Operational modal analysis of a machine-tool structure during machining operations

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Abstract :

The prediction of modal parameters of a machine-tool structure is important for an accurate knowledge of the dynamic behavior during machining. Usually, the characterization is carried out through an experimental modal analysis with impact and/or shaker tests. However, the excitation, artificially, created by a hammer or a shaker is difficult even impossible to conduct on machine-tool structure or robot structure during machining. These tests are thus performed on machines in rest. Unfortunately, the conditions of a machine in rest are significantly different from its real conditions during machining and accordingly its dynamic properties in service are different from static results of impact or shaker tests.
This paper proposes an identification methodology based on operational modal analysis for a machine-tool structure during machining operation. The modal identification is done by the transmissibility function based method whose major advantage is its independence of the exciting force's nature and its ability to eliminate harmonic components and reduce their effect on the identified modal model.

keyword: Operational modal analysis, high speed machining, machine-tool, harmonics

1 Introduction

High speeds machining (HSM) enables to increase productivity and reduce production costs. However, the use of HSM is limited by vibrations, which can affect the stability and the quality of the workpiece during machining operations [1, 2] . For this reason, it is interesting to evaluate the machining structure's modal parameters that can help to control machining operations more efficiently.
Forced vibration tests, currently by hammer or shaker excitations, are employed in order to estimate modal parameters with experimental modal analysis (EMA) techniques, and these tests are done when the machines are in rest [3]. The dynamic properties of a machine in rest can be significantly different from its real conditions during machining.
Bin et al. [4] developed a technique based on the interrupted cutting of a specific designed workpiece.

The exciting cutting force is proven to be square wave pulses signal. Hui et al. [5] proposed a random cutting excitation technique realized by cutting a specially designed workpiece. The specification of this technique comparing to the previous one, is that the surface of the workpiece has a long narrow random zigzag width, which randomizes the resulting cutting forces.
However, both methods run under artificial special random conditions rather than normal machining operations. This explicates the great need to identify operational modal parameters of machining structures in the normal machining conditions.
A powerful tool for estimating a structure's modal parameters during operational conditions was proposed, named the operational modal analysis (OMA). With this approach, it is possible to estimate the system modal parameters from only responses under unknown operational loads. The approach is therefore useful for machining operations. Over the last few decades, many operational modal identification methods have been developed. However, most of these methods are limited by the white noise excitation assumption. Practically, during machining, the exciting cutting force cannot be assumed to be a white noise excitation input because it contains also harmonic components.
In order to overcome the white noise excitation assumption, Devriendt et al. [6, 7] proposed an operational modal analysis technique based on transmissibility measurements. The authors demonstrated that this method is independent from the nature of the input excitation. This propriety is very interesting and its investigation in machining condition is the main objective of this study. In this paper, the identification of modal parameters of a machine-tool structure, using the OMA approach based on transmissibility measurements, is done. In section 1, a brief background of the operational modal identification method based on transmissibility measurements is presented. This method is then investigated on an experimental beam test, in section 2. In section 3, an experimental modal analysis of the machine-tool KX15 is investigated in order to obtain a reference modal model. Then, modal parameters are identified by the transmissibility function based (TBF) method during a machining process. Results show that the studied technique is able to identify modal parameters efficiently and also to distinguish between structural poles and spurious ones.

2 Modal identification method based on transmissibility functions

In OMA, the primary data for identification methods are in general power spectral responses or correlation function responses. These methods are based on the assumption of white noise processes for operational excitations. However, this assumption is hard to be respected in real situations and specially for high speed machining. Devriendt et al. [6, 7] proposed to use transmissibility functions of responses as primary data in order to avoid the white noise excitation assumption. A transmissibility function is defined by the ratio of response spectra between two degrees of freedom

$$T_{ij}(s) = \frac{\hat{x}_i(s)}{\hat{x}_j(s)} \tag{1}$$

2.1 Theoretical formulation of operational modal analysis based on transmissibility functions

At the system poles, the transmissibility functions are independent on excitations and equal to unscaled mode shapes. Thus, Devriendt et al. propose to consider two different load cases (single or distributed) k and l.

At the system pole λ_r

$$\Delta T_{ij}^{kl}(s) = T_{ij}^{k}(s) - T_{ij}^{l}(s) = \frac{\phi_{ir}}{\phi_{jr}} - \frac{\phi_{ir}}{\phi_{jr}} = 0 \tag{2}$$

and thus λ_r is a pole of

$$\Delta^{-1} T_{ij}^{kl}(s) = \frac{1}{T_{ij}^{k}(s) - T_{ij}^{l}(s)} \tag{3}$$

Then, $\Delta^{-1} T_{ij}^{kl}(s)$ can be used as primary data for any modal identification method in frequency domain instead of FRFs. In this study, we used the PolyMAX method to deduce system poles from $\Delta^{-1} T_{ij}^{kl}(s)$.

2.2 Practical procedure

From measurements in time, transmissibility functions are evaluated. Note that when Fourier transform is used in stead of Laplace transform, we have $s = \mathrm{i}\omega$. In application of a classical modal identification method on data of $\Delta^{-1} T_{ij}^{kl}(s)$, on can deduce its poles and then eigen-frequencies and damping ratios. Note that, the system's poles are the poles of $\Delta^{-1} T_{ij}^{kl}(s)$, but not all the poles of $\Delta^{-1} T_{ij}^{kl}(s)$ are the system's poles. Therefore, it is necessary to choose physical poles of the system from the poles of $\Delta^{-1} T_{ij}^{kl}(s)$. In order to highlight the system's poles, a third load case should be considered, let's say m and the rank of the following matrix is studied using singular value decomposition

$$\begin{bmatrix} T_{1r}^{k}(\mathrm{i}\omega) & T_{1r}^{l}(\mathrm{i}\omega) & T_{1r}^{m}(\mathrm{i}\omega) \\ T_{2r}^{k}(\mathrm{i}\omega) & T_{2r}^{l}(\mathrm{i}\omega) & T_{2r}^{m}(\mathrm{i}\omega) \\ \vdots & \vdots & \vdots \\ T_{Nr}^{k}(\mathrm{i}\omega) & T_{Nr}^{l}(\mathrm{i}\omega) & T_{Nr}^{m}(\mathrm{i}\omega) \\ 1 & 1 & 1 \end{bmatrix} \tag{4}$$

At the system pole λ_r, the rank this matrix is one. Therefore $\sigma_1(\omega) > \sigma_2(\omega) > \sigma_3(\omega) \geq 0$ and at system's poles, $\sigma_2(\omega)$ tends toward zero and the pics of $1/\sigma_2(\omega)$ indicate the system's poles.
For the mode shape $\boldsymbol{\phi}_r$, one can choose a degree of freedom as reference, for instance N, thus with a curve fitting of transmissibility functions, one can estimate the unscaled mode shape as

$$\boldsymbol{\phi}_r = \begin{Bmatrix} T_{1N}^{k}(\lambda_r) \\ T_{2N}^{k}(\lambda_r) \\ \vdots \\ 1 \end{Bmatrix} \tag{5}$$

3 Applications

3.1 Operational modal identification of a cantilever beam

In order to investigate the efficiency of the proposed procedure and its numerical implementation, experiments have been carried out for a cantilever beam, shown in Figure 1.
The necessary condition for applying the transmissibility OMA technique is that the loading conditions vary during the test. This can be realized in different ways, for example, by changing the location of the applied force, by changing the number of applied force or also by changing the amplitude of the applied force.
In these tests, the input force is a combination between an excitation by impact hammer at random loca-

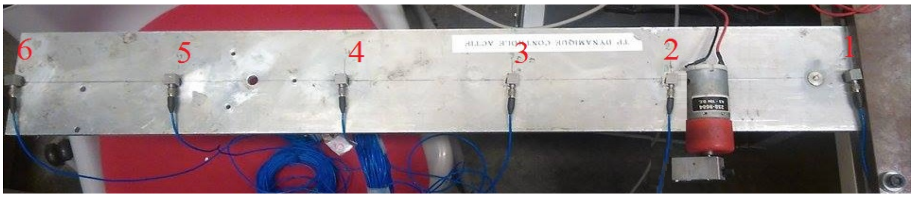

Figure 1: Experimental setup for the cantilever beam structure

tions and an harmonic excitation through a motor turning on 851 rpm (harmonic frequency: $f_h \approx 14.18$ Hz). The motor is fixed in the second point of the beam, as shown in Figure 1. So, harmonic excitations are present along with the impulse excitations.

Four free run measurements of two minutes have been taken for each experiment. The data were sampled at 200 Hz. A total of 6 accelerometers were equally distributed over the full length of the beam (the numbering of accelerometers starts from the right to the free end of the beam). In Figure 2, acceleration responses measured on six different points of the beam, under unknown excitation in presence of harmonics, are shown.

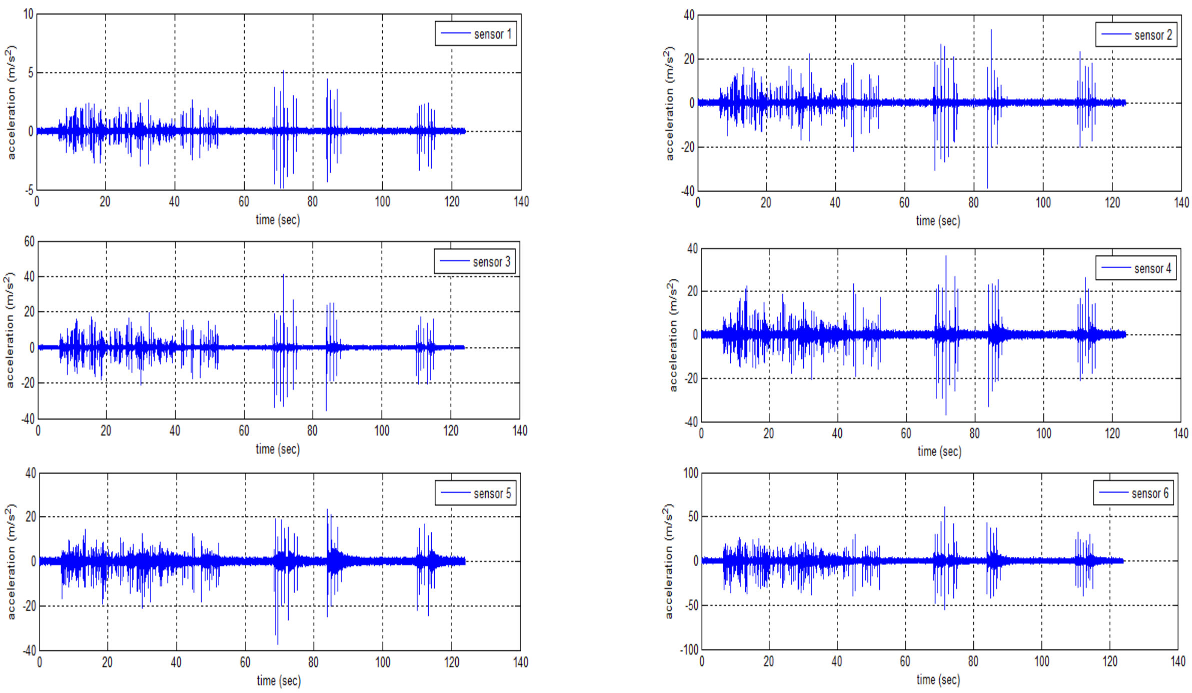

Figure 2: Acceleration responses

The first three eigen frequencies of the beam, shown on Table 1, are calculated analytically, using the equation 6:

$$f_i = \frac{1}{2\pi\sqrt{12}}\alpha_i^2\sqrt{\frac{E}{\rho}}\frac{h}{L^2} \quad (6)$$

With:
Height: h=0.8 cm,
Length: $L = 75.7$cm,
Young's modulus:$E = 70000$ MPa ,

Density: $\rho = 2698$ Kg/m^3 ,
α_0=1.8751, α_1=4.695, α_2=7.85, $\alpha_{i+1} = (2i+1)\frac{\pi}{2}$; for i>2

Table 1: Exact eigen frequency values of the studied beam

order	Exact frequency value (Hz)
1	10.8
2	67.8
3	189.85

Modal parameters are then identified by the classical PolyMAX identification method [10, 11], using power spectra. We consider the different cross power spectra with reference to the second accelerometer only. In the stabilisation diagram, shown in Figure 3, the sum of these power spectra is taken into account.

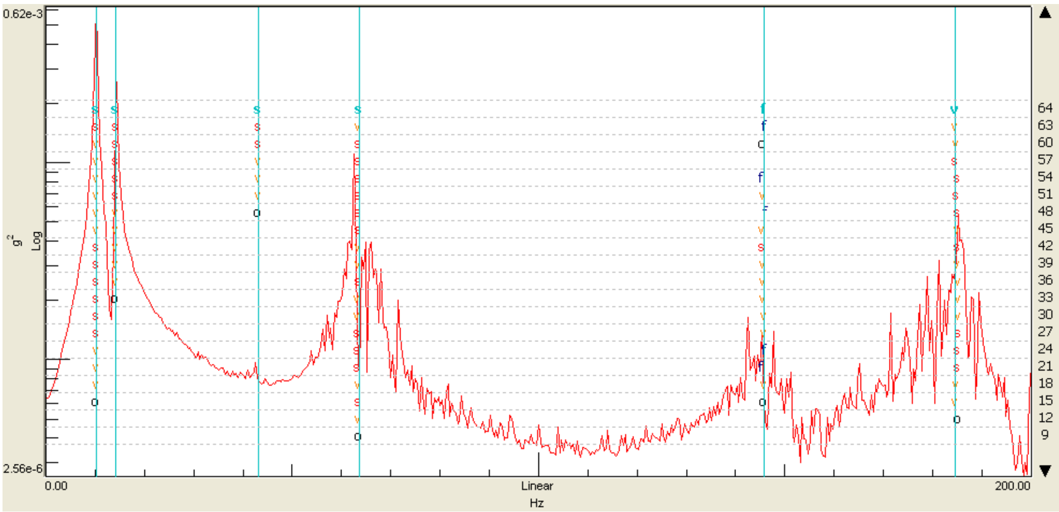

Figure 3: Cross power spectrum and the stabilization diagram by the PolyMAX method

Table 2: identified frequencies and damping factors by the PolyMAX method

order	Frequency (Hz)	Damping factor (%)
1	10,28	1.12
2	14.26	0.09
3	43.05	0.69
4	63.54	1.54
5	145.74	0.28
6	184.55	0.78

Both of structural and non structural (harmonic) poles are present. The frequency of harmonic components are multiple of $f_h \approx 14.18$ Hz.

By the PolyMAX method, an harmonic component can be considered as a stable pole despite it is not a structural pole, for example, the first harmonic frequency ($f_h \approx 14.18$ Hz) is considered as a stable pole of the beam. Consequently, the distinction between harmonic and real poles is difficult.
So the necessity of the TBF method, which solve the problem of the presence of harmonics by its ability to separate real poles from those additional. In order to select the structural poles of the studied beam,

a singular value decomposition of the transmissibility matrix T is done.

A single reference case is employed, for this, the response data of the second accelerometer is considered as reference in the calculation of transmissibility functions. The choice of measured data on the second point as reference is explained by the fact that it is the most excited place on the beam by the applied harmonic excitation created by the motor.

$$T = \begin{bmatrix} T^1_{12}(s) & T^2_{12}(s) & T^3_{12}(s) & T^4_{12}(s) \\ T^1_{32}(s) & T^2_{32}(s) & T^3_{32}(s) & T^4_{32}(s) \\ T^1_{42}(s) & T^2_{42}(s) & T^3_{42}(s) & T^4_{42}(s) \\ T^1_{52}(s) & T^2_{52}(s) & T^3_{52}(s) & T^4_{52}(s) \\ T^1_{62}(s) & T^2_{62}(s) & T^3_{62}(s) & T^4_{62}(s) \\ 1 & 1 & 1 & 1 \end{bmatrix} \quad (7)$$

In Figure 4, $\frac{1}{\sigma_2}$ is plotted as a function of frequency. Three dominant picks coinciding with the first three natural frequencies are present. Harmonic picks are attenuated if not eliminated.

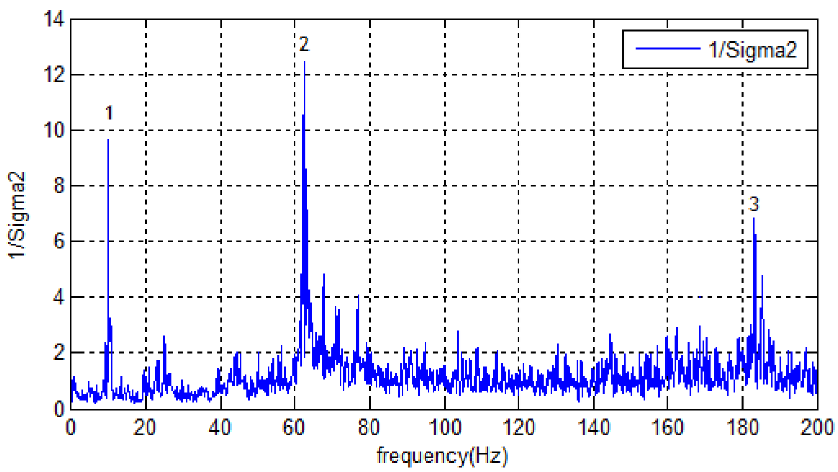

Figure 4: Selection of the system's poles by means of singular value decomposition

Once the resonnant frequencies are known, it is possible to calculate the damping factors and mode shapes from the transmissibilities as was explained in section 1.

Using the OMA approach based on transmissibility measurements, modal parameters of the studied cantilever beam are identifies and shown in Table 3.

Table 3: identified modal parameters by the TBF method

order	Natural frequency (Hz)	Damping factor (%)
1	10.12	1.128
2	64.21	2.20
3	184.39	0.39

For a comparison purpose, an operational modal identification of the beam was performed by the Polymax method. Results of the operational identification modal parameters of the studied beam, by the two different OMA techniques are shown on Figure 5 and Table 4.

Table 4: Identified modal parameters by PolyMAX and TBF techniques

Modal parameters		Exact values	PolyMAX	TBF
Natural frequency	f_1 (Hz)	10.80	10.28	10.12
	f_2(Hz)	− − −	14.26	− − −
	f_3(Hz)	− − −	43.05	− − −
	f_4(Hz)	67.80	63.54	64.21
	f_5(Hz)	− − −	145.74	− − −
	f_6(Hz)	189.85	184.55	184.3
Damping factor	ξ_1(%)	− − −	1.12	1.128
	ξ_2 (%)	− − −	0.09	− − −
	ξ_3 (%)	− − −	0.69	− − −
	ξ_4 (%)	− − −	1.54	2.20
	ξ_5 (%)	− − −	0.28	− − −
	ξ_6 (%)	− − −	0.78	0.39

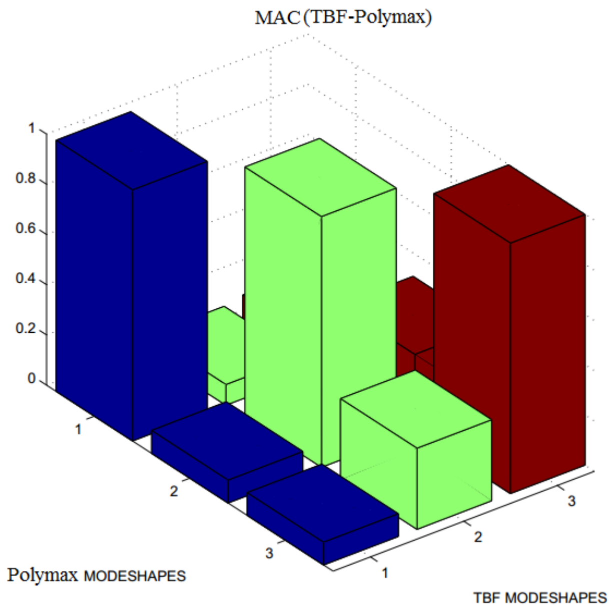

Figure 5: MAC matrix between TBF and PolyMAX modes

On the frequency band [0 200 Hz], only three structural modes are identified by the transmissibility based function (TBF) method. However, three harmonic frequencies (14.26 Hz, 43.05 Hz and 145.75 Hz) are still obtained by the PolyMAX technique. The modal assurance criterion (MAC) shows a very good correlation between the three structural mode shapes obtained by these two different techniques (Coefficient of correlation greater than 0.8). L'agreement between results obtained by the TBF method with the exact values and those of PolyMAX confirms (i) the robustness the TBF method in presence of harmonics and (ii) its numerical implementation. The TBF is now ready for application in machining operations in the next section.

3.2 Modal parameters identification of a machine-tool

3.2.1 Experimental modal analysis (EMA)

An experimental modal analysis (EMA) of the machine-tool was performed. The excitation is realized by a set of chocks on the tool by an impact hammer, while the machine is in rest. Modal parameters are identified, by the PolyMAX method, from the sum of two frequency response functions (FRF) measurements during experiments. In the stabilization diagram, shown in Figure 6, the sum of two frequency response functions (FRF) measurements is considered.

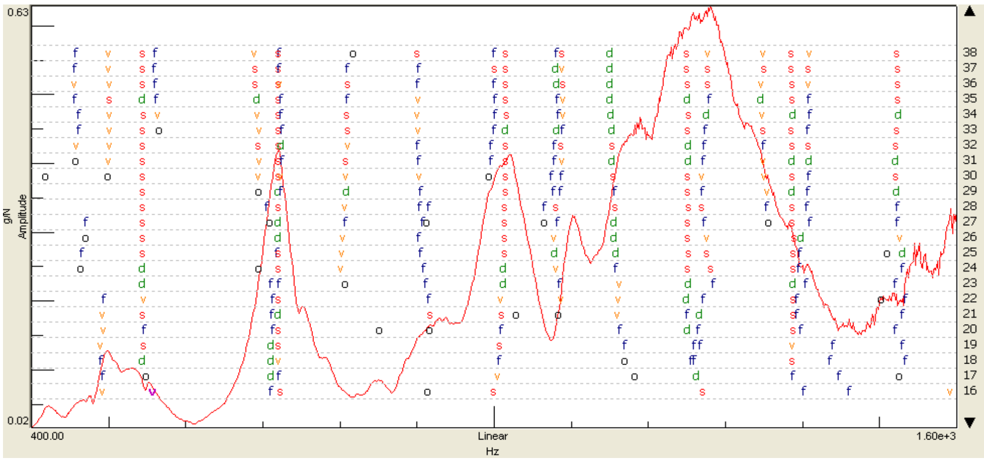

Figure 6: Cross power spectrum and stabilization diagram

Using the PolyMAX method, modal parameters on the frequency band [400 Hz, 1600Hz] are identified.

Table 5: Identified results by the PolyMax method through an EMA

order	frequency (Hz)	Damping factor (%)
1	545.56	4.83
2	721.15	2.50
3	901.68	1.39
4	1016.296	2.25
5	1153.066	1.84
6	1276.55	1.71
7	1388.83	3.63
8	1524.593	1.77

To identify modal parameters by PolyMAX method, a modal model size is considered. In this case the model size is equal to 38. When trying to fit high-size models that, much more modes than those present will appear. True physical modes are separated from the spurious numerical ones by interpreting the stabilization diagram. The poles corresponding to a certain model order are compared to the poles of a one-order-lower model. If their differences are within the tolerant limits, the pole is considered as a stable one. The spurious numerical poles will not be stable at all.

3.2.2 Operational modal analysis (OMA)

An operational modal analysis was performed on a 5-axis Huron KX15, using an end mill cutting tool, three-teeth, 16 mm tool cutter, the radial depth of cut equal to 10 mm, and the axial depth of cut equal to 3 mm. An experimental protocol is prepared in order to measure the tool's vibration during the machining process. The measured data are then treated by the transmissibility based method and the modal parameters (natural frequencies and damping factors) of the structure are identified.

3.2.3 Experimental setup and results

Four machining tests on the four different directions are done on an aluminum(2017) square piece of 100 mm of side, as shown in Figure 7.

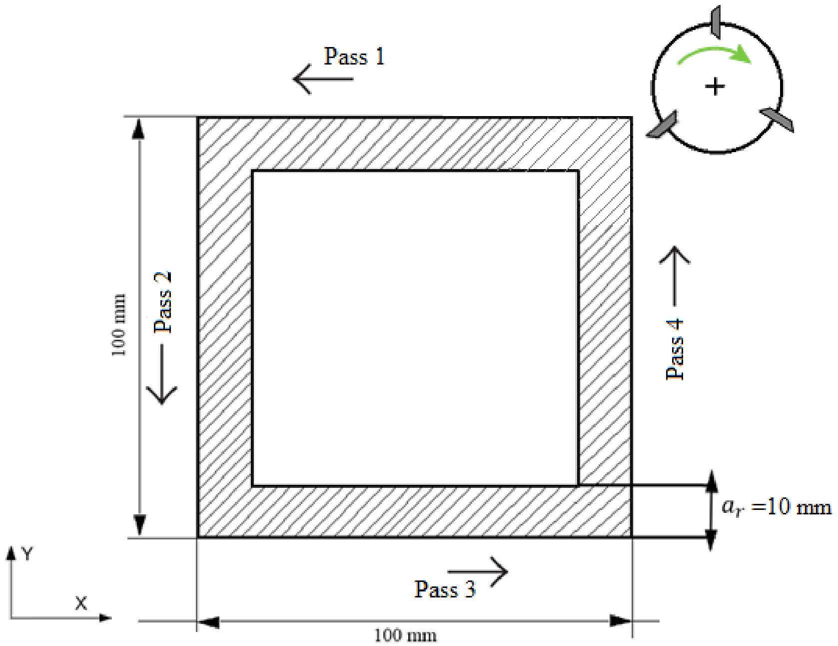

Figure 7: Cutting path

The piece is fixed on the Kistler table 9257B that measure the cutting effort in the axes (X,Y). Two accelerometers were attached to the spindle to measure the vibration during the machining tests. Finally, from an output top-tour signal of the machine-tool, the instantaneous spindle speed during test is obtained. The acquisition of these signals is done using the LMS Test-lab tool acquisition system. The temporal data concerning the vibration, the applied cutting force and the rotational spindle speed are measured using the LMS Test-Lab tool. The acquisition duration is 14 s and the sampling time is Δt=0.000244s. The frequency bandwidth during tests is [400, 1600 Hz] and the frequency resolution was 0.071Hz.
The rotational spindle speed is equal to 14000 rpm, so harmonic components will be multiple of 233.33 Hz. It is clear in Figure 8, the first harmonic frequency is around 232.28 Hz.
During machining process, the applied force contains harmonic components. In order to select only the

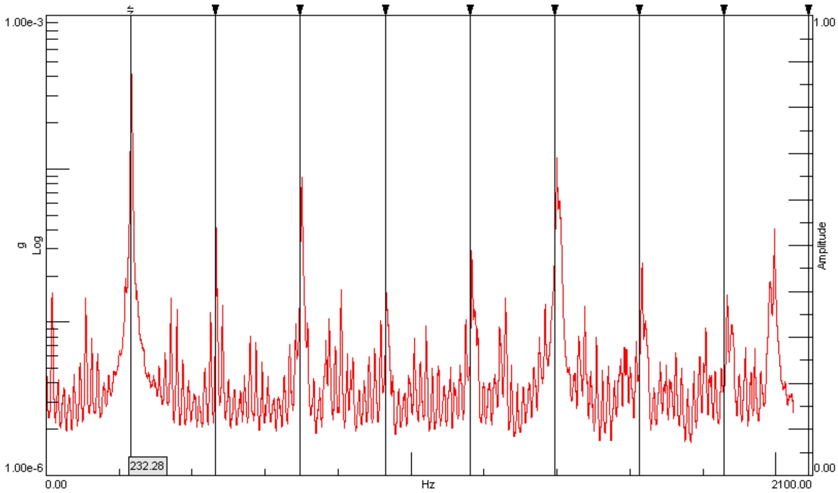

Figure 8: The frequency spectrum of vibration response of the spindle in direction X during pass 1

structural poles of the machine-tool, this transmissibility matrix T is proposed.

$$T = \begin{bmatrix} T^1_{XY}(s) & T^2_{XY}(s) & T^3_{XY}(s) & T^4_{XY}(s) \\ 1 & 1 & 1 & 1 \end{bmatrix} \tag{8}$$

$$T^i_{XY}(s) = \frac{\hat{X}_i(s)}{\hat{Y}_i(s)} = \frac{\hat{X}_i(s)\overline{\hat{Y}}_i(s)}{\hat{Y}_i(s)\overline{\hat{Y}}_i(s)} \tag{9}$$

In the transmissibility matrix T, four transmissibility functions $T^i_{XY}(s)$ are calculated during four different machining tests.

A singular value decomposition of the matrix T is done. The second singular value σ_2 of the matrix T should converge to 0 in the real poles system.

In the Figure 9, the variation of $\frac{1}{\sigma_2}$ is plotted in function of frequency. Five dominant picks are present. It can be noted that most of harmonic components are eliminated. Once the resonant frequencies are known, it is possible to calculate the damping factors from the transmissibility functions.

The modal parameters (Natural frequencies and damping factors) of the machine-tool are then identified and shown in Table 6.

Modal parameters, identified by the TBF method in machining tests, differ from those identified during

Table 6: Operational modal parameters of the machine-tool structure

order	Resonant frequency (Hz)	Damping factor
1	574.3	0.37
2	808.4	0.18
3	942.5	1.44
4	1266.5	1.69
5	1404.12	0.10

an EMA. This is because on an experimental modal analysis test, the machine is in rest and the excitation is artificially created by an impact hammer, which is totally different from the operational conditions during a machining test.

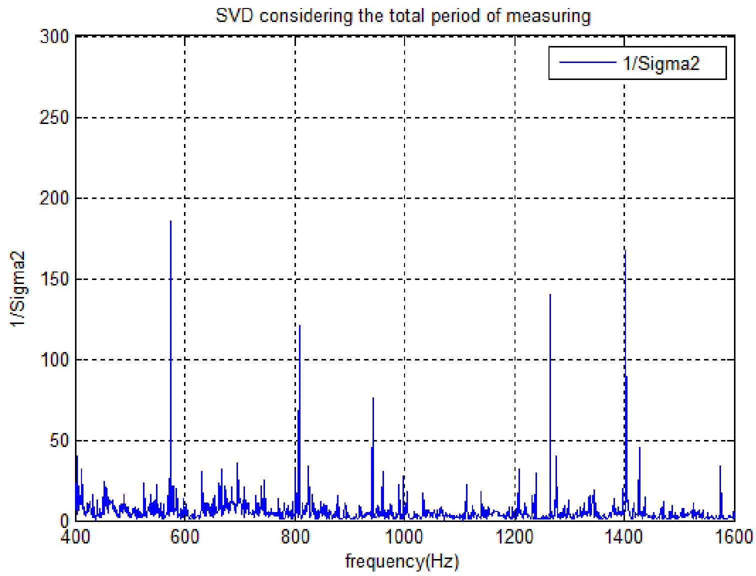

Figure 9: Selection of the system's poles by means of singular value decomposition

4 Conclusions

In this paper, the operational modal analysis of a machine tool with the transmissibility function based method (TBF) was studied. The TBF method is chosen, first, because its independence from the excitation nature, second, because of its ability to distinguish between structural and spurious poles. This can be can be adequate for machining conditions. To verify the robustness the TBF method in presence of harmonics and its numerical implementation, a modal identification of a cantilever beam is done by both of PolyMAX and TBF method. Results show that the TBF method is able to select only structural poles and to identify efficiently modal parameters. Thereafter, the OMA of the machine tool was performed by the transmissibility function based method. By a singular value decomposition of the proposed transmissibility matrix, most of harmonic components are eliminated. In order to have a reference modal base, an EMA of the machine tool was conducted. Modal parameters of the machine tool identified during machining tests are different from those identified through an EMA. This is because the conditions of the machine tool in rest are not the same from its real conditions during machining.

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