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Cutting Force Measurements in High Speed Milling – Extension of the Frequency Range of Kistler Dynamometer

D. Bachrathy[1]
G. Stepan[1]
[1] Department of Applied Mechanics, Budapest University of Technology and Economics, Budapest, H-1111, Hungary

Abstract. The knowledge about cutting force characteristic is a key factor in process optimization of cutting processes. It determines the loads on the cutting edges and the power consumption of the spindle. The derivative of the cutting force function with respect to the chip thickness strongly influences the appearance of the so-called chatter vibration. The cutting force function is usually based on a series of cutting tests for different feed rates. The function is fit only onto the averages of the measured cutting force signals. In the presented method the radial and the tangential cutting force is determined for each angular position of the cutting edge and plot as a function of the theoretical chip thickness. This way, the cutting force function can be identified in a wide range of the chip thickness with a single measurement. However, this method is limited to smaller spindle speeds (e.g. under ~4 000 rpm) due to the measureable frequency range of the force sensors (~1 kHz). In this study, we also present an advanced compensation method for a 6-channel Kistler dynamometer, which is based on a 6-degree-of-freedom dynamic transformation. The transfer function between the measured forces/moments and the modal hammer excitation is determined at a number of different locations. With the proposed dynamic compensation the measurable frequency range of the Kistler sensor is extended to ~8 kHz. The proposed compensation method was validated with successful cutting force measurements at 20 000 rpm.

Keywords: high-speed milling, cutting force, dynamometer.

1. Introduction

During the design of milling operations, it is an important task to calculate the maximal load on the cutting edge and to determine the maximal moment and power consumption of the spindle. Some of the commercial softwares (e.g.: CutPro & MachPro [1], VERICUT [2], ICAM [3]) can determine these quantities automatically based on cutting force models. These force models describe the function of axial, radial and tangtial force components as a funtion of axial depth of cut and chip width for a given cutting velocity. The corresponding parameters are fit based on a series of force signal data typically measured by Kistler dynamometers.

All type of force measuring units have a maximal measuring frequency range defined by the manufacturer, typically ~ 1 kHz. This is caused by resonance problems and necessitate the usage of a high-pass filter.

Due to the inherent intermittent nature of the milling force, the higher Fourier harmonics of the cutting force are also significant. The frequencies of these components are multiples of the tooth-pass frequency. Thus for example for a one fluted tool applied at 5000 rpm, the 10^{th} Fourier component can be measured with a 1kHz high-pass filter, which is barely sufficient to characterize the periodic cutting force function.

Therefore, parameters of the cutting force functions are usually fit based on the theoretically predicted and the measured average force only [4]. To obtain precise characteristics, several measurement points are necessary, which makes the determination of the force functions expensive and time consuming.

An additional problem of this method is, that the shape of the force function is hidden by the averaged signal, thus it is hard to decide which type of function to use (linear, shifted linear, polynomial, power [5-8]). However, the derivative and the type of non-linearity can significantly influence the stability properties of the milling operation [9].

In this study, first we will provide a method for the dynamical compensation of the Kistler force sensor, by which the measureable frequency range is radically extended. This compensation opens the way for a new type cutting force function determination even in a higher spindle speed range. The radial and tangential components of the force can be measured directly for a large range of chip width by a single cutting test.

2. Dynamic Force Compensation

In the data sheet of the Kistler 9129AA dynamometer the resonant freqency is around ~3 kHz in *x,y* and ~4 kHz in *z* direction. Based on the recommendation of the manufacturer the third or fourth of this value should be used for a high-pass filter to elimiate measurement errors. It is important to note, that the mass of the test material and the additional clamping devices can decrease the given natural frequencies, further decreasing the frequency range.

The main idea of our dynamic compensation is to consider the dynamometer as an oscillator for which the frequency response function can be defined by experimental modal analysis. The important difference is, that we define the transfer function $\boldsymbol{H_F}$ between the measured force $\boldsymbol{F}_m$ and the excitation force $\boldsymbol{F_e}$:

$$\boldsymbol{F}_e = \boldsymbol{H_F}\, \boldsymbol{F_m}\ . \tag{2.1}$$

Note, that all elements are in the frequency domain.

2.1 The 3-channel Compensation

For the cutting force models, only the three measured force components (*x*-*y*-*z*) are necessary, which can be carried out by means of a 3-channel force measurement device, thus $\boldsymbol{F}_{\mathrm{m}}$ is a 3 element vector. During modal analysis, a 4th channel is used for the detection of the force signal of the modal hammer. In one testing point, the excitation force can be defined by the normal of the excited surface $\boldsymbol{n}$ and the magnitude of the hammer force F_{h}:

$$\boldsymbol{F}_e = \boldsymbol{n}\, F_{\mathrm{h}}\ . \tag{2.2}$$

To determine precisely the complete transfer function, several test have to be carried out in several measurement points and in different directions. All the *k* measurements must fulfil Eq. (2.1) as follows:

$$\left[\boldsymbol{F}_{e,1}\ \boldsymbol{F}_{e,2}\ \dots\ \boldsymbol{F}_{e,k}\right] = \boldsymbol{H_F}\left[\boldsymbol{F}_{m,1}\ \boldsymbol{F}_{m,2}\ \dots\ \boldsymbol{F}_{m,k}\right]. \tag{2.3}$$

Denoting the matrix of all excitation elements by $\boldsymbol{E} = \left[\boldsymbol{F}_{e,1}\ \boldsymbol{F}_{e,2}\ \dots\ \boldsymbol{F}_{e,k}\right]_{3xk}$ and the matrix of all measured elements by $\boldsymbol{M} = \left[\boldsymbol{F}_{m,1}\ \boldsymbol{F}_{m,2}\ \dots\ \boldsymbol{F}_{m,k}\right]_{3xk}$, we can used the Moore–Penrose pseudo inverse calculation, which provides a least-square optimal solution for the 3x3 transfer matrix:

$$\boldsymbol{H_F} = \boldsymbol{E}\boldsymbol{M}^{-1} = \boldsymbol{E}\boldsymbol{M}^{T}(\boldsymbol{M}\boldsymbol{M}^{T})^{-1}\ . \tag{2.4}$$

Theoretically, this transfer function can now be used to dynamically compensate any measured dynamometer force signal to get the exact excitation force. However, the compensation is only valid in a finite frequency rage, defined by the highest applicable frequency of the modal hammer excitation. Nevertheless, this frequency can be ~8 kHz if an appropriately small sized hammer is used.

During test measurements we found, that this method can be applied only if a small area of the force sensor is used for the modal excitation and for the final measurement, because in the oscillations of the dynamometer not only the transversal but also the torsional oscillations are significant, which necessitate the measurement of the moments, too.

Fig. 1. Test setup. Marked points on the workpiece for modal analysis.

2.2 The 6-channel Compensation

There are dynamometers that provide the possibility to measure the moments. In a typical Kistler dynamometer plate, 8-channel measurement provides the necessary data to determine the measured force $\boldsymbol{F}_m$ and moment $\boldsymbol{M}_m$ components in the Kistler device's *x*-*y*-*z* coordinate system.

The transfer function $\boldsymbol{H_{FM}}$ for the forces and the moments is defined as:

$$\begin{bmatrix}\boldsymbol{F}_e\\ \boldsymbol{M}_e\end{bmatrix} = \boldsymbol{H_{FM}}\begin{bmatrix}\boldsymbol{F}_m\\ \boldsymbol{M}_m\end{bmatrix}. \tag{2.5}$$

In the modal tests, the excitation moment $\boldsymbol{M}_e$ of the hammer, has to be defined by a cross product

$$\boldsymbol{M}_e = \boldsymbol{r}_{OA} \times \boldsymbol{n}\, F_{\mathbf{h}}\ , \tag{2.6}$$

where $\boldsymbol{r}_{OA}$ is the position vector of the excitation in Kistler's coordinate system. The multiple measurement points are arranged as follows:

$$\begin{bmatrix}\boldsymbol{F}_{e,1} & \dots & \boldsymbol{F}_{e,k}\\ \boldsymbol{M}_{e,1} & \dots & \boldsymbol{M}_{e,k}\end{bmatrix} = \boldsymbol{H_{FM}}\begin{bmatrix}\boldsymbol{F}_{m,1} & \dots & \boldsymbol{F}_{m,k}\\ \boldsymbol{M}_{m,1} & \dots & \boldsymbol{M}_{m,k}\end{bmatrix}, \tag{2.3}$$

from which the transfer function $\boldsymbol{H_{FM}}$ is determined by a pseudo invers similarly as in Eq. (2.4).

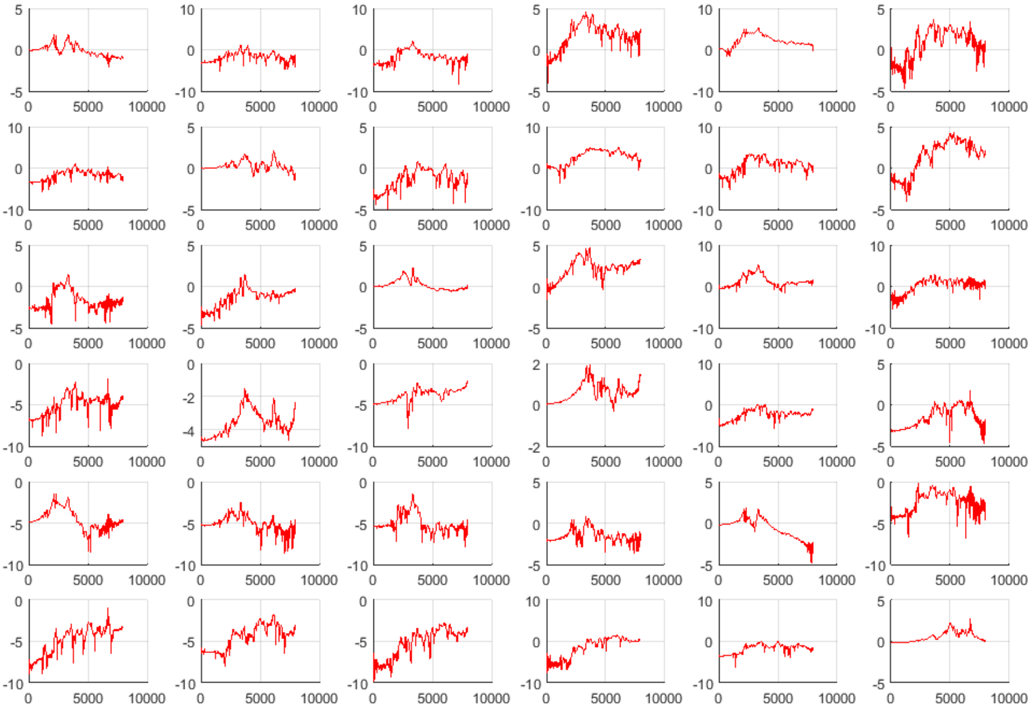

Fig. 2. Measured transfer function matrix.

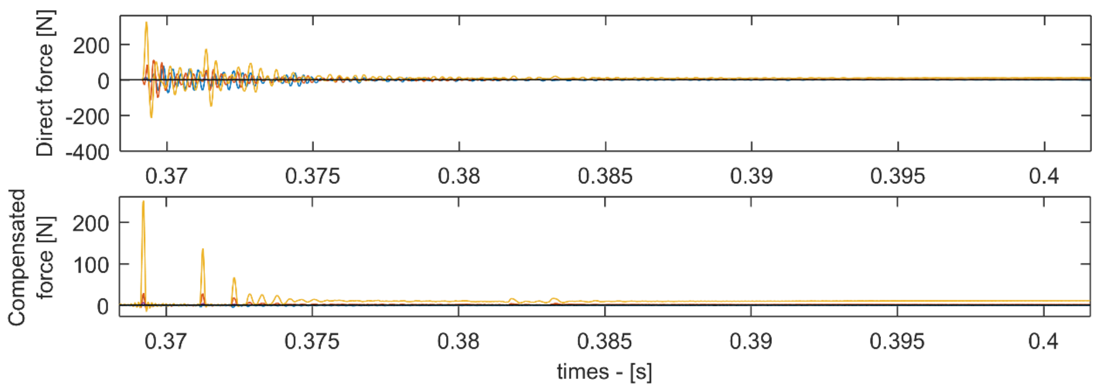

Fig. 3. Directly measured (top) and compensated signals (bottom).

Matrix $\boldsymbol{H}_{FM}$ can be used to dynamically compensate the measured signals in the full area of the plate type sensor. In the next subsection a test case is presented, where the impact like force is restored.

2.3 Test Case for Prall Phenomena

A 9129AA Kistler dynamometer was used for the modal tests, where 14 different points (see Fig. 1) were excited with a small size modal hammer with a Type 9212 sensor. This creates an appropriate excitation up to ~8 kHz.

The resultant transfer function in the frequency domain is shown in Fig. 2. Note, that the diagonal elements have almost unitary values in the low frequency range and the off-diagonal elements are almost zero. This implies that in the low frequency range (< ~1 kHz) there is no need for compensation. However, at higher frequencies all matrix elements can have high values.

As a test case, we measured a hammer contact, where multiple impact-like bounces occurred before a constant force evolved. Fig. 3 clearly shows that the direct force signal contains inertial forces due to the oscillations of the plate, while the compensated force signal provides almost the same signal as was measured by the modal hammer.

Note, that the compensated force signal still contains small oscillatory components which are caused by the ~8 kHz limiting frequency.

This compensation method can well be used in cutting force function measurements, which is presented in the next section.

3. Milling Force Measurement at High Spindle Speed

Measurement results for two simple down milling processes with a helical tool for different spindle speeds (n=5000 rpm & 20 000 rpm) are presented. See the data about the tool geometry and the technological parameters in Table 1.1.

In the lower spindle speed case (Fig. 4a), the compensated signal shows a better function, because the compensation removes the oscillatory component after the sudden force step created by the down milling operation.

In the second case at high spindle speed, the force characteristics cannot even be realized in the directly measured force function, while the compensated signal still shows an acceptable result.

Table 1.1. Parameters of the test measurements

Tool geometry		**Machining parameters**	
diameter	12 mm	radial immersion	6 mm
number of teeth	2	axial immersion	2 mm
helix angle	30°	feed per tooth	0.2 mm

4. Direct Force Characteristics Measurement

The above introduced compensation method enables to directly measure the force function.

According to our proposed method the angular position of the tool has to be measured, which can be used to transform the measured *x-y* force components to the edge's local radial and tangential (*r-t*) coordinate system:

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} \cos(\phi(t)) & -\sin(\phi(t)) \\ \sin(\phi(t)) & \cos(\phi(t)) \end{bmatrix} \begin{bmatrix} F_r \\ F_t \end{bmatrix}. \qquad (4.3)$$

Then the corresponding nominal chip thickness can be determined:

$$h(t) = f_z \sin(\phi(t)) \,, \qquad (4.3)$$

where f_z is the feed per tooth. Finally, the radial and tangential force components can be plot against the chip thickness values for the cutting phase.

During the measurements, it was sufficient to measure only a given angular position of the tool by an optical sensor and consider linearly changing angular position between two rising edges of the digital signal.

The above process is repeated for different feed-per-tooth values and plot together in Fig. 5. It can be seen, that all functions overlap with very small noise due to the appropriate compensation. Furthermore, these plots show an unexpected characteristics for the radial component, which could not be realized by traditional measurement methods.

Unfortunately, this method can only be used for radial immersion values where only one tooth is in contact with the material.

5. Conclusion

In the present study we introduced a dynamical compensation method for dynamometer data. This method enables to measure the cutting force function of a milling process at 20 000 rpm in a satisfying way.

A measurement scheme is introduced to determine the characteristics of the force function directly for a larger range of chip thickness values.

Our future goal is to improve the direct force measurement algorithm for cases, where more than one tooth is cutting simultaneously.

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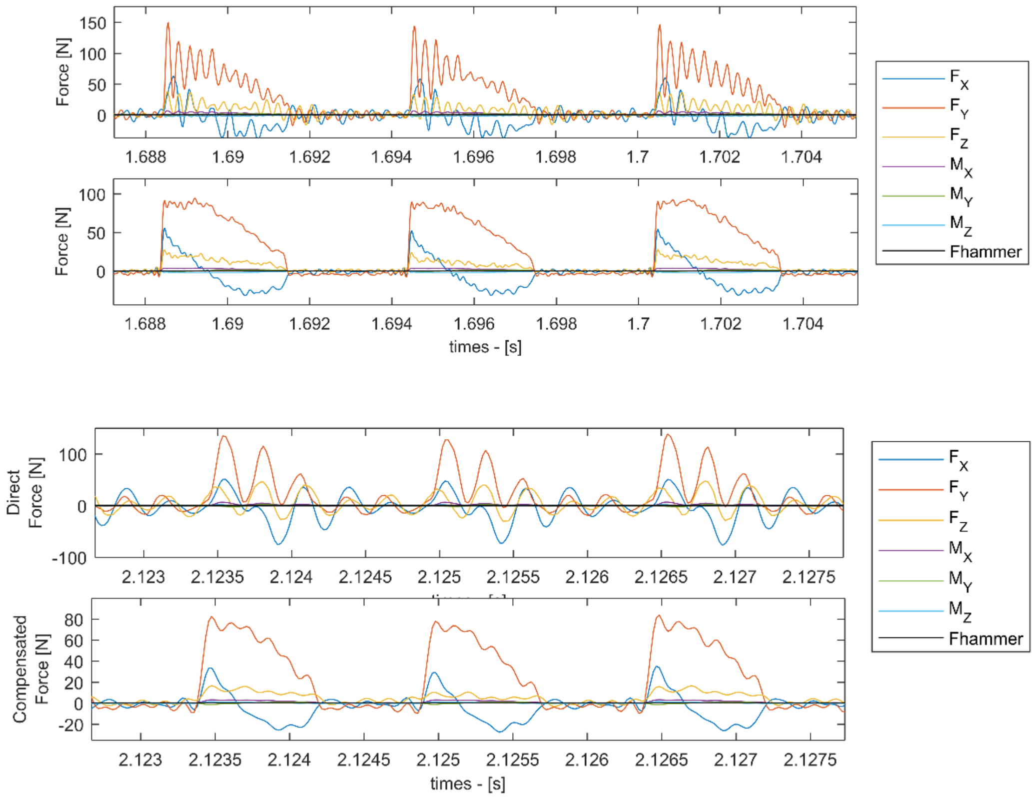

Fig. 4. Directly measured and the compensated *x-y* force components for n=5000 rpm (top) and n=20 000 rpm (bottom).

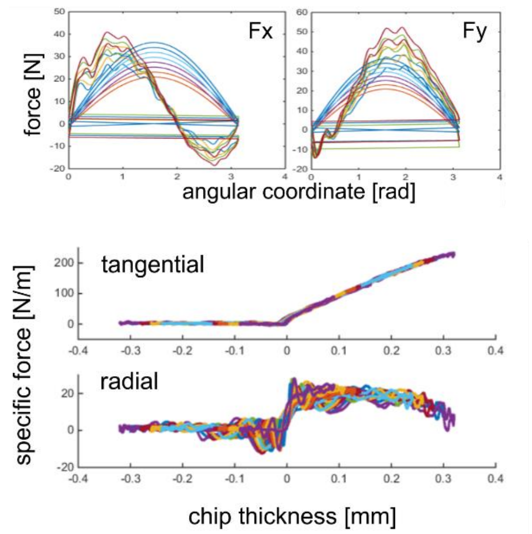

Fig. 5. Compensated cutting force in the *x-y* coordinate system with the nominal chip thickness (top) and the transformed radial and tangential component plotted againts the chip thickness (bottom). Note, the different scale! Colors represent different feed-per-tooth values in the range f_z=[0.1,0.325] mm.

6 References

[1] a1 MAL Manufacturing Automation Laboratories, Inc. CUTPRO, NPRO, MACHPRO, http://www.malinc.com/

[2] a2 CGTech, VERICUT , http://www.cgtech.com/

[3] a3 ICAM, http://icam.com/

[4] Altintas, Y. (2012). Manufacturing automation: metal cutting mechanics, machine tool vibrations, and CNC design. Cambridge university press.

[5] O. Kienzle, Spezifische schnittkräfte bei der metallbearbeitung, Werkstattstechnik und Maschinenbau 47 (1) (1957) 224–225.

[6] H. Shi, S. Tobias, Theory of finite-amplitude machine-tool instability, International Journal of Machine Tools and Manufacture 24 (1) (1984) 45– 69

[7] Endres, W. J., & Loo, M. (2002, May). Modeling cutting process nonlinearity for stability analysis—application to tooling selection for valve-seat machining. In Fifth CIRP International Workshop on Modeling of Machining (pp. 71-82).

[8] Altinta, Y., & Armarego, E. J. A. (1996). Prediction of milling force coefficients from orthogonal cutting data. Trans. ASME J. Engng Ind.

[9] Dombovari, Z., Wilson, R. E., & Stepan, G. (2008, December). Estimates of the bistable region in metal cutting. In Proceedings of the Royal Society of London A: Mathematical, Physical and Engineering Sciences (Vol. 464, No. 2100, pp. 3255-3271). The Royal Society.