340. Investigation of Vibrations of Line Scale Calibration Systems

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Abstract: The paper treats the issue of embedding the traceable length metrology directly into technological process by performing precise dynamic measurements of line scale in real industrial environment. It addresses the error-related problems specific to line scale calibration in dynamic mode of operation that are caused primarily by static and dynamic loads as well as geometrical deviations of the calibration system components. Introducing the dynamic regime of calibration leads to the dynamic calibration error originating due to vibration sources in the structure. This term should be considered and implicated in the measurement uncertainty budget. A new 3D finite element model has been developed in order to both investigate the influence of dynamical excitations of a long stroke comparator structure and evaluate possible influence of vibrations on geometrical dimensions of the line scale. The experiments have been conducted on the new interferometer-controlled comparator setup with a moving microscope in order to evaluate the performance of the precision scale calibration process.

Keywords: line scale calibration, FE modeling

Introduction

A thorny issue the designers and manufacturers of precision machines face most frequently is the control of elastic deformations induced by static or dynamic loads that allows obtaining geometrical deviations within a permissible range as well as control of deviations caused by temperature variations. The design objective is not to minimize every source of error but to keep them under control to an adequate level required by the specification for the system. This is particularly true when dealing with large machines that operate in non-ideal environmental conditions under the influence of many external factors that are not part of the desired measurement, namely, which are affected by a wide spectrum of seismic excitations, non-homogeneous temperature fields, electro-magnetic noise and other disturbances.

One of the most complicated fundamental problems that science and the high-tech applications confront today is the length calibration ones, where the new and difficult-to-implement requirements posed by the embedded metrology needs can be met only by developing novel systems that absorb recent scientific and technical findings and optimally comply with specific calibration requirements as well as by improving existing calibration systems open to complying with fundamental principles of precision engineering [1].

The need for productivity improvements in line scale calibration ultimately drives the demand for technologies, which permit to embed the traceable length metrology directly into technological processes by performing precise dynamic measurements in more demanding environments than those of calibration laboratories, providing metrological traceability of accuracy parameters of precision scales to the primary length standard in their manufacture line and reducing both the uncertainty of the scale calibration and calibration process duration.

The major improvements of machine visions systems and new tools such as fast digital cameras, new illumination systems as well as new image scanning and processing technologies and standards allow for high-speed imaging, higher data transfer rates and permit much greater scanning speeds in low light than previously available ones. With faster processors, modern systems can both acquire more information from cameras at faster frame rates and take advantage of more complex line/edge detection and error correction algorithms. This makes it possible to build a high-speed image capture and analysis system to meet particular needs of calibration of precision line scale.

The available modern technologies enable precision line/edge detection in dynamic mode of measurement with a speed up to 20 mm/s. They allow the line scale calibration with measurement uncertainty of less than 40 nm/m and ensure the graduation line detection repeatability of 5 nm, [2].

However, precision systems are often too complicated and different, and it is very difficult or nearly impossible to transfer and adapt the achievements of accuracy-related research to them straightforwardly. Throughput limitations for calibration systems in question are featured by the whole complex mechanical system including error compensation circuits. Therefore, satisfying new demanding and contradictory requirements of high-speed and accuracy for precision line scale calibration calls for the necessity of in-depth analysis of the uncertainty budget including dynamics-induced errors caused by measurement speed fluctuations, time delays, noise and vibrations especially during the graduation line detection. The broad-scope efforts are needed in order to investigate these systems in specific work environments and above all to model small deformations properly. Structures of precision length comparators often are too sophisticated to be modeled precisely by applying simple methods. Complex physical models, finite element models (FEM), and engineering ones as well as their analysis tools are to be applied in order to perform qualitative and quantitative description and analysis of determinants of the precision line calibration process.

The paper describes the recent joint work performed at Kaunas University of Technology and JSC "Precizika-Metrology" in precision length metrology aimed at the development of an interferential comparator with moving microscope for line scale calibration up to 3,5 m long. The research has been focused on the dynamic mode of calibration and simultaneous evaluation of the influence of dynamic and thermal factors upon the inaccuracies of measurement, [3]. In this work we analyze the elastic displacements of the structure as a response to applied dynamic excitations. A precision interferometer-controlled setup of high technology readiness level with NIKON modular CCD microscope has been used in order to carry out the experiments.

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FEM simulations

The state-of-the-art FE technique has been applied in order to evaluate the possible influence of dynamics upon the inaccuracies of measurement. FE models enable us to simulate all 3D displacements or vibration patterns of the structure. The investigation has been performed within ANSYS finite element software, [4].

We analyze the elastic displacements of the structure as a response to applied dynamic excitations by employing the spectral method and assuming that the acceleration $a(t)$ of the comparator foundation is known, Fig.1.

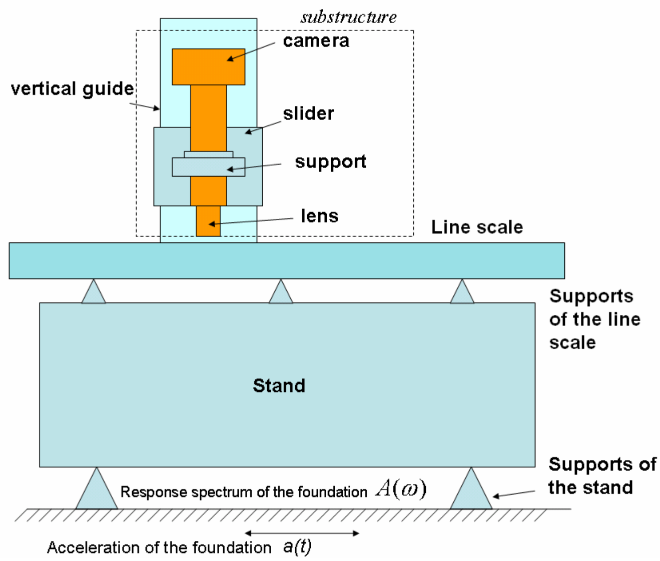


Fig. 1. The scheme of the length comparator with moving microscope

In the scope of precise measurements the main focus is the relative displacement pattern of the measurement zone rather than stresses and strains. In addition, we took into account the elasticity of thread and guide joints between certain parts of the support, which may be influenced by the temperature effects, geometrical inaccuracies and other factors. The sub-model of the thread joint and guide joint has been investigated in order to evaluate its stiffness.

Vibration analysis

In the general 3D case where vibration time law of the foundation is prescribed as the nodal displacement vector $\{U_K(t)\}$, we present the finite element structural equation in the block form as:

$$\begin{bmatrix}[M_{NN}] & [M_{NK}] \\ [M_{KN}] & [M_{KK}]\end{bmatrix}\begin{Bmatrix}\{\ddot{U}_N\} \\ \{\ddot{U}_K\}\end{Bmatrix}+\begin{bmatrix}[C_{NN}] & [C_{NK}] \\ [C_{KN}] & [C_{KK}]\end{bmatrix}\begin{Bmatrix}\{\dot{U}_N\} \\ \{\dot{U}_K\}\end{Bmatrix}+ \\ +\begin{bmatrix}[K_{NN}] & [K_{NK}] \\ [K_{KN}] & [N_{KK}]\end{bmatrix}\begin{Bmatrix}\{U_N\} \\ \{U_K\}\end{Bmatrix}=\begin{Bmatrix}\{0\} \\ \{R\}\end{Bmatrix} \quad (1)$$

where nodal displacement vectors $\{U_N(t)\}$, $\{U_K(t)\}$ correspond to unsupported and seismically excited nodes the displacements of which are investigated with respect to immovable reference systems, $[K]$, $[M]$, $[C]$ are stiffness, mass and damping matrices of the structure presented in the block form in accordance with blocks $\{U_N(t)\}$, $\{U_K(t)\}$ of the displacement vector, and $\{R\}$ is nodal reaction force vector of seismically excited nodes. The displacements of unsupported nodes are presented by separating the relative components of the displacements with respect to moving foundation as $\{U_N\} = \{U_{Nrel}\} + \{U_{Nk}\}$. The displacements described by components $\{U_{Nk}\}$, $\{U_K\}$ present the rigid body motion of the structure and do not create internal elastic forces. In the case of proportional damping $[C] = \beta[K]$, after performing the necessary algebraic manipulations the structural equation finally reads as:

$$[M_{NN}]\{\ddot{U}_{Nrel}\} + [C_{NN}]\{\dot{U}_{Nrel}\} + [K_{NN}]\{U_{Nrel}\} = \lfloor\hat{M}\rfloor \qquad (2)$$

where $\lfloor\hat{M}\rfloor = [M_{NN}][K_{NN}]^{-1}[K_{NK}] - [M_{NK}]$.

The left-hand side of equation (2) presents the structural matrices of the structure fixed at nodes of seismic excitation, and the right hand side contains inertia forces acting at each node of the structure in the case of seismic excitation $\{U_K(t)\}$.

In modal coordinates of an undamped structure equation (2) reads as:

$$\ddot{z}_i + 2\omega_i\vartheta_i\dot{z}_i + \omega_i^2 z_i = \{y_i\}^T\lfloor\hat{M}\rfloor\{\ddot{U}_K\}, \quad i = 1,2,\ldots,n \qquad (3)$$

where $\{U_N\} = \{y_1\}z_1 + \{y_2\}z_2 + \ldots + \{y_n\}z_n$, $\omega_1, \omega_2, \ldots, \omega_n$ - the natural frequencies of the structure, $[Y] = [\{y_1\}, \{y_2\}, \ldots, \{y_n\}]$ - the natural forms of the structure, ϑ_i - the damping ratio of the i-th mode.

The nodal acceleration vector on the right-hand side of (3) can be generally represented as $\{\ddot{U}_K\} = \{V\}a(t)$, where $\{V\}$ – known vector which is specific for each investigated structure, $a(t)$ – prescribed acceleration time law of the foundation.

In accordance with the spectral method, [4], the maximum displacements of each point of the structure are obtained by combining the maximum displacements of each mode. The seismic loading is transformed to the response spectrum of each node of the structure foundation. The value of the displacement response spectrum $D(\omega)$ at frequency ω is obtained as maximum deflection of the linear oscillator loaded seismically by acceleration $a(t)$ as:

$$\ddot{x} + 2\omega\vartheta\dot{x} + \omega^2 x = a(t) \qquad (4)$$

during the time interval of the seismic loading $a(t)$.

Acceleration response spectrum $A(\omega)$ is a relationship of values of maximum acceleration of the linear oscillator (4) seismically subjected to acceleration $a(t)$ against natural frequency ω during the time interval of action of $a(t)$. The displacement and acceleration response spectra are related as $D(\omega) = \dfrac{A(\omega)}{\omega^2}$.

The maximum displacement of the i-th mode is obtained as:

$$z_{i\max} = \frac{A(\omega_i)}{\omega_i^2}\{y_i\}^T\left[\hat{M}\right]\{V\} \qquad (5)$$

which gives maximum displacements of the nodes as:

$$\{q_i\} = z_{i\max}\{y_i\} \qquad (6)$$

For evaluation of total contributions of all modes to the response spectrum, the grouping formula is used as [4]:

$$q_j = \sqrt{\sum_{k=1}^{n}\sum_{l=1}^{n}\varepsilon_{kl}\left|q_{jk}q_{jl}\right|} \qquad (7)$$

where $\varepsilon_{ij} = \begin{cases} 1, & \dfrac{\omega_k - \omega_{k-1}}{\omega_k} \le 0.1 \\ 0, & \dfrac{\omega_k - \omega_{k-1}}{\omega_k} \ge 0.1 \end{cases}$, q_j – maximum displacement in j-th d.o.f., n - number of natural frequencies.

If the spectral method is to be applied to the substructure of the overall structure, the response spectrum is recalculated from the point of the foundation to the point of attachment of the substructure. The recalculation is performed by investigating a series of finite element models obtained by "scanning" the frequency interval with different values of ω. Each model provides a single point to the response spectrum of the point of attachment of the substructure (support).

The investigation of very small vibrations of the structure requires the evaluation the flexibility of the thread joint and the guide joint, which is described by the layer of solid elements of diminished stiffness modulus that relates to the stiffness modulus of the structure as $\dfrac{E}{E_j} = 1 \div 1000$. Modal analysis of the substructure fixed at the base of the support has been performed. The analysis of first 9 modes demonstrated that at $\dfrac{E}{E_j} < 100$ the physical sense of the natural forms remains unchanged and only values of natural frequencies are influenced by additional flexibility at the joint. At $\dfrac{E}{E_j} > 100$ the natural forms exhibit substantial transformations with respect to the source structure. The relationships of natural frequencies against ratio $\dfrac{E}{E_j}$ are presented in Fig 2.

The displacements should be interpreted as maximum values, which the nodes of the structure may assume over the excitation time, however, not necessarily at the same time moments.

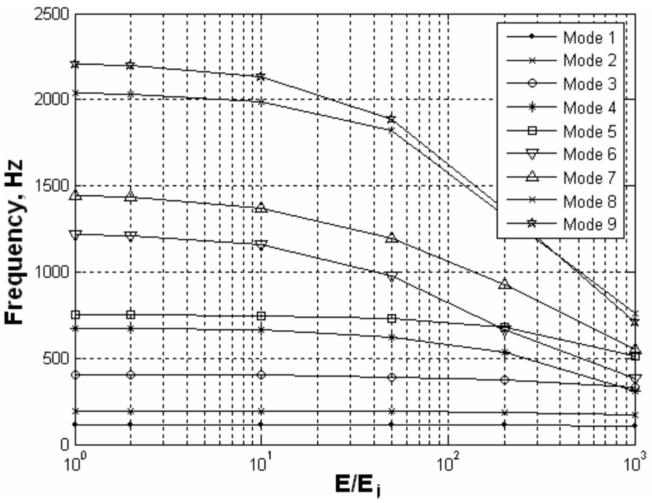


Fig. 2. The relationships of natural frequencies against ratio E/E_j

Maximum displacements are obtained at the front lens of the objective. The displacement of the front lens against the equivalent stiffness of the joints is presented in Fig. 3.

A FE model of the guided microscope carriage is presented in Fig. 4. The vertical position of the carriage is defined by aerostatic air bearings that are double-sided. Therefore the coefficient values of vertical elements COMBIN14 were doubled.

This element has been used to describe the junction between the guiding and guided carriages.

The stiffness of plate was calculated using geometrical parameters of the element. The bulk model of the microscope was connected to the guided carriage employing elements CONTA173. The optical system placed on the carriage is not shown in the FE model, but the mass of the system is estimated respectively.

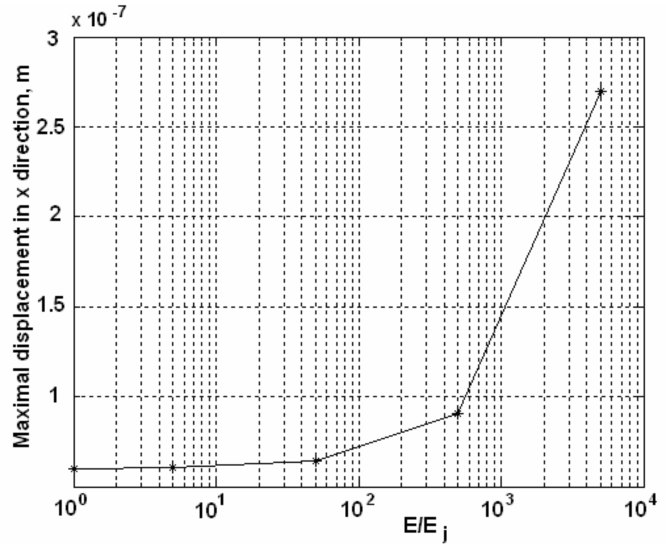


Fig. 3. The relationship of the maximum displacement of the front lens against the stiffness of the joint

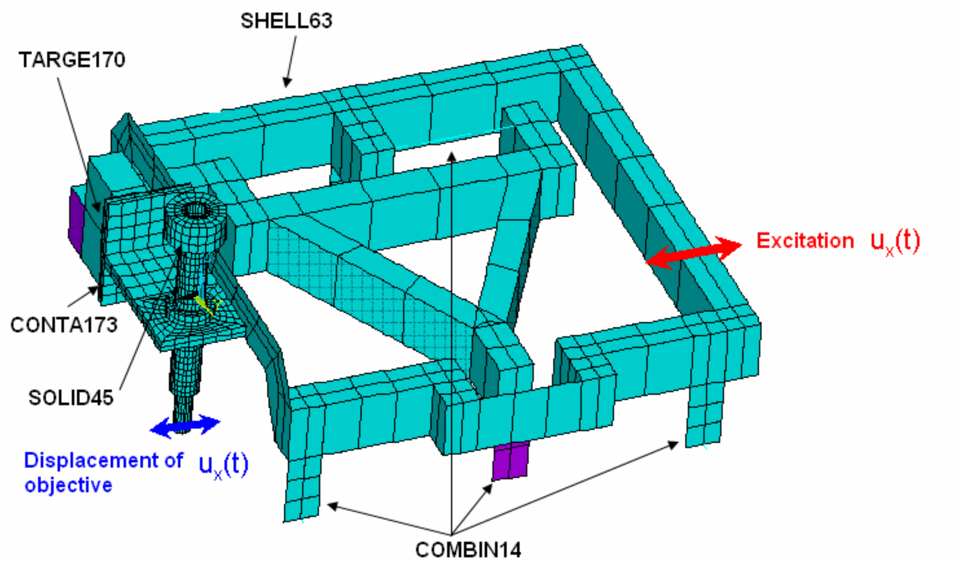


Fig. 4. FE model of the microscope carriage

Structural solid elements SHELL63 and SOLID45 used for meshing of the system components were connected applying multipoint constraint MPC method implemented in ANSYS. Thereto elements of the contact layer TARGE170 are employed to cover the surface of elements SOLID45 in the contact zone. Elements CONTA173 were used to coat the elements SHELL63. This method is a typical technique to define the mechanical contact in ANSYS where a specific key is used to secure the contacting surfaces. Thus the form of FE model equations remains linear.

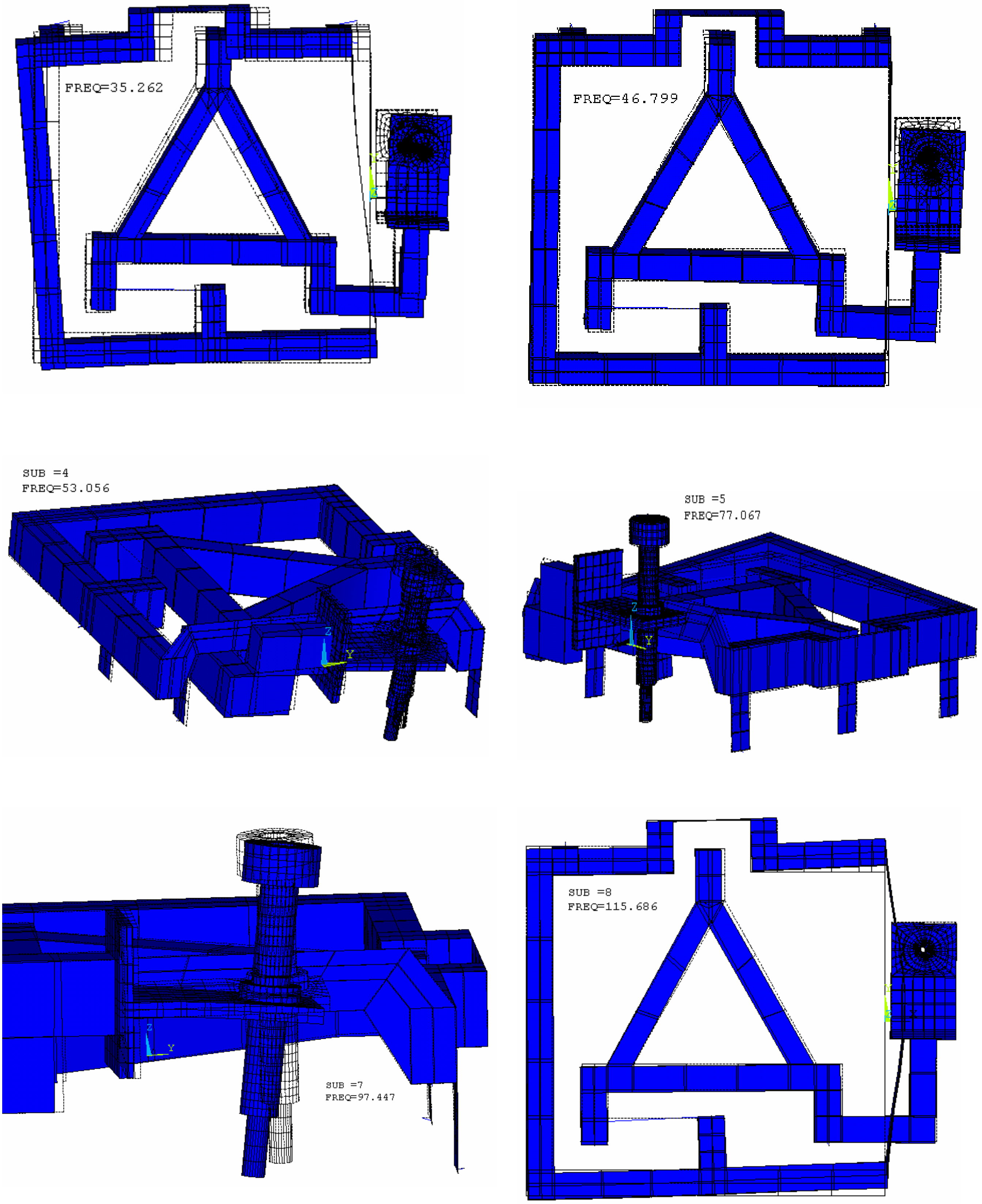


Fig. 5. The lower modes of vibration of the carriage and microscope (eigenshapes and eigenfrequencies, Hz)

The lower modes of vibration of the carriage and microscope are depicted in Fig. 5.

Subsequently the model has been used for the analysis of transitional vibrations of the microscope carriage. Applying forward-back motion to the support point of the guiding carriage the displacements of the microscope structure has been analyzed. The shortest time period, which is assumed as time constant of the microscope carriage, ascertained when the microscope traces the movements provided by the drive.

The analysis results of FE modeling have demonstrated that maximum displacements are obtained at the apex of the microscope lens and due to seismic excitations can amount to >100 nm, [5].

Experimental investigations

For precise and full-scale evaluation of the application of dynamic mode of the line scale calibration a CCD-microscope-based edge detection system has been developed. The system uses a modular focusing unit to capture image data and incorporates a set of high-quality objectives and image-processing software.

The comparator presented in Fig. 6 consists of four main parts, namely the body of the machine, a laser interferometer, a translating system and a detecting apparatus. The body of the machine, which is made of granite surface plate, is used as the base of the machine and as a guide for the moving carriage. Measurement of the displacement of the carriage is realized by laser interferometer that consist of Zygo ZMI 2000 laser head and interferometer with the single-pass arrangement. The interferometer provides a resolution of 0,62 nm. The measuring system has a configuration designed to avoid the Abbe offset between the measuring line of a linear artifact and the axis of the laser beam. The linear artifact is vertically at the same height as the two laser beams and horizontally at the centre of the two laser beams, so that it can be positioned coaxially with the centre of the two laser beams.

Line scales are calibrated by making carriage displacement measured by the interferometer, correspond exactly to scale interval lengths. The translating system, which carries interferometer mirror, angular mirror and CCD microscope for detecting the measuring points, causes relative displacements between the linear artifact and the detection system. The latter consists of a guiding carriage, guided frame, and NIKON modular CCD microscope fixed to the frame. The friction gear drives the guiding stage, which incorporates air bearings for smooth movement. It has a traverse length of 3500 mm.

The measured performances confirm that investigated measurement system can operate reliable at velocities up to 6 mm/s without appreciable loss in measurement accuracy. In order to examine the calibration process in real time, the experiments of the line scale calibration with a moving CCD microscope have been carried out in specific operating modes, and the accuracy of dynamic calibration has been studied.

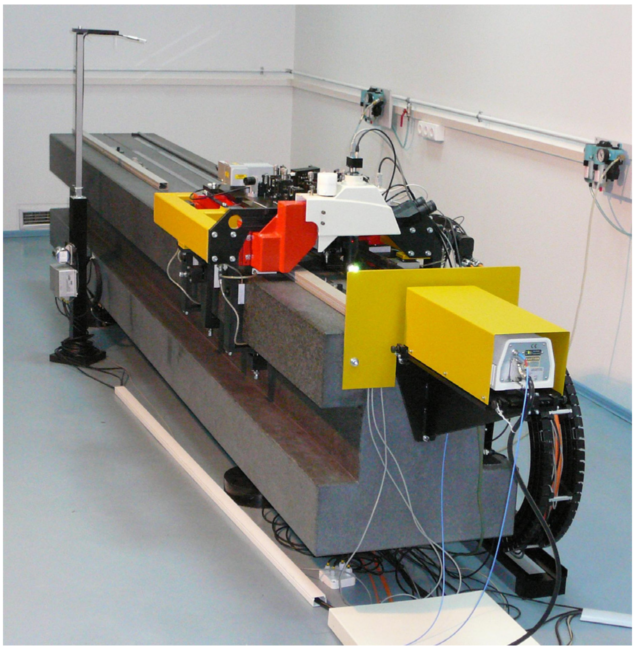


Fig. 6. Precision interferometer-controlled line scale comparator with CCD microscope

Vibration measurement

The stand of the comparator is swayed by external excitations transmitted through the laboratory foundation and supports as well as excitation of the carriage drive. Though the system is placed in the metrological laboratory it is influenced by vibrations from the technological environment: mechanisms of ventilation, machine tools operating in the next-door compartments.

The oscillations that originate due to vibration of the foundation of the precise length comparator have been analyzed in order to ensure calibration accuracy of the comparator in dynamic mode of operation.

Vibration measurement system with accelerometer 8306, vibrometer 2511 and band filter has been used for measuring vibration parameters. The accelerometer was attached to the orientation block. The later was fixed to a corresponding location on the comparator or attached to a solid measurement plate and placed on the ground. Orientation block allowed changing the measurement direction of the accelerometer.

The ascendant amplitudes of horizontal and vertical vibrations in the low frequency range (2-10 Hz) and higher frequency range (100-300 Hz) have been analyzed. Vibrations of granite table, supports of microscope and laser interferometer were additionally measured under excitations caused by the comparator drive, [7].

Typical measurement results of displacement of the comparator foundation 7-9. Conducted measurements have demonstrated that vibrations have random character and correspond to normal distribution. Amplitudes of the time signal vary from 0,4 to 2,5 µm, and amplitudes of vibration spectra fluctuate in the range of 0,04-0,9 µm at frequency 2 Hz.

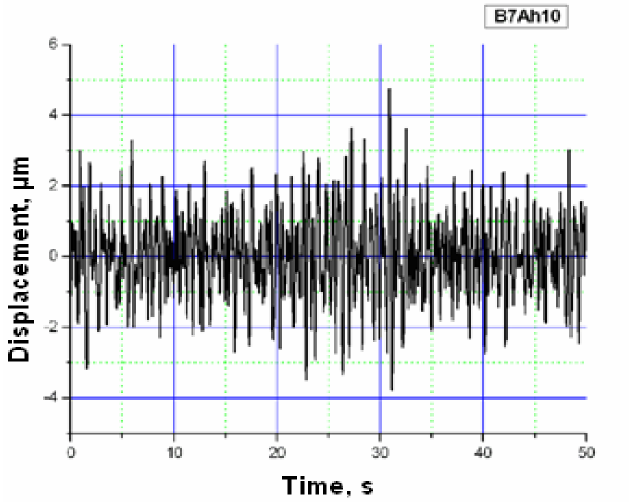


Fig. 7. Horizontal displacement signal of the comparator foundation

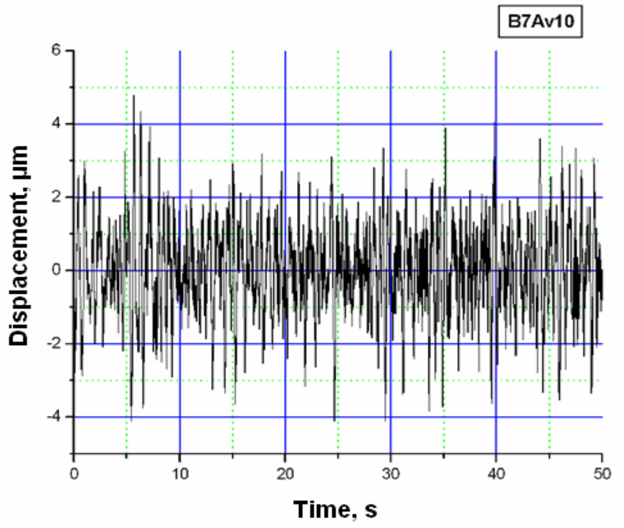
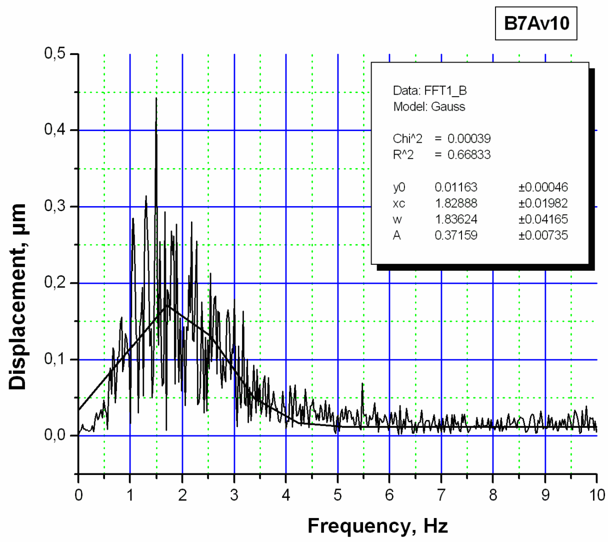


Fig. 8. Vertical displacement signal of the comparator foundation



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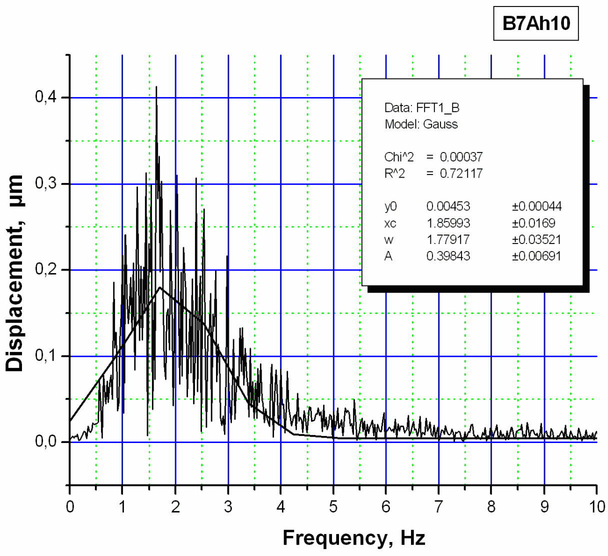

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Fig. 9. Spectrum of the displacement signal approximated by Gauss function: *a*) vertical direction; *b*) horizontal direction

Vibrations of the line detection system

A line detection system is the most sophisticated part of any length calibration system that defines metrological characteristics, efficiency and costs of a comparator. The specific uncertainty depends on the instrument, but the sources fall into a few broad categories: reference surface geometry, alignment and motion errors. Considering line detection errors, one can assume that apart from the type of a microscope and its technical characteristics precise adjustment of microscope optical axis perpendicular to the scale and focusing of a microscope will affect calibration accuracy significantly. It has been calculated that relatively small angular misalignments of the microscope optical axis and vertical vibrations of the microscope increases the error of line centre detection significantly when microscope is out of focus, [8, 9]. The vibrations of the microscope in the direction of carriage motion may cause detection errors in determining positions of graduation lines. Therefore, displacements of the structure as a result of dynamic excitations of the foundation of the comparator need to be known. They have been measured employing technique described above.

Measurements were carried out in both static and dynamic modes in horizontal and vertical directions by placing the accelerometer on the microscope carriage. The main noise sources here are vibrations transmitted through the comparator foundation and excitations induced by the microscope carriage drive. The carriage velocity sweep was from 0 to 4 mm/s that correspond to calibration speed in dynamic mode. Furthermore air supply to the comparator frame supports and aerostatic bearings of the carriage has been controlled.

The experimental results have shown that the amplitudes of vibration velocity are 0,01 – 0,04 mm/s (without air supplying) and decrease to 0,005 – 0,015 mm/s while air supply is on. The vibration velocities during dynamic calibration fluctuate from 0,2 mm/s to 0,4 mm/s. Measurement results has revealed that the maximum vibration displacements occur at frequencies 2, 4 and 10 Hz and may reach 0,045 µm; in the frequency range of 9-11 Hz these amplitudes fluctuate up to 0,025 µm .

Analysis of statistical parameters of the vibration velocity measurement signal has shown that standard deviation of the signal was 0,01224 mm/s when the microscope carriage was not moving and air supply was switched off. Standard deviation for measurement sets with air supply was reduced significantly down to 0,00549 mm/s in static mode; it considerably increased up to 0,16196 mm/s when the microscope carriage was moving at speed 3 mm/s.

Comparison of statistical results has confirmed that air-controlled frame supports and air bearings of the microscope carriage partly deaden vibrations transmitted through the foundation, but is not capable to absorb vibration components resultant due to excitations originated from the friction drive in dynamic mode.

Since the measurement mirror of the interferometer is firmly attached to the moving microscope structure any slight deviation of the measurement reflector will introduce measurement errors.

Lateral deviations of measurement mirror due to vertical vibrations will not affect the calibration accuracy significantly since they cause an offset between the measurement and reference beams at the receiver and could reduce the AC signal of the interferometer only by several percents that have no influence on measurement accuracy.

Horizontal vibrations of the measurement mirror depicted in Fig.10 will have direct influence to the measured position of the microscope. Measurement results obtained when the line detection system was not moving have shown that position deviation due to horizontal vibrations can amount up to 40 nm.

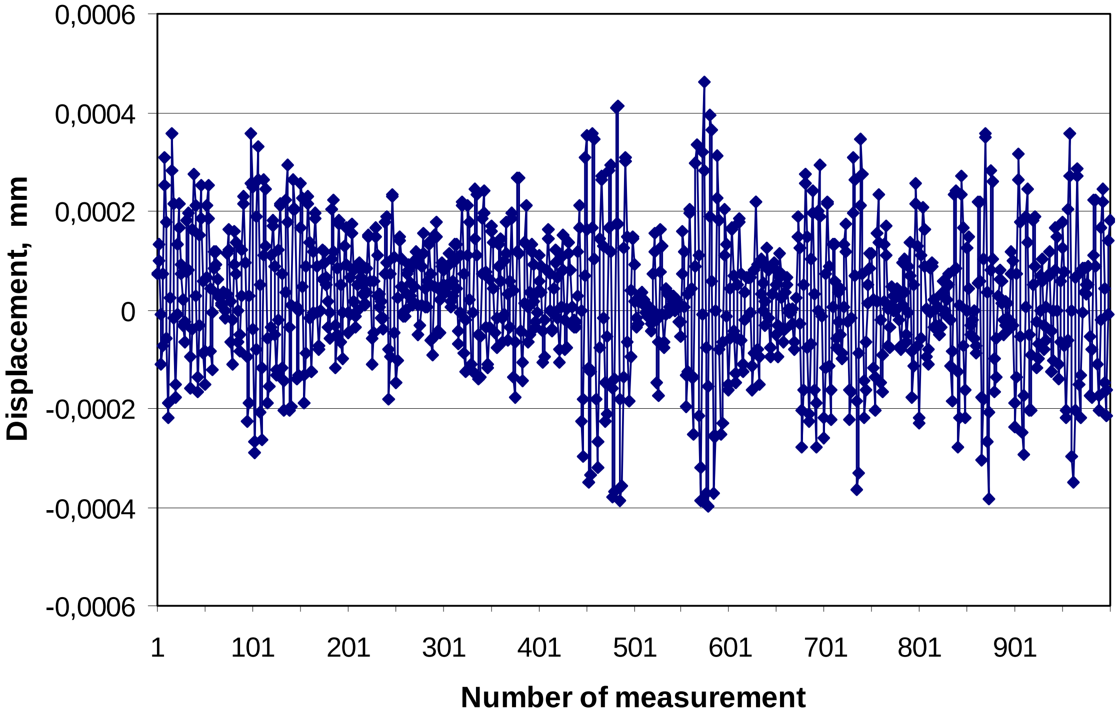


Fig. 10. Vibrations of the interferometer measurement mirror due to seismic excitations when microscope carriage is not moving

Repeatability measurements in dynamic mode have been carried out at different positions relative to the laser. Measurement results obtained at the microscope carriage speed 3mm/s and 6mm/s are depicted respectively in Fig. 11 and 12.

The initial measurement point was located 500 mm away from the laser. The overall measurement distance was 200 mm with an increment of 25 mm. The number of runs was set to 10 and all the data were collected when the microscope carriage was moving.

The analysis of experimental results revealed a slight increase of line detection error at higher speed. Line position deviation at measurement speed of 3 mm/s amounted to 0,55 µm and 0,7 µm at 6 mm/s respectively. The measured performance of the system has confirmed that dynamic calibration error originating due to vibration sources in the structure must be considered and implicated in the uncertainty budget.

The concept of designing a precision machine tool or measuring machine with displacement measuring system in line with the displacement to be measured is one of the main principles in dimensional metrology. An instrument has a moving probe, and motion in any single direction has six degrees of freedom and thus six different error components. The graduation error is the error in the direction of motion, X. The straightness errors are the motions perpendicular to the X axis. The angular errors are originated by rotations about the X axis (roll) as well as rotations (pitch and yaw) about the axis Y and Z respectively that are perpendicular to the axis of motion. If the scale is not exactly aligned to the measurement axis the angle errors produce measurement errors known as Abbe offset errors.

From economic point of view it is reasonable to secure the precise linear motion by means of presicion manufacturing when linear errors in X and Z directions do not exceed 1-2 μm, and angular deviations amount up to several arc sec. Further error compensation can be carried out by employing software error correction.

To minimize the Abbe error the anglular variation of the measurement reflector is measured and the correction for the Abbe offset is performed numerically since the offset distance is known. The structural components for Abbe error correction are shown in Fig. 13a, and measurement results of the angular deviations of the measurement mirror are depicted in Fig. 13 b. Repeated measurements revealed a good repeatability of angular error along the travelling range of the microscope carriage. The angular deviation ranged about 0,5 arc sec for the travel of 3250 mm. Fluctuations induced by elastic vibrations of the structure and other sources amount to 0,07 arc sec.

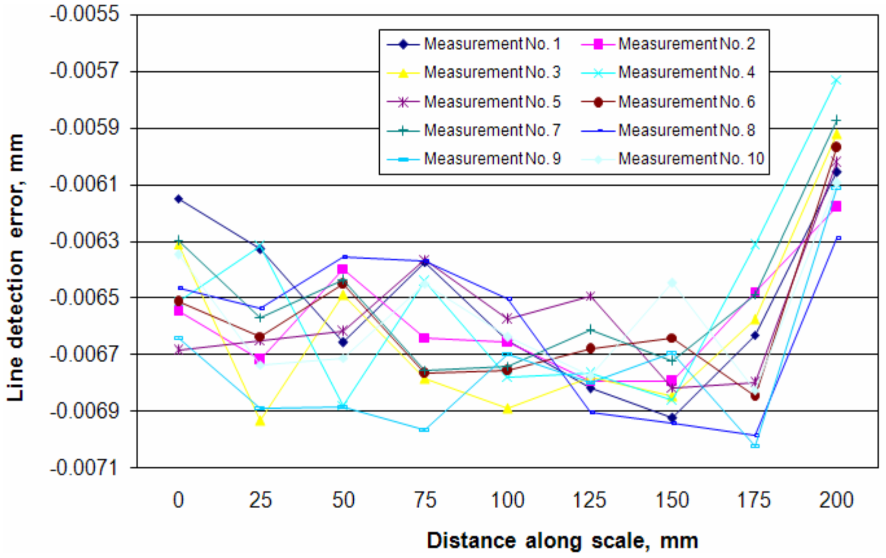


Fig. 11. Repeatability of line scale measurements at calibration speed 3 mm/s

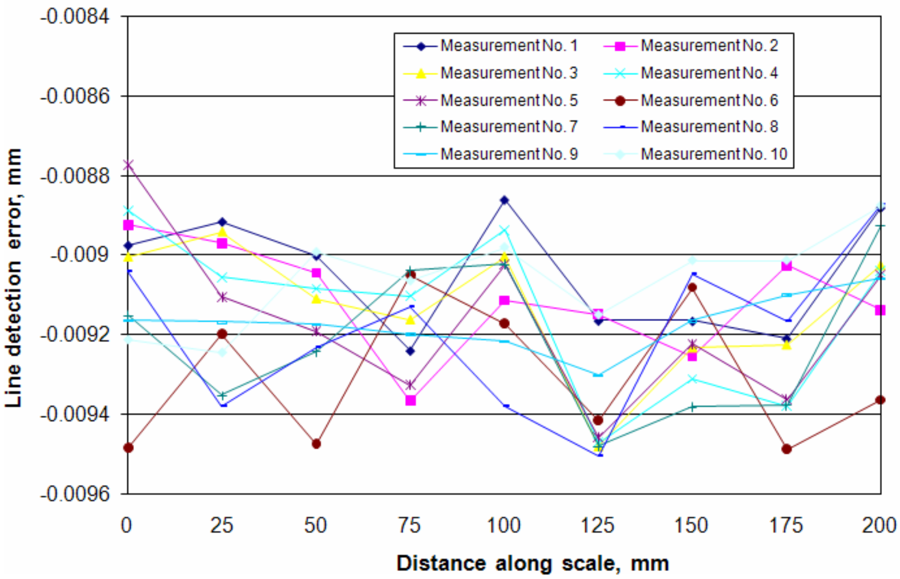
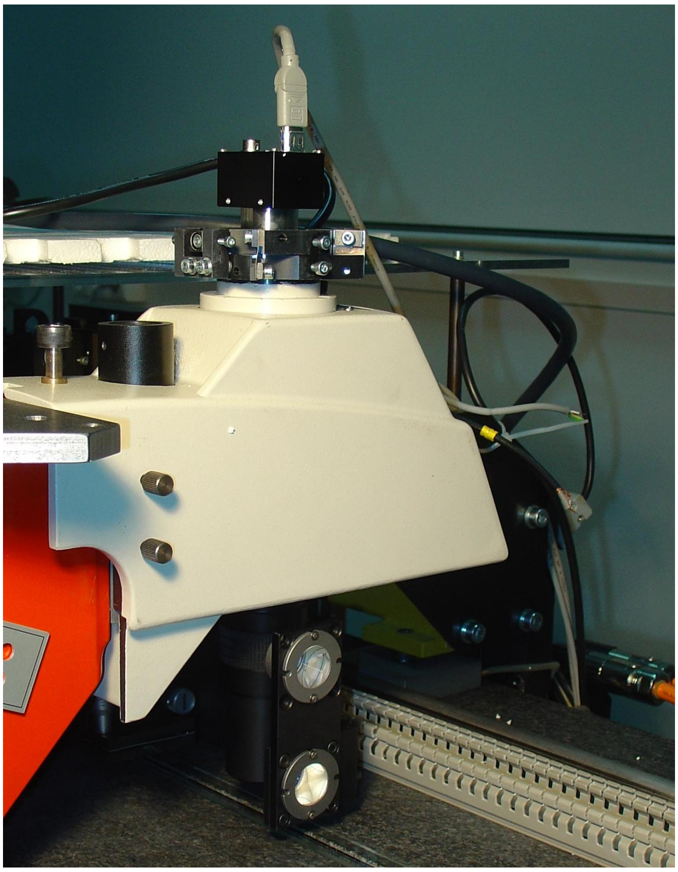


Fig. 12. Repeatability of the line scale measurements at calibration speed 6 mm/s



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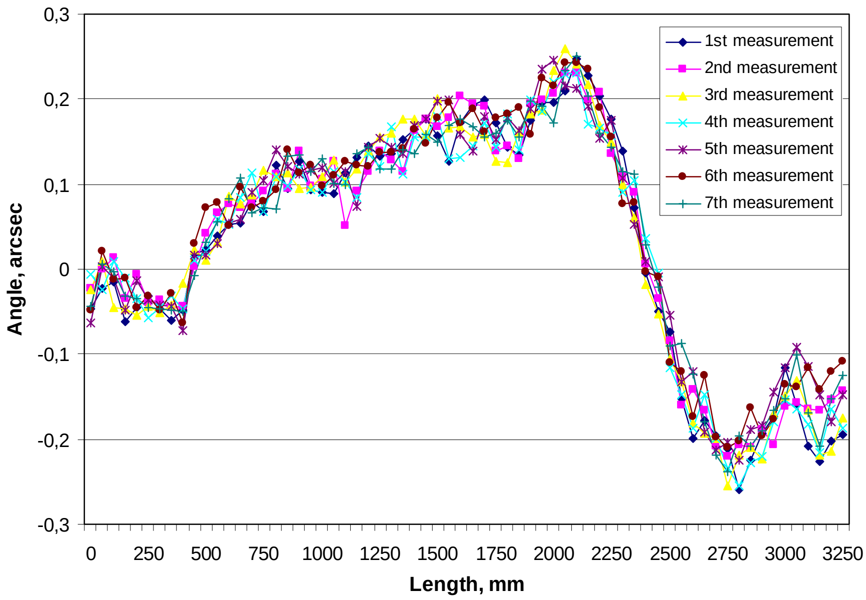

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Fig. 13. *a*) Structural components for correction of Abbe error ; *b*) interferometer signal for Abbe error correction

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Conclusions

The need for increasing the throughput of the line scale manufacturing drives the demand for embedding the traceable length metrology directly into technological processes in real industrial environment and introducing the dynamic regime of calibration. Small mechanical vibrations determined by both seismic excitations and noise caused by operation of the comparator components are important source of measurement deviations in dynamic calibration, which affect the calibration accuracy. To give some measure of confidence to the measured value, calibration errors caused by these vibrations must be identified, and their probable effect on the result estimated.

Error-related problems specific to a precision line scale calibration in dynamic regime were investigated in this paper. Advanced FE modeling techniques were applied in order to represent the structural behavior of the comparator. A new FE model has been developed and influences of dynamic processes in the length comparator structure and possible variations of geometrical dimensions of the line scale and the microscope that lead to the increase of the line scale calibration uncertainty has been evaluated. The modeling tools developed enable us to represent the structural behavior of the calibration system and ensure a high level of the model adequacy to the reality. Modeling of seismic excitations in the comparator structure has shown that maximum displacements are expected at the bottom plane of the microscope objective and can amount to more than 100 nm. The modeling and experiments conducted have revealed that the flexibility of the thread joint and the guide joint can cause the increase of the dynamical calibration errors.

Horizontal vibrations of the measurement mirror will have direct impact to the detected position of the microscope. Measurement results obtained have demonstrated that position deviation due to horizontal vibrations of the microscope can amount up to 40 nm. Deviations of the comparator components induced by dynamic processes can also significantly decrease the line detection accuracy due to defocusing. These errors can be reduced by proper alignment of the scanning system and the line scale as well as by the use of error compensating algorithms.

The dynamic calibration error originating due to vibration sources in the structure must be considered and implicated in the uncertainty budget. The analysis performed allows us to evaluate these errors and keep them in control to an adequate level required by the specification for the system.

The measured performances confirm that investigated measurement system can operate reliable at calibration velocities up to 6 mm/s without appreciable loss of measurement accuracy.

The calibration uncertainty can be further reduced by optimizing the comparator structure in compliance with the precision engineering principles and applying the error compensation techniques.

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