

参赛学生姓名：李越慕

中学：北京市第一〇一中学

省份：北京市

国家：中国

指导教师姓名：陈艳

指导教师单位：中国科学院力学研究所

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Frictional Self-Locking Mechanics of Proposed Steel-Wire Configurations: A Finite-Element Analysis

Yuemu Li *

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Abstract

The mechanical behavior of steel-wire strands depends not only on the properties of the constituent wires but also on the geometry through which they are assembled. This study investigates frictional self-locking in interlaced steel-wire structures through a combination of dynamic tensile experiments, analytical contact mechanics, and finite-element analysis. Three alternative seven-wire configurations inspired by three- and four-strand braids were first fabricated and compared with a conventional 1 + 6 strand using a Split Hopkinson Tensile Bar system. The three-strand braided configuration showed the most favorable combination of peak tensile stress and deformation capacity among the proposed designs and was selected for further investigation.

The experimental topology was subsequently idealized as a symmetric three-strand plait. A geometric contact model was developed to describe a representative wire-wire crossing, and a trial-sticking formulation was combined with the Coulomb friction criterion to define a critical friction coefficient,

$$\mu_{\text{crit}} = \frac{\|\mathbf{Q}_c^{\text{req}}\|}{N_c},$$

where N_c is the normal contact reaction and $\mathbf{Q}_c^{\text{req}}$ is the tangential reaction required to prevent relative sliding. A three-dimensional Abaqus/Standard model was then used to evaluate these reactions as the plait was progressively tensioned.

The simulations showed that, after inter-strand contact was established, both normal and tangential contact reactions increased with axial deformation, while the normal reaction increased proportionally more rapidly during the initial contact regime. Consequently, μ_{crit} decreased from approximately 4.28 at 0.894% global tensile strain to 1.93 at 1.078% and 1.78 at 1.278%. At larger strains, additional contact regions and localized plasticity developed, producing a transition to a multi-contact post-yield response.

These results indicate that frictional self-locking in braided metallic strands is governed not simply by the number of crossings, but by the evolving balance between normal geometric confinement and the tangential force required to suppress sliding. The proposed framework provides a quantitative basis for relating interlaced geometry, contact mechanics, and self-locking behavior in braided wire structures.

Keywords: braided steel wire strand, frictional self-locking, contact mechanics, braid geometry, finite element analysis

*Beijing 101 Middle School.

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1 Introduction

Steel wires, strands, and ropes are widely used in engineering applications because they combine high tensile strength with structural flexibility and efficient load-bearing capability. Individual high-strength steel wires are used in prestressed civil structures, automotive suspension systems, and power-transmission conductors [RJK⁺24, SKY⁺19, JRP24]. When multiple wires are assembled into strands and ropes, the resulting structures are further employed in lifting, bridge construction, and mining applications [YY25, XS24, RVB05]. Their mechanical behavior is influenced not only by the properties of the constituent wires, but also by the geometry and topology through which the wires are assembled.

One of the most widely used metallic strand structures is the conventional 1 + 6 configuration, in which six outer wires are helically laid around a straight central wire [Li25, Cos90]. Although this configuration is compact and convenient for industrial manufacture, its helical geometry introduces several mechanical limitations. Under axial tension, the curved paths of the outer wires tend to straighten, producing tension-torsion coupling and a tendency for the strand to rotate or unlay [Cos90]. Relative sliding can also occur between neighboring wires during loading, while non-uniform inter-wire contact conditions may lead to uneven stress distributions, fretting, and localized damage [CP83, Cha95]. These effects indicate that strand performance is strongly affected by geometric constraint and inter-wire contact.

This motivates the consideration of interlaced structures inspired by traditional braids and knots. Such structures obtain stability through repeated over-under crossings of their constituent strands [Ash44, HGB69]. These crossings constrain relative motion and can generate contact reactions and frictional resistance during loading. The mechanics of rope-like systems have long been recognized as strongly dependent on geometric configuration and inter-element interaction [MK87]. More recent numerical studies of braided ropes have also shown that braid geometry, contact, friction, and relative strand motion are closely coupled [LSZ⁺23, DLJ⁺24].

However, the presence of more crossings or contact points does not necessarily imply stronger mechanical locking. The effectiveness of frictional interaction depends on the local contact forces and on the tendency of neighboring strands to slide relative to one another. Therefore, a more useful description of braided structures requires linking braid geometry to local contact mechanics rather than treating braiding complexity alone as an indicator of performance.

In this study, three alternative seven-wire braided configurations inspired by classical three- and four-strand braids are first fabricated and compared experimentally with the conventional 1 + 6 strand using dynamic tensile testing. The most promising topology is then idealized as a symmetric three-strand plait for further theoretical and numerical investigation. An analytical geometric and contact-mechanics model is developed, and finite-element analysis is used to study the evolution of the inter-wire contact behavior under axial loading.

The remainder of this paper is organized as follows. Section 2 introduces the conventional 1+6 strand, the knot-inspired design strategy, and the proposed braided configurations. Section 3 presents the preliminary dynamic tensile experiments. Section 4 develops the theoretical model for frictional self-locking in an ideal three-strand plait. Section 5 presents the finite-element implementation and analyzes the evolution of contact reactions and frictional self-locking under axial loading. Section 6 summarizes the principal findings and limitations of the study.

2 Braiding Configurations and Motivation

Steel wires are commonly assembled into strands and ropes through helical laying. Among the most widely used strand structures is the conventional 1 + 6 configuration, in which six outer wires are helically wound around a straight central wire. Although this configuration provides a compact and highly symmetric structure, its helical geometry also introduces several mechanical limitations under axial tension. These limitations motivated the exploration of alternative wire arrangements based on interlacing rather than purely helical laying. Inspired by traditional braiding and knotting structures, three seven-wire configurations were therefore designed and experimentally compared with the conventional 1 + 6 strand. The purpose of this comparison was to identify a promising braiding topology for subsequent theoretical investigation of frictional self-locking.

2.1 Conventional 1 + 6 Strand and Its Limitations

The conventional 1 + 6 strand consists of one straight central wire surrounded by six identical outer wires. The six outer wires are helically laid around the central wire with the same pitch and lay length, producing a compact and approximately axisymmetric structure [Cos90]. This arrangement is widely adopted because of its simple geometry, manufacturing efficiency, and relatively uniform distribution of the outer wires.

However, the same helical geometry also produces several limitations under axial tensile loading. One of the most important is tension–torsion coupling. When a helically laid wire is stretched, its curved path tends to straighten. Consequently, axial extension is naturally coupled with a tendency for the strand to rotate or unlay. If end rotation is permitted, the strand may gradually rotate and lose part of its initial tightness. If the ends are rotationally constrained, the tendency to unlay instead generates additional torsional reactions and local stress concentrations [Cos90].

A second limitation is inter-wire sliding during the initial stage of loading. Since the individual wires are not geometrically interlocked, load transfer between neighboring wires depends strongly on their contact interactions. Local relative motion can therefore occur before a fully constrained load-bearing state is established, producing nonlinear behavior in the initial tensile response [CP83]. In addition, differences in local contact conditions can lead to non-uniform stress distributions and localized contact damage [Cha95].

These observations suggest that some limitations of conventional wire strands originate not only from the material properties of the wires, but also from the geometric topology through which the wires are assembled. This motivated the question of whether a different topology could suppress unlaying by introducing mutual geometric constraint between the load-bearing elements.

2.2 Knot-Inspired Alternative Braiding Strategy

Traditional knots and braids achieve structural stability through repeated interlacing of their constituent strands. Rather than allowing neighboring elements to follow nearly parallel helical paths, an interlaced topology repeatedly forces different strands to cross and constrain one another. Similar geometric principles appear in many traditional knotting and braiding techniques, including Chinese knots and classical three- and four-strand braids [Ash44, HGB69].

This observation motivated the use of an interlaced topology for metallic wire strands. Under axial tension, relative motion between neighboring strands is not determined solely by their individual axial deformation; it is also constrained by the surrounding crossings. These crossings can generate normal contact reactions, and the resulting frictional capacity may resist relative sliding. Therefore, an interlaced strand may possess a form of frictional self-locking that is absent, or substantially weaker, in a purely helically laid structure.

At this stage, this argument was treated as a design hypothesis rather than a complete mechanical theory. The immediate objective was therefore to use geometric interlocking to construct several candidate wire-strand configurations for preliminary comparison; the underlying contact mechanics are examined quantitatively later.

2.3 Proposed Seven-Wire Configurations

To permit a direct preliminary comparison with the conventional 1 + 6 strand, all proposed configurations were designed using the same total number of seven steel wires. The objective was therefore not to change the amount of material, but to reorganize the same number of wires into different topologies.

Three configurations were developed: a three-strand braided seven-wire configuration, a four-strand braided seven-wire configuration, and a four-strand braid with three straight core wires. These configurations progressively explore different combinations of interlacing, geometric confinement, and direct axial load-bearing.

2.3.1 Three-Strand Braided Seven-Wire Configuration

The first proposed topology is based on the classical three-strand braid. In a conventional three-strand braid, three strands repeatedly exchange their lateral positions through an alternating over–under sequence. This repeated crossing produces an interlaced structure in which each strand is geometrically constrained by the other two.

For the preliminary seven-wire design, the seven individual wires were first divided into three bundles containing 2, 3, and 2 wires, respectively. These three bundles were then interlaced using the standard three-strand braiding sequence. The resulting topology is illustrated schematically in Fig. 1, while the corresponding fabricated steel-wire specimen is shown in Fig. 2. This arrangement is referred to here as the three-strand braided seven-wire configuration.

The 2-3-2 grouping was introduced specifically to preserve a total of seven wires while allowing the topology to be compared directly with the conventional 1 + 6 strand. The resulting configuration contains repeated crossings between the three load-bearing bundles and therefore provides substantially different geometric constraints from the conventional helically laid structure.

The later theoretical analysis idealizes this topology as a symmetric three-strand plait, as discussed in Section 4.



Figure 1: Schematic illustration of 3B



Figure 2: Fabricated specimen of 3B

2.3.2 Four-Strand Braided Seven-Wire Configuration

The second candidate configuration was developed from a classical four-strand braid. Compared with a three-strand braid, the four-strand topology introduces a different sequence of crossings and a greater degree of geometric interlacing. The strands alternately pass over and under adjacent strands, producing a repeating four-strand pattern.

To construct a seven-wire specimen, the wires were divided into four bundles: three bundles containing two wires each and one bundle containing a single wire. The four bundles were then interlaced according to the four-strand braiding sequence. The resulting topology is shown schematically in Fig. 3, while the corresponding fabricated steel-wire specimen is shown in Fig. 4. This configuration is referred to as the four-strand braided seven-wire configuration.

The design was intended to examine whether increasing the degree of interlacing could provide stronger geometric confinement and resistance to relative sliding. At the same time, dividing the same seven wires among four strands reduces the number of individual wires contained in each load-bearing bundle. The competition between these two effects could not be determined from geometry alone and was therefore evaluated experimentally.



Figure 3: Schematic illustration of 4B



Figure 4: Fabricated specimen of 4B

2.3.3 Four-Strand Braid with Three Straight Core Wires

The third configuration was designed to combine direct axial load-bearing with an interlaced outer structure. In this arrangement, three wires remain straight along the strand axis and form an internal core, while the remaining four wires are braided around them using a four-strand pattern.

The three straight core wires were intended to carry axial tension efficiently, while the braided outer wires were introduced to provide lateral confinement and geometric stabilization. This configuration therefore represents a hybrid strategy between a conventional axial core and a braided surrounding structure.

The resulting four-strand braid with three straight core wires is illustrated schematically in Fig. 5, while the corresponding fabricated steel-wire specimen is shown in Fig. 6. Unlike the fully braided configurations, the central wires are not directly incorporated into the same interlacing network as the outer wires. The effectiveness of this hybrid load-transfer mechanism was subsequently evaluated through dynamic tensile testing.



Figure 5: Schematic illustration of 4C



Figure 6: Fabricated specimen of 4C

2.4 Role of the Preliminary Configurations

The three proposed configurations were not intended at this stage to provide a complete optimization of braided metallic strands. Instead, they served as preliminary candidate topologies for determining whether interlaced wire arrangements could provide competitive mechanical performance compared with the conventional 1 + 6 strand.

Because all four configurations contained the same total number of seven wires, their tensile responses could be compared without changing the total number of load-bearing elements. The subsequent experiments were therefore used as a screening step to identify which topology was most promising for further investigation.

3 Preliminary Dynamic Tensile Experiments

Preliminary dynamic tensile experiments were conducted using a Split Hopkinson Tensile Bar (SHTB) system to compare the tensile responses of the proposed braided configurations with the conventional 1 + 6 strand.

All specimens contained seven pearlitic steel wires. The conventional 1 + 6 strand and the three proposed braided configurations were subjected to dynamic tensile loading under comparable experimental conditions. Because conventional grips were found to be unsuitable for reliably clamping fine braided wire strands, a plate-type gripping fixture was specially designed for the experiments.

3.1 Split Hopkinson Tensile Bar System

The Split Hopkinson Tensile Bar system was used to investigate the dynamic tensile behavior of the different wire-strand configurations. The Hopkinson-bar technique determines the dynamic stress-strain response of a specimen from elastic stress waves propagating through an incident bar and a transmission bar.

3.1.1 Working Principle

The theoretical basis of the SHTB method is one-dimensional elastic-wave propagation in a slender bar. Its analysis is based on two principal assumptions. First, the cross-section of the bar remains approximately plane during longitudinal wave propagation, allowing the stress and strain fields in the bar to be treated as one-dimensional. Second, the constitutive response of the bar material is assumed to be known so that the propagating strain signals can be related to stress.

Let $\varepsilon(x, t)$ denote the longitudinal strain, $v(x, t)$ the particle velocity, ρ the material density, and $\sigma(x, t)$ the axial stress. The longitudinal motion of the bar can be described by the continuity equation

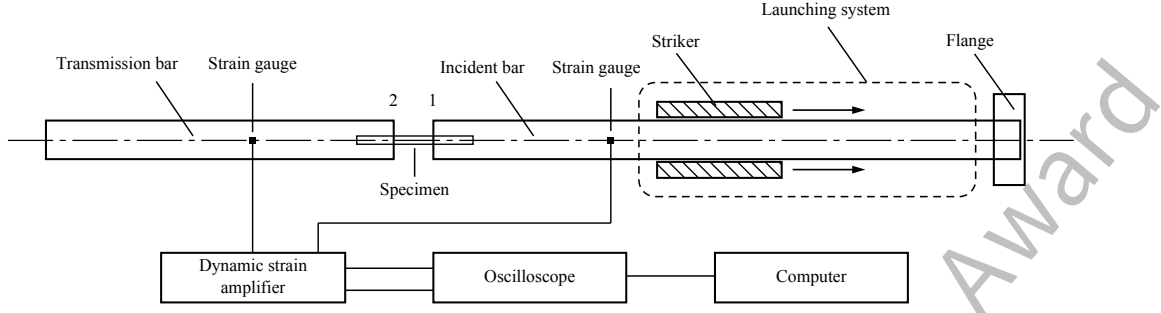


Figure 7: Schematic structure of the Split Hopkinson Tensile Bar system.

$$\frac{\partial \varepsilon(x, t)}{\partial t} = \frac{\partial v(x, t)}{\partial x}, \quad (1)$$

the equation of motion

$$\rho \frac{\partial v(x, t)}{\partial t} = \frac{\partial \sigma(x, t)}{\partial x}, \quad (2)$$

and the constitutive relation

$$\sigma = \sigma(\varepsilon). \quad (3)$$

Combining these relations gives the one-dimensional wave equation

$$\frac{\partial^2 \varepsilon(x, t)}{\partial t^2} = c_0^2 \frac{\partial^2 \varepsilon(x, t)}{\partial x^2}, \quad (4)$$

where the elastic longitudinal wave velocity is

$$c_0 = \sqrt{\frac{E}{\rho}}. \quad (5)$$

The general displacement solution can be expressed as the superposition of waves traveling in opposite directions,

$$u(x, t) = F_1(x + c_0 t) + F_2(x - c_0 t). \quad (6)$$

During an SHTB test, three strain signals are measured: the incident wave $\varepsilon_I(t)$, the reflected wave $\varepsilon_R(t)$, and the transmitted wave $\varepsilon_T(t)$. Using these signals, the dynamic response of the specimen can be calculated.

The specimen stress is given by

$$\sigma_s(t) = \frac{E_b A_b}{2A_s} [\varepsilon_I(t) + \varepsilon_R(t) + \varepsilon_T(t)], \quad (7)$$

where E_b and A_b are the Young's modulus and cross-sectional area of the Hopkinson bar, and A_s is the effective cross-sectional area of the specimen.

The specimen strain is

$$\varepsilon_s(t) = -\frac{2c_0}{L_s} \int_0^t \varepsilon_R(\tau) d\tau, \quad (8)$$

and the corresponding strain rate is

$$\dot{\varepsilon}_s(t) = -\frac{2c_0}{L_s} \varepsilon_R(t), \quad (9)$$

where L_s is the initial specimen length.

The characteristics of the incident pulse are controlled by the striker. In particular, the pulse amplitude can be approximated by

$$\sigma_i = \frac{\rho c_0 v_0}{2}, \quad (10)$$

where v_0 is the striker velocity. The pulse duration is related to the striker length L by

$$\tau = \frac{2L}{c_0}. \quad (11)$$

Therefore, the loading rate can be adjusted by controlling the striker velocity and striker length.

3.1.2 Validity for Braided Wire Strands

The SHTB method is most commonly applied to continuous solid specimens. The specimens in the present study, however, consist of multiple discrete steel wires assembled into strands. Therefore, the applicability of the method to the braided specimens was examined by considering stress equilibrium, strain-rate stability, and the consistency of the resulting stress-strain response.

A fundamental requirement is that the specimen reaches approximate axial stress equilibrium during the principal deformation stage. Stress waves entering the specimen undergo repeated transmission and reflection between the specimen-bar interfaces. If the specimen is sufficiently short and the pulse rise time sufficiently long, repeated wave reverberations allow the axial stress distribution to approach equilibrium.

The material parameters used in the original equilibrium analysis are summarized in Table 1.

Table 1: Material parameters used in the SHTB stress-equilibrium analysis.

Material	Young's modulus E (GPa)	Density ρ (kg/m ³)	Wave speed c_0 (m/s)
Hopkinson bar	200	7689	5100
Steel strand	210	7800	5189

Taking the conventional 1 + 6 strand as a representative specimen, its cross-section was approximated as a circular region with a radius of approximately 0.3 mm. The characteristic time for an elastic wave to traverse the specimen was compared with the rise time of the incident pulse. The resulting equilibrium parameters are summarized in Table 2.

Table 2: Quantitative verification of stress equilibrium for the 1 + 6 strand specimen.

Parameter	Value	Interpretation
Specimen radius	0.3 mm	Equivalent circular cross-section
One reflection time	0.862 μ s	Time for the stress wave to traverse the specimen
Incident pulse rise time	30 μ s	Experimental measurement
Number of reflections	34.8	30/0.862
Equilibrium criterion	$k \geq 18$	Requirement satisfied
Maximum specimen length	8.7 mm	Upper bound for stress equilibrium

The estimated number of wave reflections during the incident-pulse rise time was therefore greater than the required equilibrium threshold. This indicates that a sufficiently short strand specimen can reach approximate axial stress equilibrium during the relevant loading interval.

A second requirement for dynamic tensile testing is that the specimen strain rate remain approximately stable during the main deformation stage. From

$$\dot{\epsilon}_s(t) = -\frac{2c_0}{L_s} \epsilon_R(t), \quad (12)$$

a relatively stable reflected strain signal corresponds to an approximately stable specimen strain rate. The true strain can then be calculated by integration,

$$\varepsilon_s(t) = \int_0^t \dot{\varepsilon}_s(\tau) d\tau, \quad (13)$$

while the corresponding dynamic stress is obtained from the measured stress-wave signals. Together, these checks support the use of the SHTB system for comparative dynamic testing of the braided wire-strand configurations.

3.2 Design of the Plate-Type Gripping Fixture

Reliable clamping is essential for tensile testing of fine metallic wires and braided strands. During preliminary trials, conventional gripping methods were found to be unsuitable because they required precise control of the clamping force. If the clamping force was excessive, local indentation could damage the fine wires and lead to premature failure. If it was insufficient, the specimen could slip during loading. Asymmetric clamping could further cause the specimen to fracture close to one grip rather than within the effective test section.

To address these problems, a plate-type gripping fixture was designed specifically for the dynamic tensile tests. The fixture consists of two rectangular steel plates, each containing a cylindrical groove along its centerline to locate the wire strand. The two plates are compressed together by four bolts positioned near the corners. One plate is connected to the Hopkinson bar through a threaded cylindrical extension.

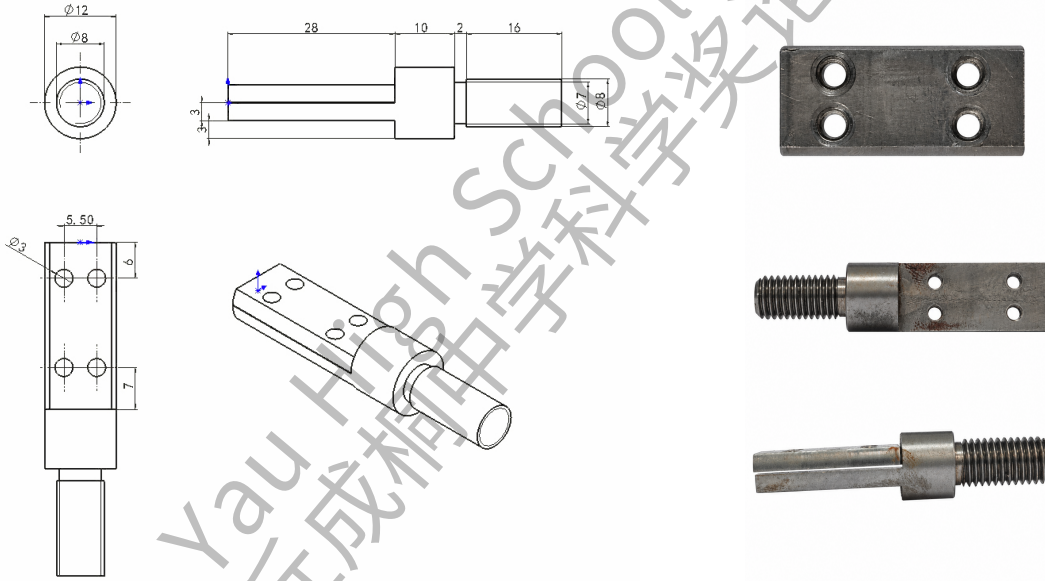


Figure 8: Schematic design

Figure 9: Fabricated fixture

The cylindrical groove helps maintain coaxial alignment between the specimen and the loading bars. The four-bolt arrangement provides approximately symmetric clamping on both sides of the specimen. A torque wrench can be used to control the bolt-tightening condition and improve repeatability between tests.

The groove diameter can also be adjusted according to the specimen dimensions. In the preliminary experiments, a groove diameter of approximately 0.5 mm was used for the seven-wire strands fabricated from 200 μm diameter pearlitic steel wires.

The effectiveness of the fixture was later confirmed by the observation that the tested specimens generally fractured within the effective specimen section rather than immediately at the gripping ends.

3.3 Experimental Procedure

Before each test, the SHTB system was prepared and aligned. The striker, incident bar, and transmission bar were adjusted so that their longitudinal axes were approximately coaxial. The pressure regulator, dynamic strain amplifier, velocity measurement system, and digital oscilloscope were then activated and checked.

The specimen was mounted between the incident and transmission bars using the plate-type gripping fixtures. A pulse shaper consisting of a small rubber sheet was attached to the incident-bar interface to reduce high-frequency oscillations generated by the impact and produce a smoother incident pulse.

The striker was inserted into the launching barrel and accelerated by compressed gas to the prescribed velocity. Upon impact with the incident bar, an elastic stress wave propagated toward the specimen. At the specimen interface, part of the wave was reflected back into the incident bar and the remaining part was transmitted through the specimen into the transmission bar.

Strain gauges mounted on the bars recorded the incident, reflected, and transmitted strain waves. The signals were amplified and recorded by a digital oscilloscope for subsequent processing.

After each test, the deformed or fractured specimen was collected and visually inspected. The recorded wave signals were then processed using the SHTB relations described above to obtain the specimen stress, strain, and strain-rate histories.

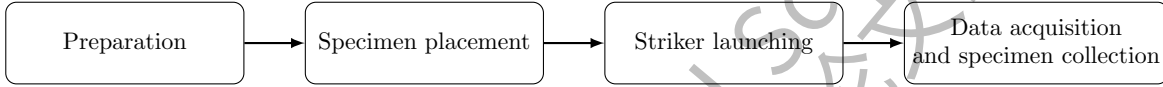


Figure 10: Experimental procedure for the dynamic tensile tests.

3.4 Experimental Results

Dynamic tensile tests were conducted on four seven-wire configurations: the conventional 1 + 6 strand, the three-strand braided seven-wire configuration, the four-strand braided seven-wire configuration, and the four-strand braid with three straight core wires.

All configurations were fabricated using pearlitic steel wires with a diameter of approximately $200\ \mu\text{m}$. Representative post-fracture specimens showed that failure occurred within the effective specimen region rather than at the gripping ends, providing further evidence that the plate-type fixture provided sufficiently stable clamping.



Figure 11: Four-strand braided specimen



Figure 12: Four-strand braid with straight core wires

Figure 13 compares the dynamic true stress-strain responses of the four configurations at a strain rate of approximately $2000\ \text{s}^{-1}$.

The peak stress of each curve characterizes the maximum tensile stress sustained by the strand, while the termination point of the curve indicates the maximum strain reached before failure. The initial slope provides an indication of the effective axial stiffness and is also sensitive to the initial compactness of the braided structure.

The three-strand braided configuration achieved a maximum stress close to that of the conventional 1 + 6 strand while reaching a larger maximum true strain. Among the newly proposed configurations, it exhibited the highest peak stress.

The four-strand braided configuration also showed a peak stress comparable to that of the conventional strand and exhibited relatively large deformation before failure. In contrast, the four-strand braid containing three straight core wires showed both a lower maximum stress and a lower maximum strain.

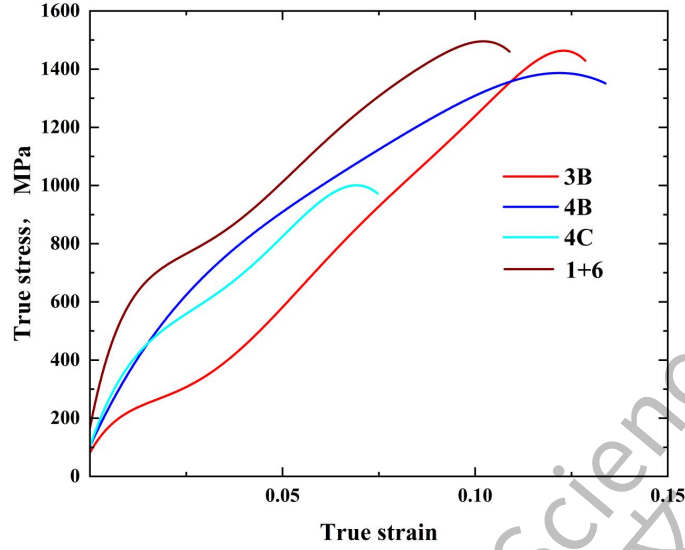


Figure 13: Dynamic true stress-strain responses of the conventional and proposed seven-wire strand configurations at a strain rate of approximately 2000 s^{-1} .

Additional tests were conducted on the four-strand braided configuration at strain rates of approximately 1775 s^{-1} and 1969 s^{-1} . Noticeable differences were observed between the corresponding stress-strain curves.

The specimen exhibiting the steeper initial slope also reached a higher maximum stress and larger failure strain. Visual comparison suggested that this specimen had been braided more tightly. The other specimen exhibited a more pronounced initial low-stiffness region, which was consistent with a looser initial configuration that required additional deformation before the strands became fully engaged.

This observation suggests that the mechanical response of braided wire structures is sensitive not only to their topology, but also to geometric parameters controlling how tightly the strands are interlaced. This observation further motivates a quantitative investigation of how inter-strand contact evolves as the braid is progressively tensioned.

3.5 Comparison, Interpretation, and Selection of the Three-Strand Plait

Among the proposed configurations, the three-strand braid showed the most favorable combination of peak stress and deformation capacity, with a peak stress comparable to that of the conventional 1 + 6 strand but a larger maximum strain.

The four-strand braided configuration also showed promising deformation capacity. However, its response exhibited greater specimen-to-specimen variation. One possible reason is that dividing seven wires into four bundles produces smaller and less uniform load-bearing groups. In addition, the four-strand braiding procedure is more difficult to manufacture consistently by hand, which can introduce variations in compactness and local geometry.

The configuration containing three straight core wires showed weaker overall performance. Although the straight wires were intended to carry axial load efficiently, they were not incorporated into the same interlacing network as the surrounding braided wires. Their mechanical interaction with the outer braid was therefore comparatively weak, limiting cooperative load transfer between the core and outer structure.

The experiments therefore suggest that the three-strand braided topology is particularly promising. However, the mechanism responsible for this behavior cannot be determined from the global stress-strain curves alone.

The improved behavior of braided configurations cannot be attributed simply to a larger number of contacts. Under Coulomb friction, the mechanical capacity of a contact depends on the normal reaction N_c

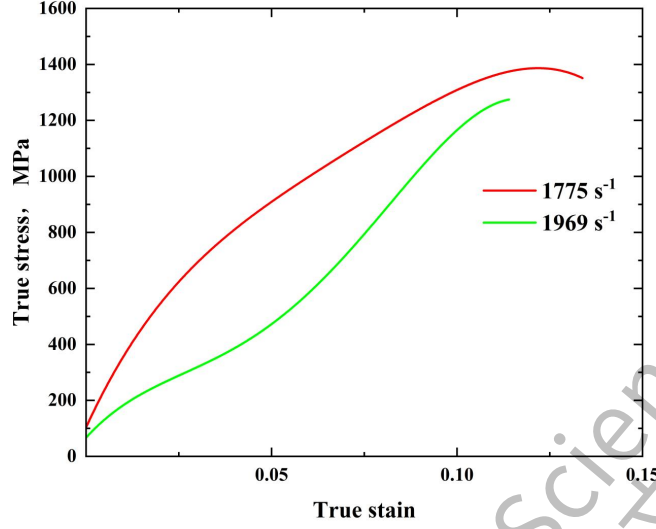


Figure 14: Dynamic true stress-strain responses of the four-strand braided configuration under different strain rates.

and the tangential reaction Q_c^{req} required to prevent relative sliding. These quantities depend on the braid geometry and evolve during loading.

For this reason, the three-strand braided topology was selected for further theoretical and numerical investigation.

The experimental three-strand specimen used an asymmetric 2-3-2 grouping so that it contained the same total of seven wires as the conventional 1+6 strand. Because this introduces unequal bundle sizes and additional internal load-distribution effects, the subsequent theory does not attempt to reproduce the complete experimental specimen.

Instead, the topology is idealized as a perfectly symmetric three-strand plait composed of three identical strands. This isolates the mechanics of repeated strand crossing from bundle asymmetry and manufacturing irregularity.

Accordingly, the following analysis focuses on how the geometry of an ideal three-strand plait determines the normal contact reactions and tangential frictional demand required for self-locking under axial tension.

4 Theoretical Model of Frictional Self-Locking in a Three-Strand Plait

This section develops an analytical framework linking the geometry of an ideal three-strand plait to its local contact reactions and Coulomb frictional locking condition.

4.1 Model Assumptions and Symmetry Reduction

The three-strand plait is idealized as consisting of three identical circular steel wires. The following assumptions are adopted in the analytical model:

1. The three wires have identical radius R , material properties, and undeformed geometric form.
2. The material is homogeneous and isotropic.
3. The plait is perfectly periodic away from the end fixtures.

4. The total externally applied tensile force F_{total} acts along the global plait axis, defined as the z -axis.
5. Owing to the symmetry of the ideal three-strand plait, the externally applied axial load is equally distributed among the three wires at the boundary.
6. End effects and manufacturing imperfections are neglected in the analytical model.
7. Repeated crossings in the ideal plait are treated as mechanically symmetry-equivalent under axial loading. Therefore, one representative crossing is sufficient for the local contact analysis.
8. Each localized crossing is represented analytically by the closest-point pair of the two wire centerlines. This local construction is used only to define the contact geometry and contact frame; the finite-element model subsequently resolves contact between the full three-dimensional wire surfaces.

Accordingly, the mechanical problem is reduced according to

$$\boxed{\text{whole three-strand plait} \rightarrow \text{one representative wire-wire contact.}}$$

The total external load vector is written as

$$\mathbf{F}_{\text{total}} = F_{\text{total}} \mathbf{e}_z,$$

where $\mathbf{e}_z$ is the unit vector along the global z -axis.

Under the equal-load-sharing assumption, the boundary load applied to each wire is

$$\boxed{\mathbf{F}_i = \frac{F_{\text{total}}}{3} \mathbf{e}_z, \quad i = 0, 1, 2.}$$

It should be emphasized that this relation describes the symmetric *boundary loading*. It does not imply that the internal force field within a curved wire is everywhere equal to $F_{\text{total}}/3$.

4.2 Geometric Model of the Three-Strand Plait

4.2.1 Periodic Centerlines

For the present analytical model, the three-strand plait is represented by three periodic centerlines with equal phase separation. To express the geometry in physical length coordinates, define the spatial wavenumber

$$\boxed{k = \frac{2\pi}{P}},$$

where P is the axial period of one labelled wire.

The phase shift of wire i is defined as

$$\boxed{\phi_i = \frac{2\pi i}{3}, \quad i = 0, 1, 2.}$$

Hence,

$$\phi_0 = 0, \quad \phi_1 = \frac{2\pi}{3}, \quad \phi_2 = \frac{4\pi}{3}.$$

The centerline of wire i is represented by

$$\boxed{\mathbf{r}_i(z) = \begin{pmatrix} a \sin(kz + \phi_i) \\ b \sin[2(kz + \phi_i)] \\ z \end{pmatrix}.}$$

Here,

- a controls the lateral amplitude of the plait;
- b controls the out-of-plane over-under displacement;
- P determines the axial periodicity;
- z denotes the global axial coordinate.

The first coordinate describes the alternating lateral motion of each wire, whereas the second coordinate produces the over-under interlacing required for a three-dimensional plait.

The parameters a , b , and P characterize the idealized geometry of the plait. The sinusoidal representation is used here to establish the geometric quantities governing local contact, rather than as an exact reconstruction of either the experimental specimen or the finite-element geometry.

4.2.2 Local Wire Tangents

Differentiating the centerline with respect to the axial coordinate gives

$$\frac{d\mathbf{r}_i}{dz} = \begin{pmatrix} ak \cos(kz + \phi_i) \\ 2bk \cos[2(kz + \phi_i)] \\ 1 \end{pmatrix}.$$

Because z is not the arc-length coordinate, this derivative is not generally a unit vector. The local unit tangent vector is therefore

$$\mathbf{t}_i(z) = \frac{\frac{d\mathbf{r}_i}{dz}}{\left\| \frac{d\mathbf{r}_i}{dz} \right\|}.$$

Explicitly,

$$\mathbf{t}_i(z) = \frac{\begin{pmatrix} ak \cos(kz + \phi_i) \\ 2bk \cos[2(kz + \phi_i)] \\ 1 \end{pmatrix}}{\sqrt{1 + a^2 k^2 \cos^2(kz + \phi_i) + 4b^2 k^2 \cos^2[2(kz + \phi_i)]}}.$$

Thus, both the magnitude and direction of the local geometric inclination of each wire can be obtained directly from the prescribed centerline.

4.3 Representative Point-to-Point Contact

4.3.1 Representative Crossing

Because all ideal crossings are related by the periodic and permutation symmetries of the three-strand plait, a representative crossing between wire 0 and wire 1 is selected.

Their centerlines are

$$\mathbf{r}_0(z) = \begin{pmatrix} a \sin(kz) \\ b \sin(2kz) \\ z \end{pmatrix},$$

and

$$\mathbf{r}_1(z) = \begin{pmatrix} a \sin\left(kz + \frac{2\pi}{3}\right) \\ b \sin\left(2kz + \frac{4\pi}{3}\right) \\ z \end{pmatrix}.$$

The projected crossing centre is first determined by requiring the two wires to have the same lateral coordinate,

$$x_0(z_c) = x_1(z_c).$$

Therefore,

$$\sin(kz_c) = \sin\left(kz_c + \frac{2\pi}{3}\right).$$

Using the trigonometric identity

$$\sin A - \sin B = 2 \cos\left(\frac{A+B}{2}\right) \sin\left(\frac{A-B}{2}\right),$$

one obtains

$$\cos\left(kz_c + \frac{\pi}{3}\right) = 0.$$

The first representative solution is therefore

$$kz_c = \frac{\pi}{6},$$

or

$$z_c = \frac{P}{12}.$$

The projected crossing centre z_c does not necessarily coincide with the true three-dimensional closest-point pair of the two wire centerlines.

Due to the local mirror symmetry of the selected crossing, the two closest centerline points are written as

$$z_0^* = z_c - \Delta, \quad z_1^* = z_c + \Delta,$$

where Δ is the axial offset between the projected crossing centre and the true closest-point locations. For convenience, define

$$\chi = \frac{\pi}{3} - 2k\Delta.$$

The corresponding centerline positions can then be written as

$$\mathbf{r}_0^* = \begin{pmatrix} x_c \\ b \sin \chi \\ z_c - \Delta \end{pmatrix},$$

and

$$\mathbf{r}_1^* = \begin{pmatrix} x_c \\ -b \sin \chi \\ z_c + \Delta \end{pmatrix},$$

where

$$x_c = a \sin\left(\frac{\pi}{6} - k\Delta\right).$$

The separation vector from wire 0 to wire 1 is therefore

$$\mathbf{d}_c = \mathbf{r}_1^* - \mathbf{r}_0^* = \begin{pmatrix} 0 \\ -2b \sin \chi \\ 2\Delta \end{pmatrix}.$$

4.3.2 Closest-Point Condition

For a local closest-point representation of two non-parallel wire centerlines, the line connecting a closest-point pair must be perpendicular to the local tangent of each centerline. Hence,

$$\boxed{\mathbf{d}_c \cdot \mathbf{t}_0^* = 0, \quad \mathbf{d}_c \cdot \mathbf{t}_1^* = 0.}$$

Since normalization does not affect orthogonality, the non-normalized centerline derivatives may be used instead.

At the two candidate closest points,

$$\left. \frac{d\mathbf{r}_0}{dz} \right|_{z_0^*} = \begin{pmatrix} A \\ C \\ 1 \end{pmatrix},$$

and

$$\left. \frac{d\mathbf{r}_1}{dz} \right|_{z_1^*} = \begin{pmatrix} -A \\ C \\ 1 \end{pmatrix},$$

where

$$A = ak \cos\left(\frac{\pi}{6} - k\Delta\right),$$

and

$$C = 2bk \cos \chi.$$

Applying the first orthogonality condition gives

$$\begin{pmatrix} 0 \\ -2b \sin \chi \\ 2\Delta \end{pmatrix} \cdot \begin{pmatrix} A \\ C \\ 1 \end{pmatrix} = 0.$$

Therefore,

$$-2bC \sin \chi + 2\Delta = 0.$$

Substituting

$$C = 2bk \cos \chi$$

gives

$$\Delta = 2b^2 k \sin \chi \cos \chi.$$

Using

$$2 \sin \chi \cos \chi = \sin(2\chi),$$

the result becomes

$$\boxed{\Delta = b^2 k \sin(2\chi).}$$

Since

$$\chi = \frac{\pi}{3} - 2k\Delta,$$

the closest-point offset satisfies the implicit equation

$$\Delta = b^2 k \sin\left(\frac{2\pi}{3} - 4k\Delta\right).$$

This equation is generally transcendental and can be solved numerically once b and P are known.

The second orthogonality condition gives the same equation because of the local mirror symmetry of the representative crossing.

The orthogonality conditions identify stationary points of the centerline-distance function. Among possible solutions, the physically relevant solution is selected as the one corresponding to a *local minimum* of the wire-wire centerline distance within the representative crossing region.

4.3.3 Finite-Radius Contact Compatibility

The minimum centerline distance is

$$d_c = \|\mathbf{d}_c\|.$$

Using the expression for $\mathbf{d}_c$,

$$d_c = \sqrt{4b^2 \sin^2 \chi + 4\Delta^2}.$$

Hence,

$$d_c = 2\sqrt{b^2 \sin^2 \chi + \Delta^2}.$$

For two identical circular wires of radius R , geometrical contact requires

$$d_c = 2R.$$

Therefore,

$$b^2 \sin^2 \chi + \Delta^2 = R^2.$$

The normal gap may equivalently be defined as

$$g_n = d_c - 2R.$$

Thus,

$$g_n > 0$$

corresponds to separated wires, whereas

$$g_n = 0$$

corresponds to geometrical contact.

A negative value of g_n would represent geometric overlap in the idealized centerline-plus-radius representation and must be resolved by contact deformation in the mechanical model.

4.4 Local Contact Coordinate System

4.4.1 Contact Normal

The unit contact normal is defined by the direction of the closest-point separation vector,

$$\mathbf{n}_c = \frac{\mathbf{d}_c}{\|\mathbf{d}_c\|}.$$

Substituting the representative-contact geometry gives

$$\mathbf{n}_c = \frac{\begin{pmatrix} 0 \\ -b \sin \chi \\ \Delta \end{pmatrix}}{\sqrt{b^2 \sin^2 \chi + \Delta^2}}.$$

Under the geometrical contact condition

$$b^2 \sin^2 \chi + \Delta^2 = R^2,$$

this becomes

$$\mathbf{n}_c = \begin{pmatrix} 0 \\ -\frac{b \sin \chi}{R} \\ \frac{\Delta}{R} \end{pmatrix}.$$

The positive normal direction is defined from wire 0 toward wire 1. Because the two centerline locations form a closest-point pair,

$$\mathbf{n}_c \cdot \mathbf{t}_0^* = 0, \quad \mathbf{n}_c \cdot \mathbf{t}_1^* = 0.$$

For a non-parallel crossing, the same contact normal may therefore be expressed as

$$\mathbf{n}_c = \pm \frac{\mathbf{t}_0^* \times \mathbf{t}_1^*}{\|\mathbf{t}_0^* \times \mathbf{t}_1^*\|}.$$

The sign is selected so that the vector points from wire 0 toward wire 1.

4.4.2 Physical Surface Contact Point

The closest-point locations $\mathbf{r}_0^*$ and $\mathbf{r}_1^*$ lie on the centerlines of the two wires rather than on their physical surfaces.

For identical wires of radius R , the physical contact point is

$$\mathbf{p}_c = \mathbf{r}_0^* + R\mathbf{n}_c.$$

Equivalently,

$$\mathbf{p}_c = \mathbf{r}_1^* - R\mathbf{n}_c.$$

The two expressions coincide when

$$\|\mathbf{r}_1^* - \mathbf{r}_0^*\| = 2R.$$

4.4.3 Contact Tangent Plane

The local tangent plane at the representative contact is the plane perpendicular to $\mathbf{n}_c$ and passing through $\mathbf{p}_c$.

It is therefore described by

$$(\mathbf{x} - \mathbf{p}_c) \cdot \mathbf{n}_c = 0.$$

For later force and displacement decomposition, define the tangential projection tensor

$$\mathbf{P}_T = \mathbf{I} - \mathbf{n}_c \otimes \mathbf{n}_c,$$

where $\mathbf{I}$ is the three-dimensional identity tensor.

For any vector $\mathbf{a}$, its normal component is

$$\mathbf{a}_N = (\mathbf{a} \cdot \mathbf{n}_c) \mathbf{n}_c,$$

while its component within the contact tangent plane is

$$\mathbf{a}_T = \mathbf{P}_T \mathbf{a} = (\mathbf{I} - \mathbf{n}_c \otimes \mathbf{n}_c) \mathbf{a}.$$

Thus,

$$\mathbf{a} = \mathbf{a}_N + \mathbf{a}_T.$$

4.4.4 Local Crossing Angle

The local crossing angle provides a consistency check for the point-to-point contact approximation.

Define

$$\alpha_c = \cos^{-1}(|\mathbf{t}_0^* \cdot \mathbf{t}_1^*|).$$

Using

$$\mathbf{t}_0^* = \frac{(A, C, 1)}{\sqrt{A^2 + C^2 + 1}},$$

and

$$\mathbf{t}_1^* = \frac{(-A, C, 1)}{\sqrt{A^2 + C^2 + 1}},$$

their scalar product is

$$\mathbf{t}_0^* \cdot \mathbf{t}_1^* = \frac{-A^2 + C^2 + 1}{A^2 + C^2 + 1}.$$

Hence,

$$\alpha_c = \cos^{-1} \left| \frac{-A^2 + C^2 + 1}{A^2 + C^2 + 1} \right|.$$

For a specified idealized plait geometry, the crossing angle can therefore be used as a consistency check for the local closest-point representation.

4.5 External Axial Loading and Local Contact Reactions

4.5.1 Symmetric Boundary Load Sharing

The externally applied tensile load is

$$\mathbf{F}_{\text{total}} = F_{\text{total}} \mathbf{e}_z.$$

Under the ideal three-fold symmetry assumption,

$$\mathbf{F}_i = \frac{F_{\text{total}}}{3} \mathbf{e}_z.$$

4.5.2 Projection of the External Load onto the Contact Frame

The normal projection of the externally assigned load is

$$\mathbf{F}_{i,N}^{\text{ext}} = (\mathbf{F}_i \cdot \mathbf{n}_c) \mathbf{n}_c.$$

The tangential-plane projection is

$$\mathbf{F}_{i,T}^{\text{ext}} = \mathbf{P}_T \mathbf{F}_i.$$

Since

$$\mathbf{F}_i = \frac{F_{\text{total}}}{3} \mathbf{e}_z,$$

these may also be written as

$$\mathbf{F}_{i,N}^{\text{ext}} = \frac{F_{\text{total}}}{3} (\mathbf{e}_z \cdot \mathbf{n}_c) \mathbf{n}_c,$$

and

$$\mathbf{F}_{i,T}^{\text{ext}} = \frac{F_{\text{total}}}{3} (\mathbf{I} - \mathbf{n}_c \otimes \mathbf{n}_c) \mathbf{e}_z.$$

These projections describe how the global axial load is geometrically resolved relative to the local contact frame.

However, they must not be identified directly with the actual contact reactions. In general,

$$\mathbf{F}_{i,N}^{\text{ext}} \neq N_c \mathbf{n}_c$$

and

$$\mathbf{F}_{i,T}^{\text{ext}} \neq \mathbf{Q}_c.$$

The actual contact reactions depend on the coupled deformation, equilibrium, stiffness, and contact constraints of the wire system.

4.5.3 Normal and Tangential Contact Reactions

Let $\mathbf{n}_c$ point from wire 0 toward wire 1.

The force exerted by wire 1 on wire 0 at the representative contact is decomposed as

$$\mathbf{R}_{c,0} = -N_c \mathbf{n}_c + \mathbf{Q}_c,$$

where

$$N_c \geq 0$$

is the magnitude of the compressive normal contact reaction, and

$$\mathbf{Q}_c \cdot \mathbf{n}_c = 0$$

is the tangential contact reaction associated with friction.

By Newton's third law,

$$\mathbf{R}_{c,1} = -\mathbf{R}_{c,0}.$$

The reactions N_c and Q_c arise from the coupled deformation–contact problem and therefore depend on the applied load. Their quantitative values are evaluated using finite-element analysis in Section 5.

4.6 Frictional Self-Locking Criterion

4.6.1 Trial Sticking State

To determine whether the representative wire–wire contact is frictionally locked, a trial sticking state is first considered.

Normal contact requires

$$g_n = 0.$$

Under sticking, the relative displacement of the two contacting material points within the local tangent plane must vanish. Denoting the relative displacement of the two contact points by

$$\Delta \mathbf{u} = \Delta \mathbf{u}_0 - \Delta \mathbf{u}_1,$$

its tangential component is

$$\Delta \mathbf{u}_T = \mathbf{P}_T(\Delta \mathbf{u}_0 - \Delta \mathbf{u}_1).$$

The sticking constraint is therefore

$$\Delta \mathbf{u}_T = \mathbf{0}.$$

The tangential reaction required to enforce this constraint is denoted

$$\mathbf{Q}_c^{\text{req}}.$$

$\mathbf{Q}_c^{\text{req}}$ is the tangential reaction required to enforce sticking at the representative contact. In the finite-element implementation, this trial sticking state is imposed using a no-slip tangential contact condition, from which the corresponding normal and tangential contact reactions are extracted.

4.6.2 Coulomb Friction Criterion

Under Coulomb friction, the magnitude of the static tangential contact reaction cannot exceed the maximum friction capacity,

$$\|\mathbf{Q}_c^{\text{req}}\| \leq \mu_s N_c,$$

where μ_s is the coefficient of static friction between the two steel wires.

A friction function may therefore be defined as

$$\Phi_c = \|\mathbf{Q}_c^{\text{req}}\| - \mu_s N_c.$$

Three contact states follow.

For

$$\Phi_c < 0,$$

the required tangential force lies inside the Coulomb friction limit, and the contact remains sticking. At

$$\Phi_c = 0,$$

the contact reaches the limiting static-friction state and is at incipient slip. If a trial sticking solution gives

$$\Phi_c > 0,$$

the required tangential reaction exceeds the available static-friction capacity. The sticking assumption is therefore inadmissible and sliding must occur.

During ideal Coulomb sliding,

$$\|\mathbf{Q}_c\| = \mu_k N_c,$$

where μ_k is the kinetic-friction coefficient, and the tangential contact force acts opposite to the direction of relative tangential sliding.

4.6.3 Critical Friction Coefficient for Self-Locking

For a contacting state with

$$N_c > 0,$$

the minimum static-friction coefficient required to sustain the trial sticking state is defined as

$$\mu_{\text{crit}} = \frac{\|\mathbf{Q}_c^{\text{req}}\|}{N_c} \quad (14)$$

where N_c is the compressive normal contact reaction and $\mathbf{Q}_c^{\text{req}}$ is the tangential contact reaction required to enforce sticking.

The Coulomb sticking condition

$$\|\mathbf{Q}_c^{\text{req}}\| \leq \mu_s N_c$$

may therefore be written equivalently as

$$\mu_s \geq \mu_{\text{crit}}.$$

Accordingly,

$$\mu_s > \mu_{\text{crit}} \Rightarrow \text{sticking is admissible}, \quad (15)$$

$$\mu_s = \mu_{\text{crit}} \Rightarrow \text{incipient slip}, \quad (16)$$

and

$$\mu_s < \mu_{\text{crit}} \Rightarrow \text{the trial sticking state is inadmissible and sliding must occur}. \quad (17)$$

An equivalent dimensionless friction-utilization parameter may be written as

$$\eta_c = \frac{\mu_{\text{crit}}}{\mu_s} = \frac{\|\mathbf{Q}_c^{\text{req}}\|}{\mu_s N_c}, \quad (18)$$

such that $\eta_c < 1$, $\eta_c = 1$, and $\eta_c > 1$ correspond respectively to sticking, incipient slip, and sliding. In the present work, however, μ_{crit} is used as the primary self-locking metric because it can be obtained directly from a trial-sticking finite-element solution without prescribing a particular value of μ_s .

Both N_c and $\mathbf{Q}_c^{\text{req}}$ evolve as the plait deforms under axial loading. Therefore, the critical friction coefficient is generally deformation-dependent:

$$\mu_{\text{crit}}(F_{\text{total}}) = \frac{\|\mathbf{Q}_c^{\text{req}}(F_{\text{total}})\|}{N_c(F_{\text{total}})} \quad (19)$$

or, equivalently, it may be evaluated as a function of the global tensile strain.

Under the ideal periodic and symmetry assumptions adopted above, mechanically equivalent crossings may be represented by a single contact during the initial contact regime. In a finite-length deformable structure, however, additional non-equivalent contact regions may develop as loading proceeds. When multiple distinct contact patches exist, the local self-locking requirement should be evaluated for each contact region. A conservative global threshold may then be defined as

$$\mu_{\text{crit}}^{\text{global}} = \max_c \left(\frac{\|\mathbf{Q}_c^{\text{req}}\|}{N_c} \right) \quad (20)$$

where the maximum is taken over all active contact regions c .

The quantities N_c and $\mathbf{Q}_c^{\text{req}}$ therefore form the direct interface between the analytical framework and the finite-element model developed in Section 5.

5 Finite Element Analysis of Frictional Self-Locking

5.1 Numerical Model and Geometry

A three-dimensional finite-element model was developed in Abaqus/CAE 2022 and solved using Abaqus/Standard. The objective of the numerical model was not to reproduce the complete asymmetric 2-3-2 experimental specimen, but to investigate the contact mechanics of the idealized three-strand plait introduced in Section 4.

The model consists of three geometrically identical solid strands arranged in a periodic interlaced configuration. Each strand was imported into Abaqus as Parasolid geometry and modeled as a deformable three-dimensional solid. The global longitudinal direction of the finite-element model is the Y -axis. The undeformed axial length of the modeled plait is approximately

$$L_0 \approx 200 \text{ mm}, \quad (21)$$

and the diameter of each solid strand is approximately

$$d \approx 9 \text{ mm}. \quad (22)$$

The finite-element geometry is therefore considerably larger in absolute scale than the fine-wire specimens used in the preliminary dynamic experiments. The numerical model is intended as an idealized mechanics model rather than a one-to-one dimensional reproduction of those specimens. The quantities of primary interest are consequently the evolution of inter-strand contact reactions and the resulting frictional self-locking criterion.

The analytical model in Section 4 represents each strand through an idealized periodic centerline. The present finite-element geometry is not generated directly from that sinusoidal representation. Instead, the imported three-dimensional geometry provides a finite-thickness deformable realization of the same three-strand interlacing topology, allowing surface contact and local deformation to be resolved explicitly.

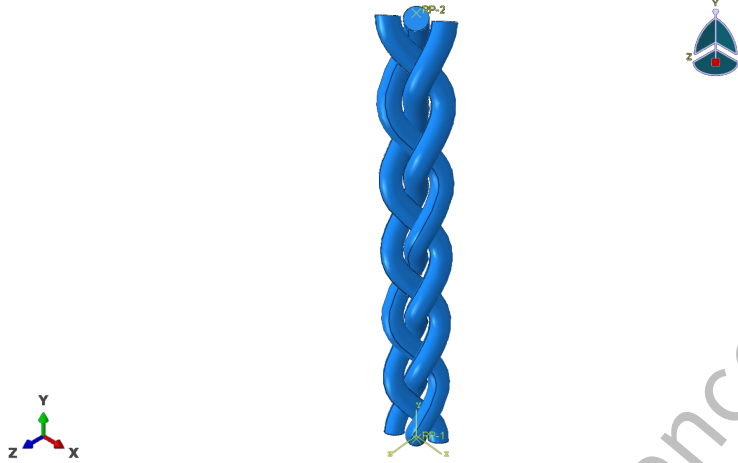


Figure 15: Undeformed three-dimensional finite-element geometry of the idealized three-strand plait. The global Y-direction is the axial loading direction.

Figure 15 shows the undeformed finite-element geometry and the global coordinate system.

5.2 Material Model

All three strands were assigned the same homogeneous isotropic steel material. An elastic–plastic constitutive model was adopted so that the numerical model could represent both the initially elastic contact response and the onset of localized yielding observed during subsequent loading.

The linear elastic response was defined by Young’s modulus

$$E = 188000 \text{ MPa} \quad (23)$$

and Poisson’s ratio

$$\nu = 0.30. \quad (24)$$

The plastic response was specified in Abaqus using true stress and plastic strain data. The input points were

$$(\sigma, \varepsilon_p) = (1540 \text{ MPa}, 0) \quad (25)$$

and

$$(\sigma, \varepsilon_p) = (1800 \text{ MPa}, 0.01057). \quad (26)$$

The first point therefore corresponds to an initial yield stress of approximately 1540 MPa, while the second point introduces subsequent strain hardening.

The material parameters used in the finite-element model are summarized in Table 3.

Table 3: Material parameters used in the finite-element model.

Parameter	Value
Young’s modulus, E	188000 MPa
Poisson’s ratio, ν	0.30
Initial yield stress	1540 MPa
Second plastic stress point	1800 MPa
Plastic strain at second point	0.01057

5.3 Boundary Conditions and Axial Loading

Reference points were introduced at the two ends of the plait to apply the boundary conditions in a controlled and approximately symmetric manner. The end faces of the three strands were connected to the corresponding reference points using kinematic coupling constraints.

At the fixed end, all translational and rotational degrees of freedom of the reference point were constrained. At the loading end, an axial displacement was imposed along the global Y -direction. The prescribed translational boundary conditions were

$$U_1 = 0, \quad U_2 = 10 \text{ mm}, \quad U_3 = 0. \quad (27)$$

Rotation about the transverse axes was constrained,

$$UR_1 = UR_3 = 0, \quad (28)$$

whereas rotation about the plait axis was left free,

$$UR_2 = \text{free}. \quad (29)$$

Allowing UR_2 prevents the boundary condition from introducing an artificial torsional constraint about the braid axis. This is important because axial deformation of an interlaced structure may naturally be accompanied by rotation, and fully fixing the axial rotational degree of freedom would introduce an additional reaction moment that is not part of the intended loading condition.

Because the initial modeled length is approximately $L_0 = 200$ mm and the final prescribed displacement is 10 mm, the maximum nominal global tensile strain is

$$\varepsilon_{\text{global,max}} = \frac{10}{200} = 0.05 = 5\%. \quad (30)$$

The displacement was ramped linearly over the normalized analysis step. Therefore, for a step time t between 0 and 1,

$$U_2(t) = 10t, \quad (31)$$

and the corresponding global tensile strain is

$$\varepsilon_{\text{global}}(t) = \frac{U_2(t)}{L_0} = 0.05t \quad (32)$$

This relation is subsequently used to express the evolution of the contact response as a function of global tensile strain rather than normalized step time.

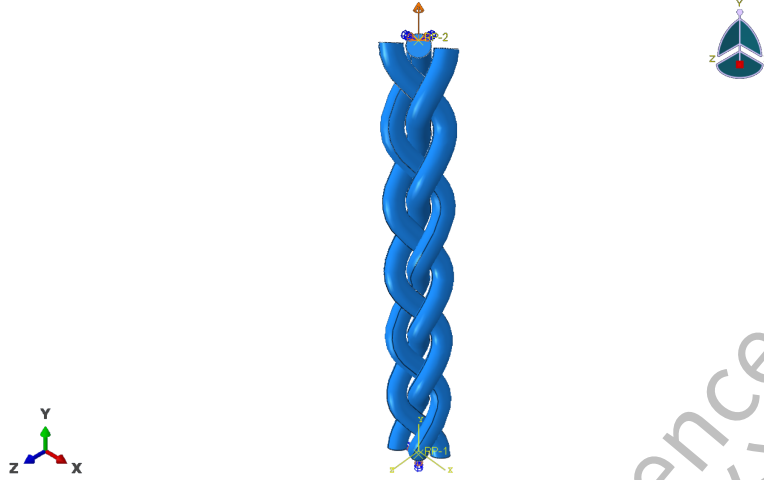


Figure 16: Boundary conditions applied to the three-strand finite-element model. The lower reference point is fully constrained, while an axial displacement is imposed at the upper reference point. Rotation about the global Y-axis is left free at the loading end.

The applied boundary conditions are illustrated in Fig. 16.

5.4 Contact Formulation and Trial-Sticking Implementation

Inter-strand interaction was modeled using the General Contact formulation in Abaqus/Standard. Contact was allowed between the surfaces of all three deformable strands.

In the normal direction, a *Hard Contact* formulation was employed. This prevents interpenetration while allowing separation when the contact pressure becomes zero.

The tangential behavior requires particular attention because the purpose of the simulation is not initially to prescribe a known physical friction coefficient. Instead, the objective is to determine the tangential reaction that would be required to maintain sticking. Accordingly, the tangential contact behavior was specified as *Rough*.

Rough contact suppresses relative tangential slip between contacting surfaces. In the present analysis, it is therefore used as a numerical implementation of the trial-sticking state introduced in Section 4.6. It should not be interpreted as an assertion that the physical steel-steel interface possesses an infinite friction coefficient. Rather, it allows the finite-element solution to determine the tangential contact force required to enforce zero relative tangential motion.

For each active contact region, Abaqus provides the contact normal-force vector and contact shear-force vector at the surface nodes. These are denoted by

$$\mathbf{CNORMF}$$

and

$$\mathbf{CSHEARF},$$

respectively.

For a contact region containing surface nodes j , the resultant normal contact-force vector is obtained from

$$\mathbf{N}_c = \sum_j \mathbf{CNORMF}_j, \quad (33)$$

and the tangential reaction required by the trial-sticking condition is

$$\mathbf{Q}_c^{\text{req}} = \sum_j \text{CSHEARF}_j. \quad (34)$$

Their resultant magnitudes are therefore

$$N_c = \|\mathbf{N}_c\|, \quad (35)$$

and

$$Q_c^{\text{req}} = \|\mathbf{Q}_c^{\text{req}}\|. \quad (36)$$

The critical static-friction coefficient corresponding to this trial-sticking state is then calculated using the criterion derived in Section 4,

$$\mu_{\text{crit}} = \frac{Q_c^{\text{req}}}{N_c}. \quad (37)$$

An important numerical detail concerns the form of the Abaqus contact output. Contact-force quantities were extracted at the *Element Nodal* position. Because a physical surface node may be associated with several adjacent element facets, the same node can appear multiple times in the exported field-output report. These repeated entries were identified by node label and removed before the nodal force vectors were summed. This avoids artificial multiplication of the resultant contact force.

The field outputs used for the contact analysis included **CNORMF**, **CSHEARF**, **CPRESS**, **CSTATUS**, and related contact quantities. The first two variables provide the primary numerical quantities required to evaluate the frictional self-locking criterion.

5.5 Analysis Step and Mesh

The axial tensile response was simulated using a *Static, General* step in Abaqus/Standard. Geometric nonlinearity was enabled through **NLGEOM=ON** in order to account for the large rotations and changes in contact geometry associated with deformation of the plait.

Automatic time incrementation was employed with an initial increment of

$$\Delta t_{\text{initial}} = 0.01,$$

a minimum increment of

$$\Delta t_{\text{min}} = 10^{-7},$$

and a maximum increment of

$$\Delta t_{\text{max}} = 0.02.$$

The normalized step time ranged from $t = 0$ to $t = 1$, corresponding to the prescribed displacement increasing linearly from zero to 10 mm.

The three strands were discretized using eight-node linear brick elements with reduced integration (**C3D8R**). A global seed size of approximately

$$h = 1 \text{ mm}$$

was used. Relative to the strand diameter of approximately 9 mm, this gives approximately nine characteristic element lengths across the diameter.

The same mesh resolution was used throughout the present analysis. A formal mesh-convergence study was not conducted; therefore, the numerical results are interpreted primarily in terms of the evolution and relative trends of the contact response rather than as mesh-independent absolute predictions.

5.6 Contact Establishment During Axial Loading

The trial-sticking simulation first reveals how inter-strand contact develops as the plait is progressively tensioned. Because the prescribed displacement increases linearly with normalized step time, the corresponding global tensile strain is obtained from

$$\varepsilon_{\text{global}} = 0.05t.$$

At a step time of

$$t = 0.1600,$$

corresponding to

$$\varepsilon_{\text{global}} = 0.800\%,$$

the extracted contact-force outputs remained zero, indicating that the initially separated strand surfaces had not yet developed a finite contact reaction.

At the next recorded contact state,

$$t = 0.1788,$$

or

$$\varepsilon_{\text{global}} = 0.894\%,$$

non-zero normal and tangential contact reactions were detected. The corresponding resultant forces were

$$N_c = 34.93 \text{ N}$$

and

$$Q_c^{\text{req}} = 149.38 \text{ N}.$$

This state is therefore identified as the first resolved contact state in the saved finite-element output.

The initial finite-element geometry consequently contains a small clearance between the contacting strand surfaces rather than an exact zero-gap condition. The contact response therefore develops only after a small amount of global extension. This differs slightly from the ideal analytical model in Section 4, where geometrical contact is represented by the condition $g_n = 0$ from the outset.

The contact region subsequently strengthens rapidly as axial extension increases. This evolution provides the numerical basis for evaluating the frictional requirement for self-locking.

5.7 Evolution of Normal and Tangential Contact Reactions

Figure 17 shows the evolution of the resultant normal contact force and the tangential reaction required to maintain the trial-sticking condition.

Immediately after contact is established at approximately 0.894% global tensile strain, the normal contact reaction is relatively small,

$$N_c = 34.93 \text{ N},$$

while the required tangential reaction is

$$Q_c^{\text{req}} = 149.38 \text{ N}.$$

As the plait is further tensioned, both quantities increase substantially. At approximately 1.078% global strain,

$$N_c = 352.33 \text{ N}, \quad Q_c^{\text{req}} = 678.38 \text{ N},$$

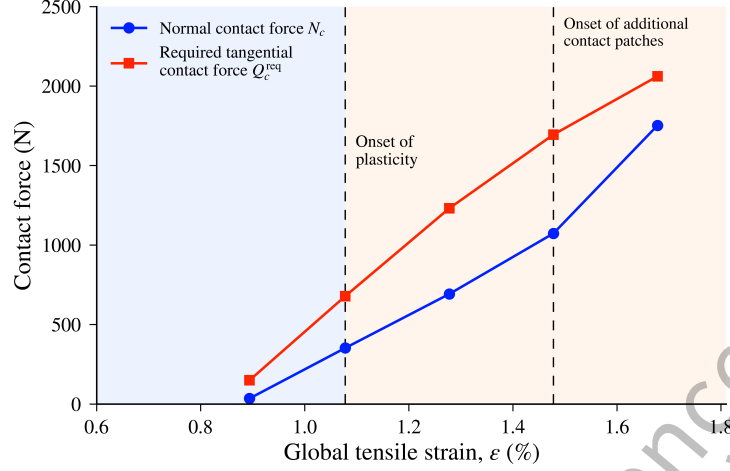


Figure 17: Evolution of the normal contact reaction N_c and the tangential contact reaction Q_c^{req} required to maintain trial sticking as a function of global tensile strain.

and at 1.278% global strain they increase to approximately

$$N_c = 692.07 \text{ N}, \quad Q_c^{\text{req}} = 1230.91 \text{ N}.$$

The contact forces continue to rise during subsequent deformation. By a global tensile strain of approximately 1.678%, the aggregated contact-force resultants reach approximately

$$N_c = 1751.68 \text{ N}$$

and

$$Q_c^{\text{req}} = 2060.90 \text{ N}.$$

Although both the normal and tangential reactions increase with loading, the normal reaction grows proportionally more rapidly after contact is established. This difference is important because the critical friction coefficient is governed by the ratio

$$\mu_{\text{crit}} = \frac{Q_c^{\text{req}}}{N_c}.$$

The increase in normal confinement therefore progressively reduces the friction coefficient required to sustain the trial-sticking condition.

At larger strains, additional contact regions develop along the strand. The force values plotted beyond the initial single-contact regime therefore represent aggregated contact reactions over the active regions of the displayed strand and are interpreted primarily as indicators of the overall evolution of inter-strand constraint.

5.8 Evolution of the Critical Friction Requirement

The critical friction coefficient was evaluated from the trial-sticking contact reactions using

$$\mu_{\text{crit}} = \frac{Q_c^{\text{req}}}{N_c}. \quad (38)$$

The resulting evolution is shown in Fig. 18.

At the first resolved contact state,

$$\varepsilon_{\text{global}} = 0.894\%,$$

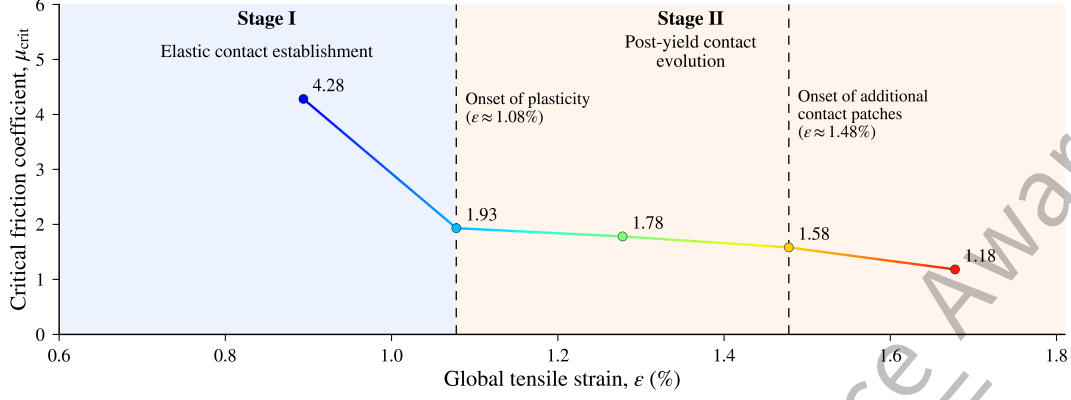


Figure 18: Evolution of the frictional requirement obtained from the trial-sticking contact reactions. Before the development of multiple contact regions, the ratio corresponds directly to the local critical friction coefficient μ_{crit} . At larger strains, it represents an aggregate contact-force ratio.

the normal contact reaction is still small, resulting in a comparatively large critical friction requirement,

$$\mu_{crit} = 4.28.$$

As contact becomes more firmly established, the ratio decreases rapidly. At

$$\epsilon_{global} = 1.078\%,$$

the critical value falls to

$$\mu_{crit} = 1.93,$$

and at

$$\epsilon_{global} = 1.278\%,$$

it further decreases to approximately

$$\mu_{crit} = 1.78.$$

The decrease arises because the compressive normal contact reaction grows more rapidly than the tangential reaction required to suppress sliding. In terms of the Coulomb criterion,

$$Q_c^{req} \leq \mu_s N_c,$$

an increase in N_c increases the available frictional capacity associated with any prescribed value of μ_s . Consequently, a progressively smaller value of the physical static-friction coefficient is required to maintain sticking.

At larger strains, the aggregated force ratio continues to decrease, reaching approximately

$$1.58$$

at 1.478% global strain and

$$1.18$$

at 1.678% global strain. These later states coincide with the development of additional contact regions and therefore no longer correspond strictly to a single representative contact patch. They are consequently interpreted as an aggregate measure of the strengthening inter-strand constraint rather than as a unique local Coulomb threshold.

Within the simulated loading range, the numerical results therefore indicate a clear overall trend: progressive tensioning of the plait strengthens normal inter-strand confinement and reduces the frictional requirement associated with maintaining a sticking configuration.

5.9 Contact-Region Expansion and Onset of Plasticity

The evolution of the active contact region provides additional insight into the reduction in the frictional requirement observed above. Figure 19 shows the number of surface nodes carrying non-zero contact-force output at different stages of loading.

No active contact nodes were detected at a global tensile strain of 0.800%. At the first resolved contact state, $\varepsilon_{\text{global}} = 0.894\%$, nine active contact nodes were identified. The number subsequently increased to 12 at both 1.078% and 1.278% global strain.

A pronounced change occurs between 1.278% and 1.478% global strain, where the number of active contact nodes increases from 12 to 40. At 1.678% global strain, 45 active contact nodes are present. Inspection of the contact distribution indicates that this increase is associated with the formation and expansion of additional inter-strand contact regions rather than only the enlargement of the original contact patch.

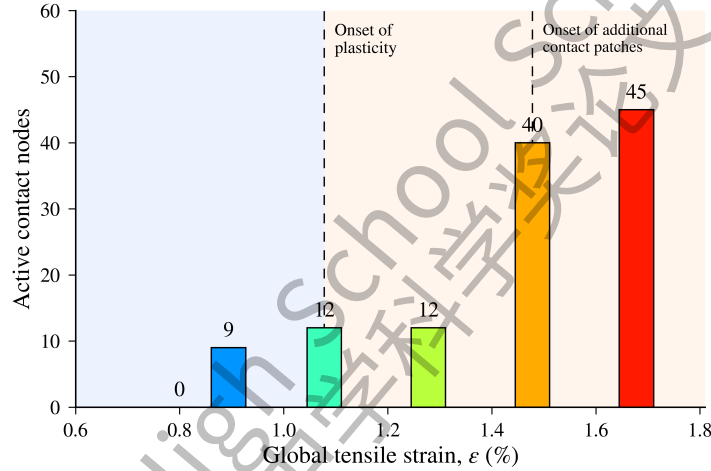


Figure 19: Evolution of the number of active contact nodes with global tensile strain. The rapid increase beyond approximately 1.48% strain indicates the development of additional inter-strand contact regions.

The onset of material plasticity occurs earlier than the major expansion of the contact network. At

$$\varepsilon_{\text{global}} = 0.894\%,$$

the maximum equivalent plastic strain remained zero. At the next saved state,

$$\varepsilon_{\text{global}} = 1.078\%,$$

a small but non-zero equivalent plastic strain was observed,

$$PEEQ_{\text{max}} \approx 7.63 \times 10^{-4},$$

indicating the onset of localized yielding.

Plastic deformation subsequently develops primarily near highly constrained regions of the interlaced strands. At a global tensile strain of

$$\varepsilon_{\text{global}} = 1.678\%,$$

the maximum equivalent plastic strain reaches approximately

$$PEEQ_{\max} = 8.96 \times 10^{-3}.$$

A representative equivalent-plastic-strain contour is shown in Fig. 20.

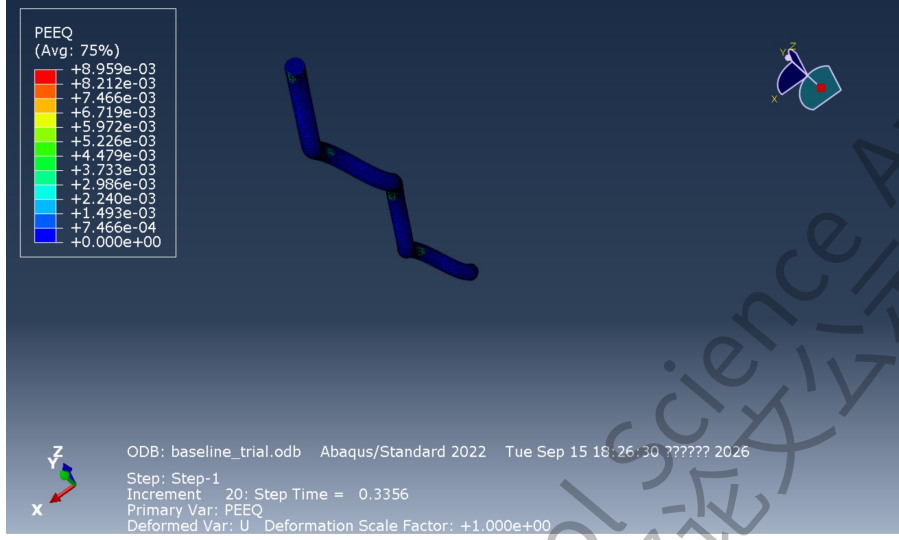


Figure 20: Equivalent plastic strain distribution at a global tensile strain of approximately 1.678%. Localized yielding develops near geometrically constrained regions of the interlaced strands.

The numerical response may therefore be divided qualitatively into two regimes. The first is an initial contact-establishment regime, in which contact develops while the material remains predominantly elastic. The second is a post-yield contact-evolution regime, in which localized plastic deformation and the formation of additional contact regions occur simultaneously.

The increase in active contact nodes should not be interpreted as a mesh-independent physical measure of contact area, since the number of active nodes depends on the finite-element discretization. Instead, it is used here as an indicator of the qualitative expansion and multiplication of the contact regions. The corresponding force and frictional trends remain the primary quantities used to characterize the self-locking response.

5.10 Analysis and Limitations

The finite-element results support the central contact-mechanics interpretation developed in Section 4. Frictional self-locking in the three-strand plait is governed not simply by the existence or number of strand crossings, but by the competition between the tangential reaction required to suppress relative sliding and the normal contact reaction that generates Coulomb frictional capacity.

The numerical results show that, once contact is established, the normal inter-strand reaction increases rapidly as the plait is tensioned. Although the tangential reaction required to maintain sticking also increases, its increase is proportionally smaller than that of the normal reaction over the initial contact regime. Consequently,

$$\mu_{\text{crit}} = \frac{Q_c^{\text{req}}}{N_c}$$

decreases as the structure is progressively tensioned.

The physical mechanism may therefore be summarized as

$$\text{axial tension} \rightarrow \text{increased geometric confinement} \rightarrow N_c \uparrow \rightarrow \frac{Q_c^{\text{req}}}{N_c} \downarrow \rightarrow \text{reduced frictional requirement for sticking.}$$

This result provides a quantitative interpretation of frictional self-locking in an interlaced metallic structure. The braid does not become more resistant to sliding merely because it contains repeated crossings.

Rather, axial deformation modifies the contact state at those crossings and generates increasing normal constraint, thereby increasing the frictional capacity available for a given physical coefficient of friction.

The later stages of loading also show that the contact topology itself evolves. Additional contact regions appear as the strands deform, while localized plasticity develops near strongly constrained regions. The finite-element response is therefore more complex than the single representative-contact model used in the analytical derivation. The analytical model should be regarded as a local description of the underlying self-locking criterion, whereas the finite-element model captures the evolution of a finite-length, deformable, multi-contact structure.

Several limitations of the present numerical analysis should be noted. First, the finite-element geometry was imported as non-parametric Parasolid geometry rather than generated directly from the analytical centerline parameters a , b , and P . A systematic parametric study of braid geometry was therefore not performed in the present work.

Second, the absolute dimensions of the finite-element model differ from those of the fine-wire specimens used in the preliminary dynamic tensile experiments. The numerical model is consequently intended to investigate the mechanics of the interlaced topology rather than to reproduce the experimental specimen quantitatively.

Third, the tangential contact condition used in the principal analysis was *Rough* contact. This condition was introduced deliberately to construct a trial-sticking state and determine the tangential reaction required to prevent sliding. It should therefore not be interpreted as a physical friction law for the steel interface. The calculated value of μ_{crit} represents the static-friction coefficient that would be required for a Coulomb interface to sustain the corresponding sticking state.

Fourth, the initial imported geometry contains a small inter-strand clearance. Consequently, finite contact reactions appear only after approximately 0.8–0.9% global tensile strain, whereas the ideal analytical model treats the representative wires as being initially in geometrical contact.

Fifth, a formal mesh-convergence study was not performed. The absolute magnitudes of local contact quantities and the number of active contact nodes may therefore retain some mesh dependence. For this reason, the principal conclusions of the present analysis are based primarily on the observed evolution and relative trends of the contact reactions.

Finally, once multiple non-equivalent contact regions develop, a single ratio obtained from aggregated contact forces cannot rigorously represent the critical Coulomb coefficient of every local contact. A fully conservative assessment would require evaluation of

$$\mu_{\text{crit},c} = \frac{\|\mathbf{Q}_c^{\text{req}}\|}{N_c}$$

for each individual contact region and identification of the largest value,

$$\mu_{\text{crit}}^{\text{global}} = \max_c (\mu_{\text{crit},c}).$$

Such local multi-contact analysis, together with parametric variation of braid geometry and explicit Coulomb-friction simulations, constitutes a natural direction for future work.

Despite these limitations, the present model establishes a consistent numerical connection between braid deformation, inter-strand contact reactions, and the frictional self-locking criterion. Within the simulated loading range, the results demonstrate that progressive tensioning can substantially strengthen normal inter-strand constraint and reduce the friction coefficient required to maintain a sticking configuration.

6 Conclusion

This study investigated whether interlaced steel-wire architectures can develop frictional self-locking under axial tension. Three alternative seven-wire configurations inspired by three- and four-strand braids were first fabricated and compared with a conventional 1 + 6 strand using preliminary Split Hopkinson Tensile Bar experiments. Among the proposed configurations, the three-strand braid exhibited the most favorable combination of peak tensile stress and deformation capacity and was therefore selected for further mechanistic investigation.

To isolate the contact mechanics of the interlaced topology, the experimental 2–3–2 configuration was idealized as a symmetric three-strand plait. A geometric model was developed to describe a representative wire–wire crossing, including the closest-point geometry, contact normal, tangent plane, and local contact reactions. Based on a trial-sticking state and the Coulomb friction criterion, a critical friction coefficient was defined as

$$\mu_{\text{crit}} = \frac{\|\mathbf{Q}_c^{\text{req}}\|}{N_c},$$

which represents the minimum static-friction coefficient required for a given contact state to remain sticking.

The finite-element model provided a numerical implementation of this criterion. After contact was established, both the normal contact reaction and the tangential reaction required for sticking increased with axial deformation. During the initial single-contact regime, however, the normal reaction increased proportionally more rapidly, causing the calculated critical friction coefficient to decrease from approximately 4.28 at 0.894% global tensile strain to 1.93 at 1.078% and 1.78 at 1.278%. At larger strains, additional contact regions and localized plasticity developed, indicating a transition from initial elastic contact establishment to a more complex multi-contact post-yield response.

These results show that the mechanical locking capability of a braided strand cannot be inferred simply from the number of crossings. Instead, frictional self-locking is controlled by the evolving balance between normal geometric confinement and the tangential reaction required to prevent relative sliding. Within the present model, progressive tensioning strengthens normal inter-strand constraint and reduces the frictional requirement associated with maintaining a sticking configuration.

The present work therefore provides a contact-mechanics framework for interpreting self-locking in interlaced metallic strands. Future work should extend the model through parameterized braid geometry, mesh-convergence analysis, local evaluation of multiple contact patches, and explicit Coulomb-friction simulations.

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Acknowledgment

Metallic wires and the ropes made from them play a crucial role in engineering because of their strong mechanical properties. They are widely used in fields such as civil infrastructure, mining operations, and energy transmission. Therefore, the mechanical performance of metallic wires has attracted extensive research interest over the past decades. This work began with my observation of potential limitations in the conventional 1 + 6 seven-wire strand configuration and was inspired by traditional knotting techniques. I proposed three alternative ways of braiding individual wires into strands. After designing the gripping fixture for the experimental setup and fabricating the samples manually, I tested their mechanical properties using a Split Hopkinson Tensile Bar system. The experimental results showed that several of the proposed braided configurations could achieve competitive tensile strength while maintaining considerable deformation capacity.

I would like to express my special gratitude to Ms. Yan Chen. Her guidance in prerequisite knowledge, topic selection, configuration design, experimental work, and research writing supported me significantly throughout this study. All of her guidance was provided for educational and research purposes, and no remuneration was charged. I also received help from Mr. Jinxi Chen in learning the fundamental knowledge of this field and in modeling the gripping fixture. Mr. Bin Xu also assisted me in conducting the experiments.

This research was mainly carried out by myself, with Ms. Yan Chen providing general guidance and specific experimental support. At the beginning of the project, she introduced me to the basic mechanics and current research related to metallic wires and wire strands. Starting from my own interest, I wanted to further explore how individual wires could be assembled into strands and investigate the relationship between braiding configuration and the mechanical properties of the resulting structures. After discussing this direction with my teacher, we developed the idea of combining traditional knotting concepts with metallic-wire braiding. I then conducted independent research on possible braiding methods and selected three alternative configurations for experimental investigation. During this process, Ms. Yan Chen provided an important suggestion: if the proposed configurations were to be compared fairly with the conventional 1 + 6 strand, the total number of constituent wires should remain the same. This consideration led me to design all candidate configurations using seven wires.

The manual fabrication of the specimens was one of the most difficult stages of the project. Initially, I attempted to braid the fine steel wires directly by hand, but their small diameter and high stiffness made them extremely difficult to manipulate, and I encountered repeated failures. To address this problem, I began using anti-slip gloves and a braiding frame to improve control of the wires. Ms. Yan Chen also suggested that I first practice the braiding sequences using strings before transferring the same procedures to steel wires. After repeated practice and adjustment, I was eventually able to fabricate the braided strands used as experimental specimens.

Another difficulty arose when adapting the specimens to the Split Hopkinson Tensile Bar system. Because the braided strands had different effective diameters from the specimens used in the original setup, the gripping fixture had to be redesigned. One challenge was how to increase the groove size while maintaining a sufficiently stable and symmetric clamping condition. After discussion with the teachers, we decided to modify only the opening dimensions required to accommodate the specimens while preserving the overall clamping structure. Through repeated adjustment, testing, and discussion, these experimental difficulties were gradually resolved.

Overall, I was responsible for developing the research question, investigating and selecting the proposed braiding configurations, fabricating the specimens, participating in the design and modification of the gripping fixture, processing and interpreting the experimental results, developing the theoretical analysis, carrying out the finite-element investigation, and writing the research paper. The teachers provided conceptual guidance, technical instruction, and experimental assistance at different stages of the project.