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How retaining forces affect spindle bearings

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Machine builders must accurately predict **spindle** preload for high-speed applications.

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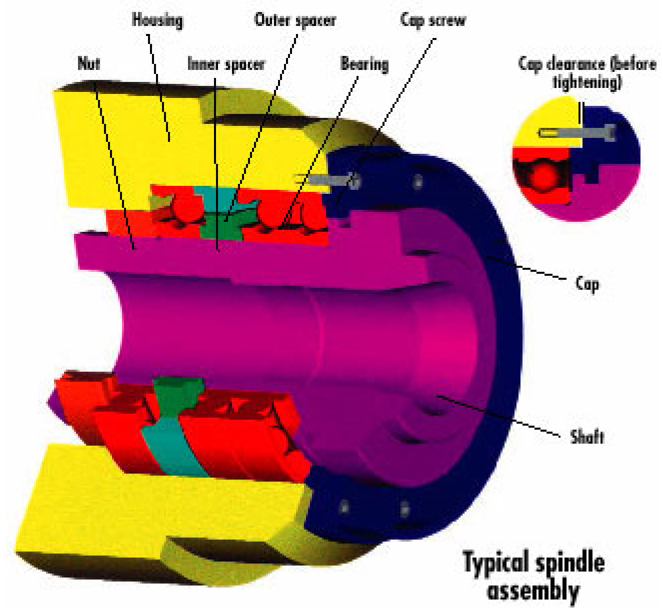
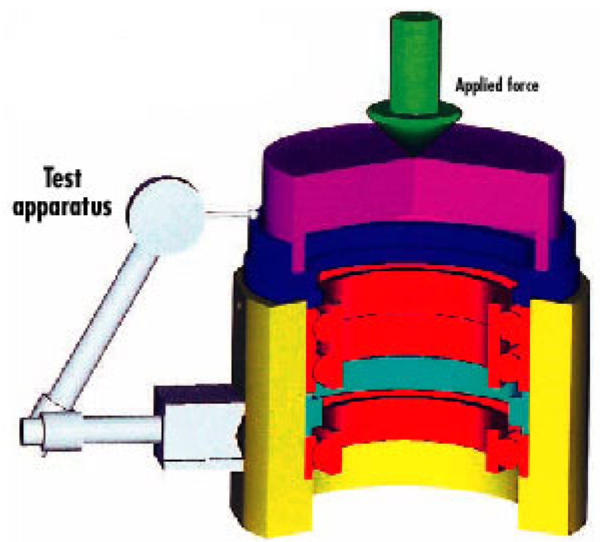
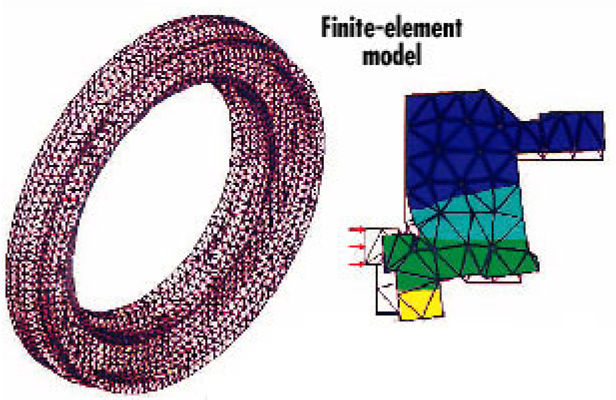


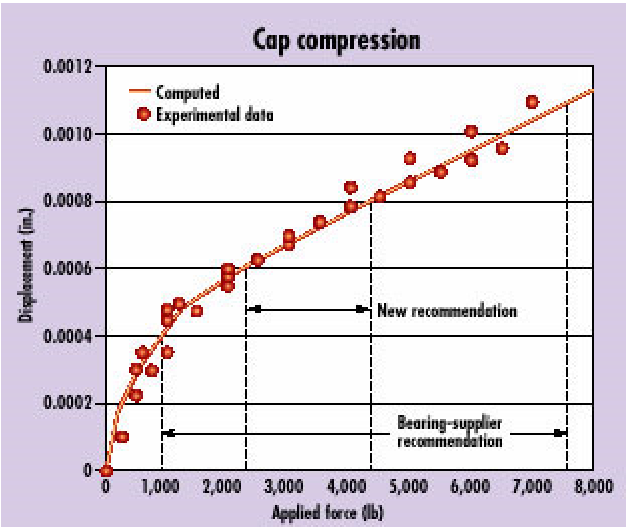
Cross-sectional diagram shows the front portion of a machine-tool **spindle** assembly with a triplex set of angular-contact **bearings**.



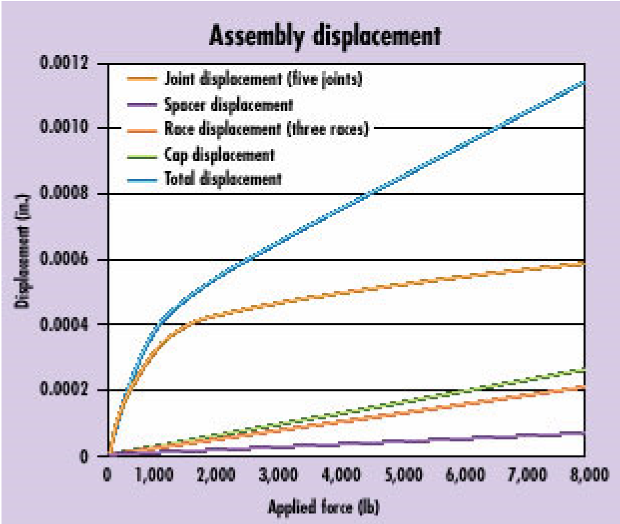
Cross-sectional diagram illustrates the test apparatus used to measure the overall deflection of the bearing assembly.



The FEA model shows a typical cap loaded with a uniform pressure along its area of contact with the outer race of the bearing. A zero-displacement boundary condition was applied to the area of the cap that contacts the housing. Although this simplification results in a higher calculated cap stiffness, the assumption should be approximately valid as long as the cap screws mount near the inner radius of the housing.



Comparison of the measured and computed cap compression.



Computed displacements of each component in the assembly are summed to yield the overall cap displacement.

Modern CNC machine tools work under increasingly diverse cutting conditions. They must handle workpieces of various sizes and materials, and produce good surface finishes while operating at ever higher metal-removal rates. This places rigorous demands on machine-tool spindles supported by rolling-element **bearings**. Spindles must operate with relatively low temperatures and sufficiently long life at high speeds, all the while providing high stiffness for lowspeed cutting.

Key to both bearing temperature and **spindle** stiffness is bearing preload. High stiffness requires high preloads and lets the **spindle** make precision cuts without chatter. This is often acceptable for low-speed operations, but highly preloaded ball **bearings** generate more heat and typically have shorter life than lightly preloaded ones. This is one reason why high-speed **spindle bearings** usually have light preloads.

Clearly, machine builders must be able to accurately predict preload to build spindles for high-speed or highstiffness applications. A typical machine-tool **spindle** includes a set of three or four angular-contact ball **bearings** mounted onto a shaft and held rigidly inside a housing. For this configuration, bearing preloads may vary from one **spindle** to another due to:

• Interference between the inner bearing races and the shaft.
• Interference or clearance between the outer bearing races and the housing.
• Torque on the nut that clamps the inner races.
• The manufacturing accuracy.

A secondary but important influence on preload is the axial force the **retaining** cap applies to the outer bearing races to hold them rigidly inside the housing. The **retaining** force compresses the outer races and reduces bearing preload. Thus, manufacturers must clamp the outer races with sufficient force to rigidly hold the bearing assembly, but not so great as to distort the races and cause irregular motion and reduced preload.

The major **spindle**-bearing suppliers typically recommend similar mounting procedures for the configuration shown. The **retaining** cap's pilot portion must provide an interference. Thus, fully tightening the cap screws seats the **retaining** cap flush to the bearing housing and compresses the outer bearing races by a specified amount. For example, both NSK and SKF recommend that the cap provide between 10 to 30 m (0.0004 to 0.0011 in.) of compression. This recommendation holds for a broad range of bearing sizes and **spindle** configurations.

This discussion presents a method to estimate the clamp force on the outer bearing races if users follow the bearing vendors' guidelines, and quantifies how this force impacts bearing preload and **spindle** stiffness. A simple analytical model lets **spindle** designers calculate clamp loads on the outer races for different **spindle** configurations.

PHYSICAL MODEL

Although manufacturers typically specify bearing retainment in terms of initial cap clearance, the desired result is a **retaining** force. For a given **retaining** compression, the force that clamps the outer races depends on:

• The compressive stiffness of the bearing races.
• The compressive stiffness of the spacer between the **bearings**.
• The stiffness of the cap in bending.
• The contact stiffness of the interfaces between **bearings** and adjacent components.

Consider the bearing assembly shown with three bearing races, a spacer, and **retaining** cap. Each component has compliance as it compresses axially. Also, the interfaces between steel parts are not perfectly rigid but have a certain compliance that depends on the contact pressure, as well as the components' surface finish and flatness. One can model the relationship between cap force and outer-race compression as a set of springs in series, acting together under the applied load of the **retaining** cap. Define the total compression of the assembly under load from the cap by:

$$\delta_t = 3\delta_R + \delta_S + \delta_C + \delta_1 + \delta_2 + \delta_3 + \delta_4 + \delta_5$$

Estimate compression of the outer bearing race by modeling it as a cylinder with a constant cross section:

$$\delta_R = \frac{FL_R}{EA_R}$$

Likewise, estimate the compression of the spacer between the outer bearing races by modeling it as a cylinder with a constant cross section:

$$\delta_S = \frac{FL_S}{EA_S}$$

Designers can determine cap compression C experimentally or analytically using finite-element analysis.

Estimate empirically the compression of joint interfaces between assembled components. For a flat joint between two rigid elements loaded in compression, approximate the relation between joint deflection and interface pressure by:

$$\delta_J = c_1\sigma^m - (c_1 - c_2)\langle\sigma^m - \sigma_0^m\rangle$$

where is the contact pressure between the two elements,

$$\sigma = \frac{F}{A_J}$$

The second half of the equation (the terms in brackets) is a step function whose result equals zero if < $_0$. This transition in joint compression accounts for the fact that joints react differently under low and high pressures.

Experimental data can be used to approximate the joint compression between steel parts. It shows that the transitional value of the contact pressure $_0$ is approximately 400 psi, and good agreement with the data is obtained with m = 0.5. The values of c_1 and c_2 vary with the application. Experimental data shows that joint behavior depends on the surface finish and flatness of mating parts, so whether the parts are turned, ground, or scraped can have a dramatic effect on the value of these constants. For ground steel parts, c_1 varies between approximately 3 to 20 in./(psi)$^{0.5}$, and c_2 between approximately 1 to 4 in./(psi)$^{0.5}$. Designers must experimentally determine actual values for each application.

PRACTICAL APPLICATION
Consider an application of the typical triplex set of **bearings**. A test apparatus was used to determine the overall bearing-assembly deflection. It applies a known, uniform force to the cap to mimic the clamping action of the cap screws. An electronic indicator with 0.000010-in. resolution measures overall cap compression with respect to the bearing housing.

The assembly uses NSK Series 7022 angular-contact **bearings** with a 110-mm bore diameter and 170-mm OD, and $AJ = 3.08$ in.^{2}, $L_R = 1.095$ in., and $AR = 4.08$ in.^{2} The steel spacer between the **bearings** has $L_S = 1.10$ in. and $AS = 4.01$ in.^{2} The spacer is precision ground with a flatness approximately 0.00005 in. and the faces are parallel within approximately 0.0001 in. The cap face which abuts onto the outer race of the bearing is a turned surface flat within approximately 0.0001 in., and the surface finish of all parts is approximately 16 in.

Measurements show that cap loads of approximately 1,000 to 7,500 lb yielded the overall cap displacement of 0.0004 to 0.0011 in. recommended by the bearing vendors.

Values of $c_1 = 4.0$ in./(psi)$^{0.5}$ and c2 = 1.25 in./(psi)$^{0.5}$ provide a good correlation between measured and computed displacement. Compression of the five joints is the largest contributor to cap deflection.

The assembly stiffness increases with applied load because contact pressure between the joints also increases. The transitional contact pressure reaches 400 psi at an applied load of approximately 1,300 lb and a displacement of approximately 0.0005 in. Above this force, the assembly stiffness (applied force divided by total deflection) remains approximately constant at 10 million lb/in.

It can be surmised that the cap pilot of this particular assembly should be sized to provide at least 0.0006 in. of **retaining** compression to ensure it exceeds the transitional joint pressure and does not compromise assembly stiffness. This corresponds to a clamping force of approximately 2,300 lb for this assembly.

The upper limit of **retaining** compression should be sized as close as reasonably possible to the lower limit but within measurement-error limits. Higher compression tends to reduce preload in the bearing set and therefore reduce assembly stiffness. Excessive **retaining** compression can distort the outer ring and result in detrimental bearing runout. Therefore, the recommended range of **retaining** compression for this specific application is 0.0006 to 0.0008 in. The next step is to verify the impact of this **retaining**-compression range on the assembly stiffness.

CAP FORCE EFFECTS
As the **retaining** cap clamps the assembly into the bearing housing, the bearing outer races, spacers, and joints between them compress. This tends to reduce the mounted preload and stiffness of the bearing assembly. To judge the impact of cap force on bearing preload, designers need an accurate method to measure axial stiffness of the **spindle**. One simple method is to apply a known force and measure the deflection, but this requires a suitable apparatus to apply large **forces** (typically >1,000 lb). A simpler approach is to orient the **spindle** vertically and measure its natural frequency in the axial direction using an accelerometer and a Fast Fourier Transform (FFT) vibration analyzer. The axial stiffness of the assembly is

$$K_a = M_t \omega^2$$

In the assembly shown, the mass of the shaft, inner spacer, nut, and inner bearing races are 76.1 lb. For any measured natural frequency, we can calculate the axial stiffness of the assembly.

A test assembly applies a predetermined force to the **retaining** cap and judges its impact on the **spindle** stiffness. The force applied to the cap by the cap screws can be approximated by:

$$F_b = \frac{T_b}{0.2D_b}$$

The assembly has ten M6 X 1.0 screws, and when torqued to 50 lb-in. exerts a force of approximately 10,600 lb to the **retaining** cap.

The table shows the measured natural frequency of the system as a function of cap-screw torque. Initially, no torque was applied to the cap screws; only the weight of the assembly held it inside of the bearing housing. In this configuration, measured natural frequency is 720 Hz, corresponding to an axial stiffness of approximately 4.0 million lb/in. Increasing torque on the cap screws determines the effect of increasing cap force on the bearing stiffness. As the clamping force increases, bearing-assembly stiffness drops until reaching a lower limit of approximately 3.4 million lb/in.

As shown, clamping the races with 3,200 lb reduces axial stiffness of the assembly by 5%. Although the contact pressure between the outer races and bearing housing increases, the reduction in bearing preload dominates the result and reduces stiffness. The test demonstrates that the recommended range of **retaining** compression, 0.0006 to 0.0008 in., which produces approximately 2,300 to 4,300 lb of clamping force, results in an assembly stiffness of approximately 3.7 to 3.9 million lb/in. — very close to the maximum possible stiffness with this bearing set. The selected range of **retaining** compression provides a balance of high stiffness and adequate bearing retainment for this **spindle**-bearing assembly.

TEST RESULTS

Torque applied to cap screws (lb-in.)	0	15	25	40	50	75	100
Calculated force applied to races (lb)	0	3,200	5,300	8,500	10,600	15,900	21,200
Measured natural frequency (Hz)	720	700	680	680	660	660	660
Axial stiffness (lb/µin.)	4.0	3.8	3.6	3.6	3.4	3.4	3.4

Test results show the measured natural frequency of the system as a function of cap-screw torque.

Nomenclature

- A_J = Joint contact area
- A_R = Cross-sectional area of the race
- A_S = Cross-sectional area of the spacer
- c_1 = Empirical constant
- c_2 = Empirical constant
- D_b = Cap screw diameter
- E = Elastic modulus of the race material
- E_S = Elastic modulus of the spacer material
- F = Applied axial force
- F_b = Force applied by cap screws
- K_a = Axial stiffness of the assembly
- L_R = Axial length of the race
- L_S = Axial length of the spacer
- M_v = Mass of the vibrating components
- m = Empirical constant
- T_b = Cap-screw torque
- δ_C = Compression of the cap as it bends under an axial load.
- δ_R = Compression of a single outer bearing race under an axial load.
- δ_S = Compression of a spacer under an axial load
- δ_T = Total compression of assembly.
- δ_1 = Compression of joint interface between outer race and housing.
- δ_2 = Compression of joint interface between outer face and spacer.
- δ_3 = Compression of joint interface between outer face and spacer.
- δ_4 = Compression of joint interface between the two bearing races.
- δ_5 = Compression of joint interface between bearing race and cap.
- σ = Contact pressure between two rigid elements.
- ω = Natural frequency of the axial-vibration mode.

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