IMPROVEMENT OF HEAVY MACHINE TOOL DYNAMICS BY PASSIVE DYNAMIC VIBRATION ABSORBER

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ABSTRACT

The article deals with evaluation of frequency responses of heavy machine tool for different attachment places of dynamic absorbers or dynamic vibration neutralizers in order to substantially suppress vibrations occurring during a machining process. The approach is based on methodology utilizing a multi-body model of a machine tool with flexible bodies in linear state-space form with added structure representing the absorber. The proposed methodology significantly decreases elapsed time requirements and makes possible to efficiently optimize even complicated problems. There are compared results obtained by utilization of two different objective functions as well as results for dynamic vibration absorber and neutralizer in the article.

KEYWORDS: machine tool, vibrations, state space LTI, dynamic vibration absorber, optimization.

1. INTRODUCTION

The self-excited oscillations occurring during a machining process often negatively influence the workpiece surface quality and their occurrence may also lead to a damage of the machine tool, see e.g. [Marek 2015]. Several methods for the suppression the oscillations exist, they are generally based on adjustments of parameters of the cutting process [Tlusty 1986] or on adding a suitable damping elements often represented by an absorber either on the workpiece [Sims 2007] or on the machine tool [Chung 1997].

Added structure in the form of dynamic absorber can be generally passive [Tarng 2000], [Rivin 1992] with parameters tuned on given frequency range or active with controller changing the parameters of the absorber during the machining [Brecher 2005], [Pratt 2001], [Vetiska 2012]. The passive absorbers tend to be less complicated and therefore also cheaper.

Authors' previous works presented studies aimed on analysis of an influence of a multi-mass dynamic vibration absorber (DVA) in different configurations [Brezina 2015a] as well as comparison of suitability of DVA against dynamic vibration neutralizer (DVN) as a specific case of DVA without damping and comparison of two objective functions for optimization [Brezina 2015b]. The numerical results obtained suggest that tuned DVA suppresses the magnitudes of resonant frequency response in given frequency range much better than tuned DVN.

These results were obtained for the DVA/DVN attached to the tool holder. In this article the analysis is completed by examination of the influence of the passive DVA/DVN attached to other places within the machine tool on the frequency response magnitudes of the tool holder.

The method used for this purpose is based on adding a structure providing compensatory action in the simplest possible form. Thus an influence of DVA or DVN on self-excited oscillations suppression will be analyzed. Namely the influence of an optimally tuned DVA/DVN attached to the selected places of the machine tool on resonant oscillations of the tool holder will be investigated.

The method utilizes the connection of black-box model of the machine tool with white-box models of absorbers with tunable parameters. The possibilities of suppression of dominant resonant magnitudes, which can form suitable conditions for appearance of self-excited oscillations, are investigated on the obtained grey-box model again with tunable parameters which enables very fast computations of frequency responses dependent on specific parameter values.

2. MODEL AND PLACES OF DVA/DVN ATTACHMENT

The basic model of the machine tool without connected DVAs/DVNs was prepared in multi-body modelling (MBS) software with modally analyzed bodies in FEM software according to the CAD documentation of the real machine tool by TOSHULIN, a. s. All of modeled bodies except the drive of the spindle, tool holder, bed and clamping plate were prepared as flexible in FEM software before the import to MBS. The machine was modelled in the position which is from the point of dynamic compliances the least favorable, i.e. with the maximally lowered slipper in the middle of the crossbeam.

The Fig. 1 shows the basic MBS model. Indicated connection of DVAs/DVNs and its local coordinate systems for the connection of the absorbers are depicted in Fig. 2 and Fig. 3. The connection points for added models of absorbers were prepared in following manner: one DVA/DVN on the tool holder at point C, two DVAs/DVNs on the upper surface of the crossbeam at points C^L and C^R, two DVAs/DVNs on the side of the support also C^L and C^R and one DVA/DVN on the top of the drive of the spindle (top of the gearbox) again at C. The connection places are based on collision free spots on the machine tool.

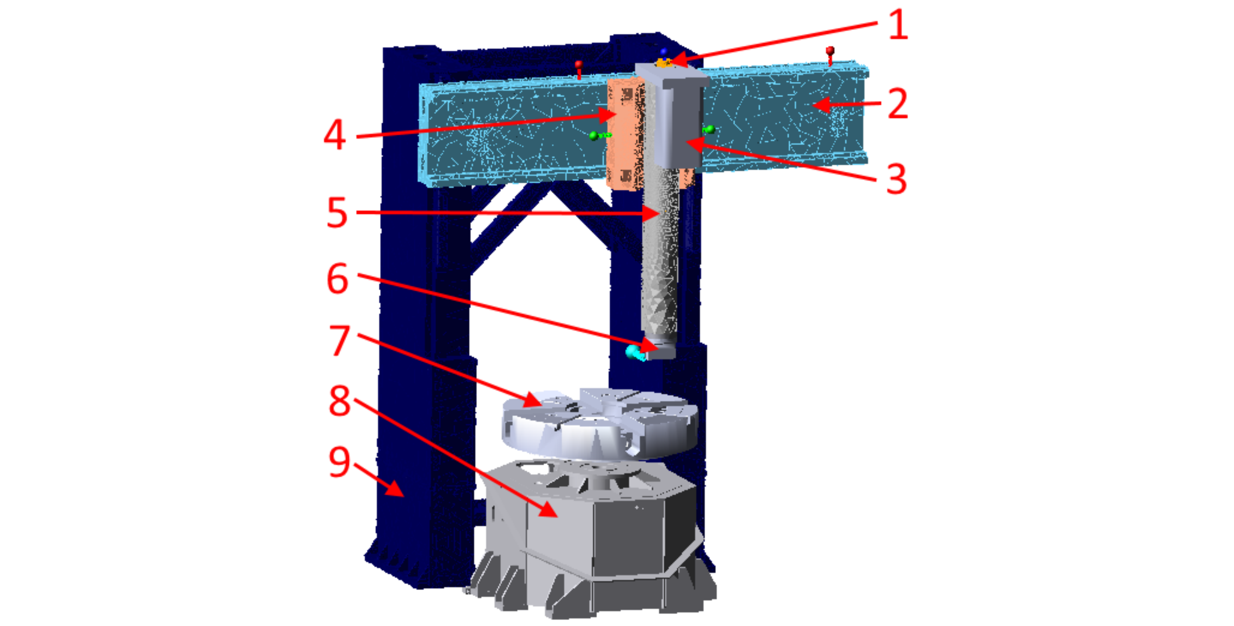

Figure 1. Machine tool MBS model with schematic connection of DVAs/DVNs; parts of the model are as follows: 1 – spindle, 2 – crossbeam, 3 – drive of the spindle, 4 – support, 5 – slipper, 6 – tool holder, 7 – clamping plate, 8 – bed, 9 – stand

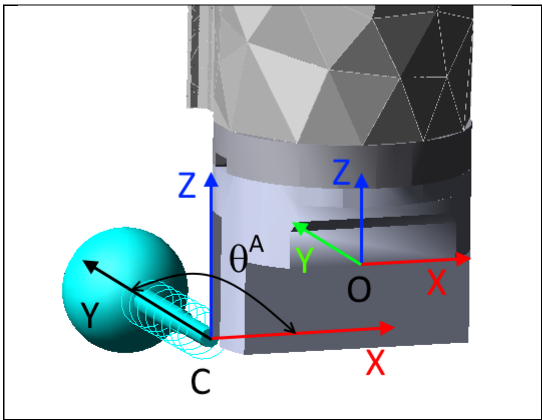

Figure 2. Detail of connection point, excitation point and coordinate systems – tool holder

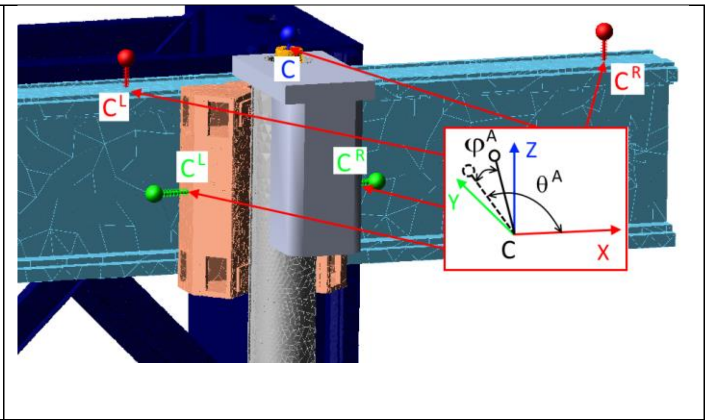

Figure 3. Detail of connection points and coordinate systems – support, crossbeam, drive of the spindle

The approximation of the machine tool MBS model at time domain is consequently exported in a form of linear time invariant model in state space (state LTI model) for inputs $\mathbf{Q}=\left[\left({}^{O}\mathbf{Q}\right)^{T},\left({}^{C}\mathbf{Q}\right)^{T}\right]^{T}$, representing exciting forces acting at X, Y, Z directions of the tool holder masspoint O and the point C where the DVA/DVN will act and for outputs $\mathbf{q}=\left[\left({}^{O}\mathbf{q}\right)^{T},\left({}^{O}\mathbf{q}'\right)^{T},\left({}^{O}\mathbf{q}''\right)^{T},\left({}^{C}\mathbf{q}\right)^{T},\left({}^{C}\mathbf{q}'\right)^{T},\left({}^{C}\mathbf{q}''\right)^{T}\right]^{T}$, representing the vectors of displacement, velocity and acceleration again at X, Y, Z of the point O and C. The model utilizes more connecting points C than one in order to make possible connection of more DVAs/DVNs. The exported linearized model has the form of matrix tetrad $\left(\mathbf{A},\mathbf{B},\mathbf{C},\mathbf{D}\right)^{B}$, which constitutes black-box state LTI model of dynamic compliance at Laplace domain (L-image) as

$$\begin{aligned} s\mathbf{x}(s)&=\mathbf{A}^{B}\mathbf{x}(s)+\mathbf{B}^{B}\mathbf{Q}(s) \\ \mathbf{q}(s)&=\mathbf{C}^{B}\mathbf{x}(s)+\mathbf{D}^{B}\mathbf{Q}(s) \end{aligned}, \tag{1}$$

or in more compact notation as $\boldsymbol{\alpha}^{B}(s)$

$$\mathbf{q}(s)=\boldsymbol{\alpha}^{B}(s)\mathbf{Q}(s). \tag{2}$$

The isolated DVA/DVN dynamic compliance $\boldsymbol{\alpha}^{A}(s,\mathbf{p})$ represented by a state LTI white-box model with tunable parameters $\mathbf{p}$ defined by the matrices $\left(\mathbf{A}(\mathbf{p}),\mathbf{B}(\mathbf{p}),\mathbf{C}(\mathbf{p}),\mathbf{D}(\mathbf{p})\right)^{A}$, is related to its own coordinate system having the state form L-image of equation of motion

$$\mathbf{q}^{A}=\boldsymbol{\alpha}^{A}(s,\mathbf{p})Q^{A}, \tag{3}$$

with kinematic output $\mathbf{q}^A = \left[q^A, sq^A, s^2q^A\right]^T$, input exciting force Q^A and matrices setting by

$$\mathbf{A}^A(\mathbf{p}) = \begin{bmatrix} 0 & 1 \\ -k^A/m^A & -b^A/m^A \end{bmatrix},\ \mathbf{B}^A(\mathbf{p}) = \begin{bmatrix} 0 \\ 1/m^A \end{bmatrix},\ \mathbf{C}^A(\mathbf{p}) = \begin{bmatrix} 0 & 1 \\ 1 & 0 \\ -k^A/m^A & -b^A/m^A \end{bmatrix},\ \mathbf{D}^A(\mathbf{p}) = \begin{bmatrix} 0 \\ 0 \\ 1/m^A \end{bmatrix}, \qquad (4)$$

where m^A, b^A and k^A mean the seismic mass of the DVA/DVN, the damping and the stiffness. So the state LTI $\boldsymbol{\alpha}^A(s,\mathbf{p})$ keeps known dependency on parameters $\mathbf{p} = \left[m^A, b^A, k^A\right]$.

Taking into account a prismatic joint between the machine tool and DVA/DVN at point C, these are then mutually influenced by interaction forces ${}^C\mathbf{Q}_i$, Q_i^A according to

$$\begin{aligned} \mathbf{q} &= \boldsymbol{\alpha}^B(s)\left[\left({}^O\mathbf{Q}\right)^T, \left({}^C\mathbf{Q}\right)^T + \left({}^C\mathbf{Q}_i\right)^T\right]^T, \\ \mathbf{q}^A &= \boldsymbol{\alpha}^A(s,\mathbf{p})\left(Q^A + Q_i^A\right) \end{aligned} \qquad (5)$$

with the white-box model of the interface describing this interaction given as

$$\begin{bmatrix} Q_i^A \\ {}^C\mathbf{Q}_i \end{bmatrix} = \begin{bmatrix} -m^A s^2 \left({}^C\mathbf{q}\right)^T \mathbf{h}^A \\ -\left(m_D^A + m^A\right)s^2\left({}^C\mathbf{q}\right) - \mathbf{h}^A\left(m^A s^2 q^A\right) \end{bmatrix}. \qquad (6)$$

Here m_D^A is frame mass of the DVA/DVN and $\mathbf{h}^A = \left[\cos\theta^A \cos\varphi^A, \cos\theta^A \sin\varphi^A, \sin\theta^A\right]^T$ describes the orientation of the coordinate system of the DVA/DVN model with respect to the coordinate system of the machine model via spherical angles (Fig. 2, 3). Note that due to the fact only pure mechanical interaction between machine and DVA/DVN without additional acting forces at the point C are studied, ${}^C\mathbf{Q} = \mathbf{0}$ and $Q^A = 0$ is set in (5).

The complete model $\boldsymbol{\alpha}(s,\mathbf{p})$ of the machine tool dynamic compliance with the DVA/DVN connected at the point C through the interface is obtained by applying the rules of interconnections of state LTIs. Now, it contains the vector of tunable parameters $\mathbf{p} = \left[m_D^A, m^A, b^A, k^A, \theta^A, \varphi^A\right]$, see (4, 6), and it represents grey-box model defined by matrix tetrad $\left(\mathbf{A}(\mathbf{p}), \mathbf{B}(\mathbf{p}), \mathbf{C}(\mathbf{p}), \mathbf{D}(\mathbf{p})\right)$.

Note that elapsed time of frequency responses for specific parameters of grey-box state LTI is significantly lower than for the same computation performed directly on the corresponding complete MBS model of the machine with the DVA/DVN. On the other hand it is often accompanied by a partial loss of the response accuracy.

The elapsed time of the frequency response from the state LTI was in this particular case approximately 30x lower than in MBS software with the equality of the responses up to 2% in case of DVA/DVN attached to the tool holder, support and drive of the spindle and up to 5% in case of the attachment to the crossbeam.

The exported black-box model of the machine tool $\boldsymbol{\alpha}$ presents the linear approximation which is also influenced by consequent reduction of its order. The equality of MBS and linear model will be deeper investigated in the further work.

3. METHOD AND OBJECTIVE FUNCTIONS

3.1 Method and its general frame

From broader perspective, the general idea is based on the extension of modeling and simulation stage in the V-cycle (standardized approach to design of mechatronic systems [VDI 2004]) by a micro-cycle. The first pass though the micro-cycle utilizes a multi-body basic model with flexible bodies, in this case of a machine tool with a nature of black-box model. A rough added white-box model, in this case of a DVA or DVN with tunable parameters, is connected to the basic model resulting into grey-box composed model using which optimal parameters are consequently searched (in our case DVA/DVN parameters for maximum suppression of dominant frequency response magnitude). The second pass of the micro-cycle then works with the best obtained results which are used now as initial parameters for another consequent optimization, alternatively other parameters making the basic model more precise can be added but most importantly the more precise added model is used. It can be generally performed several passes through the micro-cycle. The every pass of the micro-cycle makes the model more precise by adding other parameters and perform a (re)optimization procedure.

The (re)optimization procedure utilizes state LTI which makes possible to use sophisticated methods of modern control theory established for the manipulation with dynamics. Significantly shorter elapsed time of frequency responses for specific parameters of grey-box LTI model with tunable parameters offers wide range of possibilities for algorithm applications requiring large amounts of such computations. The minimization is one example of such computations.

3.2 Search for the optimal DVA/DVN parameters

The optimal parameters $\mathbf{p}_{opt}$ of the DVA/DVN are searched via a constrained minimization $\mathbf{p}_{opt} = \arg\min g(\mathbf{p})$ of scalar objective function $g(\mathbf{p})$. The choice of the specific objective function significantly influences final values of searched parameters. During the objective function assembly, it is necessary to take into account that the state LTI model of the machine tool $\boldsymbol{\alpha}^B(s)$ is based on the MBS model containing the flexible bodies. The axes of the machine are mutually influenced because of that fact, thus it is not sufficient to improve the behavior of one of them but it is necessary to introduce to the objective function a requirement for the improvement in all of axes. Contrary to that, it has the sense to simultaneously minimize only the magnitude responses which do not contain both explicit (i.e. prescribed directly by the objective function) and implicit (i.e. from/to DVAs/DVNs) transfer dynamic compliances which differ more than 10x in static compliance. A violation of that is usually just slowing down and retarding the minimization process.

Bearing in mind that the dynamic compliance with respect to point O is much lower in Z axis than in X and Y axis, the Z axis is not included into the objective function.

The frequency response of dynamic compliance $\boldsymbol{\alpha}^B(s)$ by (2) between axis k and l, $k,l = \mathrm{X,Y,Z}$ at the points $p = \mathrm{O,C}$ could be expressed and simply reached as

$$ {}^p\alpha^B_{k,l}(j\omega) = \frac{{}^p q_k(j\omega)}{{}^p Q_j(j\omega)} \tag{7} $$

and the same static compliance as

$$ {}^p\alpha^B_{k,l}(0) = \frac{{}^p q_k(0)}{{}^p Q_j(0)}\ . \tag{8} $$

Two objective functions were used. The first evaluates the higher from maxima of the frequency response magnitudes in X, Y axes of the dynamic compliance ${}^O\boldsymbol{\alpha}(s,\mathbf{p})$ at the checking tool holder masspoint O inside the frequency range $1 \le f \le 1000$ [Hz], related to the same without the DVA ${}^O\boldsymbol{\alpha}^B(s)$, see (eq3), i.e.

$$ g(\mathbf{p}) = \frac{\max\limits_{k=X,Y}\left(\max\limits_{f}\left|{}^O\alpha_{k,k}(j2\pi f)\right|\right)}{\max\limits_{k=X,Y}\left(\max\limits_{f}\left|{}^O\alpha^B_{k,k}(j2\pi f,\mathbf{p})\right|\right)}\ . \tag{9} $$

The second objective function takes again the higher from the maximum magnitudes in X, Y axes but normed separately for each axis

$$ g(\mathbf{p}) = \max_{k=X,Y}\left(\frac{\max\limits_{f}\left|{}^O\alpha_{k,k}(j2\pi f)\right|}{\max\limits_{f}\left|{}^O\alpha^B_{k,k}(j2\pi f,\mathbf{p})\right|}\right). \tag{10} $$

Note that the acting of both objective functions slightly differs. While the first does not guarantee simultaneous improvement (suppression of the maximum response magnitude) in both of X and Y axes, the second does it but often not with so deep suppression as the first one.

4. NUMERICAL EXPERIMENTS

Recall that the numerical experiments were focused on analysis of suppression of resonant magnitudes of the tool holder in X, Y axes (9, 10). There were consecutively compared and analyzed following cases: a) influence of single optimized DVA/DVN connected at point C on the tool holder (see Fig. 2), published in [Brezina 2015b]; b) influence of two optimized DVAs/DVNs simultaneously connected at their points C^L and C^R on the support (see Fig. 3); c) influence of two DVAs/DVNs simultaneously connected at their points C^L and C^R on the crossbeam and d) influence of single DVA/DVN connected at point C on the drive of the spindle; e) additionally the influence of all specified separately optimized DVAs acting simultaneously was introduced.

The numerical experiments were performed for two options. The first option presents results for fixed spherical angles θ^A and φ^A. The angles were fixing the absorber directly in one of axes X, Y or Z during the optimization process. These choices are interesting from the point of view of the realistic physical accessibility. The second option worked with absorber orientation angles in given collision free range, thus also orientation angles were optimized. Stiffness and damping of the DVAs/DVNs were optimized as $k^A \le 1.10^4\ \mathrm{N/mm}$, $b^A \le 20\,\mathrm{Ns/mm}$ to have chance to realize them as passive absorbers in the future. Masses of DVAs were kept fixed during the optimization process (except the tool holder) and frame mass was $m^A_D = 0kg$ in this first pass through micro-cycle.

4.1 Tool holder

The best results for the tool holder (Fig. 4) correspond with experiment No. 2 in Tab. 1 and they are presented for $m^A_{opt} = 5kg$ (the constraint was $m^A \le 5kg$). The numerical experiments for the tool holder were originally presented only for orientation angles $0° \le \theta^A_{opt} \le 360°$ and $\varphi^A = 0°$. The magnitudes were decreased by 23% in X axis and by 25% in the Y axis in this particular case. There were presented also better results in [Brezina 2015b] obtained for the higher mass of the DVA but the mass 5 kg was later approved as more acceptable for the construction. Note that DVA optimization utilizing objective function (9) (experiment No. 1) obtained even better results for Y axis but on the other hand improvement in X axis was very poor. DVN offered almost insignificant improvement in Y axis and in X axis behavior got even worse than of the basic model.

Table 1. Obtained optimal parameters for DVA/DVN connected to the tool holder according to [Brezina 2015b].

Experiment No.	Objective function	Improv. q_x / Q_x [%]	Improv. q_y / Q_y [%]	m^A_{opt} [kg]	b^A_{opt} [N.s/mm]	k^A_{opt} [N/mm]	θ^A_{opt} [°]
1	(9)	1,34	30,5	5	2,03	1250	88
2	(10)	23,4	25	5	1,91	1320	252
3	(9)	-1,08	5,43	5	-	47,3	121
4	(10)	-1,97	0,43	5	-	19,9	95

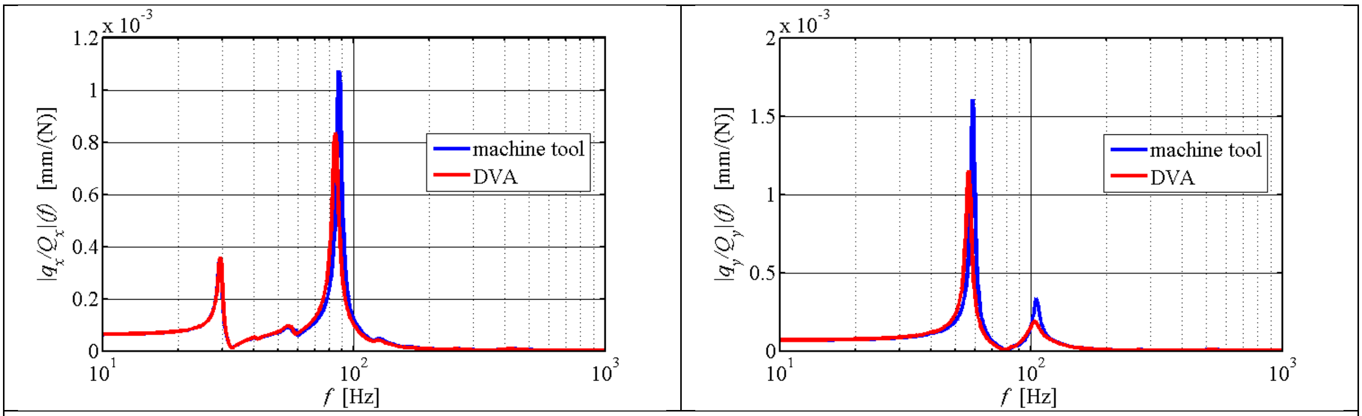

Figure 4. Frequency responses for the system with DVA connected to the tool holder (experiment No. 2)

As said above the static compliance of the machine is at connection point C on the tool holder similar in X and Y axis thus vibration suppression was quite successful in both axes. Contrary to that, it had no sense to compensate Z axis with significantly lower compliance. In the case of attachment points on the support, crossbeam (C^L, C^R) and drive of the spindle (C) is also characterized by low compliance of X axis so the vibration suppression of Y is more promising here. Results for X axis will be presented only in the form of percentage suppression of dominant resonant magnitude.

4.2 Support

The optimizations of the parameters of DVAs/DVNs attached to the support worked with fixed values of the mass for each of absorbers $m^A = 25kg$. The fixed angles option worked with $\theta^A = 180°$ for the left DVA/DVN and $\theta^A = 360°$ for the right one (see Fig. 3, connecting points C^L - left, resp. C^R - right). The second option had angles limited in range $180° \le \theta^A \le 270°$ for the left DVA/DVN and $270° \le \theta^A \le 360°$ for the right. Both options had $\varphi^A = 0°$.

The DVAs/DVNs connected with the fixed angles brought almost no improvement in the behavior (Tab. 2). On the other hand, optimization of orientation angles in the given region brought much better results and maximum resonant magnitude in the Y axis was decreased by 34 % (experiment No. 5, 9), Fig. 5. Let’s note that resulting optimal angles are placing the DVA/DVN directly to Y axis in the case of the left DVA/DVN and very close to it in the case of the right one. The difference in obtained percentage improvement was for DVA and DVN very small. Note that better results are brought for the objective function (9).

Also the important thing to note is that DVNs are introducing to the system parasitic resonances (Fig. 5) which would probably negatively influence the real system.

Table 2. Obtained optimal parameters for DVAs/DVNs connected to the support

Experiment No.	Objective function (9)	Improv. q_x / Q_x [%]	Improv. q_y / Q_y [%]	b^A_{opt} left; b^A_{opt} right [N.s/mm]	k^A_{opt} left; k^A_{opt} right [N/mm]	θ^A_{opt} left; θ^A_{opt} right [°]
5	(9)	0,36	33,9	1,03; 2,03	3360; 3360	270; 298
6	(9)	0,40	5,01	0,03; 0,03	3280; 3300	-
7	(10)	0,50	3,35	1,03; 16	7240; 8470	180; 271
8	(10)	1,37	3,24	0,03; 0,03	3330; 7510	-
9	(9)	0,32	33,85	-	3090; 3610	270; 274
10	(9)	0,40	5,24	-	3330; 3250	-
11	(10)	0,93	3,75	-	7520; 2720	180; 334
12	(10)	1,64	3,72	-	3340; 7510	-

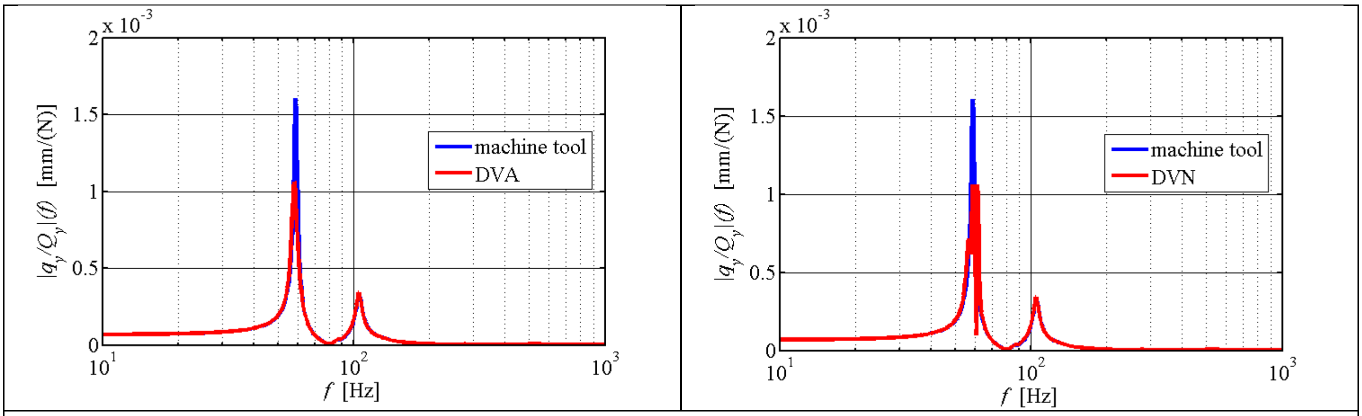

Figure 5. The best results for DVA (experiment No. 5) and DVN (experiment No. 9)

4.3 Crossbeam

Similarly to the case of the support, the mass of each of two DVAs/DVNs was fixed to $m^A = 25kg$ during the optimization process. The option for fixed angles worked with $\varphi^A = 90°$ for left and right absorber (see Fig. 3) and the option with limited angles worked with ranges $90° \le \varphi^A \le 180°$ for the left DVA/DVN and $90° \le \varphi^A \le 180°$ for the right DVA/DVN. $\theta^A = 0°$ for both options.

Contrary to the results of optimizations performed for the support, the fixed orientation angles optimization brought in case of DVA 34 % decrease of the maximum resonant magnitude for the Y axis and optimization searching for optimal orientation angles brought only mild improvement (Tab. 3). The results are obtained for the objective function (9) again.

The frequency responses for best DVA and DVN are presented in Fig. 6. Again, the parasitic frequency is observed for DVN.

Table 3. Obtained optimal parameters for DVAs/DVNs connected to the crossbeam

Experiment No.	Objective function	Improv. q_x / Q_x [%]	Improv. q_y / Q_y [%]	b^A_{opt} left; b^A_{opt} right [*N.s/mm*]	k^A_{opt} left; k^A_{opt} right [*N/mm*]	φ^A_{opt} left; φ^A_{opt} right [°]
13	(9)	0,64	18,6	0,03; 0,03	3440; 2940	87; 129
14	(9)	0,47	34	0,16; 0,28	3480; 3360	-
15	(10)	13,3	11,5	0,03; 2,03	3420; 8160	79; 92
16	(10)	17,7	15,6	0, 16; 0,78	3410; 7450	-
17	(9)	0,08	11,4	-	3440; 3450	74; 31
18	(9)	0,32	19,3	-	3490; 330	-
19	(10)	12,2	10,4	-	3430; 7940	71; 97
20	(10)	10,4	8,6	-	3440; 7940	-

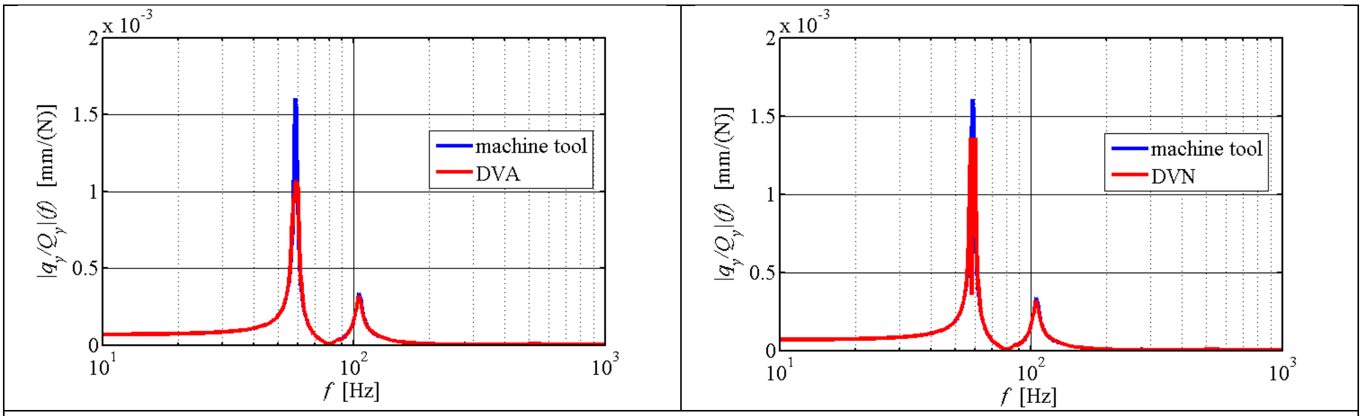

Figure 6. The best results for DVA (experiment No. 14) and DVN (experiment No. 18)

4.4 Drive of the spindle

There was a single DVA/DVN attached to the drive of the spindle (Fig. 3) with fixed mass $m^A = 20kg$. The option for fixed angles worked with $\varphi^A = 90°$ and $\theta^A = 0°$. The limited angles option worked with $0° \le \varphi^A \le 180°$ and $0° \le \theta^A \le 360°$.

The results for fixed and optimized orientation angles are very similar in this case and they brought for DVA again decrease of the resonant magnitude around 34 % (Tab. 4). Best results for DVA and DVN are presented in Fig. 7. It is obvious again objective function (9) brought better results and parasitic resonance for DVN is very noticeable.

The results for fixed and optimized orientation angles are very similar in this case and they brought again decrease of the resonant magnitude around 34 % (DVA).

Table 4. Obtained optimal parameters for DVA/DVN connected to the drive of the spindle

Experiment No.	Objective function	Improv. q_x / Q_x [%]	Improv. q_y / Q_y [%]	b^A_{opt} [N.s/mm]	k^A_{opt} [N/mm]	φ^A_{opt} ; θ^A_{opt} [°]
21	(9)	1,32	34,5	2,03	2820	56; 168
22	(9)	1,14	32,9	1,78	3220	-
23	(10)	6,6	8,83	2,03	5500	157; 244
24	(10)	1,53	6,41	0,03	5960	-
25	(9)	1,2	26,6	-	3070	62; 108
26	(9)	1,15	19,9	-	3070	-
27	(10)	4,94	7,14	-	5420	161; 227
28	(10)	1,9	6,37	-	5970	-

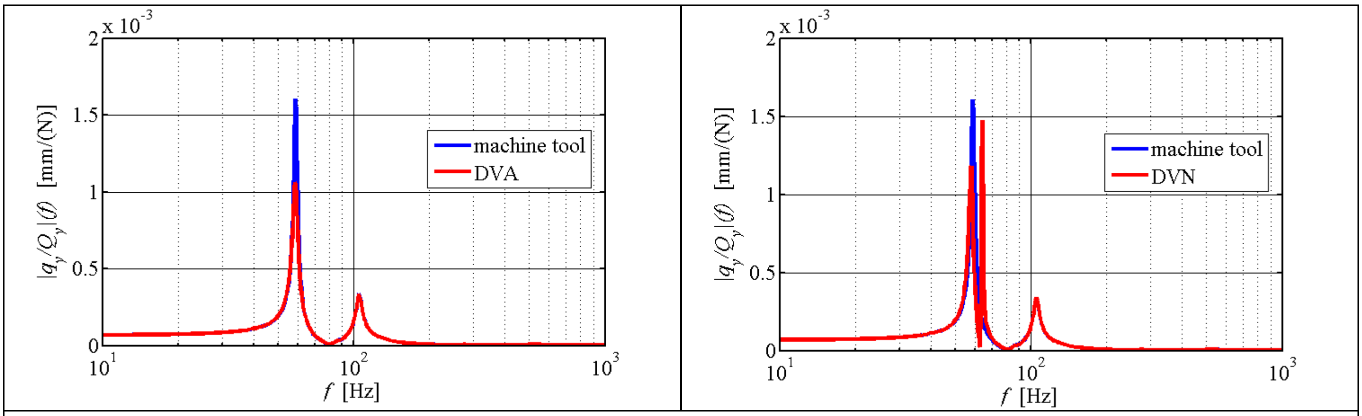

Figure 7. The best results for DVA (experiment No. 21) and DVN (experiment No. 25)

4.5 Combination of support, crossbeam, drive of the spindle and tool holder DVAs

The Fig. 8 presents the resulting frequency responses in X and Y axis for simultaneously attached best tuned DVAs (experiment No. 5, 14 and 21) together with tuned DVA connected to the tool holder according to [Brezina 2015b] (experiment No. 2).

The combination of DVAs suppresses the resonant magnitude in X axis by 23% which is mainly provided by the DVA attached to the tool holder and Y axis damped by 53% (compared to single DVA attached to the tool holder which provided 23,4% in X axis and 25% in Y).

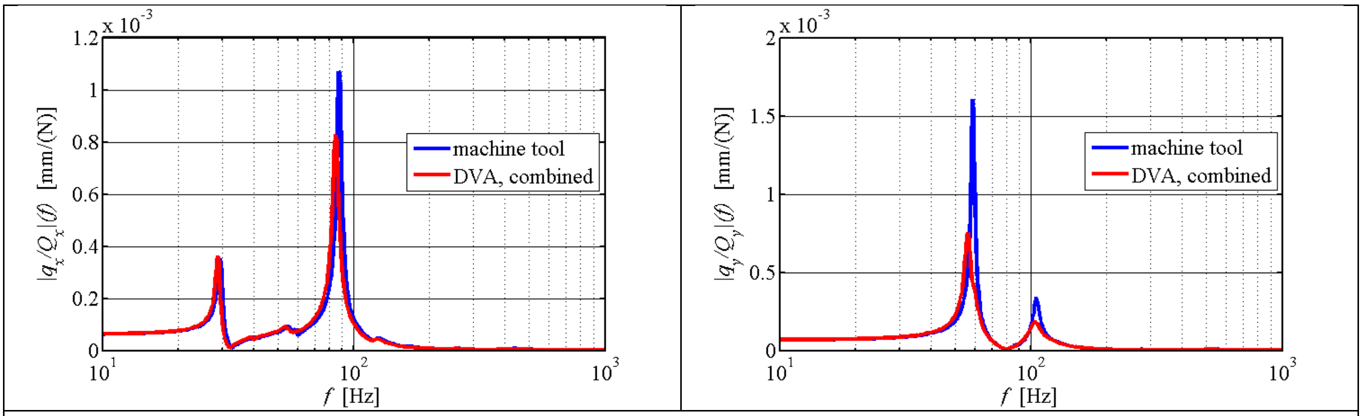

Figure 8. The resulting frequency responses for combination of DVAs

5. CONCLUSIONS

The obtained results, which should be understood as initial for the early pass through the modelling and simulation stage of V-cycle, brought among the obtained of particular values of parameters also these general knowledge:

a) it can be expected that the dominant magnitude of the Y axis of the studied machine tool can be substantially suppressed (by more than 30%) with utilization of passive DVA or DVN. Considered seismic masses are small enough that they influence dynamics of the machine only insignificantly;

b) it can be expected that the dominant magnitude of the X axis can be substantially suppressed only with DVA/DVN attached to a body with points O and C placed on tool holder in this case;

c) DVNs usage seems to be problematic. Although in all experiments DVN brings suppression of dominant resonant magnitudes, at the same time it also brings additional (parasitic) resonances which are close to the original dominant resonances. This fact covers possible risk in their later deployment;

d) DVAs/DVNs attached directly in direction of one of axes give satisfying results in case of crossbeam and drive of the spindle;

e) simultaneous attachment of the best individual DVAs found makes possible to damp X and Y axis better than only single DVA attached to the tool holder even though result of their simultaneous attachment does not comprise an expected synergic effect.

Recall that achievable impact of damping is significantly shaped by static transfer compliances playing a role both in explicit transfers used in objective function and implicit transfers from/to attached DVAs/DVNs. This results into following behavior of the objective function: for tool holder DVA/DVN which makes possible to efficiently damp X and Y simultaneously (due to similar and reasonable static compliance at X and Y axis compared with too low compliance of Z axis) objective function by (10) brings better results. Like this, the support, crossbeam and drive of the spindle DVAs make possible only to damp Y axis (characterized by too low compliance of X and Z axis against Y) and objective function by (9) gives better results than (10).

Frame mass m_D^A, defined as $m_D^A = m^A$, is added to the (re)optimized parameters in the second pass of the micro-cycle. Its influence on the observed frequency responses is evaluated after a location of more precise optimal parameters, i.e. for optimal parameters corresponding with best obtained results in the previous pass (experiment No. 2, 5, 14, 21). In the consequent passes through the micro-cycle, this initial estimation is corrected by more precise model of DVAs/DVNs with frame mass defined in a more detailed way than in (6).

Finally let us note that due to the complexity of the objective function course it is difficult to guarantee that true optimal (best possible) parameters of DVA/DVN were reached. But it can be claimed that satisfying suboptimal solutions, i.e. solutions providing significant improvement of dynamics of the studied machine tool, were found.

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