

Original Article

# Design and Analysis of a Cross-plane Solenoid Engine for Torque Ripple Stability in Electric Vehicles

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**Abstract** - The internal combustion (IC) engine is recognized as one of the major contributors to environmental pollution. Consequently, the demand for alternative fuels has been increasing due to the depletion of fossil fuel reserves and the increasing environmental concerns. However, electric motors can cause instability at lower speeds, caused by torque ripples, which cannot be effectively resolved via the original motor structure. To tackle this problem, this paper proposes utilizing the boxer engine structure with a cross-plane crankshaft. To leverage this structural advantage of the IC engine, this study designs a solenoid-based engine, integrating the boxer engine composition and cross-plane crankshaft design. Since significant motion is not observed in the prototype engine. The present study, therefore, conducts a diagnosis of the engine's failure to operate as intended, then gives a detailed analysis of the underlying causes and proposes potential solutions. Through numerical analysis, this study evaluates the efficiency of the refined engine.

**Keywords** - electric engines, electromagnetic forces, electromagnetic inductions, engine stabilities, motors

## 1. Introduction

A significant time has passed since the initial attempts to make use of fossil fuels successfully provided mankind with an abundant energy supply. However, as excessive use of fossil fuels began causing pollution and global environmental harm, efforts to decrease reliance on the fuels increased. This shift is well represented in the development of electric cars. The utilization of electric motors is noticeable in emerging vehicles, which is distinguishable from the internal combustion (IC) engines traditional cars widely adopted. Though these new systems were environment-friendly and could contribute to the reduction of fossil fuel uses, they still had mechanical issues to resolve. A slow-driving vehicle suffers from vibration and instability caused by it. Therefore, a primary objective is specified to develop and set a basis for solenoid-based engines. The new form of engine has the potential to adopt the advantages of both IC engines and motors. Solenoid-based engines have been studied in several papers. [1–5] However, these studies do not specifically address vehicle stability, and the results related to engine performance are often not clearly presented. Therefore, in this paper, a combination of a boxer engine and a cross-plane crankshaft was designed to reduce the vibration at lower speeds.

## 2. Theoretical Background

The vibration of an engine is largely classified into primary and secondary vibrations, which is a fundamental

concept that must be understood beforehand to comprehend the structural characteristics of the engine.

For pistons that are not in the power stroke and do not contribute to the crankshaft rotation at that moment, their circular motion is converted into linear reciprocating motion inside the cylinder as the crankshaft rotates. Primary vibration is caused by the fundamental reciprocating motion of the piston. Because the piston undergoes a complete up-and-down cycle for every full rotation of the crankshaft, this vibration occurs at the same frequency as the engine's rotational speed. This type of vibration is inherent in all reciprocating engines, but its effects can be minimized through careful counterweight placement on the crankshaft or the use of additional balancing mechanisms.

Secondary vibration, in contrast, results from the non-uniform acceleration of the piston due to the mechanical and geometrical constraints of the crankshaft and connecting rod. Unlike primary vibration, this oscillation occurs at twice the crankshaft's frequency, meaning the piston experiences two oscillations per revolution. This effect is particularly noticeable at high RPMs, where secondary vibration can cause additional mechanical fatigue, structural stress, and unwanted noise. While smaller in magnitude compared to the primary vibration, this secondary component plays a crucial role in engine dynamics, as it can contribute to mechanical stress, increased noise, and even resonance issues if left unaddressed



Depending on the arrangement of pistons and the relative position of the crankshafts, engines are categorized into different structures. An understanding of the various structures is crucial to ultimately selecting one fit to minimize motor vibration. The structure and explanations of the boxer engine are as follows.

Boxer engines are a sort of flat engine. The angle between each crankpin measures  $180^\circ$ , allowing a symmetrical motion about the middle of the engine. The corresponding  $180^\circ$  gap of the two cylinder banks enables a reduced height compared to other engines. A boxer engine's superiority is shown in the low height and proportioned piston motion. Its low height lowers the center of mass, increasing the driving stability. The mirror-imaged motion of the pistons promotes neutralizing secondary vibration, as demonstrated in the following section.

$$x = r\cos\theta + l\cos\Phi \quad (1)$$

Where  $x$  denotes the position of the piston,  $r$  is the distance between the main journal and crankpin,  $\theta$  is the rotated angle of the crank,  $l$  is the length of the connecting rod, and  $\Phi$  denotes the angle of the connecting rod.

Using the Fourier series to convert equation.1 into equation.2, the periodic function of the second term can be converted into a sum of trigonometric functions.

$$x = r\cos\theta + \frac{r^2}{4l}\cos\theta \quad (2)$$

The first term represents the basic motion of the piston. Because the piston's displacement follows a cosine function, this term directly corresponds to the crankshaft's rotational frequency. In other words, the piston completes one full stroke per crankshaft revolution, making this term the dominant component of its movement. The second term accounts for secondary vibration, which arises due to the geometric constraints of the connecting rod. Since the connecting rod has a finite length, the piston does not move at a perfectly uniform rate throughout its stroke. This introduces a second-order oscillation, which occurs at twice the crankshaft's rotational frequency.

One of the key advantages of a Boxer engine is its partial offset of secondary vibrations due to its unique piston arrangement. In a Boxer engine, pistons are positioned in horizontally opposed pairs, meaning that as one piston moves upward, its counterpart moves downward with an identical motion but in the opposite direction. Because these opposing pistons generate secondary oscillatory forces of equal magnitude but opposite directions, the net effect is the mutual cancellation of secondary vibrations. As a result, a Boxer engine operates with significantly reduced secondary vibration compared to L-type or V-type engines, which often require balance shafts or additional countermeasures to mitigate vibration.

In the case of primary vibrations, they are possible to be canceled out because of their symmetrical motion. However, due to its structural traits, it is easy to err in manufacturing the actual engine, demoting its ability to moderate the vibration.

The crankshaft is one of the important parts of this paper. Crankshafts have two structural types, and the decision between the two options is crucial to attain the paper's objective. In this paper, the cross-plane crankshaft, which is more effective in reducing torque ripple, is used.

Cross-plane crankshafts are structures of which the cranks holding the same crankpin gap of  $90^\circ$  with another crank group. It is known to apply more regular force compared to the other crankshaft structure. This design brings about an alteration in acceleration due to changes in crankshaft angles, suppressing secondary vibrations. As opposed to flat-plane crankshafts, there is no need for extra components to counteract vibration, showing its great performance in vibration cancellation. The downside is that it is heavier and more expensive than the other crankshafts. The unique structure and the required exhaust pipe design contribute to its high cost.

### 3. Numerical Calculate

The inductance of the coil is produced from the equation below.

$$L = \mu_0 \cdot N^2 \cdot \frac{A}{d} \quad (3)$$

The permeability of free space,  $\mu_0$  has a value of  $4\pi \cdot 10^{-7} \text{H/m}$ .  $N$  is for the number of turns of the coil,  $A$  for the cross-sectional area, which is the inertial area of the solenoid through which the magnetic field passes. And  $d$  stands for the length of the coil wrapped around the solenoid. The value of each variable used in this paper is respectively 460 turns, **1, 225mm<sup>2</sup>**, **62mm**. The calculated inductance is detailed below.

$$L = \mu_0 \cdot 460^2 \cdot \frac{1,225}{62} = 5.26 \text{mH} \quad (4)$$

The calculated resistance and current is detailed below.

$$R = \rho_{Cu} \cdot \frac{l}{A} = 0.222\Omega \quad (5)$$

$$I_{applied} = \frac{V_{applied}}{R} = 216.22 \text{A} \quad (6)$$

Where  $R$  denotes the resistance of the coil,  $\rho_{Cu}$  is the resistivity of copper,  $l$  is the length of the wire comprising the coil,  $I_{applied}$  is the applied current, and  $V_{applied}$  denotes the applied voltage.

The magnetic field strength inside a solenoid is proportional to the number of turns of the coil and applied current, while having an inverse proportionality to the length of the coil.

$$B = \mu_0 \cdot \frac{N \cdot I_{\text{applied}}}{l} = 2.016T \quad (7)$$

The magnetic moment of a permanent magnet is calculated through the equation below.

$$m = M \cdot V \quad (8)$$

Here, the volume of the magnet and the magnetization are each represented as  $V$  and  $M$ . Magnetization is the quantity for how strong a material becomes magnetized when exposed to an external magnetic field and a product of  $B_{\text{mag}}$  (magnetic field due to a material's magnetization, or magnetic flux density) over  $\mu_0$ . The magnet used in this study, a neodymium magnet, values 1.3T in magnetic flux density.

$$M = \frac{B_{\text{mag}}}{\mu_0} = 1.035 \times 10^6 A/m \quad (9)$$

$V$  equals to  $6.08 \times 10^{-7} m^3$ , therefore the magnetic moment of the magnet is as shown below.

$$m = 0.6293A \cdot m^2 \quad (10)$$

Suppose the strength of the field decreases to 0 on both ends of the solenoid. This allows us to approximate the gradient of  $B$ , the rate of change in the magnetic field over distance, by  $B$  over half the length of the coil.

$$\nabla B \approx \frac{B}{0.5l} = 65.03T/m \quad (11)$$

Consequently, the value of exerted force on the magnet is a product of  $\nabla B$  and magnetic moment.

$$F = m \cdot \nabla B = 40.92N \quad (12)$$

Torque, a rotational form of force, is produced by the expansion of Equation.12, which is needed to later process rotating systems,

$$\tau = F \cdot r \cdot \sin \theta \quad (13)$$

In equation.13,  $r$  stands for the distance between the center of the main journal and the crankpin, in this research, 35mm.  $\theta$  is the rotated angle of the crankshaft. For a convenient calculation process, we presume the sine value always to be 1 in any different  $\theta$  values.

$$\tau = 40.92N \times 0.035m \times 1 = 1.432N \cdot m \quad (14)$$

The RPM of a crankshaft is calculated using the piston's average speed. In the equation below, its speed is expressed as  $V_p$ .

$$RPM = \frac{V_p}{2r} \times 60 \quad (15)$$

The traveled distance of 31mm over a time of 1ms estimates to an RPM value of 26,570RPM.

$$RPM = \frac{31 \times 10^{-3}}{1 \times 10^{-3}} \times \frac{60}{2 \times 35 \times 10^{-3}} = 26,570RPM \quad (16)$$

The engine's horsepower results from an equation of torque and RPM.

$$P[W] = \frac{2\pi}{60} \times \tau \times RPM = 3,984.5W \quad (17)$$

$$P[HP] = \frac{P[W]}{745.7} = 5.345HP \quad (18)$$

## 4. Materials and Methods

### 4.1. Engine Structure Design

In this study, the position of the coils was established considering the mutual inductance between each solenoid coil. The length of the crankshaft was then determined to fit this spacing. In the following passages, the process to obtain mutual inductance will be described.

$$M = k\sqrt{L_1 L_2} \quad (19)$$

Above,  $k$  is a symbol for coupling coefficient and  $L_1, L_2$  for the inductance of two different coils. The mutual inductance calculation utilizes equation.5 and equation.20.

$$M = k\sqrt{L_1 L_2} = k \cdot 5.26mH \quad (20)$$

The coupling coefficient  $k$  has a maximum value of 1 when two coils are extremely close to each other, gradually decreasing as they gap further.  $k$  expressed as a function of its distance is inscribed below.

$$k(d) = k_0 \times e^{-a \cdot d} \quad (21)$$

$k_0$  and  $a$  are each coupling coefficient at zero distance and magnetic field attenuation constant. Ideally,  $k_0$  approximates to 1 and  $a$  varies between  $0.1 \sim 0.5m^{-1}$  within the air medium. In **Figure.1**, we have drawn the results of equation.22 as a function of the gaped distance, having adjusted the value of two constants as such:  $k_0 = 0.5, a = 0.1$ .

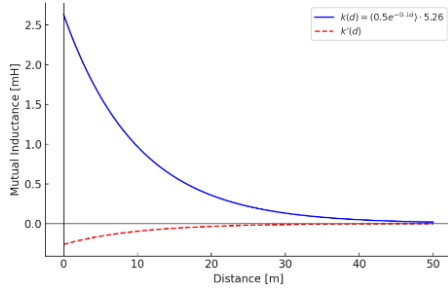


Fig. 1 Function of mutual inductance and its derivative.

For the optimization of the engine size, the longer distance between two coils was set to 1 meter. Based on these settings, the 3D modeling of the engine generated can be seen in **Figure.2**. The magnetic field of when voltage is induced in a single coil is visualized below in **Figure.3**.

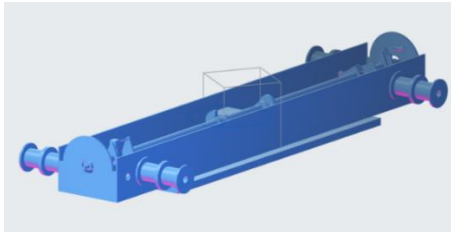


Fig. 2 3D view of the engine.

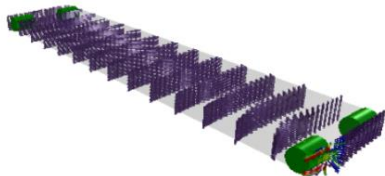


Fig. 3 Visualization of magnetic fields.

By evenly positioning the four rotary switch actuators in a 90° gap, we have enabled the voltage appliance to the coils according to the crankshaft's angle. For stable transport at slow speeds, the cross-plane crankshaft was selected, which ensures stable operation due to having more segments of angles than a flat-plane crankshaft. Also, the design was formed in a boxer engine structure in the hope of minimizing the vibration that comes from the typical IC engine structure.

**4.2. Circuit Design**

In this paper, the engine was designed to have four switch-coil pairs, each switch corresponding to one coil. The design originally intended for a relay-switched induct circuit where the coil being induced with voltage changes as the crankshaft presses different switches while spinning. The circuit diagram is illustrated below.

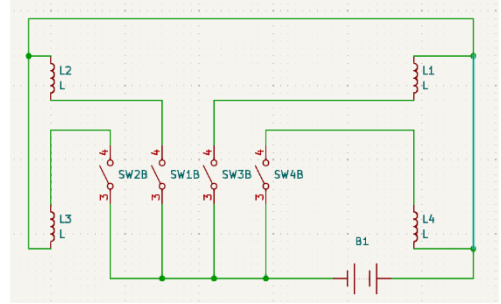


Fig. 4 Relay switch inductor circuit.

**4.3. Fabrication of the Engine**

The parts comprising the crankshaft are fabricated from a commonly used aluminum alloy, Al6061. The components are shown in **Figure.5**.

The assembled crankshaft of the paper has a stroke of 74.4mm and a bore of 10mm. Which makes it an undersquare engine specializing in low rpm motors. The engine frame is depicted in **Figure.6**.

The completed model which includes the crankshaft is shown below as **Figure.7**.

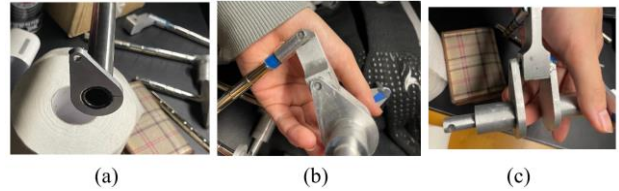


Fig. 5 Crankshaft parts. (a) crank throw, (b)&(c) piston & connecting rod & crank throw.

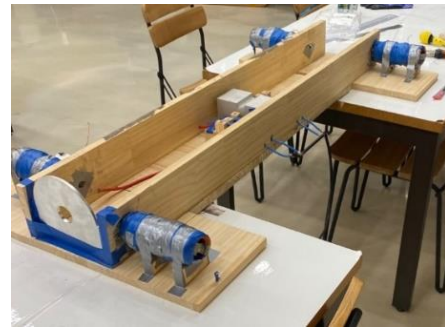


Fig. 6 Engine frame.

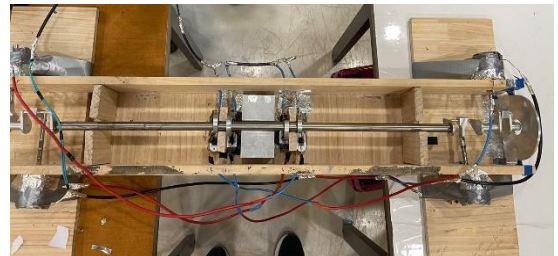


Fig. 7 Engine assembled.

#### 4.4. Analysis and Redesigning of the study

##### 4.4.1. Analysis of the Study – the Reason of Its Failure

The solenoid engine crafted for the study did not function as intended. The following text analyzes the reason for its failure.

##### *Bias of Force on the Crankshaft*

The engine utilized a cross-plane crankshaft, four-cylindrical structure, where a single piston was connected to four crank throws each displaying different angles. On the two ends of the crankshaft, about as long as 1.4 meters, locates two crank throws with a 90° gap in-between. Due to its feature where a maximum of only two coils is applied voltage over the same period, force is applied on only one end of the crankshaft. The unbalance in force and the long length of the crankshaft tends to yaw, revolving around the holding block in the middle of the engine. A certain amount of force, originally intended to spin the crankshaft, is used for a different direction, exerting lower levels of torque than expected.

With the decrease in torque, the movement of the crankshaft due to yaw makes a change in the angle of which force is applied. A shift in angle occurs in connecting rods and pistons, which must be aligned with the cylinder, resulting in an impediment in its movement. Not only does it disturb the application of forces in the intended direction, but it also has the potential to cause problems, such as separating the magnets that make up the piston or jamming the piston inside its cylinder.

##### *Error in the Structural Design*

The other problem we expected to be the reason for failure is the insufficient formation of clearance for the parts to spin. When force is applied to the piston after it is attached to the crankshaft, a large fraction of force is converted into yaw overwhelmed roll. As a cause of this, we believe it is okay to blame an action of strong friction due to the lack of distinct clearance.

By design, the length of the crankpin was just about identical to the width of the connecting rod. The friction area of the joint where the crankpin and connecting rod, the connecting rod and piston meets shows not enough room to allow for unproblematic movement. Also providing a space too tight for lubricants, defeating the idea to somewhat moderate friction. This is why we infer that a strong friction must have taken place in the rotating process.

##### 4.4.2. Adjusting Design Errors

We redesigned the four-cylindrical boxer engine structure into an eight-cylindrical one. Making the two crank throws face each other with an 180° gap between. This certain structure was devised to prevent the yaw of the crankshaft. It enabled a symmetrical application of force by dividing the rotating angle into four 90° units and connecting two pistons

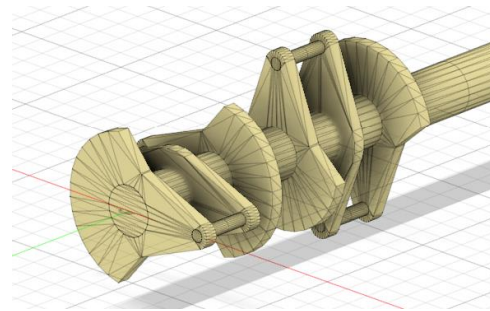
onto each unit. The design accounting for all the feedback is depicted in **Figure.8**. **Figure.9** shows the new crank throw structure joined with the crankshaft.

This design applies force from the two ends of the crankshaft symmetrically, preventing too much force from being applied to one side.

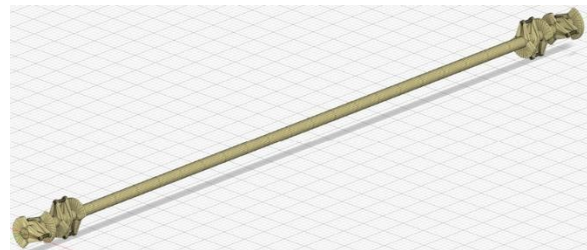
On the original design of the engine, the length of the connecting rod was half the amount of stroke. Therefore, when no electric force is induced, the junction spot of the piston and connecting rod locates along the center of the main journal. When electricity is induced, the connecting rod rotates about the crankpin due to the piston's force, rather than receiving the force while staying perpendicular to the ground surface. This not only produces the rotating motion of the crankshaft, but it also causes roll in the piston itself. The center axis of the piston and the direction of the magnetic force does not align, causing the magnets and coils to form a magnetic field weaker than what was designed.

This error can be overcome by tuning the connecting rods, pistons, and the solenoids. By adjusting the inner diameter of the solenoid coil to stroke, we design the moving piston to not get stuck with the connecting rod. Also, matching the piston's bore with the coil's diameter, the roll of the piston can be prevented. This way, we make the center axis of the piston to always be aligned with the solenoid's center axis. Finally, we match the connecting rod's length with stroke so that the piston could have the minimum length without slipping out of the cylinder pocket.

As mentioned above, by rearranging the clearance between the connecting points of each part, a space to supply lubricant can be obtained. This results in a decrease in friction which occurs as rotational action continues.



**Fig. 8 Configuration of the adjusted crank throw.**



**Fig. 9 8-cylindered crankshaft with the adjusted crank throw.**

#### 4.4.3. Numerical Analysis of the Refined Study

Referring to equation.13, the value of torque respective to angle  $\theta$  is such.

$$\tau = 40.92 \times 0.035 \times \sin(\theta) \quad (22)$$

In a cross-plane crankshaft, there is a change in force every  $90^\circ$  rotation. The equation that reflects this phenomenon can be expressed as a function of time.

$$\tau = 1.4322 \times \sin\left(\frac{\pi}{2} - \left(100\pi t \bmod \frac{\pi}{2}\right)\right) \quad (23)$$

$$\tau = 1.4322 \times \cos\left(100\pi t \bmod \frac{\pi}{2}\right) \quad (24)$$

Let the torque calculated above  $\tau_{piston}$ , then the net torque is the total accumulation of  $\tau_{piston}$  with friction torque,  $\tau_f$ .

$$\tau_{net} = \tau_{piston} + \tau_f \quad (25)$$

In equation.25,  $\tau_f$ . represents the viscous friction torque, which can be expressed as the product of the viscous friction coefficient, radius, and angular velocity. Considering the direction of torque, the viscous friction torque is given as follows.

$$\tau_f = -cr\omega \quad (26)$$

Here,  $c$  denotes the viscous friction coefficient, which typically ranges between 0.2 and 0.05 for metals lubricated with oil. In this study, a value of 0.1 is assumed.  $r$  represents the radius of the crank of which friction is applied to, which is 10mm.  $\omega$  is the angular velocity, which can be derived through the following process.

$$I \frac{d\omega}{dt} = \tau_{net} = \tau_{piston} + \tau_f \quad (27)$$

The moment of inertia  $I$  of the crankshaft is defined in terms of the mass  $m$  of the crank and its radius  $r$ .

$$I = \frac{1}{2}mr^2 \quad (28)$$

Rearranging equation.27 using equation.24 and equation.26 returns the following equation.

$$I \frac{d\omega}{dt} = 1.4322 \cos\left(100\pi t \bmod \frac{\pi}{2}\right) - cr\omega \quad (29)$$

$$\frac{d\omega}{dt} + \frac{cr}{I}\omega = \frac{1.4322 \cos\left(100\pi t \bmod \frac{\pi}{2}\right)}{I} \quad (30)$$

This equation is a first-order linear differential equation, which can be solved using the following method. The general

form of a first-order linear differential equation is given as such.

$$\frac{dy}{dt} + P(x)y = Q(x) \quad (31)$$

By applying the integrating factor method, the general solution is acquired as follows.

$$y = e^{-P(x)} \int e^{P(x)} Q(x) dx \quad (32)$$

Employing this in equation.30, with an initial condition of  $\omega = 0$  at  $t = 0$ , results in the following equation.

$$\omega = e^{-\frac{cr}{I}t} \left[ \frac{1.4322}{I} \int e^{\frac{cr}{I}t} \cos\left(100\pi t \bmod \frac{\pi}{2}\right) dt \right] \quad (33)$$

Thus,  $\tau_f$  is given as follows.

$$\tau_f = -cre^{-\frac{cr}{I}t} \left[ \frac{1.4322}{I} \int e^{\frac{cr}{I}t} \cos\left(100\pi t \bmod \frac{\pi}{2}\right) dt \right] \quad (34)$$

In equation.33, the limiting speed of the angular velocity can be confirmed by how the value appears when 't' diverges to infinity. In order to confirm this mathematically, the equation in which the term of the transient response is removed from equation.33 is as follows.

$$\omega = \frac{1.4322}{I} \int e^{\frac{cr}{I}t} \cos\left(100\pi t \bmod \frac{\pi}{2}\right) dt \quad (35)$$

Given that the crank weighs 4.27kg and has a radius of 10mm, the moment of inertia of the crankshaft can be determined using equation.28 as follows.

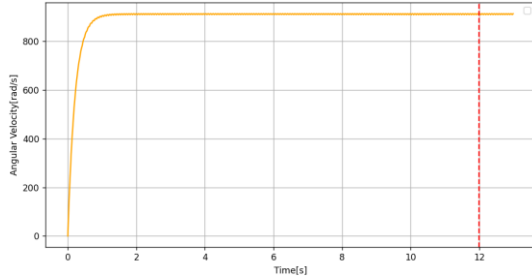
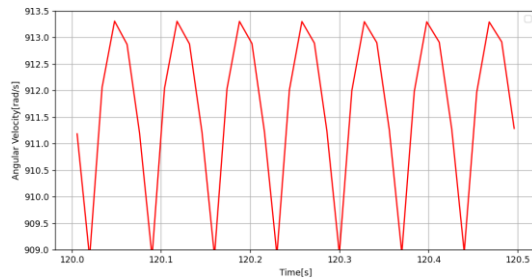
$$I = \frac{1}{2} \times 4.27 \times 0.01^2 = 0.0002135 \quad (36)$$

Therefore, the angular velocity can be summarized as follows.

$$\omega = 6707.57 \times \int e^{4.684t} \cos\left(100\pi t \bmod \frac{\pi}{2}\right) dt \quad (37)$$

## 5. Results and Discussion

The angular velocity is presented graphically in **Figure 10**. The point where the rate of angular velocity, excluding ripple effects, drops below 0.1rad/s<sup>2</sup> is approximately 11.99s. Beyond this point, the angular velocity can be considered as the limit angular velocity.

Fig. 10  $\omega - t$  graph.Fig. 11 ripple shown in  $\omega - t$  graph.

Magnifying the angular velocity within the limit angular velocity range produces a graph of **Figure 11**.

In the given graph, the ripple period is approximately 0.07s, corresponding to a frequency of approximately 14.3Hz. Additionally, the angular velocity in the stable boundary is approximately 911.2rad/s, and the ripple amplitude is approximately 2rad/s. Converting this amplitude into RPM is shown in the following value.

$$RPM_{stable\ boundary} = \frac{911.2}{2\pi} \times 60 \approx 8700RPM \quad (38)$$

$$RPM_{maximum\ ripple} = \frac{911.2+2}{2\pi} \times 60 \approx 8713.7RPM \quad (39)$$

$$RPM_{minimum\ ripple} = \frac{911.2-2}{2\pi} \times 60 \approx 8686.3RPM \quad (40)$$

The RPM fluctuation is  $\pm 13.7$  RPM, which falls well below the generally perceivable range of 50–100 RPM at high rotational speeds exceeding 8,000 RPM.

## Conclusion

Based on the above calculations, in Results and Discussion, the engine's torque ripple corresponds to

approximately 50–100 RPM, which translates to a frequency range of about 0.83–1.67 Hz. Although this falls within the range of frequencies perceptible to the human body, it is generally difficult to detect in practice. Therefore, the vibrations generated during operation of the cross-plane-applied eight-cylinder solenoid boxer engine are expected to be hard to notice to the human body. Additionally, given the shortened force application cycle, it is expected that torque ripple can be reduced compared to electric motors that operate similarly to conventional flat-plane crankshafts. This serves as theoretical evidence suggesting that the application of this engine structure can positively contribute to torque ripple reduction and enhanced driving stability.

## Conflicts of Interest

The authors declare that there are no competing interests regarding the authorship and publication of this paper. This study was conducted purely for academic purposes, without any financial benefit obtained from individuals, institutions, corporations, or other stakeholders throughout the research process. Furthermore, the study was not carried out with the intention of pursuing commercial interests, patent acquisition, or product development, and there is no possibility that the results have been influenced by any external economic or political factors. The authors affirm that the entire research process was conducted independently and autonomously to ensure objectivity and integrity.

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Author 1, Author 2, Author 3, and Author 4 contributed equally to this work.

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