CHAPTER 01 · BOLTS IN ALUMINIUM

A little thread.
A surprisingly large force.

Follow one M8 bolt from a hole in a block to a loaded connection. Each equation answers a physical question; each number has somewhere to come from.

ABOUT 12 MINUTES · NO MECHANICS BACKGROUND REQUIRED
FIXED EXAMPLE: M8 × 10 mm · 5754-H111 · CLASS 8.8

Put a block of aluminium on the bench. Drill and tap a hole. Screw in an M8 steel bolt until 10 mm of complete thread is engaged. Now imagine gripping the bolt head and pulling straight upwards.

At first, nothing visible happens. The steel stretches a little, the aluminium deforms a little, and the thread faces press against one another. Increase the pull far enough and something must give. Will the bolt break? Will the aluminium threads strip? And how far below that point should we stay?

We will build a simple model to answer those questions. Keep this one example in mind as you read. Its numbers stay fixed throughout the chapter, even when you change settings in the lab below.

BOLT M8, class 8.8 steel	THREAD PITCH 1.25 mm
ENGAGED LENGTH 10 mm	ALUMINIUM 5754-H111

M8 tells us the nominal outside diameter is 8 mm. **Pitch** is the distance from one thread crest to the next. With a 1.25 mm pitch, one complete turn advances the bolt by 1.25 mm. Our 10 mm engagement therefore spans eight turns.

A FIRST, GENTLE CALCULATION

Engaged turns = 10 mm ÷ 1.25 mm = **8**

Eight turns is useful for picturing the connection. It does not mean each turn takes exactly one-eighth of the load. Real thread loading is uneven. Our model will estimate the connection as a whole.

NOTE Engagement is the length where complete bolt and hole threads overlap. A 10 mm drilled hole usually gives less than 10 mm of full engagement because the drill point, tap lead and incomplete end threads use some space.

02 / GEOMETRY

Find the metal that has to give way

To pull the bolt out without breaking the steel, you would have to shear away the aluminium teeth holding it. Imagine a thin cylindrical surface running through the engaged threads. Unroll that surface and it becomes a rectangle.

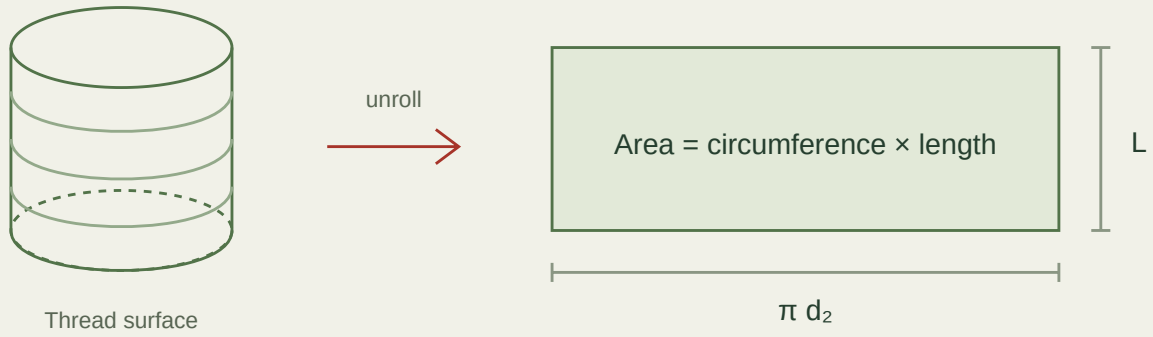


Figure 1. This is the geometry behind the area calculation. The actual resisting metal occupies only part of this imagined cylindrical surface.

The rectangle is one circumference wide and one engagement length high. Circumference is π times diameter. We use the *pitch diameter*, d_2, roughly halfway through the thread teeth, rather than the full outside diameter.

FROM NOMINAL DIAMETER TO PITCH DIAMETER

$$d_2 = d - 0.649519 \times p$$

$$= 8 - 0.649519 \times 1.25 \approx \mathbf{7.19\ mm}$$

d is nominal diameter; p is pitch. The decimal constant comes from the basic 60° metric thread geometry. It is not a material property or a safety factor.

The whole cylindrical surface is not solid aluminium resisting shear. NASA's simplified tapped-hole model uses one-third of it as an effective resisting area. The factor accounts empirically for the thread geometry and imperfect mating. [1]

EFFECTIVE THREAD SHEAR AREA

$$A_{effective} = \frac{\pi \times d_2 \times L}{3}$$

$$= \frac{\pi \times 7.19 \times 10}{3} \approx \mathbf{75.3\ mm^2}$$

We now have an area. The next question is how much force each square millimetre can resist.

03 / MATERIAL

Give that area a strength

A square millimetre of strong material can carry more force than the same area of weak material. This is what a stress value describes: force per unit area. In these calculations, **1 MPa means 1 newton per square millimetre**.

> Force capacity = resisting area × material strength.

For our 5754-H111 example, the viewer assumes a tensile strength R_m of 190 MPa. But the aluminium teeth are being sheared off, so we need a *shear* strength. The viewer estimates it as 60% of the tensile strength.

ESTIMATE ULTIMATE SHEAR STRENGTH, THEN MULTIPLY

$\tau_{ultimate} \approx 0.60 \times R_m = 0.60 \times 190 = 114\ N/mm^2$

$F_{strip} \approx 75.3\ mm^2 \times 114\ N/mm^2 \approx \mathbf{8{,}580\ N}$

$= \mathbf{8.58\ kN}$

τ (tau) means shear stress. F means force. A kilonewton is 1,000 newtons. Calculations use unrounded values internally.

Notice how the square millimetres cancel, leaving newtons. That is a useful check: an area multiplied by force per area must give a force.

Our model predicts aluminium stripping at about 8.58 kN. That is an estimated ultimate failure threshold. We have not yet chosen a sensible operating limit.

MODEL The 0.60 conversion is an assumed approximation, not a measured shear strength for this plate. The result also depends on sound threads and enough surrounding aluminium. Real pull-out tests or verified design data are stronger evidence than this simplified estimate.

04 / MARGIN

Stop before the connection is damaged

A connection can be damaged well before it comes completely apart. *Yielding* means permanent deformation: release the force and the material does not return completely to its previous shape.

The viewer therefore checks an earlier threshold. It takes the assumed tensile yield strength $R_{p0.2}$ of 80 MPa and estimates shear yield using the von Mises relationship:

AN EARLIER LIMIT: APPROXIMATE SHEAR YIELDING

$$\tau_{yield} \approx \frac{R_{p0.2}}{\sqrt{3}} = \frac{80}{1.732} \approx 46.2 \text{ N/mm}^2$$

$$F_{thread\ yield} \approx 75.3 \times 46.2 \approx \mathbf{3.48\ kN}$$

Dividing by √3 translates tensile yield to a shear-yield approximation. It does not identify the first individual thread that will deform.

Now leave additional room between the applied force and that estimate. With the viewer's screening factor n set to 2, the permitted force in this simplified check is halved.

OUR EXAMPLE, WITH A SCREENING FACTOR OF TWO

$$F_{screen} = 3.48 \text{ kN} \div 2 \approx \mathbf{1.74\ kN}$$

Number in this example	What it describes
8.58 kN	Estimated complete aluminium thread stripping
3.48 kN	Approximate thread-yield threshold
1.74 kN	**Thread-yield estimate divided by the chosen factor of 2**

These three numbers answer different questions. That is why the large number in the lab is much lower than its "strip estimate." The divisor of 3 in the geometry formula and the screening factor of 2 are separate: one belongs to the thread model, the other is an additional chosen margin.

05 / THE OTHER HALF

Remember the bolt can fail too

So far, we have watched the aluminium. Now look at the steel. Threads cut into the bolt reduce its

effective tensile area below that of a plain 8 mm rod. For a metric thread, the viewer calculates:

THE BOLT'S EFFECTIVE TENSILE AREA

$$A_{bolt} = \frac{\pi}{4} \times (d - 0.9382 \times p)^2$$

$$= \frac{\pi}{4} \times (8 - 0.9382 \times 1.25)^2 \approx \mathbf{36.6\ mm^2}$$

The 0.9382 constant comes from the conventional effective stress-area expression for metric threads. Squaring the effective diameter makes this an area.

The assumed class 8.8 bolt has an ultimate tensile strength of 800 MPa and a proof stress of 580 MPa in this size range. Proof stress gives a reference for checking the bolt without relying on its breaking strength. [2]

$$F_{bolt\ break} \approx 36.6 \times 800 \approx \mathbf{29.3\ kN}$$

$$F_{bolt\ proof} \approx 36.6 \times 580 \approx \mathbf{21.2\ kN}$$

In our example the aluminium is the limiting part. A stronger steel bolt would not fix that. The lab always selects the smallest applicable capacity before applying the screening factor:

THE COMPLETE DIRECT-PULL SCREENING RULE

$$F_{screen} = \frac{\min(F_{thread\ yield}, F_{strip}, F_{bolt\ proof})}{n}$$

"min" means choose the smallest. For our example: min(3.48, 8.58, 21.2) ÷ 2 = 1.74 kN.

06 / DESIGN CHOICES

Make it wider, or give it more depth

Return to the area equation: $\pi \times d_2 \times L \div 3$. Diameter and engagement length each appear once. Double the length and the model doubles the thread capacity. Increase diameter by a quarter and the gain is

roughly a quarter, with a small adjustment for the changing pitch.

Same aluminium; same factor	Engaged turns	Direct-pull estimate
M8, 10 mm engaged	8.0	1.74 kN
M8, 20 mm engaged	16.0	3.48 kN
M10, 10 mm engaged	6.7	2.18 kN

The M10 is wider, yet it engages fewer turns at the same depth because its coarse pitch is 1.5 mm. Counting turns alone misses the extra circumference of each thread.

Meanwhile, bolt tensile area depends on diameter *squared*. A bigger steel bolt gains tensile capacity faster than a shallow aluminium thread gains pull-out capacity. That is why increasing bolt size alone may leave the aluminium as the weak part.

LIMIT Linear growth is a simplification. Real threads load unevenly; extra depth eventually gives diminishing returns or makes another failure mode govern. The aluminium grade, available wall thickness and edge distance still matter.

07 / PRELOAD

Tightening starts the load before the work does

Change the experiment. Put a plate under the bolt head and tighten it against the aluminium block. Tightening stretches the bolt slightly and compresses the parts. The bolt behaves like a stiff spring pulling its head and threads together.

Call this initial tension F_0, the *preload*. Even with nothing pulling on the plate, the aluminium threads already carry F_0. Preload is a force, measured in newtons or kilonewtons. Wrench torque is a different quantity; friction makes the conversion uncertain.

Now pull upwards on the plate with force P. The bolt tension increases, and the compression between the parts decreases. How much goes into each depends on the relative stiffness of the bolt and joint. The simple model calls the fraction added to bolt tension **C**. [3]

WHILE THE JOINT IS STILL CLOSED

$$F_{bolt} = F_0 + C \times P$$

$$F_{clamp} = F_0 - (1 - C) \times P$$

F_0 · initial preload

P · external separating force

C · fraction added to bolt tension

1 – C · fraction releasing compression

Take a 1 kN preload and assume C = 0.20. If we apply another 0.5 kN to the plate, the bolt picks up 20% of that new pull, while 80% relieves compression:

$$F_{bolt} = 1 + 0.20 \times 0.5 = \mathbf{1.10\ kN}$$

$$F_{clamp} = 1 - 0.80 \times 0.5 = \mathbf{0.60\ kN}$$

Check the force balance: 1.10 kN bolt tension − 0.60 kN contact compression = 0.50 kN external pull.

Keep pulling. Eventually no compression remains. Setting F_{clamp} to zero gives the separation threshold:

WHEN THE JOINT BEGINS TO OPEN

$$P_{separation} = \frac{F_0}{1 - C} = \frac{1}{0.80} = \mathbf{1.25\ kN}$$

Above 1.25 kN in this example, the plate has lost contact with the block. The clamp force is zero and bolt tension equals the entire external pull. Do not keep adding the original preload on top after separation.

ASSUME C = 20% is chosen to teach the mechanism. It is not a universal value for M8 bolts or aluminium. Joint geometry, stiffness and where the force enters the parts determine the actual load sharing.

For a joint that must remain closed, the lab stops at whichever comes first: separation or the selected bolt/thread screening limit. Solving the bolt-tension equation for the available external pull gives $(F_{screen} - F_0) \div C$.

> TWO CONDITIONS; THE SMALLER PERMITTED PULL WINS
>
> $P_{limit} = \max(0, \min(P_{separation}, (F_{screen} - F_0) \div C))$
>
> $\approx \max(0, \min(1.25, (1.74 - 1.00) \div 0.20))$
>
> $= \mathbf{1.25\ kN}$
>
> "max(0, …)" prevents a negative available pull. If preload already exceeds the screening limit, the result is zero. For this example, separation governs; staying firmly closed means operating below that boundary.

08 / USING THE RESULT

Turn the numbers into a design question

Our little M8 connection has led us to three useful distinctions: the geometry supplies the resisting area; the material supplies strength per unit area; and the load case tells us what force is actually reaching the threads.

When the aluminium governs, more sound engagement or a stronger suitable parent material can help. When the bolt governs, extra thread depth does little for the bolt's tensile limit. When separation governs, the preload and joint stiffness need attention—but increasing preload also increases the force the aluminium must carry from the start.

The large figure in the lab is a **screening estimate**. It covers the simplified checks described here. It does not check edge breakout, thin-wall failure, bending, prying, head bearing, fatigue, shock or installation torsion. A chosen factor of two does not make those missing checks disappear.

Before using a result for a real part, confirm the actual material and temper, full engagement, surrounding geometry, load direction and tightening method. The kgf display is only a force conversion: it is not a lifting or supported-mass rating.

You can now read the lab's numbers as a sequence: *How much metal resists? How strong is it? What fails first? And how much of that capacity is already occupied by preload?*

Sources and assumptions. This is an original worked tutorial for the accompanying model. The full material assumptions and source documents are also available in the lab.

[1] NASA RP-1228, Fastener Design Manual, printed page 21 — empirical tapped-hole area model.

[2] Bossard, Materials: screws and nuts — class 8.8 proof and tensile strengths.

[3] Nord-Lock, What happens to preload and clamp force when external loads are applied? — elastic load sharing and separation.

Material inputs: thyssenkrupp 5754-H111 sheet/plate data. Ultimate shear strength = 0.60 × tensile strength is the viewer's assumed conversion; shear yield uses the von Mises approximation. The narrative numbers use 5754-H111 throughout.