Fasteners and Power Screws I

Introduction

- Screws and bolts are standard parts
- Details about thread geometry is given in the text book
- Different types of bolts and nuts are not interchanganable
- lead : the distance a screw advances in one full rotation
- pitch :distance between the corresponding points on adjacent threads

Figure 8–1

Terminology of screw threads. Sharp vee threads shown for clarity; the crests and roots are actually flattened or rounded during the forming operation.

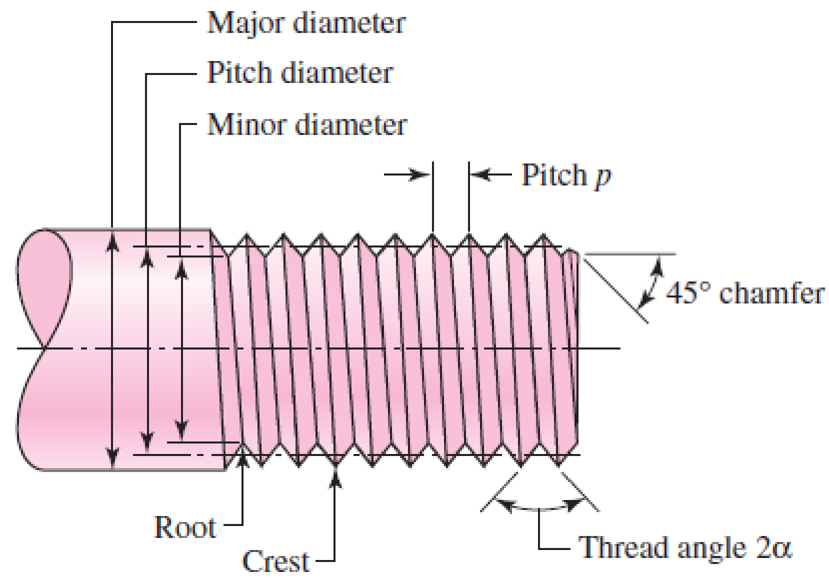


Figure 8–2

Basic profile for metric M and MJ threads.

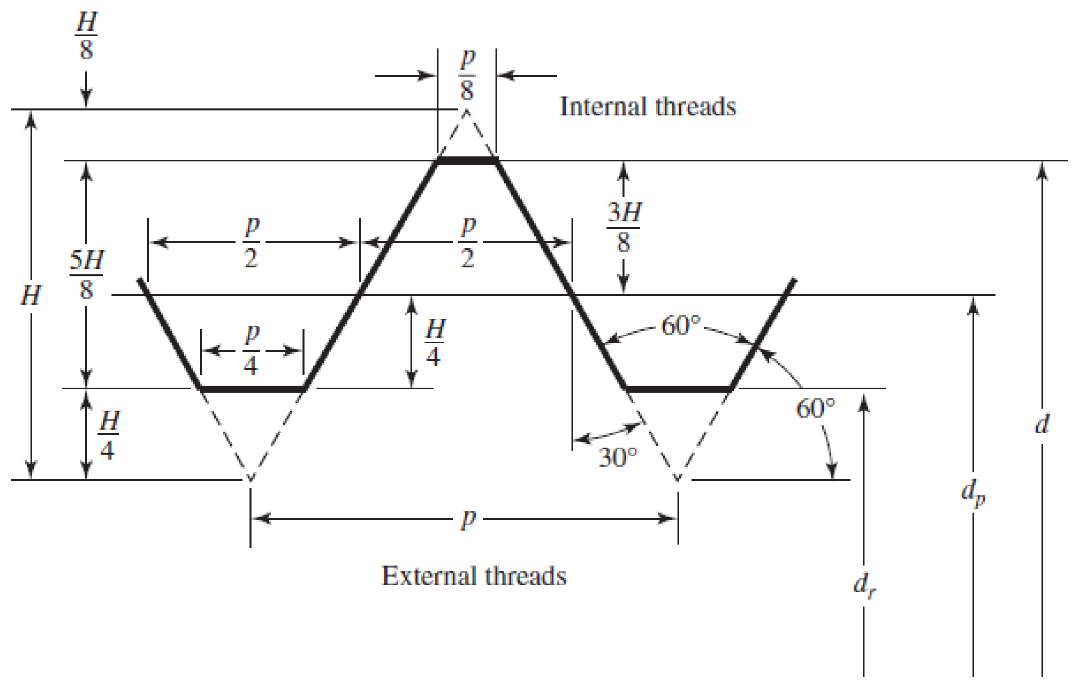
d = major diameter

d_r = minor diameter

d_p = pitch diameter

p = pitch

$H = \frac{\sqrt{3}}{2}p$



Introduction

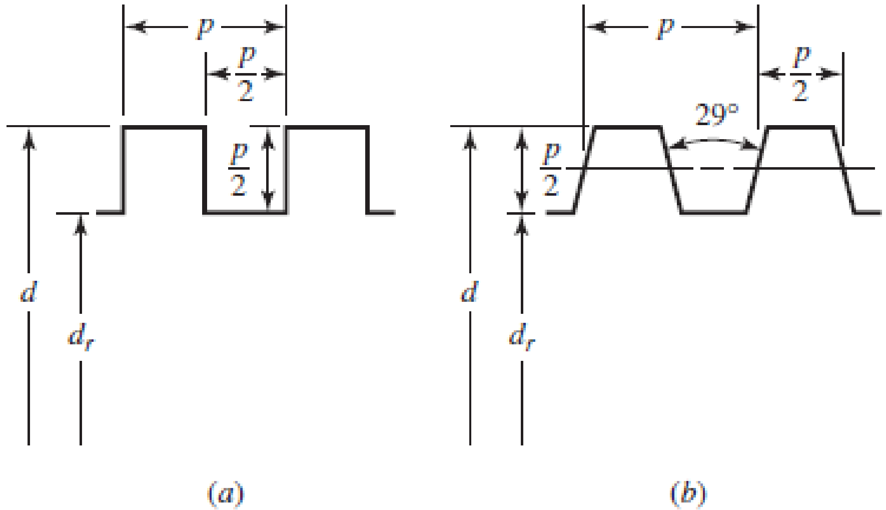
- Metric thread specification
- M12 x 1.75
- M : metric
- 12 : nominal size (major diameter)
- 1.75 : pitch
- Square and ACME (Trapezoidal) threads are used for power transmission

Introduction

- single thread : lead=pitch
- multiple thread (n) : lead=n x pitch
- For stress calculations of bolts we use "Tensile Stress Area" , A_t
- A_t is given in tables
- Threads are made according to right hand rule unless otherwise is noted.

Figure 8–3

(a) Square thread; (b) Acme thread.



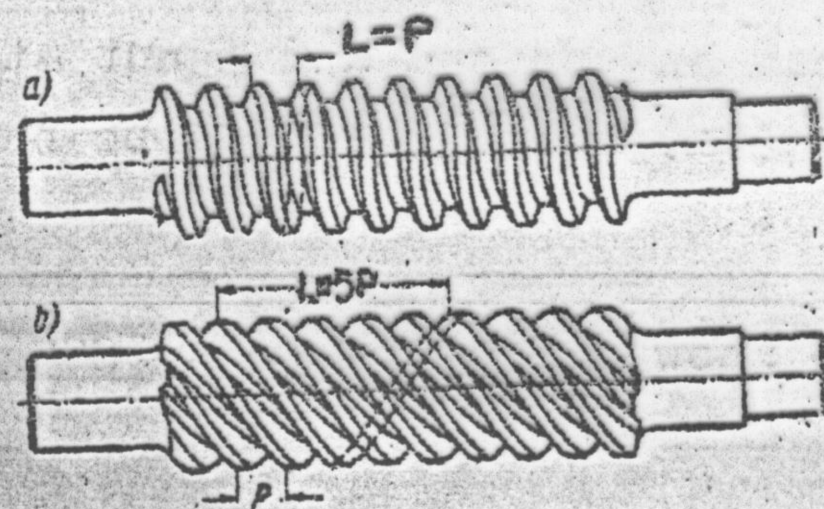


Fig. 63 a) Single start thread. b) Multiple start thread.

Table 8–1

Diameters and Areas of Coarse-Pitch and Fine-Pitch Metric Threads.*

Nominal Major Diameter d mm	Coarse-Pitch Series Pitch p mm	Coarse-Pitch Series Tensile-Stress Area A_t mm^2	Coarse-Pitch Series Minor-Diameter Area A_r mm^2	Fine-Pitch Series Pitch p mm	Fine-Pitch Series Tensile-Stress Area A_t mm^2	Fine-Pitch Series Minor-Diameter Area A_r mm^2
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884

Power Screws

- Power screws are used to change angular motion into linear motion
- They transmit power
- We want to know how much torque T is required to raise or lower the screw which is under the action of axial load F.
- If we develop the screw the thread (helix) will become an inclined plane.

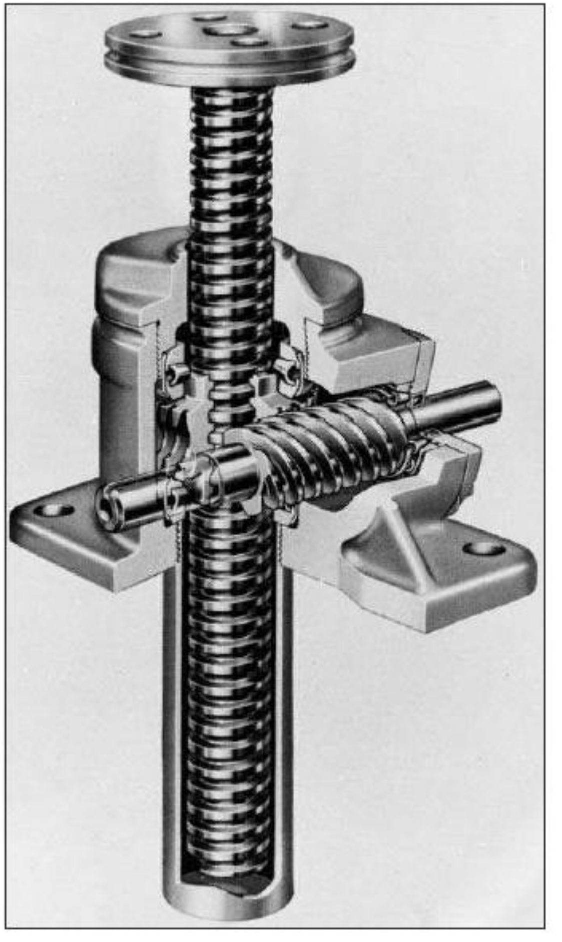
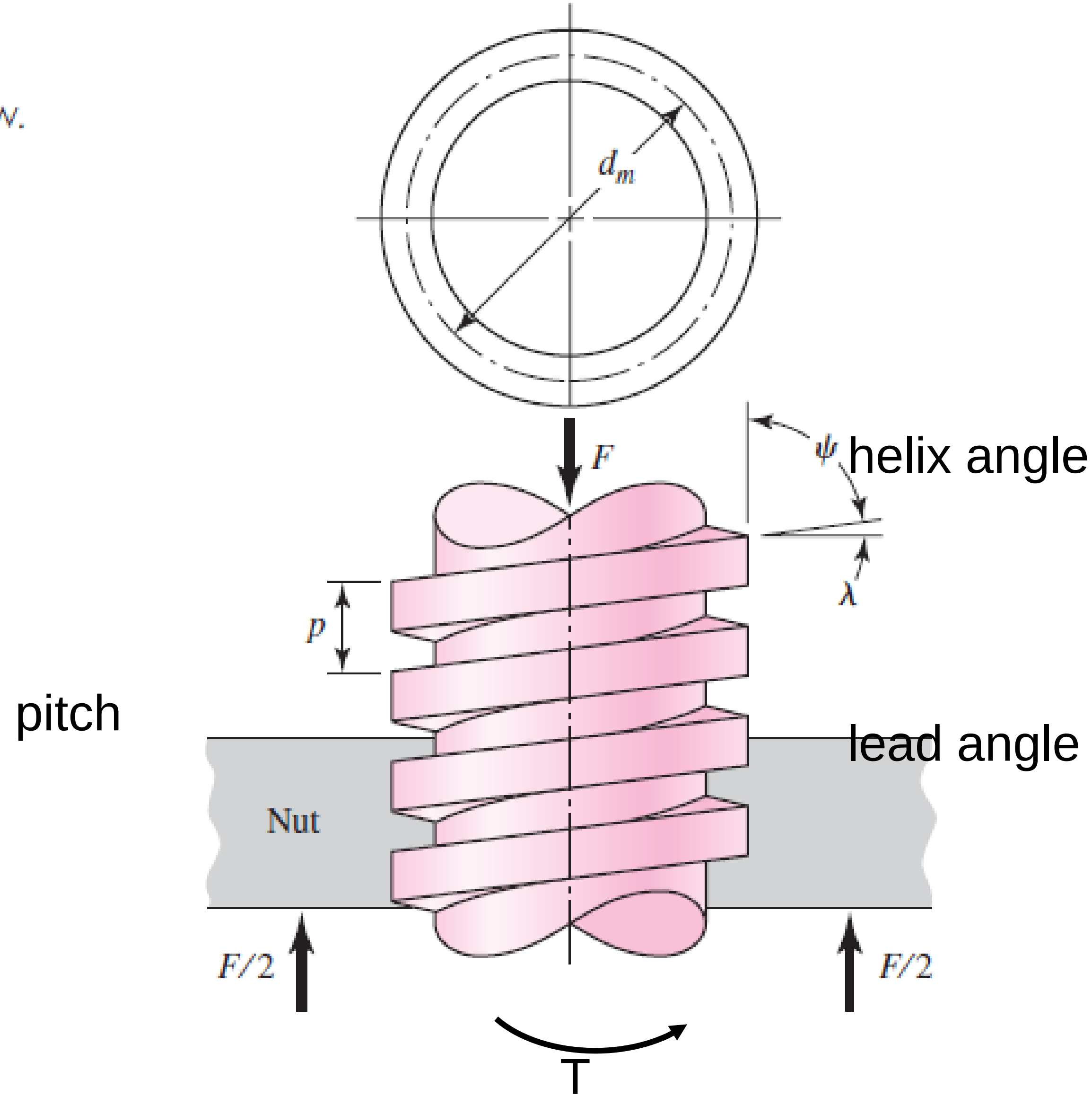
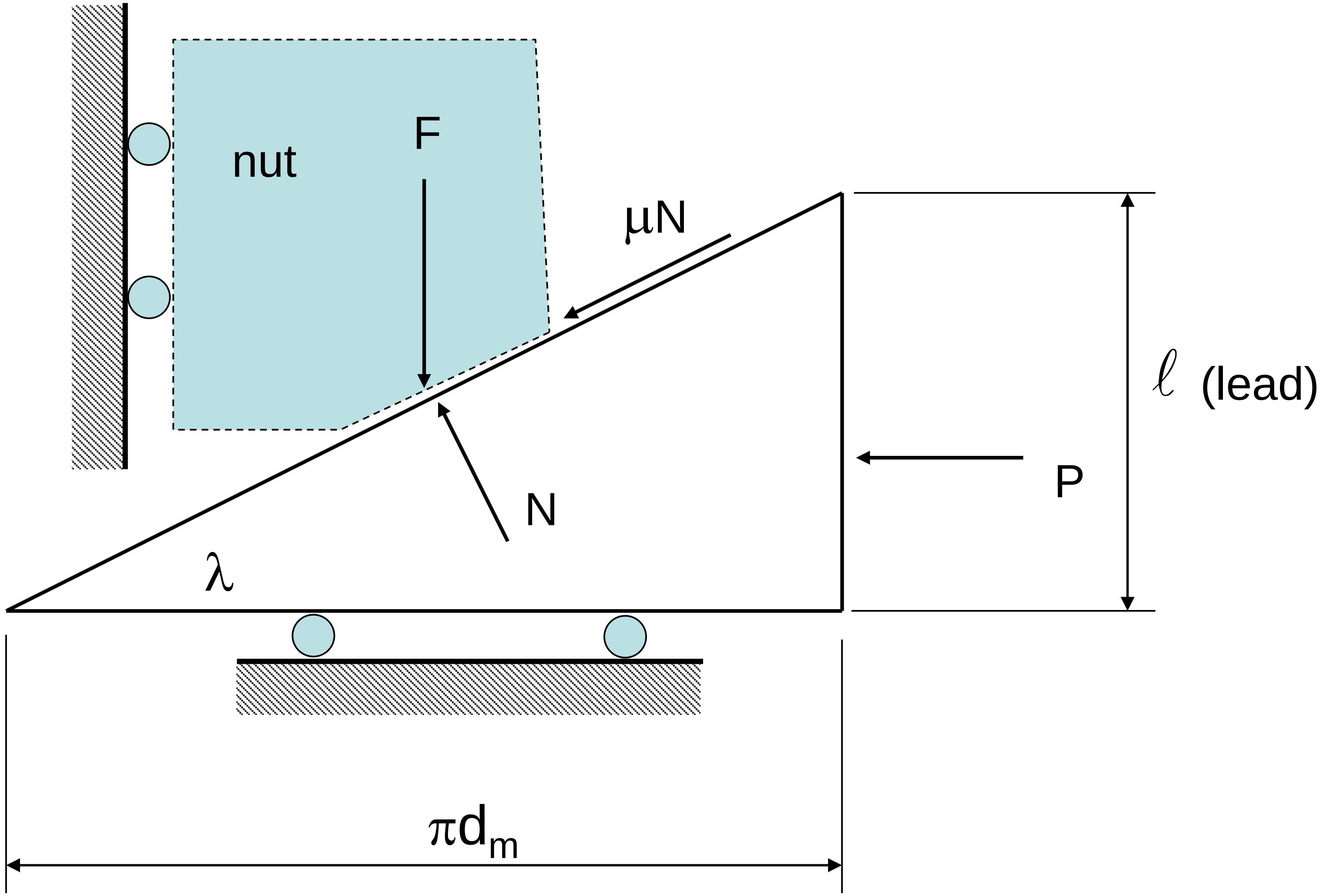


Figure 8–5

Portion of a power screw.

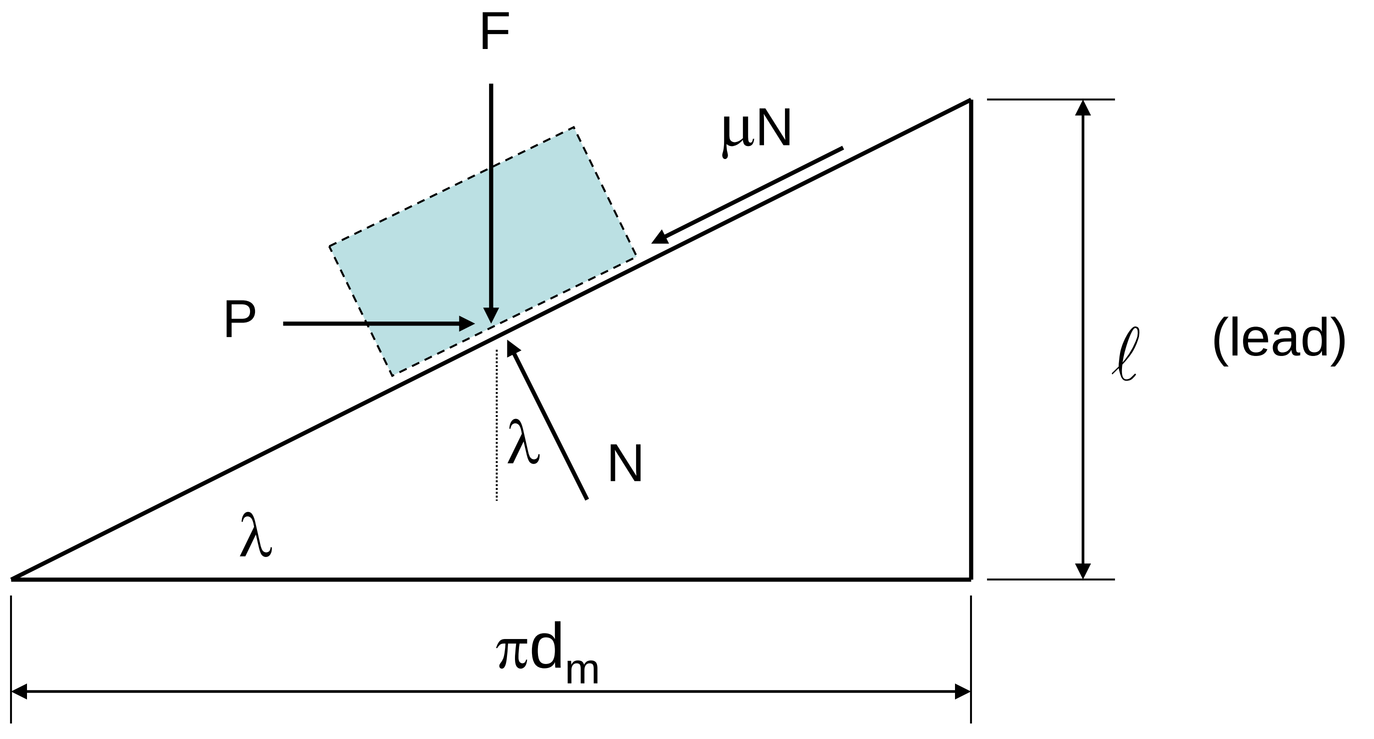




Relative motion of nut on the helical thread becomes similar to motion of a block on an inclined plane.

The problem can be approached like an inclined plane or wedge problem.

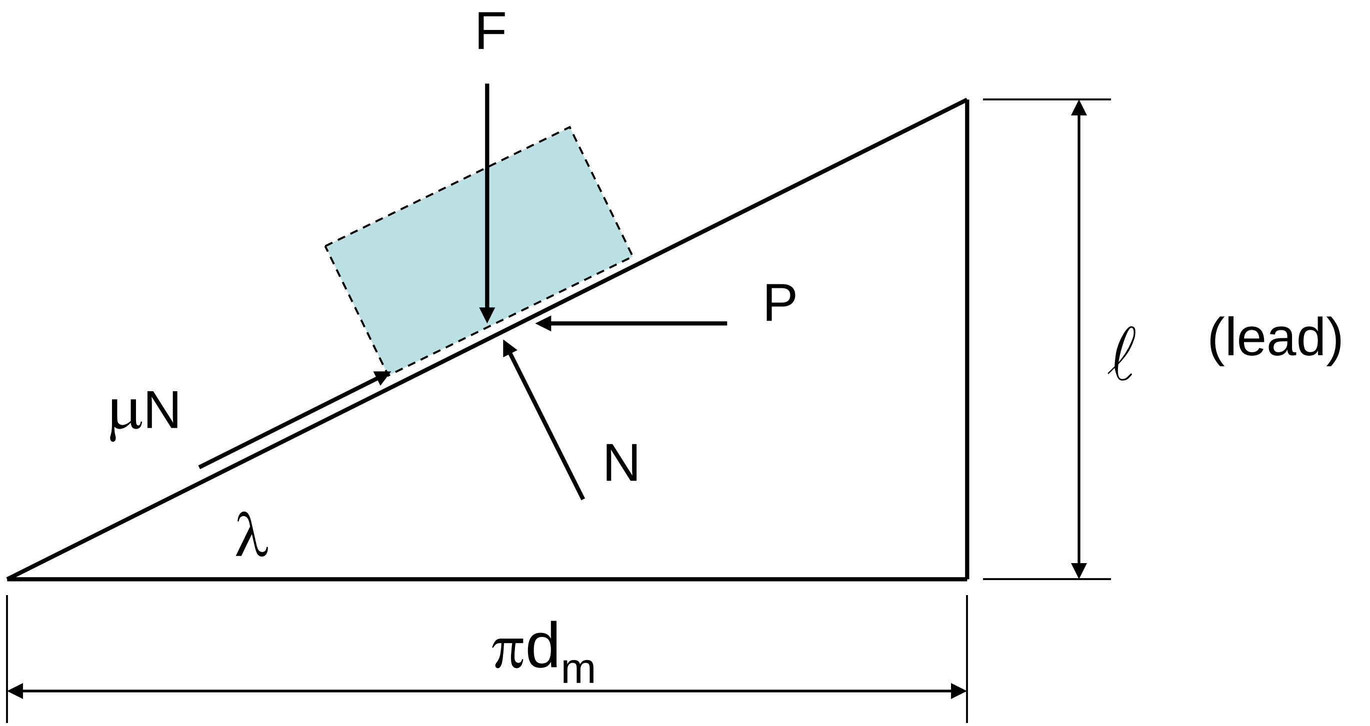
FBD for LIFTING the load :



$$\sum F_x = 0: \quad P - N\sin\lambda - \mu N\cos\lambda = 0$$

$$\sum F_y = 0: \quad F + \mu N\sin\lambda - N\cos\lambda = 0$$

FBD for LOWERING the load :



$$\sum F_x = 0: \quad -P - N\sin\lambda + \mu N\cos\lambda = 0$$

$$\sum F_y = 0: \quad F - \mu N\sin\lambda - N\cos\lambda = 0$$

we can eliminate N and solve P in terms of F

For raising the load $P = \dfrac{F(\sin\lambda + \mu\cos\lambda)}{\cos\lambda - \mu\sin\lambda} = \dfrac{F(\tan\lambda + \mu)}{1 - \mu\tan\lambda}$

For lowering the load $P = \dfrac{F(\mu\cos\lambda - \sin\lambda)}{\cos\lambda + \mu\sin\lambda} = \dfrac{F(\mu - \tan\lambda)}{1 + \mu\tan\lambda}$

Note that $\tan\lambda = \dfrac{\ell}{\pi d_m}$ and $T = P \times \dfrac{d_m}{2}$

Then

For raising the load:

$$T = \frac{Fd_m}{2}\left(\frac{\ell + \pi\mu d_m}{\pi d_m - \mu\ell}\right) \quad (1)$$

For lowering the load:

$$T = \frac{Fd_m}{2}\left(\frac{\pi\mu d_m - \ell}{\pi d_m + \mu\ell}\right) \quad (2)$$

- If lead is large or friction is low, the load F may lower itself without external torque
- Eq. (2) gives a negative value in this case
- If eq. (2) gives a positive value the screw is said to be "self locking"

$$\pi \mu d_m > \ell$$

$$\mu > \frac{\ell}{\pi d_m} \quad \text{and} \quad \mu > \tan\lambda$$

is self locking condition

- Efficiency of a power screw, e, is defined as ; $e=T_0/T$

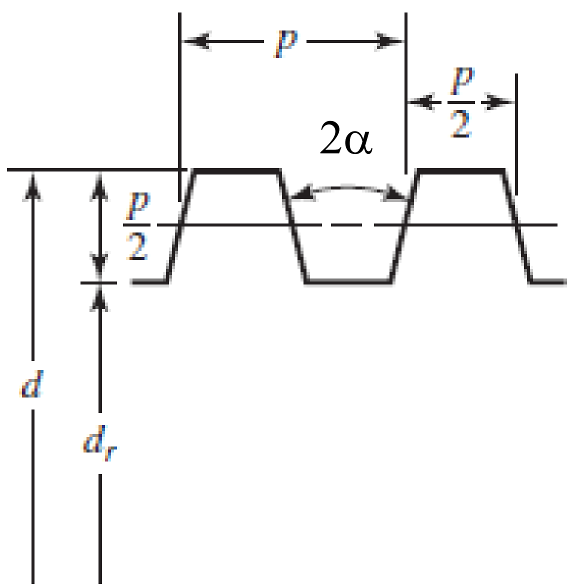
T_0 is obtained by letting μ=0 in eq. (1) $\quad T_0 = \dfrac{F\ell}{2\pi}$

$$e = \frac{T_0}{T} = \frac{F\ell}{2\pi T} = \tan\lambda\left(\frac{1-\mu\tan\lambda}{\tan\lambda+\mu}\right)$$

Typical values of coefficient of friction are between
0.06 (bronze on cast iron with lubrication) and 0.25 (dry, steel on steel)

Typical values of lead angle are between 1 to 15 degrees

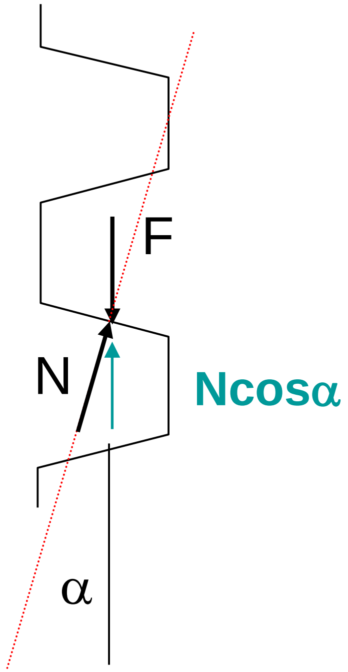
ACME Threads



Typical value of α is 14.5 degrees

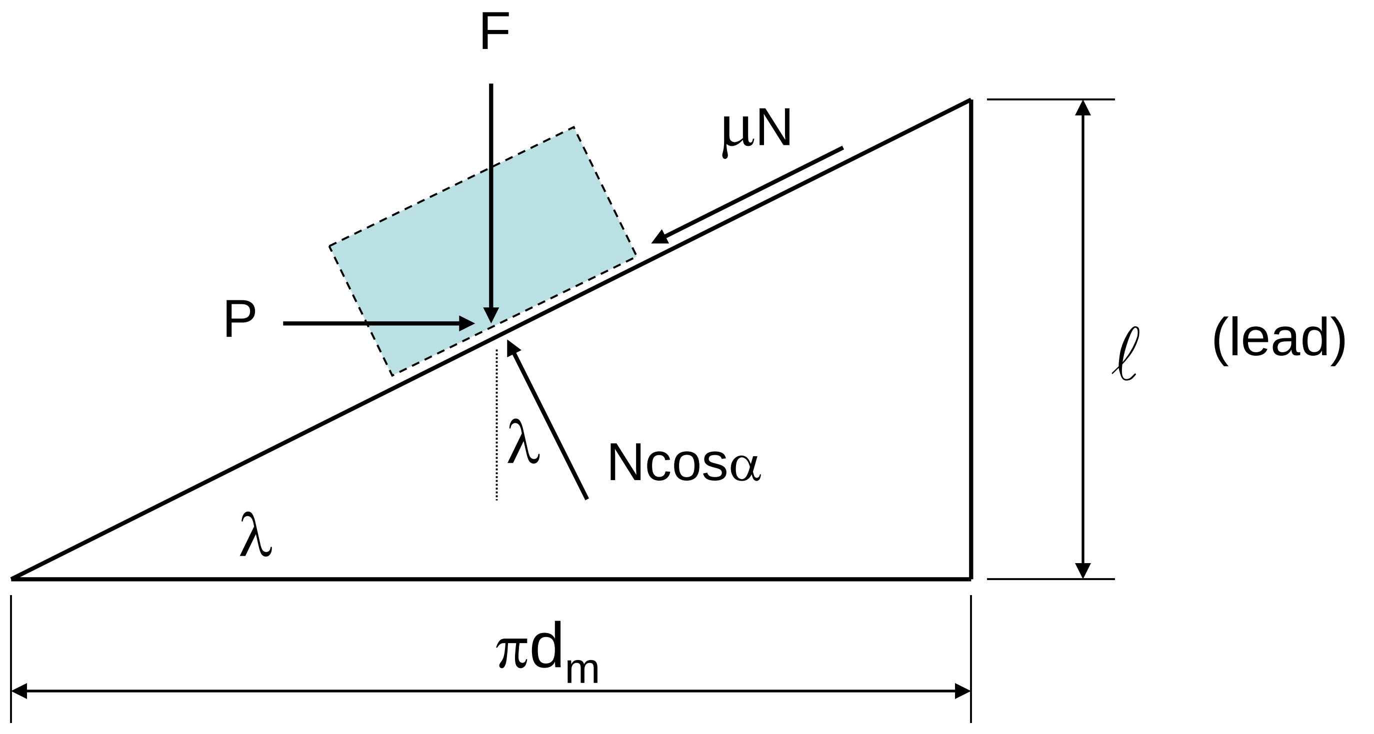
•Because of α, the force analysis needs to be modified.
•Since normal force N is perpendicular to thread surface it is no longer in the plane of load F

By recognizing this and replacing normal force in equilibrium equations accordingly, an approximate analysis valid for small λ can be done.



We note that normal force can be replaced with $N\cos\alpha$ but the friction force remains proportional to N.

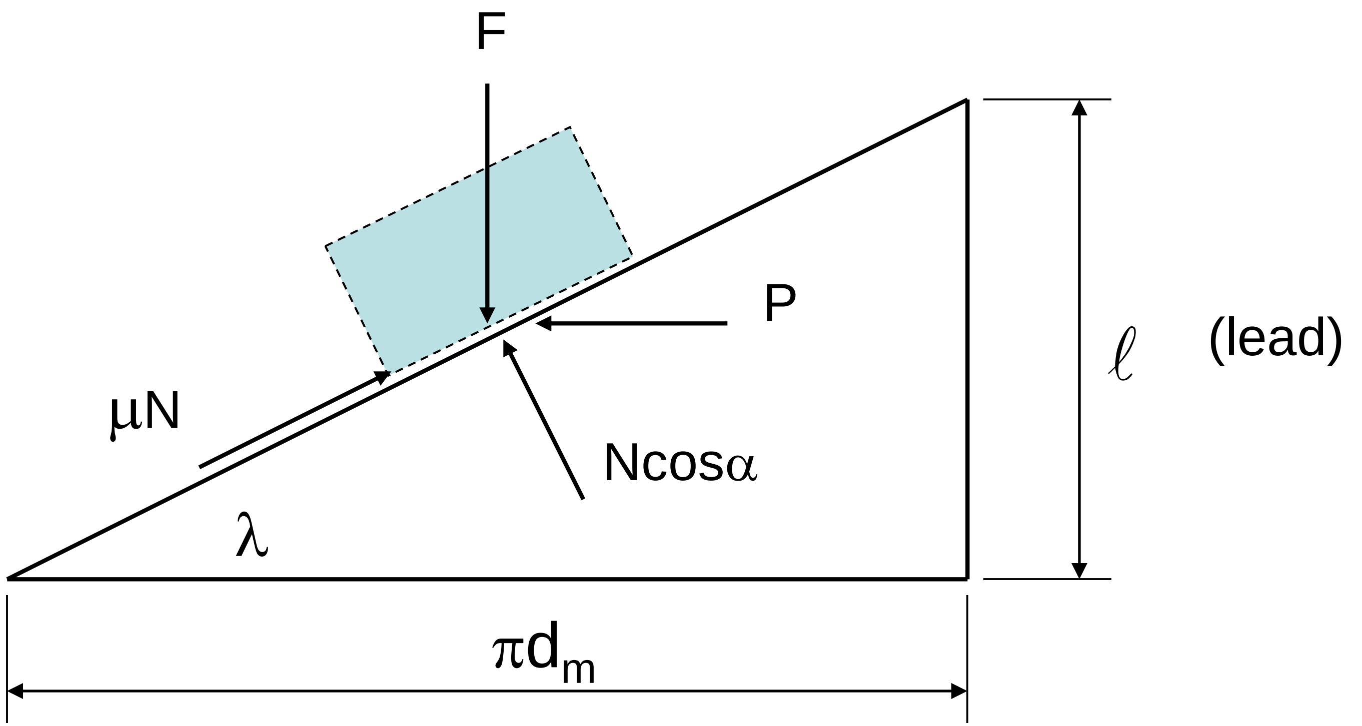
ACME thread FBD for LIFTING the load :



$$\sum F_x = 0: \quad P - N\cos\alpha\sin\lambda - \mu N\cos\lambda = 0$$

$$\sum F_y = 0: \quad F + \mu N\sin\lambda - N\cos\alpha\cos\lambda = 0$$

ACME thread FBD for LOWERING the load :



$$\sum F_x = 0: \; -P - N\cos\alpha\sin\lambda + \mu N\cos\lambda = 0$$

$$\sum F_y = 0: \; F - \mu N\sin\lambda - N\cos\alpha\cos\lambda = 0$$

we can eliminate N and solve P in terms of F again.

Noting that $\tan\lambda = \dfrac{\ell}{\pi d_m}$ and $T = P \times \dfrac{d_m}{2}$

For raising the load:

$$T = \frac{Fd_m}{2}\left(\frac{\ell + \pi\mu d_m \sec\alpha}{\pi d_m - \mu\ell\sec\alpha}\right) \quad (3)$$

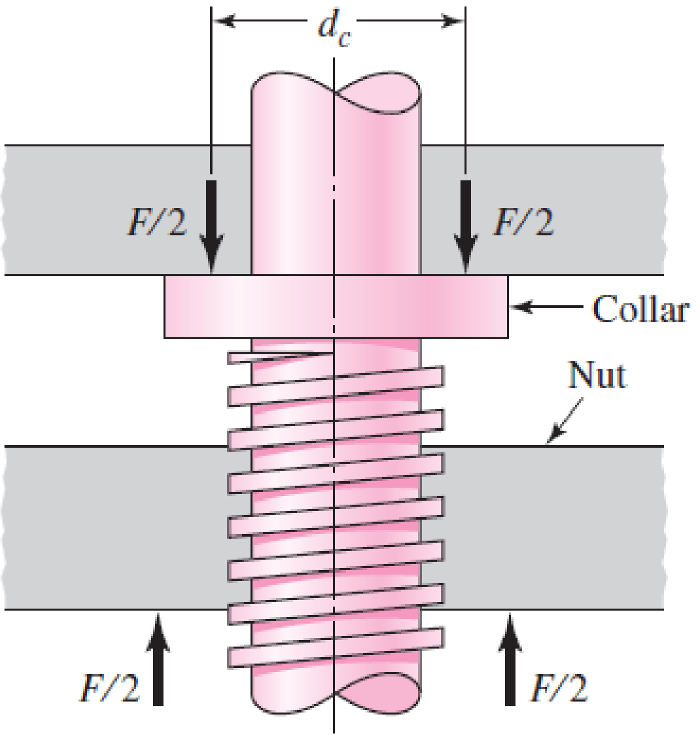
For lowering the load:

$$T = \frac{Fd_m}{2}\left(\frac{\pi\mu d_m \sec\alpha - \ell}{\pi d_m + \mu\ell\sec\alpha}\right) \quad (4)$$

These expressions are approximate
since the effect of lead angle is neglected.
ACME threads are less efficient since $\mu\sec\alpha > \mu$
and more likely to be self locking
(as if friction has increased)

Collar Torque

- When the screw is loaded axially, a thrust or collar bearing must be employed between the rotating and stationary members in order to take the axial component.



(b)

$$T_c = \frac{F\mu_c d_c}{2}$$

For large collars this equation may not be accurate

$$T_t = T + T_c$$

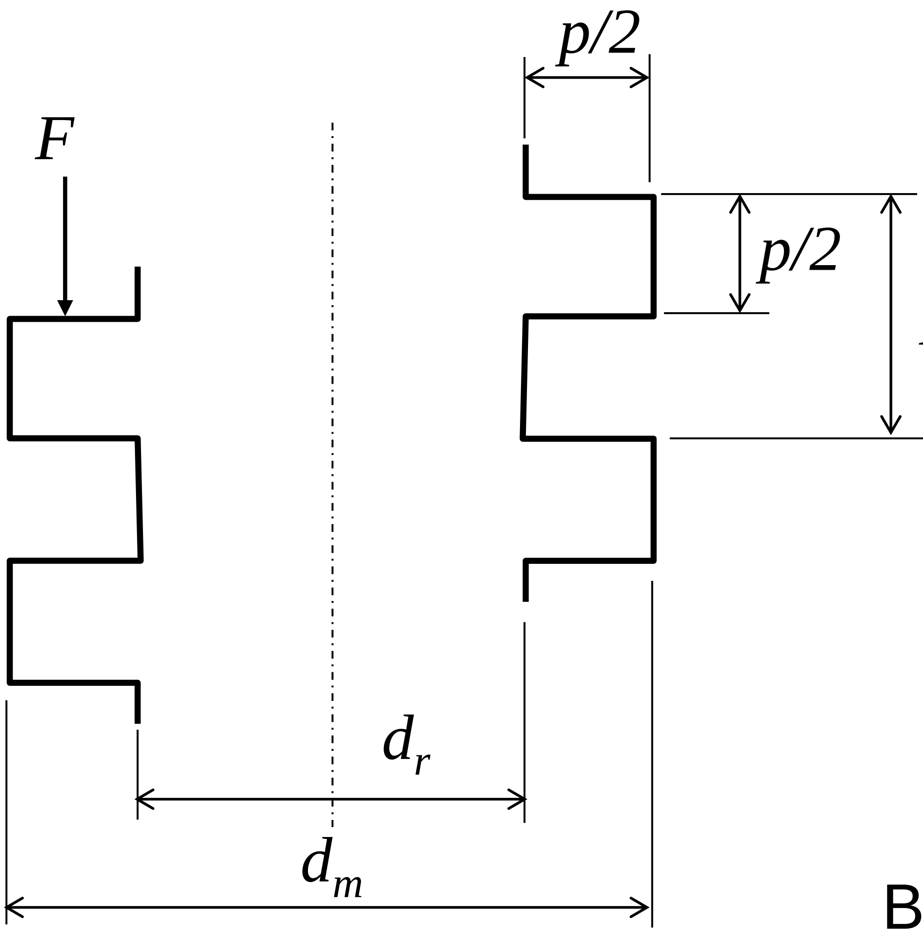
Thread Stresses

Assumption: Average screw thread shearing stress can be obtained by assuming that the load is uniformly distributed over the nut height and threads would shear of on the root diameter for the screw and on the major diameter for the nut.

Body Stresses

Buckling

$$\tau = \frac{16T}{\pi d_r^{\ 3}}$$ **(torsional shear)** $$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^{\ 2}}$$ **(axial stress)** $$P_{cr} = \frac{\pi^2 EI}{L^2}$$



Bearing Stress

$$\sigma_B = -\frac{F}{\pi d_m n(p/2)}$$

$$\sigma_B = -\frac{2F}{\pi d_m n p}$$

$$\sigma_B = -\frac{2F}{\pi d_m h}$$

n: number of threads

h:nut height

Bending Stress at thread root

$$M = \frac{Fp}{4} \qquad \sigma = \frac{Mc}{I} = \frac{(Fp/4)(p/4)}{\left(\pi d_r n (p/2)^3\right)/12} = \frac{6F}{\pi d_r n p} = \frac{6F}{\pi d_r h}$$

For nut we take major diameter d.

Shear Stress at the thread root

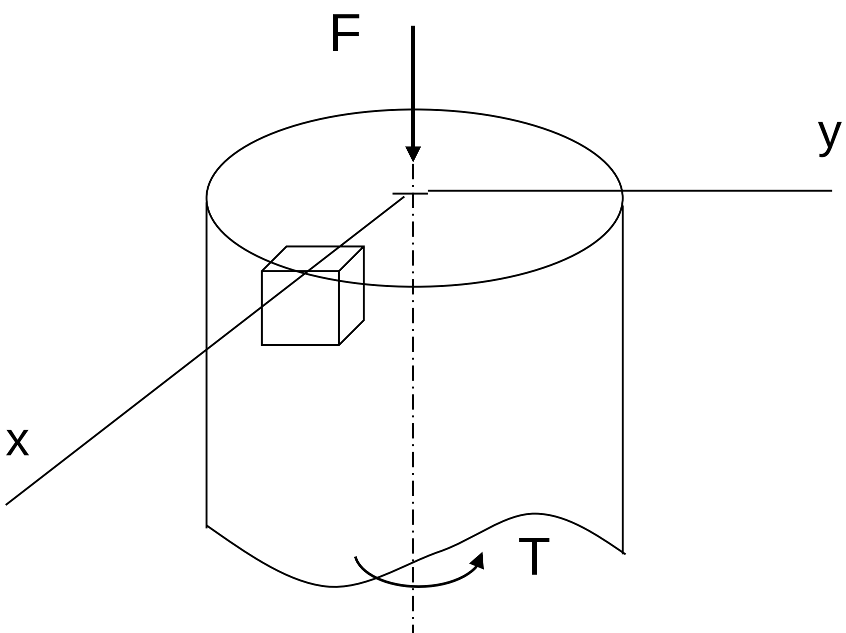
Maximum Shear Stress is in the middle of the thread root

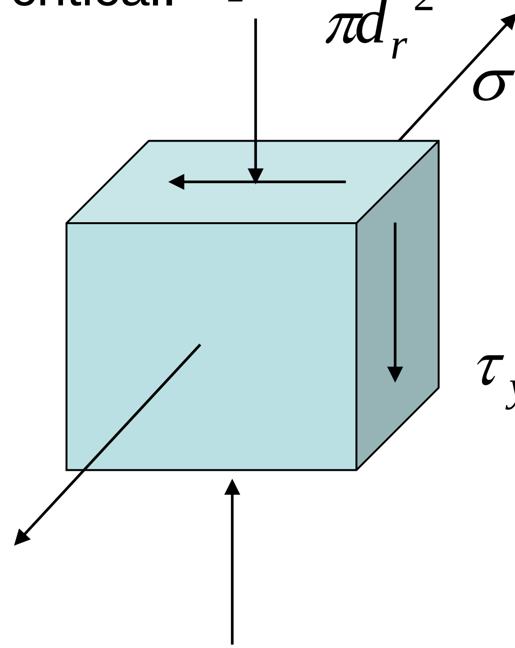
$$\tau_{ave} = \frac{F}{A} \qquad \tau_{\max} = \frac{3}{2}\frac{F}{A} \qquad \tau_{\max} = \frac{3}{2}\frac{F}{\pi d_r (p/2) n} = \frac{3F}{\pi d_r pn} = \frac{3F}{\pi d_r h}$$

For nut we take major diameter d.

If materials are the same bolt is more critical.

$$\sigma_z = \frac{4F}{\pi d_r^{\,2}}$$





$$\sigma_x = \frac{6F}{\pi d_r np}$$

$$\tau_{yz} = \frac{16T}{\pi d_r^{\,3}}$$

Remarks

- Note that thread stress calculations are rather approximate.
- More detailed studies show that first thread takes most of the load, second, third atc. take smaller portions.
- First 38%, second 25%, third 18% etc.
- Seventh and beyond take almost no load.