Chapter 13
Electrical Chatter and Modal Response of Pin-Receptacle Contacts in Air and Oil

Marcus Behling, Phillip Misterman, Robert Justin Osborne, Robert C. Flicek, Karl Walczak, Eric Robbins, Robert J. Kuether, and Jonel Ortiz

Abstract Electromechanical switches are commonly used to allow electrical circuits to be mechanically connected and disconnected as needed to meet engineering design requirements. During severe shock and vibration environments, switches are sometimes observed to *chatter*, which refers to abrupt, momentary increases in the electrical resistance across the switch. Chatter is an undesirable phenomenon that is known to adversely impact signal integrity and thus degrade electrical contact performance. This work investigates the occurrence of electrical chatter in a simple pin-receptacle connection when exposed to shock-type loading. The electrical contacts are mounted in an aluminum housing referred to as the chatter fixture. The chatter fixture can be filled with viscous fluids, such as oil, to damp the vibrations and passively control the chatter behavior. In the experiments, the chatter fixture is bolted to a large metal base plate to simulate a stiff boundary condition, together making up the test assembly. This study explores the influence of oil on both the electrical chatter response as well as the modal response of the test assembly using a combination of modeling and experimentation. Experimental results provide insight into the relationship between impact force and chatter response in oil versus in air. A simplified 2D finite element model was developed and calibrated to accurately predict observed chatter responses using the range of inputs measured from the experiments. The results of this investigation demonstrate the difference in chatter performance between electrical contacts submerged in oil versus air and provide insight into the simplified model used to predict chatter in response to shock-type loading.

Keywords electrical chatter · bifurcated receptacle · impact testing · modal analysis · finite element modeling

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Introduction

Electrical switches and relays are used extensively in engineering design to selectively allow or interrupt the flow of electrical power or data signals. Normal operating conditions often subject mechanical components to relatively severe shock and vibration environments, e.g., during aircraft takeoff and landing. These types of severe acceleration environments can cause the electrical contact resistance across a closed switch to increase abruptly, potentially corrupting or preventing the transmission of intended signals. A past study found that chatter can be caused by elastic deformation of the pin and receptacle in a switch [1]. These switches can be submerged in oil as lubricating the interface is known to improve performance, and this study seeks to characterize the chatter of the switch when submerged in oil, as the modal parameters of a structure have been observed to change when submerged in a liquid [2, 3].

As mentioned previously, physical separation between the pin and receptacle can cause chatter [4]. Initial stages of chatter research assumed that the large discrepancy between the frequency of chatter events (megahertz) and the natural frequencies of the contact pair and the test fixture (kilohertz) implied that chatter was not caused by modes of vibration. However, computational analysis by Zastrow [1] demonstrated that modes of vibration in the kilohertz range can cause very brief physical separation between the pin and the receptacle, producing chatter events on the order of megahertz. Although Zastrow's work demonstrated that the resonant modes of vibration do play a critical role in understanding chatter in electrical switches, it focused on contacts in air and did not study the effect of oil.

Johnson [5, 6] developed a novel experimental method for performing modal analysis of a pin and bifurcated receptacle submerged in oil. These experiments faced several unique challenges due to the small dimensions of these parts and the difficulty of exciting them when they are submerged in oil. However, Johnson was able to compute natural frequencies and modal damping values experimentally for the pin and receptacle in air and in oil. Johnson compared the experimental results to the natural frequencies output by a computational model, but this comparison could only be performed for tests in air, as it was not possible to include the influence of oil in the structural dynamics models available at that time.

The primary goal of this study is to understand how chatter characteristics differ for electrical switches submerged in oil versus electrical switches in air. To accomplish this, a test fixture containing the pin-receptacle is excited across a wide range of input force levels and chatter is measured in oil and air. A secondary objective is to calibrate a 2D model of the pin-receptacle to predict the chatter response. A parameter study is performed to understand the effect of each variable on predicted chatter, and experimental data is used to calibrate the model. The 2D model is then used to gain insight into the mechanisms causing chatter.

In the next section, the experimental procedure and test setup are described. In section three, a model of the fixture-pin-receptacle system is introduced, and its parameters are described. In section four, the experimentally obtained chatter results are shown and compared in oil and air. In section five, the model calibration procedure is described in detail, and the selected oil and air parameters are compared. In section six, the experimental results are compared to those predicted by the model. Finally, conclusions are given in section seven.

Experimental Setup

The experimental setup consists of a chatter fixture bolted to a steel base plate to simulate a stiff boundary condition, which is then placed on a rubber mat to isolate it from the table upon which it is placed. The fixture, shown in Figure 1, was machined out of a 6061 aluminum block to 63.5 mm x 63.5 mm x 101.6 mm and designed to hold the pin and receptacle in the inserted state of the contact pair. The cylindrical pin that has dimensions of 0.6865 mm x 6.147 mm is made of Paliney 7, and the approximately rectangular receptacle is 8.66 mm x 1.321 mm x 0.2032 mm and made of NeyoroG. The cavity is filled with Krytox GPL 101 oil that fully submerges the switch. The output of the pair is connected to a National Instruments MyRio chatter tester with a sampling rate of 40 MHz and configured to detect changes in electrical resistance greater than 120 Ohms. Because of this high resistance threshold, it is assumed that chatter occurs when there is physical separation between the receptacle and pin. An array of 500g tri-axis accelerometers is attached to the exterior of the fixture to measure the response of the experimental setup to the applied impact load. Polytec Laser Doppler Vibrometers (LDV) are used to measure the velocity of the pin and receptacle pair to measure local motions on the electrical contacts. A PCB Piezotronics 086C05 modal hammer with a range of 0-22kN is used to excite the test fixture at the red "x" location marked in the center image in Figure 1. Data from the accelerometers and LDVs is collected using a Siemens SCADAS and evaluated in the Impact Testing Module in Testlab.

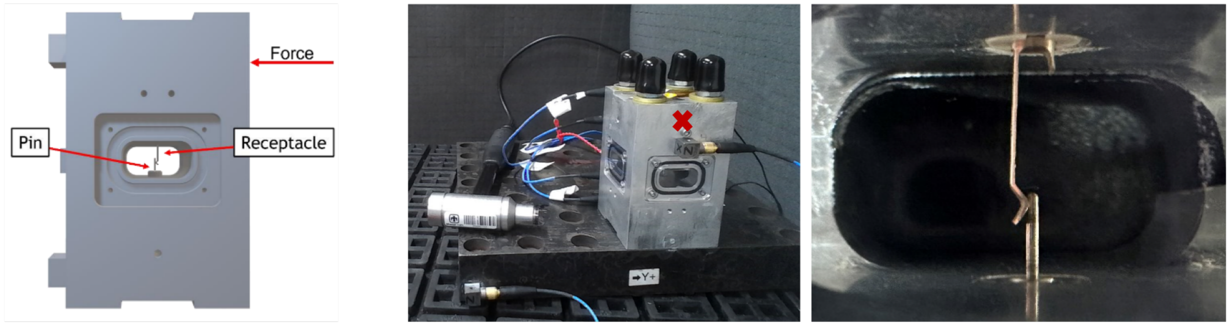

Fig. 1 Test fixture used to study chatter (left), experimental setup (center), and close-up of pin and receptacle installed in the fixture (right)

Modal Analysis and Boundary Condition

To assess the extent to which the fixture has a true fixed base in this configuration, a modal test was performed, revealing that the first elastic mode of the plate occurs at 1356 Hz. Ideally, the fixture would be attached to a larger, more rigid block to better match the desired boundary condition, but this is challenging to achieve in a laboratory setting. Despite the observed resonance of the test setup, the chosen test setup seems to reasonably approximate the boundary condition such that there was an observed first bending (cantilever) mode of the chatter fixture at 1521 Hz, as shown in Figure 2. This mode also appears in a 3D finite element model (FEM) of the test assembly, providing confidence in the understanding of the boundary conditions of the experiment. The chatter fixture was also tested in a free-free condition, in addition to the plate-mounted boundary condition, but the plate-mounted configuration was used for shock tests as chatter was more easily excited in that configuration.

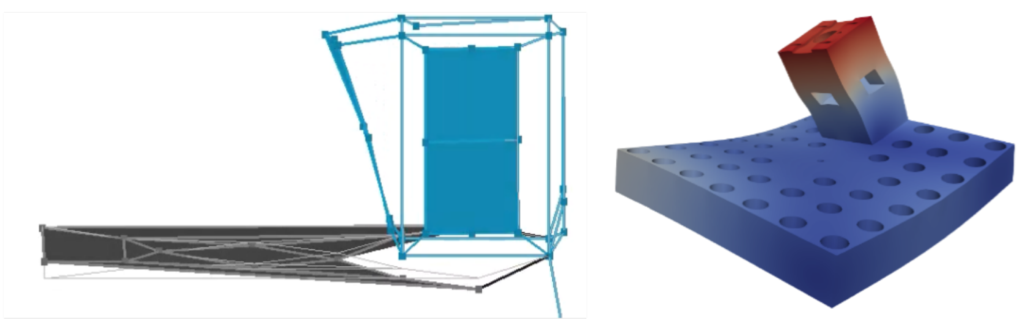

Fig. 2 First fixture bending mode at 1521 Hz (left) and 2^{nd} elastic FEM mode at 1556 Hz

Experimental Method

Chatter events are initiated by impacting the structure with a modal hammer at the location marked by the red “x” in Fig. 1. In past studies, modal shakers were used to attempt to induce chatter, though it was discovered that the hardware limitations of the shakers failed to achieve the objective of observing chatter [7]. The impact location was chosen to readily excite chatter in the direction orthogonal to the pin and receptacle. A drive-point frequency response function (FRF) was taken at the impact location to help calibrate the simplified chatter model as described in Section 5. Hammer hits were performed at various force levels and the resulting chatter events were recorded with the MyRio tester. The number of chatter events and distribution of chatter durations were noted to change as impact amplitude increased. To accurately observe and assess these phenomena, the force amplitudes were divided into bins of 200 N width. The force range of interest was 300-4000 N with an average impact duration of 350 μs. The lower limit was set to 300 N as no chatter events were observed below this input force, and the upper limit was set to 4000 N to prevent damage to the test fixture. A steel hammer tip was used to provide a consistent broadband frequency excitation up to 5000 Hz with less than 10 dB force power spectral density drop-off below this frequency. Thus, the modal hammer input has sufficiently broad frequency bandwidth to excite the relevant modal response of the test assembly.

Chatter measurements were first taken with the pin-receptacle submerged in oil. After sufficient data was collected, the oil was drained from the fixture, and measurements were taken with the pin-receptacle in air. The pin-receptacle was cleaned with a cloth to remove the oil, though there was likely some residual oil on the contact pair. It is likely that the test setup changed slightly in other ways during the draining process - for example, the preload in the pin-receptacle could have been altered - but efforts were made to reduce these differences.

Modeling

A simplified, two-dimensional FEM was previously developed for the purpose of predicting the chatter behavior of the pin and receptacle. A schematic of this model is shown in Figure 3. The chatter fixture is modeled as a single degree-of-freedom mass-spring-damper system, where the natural frequency of the mode is intended to represent that of the first cantilever bending mode of the fixture that is experimentally observed in the test assembly. Note that this model only captures a single mode of the chatter fixture even though there are other test assembly modes in the frequency band of interest. Though a significant simplification, this may be a reasonable assumption as higher order modes have been observed to affect chatter less significantly in [8]. The response of the system to the haversine input force was calculated using a nonlinear Newmark integration scheme which is described in [9]. The response was calculated from 0 to 0.05 seconds, and 50,000 time steps were used to minimize the temporal discretization error in numerical integration and capture small chatter durations predicted by the model.

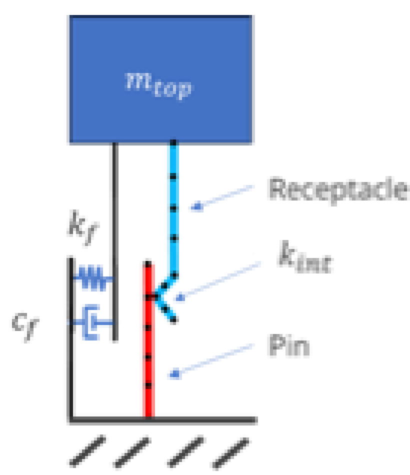

Fig. 3 2D FEM of the fixture, pin, and receptacle

The pin and receptacle represented in Figure 3 are modeled using linear beam elements according to

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\}, \tag{1}$$

where [M], [C], [K] are the mass, damping, and stiffness matrices, respectively, which are calculated using the method presented in [10]. In the preceding equation, {x} is the displacement vector, and {F} is a vector of applied forces, while the overdot represents the derivative with respect to time. The fixture is modeled as a single lumped mass element that is attached to the top of the receptacle and is connected to ground via a linear spring element (note that the viscous dashpot was set to zero). The base node of the pin is connected to ground.

The contact nonlinearity of this system is captured in the interface force between the pin and receptacle node pair, which is assumed to be Hertzian as derived in [11]. The following form is assumed,

$$F_{int} = e \cdot k_{int} \cdot \delta^{3/2}, \tag{2}$$

Table 1 Model input parameters

Parameter	Symbol	Units
Fixture stiffness	k_f	N/mm
Interface stiffness	k_{int}	N/mm
Target frequency	f_{target}	Hz
Fixture damping ratio	ζ	-
Impact duration	t_{impact}	S
Impact amplitude	F_{peak}	lb

where k_{int} is the interface spring constant, e is a boolean vector with appropriate entries to apply the contact force to the FEM, and δ is the relative displacement in the normal direction between the pin and receptacle contact node. Note that the beam elements are meshed in their undeformed configuration, so there is initial overlap in the model. A quasi-static preload simulation is required to first find the static equilibrium position of the system such that the static equilibrium satisfies

$$[K]\{x_{pre}\} + F_{int}(x_{pre}) = 0. \tag{3}$$

This is achieved by applying an outward force to the receptacle at the contact node and solving the quasi-static equations as the force ramps to zero using a damped Newton method. From this equilibrium position, the natural frequencies and mode shapes of the system around this linearized state were then solved according to the linearized eigenvalue problem

$$\left(K + \left.\frac{\partial F_{int}}{\partial x}\right|_{x_{pre}} - \omega^2 M \right) \phi = 0. \tag{4}$$

Rayleigh damping is used in this model, following the form

$$[C] = \alpha [M] + \beta [K], \tag{5}$$

where the α and β coefficients are calculated according to

$$\begin{bmatrix} \alpha \\ \beta \end{bmatrix} = \begin{bmatrix} \frac{1}{2\omega_1} & \frac{\omega_1}{2} \\ \frac{1}{2\omega_2} & \frac{\omega_2}{2} \end{bmatrix}^{-1} \begin{bmatrix} \zeta_1 \\ \zeta_2 \end{bmatrix}. \tag{6}$$

In the previous equation, $\omega_1$and ω_2 are the first two natural frequencies solved for in Eq. (4), and ζ_1 and ζ_2 are the corresponding viscous damping ratios. These damping values were assumed to be equal and are specified in the model as $\zeta = \zeta_1 = \zeta_2$.

The final nonlinear dynamic model used to simulate the time-varying contact force in the pin and receptacle due to an externally applied force is written as

$$\begin{gathered} [M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} + F_{\text{int}}(x) = \{F\} \\ x(t=0) = x_{\text{pre}} \\ \dot{x}(t=0) = 0 \end{gathered} \tag{7}$$

Chatter duration was calculated according to the time that the pin and receptacle were out of contact and the interface force is equal to zero, i.e.,

$$t_{chatter} = \sum_{i=1}^{n} t_{i,\delta>0}, \tag{8}$$

where $t_{i,\delta>0}$ is the i^{th} time segment where $\delta > 0$, indicating that the pin and receptacle are out of contact.

The parameters that inform the 2D model are fixture stiffness k_f, interface stiffness k_{int}, fixture damping ζ, target frequency f_{target}, impact duration t_{impact}, and impact amplitude F_{peak}; these are listed in Table 1 for ease of reference.

Experimental Results

The relationship between impact amplitude F_{peak} and chatter response was investigated first for the experiment involving the electrical contacts submerged in oil. The number of observed chatter events, along with the total duration of chatter, is plotted against the impact force F_{peak} in Figure 4. There appears to be an exponential relationship between peak input force and the number and total duration of chatter events below ∼800 N in oil, and the relationship is fairly linear above this force. Some of the variability in the data may be attributed to changes in pulse width t_{impact} between hits.

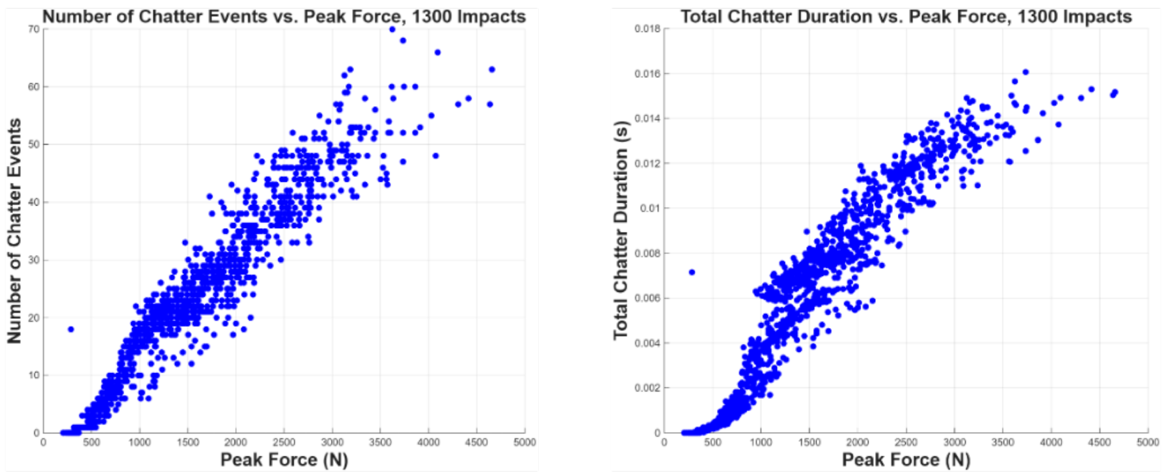

Fig. 4 Number of chatter events (left) and total chatter duration (right) versus impact amplitude in oil

The same set of chatter tests was performed with the oil drained, and these results are shown in Figure 5; as shown in this figure, more chatter events are observed in air than oil at high impact amplitudes, i.e., above 2000 N. Below this force level, there are actually more chatter events when the switch is submerged in oil, suggesting that the switch performs better in oil at high impact amplitudes but worse at lower amplitudes. In air, there is a significant increase in the number of chatter events between 1500 and 2500 N, and a similar, more understated, phenomenon is observed in oil between 500 and 1000 N.

Note that the primary intended difference between the test assemblies used for the oil and air experiments was the presence or absence of oil in the internal cavity of the chatter test fixture. However, in addition to this intended difference between the two test configurations, it is also possible that other uncontrolled factors changed between tests. These potentially uncontrolled factors are numerous, but include different bolt torque, preloaded state, or altered part alignment. Since it is not possible to isolate the effect of test-to-test variability with the present dataset, some of the observed differences between the chatter behavior in oil and air may reflect test-to-test variability.

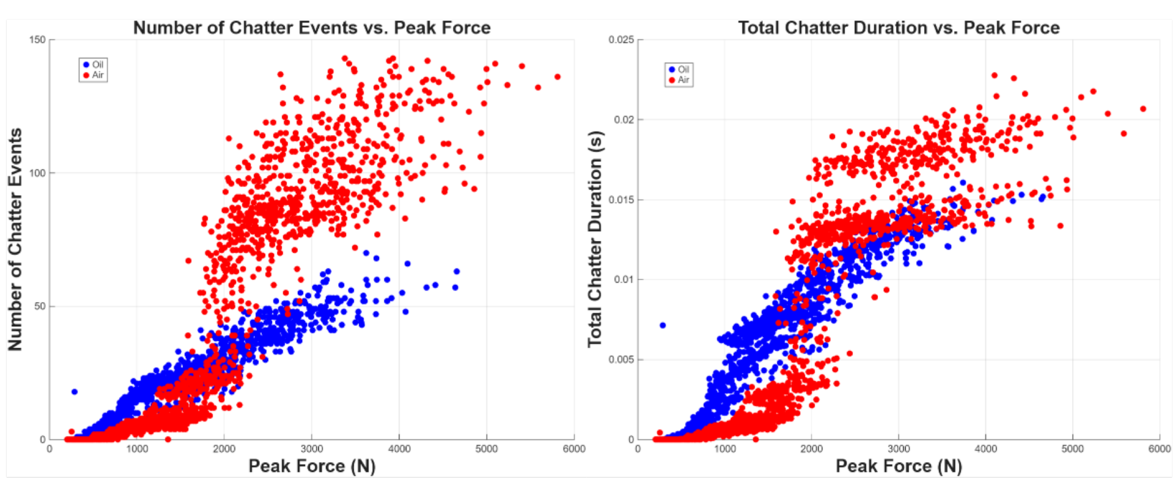

Fig. 5 Number of chatter events (left) and total chatter duration (right) vs. peak force for experiments and models in oil and air

The distributions of measured chatter durations are shown in Figure 6. They are skewed towards lower values at low force levels and are bimodal at high force levels. This suggests that there may be two distinct mechanisms causing chatter, one excited at higher force levels and the other at low force levels.

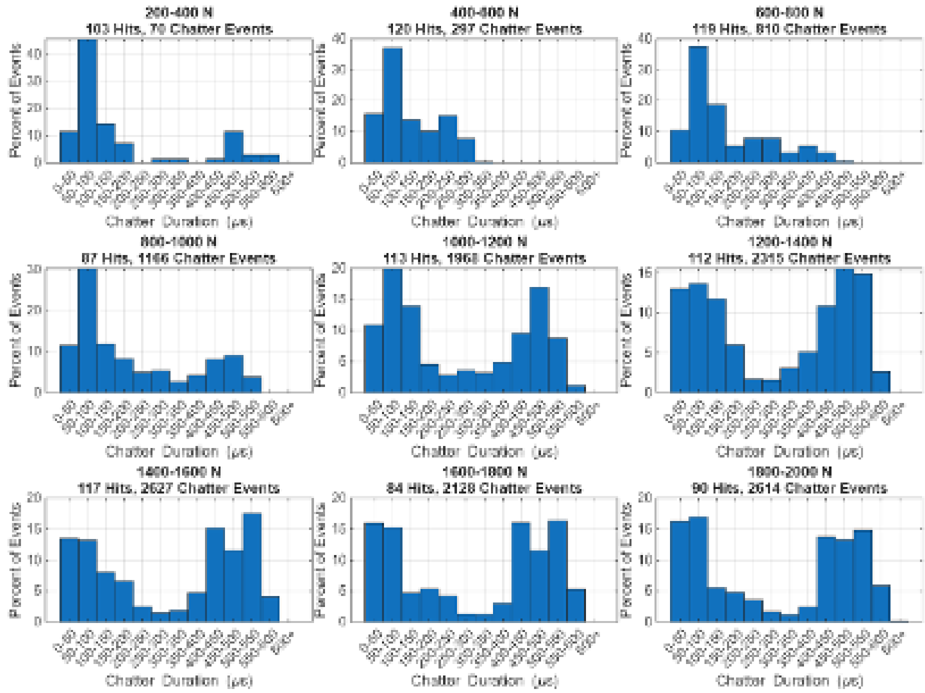

Fig. 6 Distributions of measured chatter durations in oil at varying force levels

A plot of the time history showing the chatter state measured in the experiment is shown in Figure 7 for a 500N (red) and 1928N (blue) impact. Earlier in the time response, the chatter events are longer in duration and occur in rapid succession. As the response decays, the chatter events are shorter in duration with longer time intervals between them. This supports the distributions shown in Fig. 6, as there again seem to be two different chatter regimes when the switch is excited at a high force level.

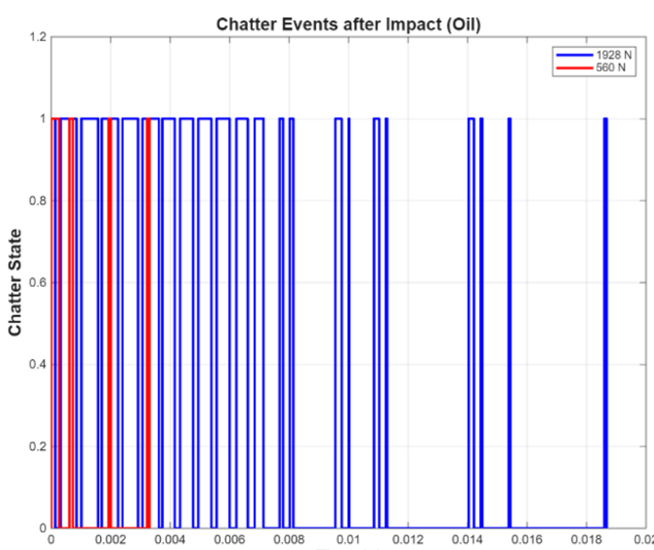

Fig. 7 Measured chatter events in oil following impacts, where “0” denotes contact and “1” denotes loss of contact

Model Calibration and Parameter Studies

Several experimental measurements were used to calibrate the simplified 2D model to match the conditions of the test. The amplitude of the impact applied to the fixture in the experiments was measured with a load cell, and the applied pulses were recorded. The impact duration was noted to vary slightly from one impact to the next but remained fairly constant, as shown in Figure 8. The impact duration was set to 350 μs to match the average value observed in the tests. The haversine force profile with this duration is overlaid on the measured forces in Figure 8. The variability in impact duration contributes to the spread of the chatter data observed in Section 4. With this determination from the tests, the input force could now be parameterized by the amplitude of the applied force.

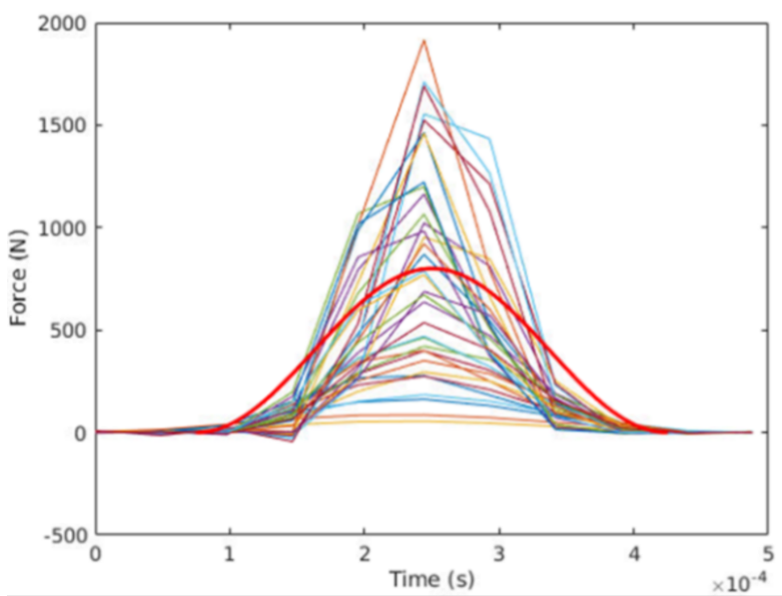

Fig. 8 Measured impact forces and haversine input used in the model (red)

It was also desired to match the natural frequency of the chatter fixture bending mode, as this mode is expected to significantly contribute to chatter, producing relative displacement between the base of the pin and that of the receptacle. The target frequency of the model f_{target} was selected to match the natural frequency of the first fixture bending mode which occurs at 1521 Hz. The fixture stiffness k_f was initially estimated using a 3D FEM but was calibrated, along with fixture damping ζ, such that the top mass drive-point FRF matched the measured drive-point FRF as shown in Figure 9. Intuitively, ζ was observed to change the width and height of the first peak. The experimentally measured damping was used as a starting point for this value but was lowered slightly to better match the chatter characteristics. As a result, the model's peak is a bit higher than that of the experimental FRF. Increasing the fixture stiffness k_f was observed to reduce the magnitude of the FRF uniformly with respect to frequency. It is noted that the FRFs do not match well past 2000 Hz as the model is only capable of capturing the first bending mode of the chatter fixture.

Parameter studies were performed by simulating the response of the 2D FEM with a selected set of parameters from 300-4000 N in increments of 50 N. The parameters were then modified, and the same process was performed to understand the effects of each parameter on the number and duration of chatter events. These effects were observed qualitatively and are shown in Table 2. In addition to the number and duration of chatter events, the table also notes the effect of the parameters on the chatter threshold, or force at which chatter starts to occur, as well as the distribution of the chatter durations. Interface stiffness and fixture damping were noted to affect the model most significantly, with interface stiffness primarily affecting the chatter threshold, and fixture damping primarily changing the slope of the curve. Using this information, this process was repeated, modifying parameters each time, until a satisfactory match was obtained between the model and experiment.

The last unknown parameter is interface stiffness of the contact penalty spring, k_{int}. Because it does not noticeably affect the shape of the drive-point FRF, it is chosen to match chatter characteristics, specifically the number and total duration of chatter events, as a function of force. The sets of parameters that best fit experimental data for the oil and air are shown in Table 2. The fixture stiffness k_f, target frequency f_{target}, and pulse width t_{impact} remain the same in oil and air, while the interface stiffness k_{int} and fixture damping ζ change significantly; the former doubles between oil and air, and the latter decreases when oil is removed.

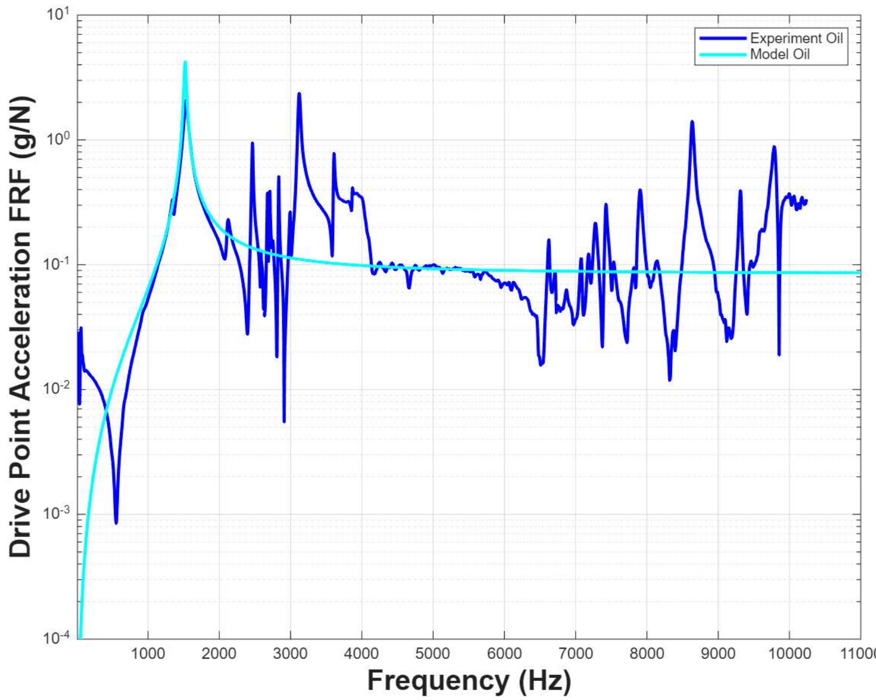

Fig. 9 Experimental (blue) and model (cyan) drive-point FRFs at top of fixture

Table 2 Observed effect of parameters on number and duration of events, chatter threshold, and chatter duration distribution

	Fixture Stiffness $\uparrow k_f$	Interface Stiffness $\uparrow k_{int}$	Target Frequency $\uparrow f_{target}$	Fixture Damping $\uparrow \zeta$	Impact Amplitude $\uparrow F_{peak}$	Impact Duration $\uparrow t_{impact}$
Number of Chatter Events N_{events}	$\downarrow$	$\downarrow$	$\downarrow$	$\downarrow$	$\uparrow$	$\uparrow$
Chatter Duration t_{dur}	$\downarrow$	$\downarrow$	$\downarrow$	$\downarrow$	$\uparrow$	$\uparrow$
Chatter Threshold	$\uparrow$	$\uparrow$	$\uparrow$	$\uparrow$	**N/A**	$\downarrow$
Chatter Distribution	**Less bimodal**	**Less bimodal**	**Less bimodal**	**Less bimodal**	**More bimodal**	**More bimodal**

Table 3 Calibrated model parameters for oil and air

	Fixture Stiffness k_f (N/mm)	**Interface Stiffness k_{int} (N/mm)**	**Target Frequency f_{target} (Hz)**	**Fixture Damping Ratio ζ**	**Pulse Width t_{impact} (s)**
Oil	110,406	7,000	1521	0.008	3.5e-4
Air	110,406	14,500	1521	0.005	3.5e-4

The change in damping ζ from .008 in oil to .005 in air seems reasonable given that oil is more viscous than air. The difference in interface stiffness k_{int} is less intuitive, and the cause of this change is unknown, though it may reflect the changes incurred in the draining process mentioned previously.

Model-Experiment Comparison

As seen in Figure 10, the 2D models accurately predict the number of chatter events as a function of force, suggesting that the model is robust and can account for changes in the medium in which the switch operates. No mass-loading term was included to account for the fluid and the model still performed quite well. The 2D model is very computationally efficient, running in about 25 seconds per simulation.

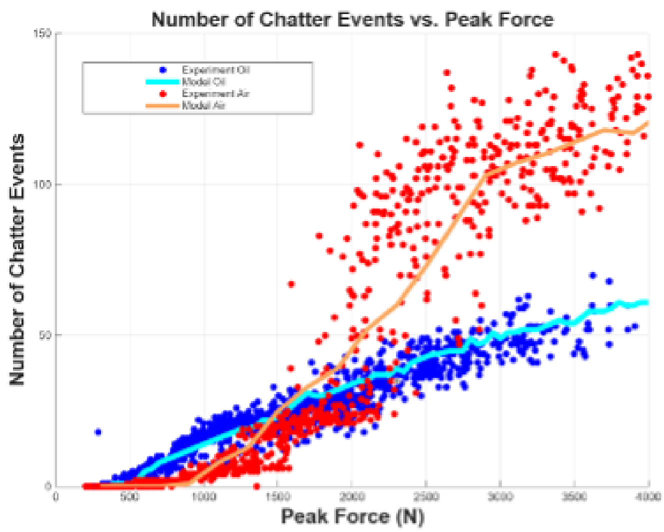

Fig. 10 Number of measured and modeled chatter events in oil and air

The distributions of chatter event duration in the calibrated oil model are shown in Figure 11. The model fails to capture the bimodal nature of the distributions of the experimental results shown in Figure 6. Though, it correctly predicts a large proportion of longer duration events, it significantly under-predicts the number of short duration events. One theory is that the long duration events are primarily caused by the bending motion that the model captures, whereas the short duration events may be caused by motion not captured in the model: sliding, lateral shearing, or torsional, for example.

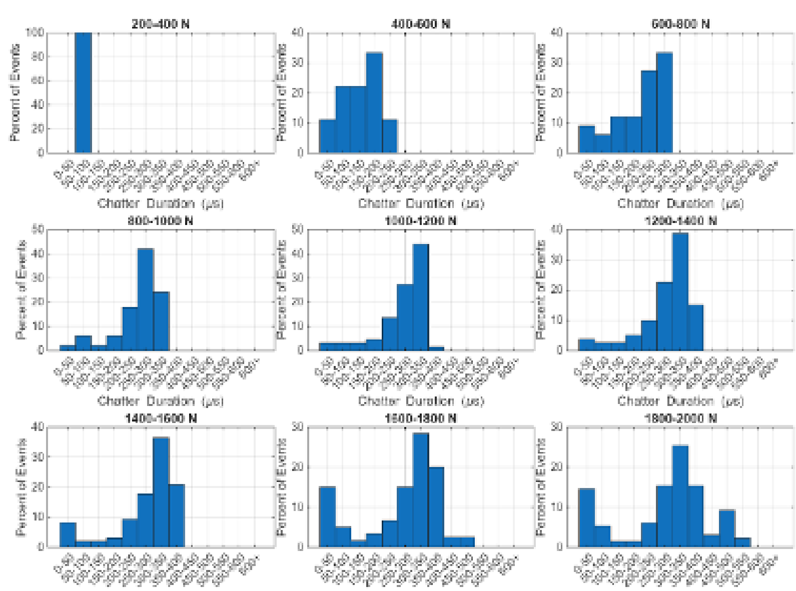

Fig. 11 Distributions of model chatter durations in oil at varying force levels

Conclusion

In this study, chatter was characterized in an electrical switch submerged in oil. This was done by impacting the chatter fixture in which the switch was installed and recording changes in resistance using a National Instruments MyRio system. Chatter events were recorded for over 1,000 hammer impacts to the system at various force levels. The data revealed that below a certain force threshold, no chatter occurred. Above this first threshold, the amount of chatter increases at a super-linear rate. As the input force increases further, another threshold is reached, above which the amount of chatter increases linearly with increasing input force.

Following the initial series of experiments in oil, the oil was removed from the fixture and the experiment was repeated with over 1,000 impacts at different force levels. The chatter data collected in air followed a similar pattern as described above, being broken into three regimes: no chatter at low forces, super-linearly increasing chatter at moderate forces, and linearly increasing chatter at high forces. However, the number of chatter events for a given input force level differed substantially for the tests in oil and in air.

A simple 2D model was used to accurately predict chatter behavior in the pin-receptacle system in the presence of air and oil. The simplicity of the model allowed us to efficiently investigate how its parameters affect chatter, and limited experimental data is required to select the model's parameters. A drive-point FRF at the impact location was used to calibrate the parameters related to the fixture's dynamics. Although many chatter measurements were obtained at various force levels during testing, the model parameters could be tuned with far fewer chatter measurements. The computational efficiency of the 2D model and limited measurements required to tune it allow chatter to be characterized quickly and efficiently, allowing for faster design iteration and ultimately, improved signal transmission.

Lastly, chatter duration was observed to follow a bimodal distribution at high force levels, suggesting two distinct chatter mechanisms. At high response levels, the model had similar distributions to those measured, suggesting that chatter is governed primarily by the fixture bending mode at higher levels. At lower response levels, the model predicted fewer chatter events than were observed, suggesting that it fails to capture a portion of the chatter events observed experimentally.

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References

1. B. G. Zastrow, R. C. Flicek, K. M. Johnson, K. A. Walczak, B. R. Pacini, B. Johnson, C. Schumann and F. Rafeedi, "Investigation of electrical chatter in bifurcated contact receptacles," in 2021 IEEE Holm Conference on Electrical Contacts, San Antonio, TX, USA, 2021.
2. G. Moraga, X. Xia, S. Roig, C. Valero, D. Valentín, M. Egusquiza, L. Zhou, E. Egusquiza, A. Presas, Experimental study on the influence of vibration amplitude on the fluid damping of a submerged disk, Journal of Sound and Vibration, Volume 569, 2024, 118099, ISSN 0022-460X, https://doi.org/10.1016/j.jsv.2023.118099.
3. D. G. Gorman, I. Trendafilova, A. J. Mulholland, J. Horáček, Analytical modelling and extraction of the modal behaviour of a cantilever beam in fluid interaction, Journal of Sound and Vibration, Volume 308, Issues 1–2, 2007, Pages 231-245, https://doi.org/10.1016/j.jsv.2007.07.032.
4. J. Brady, "Anomalous Behavior of Reed Relays Under Accelerated Life-Test Conditions," in *IEEE Transactions on Parts, Hybrids, and Packaging*, vol. 13, no. 1, pp. 51-60, March 1977, doi: 10.1109/TPHP.1977.1135170.
5. K. M. Johnson, "Characterization of a Small Electro-Mechanical Contact Using LDV Measurement Techniques (SAND2018-1206C)," Sandia National Laboratories, Albuquerque, NM, USA, 2018.
6. K. M. Johnson, "Characterization of a small electro-mechanical contact using non-conventional measurement techniques," University of New Mexico, Albuquerque, NM, USA, 2017.
7. B. Furman and L. Nester, "Final Presentation: Investigation of Electrical Contact Chatter in Pin-Receptacle Contacts," Sandia National Laboratories, Albuquerque, NM, USA, 2023.
8. C.C. Lo, A cantilever beam chattering against a stop, Journal of Sound and Vibration, Volume 69, Issue 2, 1980, Pages 245-255, https://doi.org/10.1016/0022-460X(**??**)90609-4.
9. K. Subbaraj and M. A. Dokainish, A Survey of Direct Time-Integration Methods in Computational Structural Dynamics – II. Implicit Methods, Computers & Structures, Volume 32, Issue 6, 1989, Pages 1387-1401
10. R. Craig and A. Kurdila, *Fundamentals of Structural Dynamics*, 2nd ed. Wiley, 2006.
11. K.L. Johnson, *Contact Mechanics*. Cambridge: Cambridge University Press, 1985.