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Experimental Study of the Motion Modes of a Planar Mechanical System with Multi-Clearance Revolute Joints

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Abstract

Clearances in joints of a mechanical multibody system can induce impulsive forces, leading to vibrations that compromise the system's reliability, stability, and lifespan. Through dynamic analysis, designers can investigate the effects of the clearances on the dynamics of the multibody system. A revolute joint with clearance exhibits three motions which are; free-flight, impact and continuous contact motion modes. Therefore, a multibody system with n-number of revolute clearance joints will exhibit 3n motion modes which are a combination of the three motions in each joint. This study investigates experimentally the nine motion modes in a mechanical system with two revolute clearance joints. A slider crank mechanism has been used as the demonstrative example. We observed that the experimental curve exhibits a greater impact compared to the simulation curve. In conclusion, this experimental investigation offers valuable insights into the dynamics of planar mechanical systems with multiple clearance revolute joints. Utilizing a slider-crank mechanism for data acquisition, the study successfully confirmed seven out of nine motion modes previously identified in numerical research. The missing modes are attributed to inherent complexities in real-world systems, such as journal-bearing misalignment.

Keywords

Slider Crank Mechanism, Dynamic Responses, Revolute Clearance Joints, Motion Modes

1. Introduction

In mechanical systems, the presence of clearances in imperfect joints is a natural consequence of factors such as machining tolerances, wear, material deformations, and assembly errors. Although necessary for the assembly and relative motion of components, these clearances unfortunately lead to performance degradation. They introduce problems like increased wear rate, vibrations, noise levels, and energy dissipation, impacting both the system's accuracy and longevity.

Much of the existing literature has examined the dynamic effects of clearance in planar systems where only a single joint is considered to have clearance. For example, studies by Liu *et al.* [1] and Chen *et al.* [2] have provided critical insights into the impact of clearances on individual revolute joints. These studies, while valuable, do not extend their investigation into systems with multiple clearance joints. Yet in real-world mechanical systems, multiple joints are far more common and can interact in complex ways that single-joint models cannot capture.

Other works such as those by Erkaya and Uzmay [3] have delved into some aspects of multiple clearance joints, yet these often lack an investigation into the dynamic interaction between them. There is a growing consensus ([4]-[13]) that attention must be turned toward the nonlinear dynamic analysis of systems with multiple clearance joints. However, despite these recent developments, there remains a glaring gap in understanding the collective behavior of these joints, particularly the motion modes inside the joint with clearance unlike [4] who have done a numerical investigation.

Importantly, computational methods have been used in many studies, but these often require substantial computational resources, thus limiting experimental investigations. The complexity of the analysis significantly increases as more clearance joints are considered, further dissuading extensive experimental validation [14].

This paper aims to fill this gap by conducting an experimental investigation into the motion modes exhibited by mechanical systems with multiple revolute clearance joints. The paper is organized as follows: Section 2 outlines the materials and methods for data collection; Section 3 focuses on studying the motion modes within the clearance revolute joints and their impact on dynamic response, and Section 5 concludes the study.

2. Materials and Methods

2.1. Experimental Test Rig

A slider-crank mechanism was chosen in this study which has been used by many researchers since it is relatively simple, easy to control, and can be subjected to various ranges of speeds and accelerations. In addition, the results from the numerical works which are to be compared with the obtained experimental results were obtained from a slider-crank mechanism.

The crank, connecting rod, and the slider were made entirely of mild steel which has a high rigidity in order to reduce the flexibility effects. The revolute joints between the slider-connecting rod and crank-connecting rod were constructed as dry journal bearings with variable and controlled radial clearances. The remaining revolute joint between the crank and fixed link was constructed with minimum clearance and friction in order to minimize any contamination of the measured data. A linear bearing of Winkel type was used as the sliding motion of the slider with minimum friction and less play in order also to reduce data contamination. The lengths and inertial properties of the links of the slider-crank mechanism were designed and fabricated so as to have values close to those given in **Table 1** as much as possible. These parameters had been used in the numerical work by [4] whose results were to be compared with the results of this study.

An AC motor with a variable speed control was used to rotate the crank. An AC motor was chosen because it is a viable power source for a variety of applications due to its flexibility, efficiency and quiet operation. The motor speed was controlled by varying the frequency using a variable frequency drive. The output speed of the motor was reduced by using a pulley and belt system. The driven pulley was deliberately designed to be large and heavy in order to reduce significantly the high motor output speed to lower speeds of the crank, and to act as an energy reservoir and hence help in damping the driving torque and speed fluctuations due to the high expected impacts in the system. To support the weights of the large pulley and the crank axial load, a thrust bearing was used at the bottom of the drive shaft.

The components of the slider-crank mechanism and other mechanical components were designed using Autodesk Inventor and working drawings were generated. The assembled CAD drawing of the mechanical components of the rig is shown in **Figure 1**.

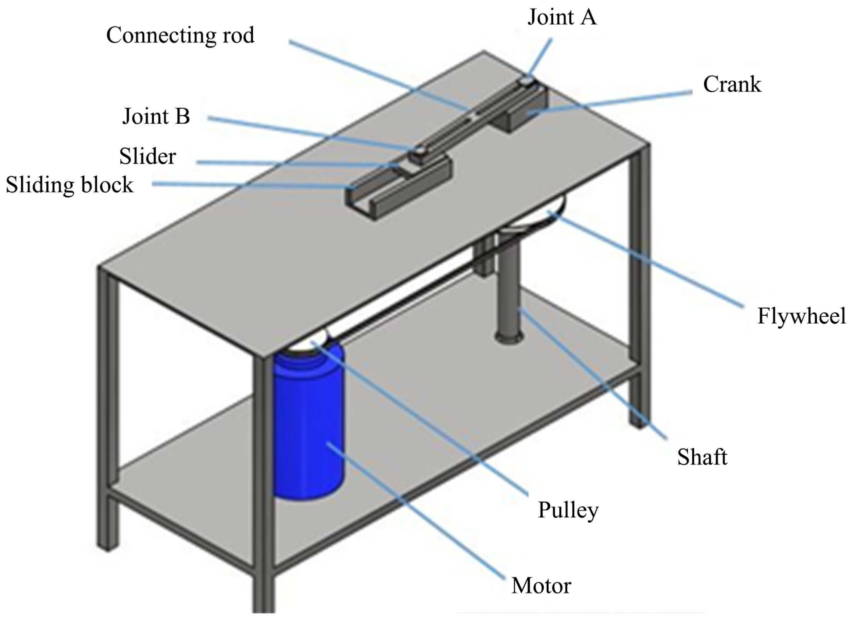


Figure 1. CAD assembly of the rig.

Table 1. Parameters used for the design and fabrication of the experimental test rig.

Description	Characteristics
Length of the Crank, s	50 mm
Length of the Connecting-rod, l	300 mm
Mass of the Crank, ml	17.9 kg
Mass of the Connecting-rod, m^2	1.13 kg
Mass of the Slider, m^3	1.013 kg
Mass of Sliding Block	2.034 kg
Nominal Bearing radius, R	15 mm
Journal Radius, r	14 mm, 14.5 mm, 14.8 mm
Radial clearance in the joints, R	1 mm, 0.5 mm, 0.2 mm

Data Acquisition

In this study, a multifaceted data acquisition system was employed to ensure rigorous and precise data capture. The system incorporated an E6 shaft encoder sensor, ADLX335 accelerometer, FlexiForce force sensor, YGX-TQ1012-50 N·m torque sensor, and an ESP32 microcontroller.

The E6 shaft encoder sensor was pivotal for recording the angular position of the shaft, generating a pulse at every 5.625 degrees of rotation. This digital sensor operated with high precision and consistency, ensuring accurate measurements.

The ADLX335 accelerometer was used to measure both static and dynamic accelerations. Calibrated to a sensitivity of 300 mV/g, this sensor demonstrated accurate performance within a bandwidth range of 0.5 Hz to 1600 Hz at a sampling rate of 1000 Hz.

Joint force measurements were made using FlexiForce sensors, which were calibrated against known weights. These sensors offered excellent linearity and were placed in the crank-connecting rod joint and slider-connecting rod joint to capture the contact forces in the revolute clearance joints.

Torque measurements were acquired with a YGX-TQ1012-50 N·m torque sensor. Operating under an excitation voltage of 10VDC, the sensor allowed for precise measurements of torque based on its output voltage.

Finally, the ESP32 microcontroller, programmed through Arduino 2.0.3, was responsible for data collection from all sensors. Data was timestamped and transmitted to a PC via a USB cable at a baud rate of 1,000,000 bits per second for further analysis, facilitated by PUTTY software, and saved in CSV format.

In summary, this data acquisition system was meticulously calibrated and deployed, ensuring a precise and reliable collection of critical variables such as shaft position, acceleration, force, and torque. This system forms the backbone of the study's empirical analysis, as depicted in **Figure 2** and **Figure 3**.

Figure 2. Experimental data collection of slider crank mechanism.

Legend

1- Pillow block bearing 1
2- Torque sensor
3- Flywheel (Pulley)
4- Pillow block bearing support
5- Pillow block bearing 2
6- Shaft
7- Encoder E6

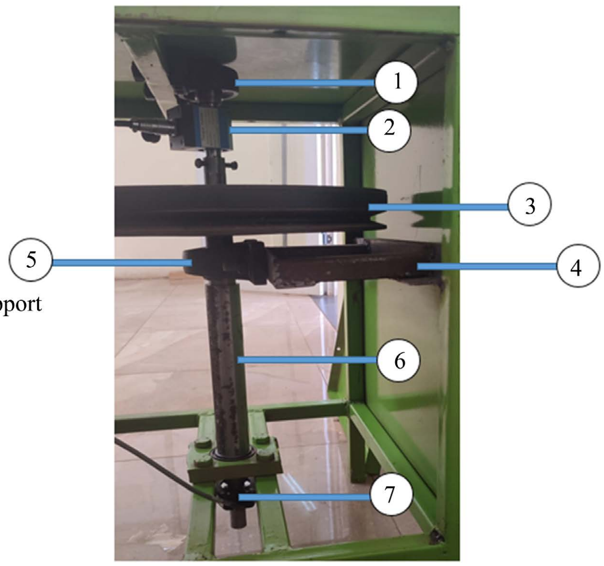


Figure 3. A section of the test-rig showing the placement of the torque and encoder.

2.2. Study of the Motion Modes inside the Revolute Clearance Joints

First, the experimental results obtained were compared with the simulation results obtained in [4] using visualization and calculation of the error between the simulation and experimental values using the Root Mean Squared Error (RMSE) which is a good measure able to quantify the deviation of the simulated values from the experiment values. The RMSE is given as:

$$\text{RMSE} = \sqrt{1/n \sum_{i=1}^{n} \left(X_i - Y_i\right)^2} \tag{1}$$

where

X_i: Experiment values;

Y_i: Simulated values;

n: Number of the data points.

For comparison purposes, radial clearance of 1 mm at both the slider-connecting rod (s-cr) and crank-connecting rod (c-cr) joints was used, and the crank was rotated uniformly at 95 rpm. The responses from the simulation and experimental studies were plotted and comparisons made. Table 2 shows a summary of the mechanism/joint parameters used.

The responses from the experimental results were analyzed in order to identify the existence of the nine motion modes exhibited by the mechanism due to the presence of the two revolute clearance joints. These nine motion modes were discovered numerically by [4] and are shown in Table 3. In the mechanical realm, as delineated in Table 3, every joint embodies three distinct motion modes: free, impact, and continuous contact. For instance, during a given experimental setup, the crank-connecting rod might be operating under the free mode, whereas the slider-connecting rod could manifest in impact, contact, or even remain in the free mode. This multifaceted interplay of motion modes within two revolute joints having clearance births the nine discernible motion modes, each emerging from the unique combinations of the inherent modes.

3. Results

3.1. Study of the Motion Modes inside the Revolute Clearance Joints

3.1.1. Comparison of Numerical and Experimental Results

As previously mentioned, the numerical results presented in this study are obtained from [4]. Figures 4-6 display the acceleration and forces profiles obtained

Table 2. Mechanism parameters used for the data collection

Description	Characteristics
Nominal bearing radius, R	15 mm
Journal radius, r	14 mm
Radial clearance in the joints	1 mm
Crank speed	95 rpm

Table 3. Motion modes inside two revolute joint with clearance [4].

		Motion inside c-cr joint		
		Free-flight motion	**Impact motion**	**Continuous-contact motion**
Motion inside s-cr joint	Free-flight motion	Free-Free Motion Mode	Impact-Free motion mode	Contact-Free Motion Mode
	Impact motion	Free-Impact Motion Mode	Impact-impact motion mode	Contact-Impact Motion Mode
	Continuous-contact motion	Free-Contact Motion Mode	Impact-Contact Motion Mode	Contact-Contact Motion Mode

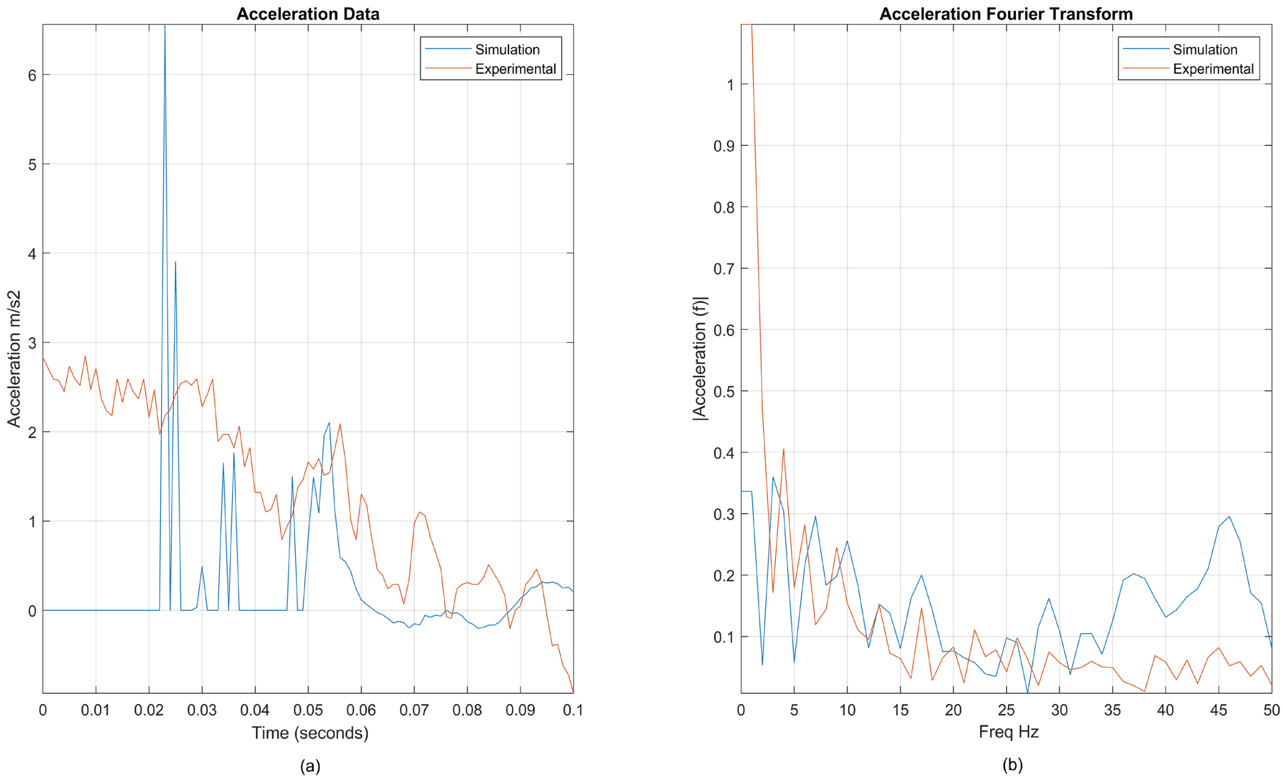


Figure 4. Dynamic response of the acceleration for simulation and experimentation (RMSE = 1.29): (a) time domain and (b) frequency domain.

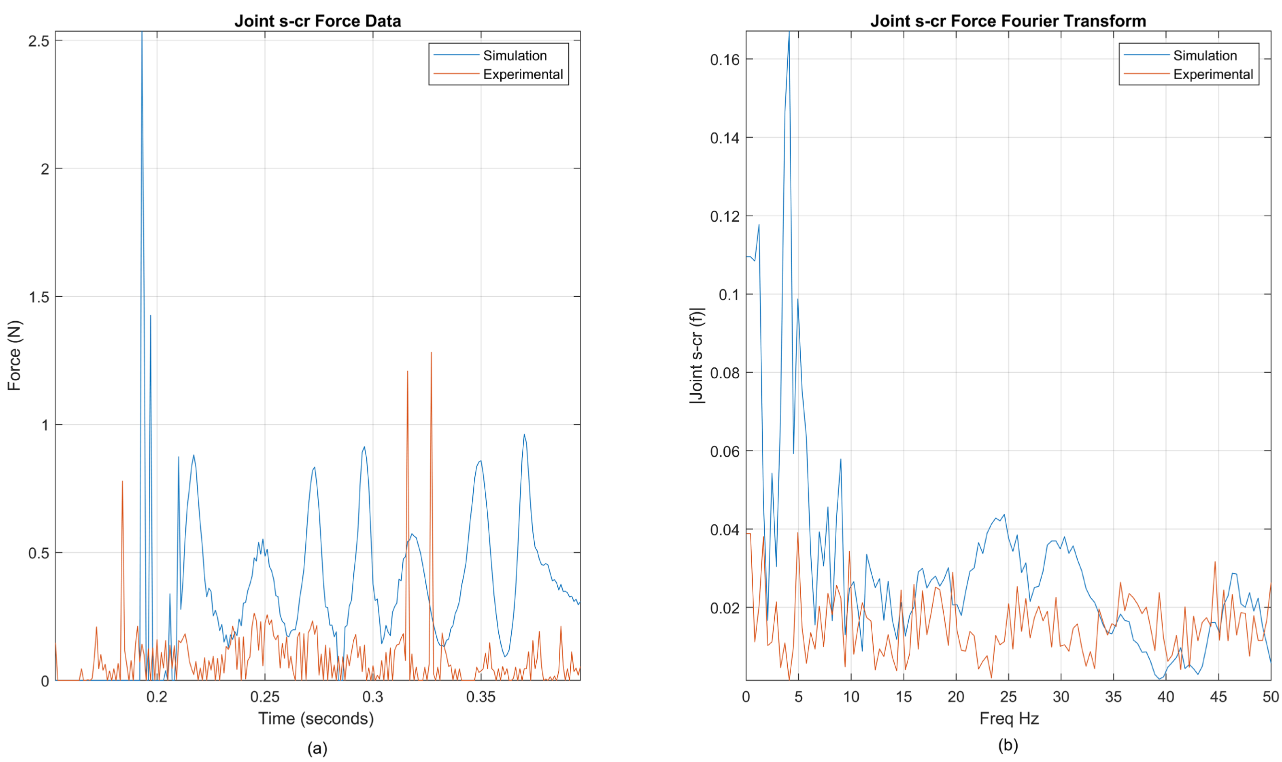


Figure 5. Dynamic response of the slider-connecting rod joint for simulation and experimentation (RMSE = 0.43): (a) time domain and (b) frequency domain.

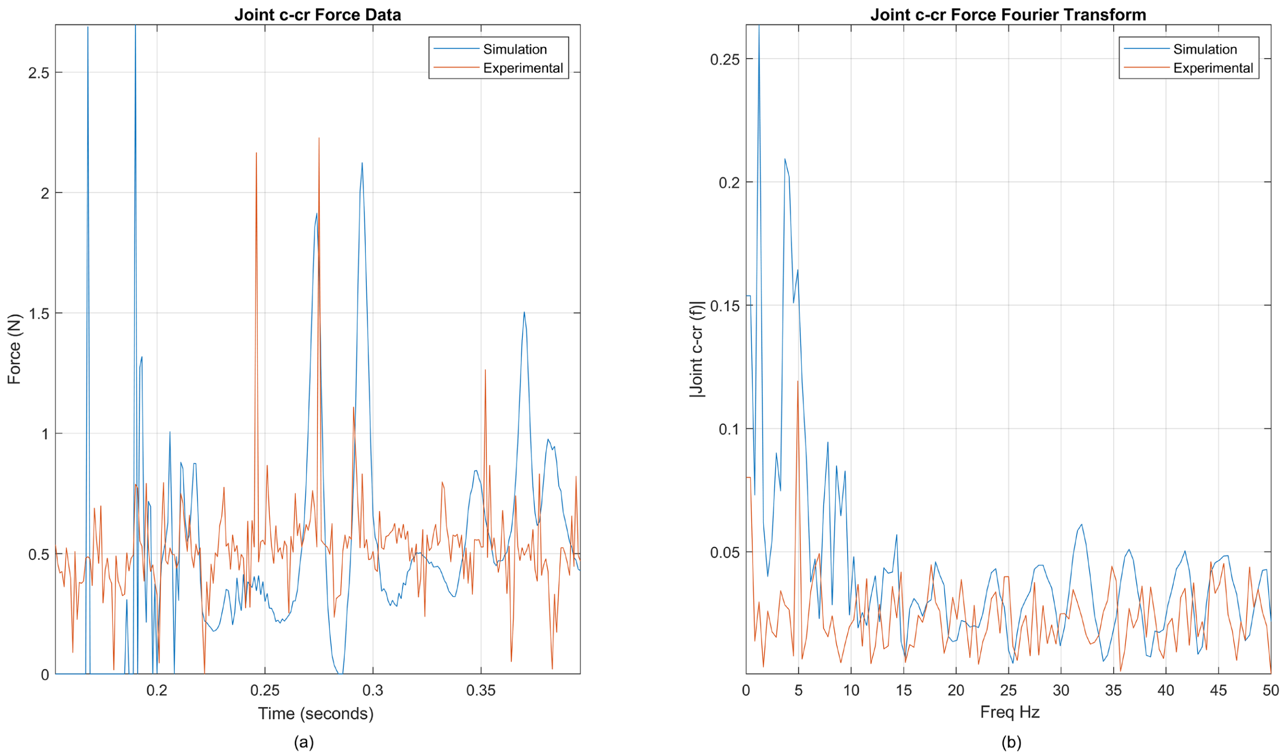


Figure 6. Dynamic response of the crank-connecting rod joint for simulation and experimentation (RMSE = 0.72): (a) time domain and (b) frequency domain.

from both the simulation and experimental results at specific time intervals. A notable resemblance can be observed between the simulation and experimental curves in Figure 4(a) and Figure 4(b), particularly in Figure 4(a) within the time range of 0.05 s to 0.095 s.

For a more quantifiable measure of this resemblance, we also calculated the Root Mean Square Error (RMSE) for the various variables. The RMSE values were 1.29 for acceleration, 0.43 for the slide-connecting rod joint, and 0.72 for the crank-connecting rod joint. These values demonstrate a fair degree of similarity between the experimental and simulation data, falling within an acceptable range for the system being studied. An RMSE value closer to zero generally suggests a more accurate numerical simulation. In our investigation, the relatively low RMSE values for the slide-connecting rod and crank-connecting rod joints affirm the efficacy of the model in capturing these dynamics. However, the RMSE for acceleration was notably higher, meriting further investigation into factors such as sensor sensitivity, measurement noise, or other complexities in the dynamic behavior not fully captured by the numerical simulations.

However, in the time domain, it is worth noting that the experimental curve exhibits a greater impact compared to the simulation curve. This discrepancy can be attributed to the presence of play in the sliding block, which could not be entirely eliminated. This play could have resulted in the contamination of the experimental data and introduced additional noise. Furthermore, the misalignment between the journal and bearing elements, which is inherent in practical mechanical systems, was not considered in the numerical simulations. Additionally, the dynamics of the flywheel were not accounted for in the numerical simulations.

Figure 5 and Figure 6 show the dynamic responses of the slider-connecting rod (s-cr) and crank-connecting rod (c-cr) joints, respectively. Notably, there exists a resemblance between the simulation and experimental curves (Figure 5(a), Figure 5(b), Figure 6(a) and Figure 6(b)). Within the time intervals of t = 0.192 s to 0.379 s for Figure 5(a) and t = 0.18 s to 0.301 s for Figure 6(a). These curves exhibit similar trends, with the exception of the experimental curve which displays a more pronounced impact. As mentioned earlier, this can be attributed to the fact that the numerical measurements neglect factors such as play and misalignment between the journal and bearing elements. These unaccounted factors contaminate the experimental data by introducing noise.

3.1.2. Motion Modes inside Revolute Clearance Joints

1) Free-free, free-impact, free-contact motion

When the journal navigates unobstructedly within the bearing, this is termed free-flight motion. This means there's a lack of forceful contact at the joint. In situations characterized by the c-cr clearance joint exhibiting such motion, Figure 7(a) provides a depiction of the forces acting on both revolute clearance joints. It also outlines the variations in crank torque and slider acceleration over specific time durations.

Figure 7(a)(i) demonstrates that the c-cr joint is experiencing free-flight motion at t = 0.143 s because contact forces are zero. This demonstrates that the crank does not rotate in touch with the connecting rod and that the journal and bearing of the c-cr joint are not in contact. During the free-free motion, the crank torque is excepted to be constant since there is no contact between the bodies, only then does an external torque need to be applied to drive the crank. This is only true if the dynamics of the flywheel were not considered. [4] were able to observe that the crank torque is constant and almost at zero when the c-cr is in free-flight motion.

In the study, the slider-crank mechanism is driven by the crank. As a result of the inertial effects of the moving components, the connection between the rod and slider detaches swiftly once the c-cr joint transitions to free-flight motion. This phenomenon is visible in Figure 7(a)(i), indicating that shortly after the c-cr joint starts its free-flight motion, the s-cr joint follows suit. This sequential initiation of free-flight motion at the s-cr joint, post the c-cr joint's transition, results in the slider achieving its peak movement. Figure 7(a)(ii) highlights these peak accelerations, particularly during the intervals when the c-cr joint is in its free-flight phase. Such observations confirm the occurrence of simultaneous free-flight motion modes.

When the c-cr joint undergoes free-flight motion, it induces a consequent impact in the s-cr joint. This impact event at the s-cr joint is depicted in Figure 7(b)(i). Such an impact causes the crank torque to reach a pronounced peak, as

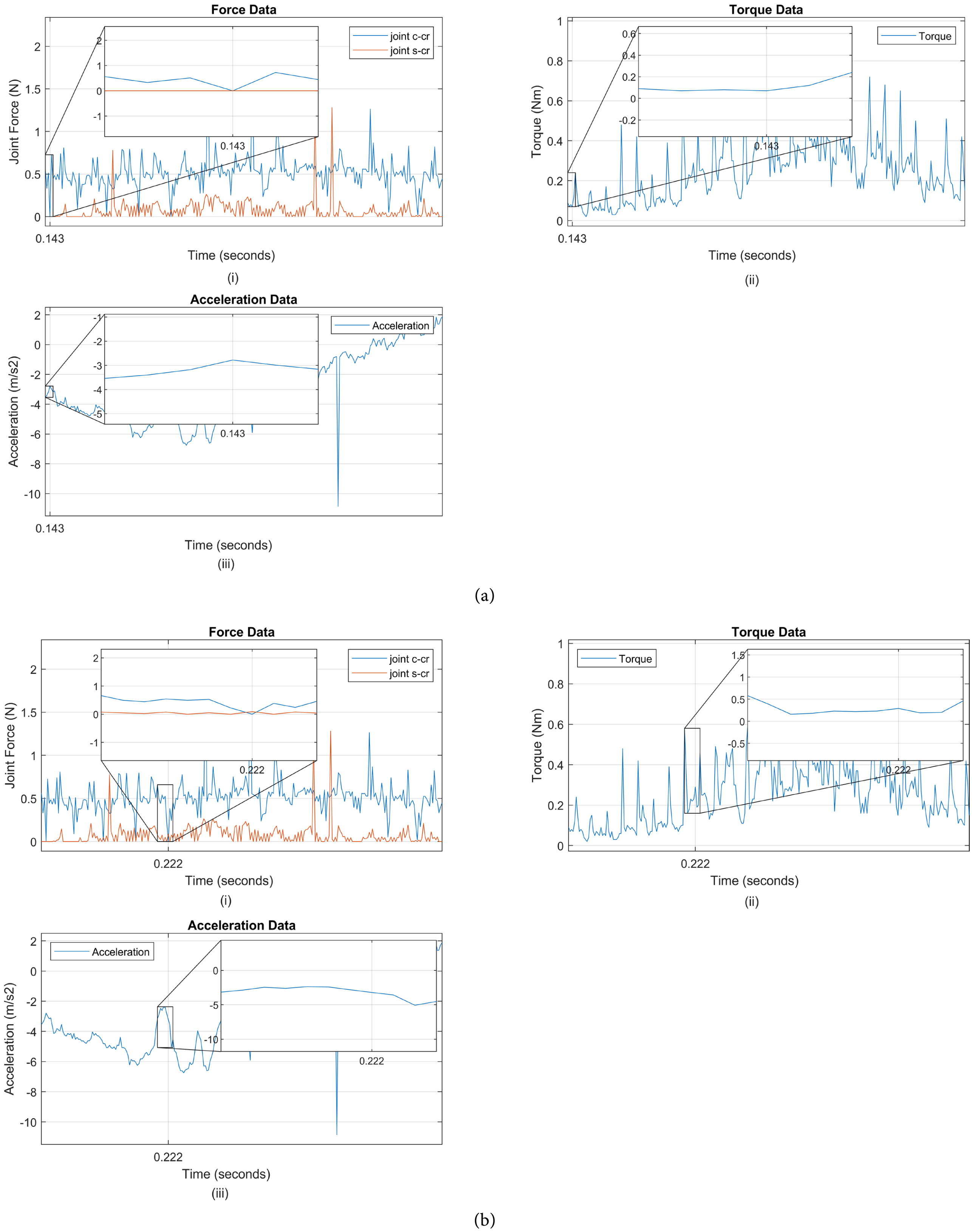


Figure 7. (a) Experimental responses showing free-free motion with (i)-joint forces, (ii)-crank torque, (iii)-slider acceleration (zoom resolution, t = 0.143 s); (b) Experimental responses showing free-impact motion with, (i)-joint forces, (ii)-crank torque, (iii)-slider acceleration (zoom resolution, t = 0.222 s).

illustrated in Figure 7(b)(ii). Following this peak, there's a noticeable reduction in slider acceleration during the free-flight phase, as displayed in Figure 7(b)(iii). These observations confirm the occurrence of free-impact motion modes.

Additionally, no evidence was found for a free-contact motion mode. This observation aligns with the findings of [4], who, via simulation, determined that when the c-cr joint of a crank-driven slider-crank mechanism experiences free-flight motion, the s-cr joint can only exhibit free-flight or impact motions.

2) Impact-free, impact-impact, impact-contact motion

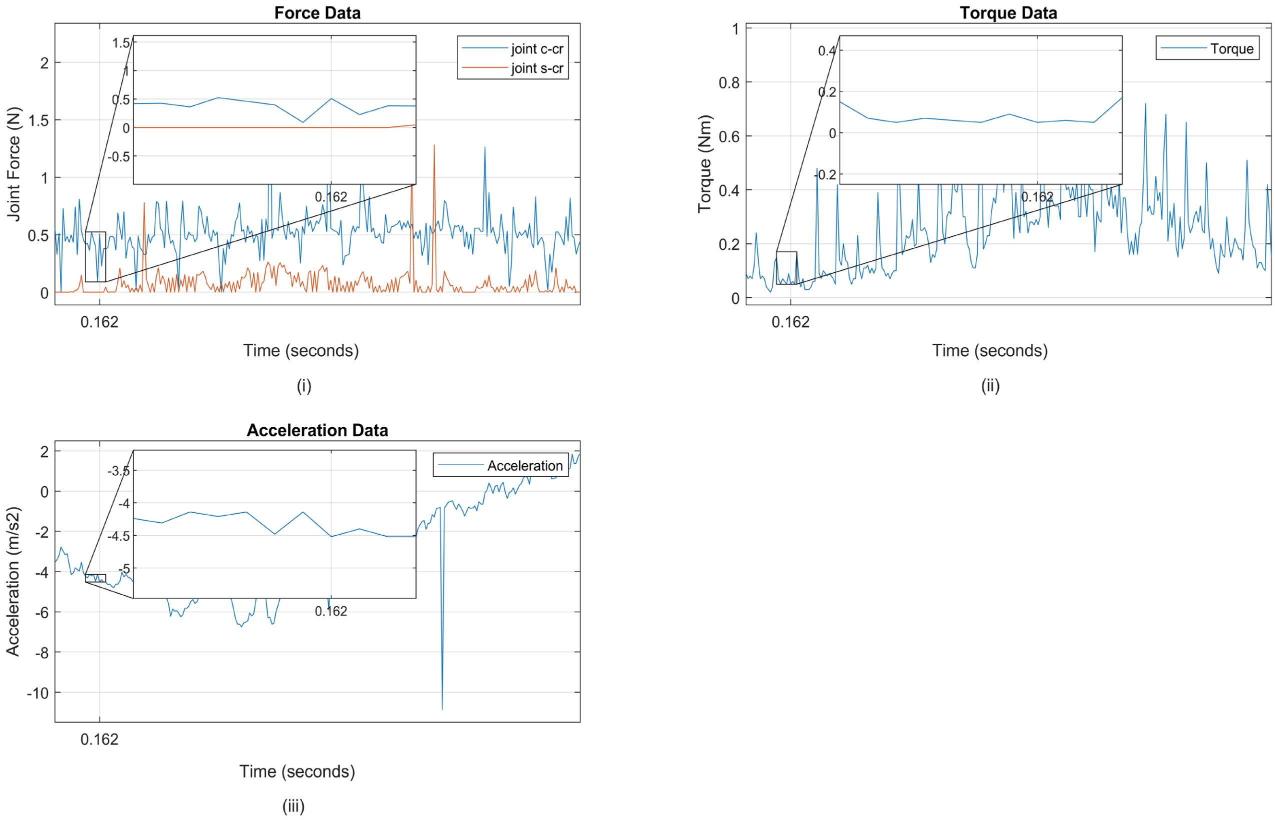
As the free-flight mode concludes, the system experiences impact motion. This is a phase during which forces are both exerted and then quickly nullified within the system. Notably, this mode introduces discontinuities in both the kinematic and dynamic attributes of the system.

Figures 8(a)-(c) serve to illustrate this. They detail the forces at the dual revolute clearance joints, variations in the crank torque, and the trends in the slider acceleration against specific time intervals. Specifically, these figures highlight the moments when the c-cr clearance joint undergoes impact motion. As depicted in Figure 8(a)(i) and Figure 8(b)(i) and Figure 8(c)(i), substantial impact forces manifest at the joint during this motion, a consequence of the mechanism's links being engineered for significant rigidity. Nonetheless, the intensity of the ensuing peaks in slider acceleration and crank torque are directly influenced by the s-cr joint's motion mode at the exact moment of the c-cr joint's impact.

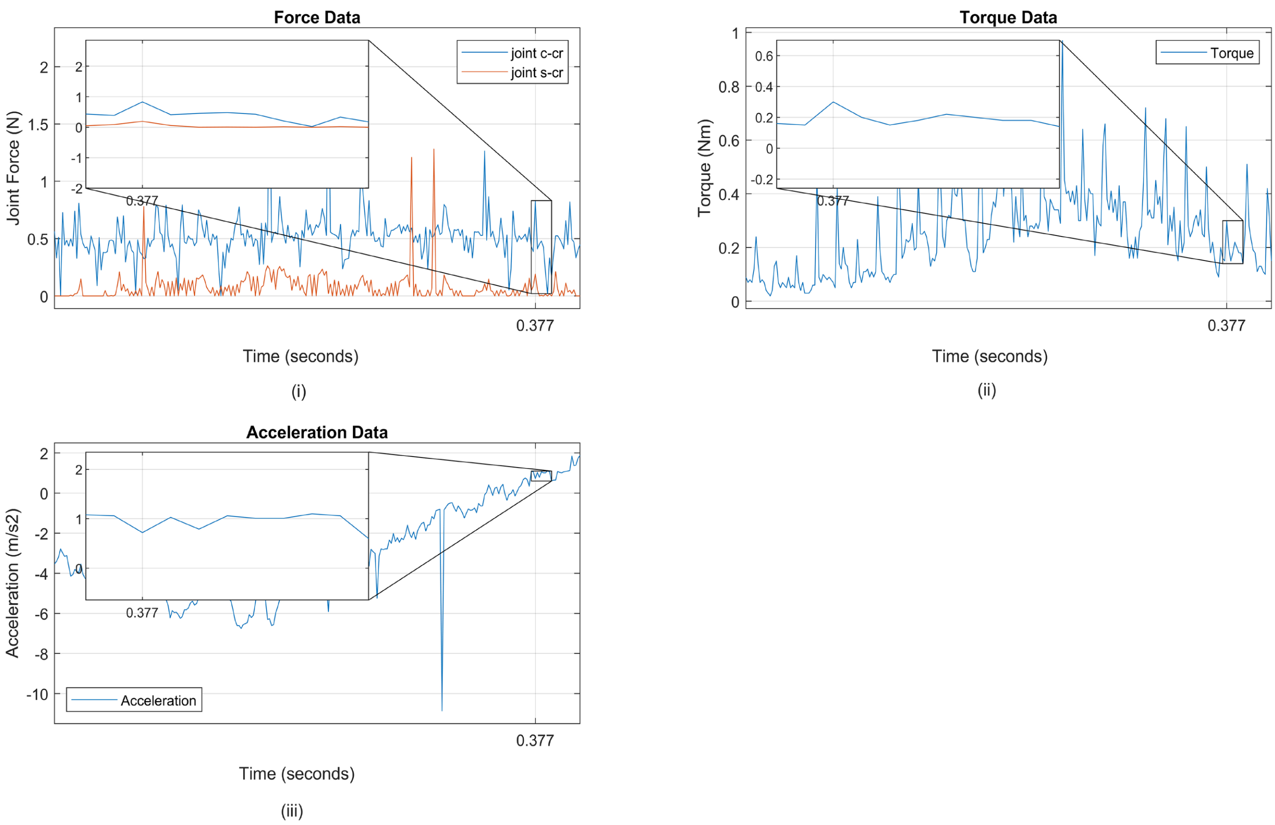
As seen in Figure 8(a)(i), at times t = 0.162 s when there is impact at c-cr during the time when the s-cr joint is in free-flight motion (impact-free motion mode), the torque and the acceleration have negative peak or impact as shown in Figure 8(a)(ii) and Figure 8(a)(iii). During impact-free motion mode, the slider acceleration is expected to be zero since the slider velocity is constant. This is only true if the friction at the sliding joint is zero. Muvengei *et al.* [4] were able to observe the zero slider accelerations during impact-free motion modes since they assumed zero friction at the sliding joint. For the practical mechanism used, it was impossible to eliminate friction at the sliding joint. Therefore, as shown in the experimental results in Figure 8(a)(iii) the slider acceleration was not zero during the impact-free motion modes observed during the sampled time.

When the s-cr joint transitions into free-flight motion, any disturbances or impacts within the c-cr joint swiftly result in a direct and immediate impact in the s-cr joint itself. This phenomenon gives rise to what's termed as "impact-impact" motion scenarios.

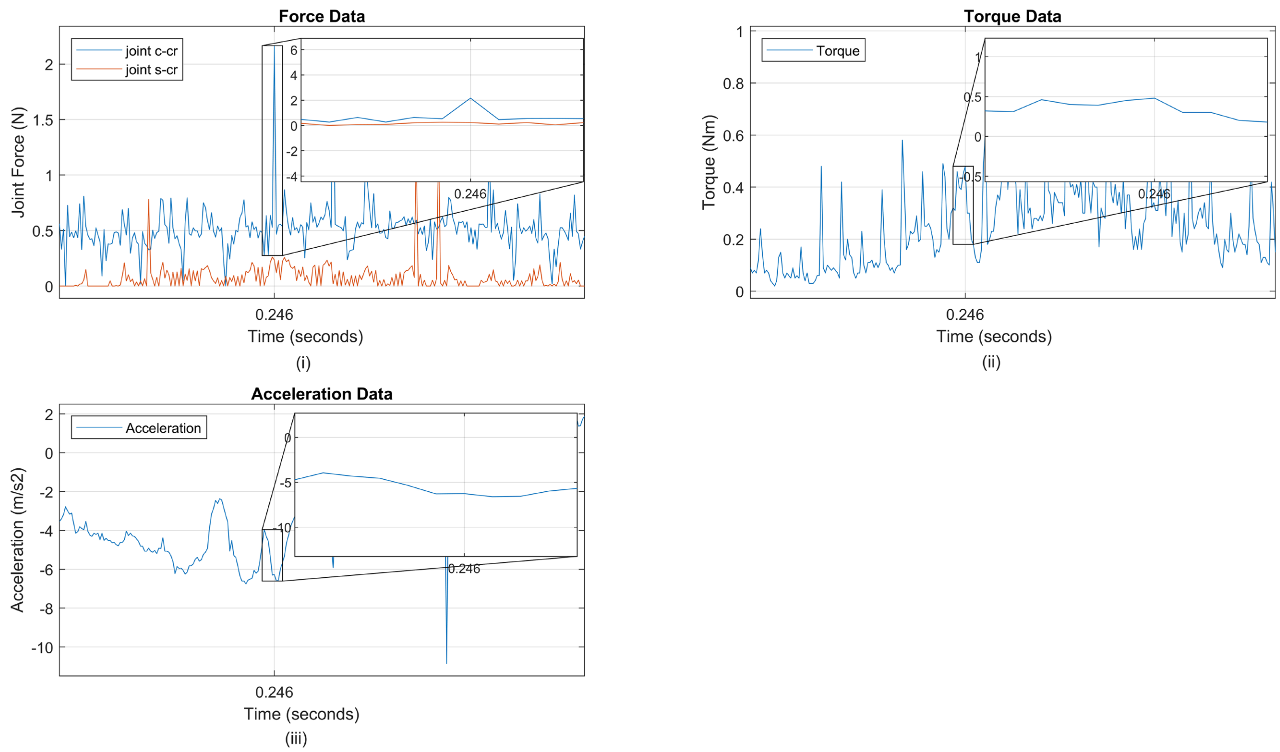
In these scenarios, an impact occurring within one joint (be it c-cr or s-cr) rapidly triggers a subsequent impact in the other. This behavior is rooted in the design approach where all components, including the connecting rod, are conceptualized as rigid entities. The prominence of these "impact-impact" motions amplifies with the mechanism operating at higher velocities. For instance, in Figure 8(b)(i) at the timestamp t = 0.377 s, there's a clear indication of an impact within the c-cr joint during the period the s-cr joint too is undergoing its


Force Data
joint c-cr
joint s-cr
Joint Force (N)
Time (seconds)
0.162
(i)
Torque Data
Torque
Torque (Nm)
Time (seconds)
0.162
(ii)
Acceleration Data
Acceleration
Acceleration (m/s2)
Time (seconds)
0.162
(iii)

(a)


Force Data
joint c-cr
joint s-cr
Joint Force (N)
Time (seconds)
0.377
(i)
Torque Data
Torque
Torque (Nm)
Time (seconds)
0.377
(ii)
Acceleration Data
Acceleration
Acceleration (m/s2)
Time (seconds)
0.377
(iii)

(b)



(c)

Figure 8. (a) Experimental responses showing impact-free motion with i-joint forces, ii-crank torque, iii-slider acceleration (zoom resolution, t = 0.162 s); (b) Experimental responses showing impact-impact motion with, (i)-joint forces, (ii)-crank torque, (iii)-slider acceleration (zoom resolution, t = 0.377 s); (c) Experimental responses showing impact-contact motion with, (i)-joint forces, (ii)-crank torque, (iii)-slider acceleration (zoom resolution, t = 0.246 s).

own impact motion (illustrating the "impact-impact" motion mode). In tandem with this, Figure 8(b)(ii) highlights a torque spike, and Figure 8(b)(iii) delineates a declining trend in the slider acceleration.

Given the design approach where the connecting rod is conceptualized as a rigid entity, an impact in the c-cr joint, especially when the s-cr joint is already in its impact phase, can prompt a sustained contact within the s-cr joint.

Illustratively, in Figure 8(c)(i) pinpointing the timestamp t = 0.246 s, the c-cr joint displays impact motion simultaneously as the s-cr joint remains in contact, signifying the "impact-contact" motion mode. During this specific phase, a minor spike is noticeable in both slider acceleration and crank torque, as depicted in Figure 8(c)(iii) and Figure 8(c)(ii) respectively. Such observations align with findings by Muvengei *et al.* [4], who identified a pronounced peak in the crank torque, the magnitude of which directly correlates with the impact force exerted on the c-cr joint during this "impact-contact" mode.

3) Contact-free, contact-impact, contact-contact motion

The "contact motion" or "continuous contact motion" describes a scenario where the journal aligns and moves alongside the bearing walls. This particular mode concludes when the journal disengages from the bearing, transitioning into the free-flight phase.

Depicting this, Figures 9(a)-(c) illustrate the force dynamics at the twin revolute clearance joints, along with the variations in crank torque and the trends in slider acceleration, indexed by specific time intervals. These figures specifically spotlight moments when the c-cr clearance joint operates in the "continuous contact" mode.

As shown in Figure 9(a)(i), when the c-cr joint exhibit a continuous-contact mode at around t = 0.168 s (contact-free motion mode), the slider acceleration is excepted to have zero value. Muvengei *et al.* [4] were able to observe the zero slider accelerations during the contact-free motion but for the practical mechanism used, it was impossible to see the zero acceleration due to the existence of friction in the sliding joint.

Given that the connecting rod is constructed to function as a rigid link, an impact is evident in the s-cr joint when the c-cr joint operates in the continuous-contact mode, as depicted in Figure 9(b).

As seen in Figure 9(b)(i), at times t = 0.308 s when there is impact at s-cr during the time when the c-cr joint is in continuous-contact motion (contact-impact motion mode), both the slider acceleration and the crank torque are showing a negative peak at that time, as shown in Figure 9(b)(iii) and Figure 9(b)(ii) which is similar to what Muvengei *et al.* [4] found through simulation results.

In the numerical study referenced from [4], a contact-contact motion mode was observed; however, this motion was notably absent in our experimental findings. This discrepancy can be largely attributed to several complexities inherent in real-world mechanical systems that are often simplified or overlooked in numerical models. Key among these is the issue of misalignment between the journal and bearing elements. While the numerical simulations may assume perfect alignment and thus predict a contact-contact motion mode, actual mechanical systems rarely achieve such ideal conditions. This misalignment can introduce additional variables like increased friction, load imbalances, or even minute shifts in the contact points, which can alter the motion modes drastically. Another contributory factor could be the material properties, which can differ from the idealized versions used in simulations. Finally, issues such as manufacturing tolerances, wear and tear, and external environmental conditions (e.g., temperature, humidity) could further exacerbate the difference between numerical predictions and experimental observations. Due to these complexities, the contact-contact motion mode was not observable in our experimental setup, underscoring the need for nuanced interpretations when transitioning from numerical models to real-world experiments.

The comparison between the simulation and experimental results revealed some similarities, although the experimental data exhibited more pronounced effects. This can be attributed to the susceptibility of experimental data to various sources of noise, including measurement errors, limitations of the instruments used and environmental disturbances. These factors introduce random fluctuations and additional peaks in the experimental curve, which may not be

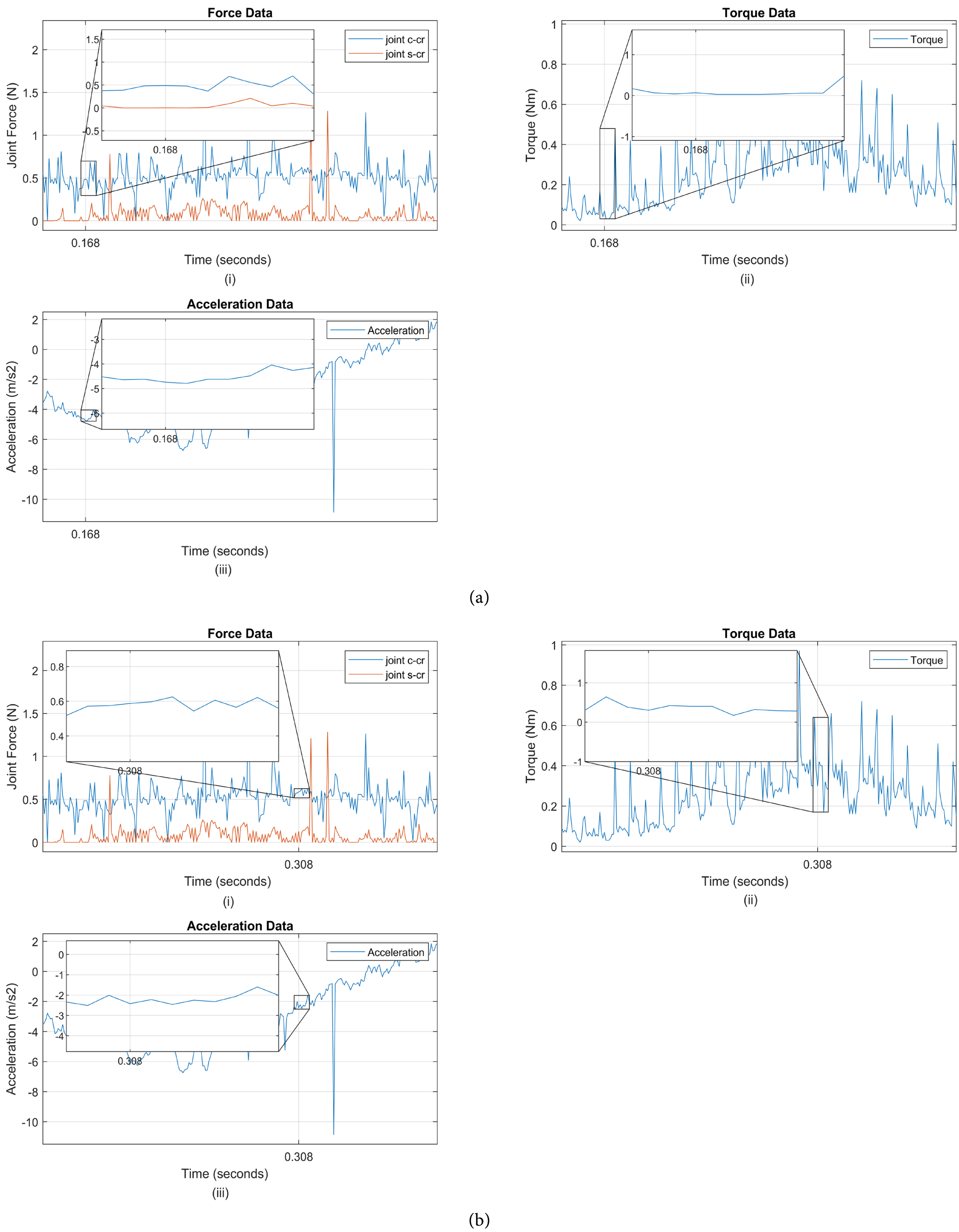


Figure 9. (a) Experimental responses showing contact-free motion with (i)-joint forces, (ii)-crank torque, (iii)-slider acceleration (zoom resolution, t = 0.168 s); (b) Experimental responses showing contact-impact motion with (i)-joint forces, (ii)-crank torque, (iii)-slider acceleration (zoom resolution, t = 0.308 s).

present in the simulation data that typically does not incorporate such noise. In addition, numerous assumptions are made during modelling which does not depict the reality.

Within a single revolute clearance joint, there are three different motion modes. With two revolute clearances, the number of motion modes increases to nine. The findings align with the work of Muvengei *et al.* [4] in terms of joint forces, but there were slight variations in the crank torque and slider acceleration. These discrepancies can be explained by:

1) Friction at the sliding joint was neglected in the simulation.

2) Play at the sliding block could not be fully eliminated. This play could have led to the contamination of the experimental data and introduced noise.

3) Misalignment between journal and bearing elements, always present in actual mechanical systems, were not considered in the numerical simulations.

4) The dynamics of the flywheel were not considered in the numerical simulations.

4. Conclusions

The intricate nature of clearance revolute joints in planar mechanical systems was experimentally explored in this study, using a slider-crank mechanism as a representative model. The dynamics between these joints proves to be highly interrelated; the behavior of one joint undeniably impacts the neighboring ones, further influencing the overall system dynamics. In light of the complex interactions, for a comprehensive understanding and to develop an effective control strategy, it's vital to recognize all these joints as clearance joints.

Our experimental setup confirmed seven out of the nine potential motion modes present in literature for such mechanical systems. The absence of the free-contact motion mode, even from numerical studies, hints at its rarity or possible non-existence. Meanwhile, the missing contact-contact motion mode from our observations may be a result of real-world intricacies, especially the misalignments between the journal and bearing parts, which numerical studies often overlook.

Future Works and Open Problems:

1) Model Refinement: A more refined model that encompasses potential real-world imperfections, such as misalignments and wear and tear, can be developed. This would bridge the gap between experimental observations and numerical simulations.

2) Effect of Material Properties: Investigate how the material properties of the revolute joint components influence the motion modes. Do certain materials or finishes exacerbate or mitigate the clearance effects?

3) Design Interventions: Based on the insights from this study, one can explore design interventions to minimize the adverse effects of clearances. Are there designs that are more tolerant of joint clearances?

4) Advanced Simulation Models: Future numerical simulations can incorporate more sophisticated models that mimic real-world misalignments and fric-

tional effects to predict motion modes more accurately.

5) Interplay of Multiple Joints: Exploring systems with more than two clearance revolute joints can offer insights into the compounded dynamics and possible emergent motion modes.

6) Control Strategies: With a deep understanding of the motion modes, future research can dive into devising control strategies that exploit or mitigate these modes for optimal system performance.

7) It's our hope that these exploratory avenues will catalyze further research in this domain, offering clearer insights and more refined control strategies for mechanical systems with clearance revolute joints.

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Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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