STABILITY ANALYSIS OF A CONSTRAINED-MOTION DYNAMOMETER

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ABSTRACT
This paper describes the stability analysis of a constrained-motion dynamometer (CMD) for milling force measurement. Multi-axis piezoelectric dynamometers are a popular choice for cutting force measurement. These dynamometers rely on specific piezoelectric transducer arrangements and the structural dynamics of the system, while stiff, are not rigid. Although these systems offer a large measuring range, high sensitivity, and fast response time, the systematic errors caused by their complex structural dynamics must generally be compensated using advanced post-processing techniques. Here, the system structural dynamics are a principal element in the design space which are easily altered with material selection, flexure element geometry and arrangement. In milling applications, a limitation on the allowable axial depth of cut is regenerative chatter. With the ability to modify the dynamic response of a cutting force dynamometer, the stability limit and bandwidth can be modified either through an increase in stiffness or damping. In this paper, a passive damping approach was selected to modify the viscous damping ratio to increase the critical stability limit of a CMD compared to the undamped design.

KEYWORDS
Force, measurement, milling, chatter

INTRODUCTION
There are many types of passive damping approaches, including: shear film damping, constrained layer damping (CLD), damping by addition of energy absorbing foams and viscoelastic materials, and dynamic absorbers [1]. For this application, a viscoelastic material was deposited between the CMD flexure elements and the mechanism frame. When milling force is applied to the CMD, energy is dissipated through the viscoelastic medium due to the relative motion between the movable platform and mechanism frame. The advantage of this approach is the ability to obtain a relatively high amount of energy dissipation in a flexure mechanism without compromising the stiffness or mass of the dynamometer. Additionally, it provides a general strategy for reducing the vibration amplitude at low cost. For this study, a widely available silicone rubber sealant was used to demonstrate its benefits as a passive damping medium for the CMD.

MILLING STABILITY VALIDATION

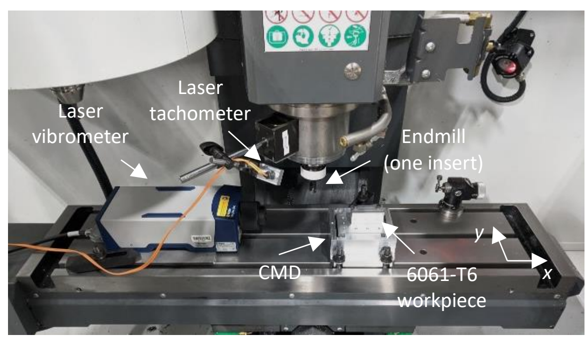

FIGURE 1. Experimental setup for stability testing.

Cutting tests were performed on a Haas TM-1 CNC vertical milling machine; see Fig. 1. A 6061-T6 aluminum workpiece was mounted to the CMD. A single flute, 15.88 mm diameter endmill (Kennametal M1D062E1401W075L150) was used to perform up-milling machining passes at a 3 mm radial depth of cut and a variable axial depth of cut. Once-per-tooth sampling was achieved using a laser tachometer, where a reflective target was attached to the rotating tool holder. In situ vibration signals were collected using the laser Doppler vibrometer. The tool and workpiece FRFs were measured by impact testing, where an instrumented hammer is used to excite the structure and the response is measured using a linear low-mass accelerometer. The FRFs are displayed in Figs. 2 and 3 for an Al 6061-T6 CMD and a damped Al 6061-T6 CMD, respectively. The modal parameters for the CMD x-direction are presented in Table 1. The addition of the silicone medium resulted in a 130% increase in damping with negligible change to the modal mass and stiffness parameters.

Table 1. X-direction modal parameters for the SDOF CMDs.

Modal Parameters	Al 6061-T6 CMD	AL 6061-T6 CMD (damped)
Direction	*x*	*x*
m (kg)	0.689	0.701
k (N/m)	2.08×10^7	2.07×10^7
c (N-s/m)	43	99

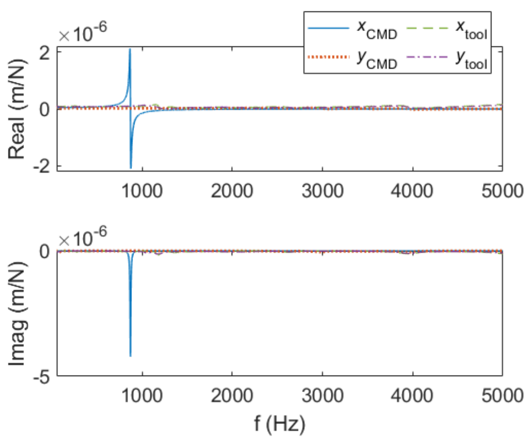

FIGURE 2. Modal fit FRFs of the tool and CMD (no added damping) for the stability validation setup.

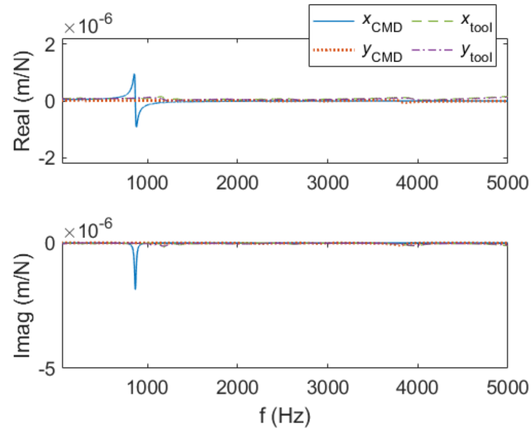

FIGURE 3. Modal fit FRFs of the tool and CMD (damped) for the stability validation setup.

The tool description and cutting conditions for the stability validation is presented in Table 2. The dominant system dynamics are provided in Table 1 for the Al 6061-T6 CMDs. The natural frequency for the CMD's x-direction changed slightly after each cut because the material was removed from the workpiece; therefore, the workpiece was replaced after every test cut. The Al 6061-T6 workpiece dimensions were nominally 30 mm x 70 mm x 30 mm with a mass of 164 ± 1 g.

Table 2. Tool description and cutting parameters for stability trials.

Diameter (mm)	Teeth	Insert material
15.88	1	Coated carbide (Kennametal EC 1402FLDJ)
Cutting parameters for up milling tests		
Spindle speed (rpm)	4900	
Feed per tooth (mm)	0.1	
Axial depth (mm)	Various	
Radial depth (mm)	2 (19% radial immersion)	

To explore the stability behavior in detail, a grid of time-domain simulations was completed at spindle speeds of 4000 rpm to 5000 rpm (10 rpm steps) and axial depth from 0.1 mm to 20 mm (0.1 mm steps). The stability behavior was automatically determined by a synchronous sampling strategy. The results of this strategy are represented by the stability maps in Figs. 4 and 5. Stable cutting behavior can be identified from periodically sampled points. If the points repeat with each revolution, then only forced vibration is present and the cut is stable. If the points do not repeat with each revolution, then either secondary Hopf or period-n bifurcations are present. The stability metric is defined by Eq. 1:

$$M = \sum_{i=2}^{N} \frac{|x_s(i) - x_s(i-1)|}{N} \qquad (1)$$

where, x_s is a vector of sampled x direction displacements and N is the number of samples. For a stable cut, the absolute value of the difference between subsequent points is ideally zero; as a result, their normalized sum is also ideally zero [2, 3].

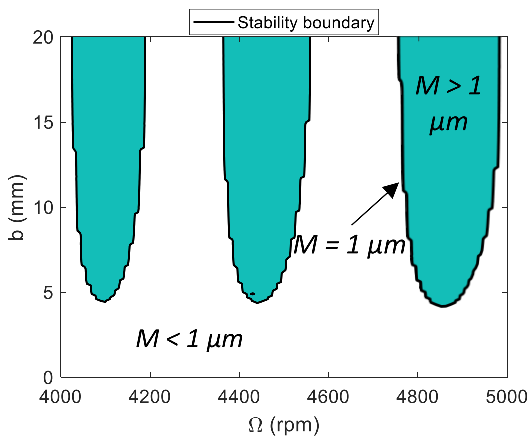

FIGURE 4. Time-domain stability limit (TDS), solid black line with teal infill, expressed as a function of spindle speed, Ω, for the Al 6061-T6 CMD (no added damping).

For an unstable cut, the difference between subsequent points is nonzero and their normalized sum is greater than zero [2, 3]. For this research, the limiting value of the metric, M, was selected to be 1 µm. To interpret the stability map, stable zones are represented by the white area bounded by the stability contour, where M < 1 µm; the stability boundary is represented by the dark stability contour, where M = 1 µm; and unstable zones are represented by the dark green region bounded by the stability contour, where M > 1 µm.

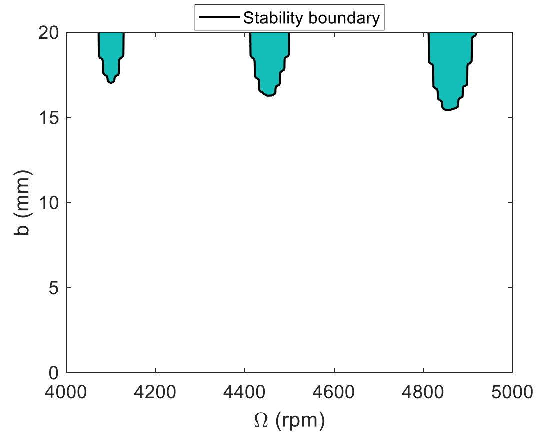

FIGURE 5. Time--domain stability limit (TDS), solid black line with teal infill, expressed as a function of spindle speed, Ω, for the Al 6061-T6 CMD (damped).

RESULTS

The milling stability validation proceeded by completing tests cuts on the Al 6061-T6 CMD until the critical depth of cut was reached. Afterwards, the same procedure was performed for the damped Al 6061-T6 CMD to demonstrate the increase in the stability limit due to the modification of the viscous damping coefficient by the addition of the passive damping medium.

To establish stability, the CMD x-direction displacement and velocity signals were sampled once per revolution at the spindle rotating frequency. This periodic sampling approach was used to determine if the milling response was synchronous with the spindle rotation or not by constructing Poincaré maps (the periodically sampled displacement versus the periodically sampled velocity) for both experiment and prediction. To interpret these Poincaré maps, if the cut is stable (i.e., it exhibits forced vibration only), the data repeats with each spindle revolution and the sampled points appear at one location. If self-excited vibration (chatter or secondary Hopf bifurcation) occurs, however, an elliptical distribution of sampled points is observed due to the presence of both the chatter frequency and the tooth passing frequency and its harmonics. The cutting test cases for the Al 6061-T6 CMDs are presented in Tables 3 and 4, respectively. The displacement and velocity profiles for the first table entry in Table 3 {4900 rpm, 1 mm} is presented in Fig. 6 to demonstrate the good agreement between the simulated (top plot) and experimental (bottom plot) results. Note that the entry and exit transients were removed before plotting.

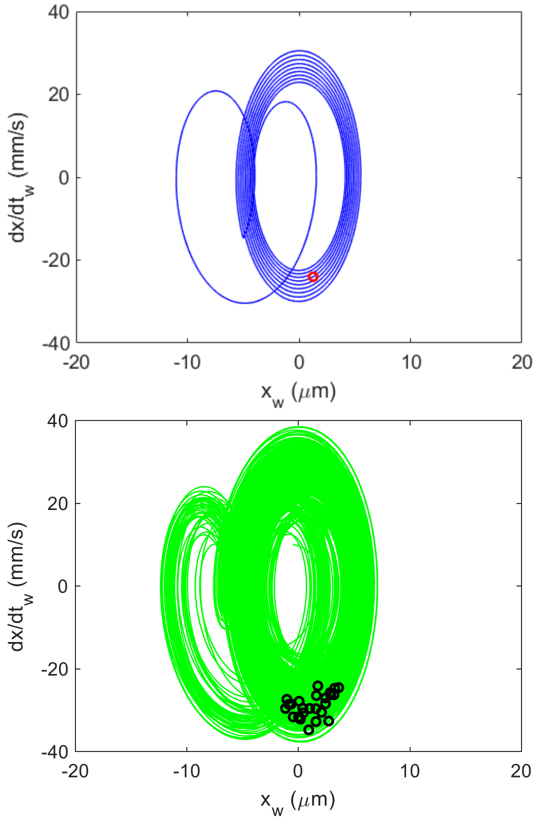

FIGURE 6. Al 6061-T6 (no added damping) Poincaré maps for stable cutting at {4900 rpm, 1 mm}. Predicted (top) and measured (bottom).

Good agreement is observed in each case. It was found that the Al 6061-T6 CMD had a critical stability limit of approximately 4.3 mm. For cases of regenerative chatter (see Fig. 7), the displacement amplitude approaches, and in some cases exceeds, the permissible displacement before exceeding the yield strength for the flexure leaf elements. Therefore, the CMD's critical stability limit is directly dependent on the flexure element design parameters.

Next, the stability results are presented for the damped AL 6061-T6 CMD, see Figs. 8 and 9. Again, good agreement between prediction and measurement was observed at all test locations. There was a significant increase in the stability limit with respect to its undamped counterpart. The maximum allowable depth of cut for the cutting tool was reached without an unstable result. The critical depth of cut for the damped Al 6061-T6 CMD was predicted to be 15.4 mm. The ability to modify the dynamic response of the dynamometer is unique to the proposed CMD and can be leveraged in future dynamometer designs.

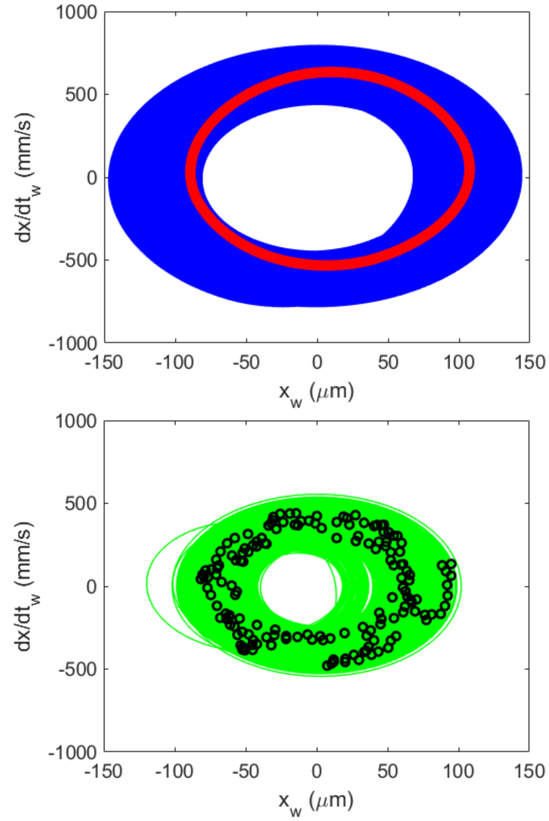

FIGURE 7. Al 6061-T6 (no added damping) Poincaré maps for stable cutting at {4900 rpm, 6 mm}. Predicted (top) and measured (bottom).

Table 3. Cutting conditions and stability metrics for the Al 6061-T6 CMD (no added damping).

Spindle speed (rpm)	Axial depth (mm)	Radial depth (mm)	Metric, M (μm)
4900	1	3	0.32
4900	2	3	0.57
4900	3	3	0.51
4900	4	3	1.16
4900	5	3	58.80
4900	6	3	49.35

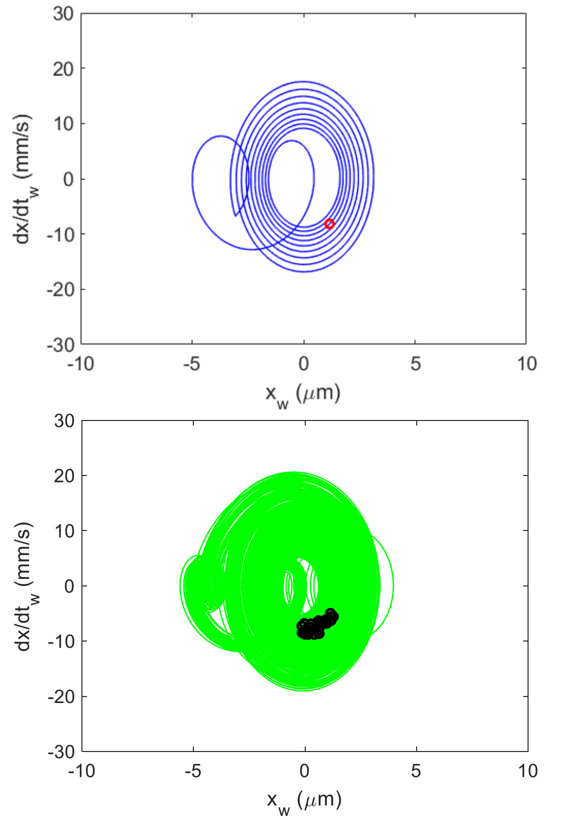

FIGURE 8. Al 6061-T6 (damped) Poincaré maps for stable cutting at {4900 rpm, 1 mm}. Predicted (top) and measured (bottom).

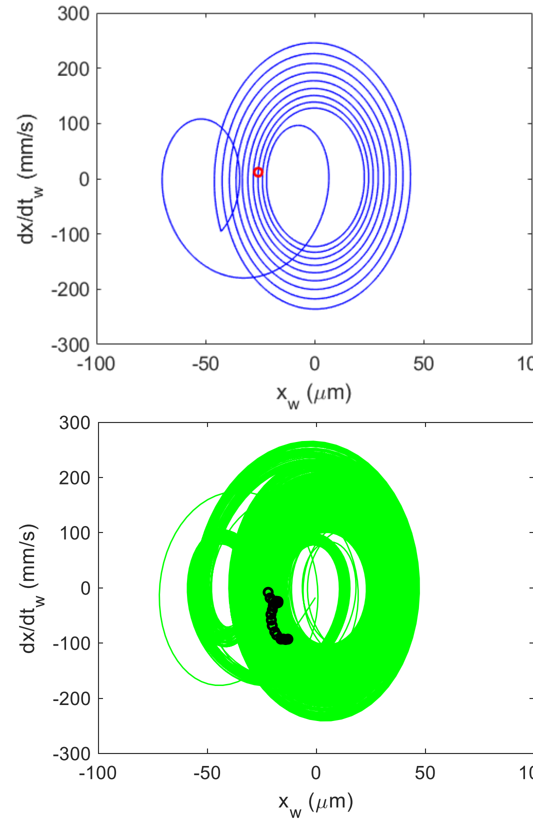

FIGURE 9. Al 6061-T6 (damped) Poincaré maps for stable cutting at {4900 rpm, 14 mm}. Predicted (top) and measured (bottom).

Table 4. Cutting conditions and stability metrics for the Al 6061-T6 CMD (damped).

Spindle speed (rpm)	Axial depth (mm)	Radial depth (mm)	Metric, M (µm)
4900	1	3	0.10
4900	2	3	0.20
4900	3	3	0.18
4900	4	3	0.24
4900	5	3	0.24
4900	6	3	0.32
4900	7	3	0.33
4900	8	3	0.30
4900	9	3	0.88
4900	10	3	0.31
4900	11	3	0.73
4900	12	3	0.36
4900	13	3	0.34
4900	14	3	0.83

CONCLUSIONS

In all machining applications, a limitation on the allowable axial depth of cut is regenerative chatter. Available dynamic force measurement (dynamometer) systems often rely on specific piezoelectric transducer arrangements and the structural dynamics of the system are often assumed to be rigid. Here, the system structural dynamics are a principal element in the design space which are altered by material selection, flexure element geometry, and flexure element arrangement [4].

With the ability to modify the dynamic response of a cutting force dynamometer, the stability limit and bandwidth can be modified through an increase in stiffness or damping. A passive damping approach was used to increase the critical stability limit for an Al 6061-T6 CMD compared to its original counterpart. It was shown that the viscous damping coefficient increased by 130%. As a result, the milling process model simulation revealed that the critical stability limit increased from 4.3 mm to 15.4 mm at a selected spindle speed. The simulation results were validated using measured displacement and velocity signals to construct Poincaré maps. Good agreement was observed for all test cases. The milling stability tests are summarized in Fig. 10 and Fig. 11 for the undamped and damped Al 6061-T6 CMDs, respectively [4].

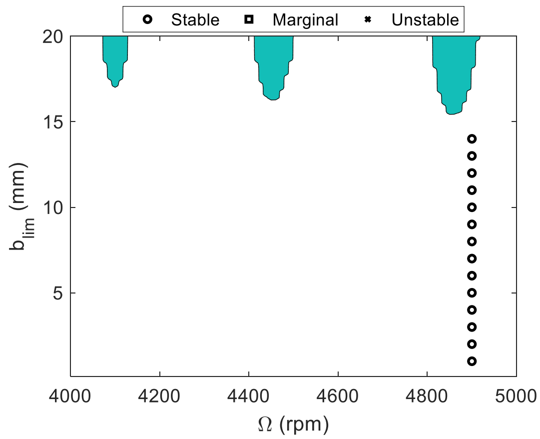

FIGURE 10. Milling stability validation for the Al 6061-T6 CMD (no added damping).

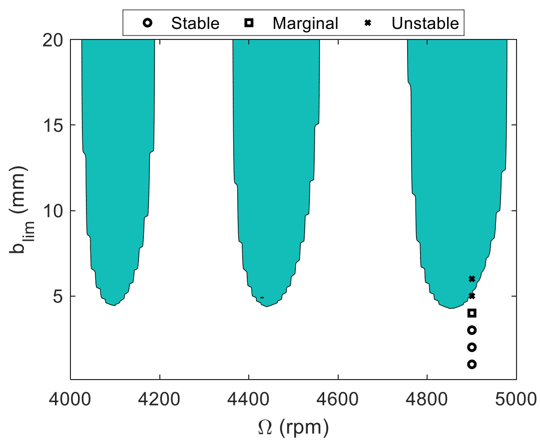

FIGURE 11. Milling stability validation for the Al 6061-T6 CMD (damped).

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