GUJARAT TECHNOLOGICAL UNIVERSITY

BE-4 SEMESTER – OLD PAPER – S22 TO S25 – QUESTION BANK SOLUTION

Subject Name & Code:
FUNDAMENTALS OF MACHINE DESIGN (3141907)

Unit 8: Power Screws and Threaded Joints

Repeated Questions

Q1. Explain the terms: Self-locking and overhauling of power screw. State the condition.
(Appeared in: SUM 25 (Q5a), WIN 24 (Q2a), WIN 23 (Q3a), SUM 22 (Q5a), WIN 22 (Q5b))

Answer:
(03-04 marks)

Self-Locking Screw:

- A screw that **will not rotate back** under the load's influence when the applied torque is removed.
- The load cannot descend by itself; it requires torque in the opposite direction to lower it.
- **Condition for self-locking:** $\mu > \tan\ \alpha$ or $\phi > \alpha$
  where: μ = coefficient of friction, φ = friction angle = $\tan^{-1}\mu$, α = helix angle

Overhauling Screw:

- A screw that **will rotate back** under the load's influence when the applied torque is removed.
- The load descends by itself due to its weight.
- **Condition for overhauling:** $\mu < \tan\ \alpha$ or $\phi < \alpha$

Neutral Condition (Borderline):

- $\mu = \tan\ \alpha$ or $\phi = \alpha$

Efficiency Relation:

- For self-locking screws: Efficiency < 50%
- For overhauling screws: Efficiency > 50%

Applications:

- **Self-locking:** Screw jacks, vices, clamps (safety requirement)

- **Overhauling:** Return mechanisms, quick-release devices

Q2. Derive an equation for finding torque required to raise the load by square threaded screws.
(Appeared in: WIN 23 (Q4b), SUM 22 (Q5b))

Answer:
(04 marks)

Step 1: Consider a Square Threaded Screw

- Mean diameter = d_m
- Helix angle = α
- Load to be raised = W
- Coefficient of friction = μ

Step 2: Forces on the Thread
When unwrapping one revolution, the thread becomes an inclined plane:

- Inclination angle = α (helix angle)
- Horizontal force required = P
- Normal reaction = N
- Friction force = μN

Step 3: Force Equilibrium
Resolving forces parallel and perpendicular to the inclined plane:

Perpendicular: $N = W\cos\,\alpha + P\sin\,\alpha$

Parallel: $P\cos\,\alpha = W\sin\,\alpha + \mu N$

Step 4: Substitute and Solve
Substitute N from first equation into second:

$$P\cos\,\alpha = W\sin\,\alpha + \mu(W\cos\,\alpha + P\sin\,\alpha)$$
$$P\cos\,\alpha - \mu P\sin\,\alpha = W\sin\,\alpha + \mu W\cos\,\alpha$$
$$P(\cos\,\alpha - \mu\sin\,\alpha) = W(\sin\,\alpha + \mu\cos\,\alpha)$$
$$P = W \cdot \frac{\sin\,\alpha + \mu\cos\,\alpha}{\cos\,\alpha - \mu\sin\,\alpha}$$

Step 5: Express in terms of φ (friction angle)
Let $\mu = \tan\,\phi$

$$P = W \cdot \frac{\sin\,\alpha + \tan\,\phi\cos\,\alpha}{\cos\,\alpha - \tan\,\phi\sin\,\alpha}$$

$$P = W \cdot \frac{\sin\ \alpha\cos\ \phi + \sin\ \phi\cos\ \alpha}{\cos\ \alpha\cos\ \phi - \sin\ \phi\sin\ \alpha}$$
$$P = W \cdot \frac{\sin\ (\alpha + \phi)}{\cos\ (\alpha + \phi)} = W\tan\ (\alpha + \phi)$$

Step 6: Torque Required
Torque = Force × Radius

$$T = P \times \frac{d_m}{2} = W\tan\ (\alpha + \phi) \times \frac{d_m}{2}$$
$$\boxed{T = W \cdot \frac{d_m}{2} \cdot \tan\ (\alpha + \phi)}$$

For collar friction included (if collar diameter = d_c, μ_c = collar friction):

$$T_{total} = W[\frac{d_m}{2}\tan\ (\alpha + \phi) + \mu_c \cdot \frac{d_c}{2}]$$

Q3. Numerical on Power Screw: Find torque, power, efficiency for given load and thread parameters.
(Appeared in: SUM 25 (Q3c & Q4c-OR), WIN 24 (Q4c & Q5c), WIN 23 (Q3c), SUM 24 (Q5c), SUM 23 (Q5c))

Answer:
(07 marks - General Solution Format)

Typical Given Data:

- Load W (N)
- Screw type (square, Acme, etc.)
- Nominal diameter d, pitch p, single/double/triple start
- Coefficient of thread friction μ, collar friction μ_c
- Collar diameters d_o and d_i
- Speed N (rpm)

Solution Steps:

1. Calculate Screw Geometry:

- For multi-start: Lead L = n × p (n = number of starts)
- Mean diameter: $d_m = d - \frac{p}{2}$ (for square threads)
- Helix angle: $\alpha = \tan^{-1}(\frac{L}{\pi d_m})$

2. Calculate Torque to Raise Load:

- **For square threads:**

$$T = W[\frac{d_m}{2}\tan(\alpha + \phi) + \mu_c \cdot \frac{d_c}{2}]$$

where $\phi = \tan^{-1}\mu$, $d_c = \frac{d_o + d_i}{2}$ (mean collar diameter)

- **For Acme/trapezoidal threads:**

$$T = W[\frac{d_m}{2} \cdot \frac{\cos\beta\tan\alpha + \mu}{\cos\beta - \mu\tan\alpha} + \mu_c \cdot \frac{d_c}{2}]$$

where β = half thread angle (14.5° for Acme, 15° for metric trapezoidal)

3. Calculate Torque to Lower Load:

- **For square threads:**

$$T = W[\frac{d_m}{2}\tan(\phi - \alpha) + \mu_c \cdot \frac{d_c}{2}]$$

(if φ > α, else negative indicates overhauling)

4. Calculate Efficiency:

$$\eta = \frac{\text{Work output}}{\text{Work input}} = \frac{W \times L}{2\pi T}$$

or

$$\eta = \frac{\tan\alpha}{\tan(\alpha + \phi)} \text{(for screw only, neglecting collar)}$$

5. Calculate Power:

Example from SUM 25 (Q4c-OR):
Double-start square thread: d = 25 mm, p = 5 mm, W = 10 kN, d_o = 50 mm, d_i = 20 mm, N = 15 rpm, μ = 0.2, μ_c = 0.15, uniform wear

Solution:

1. **Geometry:**
   - Lead L = 2 × 5 = 10 mm
   - $d_m = 25 - 2.5 = 22.5$ mm

- $\alpha = \tan^{-1}(\frac{10}{\pi \times 22.5}) = \tan^{-1}(0.1415) = 8.05°$
- $\phi = \tan^{-1}(0.2) = 11.31°$
- $d_c = (50 + 20)/2 = 35$ mm (uniform wear)

2. **Torque to raise:**

$$T = 10{,}000[\frac{22.5}{2}\tan(8.05° + 11.31°) + 0.15 \times \frac{35}{2}]$$
$$T = 10{,}000[11.25 \times \tan(19.36°) + 0.15 \times 17.5]$$
$$T = 10{,}000[11.25 \times 0.351 + 2.625]$$

3. **Power:**

$$P = \frac{2\pi \times 15 \times 65.75}{60} = \frac{6195}{60} = 103.3 \text{ W}$$

4. **Efficiency (screw only):**

$$\eta = \frac{\tan \alpha}{\tan(\alpha + \phi)} = \frac{\tan 8.05°}{\tan 19.36°} = \frac{0.1415}{0.351} = 0.403 = 40.3\%$$

Q4. Explain the different types of screw threads used in power screw / Discuss various types of power threads.
(Appeared in: SUM 22 (Q5b), WIN 24 (Q4a-OR))

Answer:
(03-04 marks)

Thread type	Profile	Efficiency	Strength	Application
Square	Flat crest & root, flanks perpendicular to axis	Highest (≈ 45-50%)	Low root strength, difficult to manufacture	Screw jacks, presses, lead screws
Acme	Trapezoidal, 29° included angle	Moderate (≈ 40-45%)	Higher strength than square, easier to machine	Machine tool leadscrews, valves
Buttress	One flank vertical, other	High (≈	Very high strength in one	Heavy presses,

Thread type	Profile	Efficiency	Strength	Application
	slanted (33°/45°)	45-50%)	direction	hydraulic jacks
Knuckle (rounded)	Curved profile	Moderate	Low friction, withstands dirt	Railway couplings, rough service

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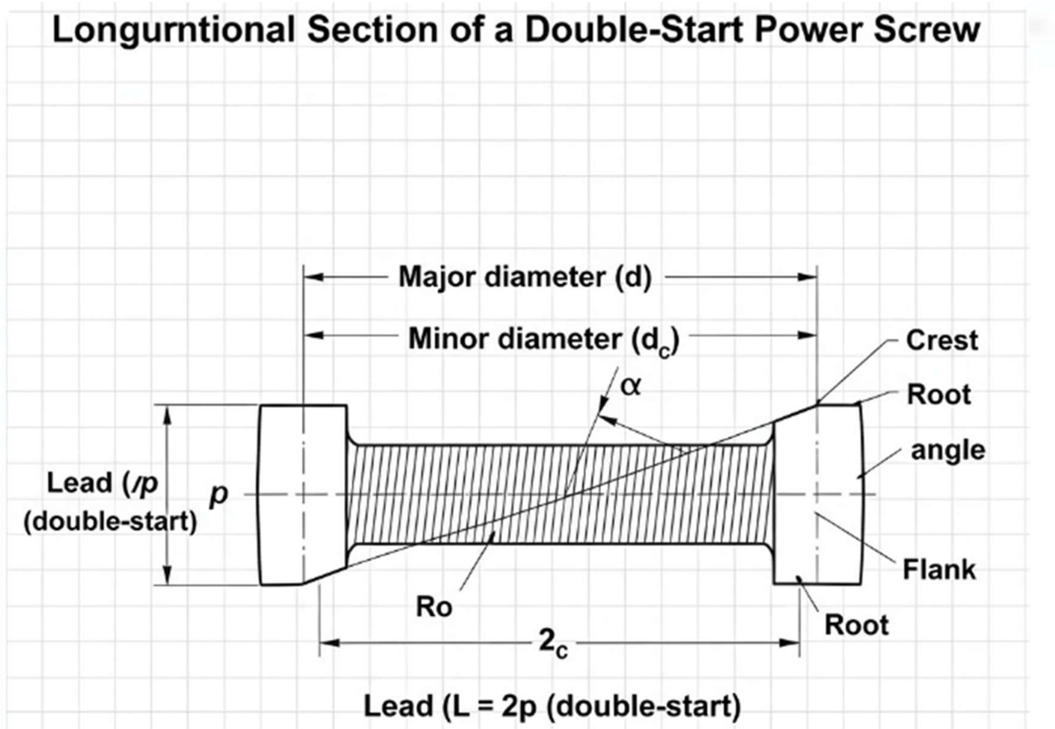
Q5. Explain any three/four Terminology of power screw with figure.
(Appeared in: WIN 23 (Q4a), WIN 22 (Q5b))

Answer:
(03-04 marks)

Important power screw terminology (with figure):

1. **Nominal diameter (Major diameter)** – Largest diameter of the thread (crest to crest).
2. **Core diameter (Minor diameter)** – Smallest diameter of the thread (root to root).
3. **Mean diameter** – Average of major and minor diameters $d_m = (d_o + d_c)/2$. Used for torque calculations.
4. **Pitch (p)** – Distance between corresponding points on adjacent threads (mm).
5. **Lead (l)** – Distance the nut advances in one complete revolution. For single start, lead = pitch; for multi-start, lead = $n \times p$.
6. **Helix angle (α)** – Angle of the thread helix at the mean diameter: $\tan \alpha = l / (\pi d_m)$.
7. **Depth of thread** – (Major dia – Minor dia)/2.



Q6. Explain: (1) Pitch (2) Lead (3) Nominal diameter (4) Core diameter for Power screw.
(Appeared in: SUM 25 (Q4b))

Answer:
(04 marks)

Term	Definition	Formula / Note
Pitch (p)	Axial distance between corresponding points on two consecutive threads	Measured in mm (e.g., p = 8 mm)
Lead (l)	Axial distance the nut moves in one complete revolution of the screw	$l = n \times p$ (n = number of starts)
Nominal diameter (d_o)	Major (outer) diameter – the largest diameter of the thread	Standard size (e.g., M50 – 50 mm nominal)
Core diameter (d_c)	Minor (root) diameter – the smallest diameter (bottom of thread)	$d_c \approx d_o - 2 \times$ depth; used for tensile stress area

Example: For a square thread with nominal 50 mm, pitch 8 mm, depth = p/2 = 4 mm, then core = 50 – 8 = 42 mm, mean = 46 mm.

Q7. Why square threads are preferable to V thread for power transmission?
(Appeared in: WIN 22 (Q1c))

Answer:
(Part of 07 marks answer)

Square threads are preferable to V-threads for power transmission because:

1. **Higher Efficiency:**
    - Square threads have **lower friction** (normal force is axial, no radial component)
    - V-threads have wedging action creating radial thrust, increasing friction
2. **No Radial Bursting Force:**
    - Square threads transmit force axially only
    - V-threads create radial component that can burst the nut
3. **Lower Wear:**
    - Square threads distribute load evenly
    - V-threads concentrate stress at crest
4. **Better for Continuous Motion:**
    - Square threads provide smooth motion
    - V-threads more suitable for intermittent fastening

However, square threads have disadvantages:

- Difficult to manufacture
- Cannot accommodate split nuts
- Weaker root compared to V-threads

Thus, Acme/trapezoidal threads are often preferred as a compromise.

Other Important Questions

Q8. Prove that efficiency of self-locking square thread is less than 50%.
(Appeared in: WIN 22 (Q4b))

Answer:
(04 marks)

Step 1: Efficiency Formula for Square Thread

$$\eta = \frac{\tan \alpha}{\tan(\alpha + \phi)}$$

where α = helix angle, φ = friction angle = $\tan^{-1}\mu$

Step 2: Condition for Self-Locking
For self-locking: $\phi > \alpha$ or $\alpha < \phi$

Step 3: Let α = φ - δ
where δ is a small positive angle (since α < φ)

Step 4: Substitute into Efficiency Formula

$$\eta = \frac{\tan(\phi - \delta)}{\tan[(\phi - \delta) + \phi]} = \frac{\tan(\phi - \delta)}{\tan(2\phi - \delta)}$$

Step 5: Using Trigonometric Identity
For small δ, $\tan(\phi - \delta) \approx \tan\,\phi - \delta\sec^2\phi$

But more importantly, since $\phi > \alpha$:
Maximum efficiency occurs when $\alpha = 45° - \frac{\phi}{2}$

Step 6: Maximum Possible Efficiency for Self-Locking Condition
For self-locking: $\alpha \leq \phi$
Let's set α = φ (borderline case):

$$\eta = \frac{\tan\phi}{\tan(2\phi)} = \frac{\tan\phi}{\frac{2\tan\,\phi}{1 - \tan^2\phi}} = \frac{1 - \tan^2\phi}{2}$$

Step 7: Show η < 0.5
Since $\tan^2\phi > 0$ for φ > 0:

$$\eta = \frac{1 - \tan^2\phi}{2} < \frac{1}{2} = 0.5$$

And for α < φ, efficiency is even lower.

Step 8: General Proof
For any self-locking screw (α ≤ φ):

- When α = 0: η = 0
- When α = φ: η = $(1 - \tan^2\varphi)/2 < 0.5$
- Maximum occurs at α = 45° - φ/2, but this violates α ≤ φ for φ < 30°

Therefore, **efficiency of self-locking screw is always less than 50%.**

Practical Implication: If a screw jack has efficiency > 50%, it will overhaul (descend by itself), which is unsafe for lifting applications.

Q9. Classify the basic types of screw fastening. Differentiate between bolt and screw.
(Appeared in: WIN 24 (Q4b))

Answer:
(04 marks)

Classification of screw fastening (threaded fasteners):

1. **Bolts** – Used with nuts; typically have a plain shank and a head.
2. **Screws** – Generally threaded fully or partially, used without nuts (tapped hole).
3. **Studs** – Threaded on both ends; one end screws into a tapped hole, other end receives a nut.

Difference between bolt and screw:

Feature	Bolt	Screw
Nut	Always used with a nut	Used without nut (into tapped hole)
Tightening	Rotating nut or bolt head	Rotating screw head (or tool)
Shank	Partially unthreaded shank common	Fully threaded or partially threaded
Application	Through holes, assembly of two parts	Blind holes, covers, machine components
Examples	Hex bolt, carriage bolt	Machine screw, set screw, wood screw

Q10. Discuss the advantages and disadvantages of threaded joints.
(Appeared in: SUM 24 (Q5a-OR))

Answer:
(03 marks)

Advantages of threaded joints:

1. **Detachable** – Can be disassembled without damaging parts.
2. **Standardized** – Wide availability, interchangeable.
3. **High clamping force** – Can generate large preload with small torque.
4. **Simple to assemble** – Using common tools.
5. **Cost-effective** – Low manufacturing cost.

Disadvantages:

1. **Stress concentration** – At thread roots, leading to fatigue failure.
2. **Loosening** – Under vibration (needs locking mechanisms).

3. **Less efficient space-wise** – Requires additional length.
4. **Corrosion** – Threads susceptible to seizing.

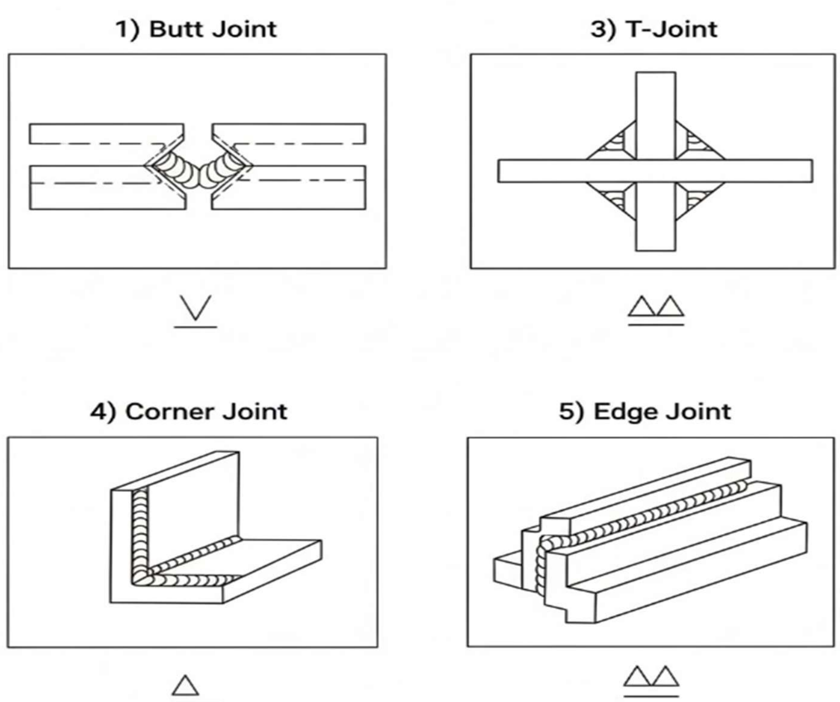
Real-world application: Threaded joints are the most common detachable fastener in machines, vehicles, and structures.

Q11. Explain different types of welded joints with neat sketches.
(Appeared in: SUM 22 (Q4a))

Answer:
(03 marks)

Types of welded joints (based on configuration):

Type	Sketch description	Application
Butt joint	Two plates in same plane, edges brought together	Pipes, boiler plates
Lap joint	Plates overlapping	Sheet metal, structural steel
T-joint	One plate perpendicular to another	Frames, stiffeners
Corner joint	Plates at right angle, forming a corner	Box sections, enclosures



Q12. The load on a bolt consists of an axial pull of 10 kN together with a transverse shear force of 5 kN. Find the diameter of bolt required according to 1. Maximum principal stress theory; 2. Maximum shear stress theory; 3. Maximum principal strain theory. Take permissible tensile stress at elastic limit = 100 MPa and poisson's ratio = 0.3.
(Appeared in: WIN 23 (Q4c))

Answer:
(07 marks)

Step 1: Given Data

- Axial pull, P = 10 kN = 10,000 N
- Transverse shear, F = 5 kN = 5,000 N
- Permissible tensile stress, σ_perm = 100 MPa = 100 N/mm²
- Poisson's ratio, μ = 0.3
- Assume: Bolt in single shear

Step 2: Stress Components
Let bolt diameter = d

Tensile stress from axial pull:

$$\sigma = \frac{P}{A} = \frac{10{,}000}{\frac{\pi}{4}d^2} = \frac{12{,}732}{d^2}\ \text{N/mm}^2$$

Shear stress from transverse load:

$$\tau = \frac{F}{A} = \frac{5{,}000}{\frac{\pi}{4}d^2} = \frac{6{,}366}{d^2}\ \text{N/mm}^2$$

Step 3: Maximum Principal Stress Theory (Rankine)

Principal stresses:

$$\sigma_{1,2} = \frac{\sigma}{2} \pm \sqrt{(\frac{\sigma}{2})^2 + \tau^2}$$
$$\sigma_1 = \frac{\sigma}{2} + \sqrt{(\frac{\sigma}{2})^2 + \tau^2}$$

Condition: $\sigma_1 \le \sigma_{perm}$

Substitute:

$$\frac{1}{2}\cdot\frac{12{,}732}{d^2} + \sqrt{(\frac{1}{2}\cdot\frac{12{,}732}{d^2})^2 + (\frac{6{,}366}{d^2})^2} \le 100$$

$$\frac{6{,}366}{d^2} + \sqrt{(\frac{6{,}366}{d^2})^2 + (\frac{6{,}366}{d^2})^2} \leq 100$$
$$\frac{6{,}366}{d^2} + \sqrt{2 \times (\frac{6{,}366}{d^2})^2} \leq 100$$
$$\frac{6{,}366}{d^2} + \frac{6{,}366}{d^2}\sqrt{2} \leq 100$$
$$\frac{6{,}366}{d^2}(1 + 1.4142) \leq 100$$
$$\frac{6{,}366}{d^2} \times 2.4142 \leq 100$$
$$\frac{15{,}373}{d^2} \leq 100$$
$$d^2 \geq 153.73$$
$$d \geq 12.40 \text{ mm}$$

Step 4: Maximum Shear Stress Theory (Tresca)

Maximum shear stress:

$$\tau_{max} = \sqrt{(\frac{\sigma}{2})^2 + \tau^2}$$

Condition: $\tau_{max} \leq \frac{\sigma_{perm}}{2} = 50 \text{ N/mm}^2$

$$\sqrt{(\frac{6{,}366}{d^2})^2 + (\frac{6{,}366}{d^2})^2} \leq 50$$
$$\sqrt{2 \times (\frac{6{,}366}{d^2})^2} \leq 50$$
$$\frac{6{,}366}{d^2} \times \sqrt{2} \leq 50$$
$$\frac{9{,}000}{d^2} \leq 50$$
$$d^2 \geq 180$$
$$d \geq 13.42 \text{ mm}$$

Step 5: Maximum Principal Strain Theory (St. Venant)

Condition: $\sigma_1 - \mu\sigma_2 \leq \sigma_{perm}$

where $\sigma_2 = \frac{\sigma}{2} - \sqrt{(\frac{\sigma}{2})^2 + \tau^2}$

From Step 3:

$$\sigma_1 = \frac{6{,}366}{d^2}(1 + \sqrt{2}) = \frac{15{,}373}{d^2}$$
$$\sigma_2 = \frac{6{,}366}{d^2}(1 - \sqrt{2}) = \frac{6{,}366}{d^2}(1 - 1.4142) = -\frac{2{,}641}{d^2}$$

Apply condition:

$$\frac{15{,}373}{d^2} - 0.3 \times (-\frac{2{,}641}{d^2}) \le 100$$
$$\frac{15{,}373}{d^2} + \frac{792}{d^2} \le 100$$
$$\frac{16{,}165}{d^2} \le 100$$
$$d^2 \ge 161.65$$
$$d \ge 12.71 \text{ mm}$$

Step 6: Summary of Results

1. **Maximum Principal Stress Theory:** $d \ge 12.40$ mm
2. **Maximum Shear Stress Theory:** $d \ge 13.42$ mm
3. **Maximum Principal Strain Theory:** $d \ge 12.71$ mm

Conservative choice: Take largest diameter → **d = 14 mm**

Check standard sizes: M14 has core diameter ≈ 11.8 mm, so need to check actual stresses. Better to use **M16** (d = 16 mm) for safety.

$$\boxed{\text{Required bolt diameter: 14 mm (minimum), use M16 for safety}}$$

Q13. Justify that "Preloading improves the fatigue strength of bolted joint."
(Appeared in: SUM 23 (Q4a-OR))

Answer:
(03 marks)

Preloading (bolt tightening to a specified initial tension) improves fatigue strength because:

1. **Reduces alternating stress amplitude** – When an external fluctuating load is applied, the bolt's tension varies between preload and (preload + proportion of load). Without preload, the bolt stress varies from 0 to maximum (full load amplitude). With preload, the mean stress increases but the amplitude decreases significantly, which is beneficial for fatigue.
2. **Prevents separation of joint** – Maintains clamping force, avoiding impact loading and fretting.
3. **Stress-strain hysteresis** – Preload moves the operating point into a region where the effective stress range is smaller relative to the endurance limit.

Explanation with a graph (conceptually):
Without preload: stress range Δσ = σ_max – 0 = high.
With preload: stress range Δσ = (σ_preload + Δσ_ext) – (σ_preload – Δσ_ext) = 2Δσ_ext, which is small if external load variation is small. The high mean stress is offset by a reduced amplitude, which lies below the Goodman or Soderberg fatigue line.

Real-world application: Cylinder head bolts, connecting rod bolts, and flange bolts are always preloaded to a specified torque to ensure long fatigue life.

Q14. Explain types of fits
(Appeared in: SUM 22 (Q4b))

Answer:
(04 marks)

Fit: The relationship between two mating parts (hole and shaft) based on the difference in their sizes before assembly.

Three Main Types of Fits:

1. Clearance Fit:

- **Condition:** Hole size > Shaft size
- **Always positive clearance**
- **Types:**
  - **Loose running:** Large clearance for relative motion with lubrication
  - **Free running:** Moderate clearance for easy assembly
  - **Close running:** Small clearance for accurate location
- **Applications:** Bearings, pivots, sliding parts
- **Example:** H7/g6, H8/f7

2. Interference Fit:

- **Condition:** Hole size < Shaft size
- **Negative clearance** (interference)
- Requires force for assembly (pressing, shrinking, expanding)
- **Types:**
  - **Light drive:** Minimal interference, hand or light press
  - **Medium drive:** Moderate interference, hydraulic press
  - **Heavy drive:** High interference, heating/cooling required
- **Applications:** Gears on shafts, bearing races, permanent assemblies
- **Example:** H7/p6, H7/s6

3. Transition Fit:

- **Condition:** May have clearance or interference
- Overlap zone between clearance and interference
- **Types:**
  - **Push fit:** Light force for assembly, zero to slight interference
  - **Wringing fit:** Very small clearance/interference, hand pressure

- **Applications:** Hubs, gears requiring accurate location
- **Example:** H7/js6, H7/k6

ISO System Notation:

- **Basic size:** Nominal size (e.g., 50 mm)
- **Tolerance grade:** IT number (IT7, IT8 - lower number = tighter tolerance)
- **Fundamental deviation:** Letter (H for hole usually, small letters for shaft)
- **Example:** 50H7/g6 means:
  - Hole: 50H7 (basic size 50, H hole, IT7 tolerance)
  - Shaft: 50g6 (basic size 50, g shaft, IT6 tolerance)