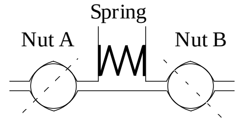


J-type preloading

- **J-type preloading uses a spring (e.g. disc spring) between the nuts to establish a constant preload.**
    - **This provides the most constant preload and hence the most uniform drive torque is obtained.**
    - **However, the carriage can only be mounted rigidly to one part of the nut.**
    - **Thus the load capacity in one direction is severely reduced.**
    - **For some axes on some machines (e.g., some turning centers) this is not a problem.**

- **Preload can also be obtained by:**
  - **Hanging a weight by a cable over the edge of the machine.**
  - **Running the cable over a pulley, and connecting it to a carriage that moves in a horizontal plane.**
    - **This is suitable only for applications where the load is much less than the preload.**
    - **One experimental setup used a carpenter's tape measure as a constant force spring!**
- **Vertically moving axes can use a portion of their weight to provide a constant preload to a ballscrew nut.**
  - **Make sure that in a vertical motion system, a brake is used to prevent the axis from falling if power dies.**

Preload selection

- The higher the preload of a ballscrew nut, the greater the stiffness, heat generation, and wear rate.
- For a compressive or tensile preloaded nut:
    - Ideally one nut will only just lose contact when the maximum force is applied.
- The displacement of a system governed by Hertzian contact is proportional to the force to the two-thirds power.
- Although a preload of 0.35 times the maximum load is the "optimal" value from a stiffness point of view:
    - It leads to excessive heat generation and wear.
- Since Hertzian contact stiffness acts as a hardening spring:
    - A good compromise for the maximum preload is 10% of the basic dynamic rated load.
- The total axial stiffness of the leadscrew system will be a function of:
    - The nut.
    - The screw shaft.
    - The journal bearings.

Load life calculations

- **Leadscrew life can be defined in terms of revolutions, hours, or meters of travel.**

- **Variables used to determine life include:**
  - **L Fatigue life in revolutions.**
  - **L_t Fatigue life in hours.**
  - **L_s Fatigue life in kilometers nut travel.**
  - **C_a Basic dynamic rated load.**
  - **f_w Load factor:**
    - **Smooth without impact 1.0-1.2**
    - **Normal operation 1.2-1.5**
    - **Impact and vibration 1.5-3.0**
  - **Fa Axial load.**
  - **n Rotation speed (rpm).**
  - **ℓ Lead (mm).**

- **The load-life equations are:**

$$L = \left(\frac{C_a}{F_a\, f_w}\right)^3 \times 10^6$$

$$L_t = \frac{L}{60\, n}$$

$$L_s = \frac{L\, \ell}{10^6}$$

- **Given a series of N axial loads F(i), each applied at a rotating speed of n(i) for a duration t(i):**
  - **The equivalent axial load F_a is:**

$$F_a = \left(\frac{\sum_{i=1}^{N} F(i)^3 \, n(i) \, t(i)}{\sum_{i=1}^{N} n(i) \, t(i)} \right)^{1/3}$$

- **Similarly, the equivalent rotational speed is given by:**

$$n = \frac{\sum_{i=1}^{N} n(i) \, t(i)}{\sum_{i=1}^{N} t(i)}$$

- **Rarely is a leadscrew designed for "infinite life". Typical machine tool applications have a 20,000 hour minimum life.**
- **The maximum static load must also be checked where:**
  - **The maximum static load is typically 1/2 of the rated static load for normal operation.**
  - **2-3 times less for operation with impact or vibration.**
- **For ballscrews, the rated static load will produce a permanent deformation at the contact points equal to 0.01% of the ball diameter.**
- **Once deformed, the leadscrew will be noisy.**

- **For very slow operation or infrequent operation of a machine or some of its axes:**
  - **Beware of fretting corrosion between the balls and the shaft.**
- **In these applications, the maximum load has to be seriously derated, sometimes by a factor of 2-10.**
- **The machine should be moved through its full range of motion several times a day.**
- **Some manufacturers can be persueded to use stainless steel or ceramic balls which prevent fretting corrosion.**
- **The leadscrew size should be the larger of the one chosen by the minimum stiffness criteria or by the load life criteria.**

Choosing the optimal lead

- **The shaft diameter is first chosen using the minimum axial stiffness criteria.**
    - **Don't forget other components that affect axial stiffness.**
- **Together with the expected motor inertia, this will establish the rotational inertia of the drive system.**
- **This is used in the determination of optimal transmission ratio (for a lightly loaded systen, lead in mm/rev):**

$$\ell = 2\pi \times 10^3 \sqrt{\frac{J_{motor}}{M_{load}}}$$

- **Quite often a small lead results from these specifications.**
- **The critical shaft speed often becomes the dominant lead selection factor.**
- **Given a maximum desired carriage velocity V_{max} (in meters per second) and the maximum shaft speed w_n:**
    - **The minimum leadscrew lead in meters is:**

$$\ell = \frac{2\pi\, V_{max}}{\omega_n}$$

- **The lead can be used with the leadscrew, coupling, and load inertia, to find the "optimal" motor inertia.**
    - **One usually does not have to worry about "windup" stiffness of the ballscrew:**
        - **The equivalent axial stiffness of the torsional stiffnesses:**

$$K_{\Gamma \text{axial eq.}} = \frac{G I_p 4\pi^2}{\ell^2 L} = \frac{\pi^3 r^4 E}{(1 + \eta)\ell^2 L}$$

Example: Preliminary selection of ballscrew diameter:

- **Determine the minimum system stiffness (spreadsheet Actuator.XLS):**

Minimum stiffness for controllability	
Max design force (N)	**10000**
Stiffness error (µm)	**0.1**
Mass error	**0.1**
ADC resolution (bits)	**10**
Minimum system mass (kg)	**500**
Maximum system mass (kg)	**2000**
Min. static axial stiffness (N/µm)	66
Min. static axial stiffness (lb/µin)	0.37
Natural frequency (Hz)	57.61
Mech. time constant (sec)	0.0174
Basic servoloop time (sec)	0.003907

Controller type and # 0 order holds	servoloop time (µsec)
PID, 2 ZOH	1953
Full state feedback	
3rd order system, 2 ZOH	1953
4th order system, 3 ZOH	1302
Motor min rpm	0.25

- **Quite often, this is the least stringent requirement for sizing a ballscrew.**

- **Ballscrew size is often driven by load capacity requirements.**

- **Time spent at load (initial estimate), can be evaluated using the Analytical Hierarchy Process:**

Best case

AHP importance of different load times

1 = column value equal importance, 9 = column value much more important

%Load	20%	40%	60%	80%	100%	200%		% importance
20%	1.00	**2.00**	**3.00**	**5.00**	**7.00**	**9.00**	3.52	43.72%
40%	0.50	1.00	1.50	2.50	3.50	4.50	1.76	21.86%
60%	0.33	0.67	1.00	1.67	2.33	3.00	1.17	14.57%
80%	0.20	0.40	0.60	1.00	1.40	1.80	0.70	8.74%
100%	0.14	0.29	0.43	0.71	1.00	1.29	0.50	6.25%
200%	0.11	0.22	0.33	0.56	0.78	1.00	0.39	4.86%
	2.29	4.57	6.86	11.44	16.01	20.59	8.04	100.00%

Worst case

AHP importance of different load times

1 = column value equal importance, 9 = column value much more important

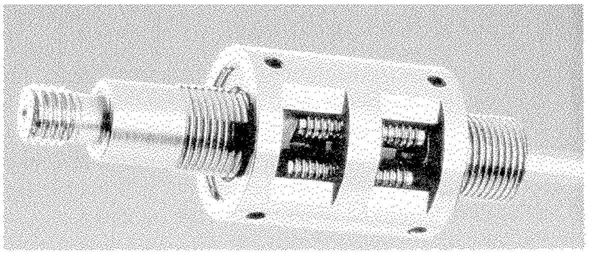
%Load	20%	40%	60%	80%	100%	200%		
20%	1.00	**0.33**	**0.14**	**0.20**	**3.00**	**7.00**	0.76	6.07%
40%	3.00	1.00	0.43	0.60	9.00	21.00	2.29	18.21%
60%	7.00	2.33	1.00	1.40	21.00	49.00	5.35	42.49%
80%	5.00	1.67	0.71	1.00	15.00	35.00	3.82	30.35%
100%	0.33	0.11	0.05	0.07	1.00	2.33	0.25	2.02%
200%	0.14	0.05	0.02	0.03	0.43	1.00	0.11	0.87%
	16.48	5.49	2.35	3.30	49.43	115.33	12.60	100.00%

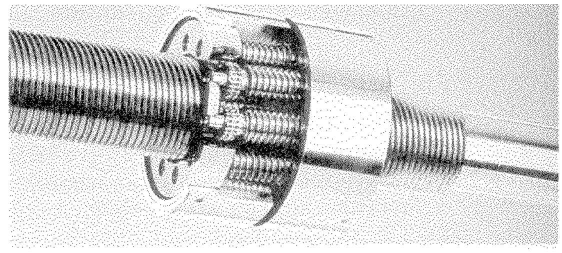
- **To obtain this information, one has to work closely with marketing and talk to potential users of the machine.**

- **This information is also used to select the proper motor and optimum transmission ratio as discussed in Section 10.2.**

Ballscrew selection (simply supported condition)	**worst case**	**best case**
Max design force (N)	**10000**	**5000**
Load life calculations (fixed-simply supported shaft)		
Shaft length (m)	**1.9**	
Minimum buckling diameter (mm) (safety factor of 2 on load)	22.2	
Optimal lead (assume motor J is 0.25 screw J)	2.97	
Actual lead selected from standard available (mm)	**10**	
Maximum allowable speed for buckling diameter (rpm)	1395	
Maximum linear velocity (m/min, ipm)	14	549
Minimum velocity (m/min, ipm)	0.0025	0.10
Maximum rotational speed for DN<70000 (rpm)	3154	
Number of cycles per minute	**1**	**0.2**
Number of hours per day	**16**	**8**
Number of days per week	**6**	**5**
Number of years of life	**5**	**5**
Number of cycles	1,497,600	124,800
Number of revolutions per shaft length	640	640
Total number of revolutions	957,837,696	79,819,808
Load factor (1-3, 3 = shock)	**3**	**1.5**
Percent time at different loads		
% time at 20% max load	6.07%	43.72%
% time at 40% max load	18.21%	21.86%
% time at 60% max load	42.49%	14.57%
% time at 80% max load	30.35%	8.74%
% time at 100% max load	2.02%	6.25%
% time at 200% max load	0.87%	4.86%
Life Load	7040	4084
Life load with load factor	21,119	6,126
Required Ca (basic dynamic load rating) (N)	208,182	26,374

Rollerscrews





- **Use rollers instead of balls to convert rotary to linear motion and transfer forces between the nut and the shaft.**
- **Rollers have a single thread with a pitch equal to the leadscrew's apparent pitch (real pitch / number of leads).**
- **The rollers mesh with both the screw and the nut.**
    - **They are spaced apart with spacer rings at their ends which act like the cage in a roller bearing.**
- **As the nut rotates relative to the leadscrew, the rollers rotate and orbit in the nut like rollers in a roller bearing.**
    - **The advance of the rollers in the nut due to orbiting is cancelled by their rotation.**

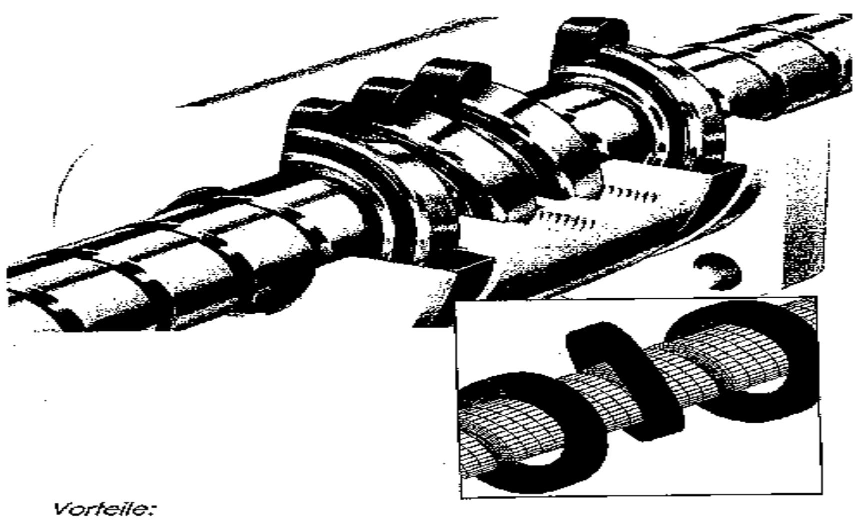
- **Planetary Rollerscrew**
  - **The ends of the rollers have external gears which mate with an internal gear in the nut which:**
  - **High speed applications**
  - **Compensates for minute inaccuracies in the actual relations between the pitch diameters.**
  - **Drives the rollers over any debris (e.g., ice) that may have gotten into the threads.**
- **Recirculating Rollerscrew**
  - **The nut has a region that allows the rollers to move radially out of the shaft thread.**
  - **A cam sends the rollers back to start a new circuit.**
  - **Allows for very fine pitch threads (2 mm).**

- **Rollerscrew size ranges:**
  - **Diameter ranging from 3.5 to 200 mm.**
  - **Pitch ranging from 1 to 40 mm.**
  - **Single nut dynamic load capacity ranging from 5300 N to 753 kN.**
- **The nut can be made in one piece without preload, or in two pieces if preload is required.**
- **Maximum speed of a planetary roller leadscrew is ω (rpm) x D (mm) = 140,000 (twice that of a ballscrew)**
- **The large number of contacting threads creates a large averaging effect**
- **This also gives rollerscrews load and stiffness capabilities many times higher than similar diameter ballscrews.**
  - **Hertz contact area is 3x that of a same size ballscrew.**
  - **Load capacity is 3x that of a same size ballscrew.**
  - **Life = $(F/F_{max})^3$, so life is 27x that of a same size ballscrew.**
- **Rollerscrews are manufactured with accuracies and efficiencies equal to those of ballscrews.**
  - **CAREFUL: Tens-of-hours run-in may be required to ensure the very stiff nuts are properly preloaded!**
- **Planetary rollerscrews are much quieter than ballscrews.**
- **Depending on the screw, a rollerscrew may cost one to three times as much as a ballscrew.**

INA's "Wallowing Thread Screw"

- **INA introduced ballscrew-like shaft with a helical gothic arch lead.**
  - **Instead of small balls rolling on a large shaft (at high speed), it uses four ball bearings:**
    - **The inner races are rounded to mate with the gothic arch groove, and the axes of rotation are inclined to the shaft.**
  - **The result is an order-of-magnitude lower rolling element velocity, and no need for lost contact during recirculation.**
  - **This actuator is half the cost of a linear motor system and can run at nearly the same speed!**
  - **It is perfect for the actuation of grinding tables at either creep-feed or reciprocal grinding speeds; However, it still has the issue of shaft rotational inertia.**

Wälzringgewindetrieb als X-Achsen-Antrieb der C740 CNC



Vorteile:

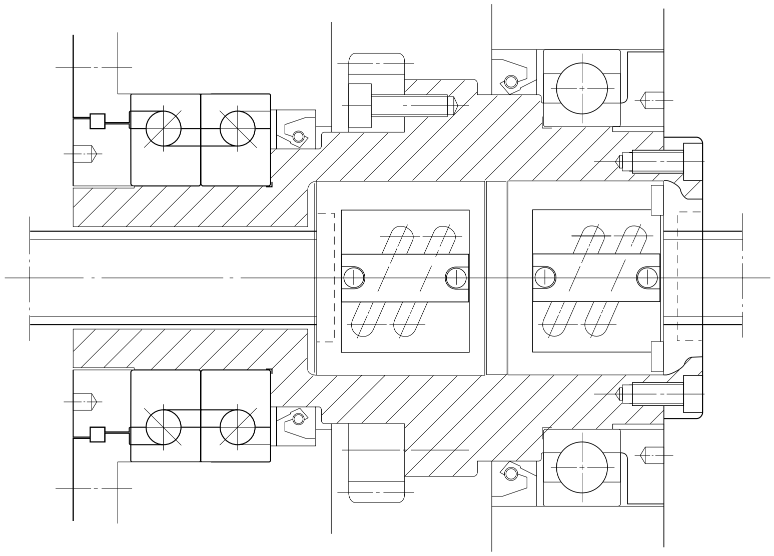
- Vorschubgeschwindigkeit 1 mm/min - 40 m/min
- Geringe Entwicklung von Geräusch, Vibration und Wärme auch bei hohen Geschwindigkeiten
- Hohe axiale Steifigkeit und Positionierbarkeit

Leadscrew mounting methods

- **Mounting tolerances are typically:**
  - **Perpendicularity to 500 μrad (target 200 μrad).**
  - **Lateral misalignment of less than 25 μm (1000 μin).**
- **There are two primary methods for mounting a leadscrew with a carriage:**
  - **Non-rotating shaft and rotating nut.**
  - **Rotating shaft and non-rotating nut.**
- **The former is most often used when:**
  - **The critical speed limit of a long shaft is below the maximum desired speed.**
  - **There is a desire to minimize rotating inertia.**
  - **Remember, a rotated nut will have an effective diameter of about twice that of the shaft:**

$$J = \frac{\pi D^4 L}{32}$$

- **Mounting method for a rotating nut** (Courtesy of NSK Corp.)**:**

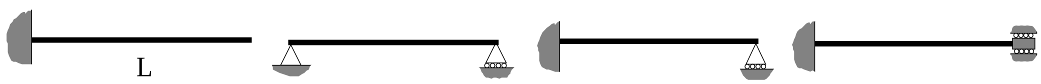


- **Some manufacturers now offer leadscrew nuts with integral rotary motion support bearings.**
- **The nut might be rotated by:**
    - **Gears.**
    - **A timing belt.**
    - **A direct drive motor that uses the OD of the nut as the shaft for the motor rotor.**

Critical shaft speed

- **If the rotational frequency equals the shaft bending natural frequency, an unstable condition known as *shaft whip* occurs.**

$$\omega_n = k^2 \sqrt{\frac{EI}{A\rho L^4}} \qquad F_{buckle} = \frac{cEI}{L^2}$$



	Cantilevered		Simply supported		Fixed-simply supported		Fixed-fixed	
n	k	c	k	c	k	c	k	c
1	1.875	2.47	3.142	9.87	3.927	20.2	4.730	39.5
2	4.694		6.283		7.069		7.853	
3	7.855		9.425		10.210		10.996	
4	10.996		12.566		13.352		14.137	
n	$(2n-1)\pi/2$		$n\pi$		$(4n+1)\pi/4$		$(2n+1)\pi/2$	

- **It is important to keep the rotational frequency below that of the first bending natural frequency of the shaft.**
- **Particular care should be taken when designing leadscrews where the ratio of shaft length to diameter exceeds 70.**
- **When evaluating the cross sectional area and moment of inertia:**
  - **Use the values of the leadscrew radius that give the most conservative result.**
- **To get a true built-into-the-wall (zero slope) effect, the bearings must have an effective center-to-center distance of 3-5 times the shaft diameter!**

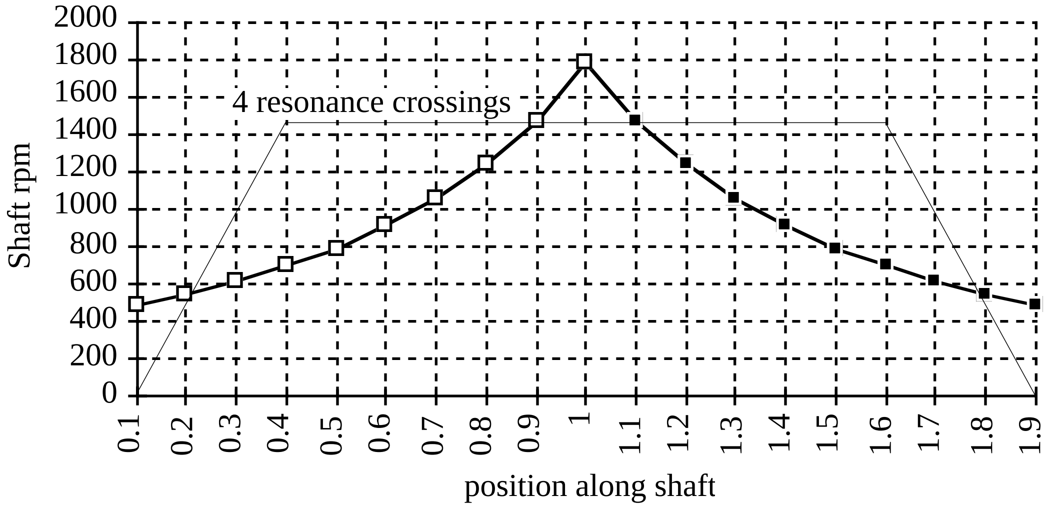
- **In addition to the critical shaft speed and buckling loads, the limiting dN value for the screw must be observed.**

Rolling Element Screws	Limiting value of n x d_0
Ball Screw with internal recirculation	70,000 (higher for specials)
Ball Screw with external recirculation	90,000 (higher for specials)
Planetary Roller Screw	140,000 (higher for specials)
Recirculating Roller Screw	20,000

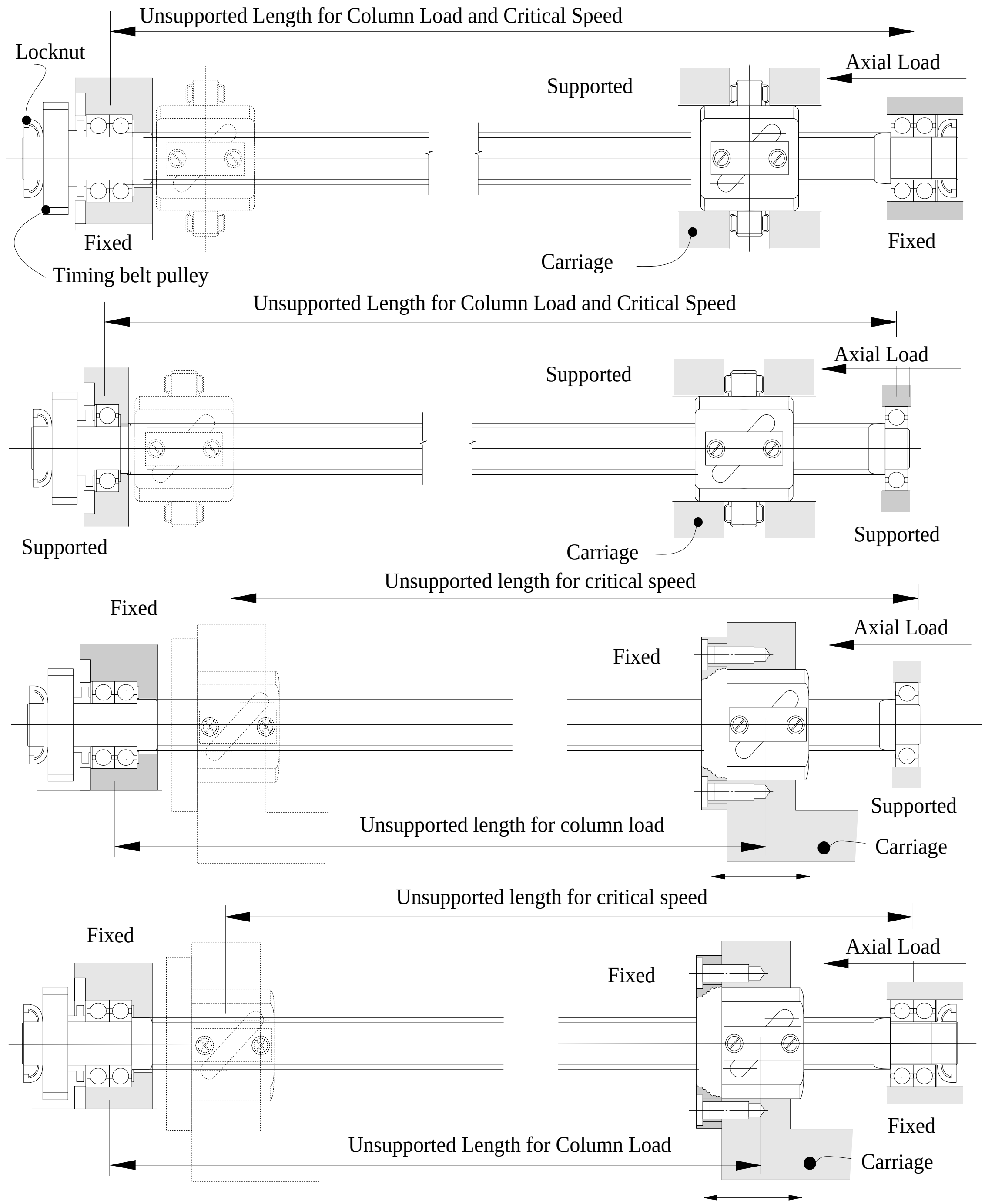
Controller design soluton to shaft whip

- **Instead of a classic trapazoidal velocity profile:**
  - **A triangular velocity profile be used to reach maximum traverse speed without exciting the leadscrew shaft.**
- **Example: Instantaneous critical speed of a simply supported leadscrew shaft as a function of nut position (saturation speed is assumed to be 2000 rpm):**

Shaft length (m)	**2**
Outer diameter (mm)	**15.0**
Root diameter (mm)	**10.00**
Maximum motor rpm	**2000**



- **Examples of leadscrew mounting methods** (Courtesy of NSK Corp.)**:**



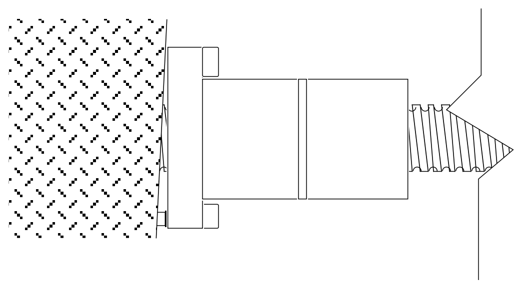
- **When a fixed-fixed mounting is used it allows the leadscrew to be stretched for thermal lead error compensation.**
    - **Stretch the ballscrew X µm, and for every µm of thermal growth, a µm of spring tension is released.**
    - **This creates a balanced system, so the lead remains constant independent of temperature.**
    - **The ballscrew has to be stretched enough so the preload is never released by thermal growth.**
- **Stretching increases the load on the support bearings which generates more heat.**
    - **It is one more item which must be controlled during the assembly operation.**
    - **A 50 mm D ballscrew 2 m long requires a stretching force of 41kN to preload for 10 C^{o} temperature rise.**
- **Stretching the leadscrew prevents the leadscrew from being solely in compression under some loads.**
    - **This increases the buckling load.**
        - **The stretching must be done without the use of compliant washers**
    - **This also slightly increases the natural frequency by 10-20%.**
- **Stretching the screw can be a good idea if a rotary encoder is used on the motor.**
- **If linear scales are used, stretching the screw is not required.**

- **To resist high axial loads and maximize stiffness, support ballscrews with special bearings:**

- **Superprecision angular contact bearings with 60^{o} contact angle for ballscrew supports.** (Courtesy of The Torrington Company.)

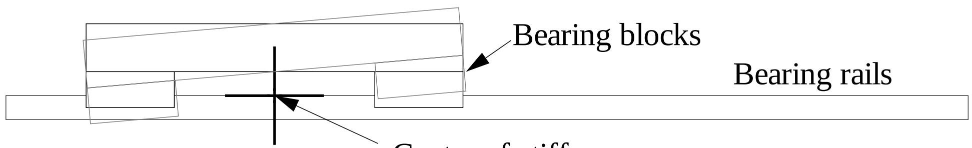
Bearing Number d	Bore +0 to - mm	Tol. D µm	OD +0 to - mm	Tol. C µm	Width Radius mm	Fillet mm	Mass kg	Preload (N)	K_{axial} torque (N/µm)	Drag thrust (N-m)	Dynamic thrust (kN)	Max. (kN)
Duplex (add suffix DU)												
MM9306WI-2H	20.000	3.8	47.0	5	31.75	0.8	0.272	3340	750	0.34	25.0	25.0
MM9308WI-2H	23.838	3.8	62.0	5	31.75	0.8	0.527	4450	1100	0.45	29.0	35.5
MM9310WI-2H	38.100	5	72.0	5	31.75	0.8	0.590	6670	1300	0.45	36.0	45.5
MM9311WI-3H	44.475	5	76.2	5	31.75	0.8	0.590	6670	1390	0.56	38.0	51.0
MM9313WI-5H	57.150	5	90.0	8	31.75	0.8	0.859	7780	1655	0.79	40.5	61.0
MM9316WI-3H	76.200	5	110	8	31.75	0.8	0.980	10000	2100	1.02	44.0	76.5
MM9321WI-3H	101.600	6.4	145	8	44.45	1.0	2.16	13340	2455	1.36	85.0	150
MM9326WI-6H	127.000	7.5	180	10	44.45	1.0	3.86	17790	3150	2.27	91.6	186
Quadruplex (add suffix QU)												
MM9306WI-2H	20.000	3.8	47.0	5	63.50	0.8	0.545	6670	1500	0.68	40.3	50.0
MM9308WI-2H	23.838	3.8	62.0	5	63.50	0.8	1.053	8900	2200	0.90	47.5	71.0
MM9310WI-2H	38.100	5	72.0	5	63.50	0.8	1.180	13340	2600	0.90	58.5	91.0
MM9311WI-3H	44.475	5	76.2	5	63.50	0.8	1.180	13340	2780	1.13	61.0	102
MM9313WI-5H	57.150	5	90.0	8	63.50	0.8	1.707	15570	3300	1.58	65.5	122
MM9316WI-3H	76.200	5	110	8	63.50	0.8	1.961	21000	4200	2.03	72.0	153
MM9321WI-3H	101.600	6.4	145	10	88.90	1.0	4.32	26700	4900	2.71	127.0	300
MM9326WI-6H	127.000	7.5	180	10	88.90	1.0	7.72	35580	6300	4.50	147.8	375

- **A precision ballscrew should be mounted with less than 20 μm radial and 500 μrad angular misalignment respectively.**
- **Regardless of the mounting method used, it is important to minimize the moment and radial force placed on the nut.**
- **Radial alignment can be achieved relatively easily.**
- **Angular misalignment is a more insidious problem because it imposes large radial loads on the nut which greatly increases wear.**



- **Angular alignment can be achieved by scraping.**
- **Compressive preloading creates a face-to-face mounting situation:**
  - **The ball loads will be lower and life will be longer than with a back-to-back preload.**
- **Seating the nut against a potting material (e.g., epoxy) is an alternative to scraping.**

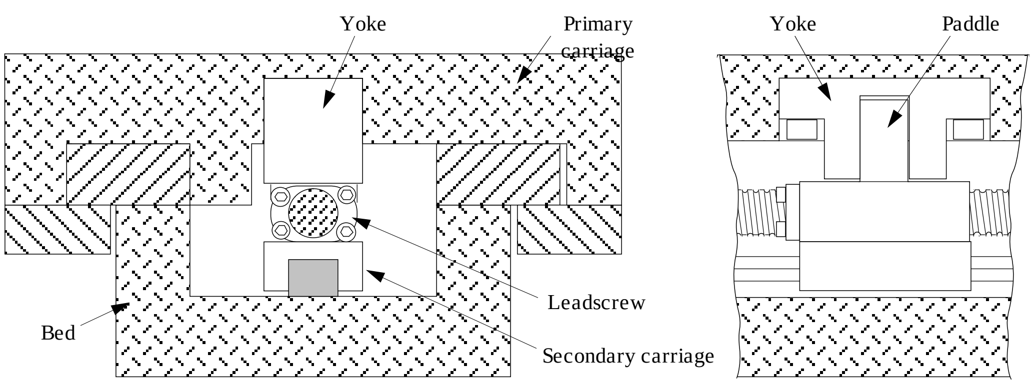
- **Since a carriage pitchs about its center of stiffness, to minimize radial displacement effects on the ballscrew:**
- **The ballscrew nut should be located at the system's center of stiffness:**



- **As the carriage pitches, if the ballscrew nut is at the center of stiffness, it will not be subject to a radial displacement.**
- **In most machines, the effect is so small, that this is not a significant problem.**

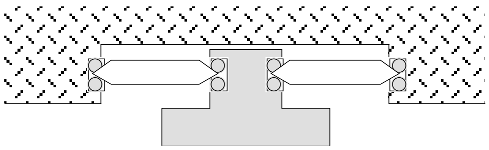
Coupling systems

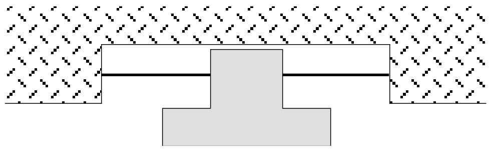
- **Prevents non-axial motion components of the actuator from causing carriage error motions:**
  - **Only one degree of freedom between the actuator and the bearing supported slide is restrained.**
  - **Filters out error motions by allowing members to slide or deflect in non-sensitive directions.**
- **An added benefit is they often help to reduce the amount of heat transferred between the actuator to the slide.**
- **Paddle coupling to isolate actuator error motions:**



- **The actuator is directly connected to a secondary carriage which has a "U" shaped yoke that straddles the carriage's paddle.**
- **For high stiffness and damping (machine tools):**
  - **Single entry fluid film thrust bearings, which cannot resist lateral or angular motions, center the paddle.**

- **For instruments:**
    - **Wire or coupling or wobble pin:**





- **The secondary carriage can also act as the anchor point for the moving end of a cable carrier.**
- **Disadvantages:**
    - **The need for an extra carriage and the bearing (coupling) between the paddle and the yoke.**
    - **If the actuator is a leadscrew, one still has to reckon with backlash and friction.**

Flexural couplings

- **Elastically accommodate error motions.**
- **Relatively stiff along the direction of the platten's axial motion.**
- **Axial stiffness is less than if the actuator coupled to the carriage rigidly or through a paddle type element.**
    - **A machine built with flexural couplings will have a slower response time.**
- **Still, the simplicity and economics of flexural couplings make them extremely useful.**
- **One must merely recognize their limitations and include them in system models before building a prototype.**

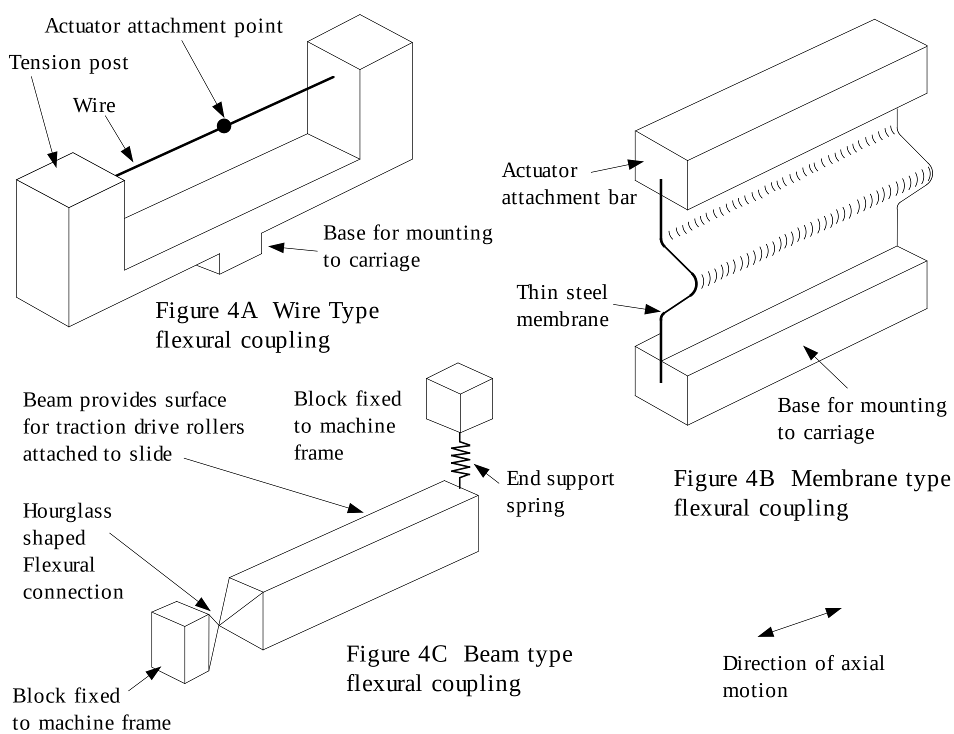


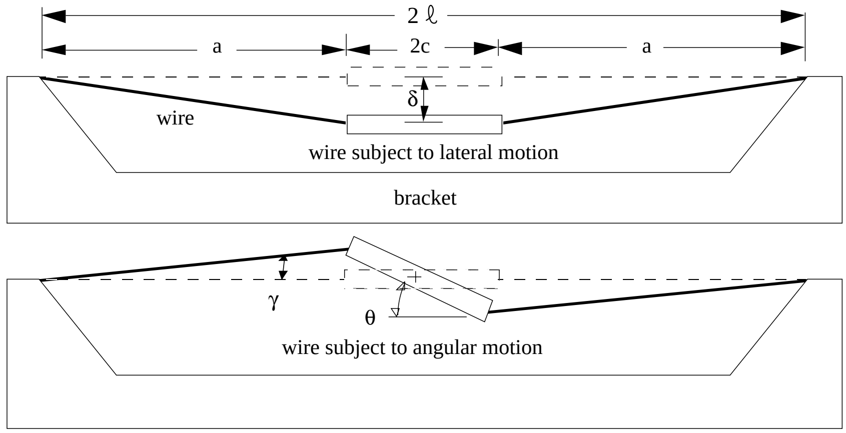
Figure 4A Wire Type flexural coupling

Figure 4B Membrane type flexural coupling

Figure 4C Beam type flexural coupling

Wire-type flexural coupling design

- **The axial stiffness of the wire coupling held at its midpoint is unaffected by the tension.**



- **For a wire cross sectional area A and wire length 2l:**
  - **The axial stiffness is K_{axial} = AE/a.**
- **As the wire is displaced laterally:**
  - **Forces are generated to resist this motion by the initial tension in the wire and the stretching of the wire.**
- **The change in tension caused by the wire stretching as it is deflected δ laterally is:**

$$\Delta T = EA\left\{\left(1 + \frac{\delta^2}{a^2}\right)^{1/2} - 1\right\}$$

Example: For an instrument platten driven by a steel wire coupling:

- **Wire tension = 10 N (2.25 lbf)**
- **Slide stiffness = 10^8 N/m (570,000 lbf/in)**
- **Length 2l = 11 cm, 2a = 10 cm (4")**
- **Wire diameter = 0.25 mm (0.010")**
- **$K_{lateral}$ = 402 N/m (2.288 lbf/in)**
- **Error motion = 100 µm (0.004")**
- **Resultant maximum lateral force = 0.0402 N (0.0090 lbf)**
- **Resultant slide lateral error motion = 4 Ångstroms**
- **K_{axial} = 203 KN/m (1,160 lbf/in)**

- **Even if the wire tension were increased by an order of magnitude.**
    - **The lateral error motion would only be 40 Ångstroms.**