MACHINE TOOL LIGHTWEIGHT DESIGN AND ADVANCED CONTROL TECHNIQUES

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Mass reduction of the machine tool movable parts is a tool for achieving lower energy demands of the machine tool operation. Lower weight can bring elevation of the machine tool dynamic properties with the application of new materials, which assure the equivalent static and dynamic compliancy compared to the usage of common materials. However, if the machine frame compliancy is increased, it is possible to compensate the missing stiffness by means of advanced control methods. The paper introduces the results of the RCMT involvement in the EU project EcoFIT, which is aimed at the development of machine tools with significantly reduced mass of movable structures.

Keywords

machine tool lightweight design, machine tool energy consumption, non-conventional materials in the machine tool design, control of flexible structures, feed drive complex model

1. Introduction

Engineering practice classical approach in the machine tool design follows the aspect of the possibly highest machine frame static stiffness achievement. Besides the need of reaching high stiffness at the tool centre point (TCP), the mentioned principle takes into account the commonly applied method of position measurement in locations relatively distant from the TCP as well. With the prerequisite of the TCP positioning control based on the mentioned measuring method the need of the position error minimization between the measuring scale and TCP arises. Fulfilment of such requirement has until now been usually attained by means of static and dynamic stiffness enhancement of the machine tool parts. Using steel and cast iron, materials commonly applied in the machine tool design, leads to the design of quite heavy structures. Decisive amount of the material applied is used to meet the stiffness criterion, whereas only a small fraction of the material provides kinematical functionality of the machine tool. Machine tool heavy movable parts consequently require feed drive design with appropriate dimensioning of its components and sufficiently powerful motor.

Mass of the machine structure and motor power decides about the overall energy demands of the machine tool operation. Energy consumed by the motor during the machine operation is besides the force spent to counteract the cutting forces proportional to the passive resistance forces magnitude and force applied to accelerate or decelerate the machine tool components. Acceleration forces component becomes very important especially with the high speed cutting (HSC) processes, for which increasing number of new machine tools are being designed. Appropriate capacity of the cooling and lubrication systems relates to the motor power and size of the machine tool components as well, whereby energy consumption of those systems represents another part of the machine tool overall energy budget.

Until now there has not been much attention paid to the question of the energy consumed during the machine tool operation. Considering the still increasing energy prices and with respect to the environmental protection issues, machine tool energy efficiency can become one of the important qualitative parameters applied with the machine tool assessment.

1.1. EcoFIT project and design of energy efficient machine tools with significantly reduced mass

With the machine tool structure weight reduction, the problem of increased frame compliancy arises as a general disadvantage. Limited amount of material used for the construction restricts the damping ability of the lite frame as well.

Solution of the mentioned topics is the issue of the 6th FP European Union funded project EcoFIT. In the international cooperation of research institutions and production companies the project follows the aim of the machine tool significant energy consumption reduction by means of drastic weight reduction, at the same time keeping the comparable machine tool accuracy and productivity. To compensate for the machine frame lower static stiffness and damping properties, advanced control techniques actively suppressing the tool vibrations are being developed. The intention of all the methods is to replace the missing machine structure static stiffness through the so called mechatronic stiffness, which actively employs the control of feed drives.

The paper presents in brief overview the RCMT contribution to the EcoFIT project solution. More information on the project can be found on the web page *http://ecofit.fatronik.com*.

2. Test bed for simulations of lightweight structure behaviour

For the purpose of the advanced modelling and control techniques development, machine tool lightweight design with increased compliancy is represented with a specifically designed test bed, called as ETB-1. Advantage of the test bed is that it can provide well known and described experimental environment, which relates in the case of the EcoFIT project to the topics of:

- Compliant structure design;
- Servodrive feedback control of compliant axis;
- Multibody simulation related command position;
- Advanced damping techniques;
- Mass reduction.

2.1. Mechanical structure of the ETB-1 test bed

There are two motion axes which correspond with a real machine tool typical feed drives on the experimental test bed ETB-1. One axis is driven with ball screw feed drive, the other one is equipped with linear motor. Each of the drives moves a table, on which compliant arm of two types can be installed. Arm tip represents the tool centre point, the movement of which needs to be controlled by means of advanced control techniques with respect to the desired vibration suppression. Whereas passive arm is of constant cross section, active arm

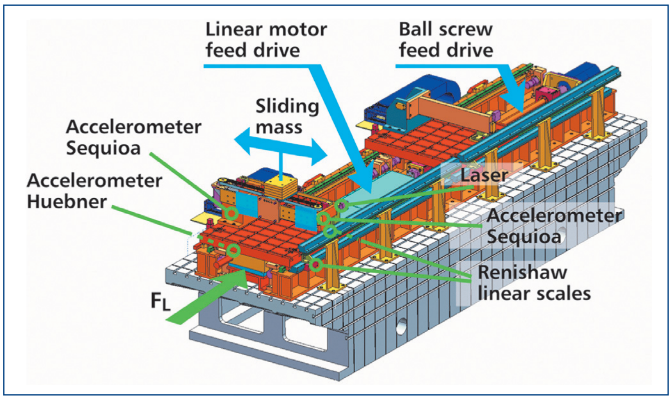

Figure 1. *Test bed ETB-1.*

provides the possibility to change its dynamic properties by means of sliding mass movement, which simulates positioning of spindle or workpiece during the machining process.

There is extensive sensor equipment on the ETB-1. Besides the linear scales there are accelerometers of the Ferraris principle (Huebner) and triaxial Sequoia or D. Electron accelerometers developed in the frame of the EcoFIT project. Oscillations of the arm tip can be measured by means of the laser distance sensor as well.

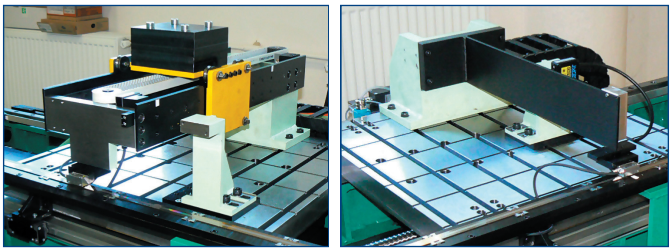

Figure 2. *ETB-1 table with active arm (left) and passive arm (right).*

View of the ETB-1 with indication of the feed drives and sensors is given on the Figure 1, Figure 2 shows detailed views of the realized design of the passive and active arm respectively.

2.2. ETB-1 Virtual model

Virtual model of the test bed is primarily created to provide a mathematical tool for development and verification of the control strategies investigated in the frame of the EcoFIT project. Another issue connected with the virtual model is to verify the technique employed for its creation.

Ball screw feed drive mechanical structure is modelled by means of FE elements and coupled with the test bed structure model directly in the FE environment, whereby boundary conditions applied at the ball screw feed drive components assure the kinematical function of the feed drive. Whole model is subject to the modal analysis, based on which transformation of the FE model into the state-space model is performed using the modal decomposition theory. This way of modelling represents an alternative to the commonly employed Flex-Multi Body System technique, which is however connected with higher modelling effort.

Basic equation of motion written in modal coordinates has the form of

$$\ddot{q} + \mathbf{C_q} \cdot \dot{q} + \mathbf{\Lambda} \cdot q = \mathbf{V}^{\mathrm{T}} \cdot f \qquad (1)$$

where ***q*** is the vector of modal coordinates, $\mathbf{C_q}$ is the matrix of modal damping, Λ is the spectral matrix, **V** is the modal matrix and ***f*** is the vector of external forces. State space input equation with the vector of modal coordinates $\boldsymbol{q_s}$ is written as

$$\dot{q}_s = \mathbf{A} \cdot q_s + \mathbf{B} \cdot u \qquad (2)$$

With the vector $\boldsymbol{q_s}$ composed of the modal coordinates ***q*** and ***q*** it can be shown, that the state space matrixes A and B are filled as follows

$$q_s = \begin{bmatrix} q \\ \dot{q} \end{bmatrix}, \quad \mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{E} \\ -\mathbf{\Lambda} & \mathbf{C_q} \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} \mathbf{0} \\ \mathbf{V}^{\mathrm{T}} \end{bmatrix}, \quad u = f \qquad (3)$$

where ***E*** stands for unity matrix and ***0*** for zero matrix.
Output equation of the system model depends on the number of physical coordinates and generally it can be expressed as linear combination

$$x_s = \mathbf{C} \cdot q_s \qquad (4)$$

where $\boldsymbol{x_s}$ stands for the vector of output displacements and velocities in the selected nodes of the system, whereby the output matrix **C** consists of zero matrixes and the modal matrix

$$\mathbf{C} = \begin{bmatrix} \mathbf{V} & \mathbf{0} \\ \mathbf{0} & \mathbf{V} \end{bmatrix} \qquad (5)$$

State space system is created with the motor torque and table external load as the input and position and acceleration output according the placement of sensors on the ETB-1.

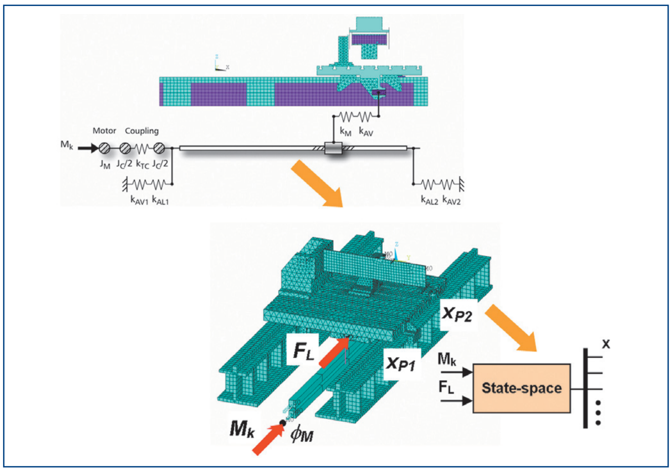

Figure 3. *ETB-1 ball screw feed drive model created in FE environment.*

2.3. ETB-1 control environment

Drives of ETB-1 are using standard electrical and mechanical components – the similar components are used in common machine tools production. There are only a few slight modifications. This is mainly because of the necessity of the additional sensors integration.

All the sensors installed on both feed drive axes (see Chapter 2.1) make it possible to perform extensive diagnostic measurements. Next, signals of the sensors are foreseen for their employment in the control to improve the feed drive behaviour. Among the sensors, attention deserves especially the ultra precise Renishaw metal linear scale and the rotary VUES motor prototype with built-in Ferraris accelerometer.

ETB-1 electrical cabinet is equipped with Control Techniques Unidrive SP servo-amplifiers. There are two basic ways of their control. The first one is depicted on the Figure 4. The amplifier is used in the current control mode and the setpoint is entered via analogue interface within the 250 us timing scale. The second way (Figure 5) uses a digital communication via CANopen interface, where the speed setpoint is transferred into the velocity control loop, closed in the amplifier.

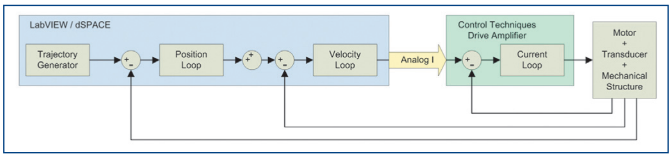

Figure 4. *ETB-1 standard control scheme – current control loop closed in the amplifier, current setpoint transferred via analog interface.*

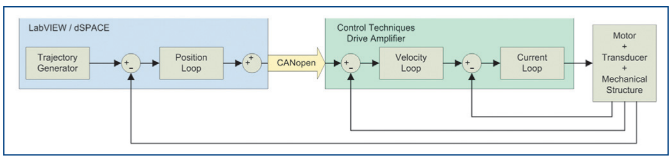

Figure 5. *ETB-1 standard control scheme – current control loop closed in the amplifier, current setpoint transfered via analog interface.*

The first analogue way is less robust, but it is used more often because it enables to control directly the motor acting force (via electric current). The velocity and position control algorithm is then realized outside of the drive amplifier and it is based on solutions from National Instruments. The control platform hardware is Real-time PXI computer with FPGA I/O card, the entire control software is programmed in LabVIEW environment. It enables to create almost arbitrary control algorithm with increased complexity (in comparison with standard drive architecture). The system is also powerful enough to calculate the simplified ETB-1 model in real-time (see Kalman Filter in State-feedback paragraph). Practically achieved timing of the control loop is 250 μs. This is equal to the drive amplifier velocity control loop timing.

Using this control equipment has also another advantage. It is useful not only for the control, but it is also very suitable for data acquisition. This system has an excellent diagnostics, so it is possible to compare efficiency of tested control algorithms.

Figure 6. *ETB-1 electrical cabinet (power equipment on the left hand side, control equipment on the right hand side).*

3. Acceleration signal employment in the velocity control loop

Dynamics of the feed drive control is influenced not only by the dynamic properties of the mechanical system, but also by the quality of signals measured and time delay connected with their acquisition. Within the common cascade control, velocity loop is closed via velocity signal, which however can not be directly measured. In common approach, velocity is calculated based on the motor position signal by means of derivation. Besides the time delay related to the derivation, signal quality is deteriorated with respect to the discrete measurement of the position as well.

The above mentioned drawbacks of the velocity calculation can be reduced with acceleration signal employment in the velocity loop. Integration of the acceleration to velocity signal is faster than the process of derivation; however some shift of the integrated signal occurs in the course of time. To avoid this shift, integrated velocity signal can be corrected by means of the position derived reference signal. Resulting velocity signal therefore arises as a mixture of low frequency component of the linear encoder signal and high frequency component from acceleration signal.

Simulations, as well as the measurements on the ball screw feed axis of the ETB-1 proved, that employment of the acceleration signal brings better quality velocity signal leading to increased dynamic stiffness of the control and reduction of the position deviation, at the same time with the lower energy consumption.

More in detail the technique is presented in [Sveda 2008].

4. State feedback control

State feedback control technique can provide higher control dynamics compared with the common cascade control technique since more states of the system are involved in the control algorithm to compute the acting force. An ideal case would occur if all the states of the system could be measured. However, this assumption can not be fulfilled with the real machine tool and therefore modifications of the state feedback control have to be researched for its practical applicability in the real machine tool control.

Among the various variants of the State feedback control, state observer with Kalman filter and LQR technique for calculation of feedback gains is investigated in the frame of the EcoFIT project.

4.1. State feedback control with observer

Generally, observer technique is used if measurement of all the states of the system is not available. In our case, employment of the observer is needed with respect to the ETB-1 model created in modal coordinates, since modal coordinates can not be directly measured. Procedure of the FEM model transformation via the modal decomposition technique into the State space description expressed by means of the A, B, C and D matrixes has been shown in the Chapter 2.2. Model needs to provide relevant description of the machine tool important frequency characteristics so that good control behaviour of the system could be achieved.

In order to create the feedback control which is capable to compensate position error, the model is additionally extended with the position error integral as a new state. General scheme of the control loop created in discrete domain with state observer is shown on the Figure 7.

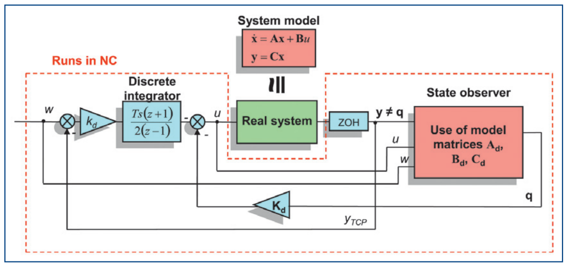

Figure 7. *General scheme of the state-feedback control with state observer.*

Signals measured on the real machine contain noise which deteriorates the control quality. System observation and at the same time filtering of the measured signals is provided by Kalman filter. In discrete domain, Kalman filter is composed of two parts [Welch 2001], predictor and corrector, described with the equations:

- Predictor

$$\hat{q}_{sk|k-1} = \mathbf{A_d} \cdot \hat{q}_{sk-1|k-1} + \mathbf{B_d} \cdot u_{k-1} \tag{6}$$

$$P_{k|k-1} = \mathbf{A_d} \cdot P_{k-1|k-1} \cdot \mathbf{A_d^T} + \mathbf{M} \tag{7}$$

- Corrector

$$K_k = P_{k|k-1} \cdot \mathbf{C^T} \cdot \left(\mathbf{C} \cdot P_{k|k-1} \cdot \mathbf{C^T} + \mathbf{N}\right)^{-1} \tag{8}$$

$$\hat{q}_{sk|k} = \hat{q}_{sk|k-1} + K_k \cdot \left(x_s - \mathbf{C} \cdot \hat{q}_{sk|k-1}\right) \tag{9}$$

$$P_{k|k} = \left(\mathbf{E} - K_k \cdot \mathbf{C}\right) \cdot P_{k|k-1} \tag{10}$$

whereby the state space matrixes of the input equation are transformed into the discrete form, denoted $\mathbf{A_d}$, $\mathbf{B_d}$ and $\mathbf{M}$ and $\mathbf{N}$ matrixes are the output and input covariance matrixes. Employment of the covariance matrixes within the Kalman filter ensures filtering of the noise contained in the measured signals.

4.2. State feedback with LQR technique

For the continuous time system described by means of (2), state-feedback law (11) is according to [The MathWorks] proposed the way it minimizes the quadratic cost function (12)

$$u = -\mathbf{G} \cdot q_s \tag{11}$$

$$\mathrm{J}(u) = \int_0^\infty \left(q_s^T \cdot \mathbf{Q} \cdot q_s + u^T \cdot \mathbf{R} \cdot u\right) dt \tag{12}$$

If discrete time regulator is considered, equivalent discrete gain matrix $\mathbf{G_d}$ is determined by transferring the continuous system (2) to the discrete form and using the weighting matrices **Q** and **R**. Sampling time $\mathbf{T_s}$ and zero-order hold approximation is considered. The discrete state-feedback law then applies

$$\boldsymbol{u}[k] = -\mathbf{K_d} \cdot \boldsymbol{q}_s[k] \tag{13}$$

Calculation of the LQR procedure is performed using the MATLAB software.

4.3. Simulation results

Simulations are performed with the common cascade and state feedback control, whereby the same maximum motor torque is used in both cases. Controlled system is considered to be stochastic one, with white noise on its input and output.

On the Figure 8 and Figure 9, examples of the ETB-1 arm tip velocity response to the motor applied velocity step and the arm tip position deviation as a response to the position ramp is shown on the comparison of the cascade and state feedback control. With cascade control it is not possible to achieve good controllability of the system due to the arm vibrations, according to which lower setting of the controller parameters would have to be chosen. It may be seen, that significantly improved feed drive dynamic properties are attained with the state feedback control using Kalman filter as the state observer. Arm tip velocity oscillations are damped and approximately 30 % smaller position deviation achieved.

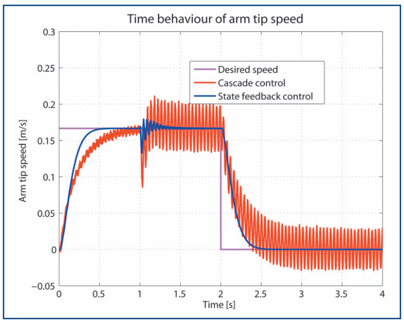

Figure 8. *Passive arm tip velocity as a response to the velocity step.*

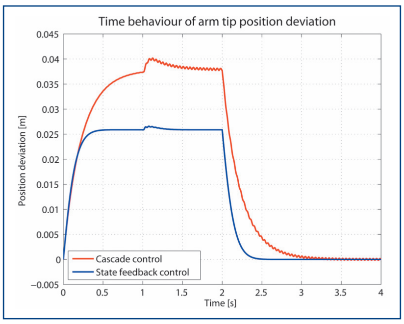

Figure 9. *Arm tip position deviation as a response to the position ramp setpoint signal.*

5. Lightweight design optimization of a real machine tool

Following the EcoFIT main paradigm aiming at the lightweight design of the machine tool structure and corresponding lower energy consumption related both to the production of the materials, use of recycled material sources and machine operation as well, case study of the Kovosvit MCU 2000 machine tool frame optimized design has been performed.

5.1. Structure of the MCU2000 machine tool

MCU 2000 machine tool features three motion axes X, Y and Z, whereby the most compliant and at the same time most heavy movable structure is represented by the cast iron X-axis portal. Portal is therefore the most convenient part of the machine to demonstrate the mass reduction potential resulting from the optimized design. Portal is driven in gantry setup with two ball screw feed drives, the configuration of which is motor – gear belt – rotating ball nut – fixed ball screw.

Optimization effort is primarily focused on the portal overall mass reduction. However, the amount of power required by the motor to move and accelerate the portal relates to the motor reflected mass $\boldsymbol{m_{RED}}$, which is determined as a sum of the total X-axis mass and inertia effects of the ball screw feed drive.

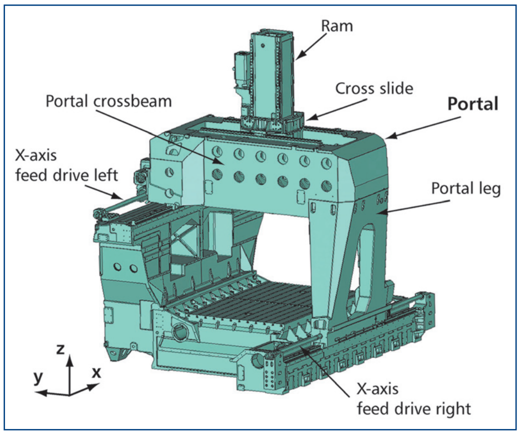

Figure 10. *MCU 2000 structure.*

5.2. New materials for the portal optimized design

For the lightweight design optimization, combination of materials with low density and high Young's modulus is considered. Composite structure is foreseen with aluminium foam ALPORAS as a core material with steel or carbon fiber (CF) laminate skin. Two variants of the laminate skin are considered; ***Lam I*** variant is based on the cheapest carbon prepreg with the T-600 fibres and it is composed of the 49 layers with the overall thickness of 21 mm. The other variant, denoted as ***Lam V***, composed of 70 layers with the overall thickness of 30 mm uses ultra high-modulus fibres. Basic material properties of the aluminium foam and CF composite are listed in the Table 1 and Table 2.

	ρ_C	**E**	$\boldsymbol{G_C}$	υ
	Density	Young's modulus	Shear modulus	Poisson's ratio
	kg.m^{-3}	MPa	MPa	-
ALPORAS	**268**	**597**	**225**	**0.33**

Table 1. *Material properties of the ALPORAS foam.*

	ρ	E_X	E_Y	G_{XY}	υ_{XY}
	Density	Young's modulus	Young's modulus	Shear modulus	Poisson's ratio
	kg.m^{-3}	MPa	MPa	MPa	-
***Lam I* skin**	**1 550**	**74 653**	**27 449**	**16 987**	**0.4577**
***Lam V* skin**	**1 550**	**231 446**	**71 807**	**47 221**	**0.565**

Table 2. *Material properties of the CF composite.*

5.3. X-axis portal optimized design

New concept of the X-axis portal form follows from the application of sandwich material structure. Optimized portal design is shown on the Figure 11, cross-section of the portal cross-beam on the Figure 12 reveals the sandwich structure.

Application of the new sandwich material structure results in cca 50 % portal weight reduction, which means approx. 33 % reduction of the whole X-axis moving mass weight. Masses of the optimized variants are listed in the Table 3.

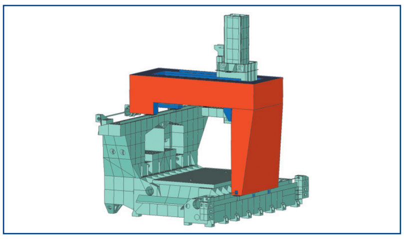

Figure 11. *Optimized design of the portal with sandwich material structure.*

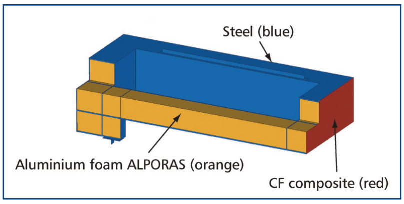

Figure 12. *Sandwich structure of the optimized X-axis portal.*

Variant	Optimized; Lam I	Optimized; Lam V
Portal weight m_P	2540 kg	2722 kg
Total X moving mass m_X	5110 kg	5292 kg
Portal weight m_P reduction	**52.3 %**	**49.0 %**
Total X moving mass m_X reduction	**35 %**	**33 %**

Table 3. *Portal weight reduction.*

5.4. Feed drive optimized design

By means of the RCMT specialized expert software system, design of the feed drive mechanical structure optimized variant has been solved. Reduction by almost 50 % of the motor reflected m_{RED} mass has been achieved; smaller motor used requires 20 % less power. High stiffness of the CF composite assures at the same time an increase of the feed drive mechanical stiffness by 15 % (see Table 4) and increase of the frequency of the first amplitude drop (so called antiresonance frequency) by 23 % in the case of ***Lam I***, or 39 % in the case of ***Lam V*** skin.

Parameter	Current design	Optimized design **Lam I**
Acceleration	3.1 m.s^{-2}	3.1 m.s^{-2}
Motor reflected mass m_{RED}	9280	4780
		Reduction by 48 %
Mechanical stiffness	131 N/µm	151 N/µm
		Increase by 15 %
Motor power	5.81 kW	4.62 kW
		Reduction by 20 %

Table 4. *Parameters of the optimized feed drive.*

For the feed drive dynamic properties simulations, machine tool complex mechatronical model has been used. Bode plot of the open velocity loop with the comparison of the original and optimized feed drive with the ***Lam I*** variant of the portal can be seen on the Figure 13.

Increased value of the first antiresonance frequency gives the presumption for higher position loop ***Kv*** gain achievement.

6. Conclusion

Knowledge gained during the solution of the EcoFIT project show, that the machine tool design with drastically reduced mass can be practically applied employing the researched advanced control techniques. RCMT contribution to the project focused on the development of the specialized test bed and research of some of the advanced control techniques.

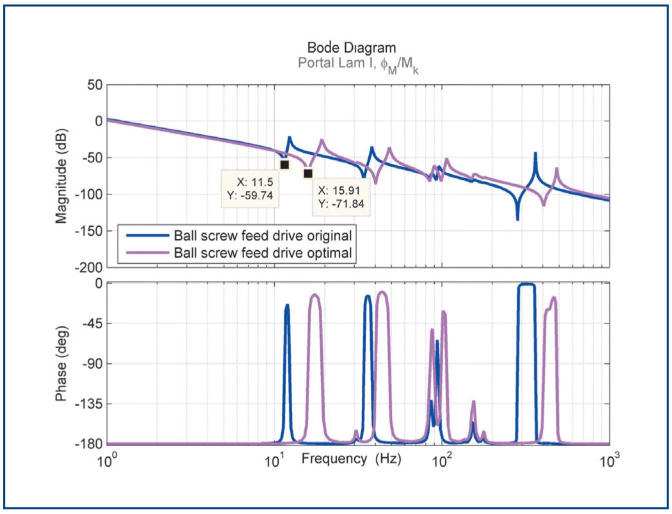

Figure 13. *Open velocity loop transfer function; original and optimized feed drive with Lam I skin portal.*

Employment of the acceleration signal in the velocity control loop results in higher feed drive dynamic stiffness and significantly increased setting of the controller parameters. By means of the state feedback control both higher dynamics of the feed drive control is attained and vibrations actively suppressed.

Case study of the Kovosvit MCU 2000 machine tool optimization proved, that using the non-conventional materials in the machine tool design, significant mass reduction can be achieved enabling to use less powerful feed drive motors. At the same time, static and dynamic stiffness of the feed drive is enhanced.

In the final stage of the EcoFIT project, all of the methods and strategies developed within the consortium will be applied on the real milling machine of the Spain producer Correa Anayak.

Acknowledgements

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