ANALYSIS OF CRANKSHAFT MECHANISM USING THE COMPLEX NUMBER METHOD WITH MATLAB PROGRAMMING

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Abstract. *This study aims to analyze the kinematic characteristics of a beverage can pressing mechanism modeled as an offset crank–slider mechanism using the complex number method. This method is applied to simplify the mathematical representation of position, velocity, and acceleration vectors of each mechanism component within a unified formulation. The analysis is conducted through mathematical modeling and MATLAB based computational simulation using specific geometric parameters and angular velocity that represent the operating conditions of the pressing device. The results show that the complex number method provides efficient and accurate kinematic calculations, particularly in determining the magnitude and direction of velocity and acceleration at critical points of the mechanism. The influence of the offset configuration on the increase in acceleration values and changes in the direction of component motion can also be clearly identified. This study demonstrates that the integration of the complex number method and MATLAB programming is highly effective for analyzing and evaluating the performance of pressing mechanisms and can serve as a reference for the design and development of similar mechanisms in the field of mechanical engineering.*

Keywords: *Beverage Can Pressing Device; Crank Mechanism; Complex Number Method; Kinematics; MATLAB.*

1. INTRODUCTION

The development of technology in the field of mechanical engineering continues to advance rapidly, particularly in moving mechanism systems that require high efficiency and analytical precision to support machine performance. Increasing demands in the manufacturing, automotive, and energy conversion machinery industries emphasize the importance of mechanical motion analysis that is more detailed, faster, and capable of describing mechanism behavior in real time (Nampira et al., 2025). One of the important mechanisms widely applied in various engineering applications is the crankshaft mechanism, which serves as a primary component in converting reciprocating motion into continuous rotational motion (Iski et al., 2022).

The crank mechanism exhibits complex motion characteristics due to the interaction among the crank, connecting rod, and slider. This complexity results in nonlinear relationships that are difficult to analyze using conventional methods such as graphical approaches or basic geometric analysis. When applied to high performance systems, even small errors in position, velocity, and acceleration analysis can lead to force imbalance, increased vibration, inefficient energy consumption, and reduced component lifespan. Therefore, an analytical method is required that can accurately and systematically describe the motion relationships among components and can be effectively implemented within computational systems.

Several previous studies have attempted to develop analytical methods for crank mechanism analysis. (Torres et al., n.d.) conducted a crank–slider mechanism analysis using a two dimensional vector approach to determine the position and velocity of the crank. Meanwhile, (Chavan & Shinde, 2021) in their study entitled *"Design & Analysis of Crankshaft for Single Cylinder Diesel Engine,"* applied crank–slider motion simulation using MATLAB based numerical calculations. These studies provide important contributions to the kinematic analysis of mechanisms; however, they remain limited to conventional trigonometric approaches and have not fully integrated the complex number method into a unified mathematical representation.

Nevertheless, most existing studies in the literature still focus on conventional crank mechanisms with a parallel axis configuration between the crank and the slider. In contrast, studies addressing crank mechanisms with offset configurations remain very limited. In fact, offset mechanisms have a significant influence on kinematic parameters such as maximum piston acceleration, inertia forces, and overall mechanism efficiency. This condition indicates the need for the development of a more comprehensive analytical model to accurately describe the kinematic phenomena of offset crank mechanisms.

In line with this, the present study focuses on the kinematic analysis of a beverage can pressing mechanism modeled as an offset crank–slider mechanism using the complex number method. The analysis is conducted to determine the position, velocity, and acceleration characteristics of each mechanism component through mathematical modeling and MATLAB based numerical simulation, serving as a basis for performance evaluation and design development of the pressing device.

The complex number method is selected because it is capable of representing the mechanism in the form of complex vectors, which facilitates the calculation of position, velocity, and acceleration through simple mathematical operations such as rotation and differentiation. This approach makes the mechanism analysis more concise, structured, and mathematically consistent. Furthermore, the complex number method is highly suitable for implementation in MATLAB programming, as the calculations can be performed automatically and the results can be visualized in the form of graphs and mechanism motion animations.

2. LITERATURE REVIEW

2.1 Slider Crank Mechanism

The crank mechanism, or slider–crank mechanism, is one of the most commonly used types of four bar linkage in mechanical systems. This mechanism is widely applied in various mechanical devices to generate or transform specific types of motion. In general, a slider crank mechanism consists of several interconnected links that move relative to one another through joints (Wahyu Aditama Y et al., 2017). According to (Norton, 2020), this mechanism is a special form of a four bar system in which one link is replaced by a sliding element (slider), thereby enabling the conversion of rotational motion into reciprocating translational motion, or vice versa.

In internal combustion engines, the crankshaft serves as the primary element that converts the energy generated from combustion into rotational motion. The combustion pressure acting on the piston causes the connecting rod to transmit the force to the crankshaft, producing torque that is used to drive other components, such as vehicle wheels (Heywood, 2019). Conversely, in applications such as reciprocating pumps and compressors, rotational energy from an electric motor is converted into translational motion to draw in or compress fluids (Juvinall & Marshek, 2017).

Structurally, this mechanism consists of four main elements: the crank with length (*r)* that rotates about the fixed axis O_2, the connecting rod with length (l) that transmits force between the crank and the slider, the slider (piston) that undergoes translational motion within the cylinder, and the ground frame, which serves as a reference and constrains the system to operate within a specified plane. The geometric relationships among the mechanism elements can be expressed through the following vector loop equation:

$$r(\cos\theta + i\sin\theta) + l(\cos\beta + i\sin\beta) = x \quad .....(1)$$

Where r represents the crank length, l the connecting rod length, θ the crank angle, β the connecting rod angle, and x the translational position of the slider. By separating the imaginary components, the relationship between the translational position of the slider and the crank angle is obtained as follows:

$$x = r\cos\theta + \sqrt{l^2 - (r\sin\theta)^2} \quad .....(2)$$

Equation (2) is used to determine the linear position of the piston with respect to the crankshaft rotation angle. By differentiating this equation with respect to time, the linear velocity and linear acceleration of the slider are obtained as follows:

$$v = \frac{dx}{dt} = -r\omega_1 \sin\theta - \frac{r^2\omega_1 \sin\theta\cos\theta}{\sqrt{l^2 - (r\sin\theta)^2}} \quad .....(3)$$

$$a = \frac{d^2x}{dt^2} = -ra_1\sin\theta - r\omega_1^2\cos\theta - \frac{r^2 r\omega_1^2(\cos\theta - \sin^2\theta)}{l^2 - (r\sin\theta)^2} \quad .....(4)$$

Equations (3) and (4) form the basis of kinematic analysis for determining the velocity and acceleration of the piston in a crank mechanism system. In addition to the basic configuration, this mechanism has several inversions, for example when the connecting rod is considered as the fixed link, or in the offset slider–crank configuration where the slider path is not parallel to the crankshaft axis. These variations result in different motion characteristics and are widely applied in quick return mechanisms and high speed motion transmission systems (Norton, 2020).

2.2 Complex Number Method

The complex number method is a mathematical approach used in the kinematic analysis of planar mechanisms. This approach allows the calculation of position, velocity, and acceleration vectors to be performed in a simpler and more efficient manner (Yogi Saputra et al., 2022). In this method, the *y*-axis of the planar mechanism is treated as the imaginary axis, enabling kinematic relationships to be expressed more concisely within a single complex number equation. Consequently, a planar vector is no longer decomposed into *x* and *y* components but is instead represented as a complex number in exponential or polar form.

$$r = Re^{i\theta} \quad .....(5)$$

where:
R = denotes the magnitude (vector length)
θ = represents the orientation angle of the vector with respect to the reference axis
$i = \sqrt{-1}$

In kinematic analysis, the complex number method is used to determine the primary motion parameters of a mechanism, which include the position of each element expressed as vectors, velocity as the first time derivative of position, and acceleration obtained from the second time derivative. For example, (Arman et al., 2020) in their study entitled *"Kinematic Simulation Analysis of a Sawing Machine Using the Complex Number Method,"* applied the complex number method to calculate the position, velocity, and acceleration of each link in the mechanism. The following presents the kinematic formulation using the complex number method:

Vector Position:

$$r = Re^{i\theta} \quad .....(6)$$

a. Velocity is obtained as the first time derivative of position:

$$v = \frac{dr}{dt} = \frac{d}{dt}(re^{i\theta}) = \dot{r}e^{i\theta} + ir\dot{\theta}e^{i\theta} \quad .....(7)$$

If ω = θ represents a constant angular velocity, then:

$$v = ir\omega e^{i\theta} = r\omega e^{i\theta(\theta+\pi/2)} \quad .....(8)$$

a. Acceleration is the second time derivative of position:

$$a = \frac{d^2r}{dt^2} = \frac{d}{dt}(ir\omega e^{i\theta}) \quad .....(9)$$

If θ = ω (constant) and θ = α (angular acceleration), then:

$$a = ir\alpha e^{i\theta} - r\omega^2 e^{i\theta} = -r\omega^2 e^{i\theta} - r\alpha e^{i(\theta+\pi/2)} \quad .....(10)$$

The main advantage of this method lies in its ability to simplify mathematical formulations in kinematic analysis, allowing the calculation of position, velocity, and acceleration to be carried out in a more systematic manner. In addition, another advantage of the complex number method is that it facilitates the development of computational algorithms and their implementation using programming languages (Nazaruddin & Fatra, 2023).

2.3 MATLAB Programming in Kinematic Analysis

MATLAB (Matrix Laboratory) is a numerical analysis and computation software that functions as an advanced mathematical programming language developed based on the properties and structure of matrices (Cahyono, 2016). As a commercial product developed by MathWorks, Inc., MATLAB has become a highly useful application for performing various linear algebra operations and other mathematical computations. The software is also equipped with a wide range of built in functions that facilitate the solution of matrix based numerical operations, which are generally difficult to perform manually (Fatwa et al., 2022).

MATLAB is capable of handling matrix operations, solving nonlinear equations numerically, performing numerical differentiation, and providing visualization and animation facilities. The combination of these capabilities makes MATLAB a highly effective tool for analyzing position, velocity, and acceleration in mechanisms (Wahyu Aditama Y et al., 2017). In the kinematic analysis of slider–crank mechanisms, MATLAB is often used to solve systems of nonlinear equations derived from the vector loop equation formulation. The *fsolve* function in MATLAB is employed to obtain numerical solutions for the connecting rod angle and slider position based on the crank rotation angle using iterative methods such as Newton–Raphson.

(Laribi, 2023) in a journal article entitled *"Optimal Synthesis of a Planar Mechanism Using MATLAB: Example of Slider–Crank in a Rowing Motion,"* employed the complex number method fully implemented through MATLAB programming to analyze and optimize a slider–crank mechanism. In that study, the kinematic formulation and optimization process were expressed in the complex number domain, resulting in a simpler and more structured computational procedure. Furthermore, all stages of the analysis, from parameter determination and motion calculation to visual simulation, were performed using MATLAB, producing a comprehensive and efficient simulation of the mechanism motion.

3. RESEARCH METHODS

This study is a quantitative study with a descriptive analytical approach that employs computational methods through mathematical modeling and MATLAB based program simulation. The complex number method is applied to analyze the kinematics of the driving crank mechanism, including the calculation of position, velocity, and acceleration of each mechanism component. The research stages are conducted systematically to ensure that the simulation results can be easily validated and replicated.

The research begins with the determination of parameters and the design of an offset slider–crank mechanism model based on the system configuration. Subsequently, the kinematic equations for position, velocity, acceleration, and the connecting rod angle are derived using the complex number method. The obtained equations are then implemented in the form of MATLAB scripts to perform calculations and visualize the simulation results. If the program does not operate correctly, a debugging process is carried out until the expected results are obtained. Finally, the simulation results in the form of position, velocity, and acceleration graphs are analyzed to identify the motion characteristics of the investigated mechanism. The complete research workflow is illustrated in **Figure 1**.

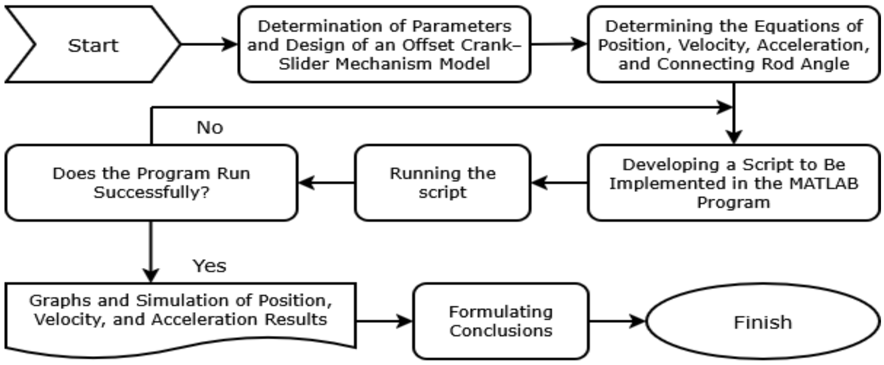

Figure 1. Research Flowchart

4. RESULTS AND DISCUSSION

The mechanism analyzed in this study represents a model of a beverage can pressing machine mechanism. The analysis is carried out using the complex number method to mathematically describe the kinematic relationships among the components of the mechanism. At this stage, the model design is not based on a physical object or actual machine components, but rather on numerical data and geometric parameters that represent the mechanism. The parameters used in this analysis are presented in **Figure 2**, which illustrates the dimensions and basic configuration of the analyzed mechanical system.

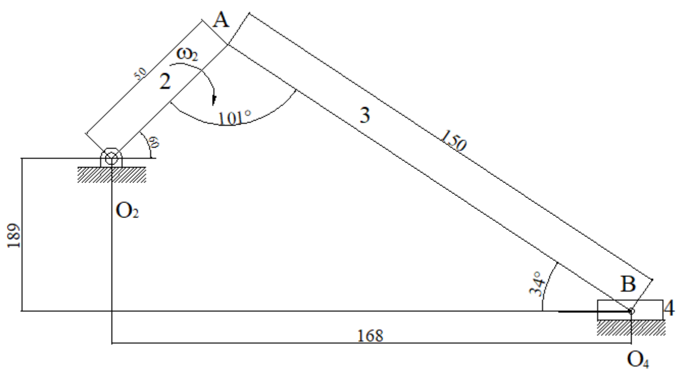

Figure 2. Geometric Model of the Offset Slider–Crank Mechanism

Mechanism Geometry

a) Length of link 2

r_2 = 50 mm

b) Length of link 3
r_3 = 150 mm

c) Horizontal distance $O_2 \rightarrow O_4$
xo_4 = 168 mm

d) Vertical offset (O_2 is lower than O_4)
yo_4 = -189 mm, this offset value represents the main difference compared to a non offset slider crank mechanism.

e) The absolute angle of link 2 with respect to the horizontal axis is:
$\theta_2 = 10^0$

The angular velocity of link 2 is:
ω_2 = 1200 rpm

$$\omega_2 = 1200 \cdot \frac{2\pi}{60} = 40\pi = 125{,}66 \text{ rad/s}$$

Since the rotation is clockwise (CW), the angular velocity is taken as negative:
$\omega_2 = -125{,}66$ rad/s

4.1 Position Analysis

The positional relationship among the mechanism components is expressed using the vector loop equation as follows:

$$\vec{r_2} + \vec{r_3} = \overrightarrow{r_{O_2O_4}}$$

When expressed in the form of complex numbers, the equation can be written as follows:

$$r_2 e^{j\theta_2} + r_3 e^{j\theta_3} = 168 - j189$$

By substituting the values, we obtain:

$$50e^{j\theta_2} + 150e^{j\theta_3} = 168 - j189$$

By separating the real and imaginary components of link 2, the following expression is obtained:

$$50(\cos 10^0 + j\sin 10^0) = 49{,}24 + j8{,}68$$

By transferring the terms to the right hand side, the equation becomes:

$$50e^{j\theta_3} = (168 - 49{,}24) - j(189 + 8{,}68)$$

$$50e^{j\theta_3} = 118{,}76 - j197{,}68$$

The angle of link 3 is then determined as follows:

$$|r_3| = \sqrt{118{,}76^2 + 197{,}68^2} = 230{,}6 \text{ mm}$$

$$\theta_3 = \tan^{-1}\left(\frac{-197{,}68}{118{,}76}\right) = -1{,}67$$

$$\theta_3 = -59^0$$

This is the absolute angle of link 3, not the relative angle.

4.2 Velocity Analysis

The mathematical model of velocity is based on the mechanism loop equation:

$$j\omega_2 r_2 e^{j\theta_2} + j\omega_3 r_3 e^{j\theta_3} = 0$$

Subsequently, the imaginary unit (j) is eliminated from the equation:

$$\omega_2 r_2 e^{j\theta_2} + \omega_3 r_3 e^{j\theta_3} = 0$$

Next, the corresponding values are substituted.

$$(-125,66)(50)e^{j10^0} + \omega_3(150)e^{-j59^0} = 0$$

$$-6283e^{j10^0} + \omega_3(150)e^{-j59^0} = 0$$

The equation resulting from the substitution is decomposed into two components, namely the real component and the imaginary component.

Real, $(-6283\cos 10^0 + 150\omega_3 \cos 59^0) = 0$

Imajiner, $(-6283\sin 10^0 - 150\omega_3 \sin 59^0) = 0$

Accordingly, the following equations are obtained:

$$150\,\omega_3 \cos 59^\circ = 6283\cos 10^\circ$$

$$\omega_3 = \frac{6283\cos 10^\circ}{150\cos 59^\circ}$$

$$= \frac{6187,54}{77,25}$$

$$= +80,09 \text{ rad/s}$$

From the calculation results and the interpretation of the obtained angular velocity, it is determined that link 3 rotates in the counterclockwise (CCW) direction. Further analysis is conducted to determine the linear velocities at specific points of the mechanism.

Point A is located at the end of link 2, where the direction of its velocity is perpendicular to link 2.

$$v_A = |\omega_2| r_2 = (125,66).(50) = 6283 \text{ mm/s}$$

Meanwhile, point B is a point on link 3 that has a velocity relative to point A, with the direction of motion perpendicular to link 3.

$$v_{B/A} = \omega_3 r_3 = (80,09).(150) = 12013,5 \text{ mm/s}$$

4.3 Acceleration Analysis

In this analysis, link 2 is assumed to rotate with a constant angular velocity; therefore, the angular acceleration of link 2 is considered to be zero. The loop equation is obtained by differentiating the velocity loop equation with respect to time, which can be expressed as follows:

$$(-\omega_2^2) r_2 e^{j\theta_2} + (-\omega_3^2 + j\alpha_3) r_3 e^{j\theta_3} = 0$$

The acceleration equation obtained is substituted with known values, then separated into real and imaginary parts to facilitate analysis of the direction and magnitude of the acceleration of each mechanism bar, thus obtaining:

$$\alpha_3 = -2850 \text{ rad/s}^2$$

the negative sign (-) in the calculation results indicates that the direction of rotation of the rod is clockwise (CW).

The acceleration at point A is purely centripetal, because this point is at the end of rod 2, which rotates at a constant angular velocity relative to point O_2. The direction of acceleration at point A is toward the center of rotation, namely toward point O_2 following the axis of rod 2. Thus, the acceleration acting at point A has only a radial component without any tangential component, thus obtaining:

$$a_A = \omega_2^2 r_2 = (125,66)^2.50$$
$$a_A = 789521,78 \text{ mm/s}$$

The pure centripetal acceleration whose direction is towards point B is determined by considering the relative motion between point B and point A on rod 3. The acceleration of point B consists of two main components, namely the acceleration due to the rotation of rod 3 towards point A and the tangential acceleration that arises due to the angular acceleration of rod 3. Thus, the direction of acceleration of point B does not only depend on the direction of motion of rod 3, but also on the direction relative to point A. Through the calculation process using the complex number method, the results of the acceleration analysis of point B are obtained as follows:

$$a_B = \overrightarrow{a_A} - \omega_3^2 r_3 e^{j\theta_3} + j\alpha_3 r_3 e^{j\theta_3}$$

Where:

$\overrightarrow{a_A}$ = translational acceleration A

$\omega_3^2 r_3 e^{j\theta_3}$ = centripetal B relative to A

$j\alpha_3 r_3 e^{j\theta_3}$ = tangential B relative to A

Next, the calculation of each acceleration component at point B with respect to point A is carried out.

a) Relative centripetal component B/A

$$\omega_3^2 r_3 = (80,09)^2.(150)$$
$$\omega_3^2 r_3 = 962161,21 \text{mm/s}^2$$

Vector:

$$-962161,21(\cos 59^0 - j\sin 59^0)$$
$$= -495549,65 + j824733,12$$

b) Relative tangential component B/A

$$\alpha_3 r_3 = (-2850)(150) = -427500 \text{ mm/s}^2$$

Multiplied by j, we get:

$$j\alpha_3 r_3 e^{j\theta_3} = -427500(\sin 59^0 + j\cos 59^0)$$
$$= -366439,02 - j220178,77$$

c) The relative acceleration B/A

$$\vec{a}_{B/A} = (-495549,65 - 366439,02) + j(-824733,12 - 220178,77)$$
$$\vec{a}_{B/A} = -861988,67 - j1044911,89 \text{ mm/s}^2$$

d) Add the acceleration of point A

The acceleration of A is directed along rod 2 (sudut 10^0), then:

$$\alpha_A = -789521,78(\cos 10^0 + j\sin 10^0)$$
$$\alpha_A = -777527,17 - j137099,01$$

e) Final result of acceleration of point B

$$\vec{a}_B = \vec{a}_B + \vec{a}_{B/A}$$
$$\vec{a}_B = (-777527,17 - 861988,67) + j(-137099,01 - 1044911,89)$$
$$\vec{a}_B = -1639515,84 - j1182010,9 \text{ mm/s}^2$$

f) The magnitude of the capture of point B

$$|a_B| = \sqrt{(1639515,84)^2 + (1182010,9)}$$
$$|a_B| = 1,63x10^6 \text{ mm/s}^2$$

g) Direction (downward from the horizontal direction)

$$\theta_{aB} = \tan^{-1}\left(\frac{1182010,9}{1639515,84}\right)$$
$$\theta_{aB} = 35,7^0$$

From these results, it can be concluded that the acceleration of point B is relatively large, which can still be considered reasonable because it is influenced by the high angular velocity ω_2 (1200 rpm), the presence of offset, and the asymmetrical shape of the mechanism. The direction of the acceleration of point B is also not in line with the rod, because it is the result of the sum of several acceleration vector components, not originating from a single component alone.

4.4 Simulation Results and Graphical Visualization Using MATLAB

The simulation was carried out using MATLAB software with the crank angle varied from 0° to 360° in increments of 10°. The geometric parameters of the mechanism were defined according to the design data described in the previous subsection. The slider position equation was derived from the vector loop equation using the complex number method, while the velocity and acceleration equations were obtained by differentiating the position equation with respect to time. The graphs obtained from the MATLAB simulation include the relationship between slider position and crank angle, slider velocity and crank angle, and slider acceleration and crank angle, as explained in the following discussion.

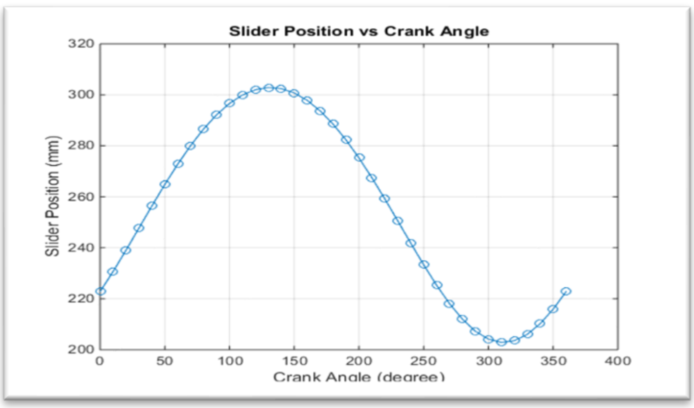

Figure 3. Graph of the Relationship between Slider Position and Crank Angle

Based on Figure 3, the relationship between slider position and crank angle shows a nonlinear and asymmetric pattern due to the presence of an offset between the crank center and the slider path. This offset alters the geometric relationship between the crank angle and the slider position. The maximum and minimum positions no longer occur at the classical angles of 90° and 270°, but shift according to the mechanism configuration. The varying slope of the curve indicates that the change in position with respect to the crank angle is not uniform. Differences between the forward and return strokes confirm that the mechanism exhibits an asymmetric motion characteristic.

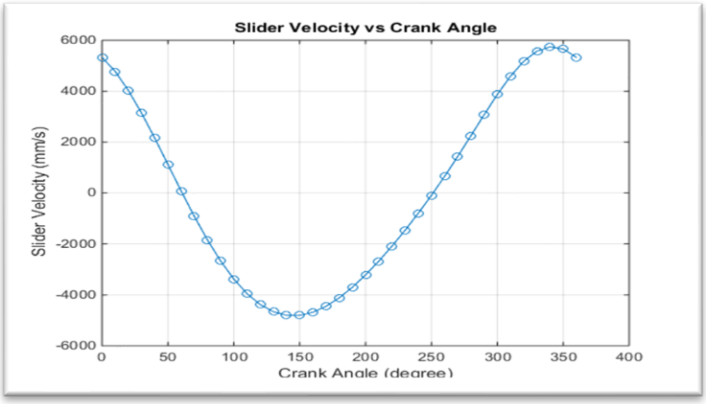

Figure 4. Graph of the Relationship between Slider Velocity and Crank Angle

From the graph of the relationship between slider velocity and crank angle, the slider velocity varies periodically but not symmetrically over one crank revolution. The velocity becomes zero at certain angles corresponding to the extreme positions of the slider, where the direction of motion changes from forward to return or vice versa. The maximum velocity during the forward stroke differs from that during the return stroke, indicating the influence of the offset on the motion characteristics. In addition, the peaks and valleys of velocity do not occur at the classical angles but shift according to the offset configuration. This confirms that the offset causes an uneven distribution of slider velocity with respect to the crank angle.

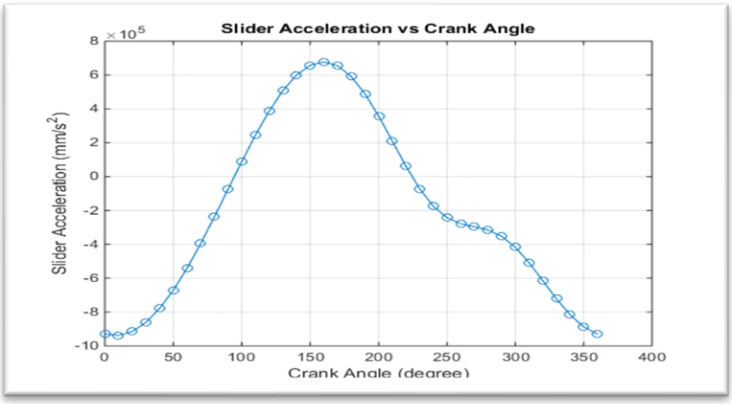

Figure 5. Graph of the Relationship between Slider Acceleration and Crank Angle

Referring to the graph of slider acceleration as a function of crank angle, the acceleration changes sharply and asymmetrically over one crank revolution. The maximum and minimum acceleration values occur at specific angles that differ between the forward and return strokes, indicating a strong influence of the offset configuration. Large acceleration variations generally occur when the crank and connecting rod approach an almost aligned position, causing the inertial components to dominate. Moreover, the irregular shape of the curve shows that acceleration is highly sensitive to changes in crank angle, highlighting the important role of the offset in determining the magnitude of inertial forces acting on the mechanism.

CONCLUSION

Based on the kinematic analysis of the can pressing mechanism with an offset crank–slider model, it can be concluded that the complex number method is capable of representing the relationships of position, velocity, and acceleration of the mechanism mathematically and consistently. The analysis was conducted on a mechanism with a crank length of 50 mm, a connecting rod length of 150 mm, a horizontal distance between supports of 168 mm, a vertical offset of −189 mm, and a crank angular velocity of 1200 rpm. Based on the MATLAB simulation results, graphs were obtained showing the profiles of slider position, velocity, and acceleration with respect to the crank angle, which help visualize the overall motion characteristics of the mechanism.

The acceleration analysis shows that point A experiences purely centripetal acceleration toward the rotation center O_2, while point B experiences resultant acceleration due to the combination of centripetal and tangential components. The acceleration at point B is relatively higher than at point A because of the effects of high angular velocity, vertical offset, and the mechanism's geometry. The MATLAB graphs indicate that the slider motion is nonlinear and asymmetric, with the maximum and minimum positions as well as peak accelerations shifted according to the offset configuration, highlighting the influence of the offset on the kinematic characteristics.

The graph results indicate that the offset crank–slider mechanism exhibits flexible motion characteristics. The position graph shows differences in the duration of the forward and return strokes, the velocity graph demonstrates uneven motion rates, and the acceleration graph displays high peak accelerations at specific angles. The conclusions drawn from these three graphs confirm that the offset allows for adjustment of the slider motion according to process requirements and influences the distribution of forces and inertia throughout a complete crank revolution.

The advantages of the offset mechanism are evident in its flexibility to regulate slider motion, stroke duration, dead center positions, and the distribution of velocity and acceleration. The offset also reduces lateral forces, lowers friction and wear, and improves efficiency and component lifespan. The application of the complex number method based on MATLAB has proven effective for calculating and evaluating kinematic characteristics, making it a reliable reference for the analysis and design of can pressing mechanisms as well as other offset crank–slider mechanisms.

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