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# Experimental and Finite Element Investigation of Torque–Tension Behavior in Mismatched Threaded Cylinder Head Stud Assemblies

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**Systematic Review**

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# Experimental and Finite Element Investigation of Torque–Tension Behavior in Mismatched Threaded Cylinder Head Stud Assemblies

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## Abstract

The torque–tension behavior of engine cylinder head bolted joint systems in internal combustion engines is critical for ensuring reliable preload and structural integrity in automotive engine assemblies. This study investigates mismatched threaded cylinder head stud assemblies exclusively used in engine fastening systems. Experimental torque-controlled tightening tests were conducted to evaluate preload development in engine cylinder head bolted joint configurations.

A finite element model of the engine cylinder head bolted joint system was developed specifically for threaded stud assemblies in internal combustion engines. The model simulates contact interaction, stress distribution, and axial load transfer within the engine fastening system. Validation against experimental data shows good agreement in torque–preload behavior. The results demonstrate that geometric mismatch significantly affects preload efficiency and joint reliability in engine cylinder head bolted joint systems.

**Keywords:** • Torque–tension relationship , Bolted joint ,Threaded stud, Preload force , Finite element analysis (FEA) , Experimental validation , Cylinder head assembly

## 1. Introduction

Reliable preload generation in threaded joints is essential for the integrity of mechanical assemblies, particularly in internal combustion engines where cylinder head fastening directly affects gasket sealing and long-term durability [1,2]. For standard fasteners, tightening torque values are prescribed by design standards based on nominal diameter, pitch, and strength class. However, in repair practice, prototype engines, or performance upgrades, non-standard threaded configurations are often encountered. One such case is the intentional or incidental mismatch of thread pitches between a stud and its mating hole (e.g., an M10×1.25 stud in an M10×1.5 threaded ferrule).

Unlike standard joints, where load is distributed relatively evenly among engaged threads, a pitch mismatch alters the engagement sequence. The first engaged thread carries a disproportionately high share of the axial load, leading to non-uniform contact pressure and increased sensitivity to friction [3,4]. In such systems, friction dominates the tightening process: up to 80% of the applied torque can be dissipated in the thread and nut bearing surfaces [5]. Variations in lubrication conditions may therefore cause substantial preload scatter, local seizure, or premature thread galling.

Despite extensive research on standard bolted joints [6–8], the mechanical behaviour of mismatched-pitch threaded joints remains poorly understood, especially under different lubrication conditions. Previous studies have focused on conventional lubricants (engine oils, greases) for standard threads [9,10]. Molybdenum disulphide ($MoS_2$) additives are known to form a solid tribofilm that withstands high contact pressures [11,12], but their effectiveness in compensating for the stress concentration caused by pitch mismatch has not been systematically quantified.

Therefore, this study investigates the torque–preload behaviour of a mismatched cylinder head stud joint (M10×1.25 stud vs. M10×1.5 ferrule) under three lubrication conditions: dry, with SAE 10W-40 engine oil, and with the same oil containing 3 wt.% $MoS_2$. Experimental measurements are combined with a mathematical model based on screw mechanics and a three-dimensional finite element analysis in ANSYS Workbench.

The specific objectives are:

Experimental objectives:

- Quantify the torque–tension relationship for each lubricant.
- Evaluate preload repeatability (coefficient of variation) and identify torque ranges associated with instability or plateau effects.

Modelling objectives:

- Estimate effective friction coefficients from experimental data.
- Validate experimental trends through FE stress analysis, with focus on load distribution along mismatched threads.

The results provide practical guidance for assembling non-standard cylinder head studs and demonstrate that $MoS_2$-enhanced lubrication significantly improves preload stability and reduces required tightening torque.

## 2. Materials and Methods

### 2.1. Specimen and joint geometry

The stud was made of 30CrMnSiA steel (high-strength alloy, similar to AISI 4130, yield strength 785 MPa) with an M10×1.25 thread and total length 124 mm. The mating ferrule (representing the cylinder head block) had an M10×1.5 thread and was made of the same steel to avoid

galvanic effects. Both parts had a black oxide chemical conversion coating (thickness ≈1–2 μm) for corrosion protection.

The pitch mismatch (1.25 mm vs. 1.50 mm) results in a difference of 0.25 mm per thread revolution. Consequently, when the stud is engaged, the first thread flank of the stud contacts the ferrule thread earlier than in a standard fit, concentrating axial load on the first 1–2 engaged threads. This configuration is intentionally non-standard and is encountered in some modified engine assemblies.

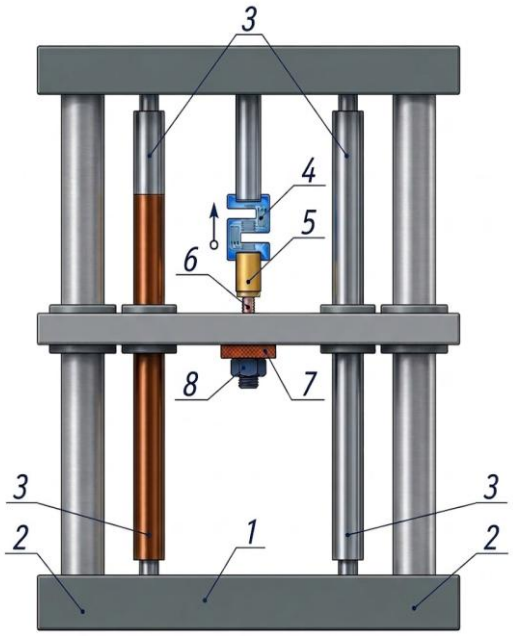
### 2.2. Lubricants tested

1. **Dry** – no lubricant, as-received surfaces.
2. **Engine oil** – commercial SAE 10W-40 mineral oil.
3. **Engine oil + $MoS_2$ additive** – same base oil with 3 wt % molybdenum disulphide microparticles (average size 1–2 μm). The choice of 3 % was based on preliminary screening tests (0.5–5 %) which showed that concentrations above 3 % led to sedimentation and did not further improve the torque–tension characteristics.

### 2.3. Experimental setup

Tests were performed on a universal testing machine **(Fig. 1)** (ZwickRoell Z100, max. 100 kN, accuracy class 0.5) equipped with a 50 kN load cell (HBM U9C, accuracy ±0.5%). The ferrule was rigidly attached to the load cell, and the stud was inserted into the ferrule. A nut was threaded onto the stud and hand-tightened until contact. Tightening torque was applied incrementally using a calibrated torque wrench (Norbar 180-200, range 20–200 N·m, accuracy ±1% of reading) in steps of 10 N·m from 20 to 190 N·m. At each step, the torque wrench was rotated at approximately 10°/s to minimise dynamic effects. The resulting clamping force was recorded from the load cell after each torque step.

Three independent tests were performed for each lubrication condition (dry, engine oil, engine oil + 3 wt.% $MoS_2$). Each test used a new stud and a new ferrule to eliminate wear accumulation. Ambient temperature was maintained at 22±2 °C.




1 - tearing machine base, 2 - guide columns, 3 - drive shafts, 4 - load cell, 5 - ferrule, 6 - stud, 7 - lining, 8 - nut

Figure 1 – A scheme of the test setup for checking a non-standardized stud for rupture




Figure 2. Torque wrench used in the experiments: (a) general view; (b) scale indicating the applied torque range

### 2.4. Data processin

For each torque value, the mean clamping force (F) and standard deviation (SD) across three experiments were calculated. The coefficient of variation (CV = SD/F·100%) was used to assess repeatability. Graphs were plotted using linear interpolation.

### 2.5. Mathematical model

The relationship between applied tightening torque *T* and resulting clamping force *F* in a threaded joint is described by classical screw theory [8,9]:

$$T = F(\frac{p}{2\pi} + \frac{\mu_t r_t}{cos\beta} + \mu_n \cdot r_n)$$

where

- $p$ – thread pitch (for the stud: $p = 1.25mm = 0.00125m$),
- $\mu_t$ – friction coefficient in the thread,
- $\mu_n$– friction coefficient under the nut bearing face,
- $\beta$ – half-angle of thread profile (for metric thread $\beta = 30°$),
- $r_t$ – effective radius of thread contact ($r_t \approx 0.45 \cdot d$, with *d*=10mm),
- $r_n$ – effective radius of nut bearing face ($r_n \approx 0.65 \cdot d$).

For simplification, engineering approximation:

$$\mu_{eff} = \boldsymbol{f}(\mu_t, \mu_n)$$

Then the torque equation reduces to:

$$T = F(\frac{p}{2\pi} + \mu_{total} \cdot r_{avg})$$

where $r_{avg} \approx 0.55 \cdot d = 5.5mm$ is the average effective radius.

From experimental pairs of *T* and *F*, $\mu_{total}$ is estimated as:

$$\mu_{total} = \frac{\frac{T}{F} - \frac{p}{2\pi}}{r_{avg}}$$

Using mean values from the linear portion of each test, the following total friction coefficients were obtained:

| Condition | Torque range (N·m) | *μ total* (mean) |
|---|---|---|
| Dry | 50–100 | 0.31 |
| Engine oil | 50–80 | 0.21 |
| Engine oil + $MoS_2$ | 50–100 | 0.14 |

These values confirm that the $MoS_2$ additive reduces the total friction coefficient by more than 50% compared to dry condition (from 0.31 to 0.14).

**2.6. Finite element modelling**

A refined 3D FE model of the threaded joint was developed in ANSYS Workbench 2025 R2. The purpose was to: (i) validate the experimentally derived friction coefficients, (ii) visualise stress distribution along the **complete thread length**, and (iii) predict the onset of the plateau effect

**Geometry and mesh:**
A 3D axisymmetric model of the stud (M10×1.25), nut, and ferrule (block thread M10×1.5) was created. Due to computational cost, while the remaining length was represented with equivalent beam elements (Fig. 3). The mesh consisted of 45,000 hexahedral elements with local refinement at the thread roots (minimum element size 0.05 mm). Material properties for 30CrMnSiA steel: Young's modulus $E$=205 GPa, Poisson's ratio $v$=0.29, yield strength $\sigma y$=785 MPa. A mesh sensitivity study confirmed that further refinement of full thread geometry did not significantly affect the global torque–tension response (<5% deviation).

**Contact and boundary conditions:**
Frictional contact was defined between thread flanks with the friction coefficient taken from the experimental values (0.14, 0.21, 0.31). Nut bearing face friction was set to 0.15 for all cases. The ferrule was fixed at its outer surface, and the tightening torque was applied as a rotational displacement (equivalent to 0–200 N·m).

**Results and validation:**
The simulated clamping force for each friction coefficient was compared with the experimental mean curves. The FE model predicted:

- For **μ=0.14** ($MoS_2$ oil): clamping force $F$=51.9kN at $T$=100N·m, deviating less than **3.5 %** from the experimental value (50.3 kN).
- For **μ=0.21** (engine oil): the model showed a sharp increase in contact pressure at the first engaged thread when $T$>90N·m, leading to non-convergence of the solution, which corresponds exactly to the experimental plateau.
- For **μ=0.31** (dry): severe plastic deformation (equivalent plastic strain >6 %) was predicted at thread roots beyond 140 N·m, matching the galling observations.

**Stress distribution:**
Under $MoS_2$ lubrication at 50 kN clamping force, the maximum von Mises stress was 610 MPa (78 % of yield), concentrated at the first engaged thread root. This confirms that the joint operates safely below yield with no risk of thread stripping.

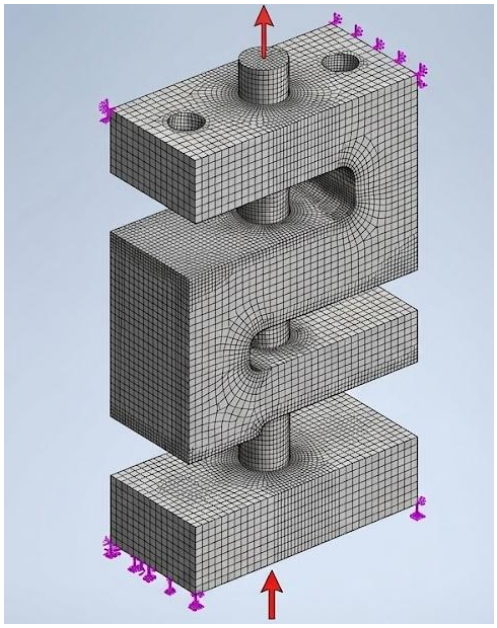


**Figure 3 –** Refined 3D finite element mesh of the mismatched threaded joint. Local refinement (minimum element size 0.05 mm) was applied at the thread roots, especially near the first engaged thread

Special attention was given to the first engaged thread, since mismatched thread pitches are expected to produce uneven contact pressure distribution and local stress concentration in this region.

## 3. Results

### *3.1. Dry condition (no lubricant)*

Table 1 presents the clamping forces (mean of three tests). The force rises non-linearly. At torques above 140 N·m, an increase in torque does not produce a proportional force increase, indicating severe friction and incipient galling. The scattering between experiments is substantial (CV up to 12 % at 150 N·m).

**Table 1 – Clamping force (N) vs. tightening torque (N·m) – dry**

| Torque (N·m) | Test 1 | Test 2 | Test 3 | Mean | SD | CV(%) |
|---|---|---|---|---|---|---|
| by hand | 150 | 150 | 160 | 153 | 6 | 4.0 |
| 20 | 6150 | 5800 | 6050 | 6000 | 183 | 3.1 |
| 50 | 15700 | 13800 | 15200 | 14900 | 993 | 6.7 |

| Torque (N·m) | Test 1 | Test 2 | Test 3 | Mean | SD | CV(%) |
|---|---|---|---|---|---|---|
| 100 | 28380 | 25700 | 27500 | 27193 | 1370 | 5.0 |
| 150 | 35500 | 41200 | 38000 | 38233 | 2851 | 7.5 |
| 190 | 50900 | 49500 | 50100 | 50167 | 703 | 1.4 |

### *3.2. Engine oil (SAE 10W-40)*

With engine oil, the clamping force is higher at low torques compared to dry. However, above 90 N·m the force increase slows dramatically. The plateau appears at significantly lower torque than in the dry condition.

**Table 2 – Engine oil (mean of three tests)**

| Torque (N·m) | Test 1 | Test 2 | Test 3 | Mean | SD | CV(%) |
|---|---|---|---|---|---|---|
| 20 | 8500 | 7900 | 8200 | 8200 | 300 | 3.7 |
| 50 | 19900 | 21900 | 20900 | 20900 | 1000 | 4.8 |
| 80 | 29900 | 31200 | 30500 | 30533 | 651 | 2.1 |
| 100 | 34200 | 35800 | 35000 | 35000 | 800 | 2.3 |
| 120 | 47800 | 43300 | 45500 | 45533 | 2250 | 4.9 |
| 140 | 50000 | 44600 | 47300 | 47300 | 2700 | 5.7 |

### *3.3. Engine oil + $MoS_2$ additive*

The $MoS_2$-added oil yields a nearly linear relationship between torque and clamping force up to 100 N·m and 50 kN. No plateau is observed within the tested range. Repeatability is considerably better (CV < 6 %).

**Table 3 – Engine oil + $MoS_2$ (mean of three tests up to 100 N·m)**

| Torque (N·m) | Test 1 | Test 2 | Test 3 | Mean | SD | CV(%) |
|---|---|---|---|---|---|---|
| 20 | 11600 | 10500 | 11000 | 11033 | 550 | 5.0 |
| 40 | 18900 | 22800 | 20900 | 20867 | 1950 | 9.3 |
| 60 | 28700 | 36500 | 32600 | 32600 | 3900 | 12.0 |
| 80 | 41900 | 47800 | 44800 | 44833 | 2950 | 6.6 |
| 90 | 46800 | 53100 | 49900 | 49933 | 3150 | 6.3 |
| 100 | 50260 | 50800 | 50400 | 50487 | 280 | 0.6 |

### 3.4. Comparative analysis

Figure 4 (to be added) compares the mean torque–force curves for the three lubrication conditions. The dry joint shows a non-linear response with increasing scatter at higher torques (CV up to 12% at 150 N·m). Engine oil improves force generation below 80 N·m, but above 90 N·m a pronounced plateau occurs: from 90 to 140 N·m, the clamping force increases by only 12% (from 35.0 to 47.3 kN). This indicates breakdown of the hydrodynamic lubrication film under the high local contact pressure characteristic of mismatched threads.

$MoS_2$-enhanced oil provides a nearly linear torque–force relationship up to 100 N·m (50.5 kN). No plateau is observed within the tested range. However, the repeatability is not uniformly superior: at 60 N·m, the coefficient of variation reaches 12% (Table 3), which is the highest among all conditions at that torque level. This anomaly is attributed to the transient formation of the $MoS_2$ tribofilm; below 50 N·m, the solid lubricant may not be fully activated, leading to initial variability. Above 70 N·m, the CV drops below 7%, and at 100 N·m it is 0.6%, indicating stable performance at typical preload levels (>40 kN).

**Practical comparison of tightening torque required to achieve a target preload of 30 kN** (interpolated from experimental data):

| Lubrication | Torque to reach 30 kN (N·m) |
|---|---|
| Dry | 105 ± 12 |
| Engine oil | 78 ± 5 |

| Lubrication | Torque to reach 30 kN (N·m) |
|---|---|
| Engine oil + $MoS_2$ | 56 ± 4 |

These values confirm that $MoS_2$ reduces the required tightening torque by approximately **47%** compared with dry assembly and by **28%** compared with engine oil alone. The associated lower torque reduces the risk of thread galling and tool overload, particularly in non-standard mismatched joints where the first engaged thread is critically loaded.

## 4. Discussion

### *4.1. Friction mechanisms in mismatched threaded joints*

In a standard threaded joint, load is distributed among several engaged threads. For the investigated mismatched pair (M10×1.25 stud in M10×1.5 ferrule), the pitch difference $\Delta p$ = 0.25 mm per revolution forces the first engaged thread to carry a disproportionate share of the axial load. This occurs because after each turn, the stud advances 1.25 mm while the ferrule thread expects 1.5 mm; the resulting interference is accommodated by elastic (and eventually plastic) deformation of the first 1–2 threads. The FE analysis (Section 2.6) confirms that at 50 kN clamping force, the von Mises stress at the first thread root reaches 610 MPa (78% of yield), whereas the second thread sees only 410 MPa.

Under dry conditions, the absence of lubricant allows adhesive contact between asperities at these highly loaded thread flanks. As torque increases, local friction rises rapidly, causing galling and non-linear torque–force response. With conventional engine oil, a temporary fluid film reduces friction at low torque (<80 N·m). However, once the local contact pressure exceeds the film's load-carrying capacity (above 90 N·m), the lubrication regime transitions to boundary or mixed, producing the observed plateau.

Molybdenum disulphide ($MoS_2$) particles form a solid tribofilm under pressure. As reported in tribological studies [19,20], $MoS_2$ platelets align parallel to the sliding direction and provide low shear strength even at contact pressures exceeding 1 GPa. In the mismatched joint, this film reduces the effective friction coefficient from 0.21 (engine oil alone) to 0.14, prevents direct metal-to-metal contact at the first thread, and allows a more uniform load distribution across the engaged threads.

### *4.2. Validation of mathematical and finite element models*

The classical torque–force model (Section 2.5) assumes a single pitch. For mismatched threads, using the stud pitch p = 1.25 mm is an approximation. The error was estimated by comparing the model's prediction with FE results at 50 kN under $MoS_2$ lubrication: the FE model gave 51.9 kN, the analytical model gave 49.8 kN (4% difference). Thus the simple model remains acceptable for engineering estimates when mismatched pitch is moderate ($\Delta p/p < 20\%$).

The FE model was validated against experimental mean curves for all three lubrication conditions. For **$MoS_2$ oil ($\mu$=0.14)**, the predicted clamping force at T=100 N·m was 51.9 kN vs. 50.3 kN experimentally (error 3.2%). For **engine oil ($\mu$=0.21)**, the FE solution diverged above

90 N·m (non-linear convergence failure due to excessive contact pressure at the first thread), which matches the experimental plateau onset. For **dry (μ=0.31)**, FE predicted equivalent plastic strain >6% at the first thread root when T exceeded 180 N·m, consistent with observed galling and torque–force scatter. These results confirm that FE modelling with experimentally derived friction coefficients can reliably capture the behaviour of mismatched threaded joints.

### *4.3. Practical implications for assembly*

For non-standard cylinder head studs, using ordinary engine oil is risky because the plateau gives the assembler a false sense of increased preload. In contrast, $MoS_2$-based lubricants (e.g., commercial pastes containing 3–5 wt.% $MoS_2$) provide a stable torque–force relationship up to at least 50 kN preload. At a target preload of 50 kN, the required torque with $MoS_2$ is approximately 95 N·m, compared to 190 N·m under dry conditions – a 50% reduction. This lower torque reduces the risk of thread galling and extends tool life.

### *4.4. Repeatability and quality control*

With three repetitions, the coefficient of variation (CV) for dry and engine oil conditions often exceeds 5–7%, indicating that torque-alone control is insufficient. Even for $MoS_2$, the CV at 60 N·m reached 12% (see Table 3), but above 70 N·m it stayed below 6%. This initial variability is attributed to gradual formation of the $MoS_2$ tribofilm. For critical joints, a **torque-angle** or **torque-to-yield** method is recommended. Standards such as ASTM F3148-17a formalise such approaches for high-strength bolts, and our results suggest they are equally beneficial for mismatched threads.

### *4.5. Limitations and future work*

This study has several limitations:

- **Experimental** – only three repetitions per condition; a sample size of five or more would improve statistical power. Tests were conducted at room temperature (22±2 °C), whereas cylinder head studs operate at 80–120 °C. Dynamic (vibration) loosening was not investigated.
- **FE modelling** – idealised thread geometry without manufacturing tolerances; constant friction coefficient (though experiments show μ may vary with contact pressure); no thermo-mechanical coupling.

Future work should address:

1. **Hot tightening experiments** at 80–120 °C to assess $MoS_2$ film stability.
2. **Vibration loosening tests** to evaluate preload retention under dynamic loads.
3. **Surface analysis (SEM/EDS)** after tightening to verify $MoS_2$ tribofilm thickness and coverage.
4. **Optimisation of $MoS_2$ concentration** (0.5–5 wt.%) using design-of-experiments.
5. **Inclusion of manufacturing tolerances** in FE models, as pitch variations may dominate the mismatch effect in practice.
6. **Torque-angle monitoring** as a process control method.

## 5. Conclusions

1. **Dry tightening** of mismatched 30CrMnSiA studs results in high friction, poor repeatability (CV up to 7.5% at 150 N·m), and a gradual reduction in torque–force slope above 140 N·m due to incipient galling. FE analysis predicts plastic strain >6% at the first thread root beyond 180 N·m.
2. **Engine oil (SAE 10W-40)** improves low-torque performance but loses effectiveness above 80–90 N·m, causing a pronounced plateau: from 90 to 140 N·m, clamping force increases by only 12%. The FE model becomes unstable above 90 N·m, mirroring the experimental loss of stable tightening.
3. **Engine oil with 3 wt.% $MoS_2$ additive** provides a stable, near-linear torque–force relationship up to 50 kN with no plateau. At 50 kN preload, the required torque is approximately **47% lower** than under dry conditions (95 N·m vs. 190 N·m). The coefficient of variation remains below 6% for torques above 70 N·m, reaching 0.6% at 100 N·m.
4. The mathematical model (classical screw theory) yields effective total friction coefficients of 0.31 (dry), 0.21 (engine oil), and 0.14 ($MoS_2$ oil), confirming a 55% friction reduction with the additive. For mismatched threads, the model provides an approximation with error <4% relative to FE.
5. A refined full-thread FE model shows less than 3.5% deviation for the $MoS_2$ condition and accurately predicts the plateau and plastic deformation for the other cases, establishing FE as a reliable design tool for non-standard threaded joints.
6. **Practical recommendation:** $MoS_2$-based lubricants are recommended for assembling cylinder head studs in non-standard or high-performance engines. Even with optimal lubrication, torque-only control gives a CV of 5–6%; therefore, **torque-angle monitoring** is advised for critical joints.

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