# CORRELATION BETWEEN DYNAMIC PROPERTIES OF LINEAR GUIDE AND RADIAL FORCE LOAD

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A dynamic behavior of machine tools always depends on dynamic properties of machine tool individual parts; for example, a spindle unit, frame, ball screw, linear guide, etc. The present paper deals with an analysis of dynamic properties of linear guide RA55AN. The guide was fixed to a special testing stand which enabled us to change and measure radial force loads of the linear guide via a thread mechanism and a force washer. The aim of this article is to define the correlation between the value of radial force load and the linear guide dynamic properties. A dynamic compliance and modal analysis were used as criteria for evaluation of linear guide dynamic properties.

**KEYWORDS**

modal analysis, natural frequency, damping, linear guide, force load, dynamic compliance

## 1 INTRODUCTION

Machine tools are widely used in all industries. Demands for higher accuracy, production speed, and costs reduction increase the importance of deeper investigations into this field. Currently, machine tool structures are designed with a modularity concept to satisfy the specific industrial applications. Linear rolling guides are widely used for high-precision positing systems in machine tools because of their high carrying capacity and low friction compared to the sliding guides [Rahmani 2016].

Machine tools are complicated systems composed of many components, such as a frame or spindle, with many joints. Dynamic properties of machine tools are not only determined by these components, but also by dynamic properties of various joints [Li 2012]. However, the machine tools cannot be easily formulated as a composition of individual parts because there are various interfaces at connection joints, such as bolted components, which form weak links between the components and affect the overall dynamics and statics of the assembled structures [Yuan Li 2010, Mi 2011]. Zhang in [Zhang 2003] presented that about 60% of total dynamic stiffness and about 90% of total damping in the entire machine tool structure originates in joints. For this reason, the analysis of dynamic behaviour could be split into two parts: identification of joints characteristics and modelling of dynamic parameters of the entire machine tool structure.

In the linear guide components, rolling interfaces exist between the rolling balls and the grooves, which exhibit a nonlinear contact characteristic of Hertz type [Johnson 1985]. Due to the influence of bearing stiffness, spindle-bearing systems have been a research topic of many scientists investigating the variations of dynamic characteristics during operation [Akturk 1997, HARASHA 2003, CAO 2004 and CAO 2007]. In general, the dynamic behaviour of the spindle is mostly determined by the initial preload of the bearings. However, it could deviate from the original behaviour because of variations in preload and stiffness under different operating conditions [Alfares 2007].

Considering nonlinearities of the machine tool joints, the interface characteristic could not be directly obtained using an analysis method unless experimental measurements were made [Huang 1993, Lin 2003].

## 2 EXPERIMENTAL SETUP

To carry out a comprehensive experimental measurement on the linear guide, an experimental setup ensuring repeatability was designed; this setup consists of linear guide, flange, testing stand, thread mechanism, force washer, and equipment for modal analysis.

### 2.1 *Linear guide RA55AN*

A subject of interest was a linear guide RA 55 500 ANC1-0P, which was designed for super-high load or high load depending on the preload setting. The tested linear guide was preloaded for 12.9 kN, i.e. it is suitable for high load.

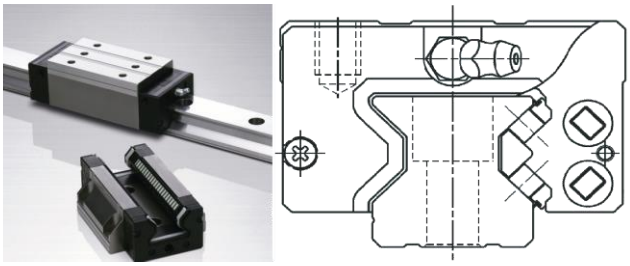

**Figure 1**. Schematic representation of linear guide RA 55 500 ANC1-0P.

### 2.2 *Testing stand*

A special testing stand was designed for the experimental measurement of linear guide dynamic properties at determined radial force load. The testing stand enabled us to set a vertical and horizontal force load with the positive and negative values; tension and pressure load. The thread mechanism allowed us to set the force load from -40 kN to 40 kN.

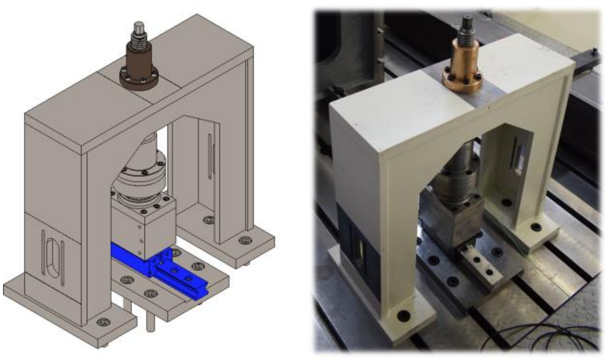

**Figure 2**. CAD model of testing stand and real design.

### 2.3 *Measuring equipment*

The experimental tests were performed via a measurement of force load by Kistler force load washer 9081B and the dynamic properties by Brüel & Kjær equipment based on Pulse platform.

### 2.3.1 *Force washer 9081B*

The force washer was placed between the linear guide and thread mechanism fixed to the testing stand. The measuring range of the force washer was 0 – 650 kN. The force sensor was preload by 47.1 kN to ensure linearity and proper functionality. Owing the preload, it could realize not only the pressure, but also the tensile measurement.

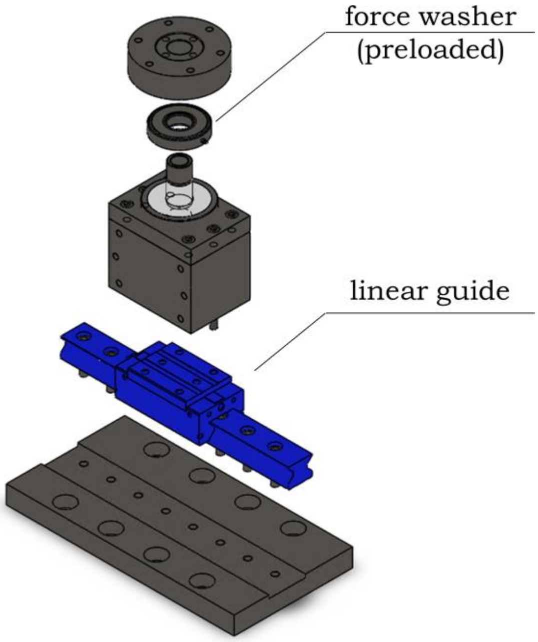


**Figure 3**. Position of the force washer

### 2.3.2 *PULSE platform*

The Brüel & Kjær equipment consisted of a four - channel analyser that enabled us to connect a three-axis accelerometer 4524B and a modal hammer 8206-003 to measure the modal parameters.

The measurement of frequency response functions (FRF) were performed by one-axis accelerometer 4507B and modal hammer 8206-003.

The modal hammer was fitted with a plastic tip that enabled us to measure in the frequency range up to 1.6 kHz. The accelerometers were fixed to the testing stand by magnetic head. In the case of modal analysis, the accelerometer was placed outside the expected nodes. The positions of accelerometer during FRF measurements were always chosen against the direction of the modal hammer hit.

## 2.4 *Experimental procedure*

The force load of the linear guide was systematically varied from -40 kN to 40 kN; the chosen step was 10 kN. There was always a release of the thread mechanism to the zero value of the force load between the two settings of force load. FRFs measurements were performed on the linear guide, but the modal analysis was carried out for the entire testing stand.

### 2.4.1 *Modal analysis setup*

A simplified model of the testing stand and linear guide were made in a Pulse consultant software. The simplified model and measurement strategy are shown in Fig. 4. The three-axis accelerometer was fixed on the structure where the modal node is not expected. The modal hammer successively hit the entire testing stand.

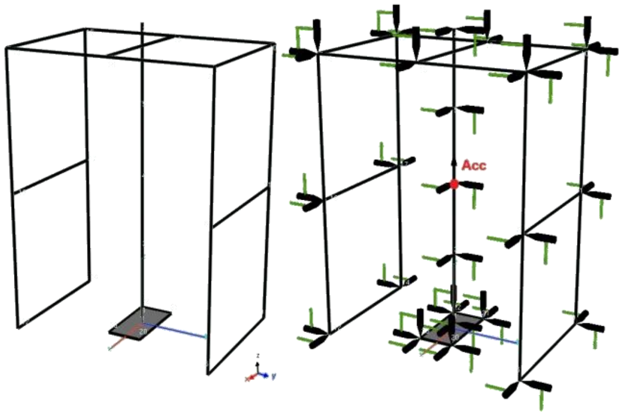


**Figure 4.** Simplified model of testing stand and linear guide.

The modal analysis was performed only for two variants of the force load: a) Force load F = 40 kN and b) Force load F = - 40 kN. The two variants of the force load were made because there were expected no significant changes in the natural frequencies and mode shapes.

### 2.4.2 *FRFs - dynamic compliance setup*

FRFs were measured in directions X, Y and Z; the directions and points of the measurements are presented in Fig. 5. The accelerometer measured the acceleration of vibrations. The measured FRF functions had to be integrated twice to obtain a dynamic compliance.

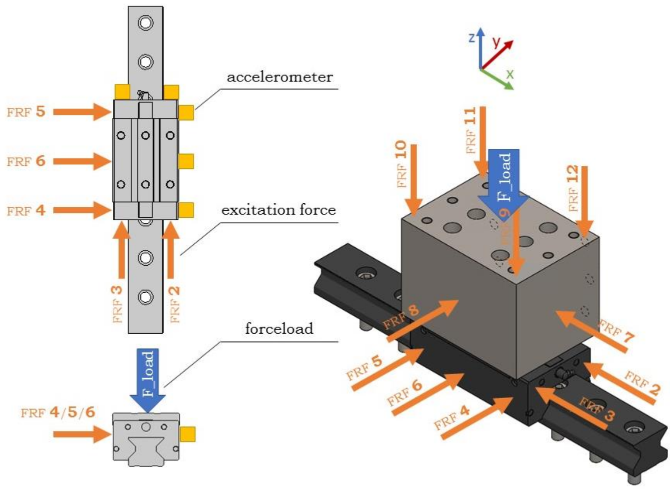


**Figure 5.** Position of FRFs measured on the linear guide

## 3 RESULTS

This chapter describes the results obtained from the modal analysis and FRF – dynamic compliance measurements. The modal analysis allowed for definition of the natural frequencies, damping and mode shapes. However, for identification of the frequencies vital for dynamic structures behaviour, the FRFs – dynamic compliance measurements were needed .

## 3.1 *Modal analysis*

The obtained data showed that the change of force load orientation from the positive to negative value caused a decrease in the linear guide natural frequencies about 15 Hz, except for one mode (no. 5), see Tab. 1.

| Mode | Force load variant a) F = 40 kN | | Force load variant b) F = - 40 kN | |
|---|---|---|---|---|
| | Frequency [Hz] | Damping [%] | Frequency [Hz] | Damping [%] |
| 1 | 58.5 | 11.4 | 41.0 | 13.3 |
| 2 | 124.6 | 3.3 | 107.5 | 2.3 |
| 3 | 150.1 | 2.8 | 135.7 | 4.5 |
| 4 | 175.1 | 1.3 | 172.1 | 1.2 |
| 5 | 317.6 | 3.0 | 229.3 | 4.6 |
| 6 | 452.8 | 3.5 | 438.2 | 3.9 |

**Table 1.** Modal parameters of linear guide and testing stand

There were insignificant changes of the mode shapes obtained from the modal analysis of both variants of the force load. The mode shapes of the natural frequencies are shown in Tab. 2.

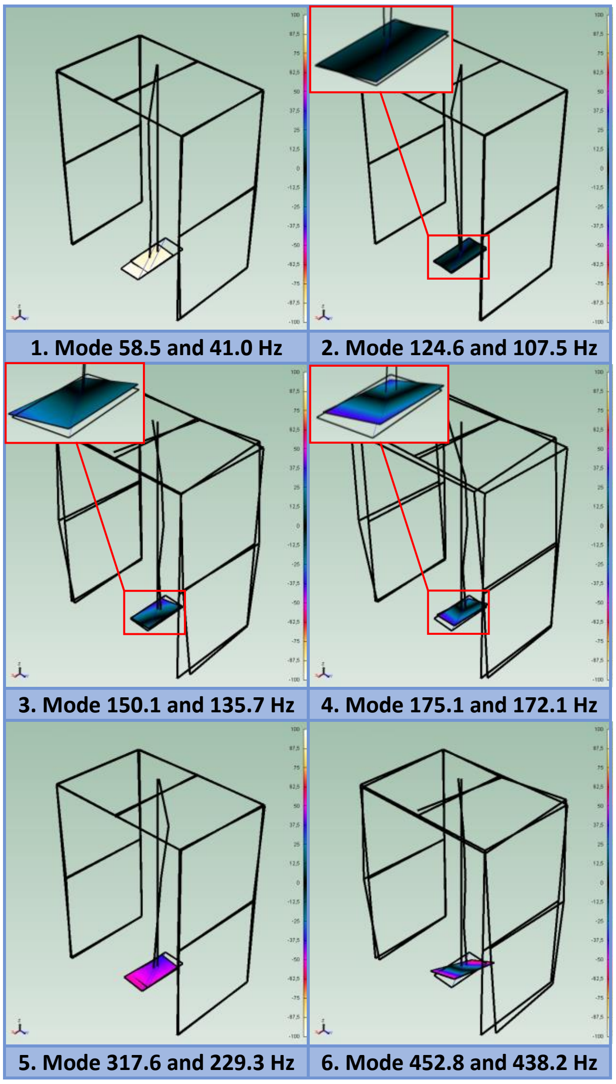


**Table 2.** Vibration mode shapes of testing stand and linear guide

The first mode at the lowest natural frequency had a linear movement in the X axis. In production machines, this is usually eliminated or restricted by a ball screw. The most typical movement of the force load of linear guide is a tilt around the X axis, see the mode shape 2 in table 2.

### *3.2 FRFs - dynamic compliance*

As expected, a dynamic compliance reached the highest values in the X direction. The value of the pressure force load had no effect on the frequency and the amplitude of dominant frequencies, see Fig. 6.

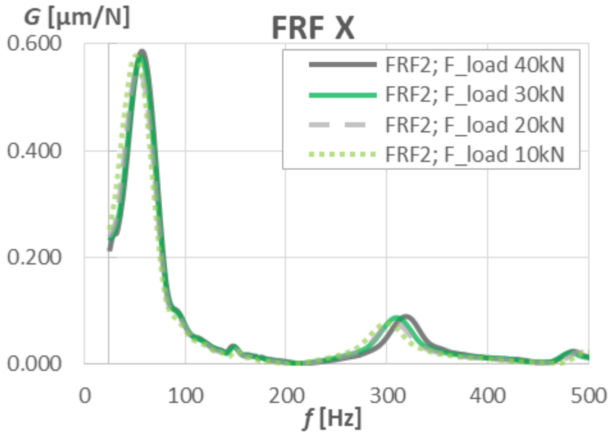


**Figure 6.** FRFs - point 2, direction in axis X, force load from 10 to 40kN

A notable dominant frequency amplitudes decrease of FRFs in the X axis occurred in the course of a change of force load orientation, see Fig 7.

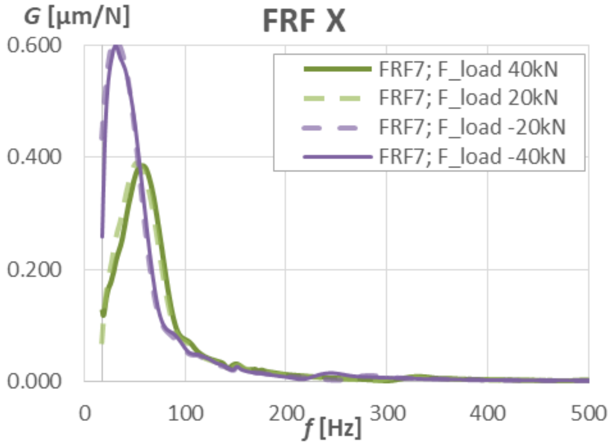


**Figure 7.** FRFs - point 2, direction in axis X, positive and negative force load

There were no differences in the dominant frequencies during a systematic increase in force load in the Y axis, see Fig. 8. The dominant frequency corresponded to the second mode shape – a tilt around the X axis, see Tab. 2.

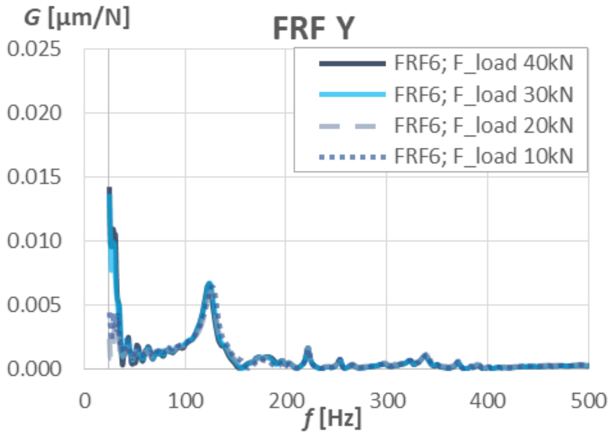


**Figure 8.** FRFs - point 6, direction in the Y axis, force load from 10 to 40kN

The point 6 (middle of the linear guide) was chosen for comparison of the results because the linear movement of the production machine is never provided by a single linear guide but by multiple linear guides. A higher number of the linear guides restrict the mode shapes which could be described as a tilt around the Z axis.

A different behaviour of the dominant frequencies was observed when the negative values of force load were set. The frequency and amplitude of the dominant frequencies changed significantly when the force load was changed from -20 kN to -30 kN, see Fig. 9. An increase in the force load to - 40 kN made no difference.

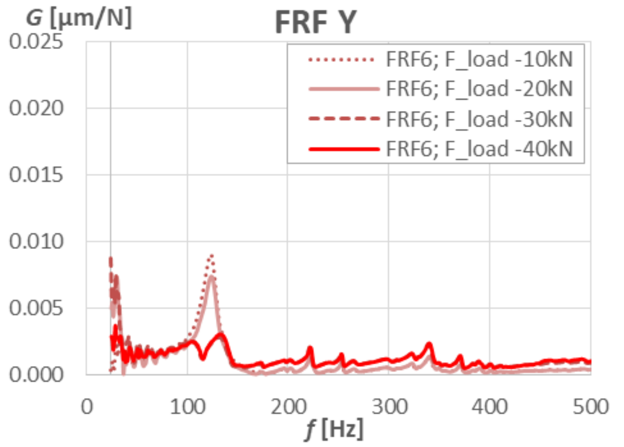


**Figure 9.** FRFs - point 6, direction in axis Y, force load form -10 to -40kN

In general, a dynamic compliance in the Z axis is the most important feature of dynamic properties. FRFs of the linear guide in the Z axis differ significantly for the positive and negative values of the force load, see Fig. 10 and Fig. 11.

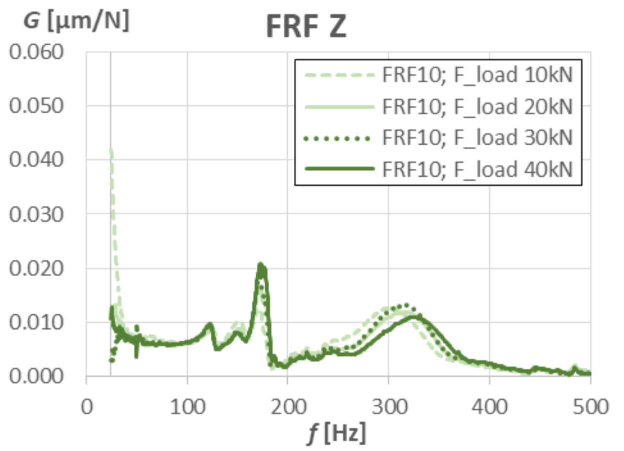


**Figure 10**. FRFs - point 10, direction in axis Z, force load: 10 - 40kN

The frequencies and amplitudes of dominant frequencies were systematically changed during the increase in the values of positive force load. In this case, the dominant frequency corresponds to the fourth mode shape.

When the linear guide was loaded by negative force load values, the frequency and amplitude of the dominant frequencies changed significantly when the force load was changed from -20 kN to -30 kN.

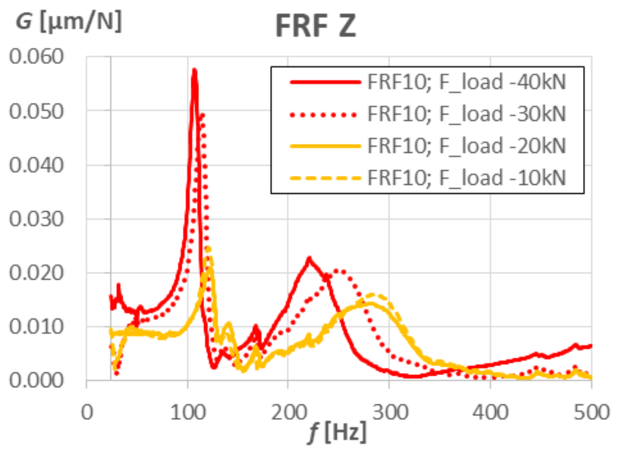


**Figure 11.** FRFs - point 10, direction in axis Z, force load form -10 to -40kN

Other experimental measurements of the modal analysis, FRFs measurement and static stiffness will be performed on the linear guide with force load applied in the Y axis. The goal is to describe all dynamic properties of the linear guide under different types of force load.

## 4 CONCLUSIONS

The analysis of dynamic properties of RA 55 500 ANC1-0P linear guide was performed using a modal analysis and FRFs – dynamic compliance measurements. A combination of modal analysis and FRFs enabled us to identify the dominant frequencies and the corresponding modal parameters. The obtained data will be used to verify and improve of mathematical models. The results are going to be used as a part of the materials for predicting of the dynamic properties of machine tools during force load caused by machining process.

The following conclusions were based on the experimental measurements:

- Dynamic properties had a different character in the middle and on the edges of the linear guide. On the edges, there were different frequencies which corresponded to higher mode shapes.
- The linear guide had better dynamic properties under the positive force load compared to the negative force load.
- An increase in the value of negative force load caused irregular changes of the frequencies and amplitudes of dominant frequencies. This change occurred around the value of force load which corresponds to the preload.
- An increase in the value of positive force load in the Z axis did not result in a significant change of the frequency and amplitude of dominant frequencies. This is essential because the production machines could be forced in a wide range depending on technologies and cutting conditions which directly affect the force load of the linear guides.

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