Technical Information

Accuracy of Feed Axes

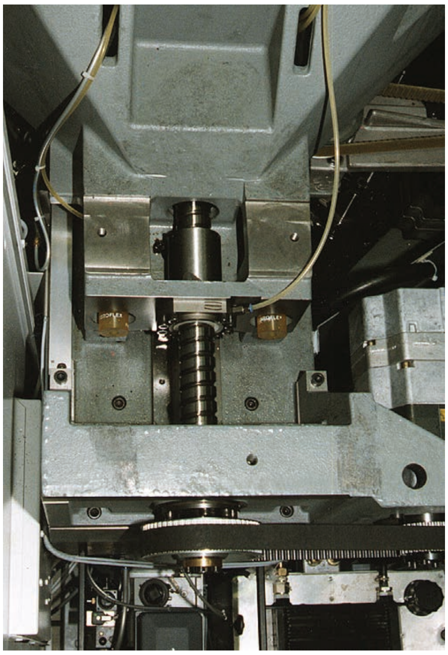
Every year's new machine tools show improvements in efficiency and power. Increasing feed rates and acceleration values reduce machining times. At the same time, increasing accuracy permits ever-closer tolerances. On the one hand, these developments enable the machining of increasingly critical parts; on the other hand they simplify the manufacture of complex assemblies. Selective manual assembly and bench work can often be reduced. Also, higher accuracy of parts normally results in improved function of component assemblies. Dimensional accuracy in motor transmissions, for example, increases service life and reduces noise emission.

In the total error budget of a machine tool the positioning error values of the feed axes play a critical role. The following text discusses these errors and compares them with other types of error. The primary problem involved with position measurement using rotary encoder and ball screw is the thermal expansion of the ball screw. The resulting positioning error often outweighs the thermally induced structural deformation and geometric error of machining centers. Several tests show the influence of the heating of the ball screw on the results of machining – also with respect to the ball-screw bearing. When a linear encoder is used for position measurement, the thermal expansion of the ball screw has no influence and the position drift is negligible. As requirements for machine tool accuracy and velocity increase, the role of linear encoders for position measurement grows increasingly important.

The accuracy of modern machine tools is measured with an increasing number of new and revised inspection and acceptance tests. Where years ago purely geometric acceptance tests predominated, today's routine methods include dynamic tests such as circular interpolation and free-form tests, thermal tests such as described in ISO/DIS 230-3, and for production machines, capability testing during acceptance or regular inspection. The influences of the cutting processes, the geometric and thermal accuracy of the machine, its static and dynamic rigidity, and the positioning response of the feed axes on the attainable accuracy of the workpiece can be more specifically analyzed. Machine errors therefore become increasingly transparent to the user.

Considering the increasing frequency of changing jobs and the concomitant reduction in batch sizes, reducing the thermal or systematic error of a machine tool through tedious optimization of individual production steps is seldom feasible. *"Accuracy of the first part"* is gaining in importance. In particular the thermal error of machine tools is drawing ever more interest.

The following text shows that thermal error can be quite significant, especially for the feed axes. Unlike structural deformation, errors of the feed axes can be dramatically reduced through a choice of simple and readily available measuring devices.



Feed drive mechanism of a milling machining center

03/2006

Feed-drive system design

An exact error analysis of position measurement via rotary encoder and feed screw begins with a consideration of prevalent mechanical feed-drive systems. Although machine tool designs vary immensely, the mechanical configuration of their feed drive is largely standardized (Fig. 1). In almost all cases, the recirculating ball screw has established itself as the solution for converting the rotary motion of the servo motor into linear slide motion. Its bearing takes up all axial forces of the slide. The servo motor and ball screw drive are usually directly coupled. Toothed-belt drives are also widely used to achieve a compact design and better adapt the speed.

For position measurement of feed axes on NC machine tools it is possible to use either linear encoders or recirculating ball screws in conjunction with rotary encoders. A position control loop via rotary encoder and ball screw includes only the servo motor (Fig. 1 dashed line). In other words, there is no actual position control of the slide, because only the position of the servomotor rotor is being controlled. To be able to extrapolate the slide position, the mechanical system between the servo motor and the slide must have a known and, above all, reproducible mechanical transfer behavior. A position control loop with a linear encoder, on the other hand, includes the entire mechanical feed-drive system. Transfer errors from the mechanics are detected by the linear encoder on the slide, and are corrected by the controller electronics.

Differing terminology
Different terms are used to distinguish between these two methods of position control. German-speaking and some English-speaking communities generally refer to them somewhat inaccurately as "direct and indirect measurement." However, these terms are rather poorly chosen because, strictly speaking, both methods are direct. One method uses the line grating on the linear scale as the measuring standard, the other the pitch of the ball screw. The rotary encoder simply serves as an interpolating aid. Here the Japanese concepts of "Semi-Closed-Loop and Closed-Loop control" seem appropriate, since they more aptly describe the actual problem.

Trend toward digitally driven axes
As a result of the trend toward digital axes in drive technology, a large share of new servo motors feature rotary encoders, which in principle can serve together with the feed screw for position control. With such a drive configuration the decision must be made as to whether to add a linear encoder or simply to use a ball screw working in combination with the already existing motor encoder.

One should remember to consider the problems discussed in the following text regarding position measurement using a rotary encoder/ball-screw system. They can quickly increase the cost of an "economical" machine if the owner finds that the accuracy does not suffice in certain applications.

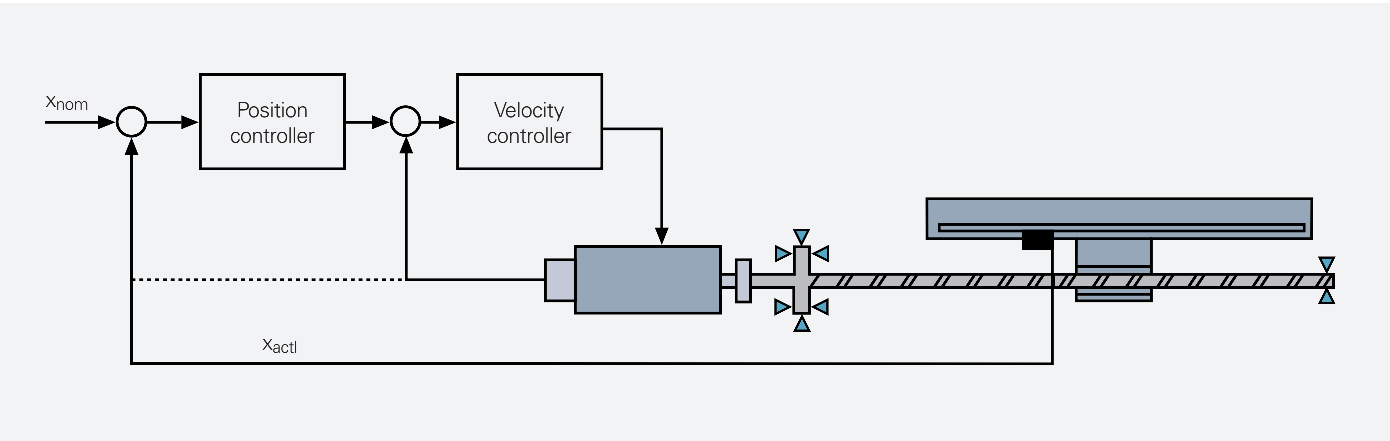


Fig. 1: Typical drive system of a numerically controlled machine tool with linear scale on the slide and rotary encoder on the motor. Unlike in position feedback control with rotary encoder and ball screw, a linear encoder includes the feed drive mechanism in the control loop.

Positioning errors caused by mechanical influences

Kinematics error

Kinematics error that can be directly attributed to position measurement using feed screw and rotary encoder result from ball-screw pitch error, from play in the feed elements, and from pitch loss. Ball-screw pitch errors directly influence the result of measurement because the pitch of the ball screw is used as a standard for linear measurement. Play in the feed transfer elements causes backlash. The pitch loss results from a shift of the balls during the positioning of ball-screw drives with two-point preloading and can lead to reversal error in the order of 1 µm to 10 µm [see Schröder, Wilhelm: Feinpositionierung mit Kugelgewindetrieben].

Error compensation

Most controls are capable of compensating such pitch error and reversal error. However, to determine the compensation values it is necessary to make elaborate measurements with comparative measuring devices such as interferometers and grid encoders. In addition, the reversal error is often unstable over long periods of time and must be regularly recalibrated (Fig. 2).

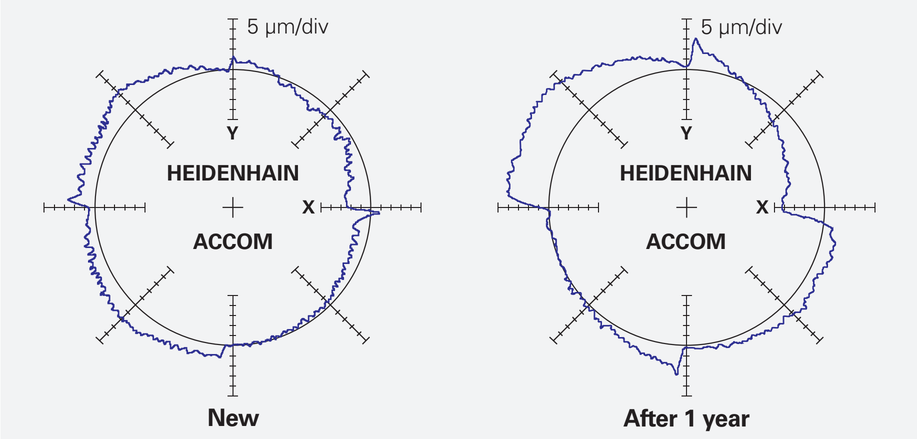


Fig. 2: Circular tests of a machining center without linear encoders in new condition and after one year. The reversal error of the X axis has increased significantly.

Strain in drive mechanisms

Forces leading to the deformation of feed drive mechanisms cause a shift in the actual axis slide position relative to the position measured with the ball screw and rotary encoder. They are essentially inertia forces resulting from acceleration of the slide, cutting process forces, and friction in the guideways. The mean axial rigidity of a feed drive mechanism as shown in Fig. 1 lies in the range of 100 N/µm to 200 N/µm (with a distance between ball nut and fixed bearing of 0.5 m and a ball-screw diameter of 40 mm).

Cutting force

The cutting force can quite possibly lie in the kN range, but its effect is distributed not only in the feed drive system, but also over the entire structure of the machine and therefore between the workpiece and the tool. The deformation of the feed drive system therefore normally has only a small share in the total deformation of the machine. A linear encoder can recognize and correct only this small portion of the total deformation. Critical component dimensions, however, are normally finished at low feed rates with correspondingly low deformation of the feed drive system.

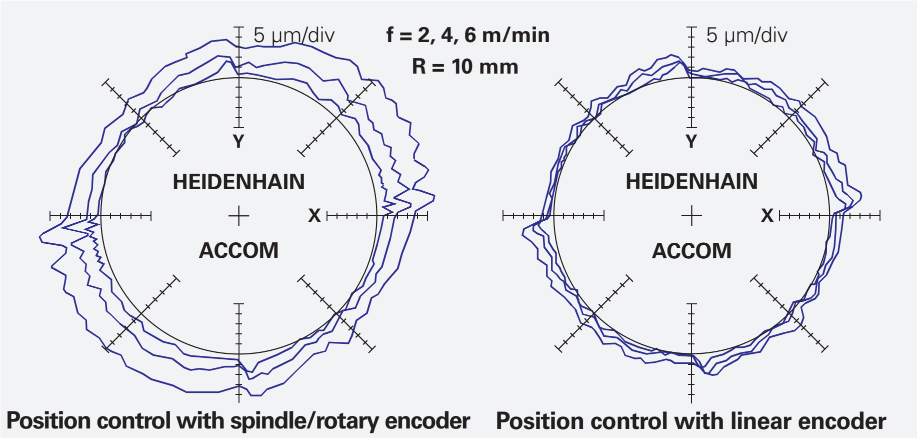


Fig. 3: Circular tests of a machining center that has been retrofitted with linear encoders. With position feedback control by rotary encoder and ball screw, the circles deviate significantly from the ideal path at high feed rates. With linear encoders, the contour accuracy is considerably better.

Force of acceleration

A typical slide mass of 500 kg and a moderate acceleration of 4 m/s^2 result in deformations of 10 µm to 20 µm that cannot be recognized by the rotary encoder/ball-screw system. The present industry trend toward accelerations in significantly higher ranges will result in increasingly large deformation values.

Force of friction

The force of friction in the guideways lies between 1 % and 2 % of mass for roller guideways and 3 % to 12 % of mass for sliding guideways [see VDW-Bericht 0153: Untersuchung von Wälzführungen zur Verbesserung des statischen und dynamischen Verhaltens von Werkzeugmaschinen]. A mass exerting 5000 N therefore results in feed drive deformation of only 0.25 µm to 6 µm.

Circular interpolation test for inspecting machine tools

A typical example for errors dependent on acceleration and velocity can be recorded in a circular interpolation test on a vertical machining center (Fig. 3). With position feedback control by rotary encoder and ball screw, the circles deviate significantly from the ideal path at high contour speeds. The same machining center shows significantly better contour accuracy when equipped with linear encoders.

Positioning error due to rising ball-screw temperature

Positioning error resulting from thermal expansion of the ball screw presents the greatest problem for position measurement via rotary encoder and ball screw. This is because the ball-screw drive must serve a double function: On the one hand it must be as rigid as possible to convert the rotary motion of the servo motor to linear feed motion. On the other hand it must serve as a precision measuring standard. The two-fold function therefore forces a compromise because both the rigidity and the thermal expansion depend on the preloading of the ball nut and the fixed bearing. Both the axial rigidity and the frictional moment are roughly proportional to the preloading.

Friction in the ball nut

The largest portion of the friction in a feed drive system is generated in the ball nut. This is because of the complex kinematics of a recirculating ball nut. Although at first glance the balls may seem only to be rolling, they are in fact subjected to a great deal of friction. Besides the microslip resulting from relative motion in the compressed contact areas, the greatest effect is from the macroslip due to kinematics exigencies. The balls are not completely held in the races and wobble much like tennis balls rolling down a gutter. The result is a continual pressing and pushing with occasional slipping of the balls. The friction among the balls is aggravated by high surface pressure due to the absence of a retaining device to separate them. As in every angular-contact ball bearing a spinning friction results from a contact diameter that is not orthogonal to the axis of ball rotation. Each ball therefore rotates about its contact diameter. Studies have also shown that the balls can move in the thread only because of an additional slip component brought on by the thread pitch [see Weule, Hartmut /Rosum, Jens: Optimization of the friction behaviour of ball screw drives through WC/C coated roller bodies].

The recirculation system is a special problem zone for ball screws. With every entrance into the recirculation channel, just as with every exit, the movement of the ball changes entirely. The rotational energy of the balls, which in rapid traverse typically rotate with 8000 rpm, must be respectively started and stopped. In contrast to the preloaded thread zone, in the recirculation zone the balls are not under stress. The play of energy causes the balls to collect in the recirculation channel. Without elaborate measures to reintroduce the balls into the thread at the end of the channel it tends to congest, causing the familiar jamming of the ball-screw drive.

The frictional moment of a ground precision recirculating ball screw with 40 mm diameter and 10 mm pitch was measured by Golz [see Golz, Hans Ulrich: Analyse, Modellbildung und Optimierung des Betriebsverhaltens von Kugelgewindetrieben] for various preload forces and rotational speeds (Fig. 4). The Stribeck characteristic of frictional moment is clearly recognizable. It confirms the high share of solid-body friction and mixed friction in ball-screw drives at low speeds. Viscous friction dominates at high speeds. It is interesting to note that for this typical ball screw the normal machining feed rates lie far below the speeds at which the frictional moment is at its minimum. The rapid traverse feed rates, however, lie far above it. The feed rates at which this ball screw is at optimum efficiency therefore seldom occur. The frictional moment is only slightly dependent on the axis load of the ball nut [see ibid.].

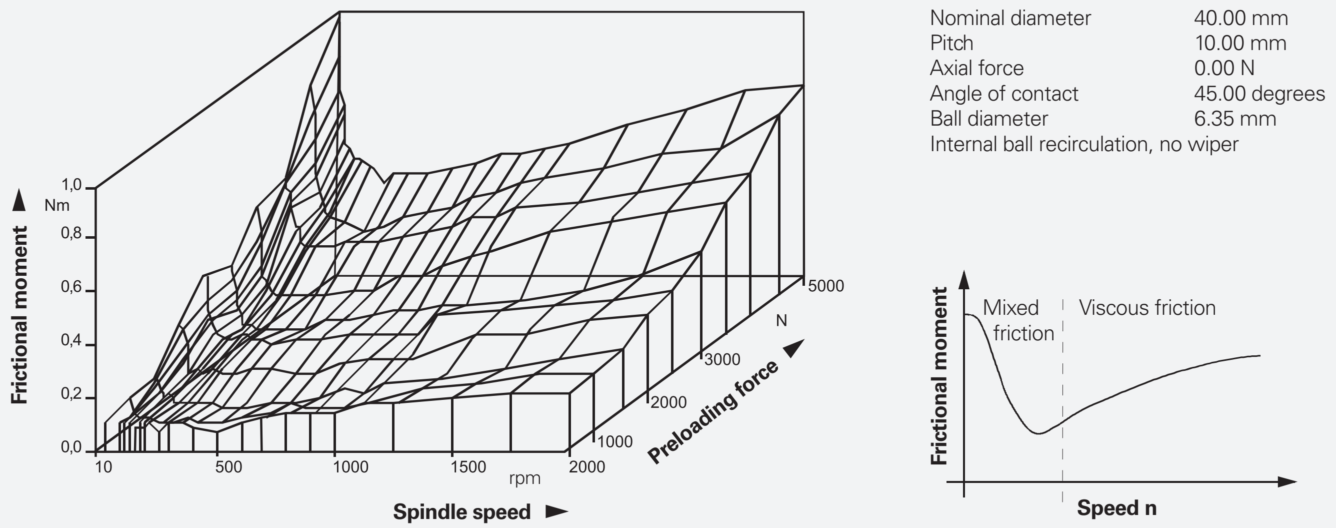


Fig. 4: Measured moment of friction of a two-point preloaded ball screw [see ibid.]. The Stribeck characteristic is plainly visible.

Frictional heat generated in the ball nut
With a typical preload of 3 kN and allowing for the missing wiper, this results in a no-load or frictional moment of 0.5 Nm to 1 Nm. This means that in rapid traverse at a ball screw speed of 2000 rpm approximately 100 W to 200 W of frictional heat is generated in the ball nut.

More frictional heat to be expected
To increase rapid traverse velocity, either the pitch or the rotational speed of the recirculating ball screw must increase. In the last five years the maximum permissible speed of recirculating ball screws has doubled. Due to the continually increasing requirement for acceleration, the preloading and therefore the friction of the ball nut could not be reduced. Recirculating ball-screw drives therefore generate significantly more heat than before and will generate even more in the future.

Measurement of positioning accuracy according to ISO 230-3
The influence of frictional heat on the positioning response of the feed axis becomes apparent when positioning tests are conducted in accordance with the international standard ISO/DIS 230-3. This standard contains proposals for making uniform measurements of thermal shifts of lathes and milling machines as a result of external and internal heat sources (Fig. 5).

Deformation in the machine structure resulting from changes in ambient conditions or through heat generation in the spindle drive are recorded with the aid of five probes that measure against a cylinder mounted in the tool holder. This makes it possible to measure all five relevant degrees of freedom. To test the feed axes, it proposes a repeated positioning to two points that lie as near as possible to the ends of the traverse range at an agreed percentage of the rapid traverse velocity. The change of the positions with respect to the initial value is recorded. The test is to be conducted until a satiation effect is clearly observable. Simpler test equipment than a laser interferometer, such as dial gauges, can also be used for the axis test. These tests enable any workshop to conduct such inspections at a reasonable cost.

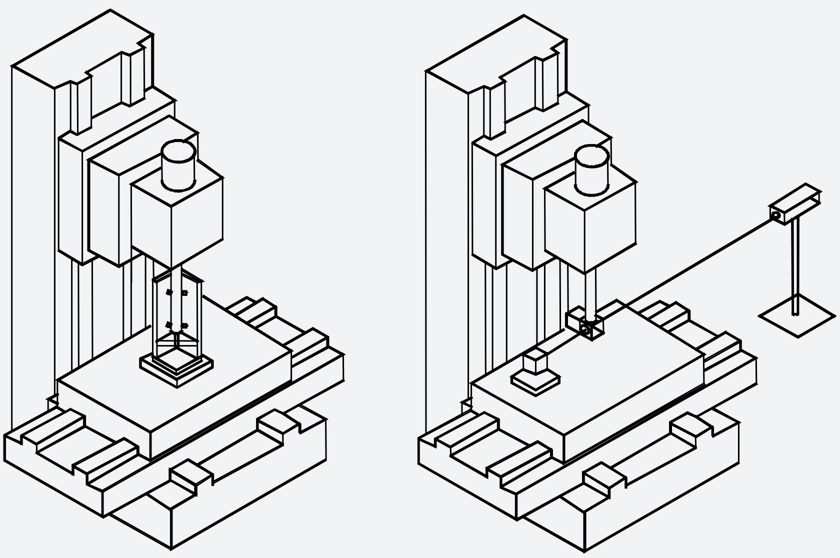


Fig. 5: Measurement of thermal displacement of a machining center in accordance with ISO/DIS 230-3.

Influence of the ball-screw bearing on positioning accuracy

Differing types of behavior can be expected depending on whether the ball screw can expand freely. The various types of bearings for recirculating ball screws are illustrated in Fig. 6.

Fixed bearing at one end
In the case of fixed/floating bearings, the ball screw will expand freely away from the fixed bearing in accordance with its temperature profile. The thermal zero point of such a feed axis is at the fixed bearing. This means that theoretically no thermal shift would be found if the ball nut is located at the fixed bearing. All other positions are affected by the thermal expansion of the ball screw.

Figure 7 shows the result of a positioning test as per ISO/DIS 230-3 on a vertical machining center (built in 1998) retrofitted with linear encoders. The X axis was positioned to three points a total of 100 times at 10 m/min. Taking the standstill periods for measured value acquisition into account, the mean traversing speed during the test was approx. 4 m/min. In addition to the two positions at the ends of traverse as recommended in the standard, a third position at the midpoint of traverse was measured. Figure 7 shows the position values with respect to their initial values. At first the ball screw / rotary encoder system was used for position feedback control. In a second test under otherwise identical conditions, linear encoders were used. The comparator system was a VM 101 from HEIDENHAIN.

In spite of the moderate feet rate of 10 m/min (rapid traverse 24 m/min), the position farthest from the fixed bearing of the ball screw shifted by more than 110 µm within 40 minutes. It is interesting to note that the drift increases very quickly immediately after switch-on. Any change in the mean feed rate in a production process therefore immediately affects positioning accuracy. Similar results were published by Schmitt [see Schmitt, Thomas: Modell der Wärmeübertragungsvorgänge in der mechanischen Struktur von CNC-gesteuerten Vorschubsystemen].

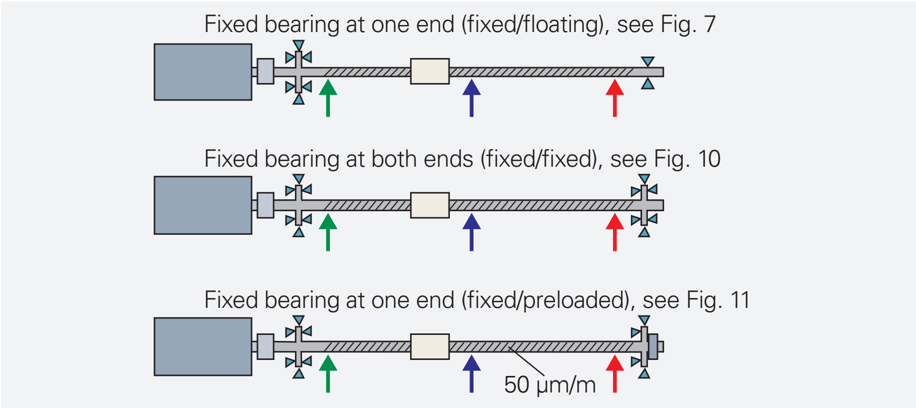


Fig. 6: The various types of bearings for recirculating ball screws.

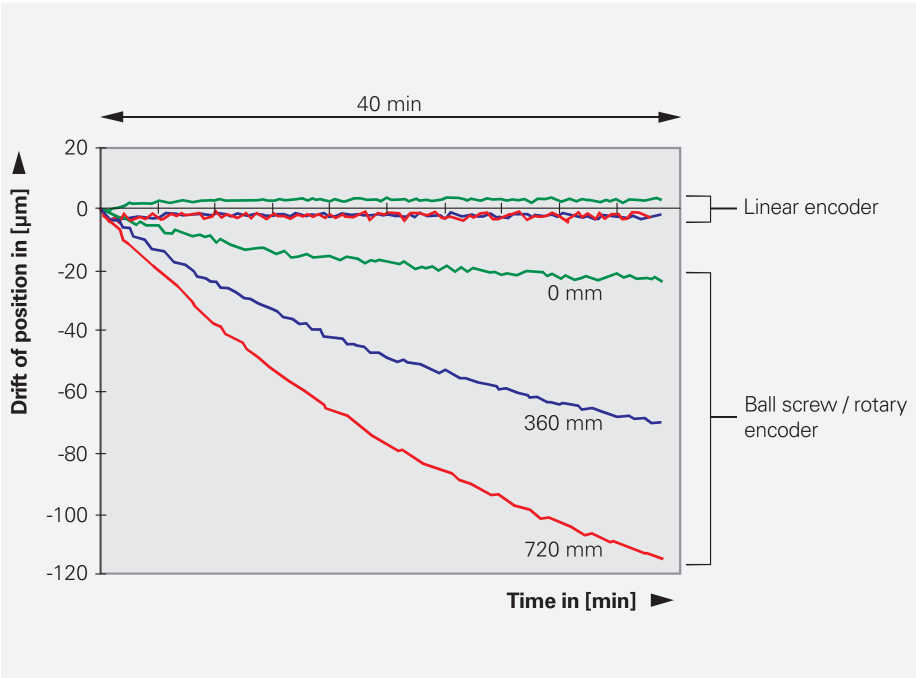


Fig. 7: Drift of three positions during positioning accuracy measurement in accordance with ISO/DIS 230-3 on a machining center with ball screw in fixed/floating bearings. Position measurements via rotary encoder and ball screw result in a distinct drift of positions due to the thermal growth of the screw.

No drift in position values measured with linear encoders
The measured positioning accuracy depends directly on the number of repetitions, particularly if there are only few repetitions. The measurements taken by the retrofitted linear encoders show no drift.

Series production
To demonstrate the applicability of this experiment in actual production conditions, a small batch of aluminum workpieces were machined on the same machine. Eight 70 mm x 70 mm workpieces were fixed on a vertical machining center. Four pockets and two radii were machined using four tools with an infeed of 1 mm in the Z axis (Fig. 8).

After the 6-minute machining operation the eight parts were not exchanged. Rather, the infeed in Z was increased by 1 mm and the operation repeated. As a result of the thermal expansion of the ball screw, all workpieces show a step pattern on the left side. This pattern is particularly pronounced on the workpiece farthest away from the fixed bearing. The right sides of the work-pieces are smooth because with each shift in the positive X direction the previous step was also removed. In principle the same effect could be observed in the Y direction as in the X direction, but because of the lesser amount of movement in the Y axis the step pattern is significantly less pronounced. In the X direction the comparative measurement of the step pattern shows a drift of approx. 90 µm with a time constant of thermal expansion slightly less than one hour (Fig. 9).

If additional work is to be done on previously machined workpieces with critical dimensions, the machine datum must be continually inspected and corrected. The machine achieves thermal equilibrium after one hour, but after an interruption in machining it begins to drift in the reverse direction. If the part program and with it the mean feed rate are changed, it again takes approx. one hour for the ball screw to regain thermal equilibrium.

Fixed bearing at both ends

The situation is more complex in the case of fixed/fixed bearings. Ideally rigid bearings would prevent expansion of the ball screw at its end points. However, this would require considerable force. To prevent expansion of a ball screw with 40-mm diameter, 2.6 kN must be applied per degree Celsius of temperature increase. A typical angular-contact ball bearing would quickly fail under any large increase in temperature. Under real conditions, the rigidity of the purportedly fixed bearings with their seats lies in the area of 800 N/µm. This means that as the temperature of the ball screw increases, the bearings deform significantly. The end points of the ball screw do not remain at their original position. The same experiment as in Fig. 7 was conducted on a vertical machining center (built in 1998) with fixed bearings at both ends. The tested 1-m long feed axis was mechanically designed to be very rigid. At each end of the ball screw the same bearing was built into seats that were machined directly into the machine's cast frame.

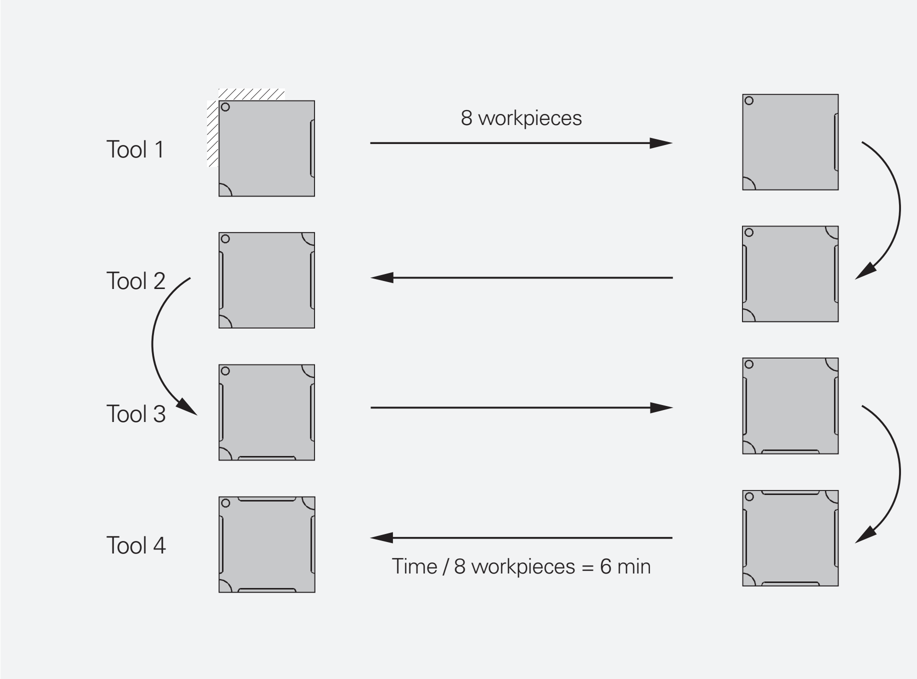


Fig. 8: Experimental setup for batch production with multiple workpieces. Four pockets and two radii were machined using four tools to a depth of 1 mm each. To illustrate drift resulting from thermal expansion of the ball screw, the workpieces were not exchanged after machining. Instead, the part program was run repeatedly at successively increasing depth.

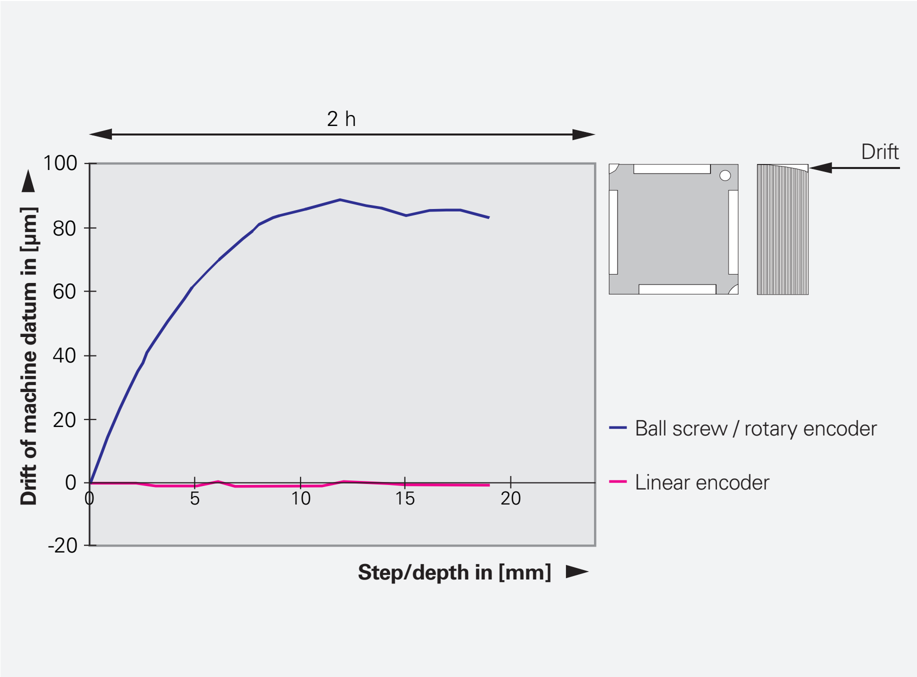


Fig. 9: Result of the experiment in Fig. 8. The left pocket of the finished workpieces plainly shows a step pattern resulting from thermal expansion of the ball screw.

The results of measurement in Fig. 10 show curves as theoretically expected. The end points of the ball screw cannot be kept in their original positions. They each move by 20 µm to 30 µm in the direction of the force generated by heat. The total expansion of the ball screw is about 50 % less than that shown in Fig. 7. This means that by designing fixed bearings at both ends, the expansion could be halved. The thermal zero point of the feed axis seems to lie at the midpoint of the traverse range. This is also expected because the bearings have approximately equal rigidity and the ball screw was heated evenly over its entire length.

Fixed/preloaded bearing

This type of bearing causes problems for traversing programs with high mean velocities because the bearing load is detrimental to service life and the forces to be withstood result in deformation of the machine structure. A fixed/preloaded bearing design is therefore often used as a sort of pressure valve (Fig. 6). With a typical preload of 50 µm/m, one would expect that such a bearing configuration would behave like a fixed/fixed combination up to a temperature increase of approx. 5 K, and beyond that, like a fixed/floating combination.

Figure 11 shows the results of a positioning test on a machining center with a ball screw with fixed/preloaded bearings conducted along the pattern of the previously described experiments. Surprisingly, in spite of the fixed/preloaded bearing configuration, a position drift similar to that in Fig. 7 becomes apparent. This means that the feed axis behaves roughly like one with fixed/floating bearings. The thermal zero point seems to lie near the fixed bearing. Unlike the axes in the two previous experiments, this axis had a travel of only 500 mm instead of one meter. The magnitude of the drift is therefore not comparable.
This experiment shows that the simple model of the ball screw with fixed/preloaded bearings does not stand up to reality. As a rule, the end with the movable bearing is much less rigid than the end with the fixed bearing. The cause lies in the difference in the bearing designs. While the first end with a genuine, inherently preloaded fixed bearing must continue to remain rigid when the second end has begun to move, with increasing temperature the second end loses its preload and therefore also its rigidity.

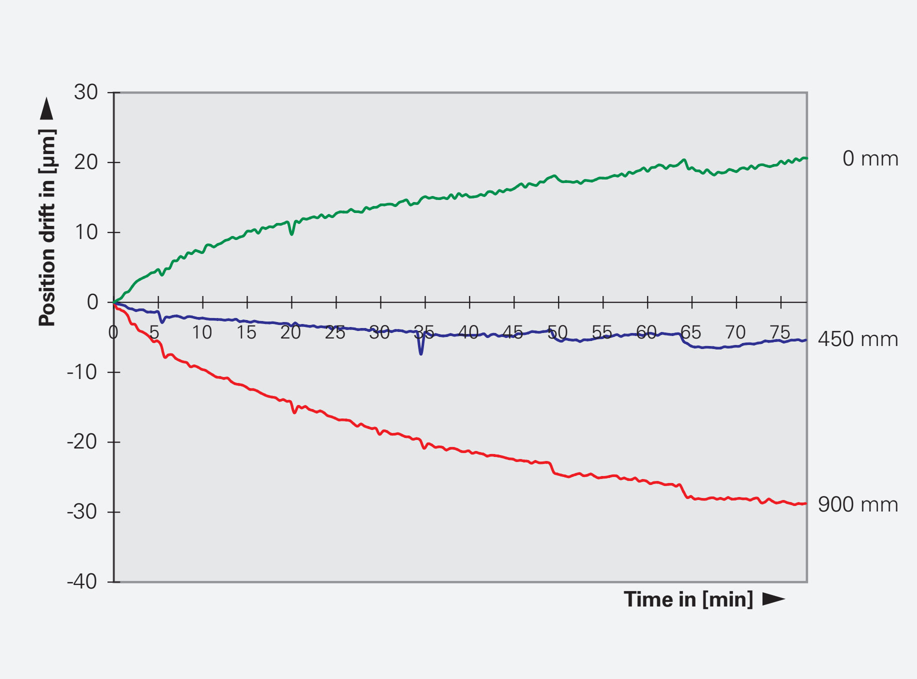


Fig. 10: Drift of three positions during positioning accuracy measurement in accordance with ISO/DIS 230-3 on a feed axis with a ball screw in fixed bearings at both ends. Position measurements via rotary encoder and ball screw result in a distinct drift of positions due to the thermal growth of the ball screw.

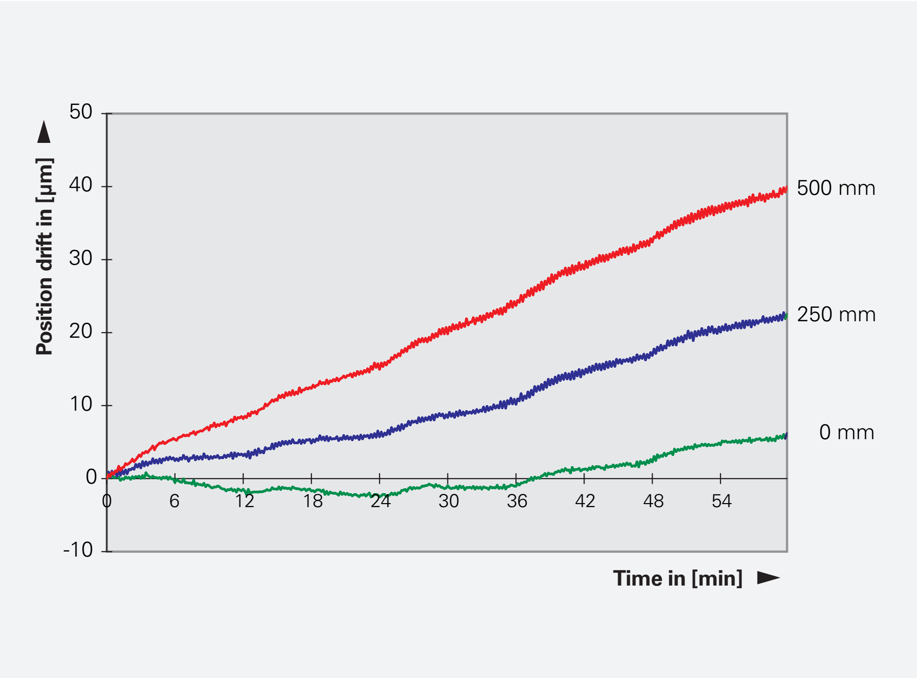


Fig. 11: Drift of three positions during positioning accuracy measurement in accordance with ISO/DIS 230-3 on a feed axis with a ball screw in fixed/preloaded bearings. The results show a distinct drift due to position measurement by ball screw and rotary encoder.

Influence of the temperature distribution along the ball screw

Apart from the ratio of bearing stiffness, the position of the thermal zero point depends particularly on the distribution of temperature along the ball screw. Figure 12 shows a thermographic snapshot of a ball-screw drive after several hours of reciprocating traverse between two points separated by 150 mm. As the thermograph shows, even after several hours the temperature increase remains almost exclusively in the area of the ball nut traverse. The temperature of the ball screw and therefore the thermal expansion is very local.

Because the bearings of the ball screw can provide at best only an evenly distributed mechanical tension and ensure constant expansion along the ball screw, they cannot compensate the expansion resulting from local temperature changes.

A simple calculation shows this clearly (Fig. 13). On a 1-m long ball screw with a fixed bearing at one end, a local temperature increase of 10 K as indicated by the red curve would result in a positioning error as indicated by the green curve. A fixed/fixed bearing configuration with a rigidity of 700 N/µm results in an error curve as indicated by the blue curve. As a result of the forces exercised by the bearing, the ball screw is compressed at its ends where the temperature is not increased. The area of the ball screw near its midpoint expands due to the temperature increase at almost the same rate as with the fixed/floating configuration.
At 22 µm, the maximum positioning error from the fixed/fixed configuration is roughly 2/3 of the error that occurs from the fixed/floating configuration.

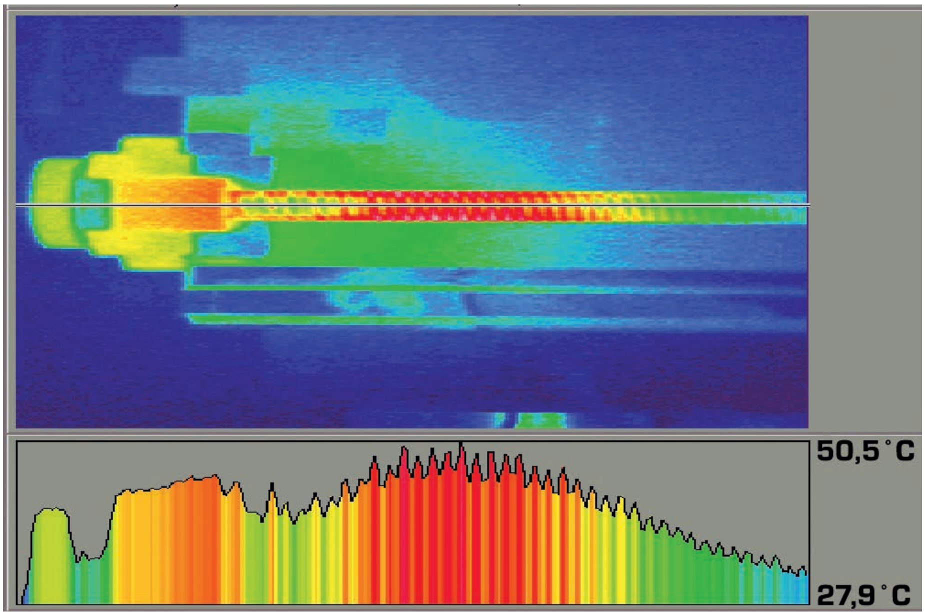


Fig. 12: Local heating of a ball screw in the traverse range of the ball nut after 6 hours of reciprocating traverse at 24 m/min between two positions separated by 150 mm [see A. Frank / F. Ruech: Position measurement in CNC Machines]. For this thermographic snapshot, the machine table was moved aside at the end of the traverse program. The illustration shows the higher temperatures of the belt drive, locating bearing, and ball screw.

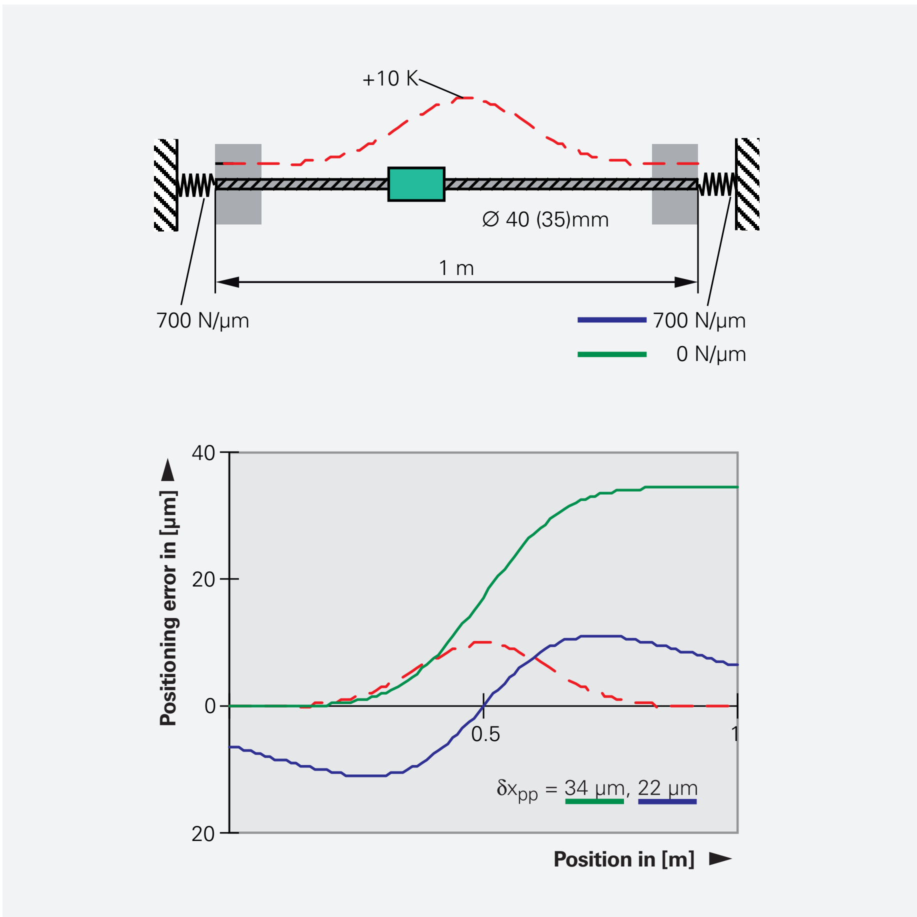


Fig. 13: Positioning error in a Semi-Closed Loop as a result of local temperature rise in the recirculating ball screw.

Countermeasures

The test results show that the thermal expansion of the ball screw as a result of friction in the bearings and particularly in the ball nut results in significant positioning errors if the axis is controlled in a Semi-Closed Loop. Besides the use of linear encoders, countermeasures aimed at avoiding this error include coolant-conducting hollow ball screws and purely electronic compensation in the control software.

Cooled ball screws
The circulation of the coolant requires a hole in the ball screw and rotating bushings near the screw bearings. Apart from the sealing problems, the hole reduces the ball screw's mechanical rigidity in its already weak axial direction. The greatest problem, however, is a sufficiently accurate temperature control of the coolant. A 1° C change in temperature changes the length of a 1-m long ball screw by 11 µm. In light of the considerable amount of heat to be removed it is not an easy task to maintain a temperature stability of < 1 K. This is particularly so when the spindle or its bearings are cooled with the same system. In such a case, the required cooling capacity can easily lie in the kilowatt range. The temperature constancy of existing ball-screw chillers is usually significantly worse than 1 K. It is therefore often not possible to use them to control the temperature of the ball screw. Switching controllers are often used in the chillers to reduce costs. Since each switching operation is triggered by a violation of temperature limits, the individual switching operation can be considered to be an expansion of the cooled ball screw and therefore an axis positioning error. Figure 14 shows the result of a positioning test on a vertical machining center with liquid-cooled ball screws in fixed/floating bearings. During the test, the axis was moved slowly at 2.5 m/min between two points separated by 500 mm. The maximum traverse range was 800 mm. The position drift of the position farther away from the fixed bearing was recorded. The switching of the chiller is plainly visible. Its hysteresis was 1 K. Compared with the noncooled Semi-Closed Loop design the position drift was significantly reduced. However, the switch operations produce relatively quick changes in position, which have a stronger effect during the machining of workpieces with short machining times than the slow position drift evident in the noncooled Semi-Closed Loop design.

Software compensation
Research is underway on compensation of thermal deformation with the aid of analytic models, neural networks, and empirical equations. However, the main focus of these studies is on the deformation of the machine tool structure as a result of internal and external sources of heat. There is little interest in investigation into compensation of axis drift.

As a whole, the possibilities of such software compensation are frequently overestimated in today's general atmosphere of enthusiasm for software capabilities. Successful compensation in the laboratory is usually achieved only after elaborate special adjustments on the test machine. It is usually not possible to apply such methods to machines from series production without time-consuming adjustment of the individual machines. The example of the feed axis shows the variations of the input parameters to be considered.

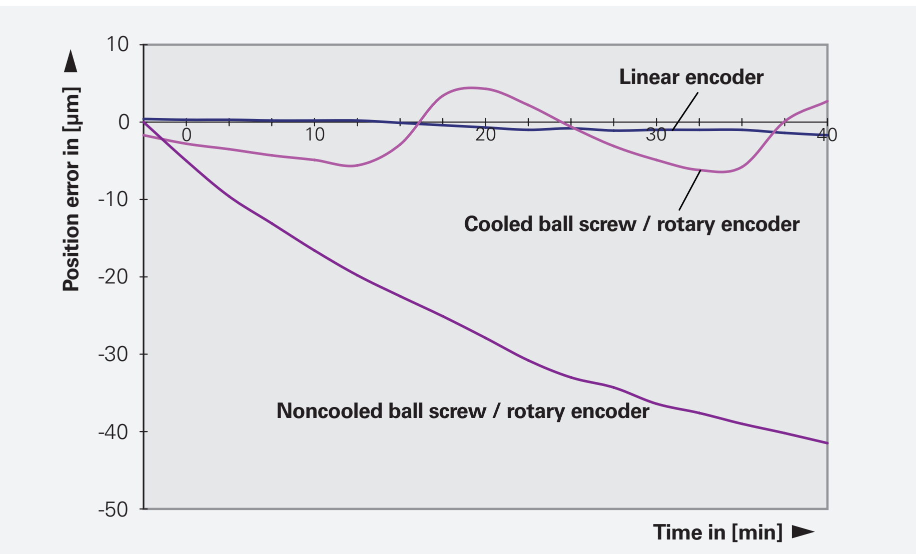


Fig. 14: X axis of a vertical machining center with liquid-cooled ball screw in fixed/floating bearings. The diagram shows the drift of the position farthest away from the fixed bearing during reciprocating traverse over 500 mm (800-mm traverse range) at 2.5 m/min. The axis was also equipped with a linear encoder for test purposes.

To compensate the expansion of the ball screw, its temperature must be known with respect to its position, since the local temperature depends on the traversing program. Direct temperature measurement of the rotating ball screw, however, is very difficult. Machine tool builders therefore often attempt to calculate the temperature distribution. This is theoretically possible if a heat analysis can be prepared for individual sections of the ball screw. The heat in such a section is generated by friction in the ball nut through thermal conductance along the ball screw, and through heat exchange with the environment. The friction of the ball nut depends almost proportionately on the preload of the ball nut and, in a complex manner, from the type, quantity, and temperature of the lubricant. The preload of the ball nut normally changes by ±10 % to ±20 % over its traverse range in an manner depending on the individual ball screw. In the course of the first six months, the mean preload typically decreases to 50 % of its original value. Due to the complex interaction of static forces at play on the ball screw, certain jamming effects and an associated increase in friction are unavoidable. Even these few examples show that the calculation of the actual frictional heat presents formidable problems. Calculating the heat dissipation is similarly difficult because is depends strongly on largely unknown ambient conditions. Even the temperature of the air surrounding the ball screw is normally unknown, although it plays a decisive role in any calculation of heat dissipation.

On the whole it seems certain that, even in the relatively simple case of a fixed/floating bearing, software compensation of ball-screw expansion without additional temperature sensors has little chance of success. In the case of fixed/fixed and fixed/preloaded bearing one must additionally take into account the bearing rigidity and the preload-dependent friction in the bearings. These factors make compensation even more difficult.

Comparison of positioning error with other types of error

After this discussion of temperature-dependent positioning error of feed drives, it remains to classify these types of error with the other types of static and quasistatic error in the total error budget of the tested machining centers. The frame deformation resulting from the heat generated by the spindle drive was examined on all three machines in accordance with ISO/DIS 230-3. After several hours of operation at a maximum spindle speed of 6000 rpm, the first machining center showed a linear deformation of {X: 5 µm, Y: 60 µm, Z: 15 µm}. The rotational deformation reached a maximum of {A: 40 µm/m, B: 70 µm/m}.

The deformation of the second machining center is shown in Fig. 15. Under the same conditions, also with 6000 rpm, it shows a maximum linear deformation of {X: 5 µm, Y: 45 µm, Z: 55 µm}. The rotational deformation reached a maximum of {A: 25 µm/m, B: 10 µm/m}. The third machine was equipped with a high-speed spindle and jacket cooling. At 12000 rpm it showed linear deformations of {X: 5 µm, Y: 5 µm, Z: 40 µm} and rotational deformations of max. {A: 20 µm/m, B: 30 µm/m}. The measured axis drift values attain at least the same magnitude as the structural deformation. Particularly on spindles with fixed/floating bearings, or machines with effective cooling of the spindle, the positioning error of the feed axes driven in a Semi-Closed Loop is significantly greater than the measured structural deformation.

A comparison with the usual geometric error leads to similar results. If one observes the pitch, roll, and yaw error of the feed axes of 16 different NC machines one sees that these types of error usually lie in the range of 10 µm/m to 50 µm/m (Fig. 16). The positioning error is found by multiplying these values by the respective Abbe distance. The error does not attain the values of the feed axis until over 1 meter traverse.

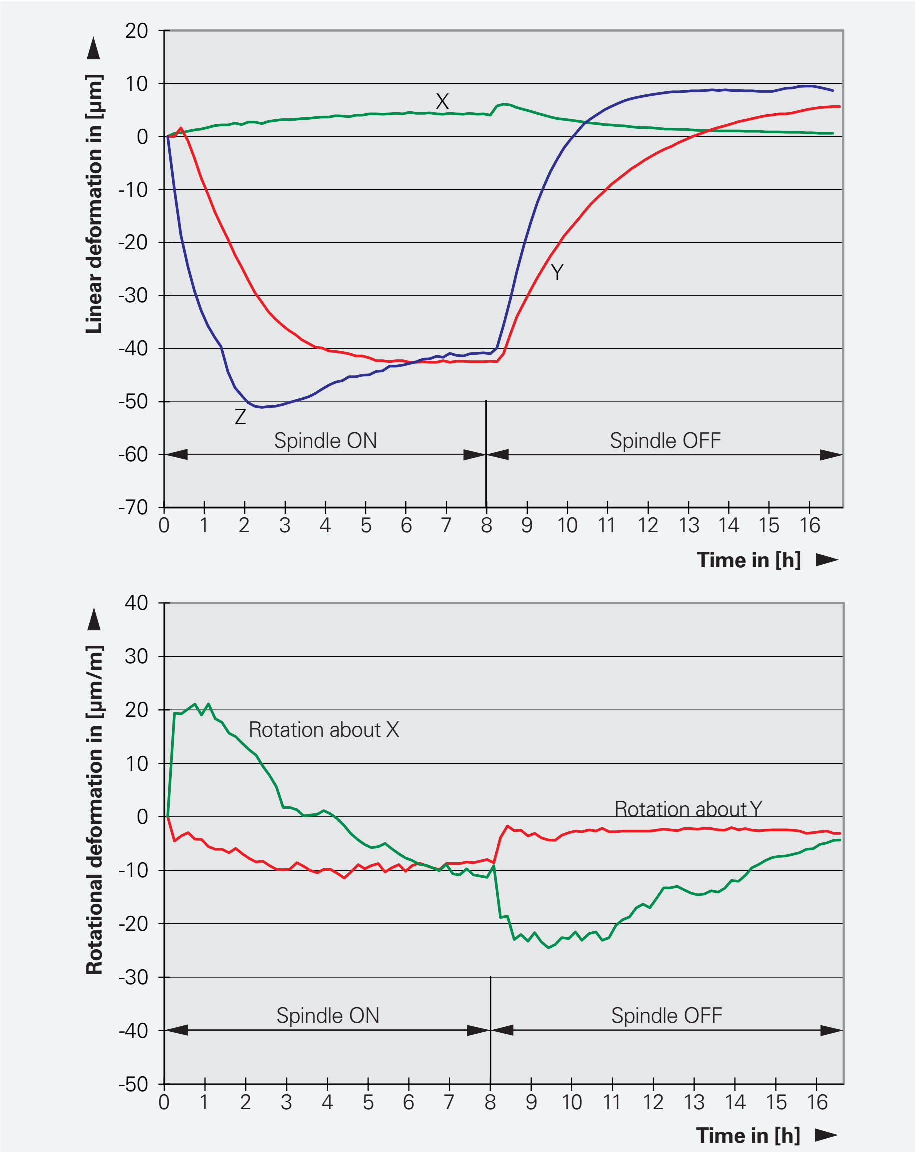


Fig. 15: Structural deformation of a vertical machining center as a result of heat generated in the spindle drive at 6000 rpm without load.

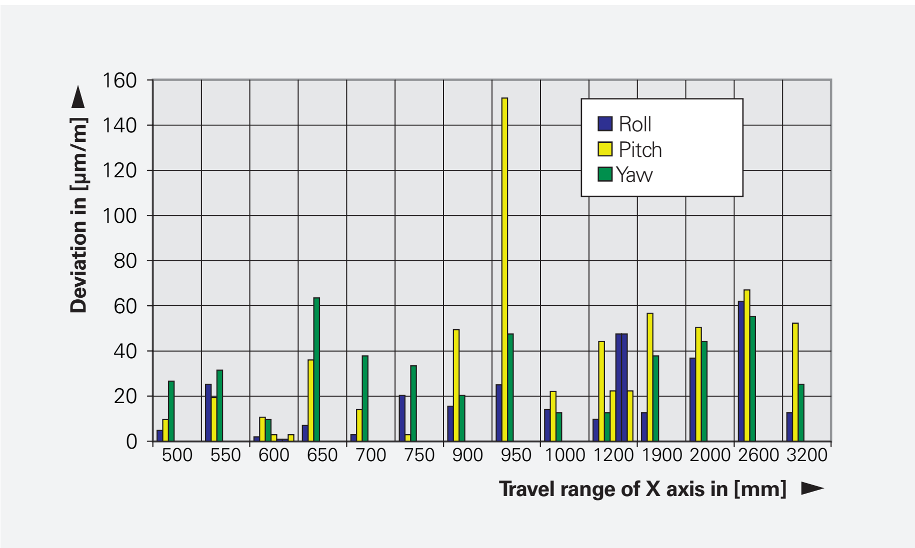


Fig. 16: Pitch, roll, and yaw of feed axes of 16 different NC machines

Conclusion

The primary problem involved with position measurement using rotary encoder and ball screw is the thermal expansion of the ball screw. With typical time constants of 1 to 2 hours, thermal expansion causes positioning error in the magnitude of 0.1 mm, depending on the nature of the part program. This positioning error therefore outweighs the thermally induced structural deformation and geometric error of machining centers.

After every new part program the ball screw requires approx. 1 hour to attain a thermally stable condition. This also applies to interruptions in machining. A rule of thumb for thermal expansion is that, over the entire length of a cold ball screw 1 meter in length, the ball screw grows by approx. 0.5 µm to 1 µm after every double stroke. This expansion accumulates within the time constant.

As requirements for machine tool accuracy and velocity increase, the role of linear encoders for position measurement grows increasingly important. This should be taken into consideration when deciding on the proper feedback system design.

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394843 · 01 · A · 02 · 10/2018 · PDF