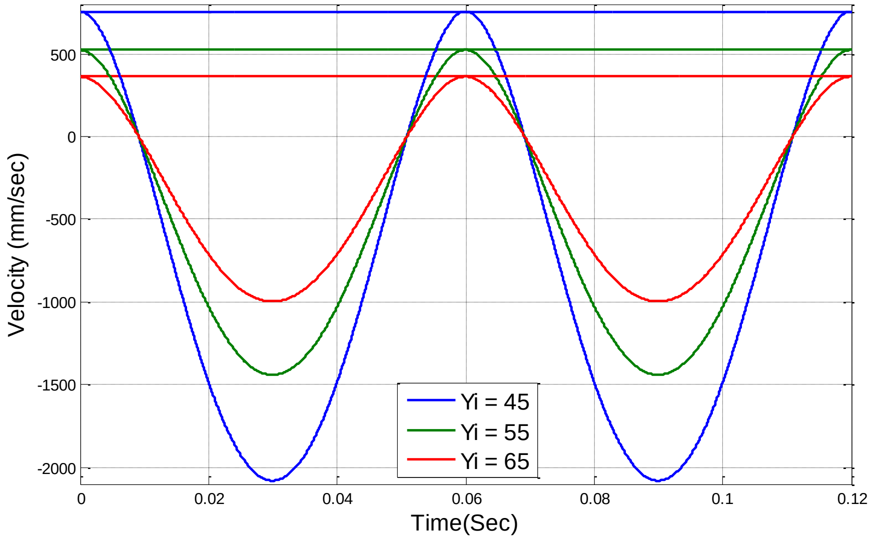

Figure 5-13: Theoretical liner velocity of the Grooved Wheel Follower at (CV & poly (1-5) Cam profile) at Yi= (45, 55, and 65) mm.

5.3.7 Angular Velocity of the Grooved Wheel

Figure (5.14) shows the angular velocity of the Grooved Wheel for the system of two units using the same condition that applied in obtaining the results shown in figure (5.13). It can be noticed that the spiral grooved used gives an angular velocity of an amount equal to 2.7-times of the input shaft speed at Yi=65mm, 4-times at Yi=55mm and 6-times at Yi=65mm. The curves shown in the figure are smooth at all times of operation. At the time 0.06 &0 S, all curves are changed from constant velocity cam profile to poly (1-5). The smoothness of the velocity curves is crucial in determining the acceleration on the output shaft.

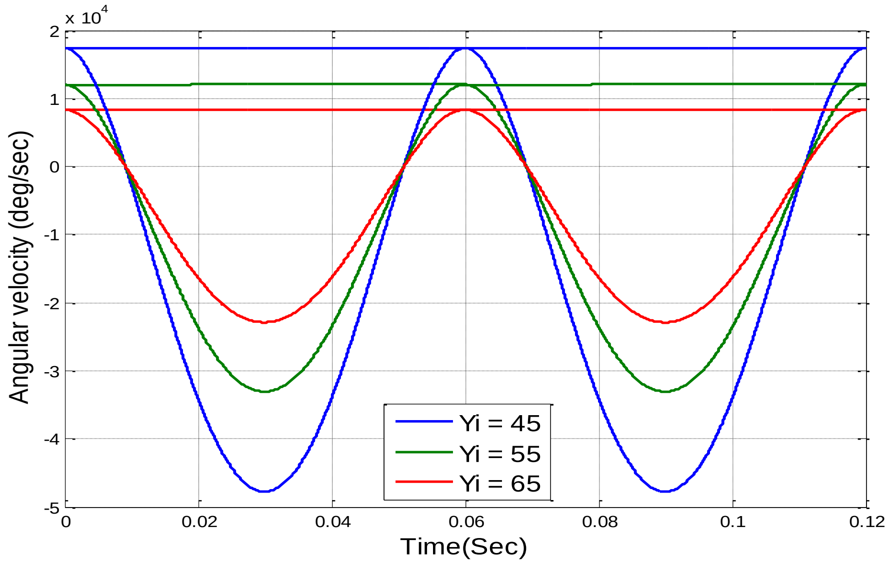

Figure 5-14: Theoretical angular velocity of the Grooved Wheel at (CV & poly Cam profile) (1-5) at Yi= (45, 55, and 65) mm.

5.3.8 Angular velocity output shaft

Figure (5.15) shows the theoretical& simulated angular velocity of the output shaft for Yi values of 45, 55& 65 mm.

The results show inverse proportionality between the value of Yi and the angular velocity of the output shaft. Another important aspect that can be drawn from these results is a drop in the velocity of the simulation results. Theoretically, the speed of the output shaft should continue at a constant value. This drop is not large compared with the output speed, and it is increased when the output speed is decreased. It might result from the ratchet used in the system.

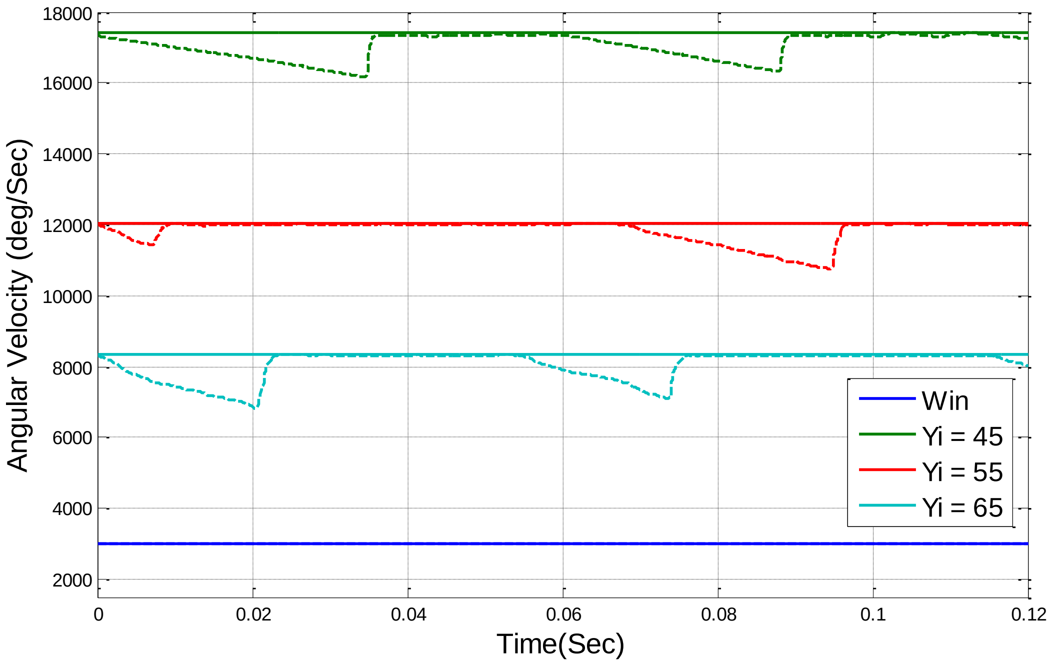

Figure 5-15: Theoretical & simulated angular velocity of the output shaft at (CV & poly Cam profile) (1-5) using Yi= (45, 55, and 65) mm.

5.3.9 Contact Force only solidworks

Figure (5.16) illustrates the contact force at the cam-Follower and Grooved Wheel- Follower contact regions. The results shown in the figure are based on the simulation model for different values of Yi of (45, 55, & 65) mm.

In general, the trend of the contact force for both zones is similar. However, the force amplitude of the cam-follower zone is slightly higher than that at the grooved wheel-follower zone. This might result from the inertia force part on the zones. The fluctuation seen in the curves is probably caused by using a spline projected on both cam profiles (constant velocity and polynomial). It can also be noted that the value of Yi has a small effect on the contact force value.

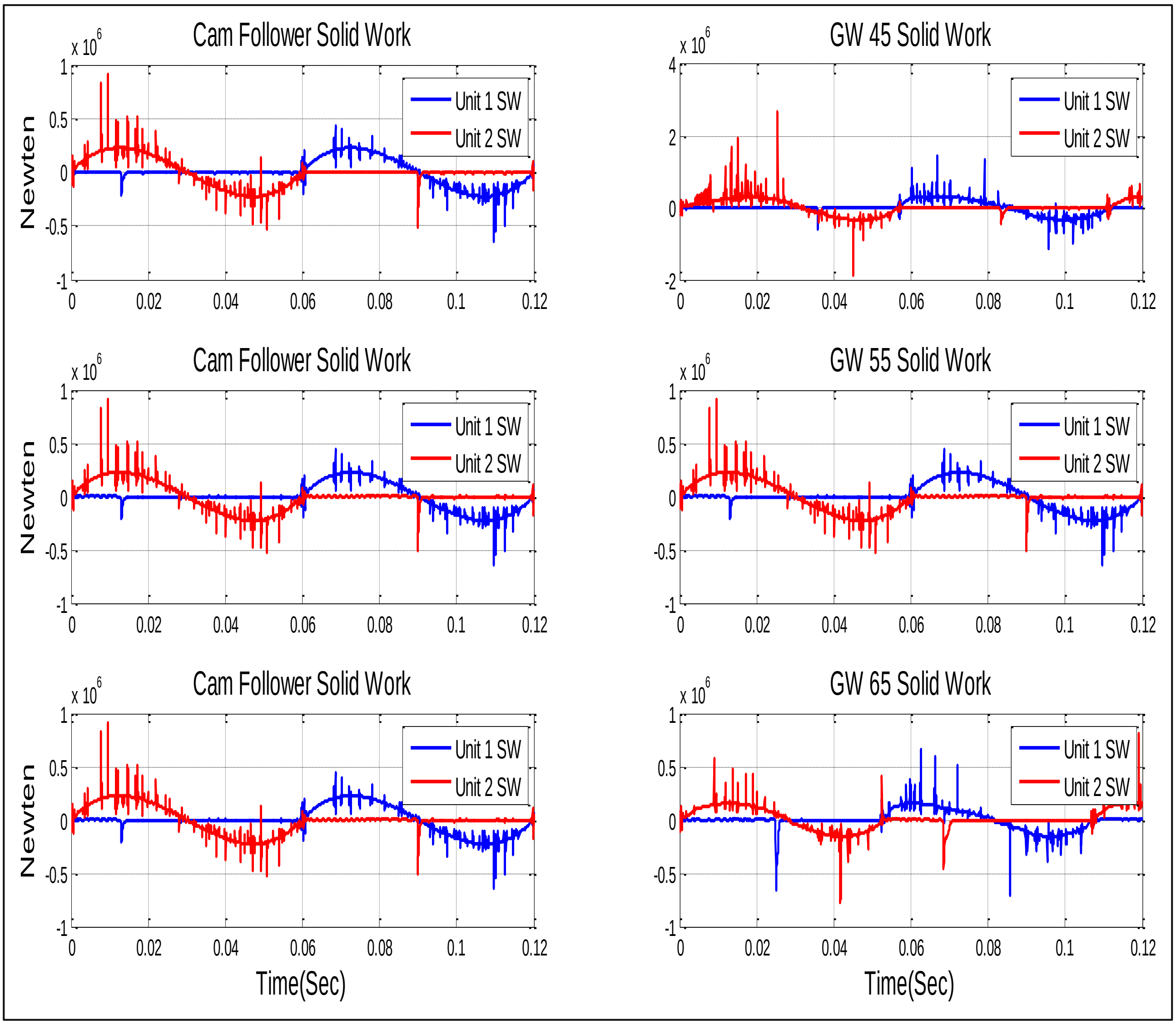

Figure 5-16: Contact force at the contact zones.

5.4 Variable Transmission Ratio

In the previous Section, the results were obtained at different fixed transmission ratios, whereas in this Section, the ratio is changed during operation. This means that the transmission ratio alterable is moved with a particular velocity value (constant velocity is considered in this study). In the next Sections, the results are shown for the different parts of the system except for the cam-follower as it will not be affected by the linear velocity of the alterable.

5.4.1 Angular velocity of the slotted link.

Figure (5.17) illustrates the angular velocity of the slotted link (theoretically & simulation) for two units through three Seconds of the time interval. The movement of the slotted link is carried out from the lower base (Yi= 65 mm) to the upper value (Yi=45 mm) and then to the lower position again (Down Up Down). The typical velocity of the slotted link is taken at 12.5mm/S for both cases, considering the minus sign for the reverse direction.

From the results achieved for this case, The error between theoretical and simulation study of about 1.92% at constant velocity cam profile part. However, at the polynomial cam profile part. The result were matched without any error. The error in the first case might come from installing a flywheel on the output shaft.

For more illustration of the above error, figure (5.18) is used as a focus schematic.

Angular velocity slotted link change (480-600) deg/S these properties take advantage to the device can operate under load without any issues.

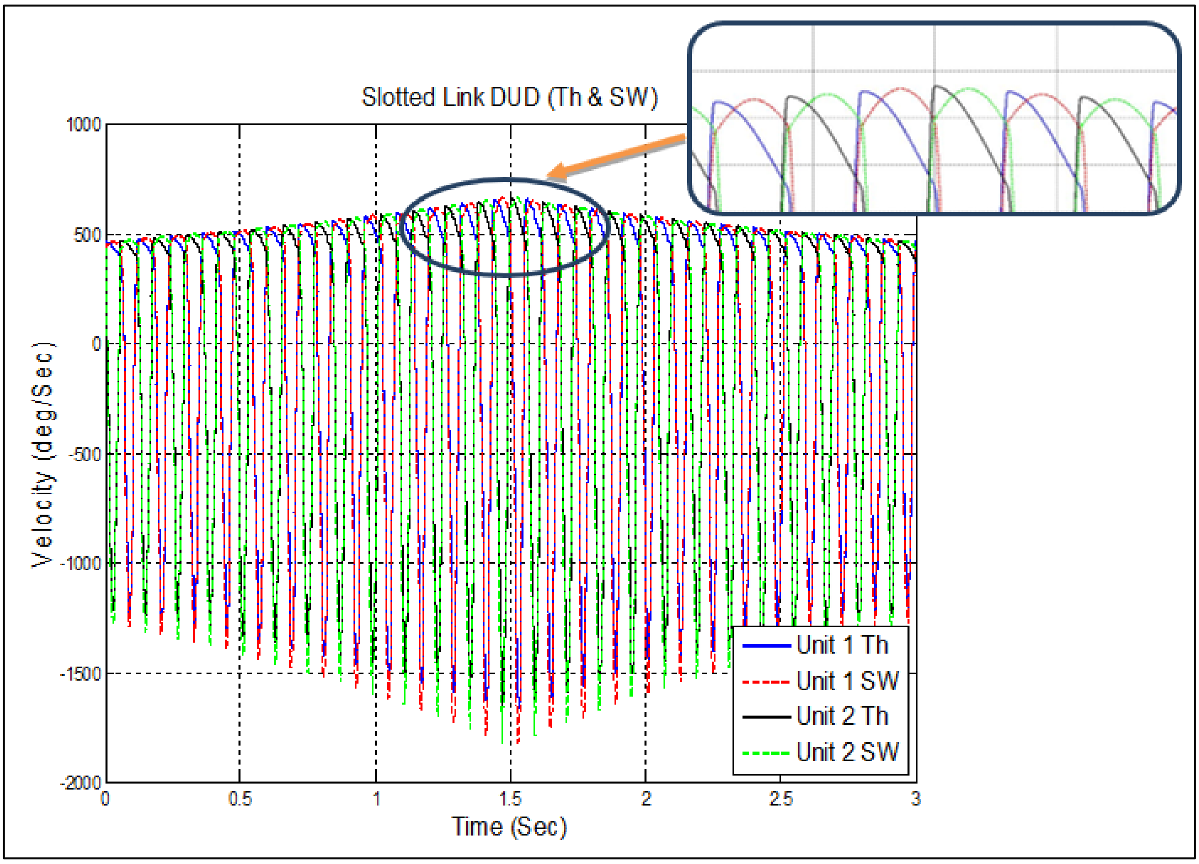

Figure 5-17: Angular velocity of the slotted link for variable transmission ratio.

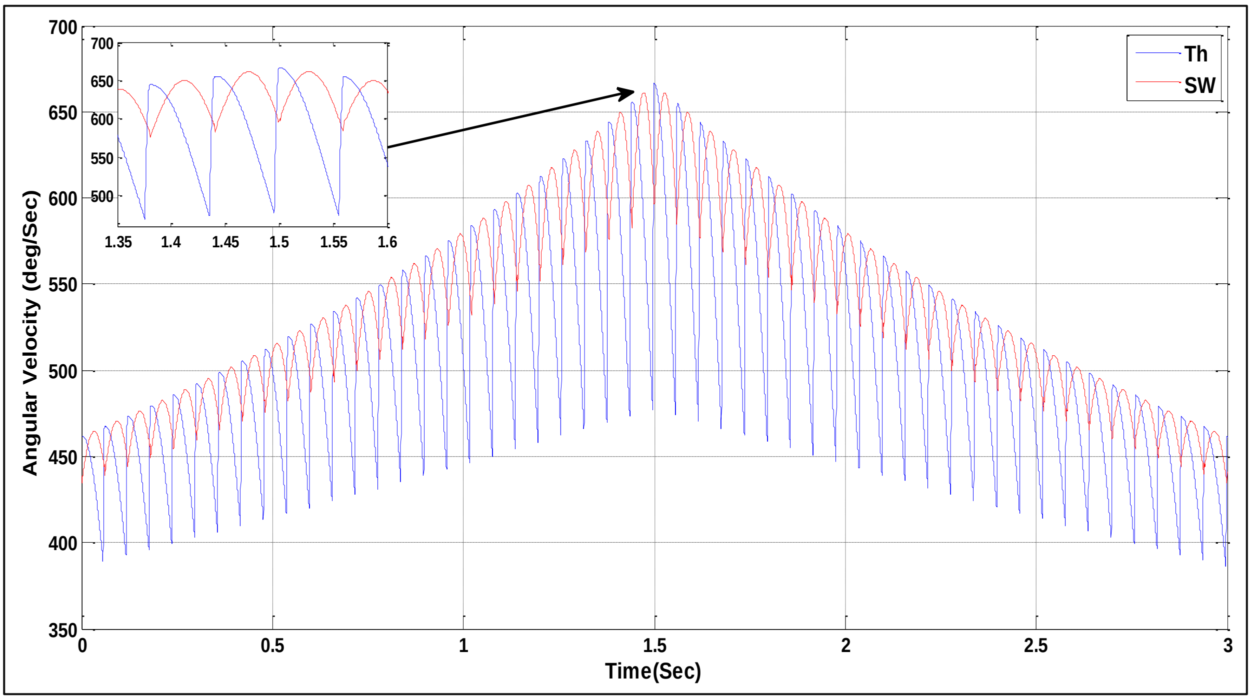

Figure 5-18: Focus schematic max angular velocity slotted link at (CV & poly Cam profile) (1-5) start from Yi=65mm end at Yi=45mm.

5.4.2 Linear velocity Grooved Wheel Follower

Figure (5.19) displays the linear velocity of the Grooved Wheel follower. The dashed line represents the simulation data by SOLIDWORKS, and the solid line represents the theoretical data.

The max-error between the theoretical and simulation results is about 1.3% through the constant velocity stroke. This error might come from the style of mate used in assembling the parts in SOLIDWORKS, in addition to the use of flywheel at the output shaft. However, a perfect match at the stroke of the polynomial profile is achieved between the theoretical and the simulation results without any error.

The discrepancy between the errors is concentrated at the beginning of the constant velocity period for both units. This might be caused from the zero overlap between the two units.

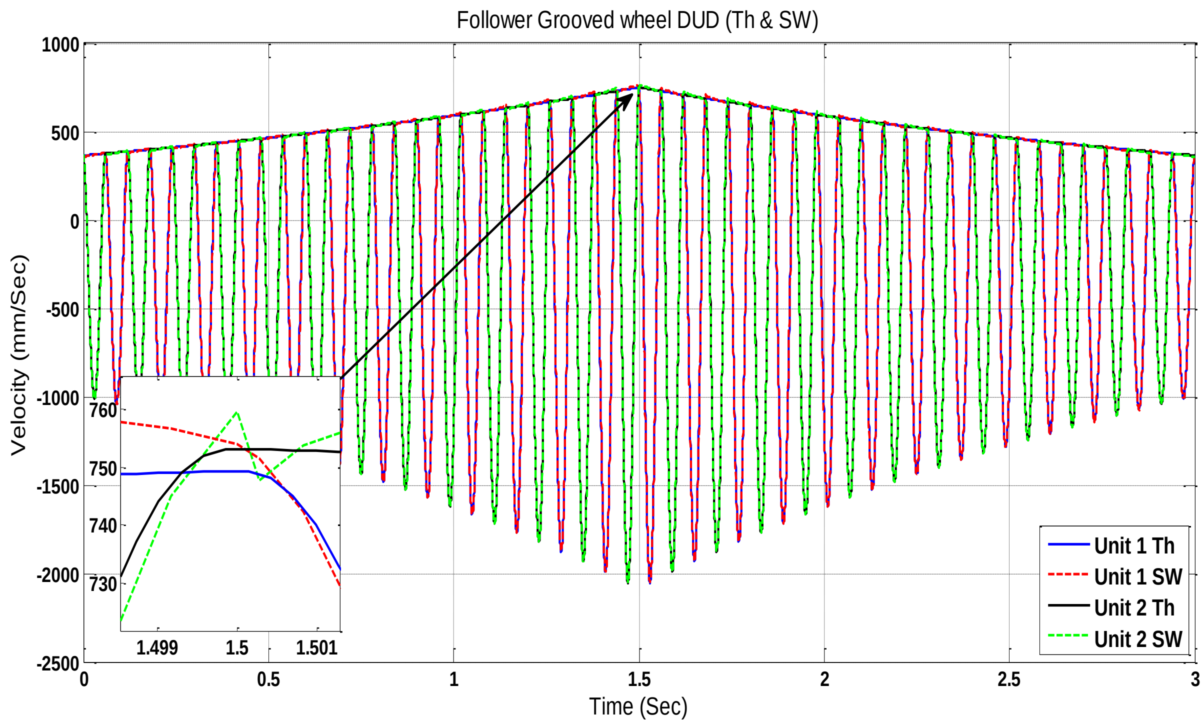

Figure 5-19: Grooved wheel Follower liner velocity for variable transmission ratio.

5.4.3 Linear velocity of the Follower Grooved Wheel

Figure (5.20) illustrates the linear velocity of the Follower Grooved Wheel for variable transmission ratios at different alterable velocities (6.25, 12.5, 18.75 & 25) mm/S.

It is noted from the results that there is an inverse proportion. between the angular velocity (slotted link) and the amount of fluctuation in the linear velocity (grooved wheel follower). This is due to a sudden change in the position of the slotted link axis.

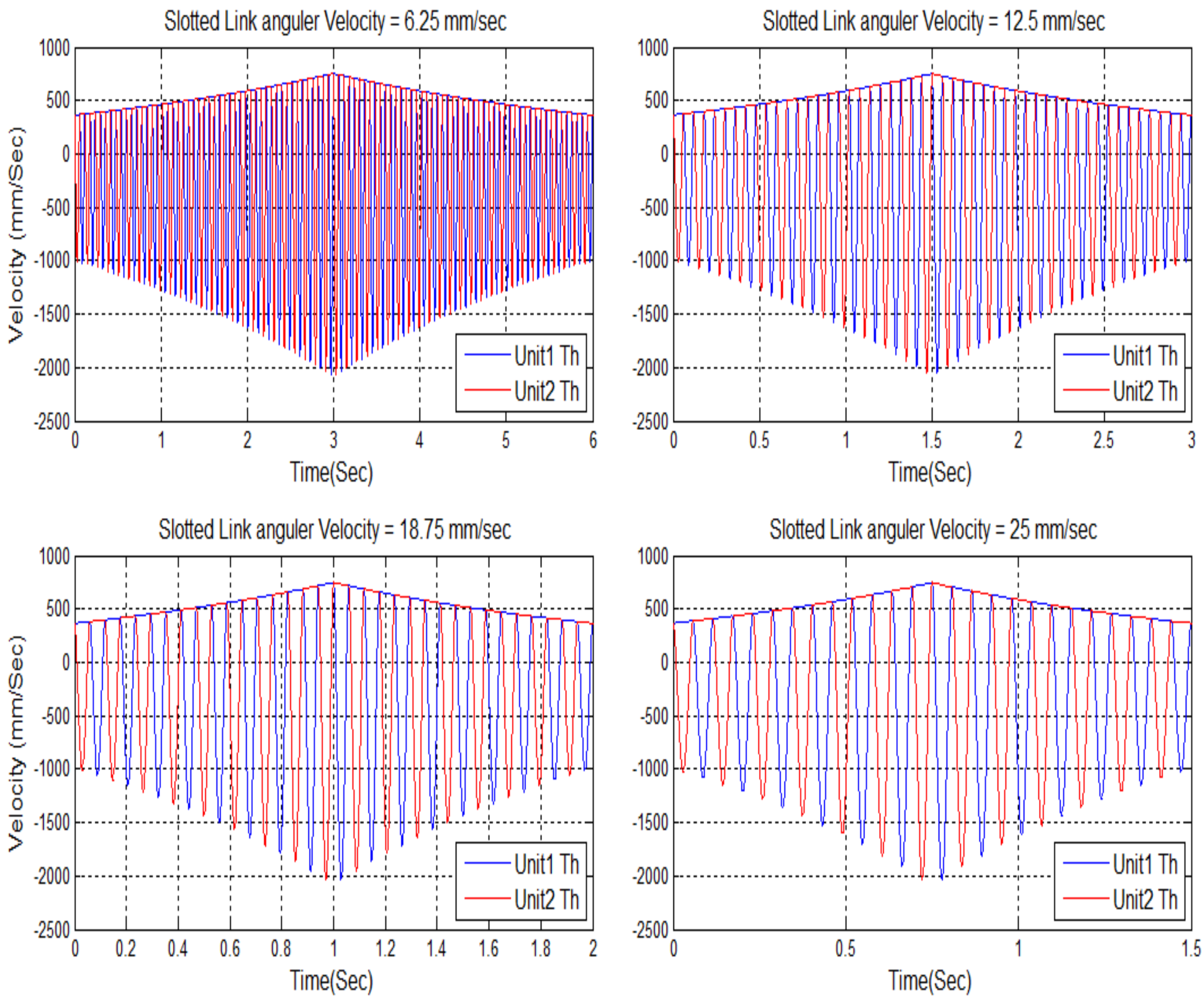

Figure 5-20: Grooved wheel Follower linear velocity at case (DUD) (start from Yi =65mm end in Yi=45mm) & Velocity Slotted Link = (6.25,12.5,18.75&25) mm/S.

Figure 5.21 shows the maximum speed featured in figures 5.20 for the same values of the speed of the Slotted Link of 6.25, 12.5, 18.75 & 25 mm/S.

It is noticed from the figure below, which represents the highest values of linear velocity (Grooved Wheel Follower). As the oscillation value increases with increasing speed (slotted link) and vice versa.

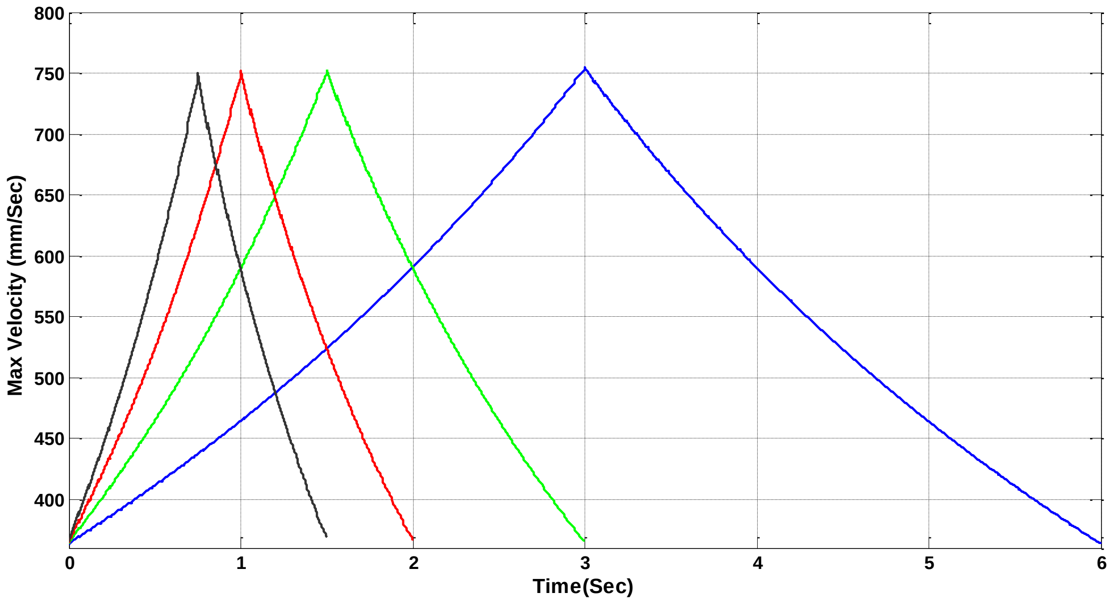

Figure 5-21: Follower Grooved wheel Max Angular velocity at Velocity Slotted Link of 6.25, 12.5, 18.75 & 25 mm/S.

5.4.4 Angular velocity of the Grooved Wheel

Figure (5.22) shows the angular velocity of Grooved Wheel for variable transmission ratio with a uniform alterable speed of 12.5 mm/S. An error of 1% between the theoretical& simulation was observed at the constant velocity period of the cam, which might be caused by using a knife-edge follower. A perfect match between the theoretical and simulation results was obtained during the polynomial period.

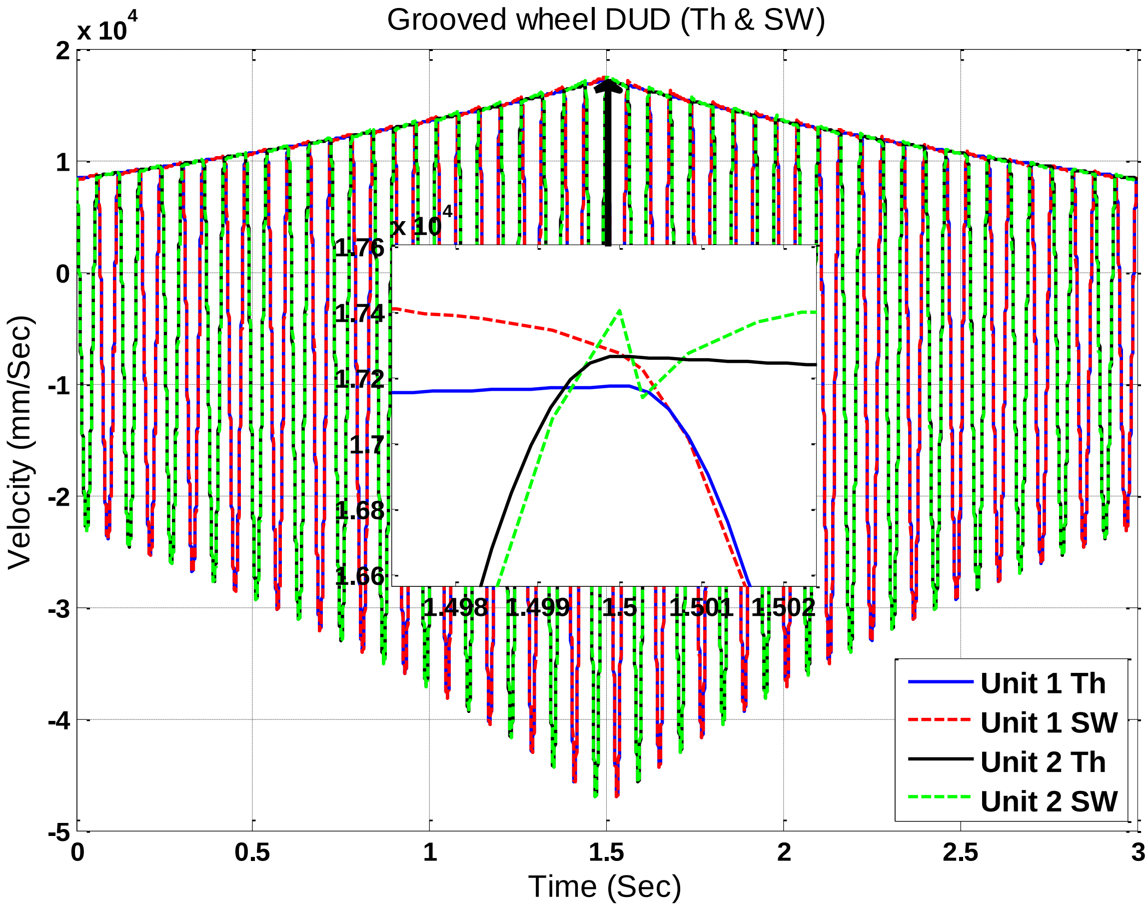

Figure 5-22: Grooved wheel Angular velocity for variable transmission ratio at 12.5 mm/s off alterable velocity

5.4.5 Angular velocity Grooved wheel multi-velocity slotted link

Figure (5.23) illustrates the angular velocity of the Grooved wheel at different values of the slotted link velocity (6.25, 12.5, 18.75 & 25) mm/S.

It was observed that a direct proportion between the slotted link velocity & the value of fluctuation of the output angular velocity. Accordingly, a velocity of 6.25mm/S produces the least fluctuation at the output angular velocity. The direct proportion between max displacement & velocity slotted link this effect on time need to change ratio.

In other words, the number of revolutions of the Grooved wheel is directly proportional to time and inversely proportional to the slotted link speed to reach the desired speed, as shown in Figure (5.24).

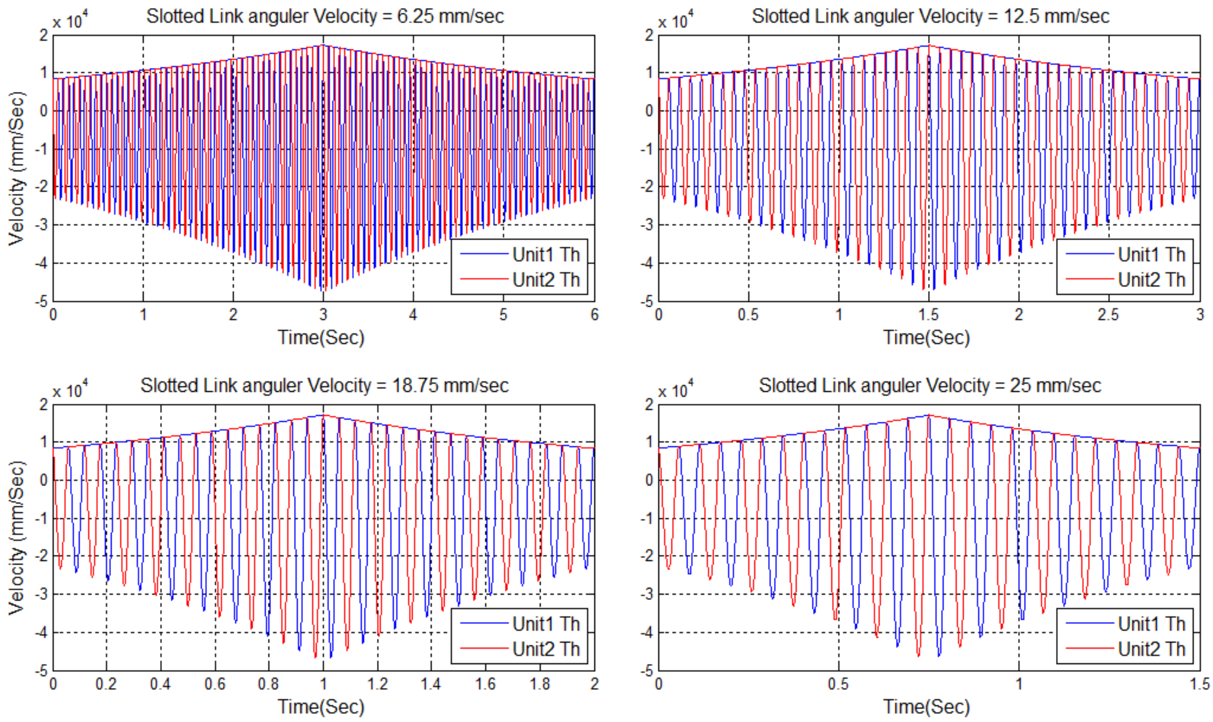

Figure 5-23: Grooved wheel Angular velocity at case (DUD) (start from Yi =65mm end in Yi=45mm) & Velocity Slotted Link = (6.25, 12.5, 18.75 & 25) mm/S.

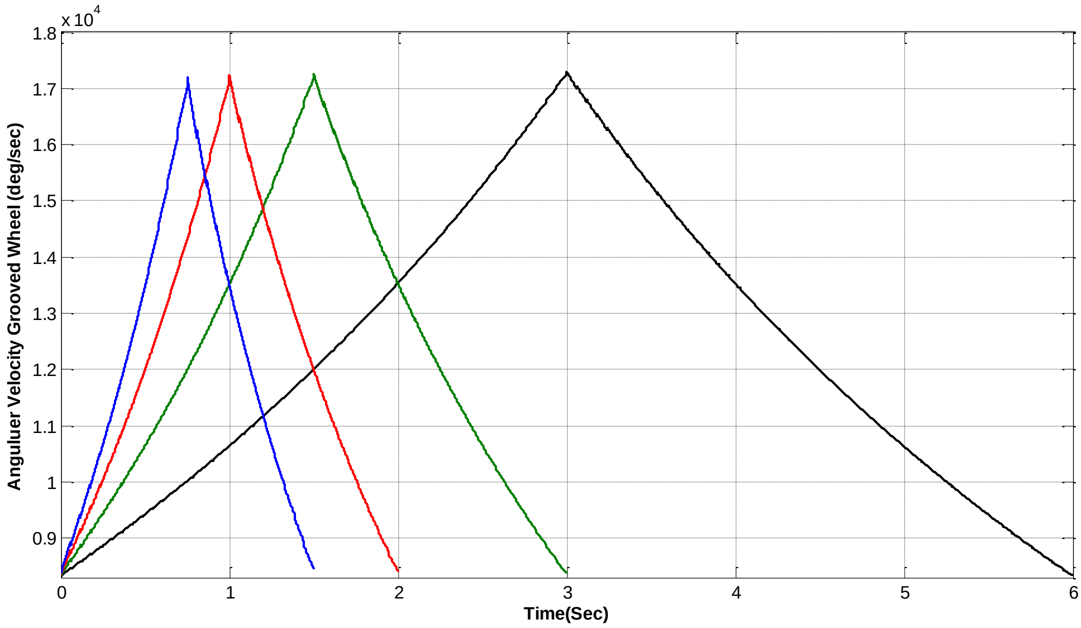

Figure 5-24: Grooved wheel Max Angular velocity for variable transmission ratio and different slotted link velocity
(6.25, 12.5, 18.75 & 25) mm/S.

5.4.6 Angular Acceleration of the Grooved wheel at the multi-velocity slotted link

Figure (5.25) shows the theoretical results of the value of the acceleration of the Grooved wheel at different values of the slotted link velocity (6.25, 12.5, 18.75 & 25) mm/S. It can note that when increasing the velocity of the slotted link the value of the acceleration is increased too (directly proportional). At Vel 6.25 mm/S the Acc reaches 40 deg/S^2, while at Vel 25 mm/S the Acc reaches 150 deg/S^2, Figure (5.25) illustrate the Max value of Acc show in figure (5.26)

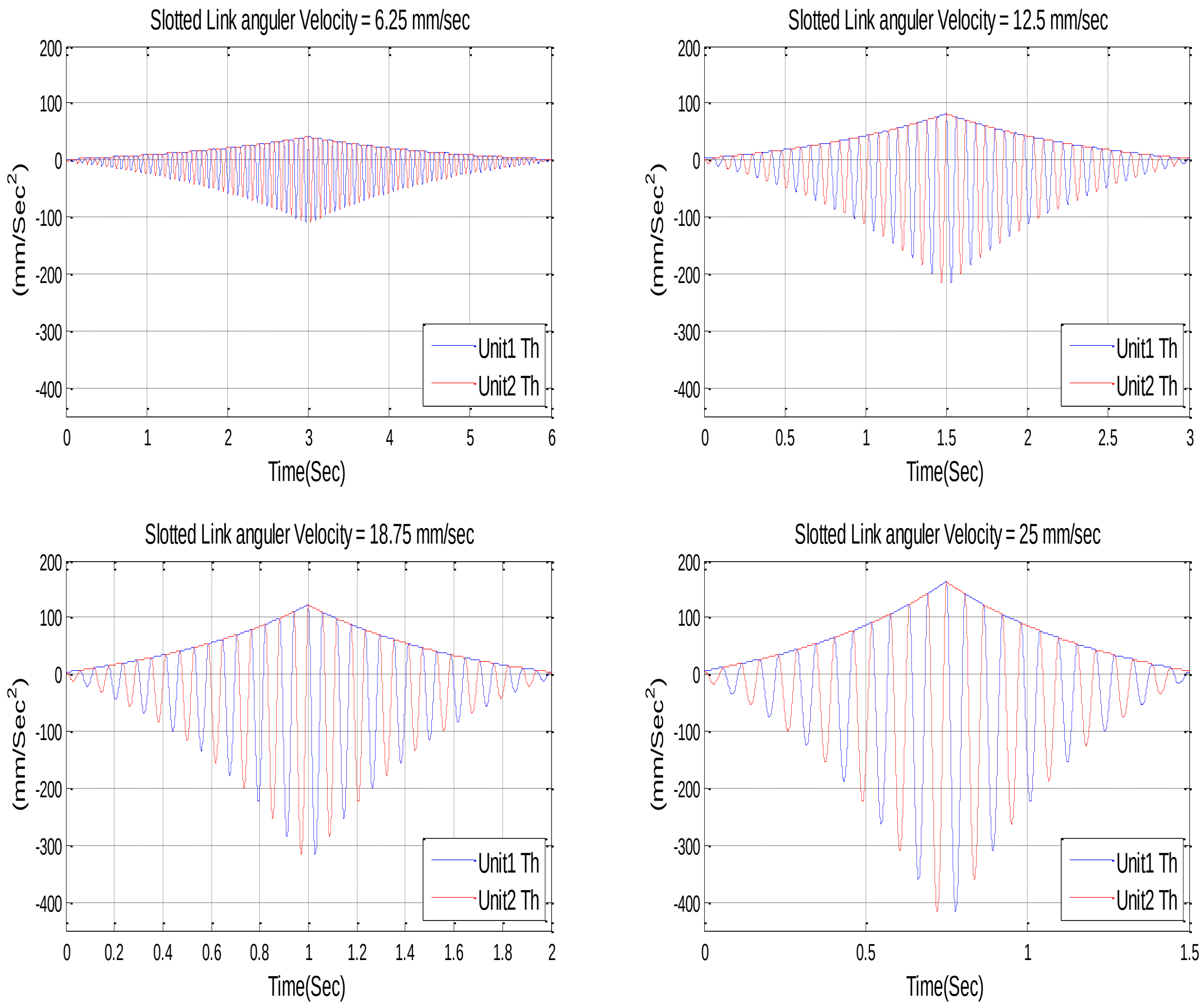

Figure 5-25: Acceleration at a variable velocity of Slotted link.

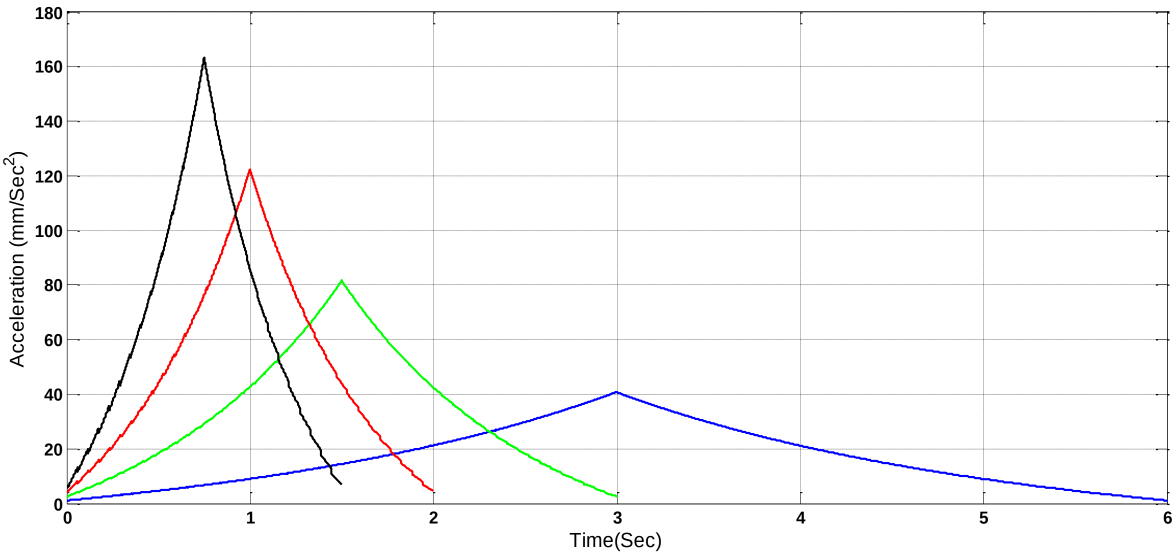

Figure 5-26: The Maximum angular Acceleration of the Grooved wheel for different alterable velocities (6.25, 12.5, 18.75 & 25) mm/S.

5.4.7 Output angular velocity

Figure (5.27) shows a comparison between theoretical and simulated output angular velocity at slotted link velocity of,

It can be seen that as the Yo value is increased, it leads to an increase in the output velocity.

It was found that there is a max error between the theoretical and the simulation results of about 19.3%. This error might result from using a ratchet connecting the output shaft and the Grooved Wheel. The ratchet is responsible for transforming the reciprocating rotational movement into a one-way rotational motion at the output shaft, and therefore, it cannot be dispensed.

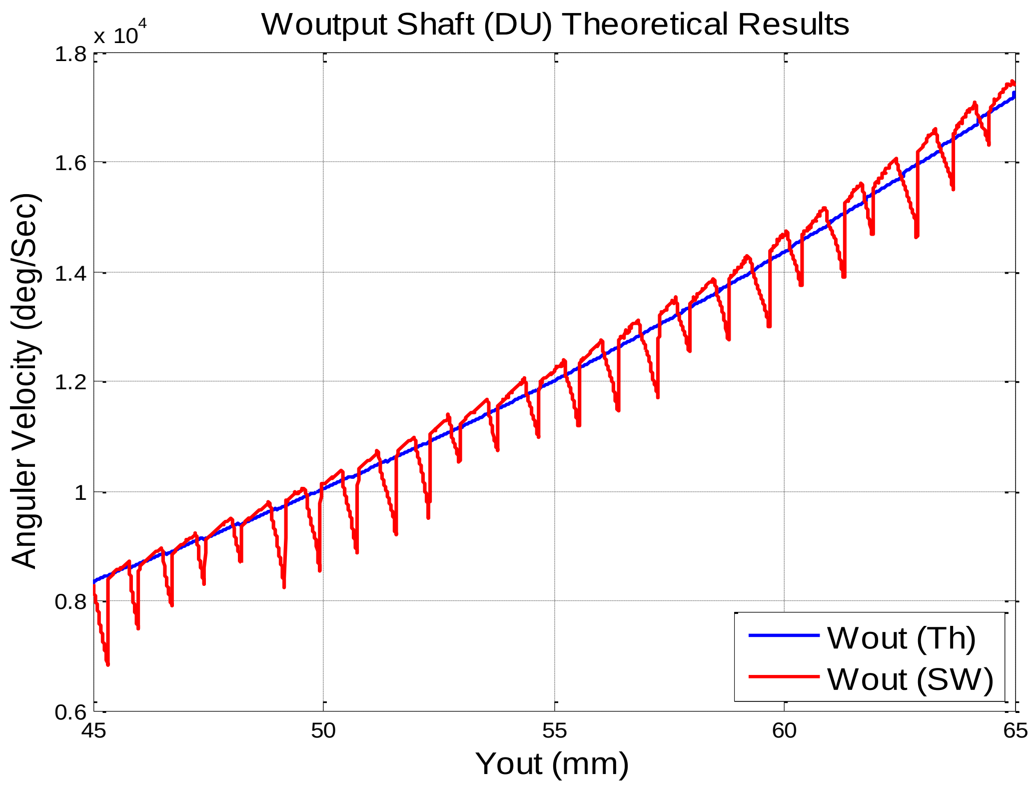

Figure 5-27: Angular velocity output shaft at case DU From Yi=65mmto Yi=45mm.

Figure (5.28) shows the theoretical result of the relationship between the Yo/ Y ratio and the value of the transmission velocity (Wo/Wi). Results were calculated at Yo=0-110 mm and Y=110 mm. The slotted link moves with a uniform velocity of 12.5 mm.

Noting that when Yo value = 0, the output shaft angular velocity is also zero. When Yo reaches 95% of its maximum position, the transmission ratio approaches infinity. This feature, in addition to continuous smooth function, puts the device under the category of the infinitely variable transmission (IVT).

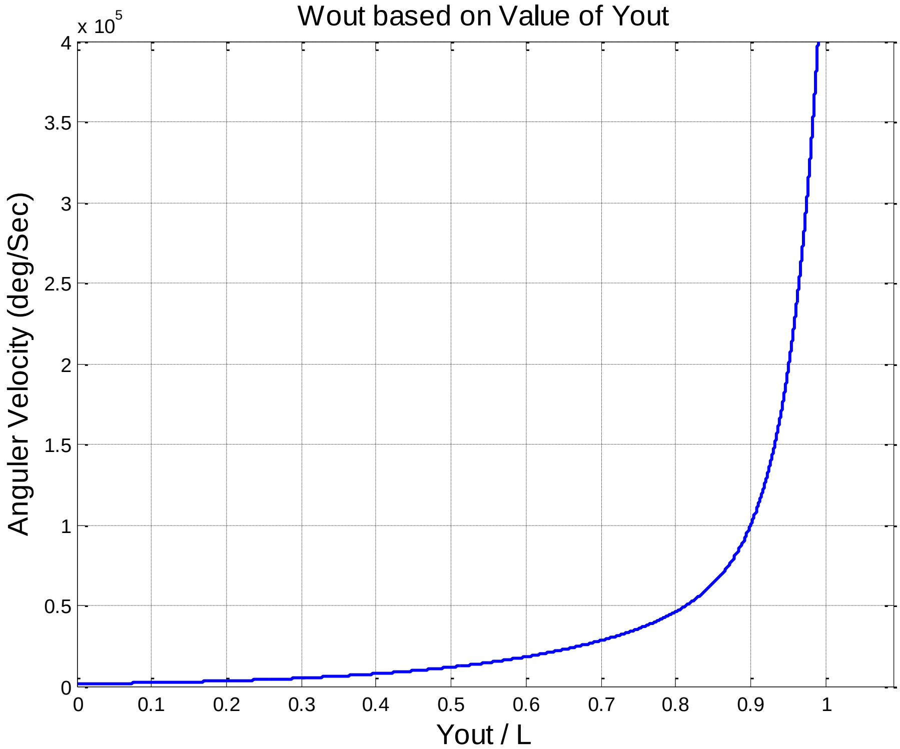

Figure 5-28: output transmission ratio against the alterable position.

5.4.8 Contact force at different values of slotted link velocity

Figure (5.29) shows the simulation results of the contact forces at cam-follower and Grooved wheel-follower x) contact zones for uniform slotted link velocity of (6.25 mm/S, 12.5 mm/S & 18.75 mm/S)

It can be noted that the minimum value of the contact forces is at a speed of 6.25 mm/S. In addition, the shape of the contact forces curve takes the same shape as the cam follower acceleration curve because the acceleration is directly proportional to the dynamic forces transferred between the parts. The presence of a high-value force (noise) might be caused by using aknife-edge cam follower. An approximately equal contact force for both zones can be observed at 6.25 mm/s because minimum acceleration is achieved in this case, which reduces the inertia forces.

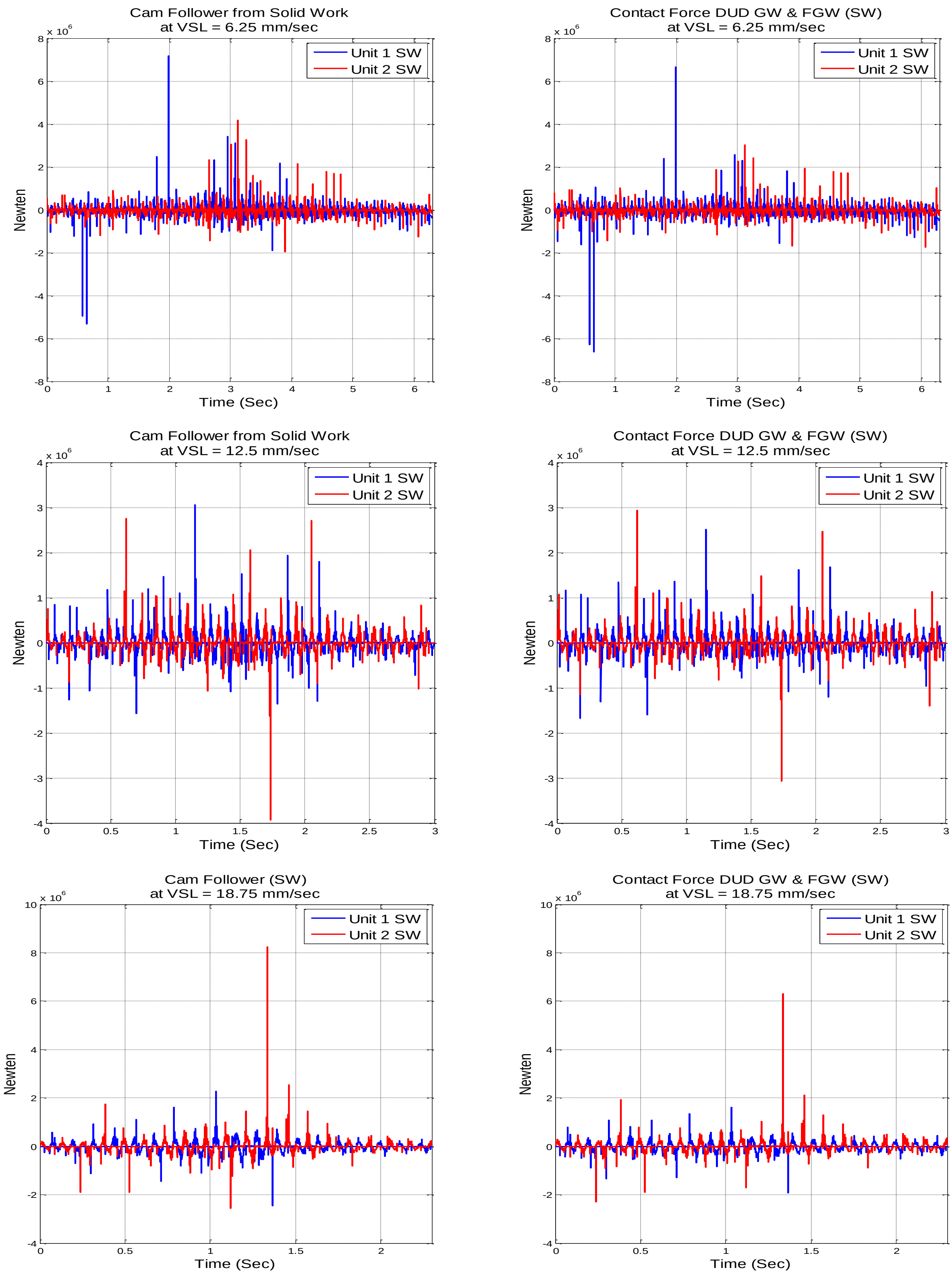

Figure 5-29: Contact force at different values of slotted link velocity at (6.25, 12.5 & 18.75) mm/S.

6 Chapter Six: Conclusions and Future Works

6.1 Conclusions and Remarks

In this work, a theoretical analysis and simulation model was established and applied to a recently developed cam based infinitely variable transmission system (IVT). The system contains two identical units, at least for transmitting continuous power. Each of these units contains cam, cam-Follower, slotted link, follower grooved wheel, grooved wheel, and ratchet. From the analysis and results in this work, the following conclusions can be drawn:

1. In the ratcheting type CVT and IVT, including the system under consideration, they should have multi-units to ensure transmitting continuous power.

2. In order to achieve regular output rotational speed for uniform input, a constant velocity cam profile should be developed, especially for the outstroke of the cam in each unit. Using a constant velocity cam profile for both strokes (outward and return) produce infinite acceleration at the points of changing velocity. This will lead to high forces and stresses in the system, especially at the contact zones. Developing a fifth-degree polynomial cam profile for the return stroke introduce a smooth change in the velocity of the cam-follower. This led to a smooth change in the acceleration and then the forces and stresses within the system.

3. A cam follower of the knife-edge on one side and round on the other side gives acceptable results in terms of the geometrical design of the system.

4. Theoretically, the IVT mechanism creates regular output velocity for regular input at fixed transmission ratios. However, little fluctuation of results appeared in the simulation results.

5. The vertical position and velocity of the slotted link control the transmission ratio of the system, and this needs to be studied carefully.

6. For the grooved wheel profile, constant velocity type is only required. This is due to that the return stroke. The wheels are disconnected from the output shaft via the ratchet.

7. The groove in the grooved wheel can cover more than 2π angular displacement depending on the maximum transmission ratio required. In this study, $8\ \pi$ angular displacement was considered

8. The ratchet is the main part of these types of IVT systems. Its type and features directly affect the performance of the system. Fluctuation of results might be produced from the ratchet used.

9. flywheel might be required in these types of IVT to reduce the fluctuation of speed at the output shaft.

6.2 Recommendations for Future works

1. A comprehensive study based on the field of Multibody Dynamic (MBD) can lead to the theoretical appreciation of the kinematics and kinetics for these types of systems.

2. To obtain optimal performance, a detailed study on the type and design of the ratchet is required.

3. The geometry of the follower ends need to be studied carefully, that will affect the dynamics of the system and the stresses at the contacting zones.

4. Studying the effect of the number of units on the performance and design of the system.

5. Studying the effect of overlap of strokes between the units in the system, where this aspect is similar to the contact ratio in toothed gearing analysis.

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Appendix

calculate dimension of ratchet pawl

Z=number of rachet teeth. For independent type ratchet =(12to 20) teeth.

M_b =ph

p = force apply by ratchet

h = tooth Hight

a= tooth thickness

m= module

m=$\frac{D}{Z}$

D= diameter of ratchet.

$\Psi = \frac{b}{m}$ = ratio range (1.5 to 3.0)

b = thickness of ratchet.

On this thesis (z=20 tooth, D=130 mm)

m=$\frac{130}{20}$=6.5

p= $\frac{2m_t}{D}$

or if we stress of steel (300 to 500) kgf/cm^2

m=2$\sqrt[3]{\frac{m_t}{z\Psi(\sigma_b)}}$

calculation dimension of pawl.

Dimeter of pin pawl

D=$2.71\sqrt[3]{\frac{p}{2(\sigma_b)}} * \left(\frac{b}{2} + a_1\right)$

For better sliding of pawl on ratchet

$\phi > \rho$

Where

ϕ= ratchet cog angle

ρ= friction angle =$\tan^{-1}\mu$ (specification of metal).

الخلاصة

جهاز نقل الحركة بنسب لامتناهية (IVT) .هو منظومة تنقل السرعة الدورانية بين محوري الأجزاء الدوراة مع إمكانية الحصول على تدرج في نسب السرع .كذالك ممكن الحصول على سرع محايدة (سرعة صفرية).

في هذه الرسالة تم دراسة وتصميم ابعاد الأجزاء وكذالك تم تجميعها .وبعدها تم تطوير الجهاز ليتكون من مجموعتين متشابهتين :كل مجموعة تتكون من حدبة ،تابع حدبة ،محور محزز،تابع عجلة محززة ،عجلة محززة وسقاطة .

حسابات التحليل الحركي تم اجرائها نظريا للمنظومة .اما المحاكاة تمت بواسطة برنامج (SOLIDWORKS 2018) والذي يعتبر قريب للواقع العملي .شكل وابعاد الحدبة تم اختيارها وتصميم وذالك للانه يعتبر تصميمها المهمة الأساسية لهذه الرسالة .وللسبب المذكور تم اختيار شكل الحدبة ليكون (ثابتة السرعة في شوط العمل و(poly 1-5) في شوط الرجوع .

بالإضافة الى الحسابات النظرية تم حساب المحاكاة لكل أجزاء الجهاز .كل جزء في الجهاز منفصلا .وذالك للاجراء مقارنة بين نتائج النظرية والمحاكاة .ولضمان صحة ودقة كلا النتائج المستحصلة .اما النتائج النظرية تم اجرائها لكل الجهاز .حيث تم تم اجرائها عند الحالتين :الحالة الثابتة لمحور دوران المحور المحزز،الحالة المتغيرة للحصول على نسب متغير وكما ستوضح ادناه .الحالة الأولى المسافة بين محور دوران المحور المحزز ومسار الحركة الخطية لتابع حدبة والتي تسمى (Yi) تكون عند القيم التالية (65,55,45) ملم .النتائج تبين تقارب بين النتائج النظرية المحاكاة .اما الحالة الثانية التي تم اخذ النتائج النظرية والمحاكاة أيضا عندها كانت عند تغير مستمر لقيمة (Yi) كذالك النتائج المستحصلة من الحسابات النظرية المحاكاة جيد وبنسبة خطاء 2%

بالإضافة الى النتائج أعلاه تم حساب القوى في منطقة الاتصال التالية (الحدبة وتابع الحدبة ،القرص المحززوتابع القرص المحزز) .هذه الحسابات تم فقط بواسطة المحاكاة بواسطة برنامج (SOLIDWORKS 2018) .وكذالك عند الحالات المذكورة أعلاه لقيمة (Yi).

جمهورية العراق

وزارة التعليم العالي والبحث العلمي

جامعة كربلاء- كلية الهندسة

قسم الهندسة الميكانيكية

دراسة وتطوير ناقل الحركة المتغير بلا حدود يعتمد على مبدأ الكامة

رسالة مقدمة الى قسم الهندسة الميكانيكية - جامعة كربلاء كجزء من متطلبات نيل درجة الماجستير في الهندسة الميكانيكية

(ميكانيك تطبيقي)

من قبل

مهيمن فالح حمد

بكالوريوس في علوم الهندسة الميكانيكية لسنة 2009

بإشراف

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