



Self-Forcing Braided Sleeve Gripper

A Soft Robotics Approach
to Offshore Conductor Removal

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Self-Forcing Braided Sleeve Gripper: A Soft Robotics Approach to Offshore Conductor Removal

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Preface

This thesis concludes my Master's studies in Mechanical Engineering at Delft University of Technology, within the Multi-Machine Engineering track. The work presented here was carried out in cooperation with and under the supervision of Allseas in Delft, between November 2025 and July 2026.

This project reached me through an earlier internship at Allseas, and I was immediately motivated to pursue it. It offered the possibility to take a pressing issue and technology gap in the decommissioning of structurally damaged conductors and to approach it from the creative perspective of soft robotics, arriving at a gripper inspired by a children's toy and scaled to multi tonne lifting loads. The final results are promising. To my knowledge, this is one of the first soft grippers to find an application in offshore decommissioning at this scale, and I hope it adds something to the soft robotics literature. The work is also intended as a solid first step for future research and development efforts. Through this project I improved my ability to apply creativity in engineering, and I expanded my capacity to take an idea and turn it into a working concept while managing the different aspects of a project.

Many people supported me throughout this project and I am grateful to all of them. First, I want to thank Jeroen Breukels from Allseas, who proposed this project and supervised it throughout. Our weekly meetings helped me stay focused and think creatively, and the feedback he gave was essential. Without it, the prototypes would never have taken shape. I am also grateful to Dr. Jovana Jovanova for supervising this work and for allowing me to look at things from a different perspective, for helping me make the right simplifications and for proposing alternatives whenever the work grew complex or the scope threatened to get out of hand. I want to thank Vera Kortman for the inspiration and for the input she provided on fabricating the prototype and on braiding. A special thanks goes to Angelo Stankov for his assistance during the test campaign.

On a more personal note, I want to thank all my friends in the Netherlands, in Greece and elsewhere, for always being there and for their support. To my parents, my sister and the rest of my family, I cannot begin to imagine how any of this, and much beyond this thesis, would have been achievable without you. Thank you for every sacrifice you have made over the years. Finally, to Dimitra, without your patience, your constant encouragement and your love through all the difficult times, none of this would have been possible.

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Delft, July 2026*

Abstract

Offshore decommissioning is accelerating as ageing oil and gas infrastructure reaches the end of its service life. In the North Sea alone, over 2,650 wells are scheduled between 2025 and 2034, and well plug and abandonment accounted for 49% of decommissioning expenditure on the UK Continental Shelf in 2024. Within this scope, conductor removal, the extraction of large-diameter vertical steel pipes connecting subsea wells to surface facilities, remains a persistent operational bottleneck. Current practice relies on the cut-and-pull method, in which the entire lifting load is transferred through a single conductor cross-section. Its applicability is therefore governed by the structural condition of that cross-section, as corroded casings and degraded connections may not sustain the concentrated load.

This thesis proposes a compliant gripping mechanism that distributes the lifting load over the outer conductor surface, and examines whether it scales to the multi-tonne loads of conductor removal, a regime far beyond the small-scale applications to which soft robotic grippers have been confined. Following a literature study into soft gripping methods and a conceptual design process, four concepts were evaluated, from which a braided constriction sleeve gripper was selected. Its defining property is a self-forcing mechanism, in which the weight of the conductor tightens the braid around its surface, so that a heavier load produces a stronger grip.

An analytical model relates the braid geometry to the holding capacity, which grows exponentially with a dimensionless self-forcing coefficient. This coefficient is the product of the friction coefficient and a geometric coefficient scaling with the conductor radius and the square of the number of braid turns, and inversely with the gripping length. The capacity is supplied passively by the weight of a bottom ring, requiring no continuous actuation. A parametric campaign across twelve configurations, each tested five times, supported the model within the range examined. The measured capacity scaled linearly with the ring weight at fixed geometry, rose far more steeply with the geometric coefficient than preload friction alone permits, following the exponential form characteristic of self-forcing, and followed the predicted direction for each geometric parameter varied in isolation. A single friction coefficient of approximately 0.135 reproduced the measured capacity across differing geometries, supporting the contact assumptions. The most tightly braided configuration sustained more than 75 kg without slipping, giving a payload-to-self-weight ratio of at least 337:1 against 16:1 at the baseline.

Applied to the North Cormorant platform, the model yields an early-phase full-scale design employing a steel wire rope braid of seven turns over a gripping length of 14 metres, preloaded by a 10 tonne bottom ring. Under a conservative design friction coefficient of 0.10, the holding capacity reaches approximately 18,800 kN against a design lifting weight of 634 kN per conductor section. Slip is therefore not the governing constraint, which is set instead by the rope strength at the top ring connection and by the deployment of the braid. To the author's knowledge, this constitutes the first compliant gripping concept modelled, experimentally supported, and dimensioned for offshore removals at the multi-tonne scale.

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List of Symbols

Latin Symbols

Symbol	Unit	Definition
A	m^2	total contact area between braid and cylinder
A_1	m^2	contact area of the braid over one full turn
A_{contact}	m^2	contact area (Ch. 3)
A_{rhombus}	m^2	overlap area at a single weaving crossing
A_{S1}	m^2	uncorrected rectangular contact area over one turn
C_A	-	covered area ratio
CSR	-	closed-state ratio
CV	-	coefficient of variation
c	m	radial deployment clearance
d	m	fibre thickness (rope or wire diameter)
d_c	m	critical fibre thickness of the fully compact braid
E	Pa	Young's modulus
F	N	axial force in the braid
$F(x)$	N	axial tension distribution along the sleeve
$F(0)$	N	axial tension at the top of the braid
F_f	N	friction at the contact interface
F_{fibre}	N	maximum tension in a single fibre
F_{lift}	N	total friction lifting force (Ch. 3)
F_p, F_N, F_f, F_H	N	pinch, normal, friction, and holding forces (Ch. 3)
F_r	N	total radial force over one braid turn
F_{sf}	N	self-forcing tension contribution
G	Pa	shear modulus
H	m	gripping length
$H_{\text{closed}}, H_{\text{open}}$	m	braid length in the closed and open states
H_{eff}	m	effective contact length under necking
ΔH	m	height change between open and closed states
h	m	braiding disc height increment per cycle
I	m^4	second moment of area of the fibre cross section
I_p	m^4	polar second moment of area of the fibre cross section
K_1, K_2, K_3	various	helical spring constants
k	m^{-2}	force conversion factor from axial force to radial pressure
L_b	m	jamming pad radius (Ch. 3)
M	Nm	driving torque on the top ring (Ch. 3)
MBL	N	minimum breaking load
$m_{\text{braid}}, m_{\text{ring}}$	kg	mass of the braid and of the bottom ring
N	-	number of full turns
N_{cycles}	-	total number of braiding cycles
N_j	-	number of jaw pairs (Ch. 3)
n	-	number of fibre sets, giving $2n$ fibres in total
n_s	-	number of straps (Ch. 3)
OSR	-	open-state ratio
P	Pa	radial pressure between braid and cylinder
P_0, P_{sf}	Pa	preload and self-forcing pressure contributions
$P_{\text{gen}}, P_{\text{req}}, P_b, P_h, P_{\text{lift}}, \Delta P_{\text{end}}$	Pa	concept screening pressures (Ch. 3)
p	m	pitch of the helix
$p_{\text{max}}, p_{\text{min}}$	m	pitch at the extreme braid states
R_{pw}	-	payload-to-self-weight ratio
R^2	-	coefficient of determination
r	m	braid radius
r_0	m	braid radius at the fabrication state
r'	m	braid radius at an arbitrary deformed state
r_c	m	conductor or cylinder radius

r_{closed}, r_{open}	m	braid radius in the closed and open states
r_{max}, r_{min}	m	braid radius at the extreme states
r_s	m	sleeve or ring radius (Ch. 3)
S	m	length of one fibre over N turns
S_1	m	fibre length over one full turn
S_{total}	m	total fibre length in the braid
SF_{lift}, SF_{break}	-	lift and breaking safety factors
s	N	sample standard deviation of the slip load
T	N	axial transition (restoring) force of the braid
T_c, T_e	N	compacting and elongating restoring forces
T_0, T_L, T_{strap}	N	strap tensions (Ch. 3)
t	m	strap thickness (Ch. 3)
W_c	N	conductor section weight
$W_{c,max}, W_{c,min}$	N	maximum and minimum holding capacity
W_{eff}	N	effective ring weight
$W_{gripper}$	N	gripper self-weight
W_{rest}	N	weight of remaining equipment
W_{ring}	N	bottom ring weight
$W_{slip}, \overline{W}_{slip}$	N	slip load and mean slip load
w	m	strap width (Ch. 3)

Greek Symbols

Symbol	Unit	Definition
α	deg	braiding angle
α_0, α'	deg	braiding angle at the fabrication and arbitrary states
$\alpha_{max}, \alpha_{min}$	deg	braiding angle at the extreme states
β	-	self-forcing coefficient
Γ	-	geometric coefficient
γ	m^{-1}	axial decay coefficient
Δ	m^4	discriminant of the pitch equation
$\delta W, \delta U$	J	external virtual work and internal virtual energy
$\delta p, \delta r$	m	virtual pitch and radius changes
θ	deg	angular position along the helix
μ	-	friction coefficient
σ_j	Pa	applied jamming stress (Ch. 3)
ϕ_{ring}	deg	top ring rotation (Ch. 3)

List of Abbreviations

Abbreviation	Definition
API	American Petroleum Institute
CFT	Chinese Finger Trap
CSR	Closed-State Ratio
CV	Coefficient of Variation
DAF	Dynamic Amplification Factor
DNV	Det Norske Veritas
ISO	International Organization for Standardization
LAT	Lowest Astronomical Tide
MBL	Minimum Breaking Load
MRQ	Main Research Question
ODE	Ordinary Differential Equation
OEUK	Offshore Energies UK
OSR	Open-State Ratio
P&A	Plug and Abandonment
PLA	Polylactic Acid
PVC	Polyvinyl Chloride
ROV	Remotely Operated Vehicle
SRQ	Sub-Research Question
UK	United Kingdom
UKCS	UK Continental Shelf
VDI	Verein Deutscher Ingenieure

1 Introduction

As the global energy system transitions away from fossil fuels toward renewable sources, a growing challenge has emerged in the offshore sector: how the thousands of tonnes of ageing oil and gas production infrastructure can be safely and efficiently removed. Within decommissioning efforts, well plug and abandonment (P&A) accounted for 49% of total decommissioning expenditure in the UK Continental Shelf (UKCS) in 2024 [1]. The scale of work ahead is substantial. Between 2025 and 2034, over 2,650 wells are scheduled for decommissioning in the North Sea alone, representing both a significant technical challenge and a major business opportunity for the supply chain [1].

Within well P&A operations, conductor removal, the extraction of large-diameter vertical steel pipes that connect subsea wells to surface facilities, presents one of the most persistent operational bottlenecks. Conductors are essential structural elements during production, but they become major obstacles during decommissioning. Current removal methods suffer from extended durations, high costs and operational difficulties that are not fully resolved by existing technology.

This thesis investigates the application of compliant gripping mechanisms to the conductor removal problem, where conductors weighing several hundred tonnes must be lifted from offshore platforms in conditions that can render conventional methods ineffective. A gripper concept is therefore proposed, analytically developed, and validated through prototype testing. The following sections provide the relevant background and motivation, identify the research gap, define the research questions and methodology, and outline the scope and structure of this report.

1.1 Background and Motivation

Conductor removal is one operation within a broader decommissioning sequence that involves topsides removal, jacket dismantling, and well abandonment, each of which carries its own technical and regulatory constraints. To establish the research gap that this thesis addresses, the following sections first provide context on offshore decommissioning in general, then examine the current state of the art in conductor removal specifically. The limitations of existing methods are presented in detail, motivating the investigation into compliant gripping as an alternative through technology transfer from adjacent fields.

1.1.1 Offshore Oil and Gas Platform Decommissioning

As of March 2023, 590 oil and gas platforms exist in the North Sea, with the first one commissioned in 1966. Of these, 95% are fixed structures anchored to the seabed, while the remainder are floating structures. Since 1966, 138 platforms have been successfully decommissioned and removed, while 113 were decommissioned but not yet removed. Platform installation numbers increased through the early 2000s but have declined in the last two decades [2]. Although these numbers are specific to the North Sea, they provide an instructive example that can be extrapolated to other regions worldwide, where similar trends are emerging.

These numbers indicate two key realities. First, the importance of decommissioning these structures is growing. Second, the process presents significant challenges, as these structures are ageing, with over 40 years of constant fatigue loading from harsh sea conditions, designs that did not primarily account for eventual decommissioning, and structural integrity that may now be uncertain.

This thesis focuses on the fixed jacket platform, representing approximately 95% of installed offshore structures. The topside is supported by the jacket, a truss structure of welded tubular members piled to the seafloor, typically installed in water depths not exceeding 500 metres [3]. As presented in Figure 1.1, the three major components are the topside, which supports drilling, production, and life support equipment, the jacket, which resists

environmental loads and protects well conductors and pipeline risers, and the foundation, consisting of steel piles driven multiple metres below the seabed [4].

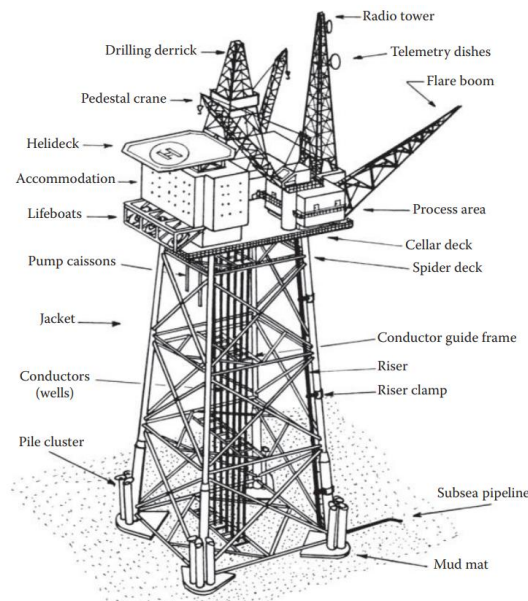


Figure 1.1: Fixed offshore jacket platform [4]

Finally, as seen in Figure 1.1, the conductors extend vertically from the mudline to the topside through conductor guides positioned at multiple elevations, which constrain lateral movement while maintaining an annular clearance of approximately 25 mm to accommodate thermal expansion and construction tolerances [5]. The conductors themselves are examined in detail in the following section.

Since no two platforms are alike, there is no single standardised process for decommissioning. Each platform removal is therefore considered a unique project, requiring years of preparation and coordination between multiple stakeholders. However, high-level guidelines exist that define the general scope of decommissioning projects. One such framework is provided by Offshore Energies UK (OEUK), which identifies eleven decommissioning work breakdown steps, as presented in Figure 1.2.

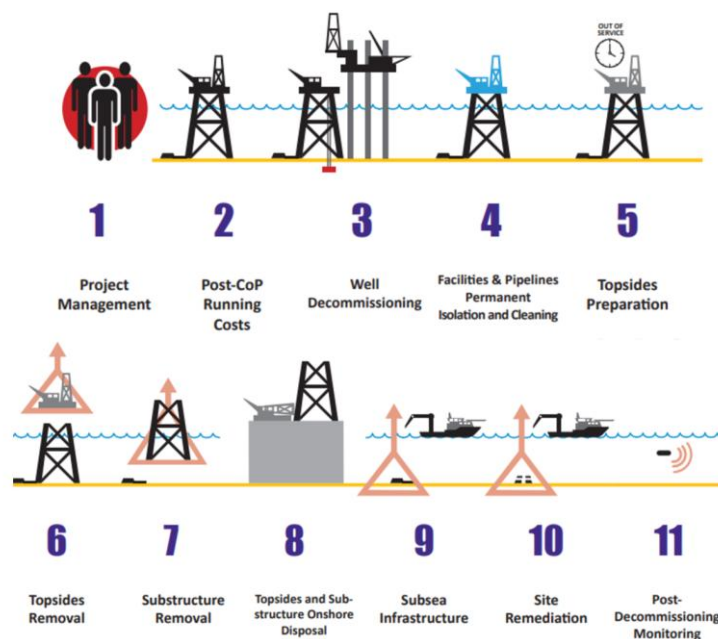


Figure 1.2: High-level decommissioning work breakdown structure [6]

Relevant to this thesis are the removal activities, which include the extraction of the topside, the jacket, and all associated equipment such as conductors, risers, wellheads, and subsea infrastructure. Three principal removal methods exist for these operations. Piece-small removal involves dismantling components into sections generally below 20 tonnes, requiring multiple lifting cycles and increased subsea intervention. Piece-large or modular removal includes lifting sections exceeding 20 tonnes, reducing offshore hours but introducing greater complexity and the potential need for structural reinforcement. Single-lift removal extracts the entire topside or jacket as one unit, minimising offshore hours and safety exposure, but presenting significant logistical challenges given the equipment required to handle structures weighing several thousand tonnes [7].

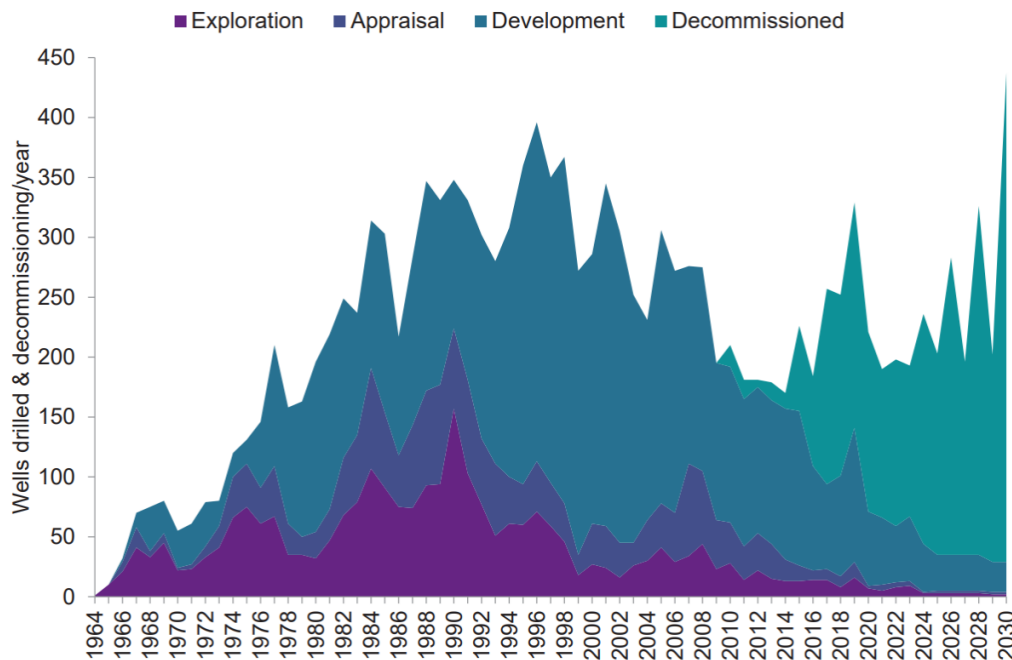


Figure 1.3: Well drilling and decommissioning activity in the UKCS [1]

Arguably the most technically challenging and costly step of the decommissioning process illustrated in Figure 1.2 is well decommissioning, which involves isolating the reservoir and all intermediate zones, followed by removing the wellhead and conductor [6]. In the UKCS, well decommissioning operations now outpace exploration, appraisal, and development activities combined, representing a 220% increase relative to production-adding activity since 2022. This shift is driving a significant surge in workload, with forecasts projecting 210 wells scheduled for decommissioning in 2025 compared to just 124 in 2024, as illustrated in Figure 1.3. These escalating activity levels are accompanied by rising costs, with platform well decommissioning expenditure increasing from £2.56 million per well in 2022 to £4.08 million in 2025, while subsea well costs have risen by approximately £1 million per well over the same period [1].

Within the well decommissioning process, conductor removal presents a particularly challenging activity that significantly impacts both project timelines and overall costs. The technical reasons for this are examined in the following section.

1.1.2 Conductors

The primary purpose of the conductor is to protect the production tubing, through which hydrocarbons are extracted from the reservoir, from environmental and geological hazards. Below the mudline, these include high-pressured zones, weak and fractured formations, unconsolidated formations, and sloughing shales encountered during drilling. Above the mudline, the conductor provides structural support to the inner casing strings and shields them from waves, currents, and corrosion throughout the platform's operational life [8].

In practice, wells are drilled and cased in several steps, with the conductor representing the outermost casing string. Typical conductor outer diameters range from 20 to 30 inches, though values up to 42 inches are encountered in more challenging environments. Inside the conductor, as illustrated in Figure 1.4, several progressively smaller casing strings are installed as drilling progresses deeper, with common inner diameters of 13 3/8, 9 5/8, and 7 inches, among others [8], [9]. All casing dimensions and material grades are governed by API and ISO standards, which specify acceptable ranges for outer diameter, wall thickness, yield strength, and chemical composition, selected based on expected loading conditions and operational stresses [9].

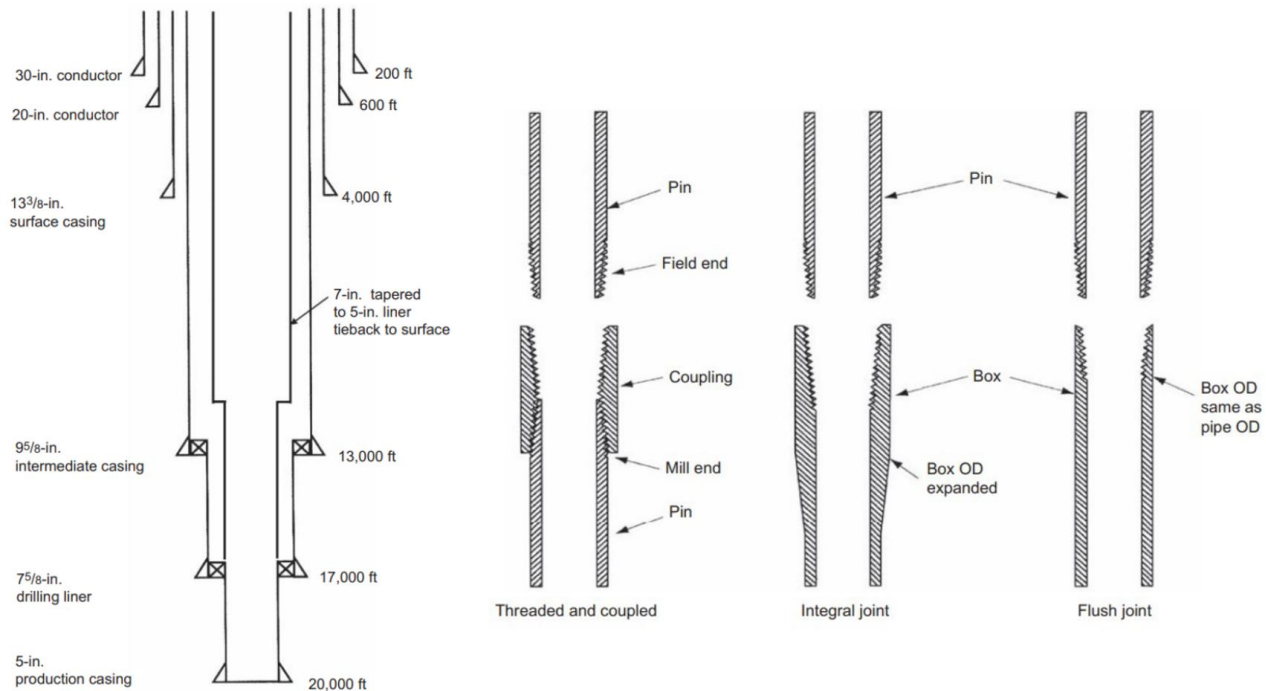


Figure 1.4: Typical casing program with typical dimensions (left) and casing connection types (right) [9]

Also depicted in Figure 1.4 are the typical connections between individual casing sections, a design characteristic that directly affects removal. Casing sections are delivered in approximately 12-metre lengths and joined by threaded pin-and-box connections, forming the conductor string. The most common type is the threaded and coupled connection, where pins at both ends of the pipe screw into a separate coupling piece. Beyond these standardised connections, proprietary premium connections are attractive to the industry as they allow for characteristics such as a flush joint, maximising clearance when the casing passes through the conductor guide frames [9]. However, these connections represent the weakest point in the conductor string and pose the greatest challenge during removal. After decades of offshore service, connector threads experience significant corrosion and fatigue degradation. The proprietary Talon connector, which appears frequently in conductor removal literature, is particularly problematic, as it was designed to resist compression but lacks the tensile strength required for extraction under surface pulling loads. Combined with age-related deterioration, pulling conductors with Talon connectors under tension from the surface is not feasible [10].

The internal configuration of a conductor assembly varies considerably between platforms. The annuli between individual casing strings may be fully cemented, partially grouted, or left open, depending on the operator and site conditions [11]. Additionally, the inner casings may be eccentric within the conductor, introducing uncertainty during removal operations regarding the exact position of the innermost string. Figure 1.5 showcases examples of single, triple, and quadruple casing configurations in both grouted and ungrouted arrangements, alongside an example of casing eccentricity.

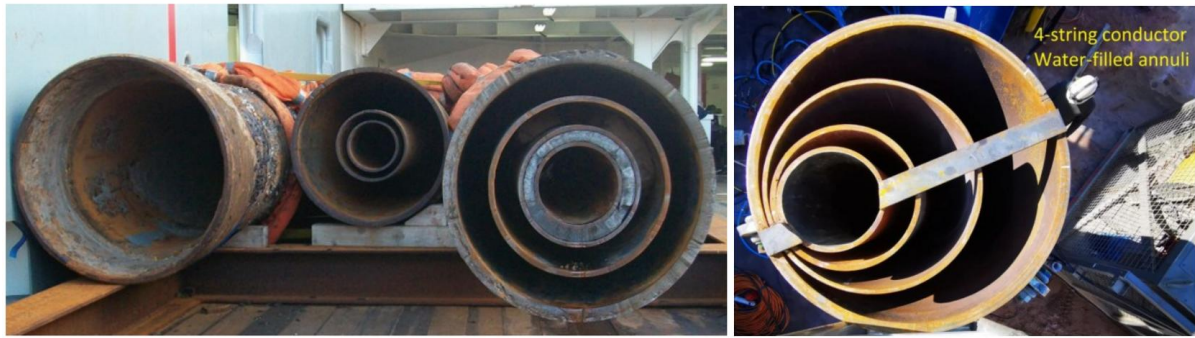


Figure 1.5: Single, triple, and quadruple well grouted and ungrouted casing configurations (left) [12] and eccentric multi-string conductor (right) [13]

Given the variability in diameter, connection type, grout configuration, and casing eccentricity described above, no two platforms present identical conductor specifications, and considerable variation can exist even between conductors on the same platform. These characteristics collectively drive the technical challenges of conductor removal, which are examined in the following section.

1.1.3 Conductor Removal Methods

Conductor removal currently relies predominantly on the piece-small removal approach, in which the conductor string is extracted in sections through a sequence of cutting and lifting operations commonly referred to as the cut-and-pull method. The general sequence begins with severing the conductor below the mudline, or in some specific circumstances at a defined elevation above it. Following this initial severance cut, the steps illustrated in Figure 1.6 are repeated until the conductor is fully removed. The conductor is first pinned so that the internal casing strings are mechanically secured and cannot drop during extraction. Using a lifting frame, the string is then raised to the required height above the deck and held in place through slips. The next section below is pinned, the section above is cut free and positioned on the platform deck or transferred to a nearby construction support vessel, and the lifting frame is repositioned to the next section. This cycle continues until the conductor string is completely removed [14].

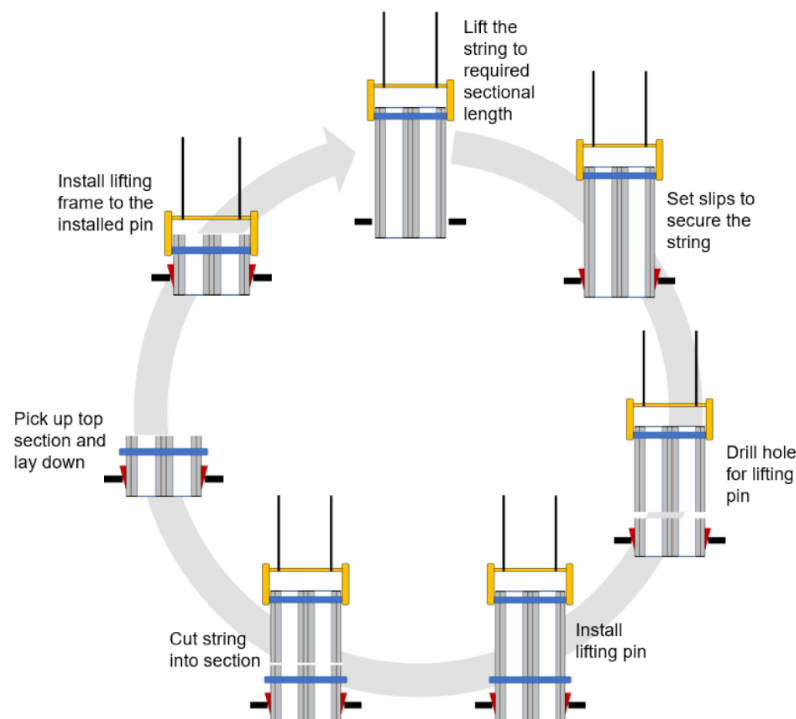


Figure 1.6: Process cycle of the cut-and-pull method for removing a multistring conductor [14]

Conductor removal is typically performed at the third step of the OEUK framework presented in Figure 1.2, although in practice it may also take place after topside removal at the seventh step. This sequencing introduces the additional complication that conductors are free to move under environmental loading once the topside is no longer present, increasing the risk of damage.

For each of the three principal operations, cutting, pinning, and lifting, a range of tools and technologies currently exists. Cutting can be performed using either internal or external tools, with the main categories being diamond wire cutting systems, shear cutters, internal milling cutters, abrasive waterjet systems, and laser cutting. External tools operate around the outside of the conductor and are well suited to multi-string cutting in a single pass, while internal tools are lowered through the innermost casing and are therefore limited by its relatively small inner diameter. For all cutting methods, confirmation that a complete cut has been achieved is not always straightforward, and dedicated cut-verification systems have been developed to address this [15].

Pinning is achieved either by drilling a hole through the full multi-string assembly and passing a pin through it, or by directly punching the pin through the casing walls without prior drilling. Both approaches can be executed above the waterline at the platform deck or subsea with ROV assistance. Once the casings are pinned and mechanically connected to one another, lifting can proceed. Lifting options include frames attached directly to the pins, internal lifting tools such as ball grabs that engage the innermost casing from below, and external clamps that grip the conductor in compression. In addition, several integrated systems combining cutting, pinning, and lifting into a single conductor removal platform have been developed, typically mounted on a jack-up style structure that spans the conductor and enables sequential operations across multiple wells [15].

Where connector integrity is insufficient for tensile lifting, as is the case with the Talon connector, an alternative approach involves installing a sacrificial casing, or lifting pipe, through the innermost casing and pinning it above and below the problematic connection, effectively bypassing the weak joint and transferring the extraction load through the internal pipe [15].

Finally, a fundamentally different removal strategy involves extracting the conductors together with the jacket in a single lift. In this approach, the conductors are first secured to the jacket structure through welding them to a dedicated conductor hang-off frame, with conductors that cannot be lifted in tension supported from below using internal lifting tools. The conductors are then cut at the required depth and the entire jacket section, conductors included, is extracted in one operation by a heavy-lift vessel. While this approach eliminates the repetitive cut-and-pull cycle entirely, it is highly constrained by vessel lifting capacity, regulatory requirements regarding the permitted cutting depth, and the significant planning effort required. To date, this method has been implemented only once during the decommissioning of the Brent Alpha platform in the North Sea [16].

1.1.4 Challenges and Limitations of Current Methods

Through a review of academic literature, case studies, supplier specifications, and patents, the key challenges that the offshore decommissioning industry encounters during conductor removal can be derived and categorised. While not all challenges apply to every conductor given the variability between offshore platforms, the identified issues provide a comprehensive overview of the typical difficulties encountered, largely independent of conductor-specific characteristics [15]. These challenges can be grouped into five categories, namely design and age-related issues, uncertainties, access constraints, operational limitations, and safety concerns.

From a design and age perspective, offshore platforms and their conductors were not originally designed with decommissioning in mind, and after decades of service the structural integrity of both the conductors and the conductor guide frames is often compromised. Corrosion and fretting damage commonly occur at contact points

with the guide frames and within the splash zone, while certain connectors, such as the Talon connector, were not designed to be lifted in tension and may disengage under tensile loads.

Uncertainties represent a further significant category of challenges. Grout levels in the annuli between casing strings are often unknown, casing eccentricity may be undetected until operations are already underway, and platform documentation is frequently incomplete or unreliable for older installations. Additionally, since friction between the surrounding soil and the steel casings cannot be relied upon, pinning is required to secure the casing strings even in fully grouted conductors.

Access constraints introduce additional complexity at every stage of the operation. The radial clearance between the outermost casing and the conductor guide frame is typically around 25 mm, significantly complicating lifting and handling operations. Obstructions on the conductor, including welded repair patches, installed repair sleeves, or cutting-induced outward flaring caused by certain cutting methods, may further prevent the conductor from passing through the guide frame during extraction. Internal tools are additionally constrained by the relatively small inner diameter of the innermost casing, commonly between 7 and 13 inches, while external access may be restricted by jacket structural members or the close spacing between adjacent conductors.

Operational limitations further compound the difficulties described above. Marine growth increases weight and may obstruct passage through the guide frames, welding operations are not always feasible given the corrosion state and limited access, and cut confirmation is not straightforward for certain cutting methods. When conductors are removed after topside extraction, they are additionally free to move under environmental loading, increasing the risk of damage at the guide frames.

Finally, safety concerns related to residual hydrocarbons within the casings and the risk of damage to seabed infrastructure including pipelines and risers during handling impose strict constraints on the available methods and procedures.

From these identified challenges, clear limitations of the current methodologies emerge. Cutting remains the bottleneck of the overall removal process, with cut verification representing a persistent challenge particularly for below-mudline operations. However, active research and ongoing development within the industry continue to address these cutting limitations, and incremental improvements are expected to eventually yield more efficient solutions. This thesis therefore tackles a different and less explored limitation, namely the inability of existing methods to reliably handle the structural uncertainty and physical variety of conductors encountered in practice.

This gap persists largely because the offshore industry is inherently risk averse, and operators tend to rely on proven methods despite their inefficiencies, driven by the high costs and safety concerns associated with offshore operations. Furthermore, almost all technologies currently applied to conductor removal were originally developed for other offshore applications such as jacket or pipeline cutting and lifting and subsequently adapted for conductors. The highly case-specific nature of the operation, where no two platforms or conductors are identical, makes generalised solutions difficult to develop and further reduces the incentive for dedicated innovation, leaving many gaps unresolved.

A common characteristic of all current removal approaches is their reliance on point load lifting. Some conductors are lifted in tension while others, where connectors or structural condition prohibit tensile loading, must be lifted in compression. In both cases the load is transferred through a single cross-section of the conductor, introducing additional risks. Point loading may be insufficient for severely corroded or damaged conductors, where weakened casings or internal grout may collapse under self-weight. Furthermore, compression lifts using point loads are inherently unstable, as the centre of gravity of the conductor lies above the lifting point.

The research opportunity identified in this thesis therefore lies in the development of a distributed-load lifting concept, one that applies lifting forces along the outer surface of the conductor over its entire length or a significant portion of it, reducing local stresses, improving stability, and mitigating the risk of collapse at damaged sections or problematic connections. Such a solution would be inherently more robust against the variability between conductors and platforms, potentially enabling the removal of conductors that current methods cannot safely handle. To develop such a concept, the field of soft robotics and more specifically compliant gripping mechanisms was identified as the domain with the greatest potential for a transferable solution, as examined in the following section.

1.1.5 Soft Robotics as a Technology Transfer Opportunity

The identified limitation of current conductor removal methods, namely the absence of a solution capable of applying distributed surface loads, motivates the examination of soft robotics as a source for technology transfer. Soft grippers and compliant mechanisms are constructed from materials with relatively low Young's modulus, giving them an inherent flexibility and adaptability that rigid grippers cannot replicate. Unlike traditional rigid grippers, which require precise control and well-defined object geometries, soft grippers can conform to irregular or variable shapes without detailed geometric information [17]. This adaptive capability makes them particularly suitable for handling objects with uncertain configurations, such as conductors with unknown internal structures, varying corrosion states, and irregular outer surfaces.

However, soft robotics research remains predominantly confined to laboratory-scale applications and to fields such as industrial automation, medicine, and underwater sampling, where manipulation of relatively lightweight objects is expected [17]. The highest reported load-bearing capacities in the literature do not exceed a few hundred kilograms, far below the multiple tonnes required for conductor removal. Scaling soft gripping mechanisms to this magnitude introduces fundamental challenges, as the gripping force, which depends primarily on contact area, scales with the square of the characteristic length, while the mass of the gripper itself scales with its volume, growing with the cube, compounding difficulties related to power supply, actuation, structural integrity, and deployment at scale [18]. The transferability of soft gripping principles to heavy offshore lifting therefore represents a significant and largely unexplored research opportunity, one that directly guides the research objective of this thesis. A more detailed examination of soft gripper concepts, including gripping methods, actuation strategies, materials, and current capabilities, is presented in the literature study of this report.

1.2 Research Objective and Questions

Having established the necessary background and motivation, the research objective of this thesis is to propose a method for distributing the lifting load over the surface of a conductor using a compliant gripping mechanism. The expected outcome is twofold. The first objective is to design a gripping concept that improves both the safety and efficiency of conductor removal by addressing the limitations of current point-load approaches. The second objective is to demonstrate the scalability of such a compliant mechanism to the load magnitudes encountered in offshore conductor removal, given that soft robotic gripping solutions currently remain confined to small-scale applications. To support both objectives, the design, fabrication, and testing of a prototype forms an integral part of the research.

To structure these objectives in concrete and measurable goals, the following main research question (MRQ) is defined.

MRQ: How can a compliant gripping mechanism be designed to distribute lifting loads over the surface of conductors during removal from fixed offshore jacket platforms?

To further detail the MRQ and formally define the sub-objectives, the main research question is divided into three sub-research questions (SRQ).

SRQ1: Which compliant gripping concept is most suitable for conductor removal?

SRQ2: How does the selected concept scale to full-load conductor removal and what does a feasible full-scale design require?

SRQ3: How can a scale prototype be designed and tested to demonstrate the feasibility of the selected concept?

Answering these research questions is expected to yield three principal deliverables. The first is a concept selection for a compliant gripping mechanism, identifying the most suitable gripping principle from a structured evaluation of candidate concepts. The second is an analytical model describing the governing mechanics of the selected concept, accompanied by a full-scale feasibility assessment including dimensioning and a definition of the removal procedure. The third is a scaled prototype, fabricated and tested to validate the analytical model and to demonstrate the physical feasibility of the gripping mechanism at laboratory scale.

1.3 Methodology

The research methodology is structured around the three defined sub-research questions and follows an iterative design approach grounded in the methodical design model of Pahl et al. (2007), consistent with the VDI Guidelines 2221 and 2222.

The first phase of the research addresses SRQ1 through a literature study into soft robotic gripping concepts, examining actuation strategies, gripping mechanisms, materials, and load-bearing capabilities across different technologies. The knowledge gained from this study feeds directly into the conceptual design phase, in which the requirements on the gripper are first formalised, followed by the development of a functional structure that decomposes the design problem into individual functions. For each function, a range of technical solutions is identified and subsequently combined into candidate concepts. These concepts are then evaluated against technical and quality criteria derived from the defined requirements, yielding a selected concept and an answer to SRQ1.

The second phase addresses SRQ2 through the detailed design of the selected concept. An analytical model describing the governing mechanics of the concept is developed, and critical design parameters are determined through successive design iterations, each adding further detail and refinement. The outcome of this phase is an estimated dimensioning of the full-scale design, a definition of the operational procedure, and a feasibility assessment of the concept at the load magnitudes relevant to conductor removal.

The third phase addresses SRQ3 through the design, fabrication, and testing of a scaled prototype. The prototype serves two purposes. It demonstrates the conceptual feasibility of the gripping mechanism, and it provides experimental data against which the analytical model can be validated. The results of prototype testing feed back into the full-scale design, informing any necessary corrections or revised assumptions and thereby further strengthening the answer to SRQ2. This feedback loop between analysis, prototyping, and design refinement continues iteratively until the research objectives are met to the degree defined in the scope of the thesis, presented in the following section.

1.4 Scope

Given the breadth of the topic, a defined scope is necessary to keep the thesis focused on answering the research questions within the available timeframe. The following boundaries apply throughout the work.

The scope is limited to the gripping and lifting mechanism itself. All other aspects of the conductor removal procedure, including cutting, pinning, and deck handling, are considered out of scope. Where relevant, these are briefly discussed in the context of the operational procedure or identified as open points for future research.

The thesis focuses exclusively on conductors from fixed jacket offshore platforms, as these represent the majority of installed structures currently awaiting decommissioning and present the most clearly defined set of design constraints.

The research is conducted in collaboration with Allseas, an offshore contractor specialising in heavy lift and decommissioning operations, and owner of the *Pioneering Spirit*, one of the world's largest heavy-lift vessels [19]. Allseas currently relies on subcontractors for conductor removal, and the development of proprietary methods and equipment in this area represents a direct business interest. As a result, the scope and design requirements of this thesis are partly shaped by requirements set by Allseas. To develop a relevant full-scale design, the upcoming decommissioning of the North Cormorant platform is used as the reference case, with conductor dimensions, weight, and placement taken from documentation provided by Allseas.

Finally, given that this is, to the author's knowledge, the first compliant gripping concept developed specifically for offshore conductor removal, this thesis is positioned as an early-phase design study. Accordingly, a number of simplifying assumptions are adopted to manage scope. From an operational perspective, these include the neglect of buoyancy effects, the idealisation of the conductor as a perfect cylinder, and the assumption that all preparatory offshore work is completed prior to gripping. From an analytical perspective, simplifying assumptions regarding contact mechanics, friction behaviour, and loading conditions are also adopted. All assumptions and simplifications are stated and justified where they arise during the development process.

1.5 Structure of the Report

This report is structured in eight chapters. Following this introduction, Chapter 2 documents the literature study into soft robotic gripping concepts and presents the key findings relevant to the concept development phase. Chapter 3 covers the conceptual design process in full, beginning with the formalisation of the design requirements and concluding with the evaluation of candidate concepts and the selection of the most suitable one, thereby answering SRQ1. Chapter 4 expands on the selected concept by developing the analytical model in detail. Chapters 5 and 6 address SRQ3, with Chapter 5 presenting the prototype design and Chapter 6 reporting and discussing the experimental validation of the analytical model. Building on these results, Chapter 7 provides a full-scale design recommendation and defines an exemplary operational procedure. Together, Chapters 4 and 7 collectively answer SRQ2. Finally, Chapter 8 summarises the work and its findings and presents recommendations for future research.

2 Literature Study

Following the motivation established in Chapter 1, the field of soft robotics was identified as the most promising source for a transferable gripping solution, given the inherent ability of compliant mechanisms to generate distributed gripping forces over irregular, unknown, or damaged surfaces. This behaviour is directly relevant to conductor removal, where outer casing surfaces are frequently corroded, geometrically imperfect, may carry repair patches, and vary considerably between platforms.

The goal of this chapter is therefore to identify gripping technologies and concepts from the soft robotics literature that show potential for transfer to the conductor removal problem. The study is targeted rather than exhaustive, designed to produce results that feed directly into the concept selection process in Chapter 3, providing both design requirements and inspiration for candidate solutions in the morphological chart. Accordingly, the review is specifically tailored to gripping technologies applicable to vertical cylindrical objects and does not represent a complete survey of the soft robotics field, which spans a vast and rapidly developing range of applications and solutions. In the following sections, the fundamentals of soft robotic gripping are first presented, covering gripping methods, actuation strategies, and materials. Relevant gripping concepts that demonstrate transferability and scalability potential are then examined as inspiration for the further design. Finally, the current capabilities and limitations of compliant gripping technologies are discussed, concluding with a summary of the key findings that inform the design phase.

2.1 Fundamentals of Soft Robotics

Soft robotics is a broad and rapidly developing field with no single universally accepted definition. Definitions are often centred around the use of inherently compliant materials, the ability to exhibit large strains, or the capacity for variable stiffness [20]. Classification follows a similar variety, with most frameworks organising soft grippers by their actuation method, gripping principle, or the materials they are made from. The following sections examine each of these aspects, focusing on soft gripping applications relevant to the conductor removal problem.

2.1.1 Soft Gripping Methods

Gripping forces in soft robotic systems can be generated through a range of physical phenomena. Hirai (2024) classifies manipulation forces into contact-based forces, including normal, friction, and adhesion forces, and non-contact forces such as suction, Bernoulli, electrostatic, and magnetic forces [21]. Given the significant load requirements of the conductor removal problem, not all of these are equally relevant. The following subsections therefore categorise the gripping methods with the intended application in mind, covering mechanical grasping, suction gripping, electromagnetic gripping, and methods identified as non-relevant for conductor removal.

Mechanical Grasping

Mechanical grasping relies on direct contact between the gripper and the object, with friction and normal forces responsible for holding and manipulating the object. In friction-based gripping, the gripping mechanism applies normal forces to the object surface, through which friction is generated to counteract the weight of the object and resist relative movement [18]. In the context of conductor removal, the friction conditions between any gripper material and the conductor surface are inherently uncertain, as the outer casing condition is largely unknown prior to operations. However, this uncertainty is less problematic for a friction-based compliant gripper than it might first appear. Features that are considered challenges in conventional removal methods, such as marine growth, surface corrosion, rough patches, and welded repairs, would in fact be beneficial for a friction-based gripping approach, as they increase the effective friction coefficient at the contact interface and thereby increase the holding capacity of the gripper.

The second method of mechanical grasping is form closure gripping. Rather than relying on friction, form closure uses geometric interlocking between the gripper and the object, with normal contact forces sufficient to resist the applied load. While inducing geometric interlocking on a smooth vertical cylinder such as a conductor might appear inherently difficult, two practical approaches exist. The gripper can engage with existing surface features such as marine growth, welded repair patches, or the damaged conductor surface, again benefiting from the characteristics that challenge traditional removal methods, or alternatively engage with purposefully introduced geometries such as cut holes or windows. Both approaches are illustrated in Figure 2.1, where the generated forces counteracting the weight of the cylinder are depicted.

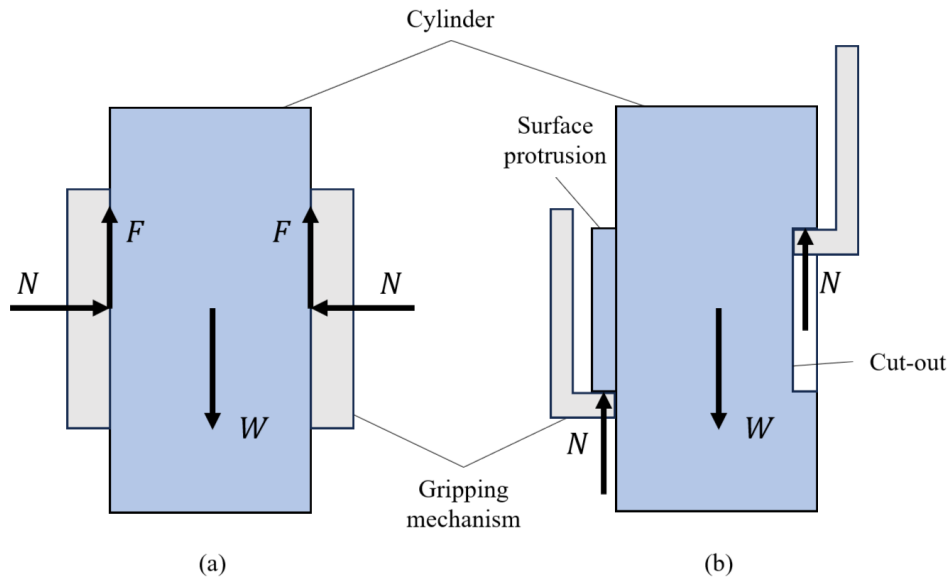


Figure 2.1: Mechanical grasping methods applied to a vertical cylinder. (a) Friction gripping. (b) Form closure gripping through interlocking with a surface protrusion or cut-out

Suction Gripping

Suction gripping generates holding forces through a pressure differential between the interior of a suction cup and the object surface. Once the suction cup contacts the object, the pressure inside the cup is reduced, generating a suction force that holds the object in place. Suction cups are constructed from soft materials, allowing them to deform passively upon contact and conform to the object surface to maintain the pressure seal. A suction gripper can generate substantial lifting forces when the contact surface is smooth and flat. However, when the surface is uneven or irregular, gaps between the cup and the surface reduce the internal pressure differential, significantly limiting the generated holding force [21]. For conductor removal, this represents a fundamental limitation, as the outer casing surface is both convex and irregular, making a reliable pressure seal difficult to guarantee. The subsea environment further complicates matters, introducing additional pressure equalisation challenges. Nevertheless, suction-based pipe gripping concepts do exist and will be examined in Section 2.2.3. The suction gripping principle is illustrated in Figure 2.2.

A further gripping method worth introducing at this point is jamming gripping, first proposed by Brown et al. (2010). Jamming gripping is notable as it combines all three of the mechanical gripping principles discussed above, namely friction, form closure, and suction, into a single mechanism. A flexible membrane filled with granular material is pressed against the object, after which vacuum is applied inside the membrane. This causes the granular particles to jam against one another, stiffening the gripper around the object. The resulting holding force is generated simultaneously through friction at the contact interface, suction where an airtight seal forms between the membrane and the object surface, and geometric interlocking where the gripper envelops existing surface features [22]. Jamming gripper designs will be examined in more detail in Section 2.2.4.

Electromagnetic Gripping

Electromagnetic gripping generates holding forces through magnetic attraction between the gripper and a ferromagnetic object, using either permanent magnets or electromagnets. In principle this is applicable to conductor removal given that the outer casing is a steel structure. However, the load magnitudes required demand a large number of magnets or a single magnet of impractical size and weight. Combined with the presence of marine growth between the steel surface and the magnets, the convexity of the conductor, and the challenges of the subsea environment, electromagnetic gripping is not considered a viable primary gripping mechanism for conductor removal. The principle is depicted in Figure 2.2.

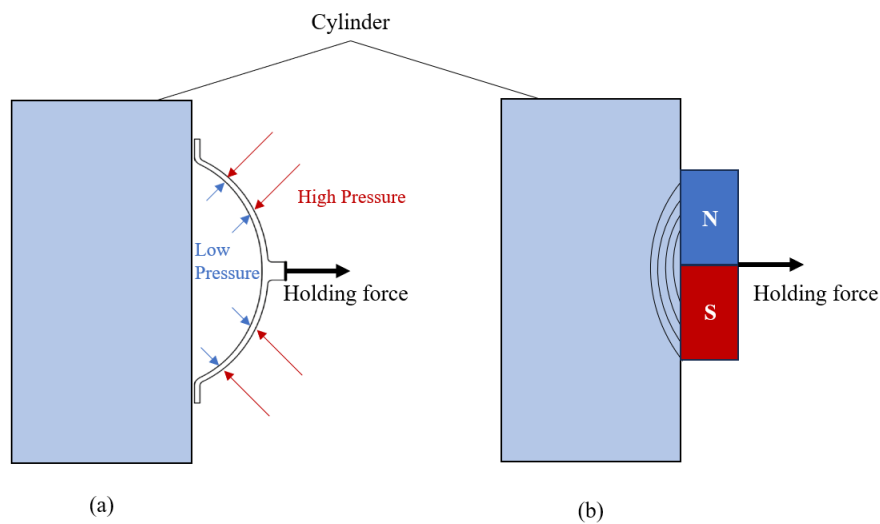


Figure 2.2: Gripping methods applied to a vertical cylinder. (a) Suction gripping. (b) Electromagnetic gripping

Other gripping methods

A range of further gripping methods exist in the literature that rely on small-scale force phenomena. Adhesion-based methods include capillary forces generated by liquid bridges between the gripper and the object, van der Waals forces arising from molecular attraction at very small separations, and dry adhesion through the microstructure of surfaces. Electro-actuated methods generate holding forces through electrical phenomena, including electrostatics, ultrasonic pressure, and optical pressure. These methods remain largely confined to research settings and are applicable only at microscale. Scaling such principles to the load magnitudes required for conductor removal is not considered feasible.

In summary, the most promising gripping principles for lifting large cylindrical structures are friction and form closure. As established above, the surface characteristics of aged conductors, including corrosion, marine growth, and repair patches, represent potential benefits rather than obstacles for both of these methods, increasing friction at the contact interface and providing geometric features for interlocking. Suction gripping and electromagnetic gripping are considered less promising due to the surface irregularity of the conductor and the impractical equipment sizes required at the relevant load magnitudes.

2.1.2 Actuation Methods

The actuation of compliant grippers is a broad research field with a wide variety of methods, however the options relevant to the conductor removal problem are considerably narrowed by the offshore subsea environment and the required load magnitudes.

Pneumatic and hydraulic actuation

Pneumatic actuation relies on pressurised air applied to elastic tubular structures, causing them to bend, twist, or elongate depending on their geometry and material properties [20]. For large-scale subsea operations,

pneumatic actuation is generally avoided, as the increasing hydrostatic pressure with depth requires ever higher internal gas pressure to maintain actuation, introducing buoyancy effects and significant safety concerns related to compressed gas subsea.

Hydraulic actuation operates on the same principle but uses an incompressible liquid such as oil or water as the working fluid, eliminating the buoyancy and gas compression concerns associated with pneumatics [20]. However, the requirement for high operating pressures at depth remains a challenge, and hydraulic systems introduce complexity through the need for pumps, sealing mechanisms, and fluid management in a subsea environment.

Tendons

A promising actuation method for the conductor removal application is tendon actuation, in which wire or cable structures deform or change the overall shape and stiffness of the gripper through tension. Unlike pneumatic and hydraulic actuators, which can apply both positive and negative pressures, tendons operate exclusively in tension. Tendons are most commonly controlled through servo motors and pulleys, while shape memory alloys, materials that return to a predefined geometry upon heating, represent an alternative actuation source, though their application remains limited to small-scale systems far below the load magnitudes required for offshore conductor removal [20].

The main challenges associated with tendon actuation at the relevant scale are twofold. First, tendons have a tendency to straighten along curved surfaces, making precise shape control around a cylindrical conductor difficult. Second, at the scale of conductor removal the self-weight of the tendon system becomes significant, and providing adequate structural support for the tendons themselves introduces additional design complexity that is typically ignored in small-scale research applications. Nevertheless, tendon actuation remains promising for this application given its potential to generate large forces through simple and robust mechanical means.

Other actuation methods

Several further actuation methods appear in the soft robotics literature, including dielectric elastomers, piezoelectric actuators, and hydrogels. Dielectric elastomers deform under an applied voltage through electrostatic squeezing of an elastomer layer, piezoelectric actuators produce micrometre-scale deformations through electromechanical coupling, and hydrogels generate volume changes through swelling and deswelling of a polymer mesh [20]. All three are fundamentally limited in force output and operating scale, with applications confined to micro and small-scale robotics, and are therefore not considered further for the conductor removal application.

In summary, the most promising actuation method for a compliant conductor gripper is tendon actuation, which is relatively simple, capable of delivering large forces, and potentially integrable with existing offshore equipment. Pneumatic and hydraulic actuation are not discarded however, as they remain widely used in soft gripper applications and may prove relevant depending on the selected concept. Finally, conventional mechanical drive systems such as motors, winches, and hydraulic cylinders must also be considered, as they represent proven offshore technology and may provide a practical and robust means of actuating a compliant gripping mechanism at the required scale.

2.1.3 Gripper Materials

The materials used in soft robotic grippers span a wide range. Common choices in research include silicone-based elastomers, hydrogels, shape memory alloys, and fabric-rubber composites, each offering different combinations of compliance, stretchability, durability, and in some cases self-healing capability [23]. For underwater applications specifically, silicone remains the most widely used material due to its robustness in wet environments and ease of manufacture.

However, a key consideration for this thesis is that compliant grippers are not defined by their material but by their ability to conform around irregular objects, distributing contact forces and offering the adaptability that rigid grippers cannot provide. In this sense, a gripper constructed from polyester rope or even steel wire rope can equally be considered a compliant gripper, as it conforms to unspecified surfaces and distributes load along its contact length. This distinction is particularly important at the scale of conductor removal, where the specialised elastomers and polymers common in laboratory research may not be manufacturable at the required dimensions or may prove cost inefficient. The material selection for a full-scale compliant conductor gripper should therefore be guided by load-bearing capacity, durability in the offshore environment, and practical manufacturability and availability at scale. In this regard, materials that are unconventional in soft robotics research but well established in the offshore industry, such as steel wire rope and synthetic fibre rope, should not be overlooked.

2.2 Gripper Designs

This section presents existing gripping concepts centred around the grasping and manipulation of cylindrical objects, selected for their relevance and potential transferability to the conductor removal problem. The focus lies on concepts that may be applicable to the full-scale removal problem when appropriately scaled. For clarity, the concepts are grouped using a simplified appearance-based categorisation into sleeve-like, wrapping, suction-based, and jamming gripper designs.

2.2.1 Sleeve-Like Gripper Designs

Sleeve-like gripper designs present a potentially viable gripping concept for conductor removal, as they conform around a cylindrical structure and generate gripping forces through a combination of friction and form closure. Such sleeve concepts already exist in the offshore industry under various names, including cable sleeves, wire socks, and Chinese fingers, where they are used for handling, pulling, and suspending umbilicals, power cables, and other cylindrical elements, as seen in Figure 2.3 [24].



Figure 2.3: Braided sleeve for handling cylindrical objects [24]

The working principle is that as the braided sleeve elongates axially under tension, the interwoven ropes contract radially, generating radial pressure and therefore friction that counteracts the weight of the suspended object. The same mechanics are present in the well-known Chinese finger trap (CFT), a toy that tightens its grip as both ends are pulled. Lu et al. (2024) investigate the geometry and mechanics of the CFT through an analytical model validated by numerical simulations and experiments, demonstrating that as the structure compacts under load, the axial force transitions from linear to exponential behaviour. The authors further note that the developed model and working principle present potential for the development of a compliant gripper [25]. This self-tightening behaviour, combined with the ability of the CFT to distribute loads over the surface of a cylindrical object, makes it particularly interesting for the conductor removal problem.

Building on the same self-forcing mechanics, several braided sleeve gripper concepts exist in the literature. Shang et al. (2019) present a braided tube gripper that exploits a self-forcing mechanism in which the holding

force increases proportionally with the applied load, demonstrating a 120% increase in holding force due to this effect and successfully handling objects of varying shapes, weights, and rigidities [26]. A further concept by Pan et al. (2025), inspired by the swallowing behaviour of snakes, achieves adaptive gripping through controlled contraction of a braided fibre tube, operating on the same self-forcing principle as Shang et al. though relying on manual actuation rather than an integrated actuation mechanism [27].

Furthermore, Phan et al. (2022) present a soft tubular gripper constructed by braiding multiple hydraulic filament artificial muscles, which upon pressure release contract and grip a target object through friction, successfully lifting objects up to 83 times the weight of the gripper [28]. At a slightly larger scale, Stoy et al. (2021) demonstrate that braided structures can be constructed at the metre scale while remaining lightweight and self-supporting, designing a braided manipulator made from glass-fibre reinforced rods measuring over a metre in height and weighing only 35 grams [29]. Although not developed as a gripper, this work provides evidence that braided compliant structures are potentially scalable to dimensions relevant to offshore applications while maintaining their ability to contract. All introduced concepts are presented in Figure 2.4.

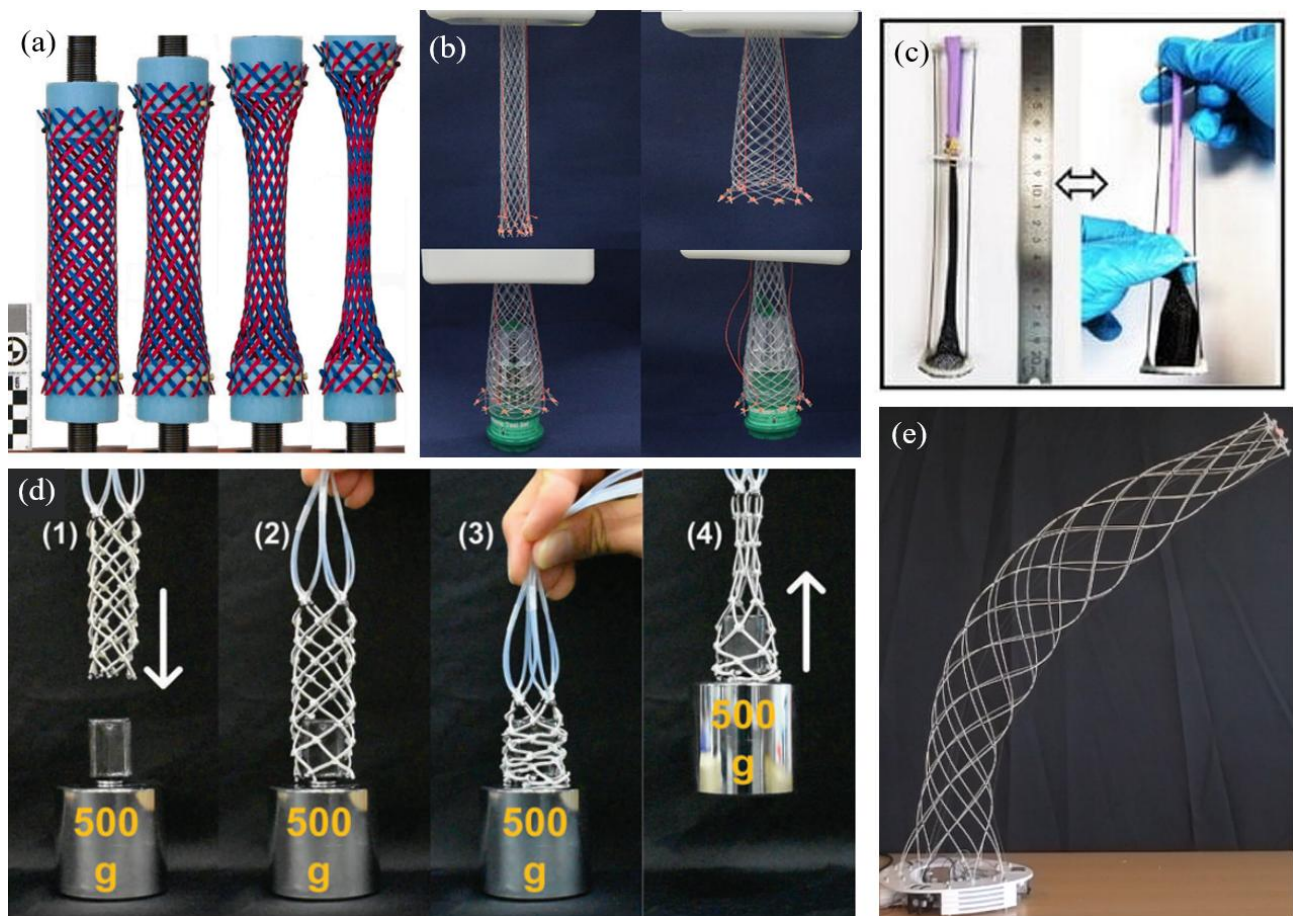


Figure 2.4: Sleeve-like braided gripper concepts. (a) Chinese finger trap prototype during tensile testing [25]. (b) Braided tube gripper in its normal and deformed state and while gripping and lifting an object [26]. (c) Variable caliber braided tube gripper prototype [27]. (d) Hydraulically actuated tubular braided gripper lifting a weight [28]. (e) Braided manipulator [29]

Although not sleeve-like in its form, the band-based gripper developed by Lee et al. (2025) operates on the same mechanical principle as the braided sleeve concepts described above. Inspired by the human iris mechanism, the gripper consists of two concentric rings, a stationary inner ring where the bands are fixed and a rotating outer ring onto which the opposite ends of the bands are spooled. As the outer ring rotates relative to the inner ring, the bands twist around and conform to the target object, generating a gripping force through friction in a manner analogous to the constriction of a braided sleeve. The authors demonstrate that the gripper can handle a wide variety of object geometries and sizes, and exhibits a self-centring behaviour that reduces the precision

required during deployment. A further important finding is that stretching the bands into their elastic region through continued rotation significantly increases the generated gripping force [30]. The self-centring property and the ability to conform around irregular geometries are characteristics of particular relevance to the conductor removal problem, where precise positioning of the gripper around the conductor may not always be achievable.

A further concept operating on the same relative rotation principle is the multi-stem twining gripper proposed by Shan et al. (2023). Rather than using individual bands, multiple longitudinal fibre ropes are arranged into a sleeve-like structure and twisted around the object through the relative rotation of two concentric flanges on the same side of the gripper. As the fibres wind around the object, the gripping force increases exponentially with the twining length, the curvature of the object, and the friction coefficient, with the authors demonstrating a high load capacity and strong adaptability across different object geometries [31]. A similar rotational principle is employed by the ROSE gripper proposed by Bui et al. (2023), where a silicone membrane in the shape of a ferrule sleeve buckles and conforms around the object through a single rotation of the base, generating a large contact area and achieving a high payload-to-weight ratio [32].

Moreover, He et al. (2025) propose an Omega-shaped enveloping gripper in which a silicone sleeve with a stiffer outer and more compliant inner membrane buckles radially inward through a single base rotation, conforming around the object. A notable feature is that the buckling simultaneously generates radial compression and an axial lifting force, reducing the friction required to sustain the grip and thereby minimising contact pressure on the grasped surface [33]. A final rotation-based sleeve-like gripper is the BerryTwist proposed by Elfferich et al. (2025), designed for berry harvesting. The gripper consists of a fabric tube whose two ends are rotated in opposing directions, causing the tube to contract radially and envelop the object with uniform pressure distribution [34].

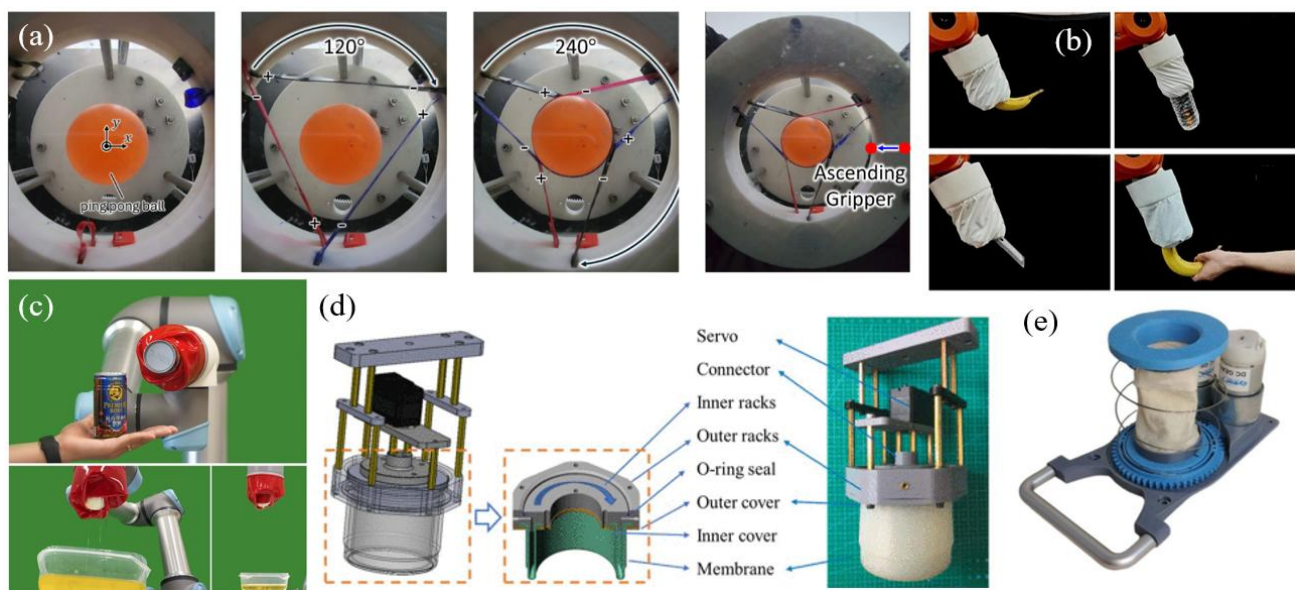


Figure 2.5: Sleeve-like torsional gripper concepts. (a) Band-based iris gripper in the process of gripping a ball [30]. (b) Multi-stem twining gripper grasping objects of different shapes [31]. (c) ROSE gripper picking a coffee can and gripping an egg submerged in an oil container [32]. (d) Omega-shaped torsional buckling gripper [33]. (e) BerryTwist tube gripper [34]

All of the twisting gripper concepts presented above are particularly interesting for conductor removal, as they distribute the gripping load over the outer surface of a cylindrical object through friction, with a relatively simple rotational actuation that is compatible with conventional offshore drive systems. The concepts are illustrated in Figure 2.5.

A final category of sleeve-like grippers are inflatable designs, where a soft membrane is inflated to engage with and conform around the object, generating gripping forces through friction. A notable example is the multimodal

enveloping gripper proposed by Hao et al. (2021), built as an accordion structure with internal fluid chambers that can contract around an object, expand within an internal opening, or generate suction against a flat surface, with a gecko-inspired interior surface further enhancing friction and load capacity [35]. A further inflatable design is presented by Li et al. (2021), where pneumatic artificial muscles wound around a soft frame contract upon inflation and tighten around the object, achieving a high payload-to-weight ratio [36]. Building on the same winding principle but at a significantly larger scale, Li et al. (2023) present a soft-rigid enclosed gripper capable of grasping the human body. The design combines a soft pneumatic winding structure with a rigid outer shell to achieve the omnidirectional load capacity that pure soft structures cannot provide at this scale [37]. The three gripper designs are showcased in Figure 2.6.



Figure 2.6: Inflatable sleeve-like gripper concepts. (a) Multimodal enveloping gripper demonstrating three grasping modes [35]. (b) Untethered high-load soft pneumatic gripper [36]. (c) Soft-rigid enclosed gripper [37]

A variety of further inflatable sleeve-like gripper designs exist in the literature beyond the examples presented above. The inflatable sleeve concept shows potential scalability to the conductor removal problem, as the reviewed designs already demonstrate progression toward larger payloads and higher payload-to-weight ratios while simultaneously addressing the structural support requirements of the gripper itself. However, the concerns regarding pneumatic and hydraulic actuation in a subsea environment, as discussed in Section 2.1.2, remain and represent a significant practical challenge for large offshore applications.

2.2.2 Wrapping and Envelopment Gripper Designs

A different category of compliant grippers encompasses wrapping and envelopment designs, where fingers, arms, or tendrils wrap around objects and grasp them through friction, distributing the gripping load over the contact surface. A directly relevant example is the boa-type gripper developed by Galloway et al. (2016), deployed on an ROV at depths greater than 100 metres to retrieve coral specimens. Through hydraulic actuation, the gripper transitions from a straight state into a coiled configuration, wrapping around objects with uncertain size, shape, and stiffness and distributing the gripping load along the contact surface [38]. The underwater deployment and the requirement to grasp and lift an irregular cylindrical object share conceptual similarities with the conductor removal problem, and the hydraulic actuation is directly compatible with subsea operational requirements.

In their majority, wrapping gripper concepts are inspired by biological structures such as octopus arms, elephant trunks, and snakes, which are soft appendages capable of coiling around objects with considerable adaptability. A notable example is the OctArm continuum manipulator developed by McMahan et al. (2006), a pneumatically actuated multi-section arm that successfully demonstrated adaptive grasping of objects across a wide range of sizes and shapes, including in an underwater environment [39]. Inspired by the winding locomotion of arboreal snakes, Liao et al. (2020) propose a soft rod-climbing robot consisting of two winding actuators and a telescopic actuator. The winding actuators anchor the robot onto cylindrical objects while the telescopic actuator drives linear climbing motion, demonstrating a payload capacity exceeding 25 times the robot's self-weight [40]. Similarly inspired by the octopus, Ito et al. (2020) develop a soft multi-legged climbing robot in which flexible silicone legs conform around cylindrical objects with a simple tendon-pulling mechanism, enabling adaptive grasping and climbing of objects with unknown geometries [41].

A further category of wrapping grippers, less directly relevant to conductor removal due to their small scale but notable for their working principles, are coiling and entanglement grippers. Chen et al. (2025) present a snake-inspired coiling gripper driven by a bilayer structure that responds to external stimuli, enabling dynamic coiling around and release of cylindrical objects including within confined spaces [42]. Qing et al. (2024) propose a tendril-like hydraulic gripper capable of grasping cylindrical objects through both external coiling around the object and internal entanglement within confined spaces [43].

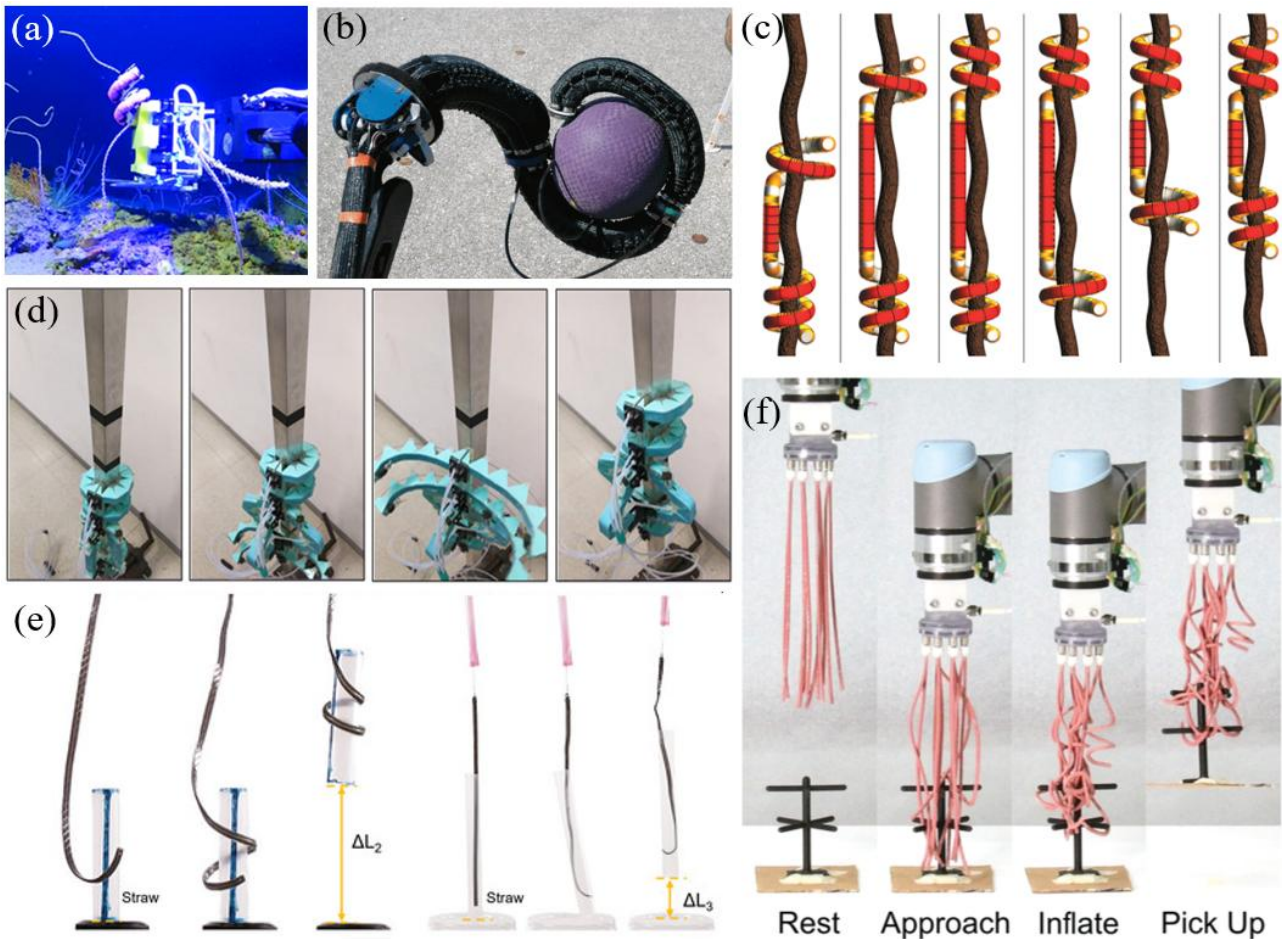


Figure 2.7: Wrapping and envelopment gripper concepts. (a) Boa-type gripper collecting a coral specimen [38]. (b) OctArm manipulator grasping a sphere [39]. (c) Soft rod-climbing robot [40]. (d) Soft multi-legged octopus-inspired robot climbing a pipe [41]. (e) Hydraulically actuated tendril-like gripper performing external and internal grasping [43]. (f) Filament-array gripper entangling around object [44]

A final concept of interest is the filament-array gripper developed by Becker et al. (2022), consisting of pneumatically actuated slender elastomeric filaments that curl and coil under pressure, entangling with one another and with the target object to achieve stochastic, topological grasping without perception, planning, or feedback [44]. The gripper successfully handles objects of varying size, shape, and topological complexity by creating distributed soft contacts across multiple filaments. While direct lifting of a smooth vertical cylinder such as a conductor is limited, the concept could be of interest if the conductor features surface protrusions or cut-outs that the filaments could entangle with, as discussed in Section 2.1.1.

The wrapping and envelopment gripper concepts presented in this section are illustrated in Figure 2.7. Collectively, they demonstrate that distributed load transfer over the surface of cylindrical objects through friction and form closure is achievable across a wide range of scales and actuation principles, providing further inspiration for a conductor removal gripper concept.

2.2.3 Suction-Based Gripper Designs

Suction-based gripping is already applied in practice for lifting and handling pipes and casings exceeding one metre in diameter and approaching two tonnes in weight [45]. However, such implementations are limited to horizontal lifting, where the suction force directly counteracts the object weight. For vertical lifting as required in conductor removal, significantly higher suction forces are needed to generate sufficient friction to resist the conductor weight, and the surface irregularities and subsea challenges discussed in Section 2.1.1 further complicate a reliable seal. Despite these challenges, some suction-based gripper concepts addressing curved or irregular surfaces are worth examining for their potential transferability.

First, Jo et al. (2024) present a suction cup-based gripper designed for harvesting cucumbers, which are vertical cylindrical objects with irregular surfaces. The gripper addresses surface irregularity through optimised suction cup geometry and surface parameters, enabling secure attachment while minimising dropped objects [46]. Regarding underwater suction gripping, Zhang et al. (2023) review adhesion-based soft robotics for underwater applications, noting that octopus-inspired suckers offer adjustable pressure and wide applicability but require improved shear resistance for vertical grasping [47]. Accordingly, Wang et al. (2025) combine hydraulically actuated tentacles with octopus-inspired suction cups into a hydrogel gripper capable of switchable adhesion and manipulation of objects with irregular and curved surfaces in underwater environments [48].

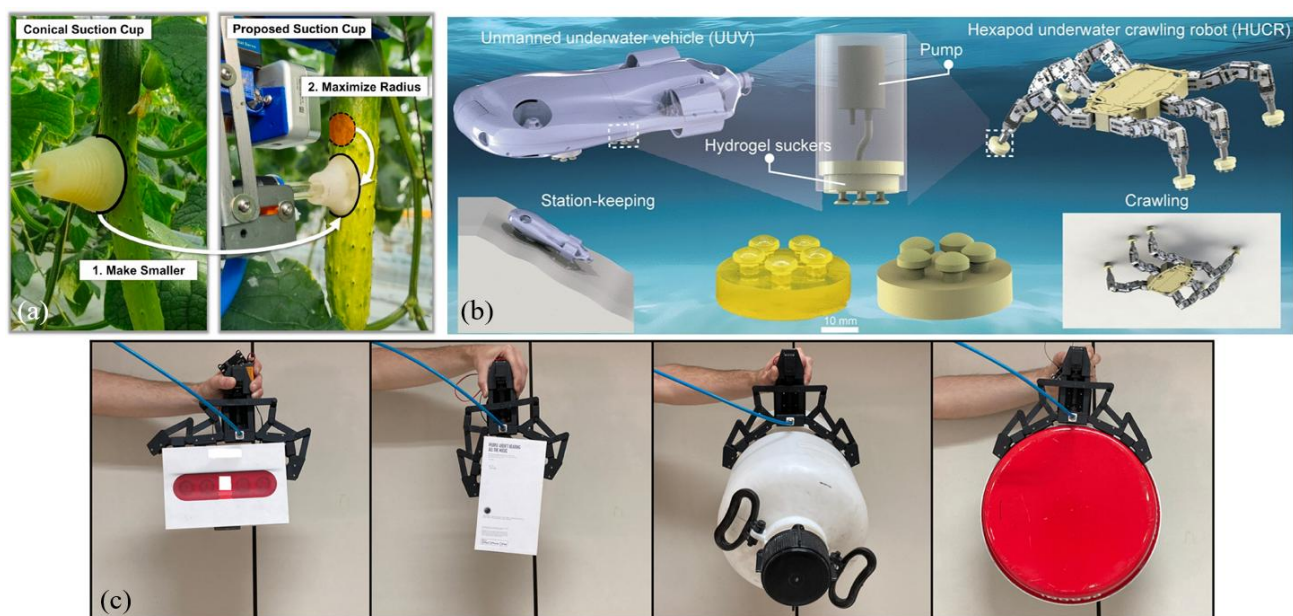


Figure 2.8: Suction-based gripper concepts. (a) Optimised suction cup gripper for cucumber harvesting [46]. (b) Underwater vehicle and crawling robot equipped with an array of octopus-inspired suckers [48]. (c) Underactuated vacuum gripper [49]

Finally, combining suction gripping with other gripping methods presents a promising avenue for increasing overall gripping capability. Maggi et al. (2024) demonstrate this by integrating suction cups into linkage-driven phalanges, achieving a six-fold increase in stability over standard underactuated grippers [49]. The concepts discussed in this section are illustrated in Figure 2.8. Overall, suction-based gripping presents potential relevance to conductor removal, though challenges related to surface irregularity, subsea sealing, and vertical load capacity remain, with hybrid approaches offering the most promising direction.

2.2.4 Jamming Gripper Designs

As discussed in Section 2.1.1, an interesting gripping concept is the jamming of granular material. Such a design was first introduced by Brown et al. (2010), where a bag filled with granular material is pressed against an object, conforming to its shape, and subsequently jammed by applying vacuum, hardening around the object to generate gripping forces through a combination of friction, suction, and geometric interlocking [22].

A significant body of research has since been conducted on jamming grippers, with Cortes and Miranda (2025) providing a comprehensive review of their design, control, and applications, noting their wide versatility across object shapes and sizes and their demonstrated use in underwater environments [50]. Such an interesting application is the one developed by Miettinen et al. (2019), who specifically investigated the scalability of jamming grippers for crane-scale lifting operations, demonstrating successful lifting of objects up to 120 kilograms including a steel cylinder, with analytical modelling suggesting a maximum lifting capacity approaching 270 kilograms for appropriately sized configurations [51].

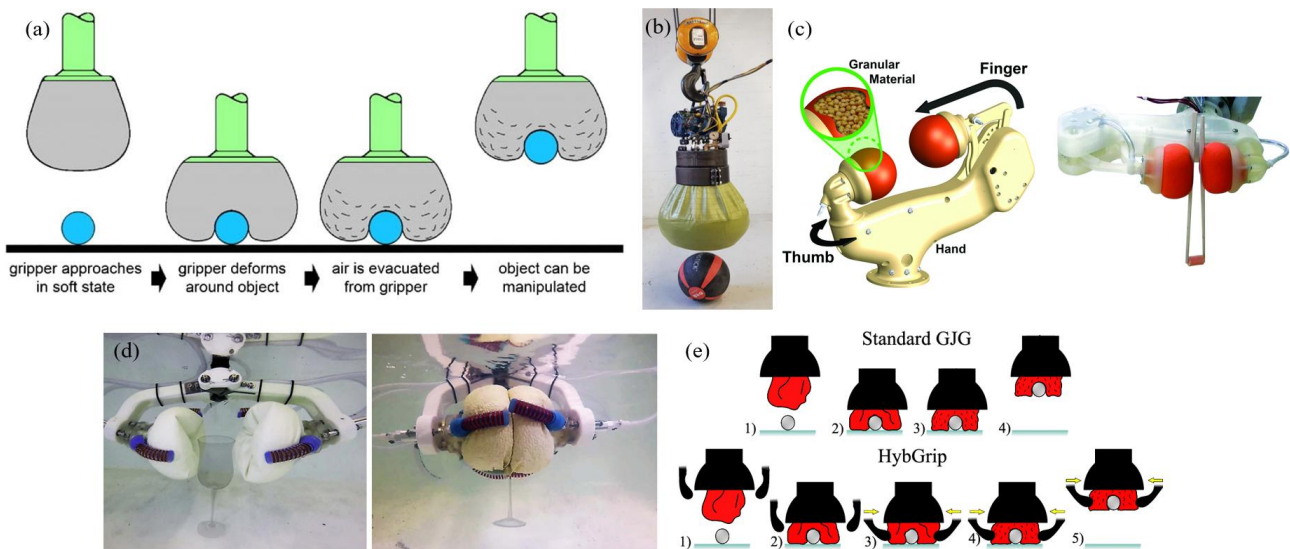


Figure 2.9: Jamming gripper concepts. (a) Schematic of the granular jamming gripping operation [22]. (b) Granular jamming gripper for lifting of heavy objects [51]. (c) JamHand manipulator [52]. (d) Underwater two-pad jamming gripper with neutrally buoyant pads [54]. (e) Grasping stages of a standard granular jamming gripper and the HybGrip with integrated pinching mechanism [57]

However, cylinders are difficult objects to grasp with jamming grippers, as they offer limited capability for the gripper to interlock with surface features and suction may not always be reliably formed. To address this, jamming hands employing two opposing jamming grippers are designed to increase the generated friction force on cylindrical surfaces. Such a design was first introduced by Amend and Lipson (2017) with the JamHand, which combines two granular jamming fingers to achieve dexterous manipulation with minimal actuation, demonstrating a wide range of grasping capabilities while remaining significantly simpler than conventional multi-fingered robot hands [52]. Building on the same principle, Hou et al. (2019) present a two-fingered gripper with jamming fingertips that demonstrates higher grasping stability and up to 40% lower motor power consumption compared to a rigid-fingered equivalent [53]. Finally, Capalbo et al. (2022) present an underwater

two-pad jamming gripper combining neutrally buoyant jamming pads, demonstrating successful grasping of objects with widely varying weight, size, and geometry in an ROV-mounted configuration [54].

Further research by Gómez-Paccapelo et al. (2020) demonstrates that while the selected granular material influences the activation force required to conform the gripper around an object, the maximum holding force is governed primarily by the penetration depth achieved rather than the specific granular material used [55]. Simultaneously, research by Götz et al. (2021) demonstrates that using soft granular particles such as expanded polystyrene beads generates a squeezing effect upon vacuum application that significantly increases friction forces, achieving approximately 19 times higher holding force compared to rigid glass beads of equivalent size [56]. Finally, gripping performance can be further increased by combining the jamming gripper with a pinching mechanism that actively forces the granular material around the object, as demonstrated by the HybGrip proposed by Badilla-Solórzano et al. (2024) [57].

The jamming gripper concepts discussed in this section are illustrated in Figure 2.9. Collectively, the reviewed designs demonstrate that granular jamming grippers present a credible and scalable gripping approach for conductor removal, with the two-pad jamming hand configuration representing the most promising direction. However, as with all other reviewed gripper categories, jamming grippers present their own set of challenges when scaled to the required load magnitudes for conductor removal.

2.3 Current Capabilities

The soft robotic gripper designs reviewed in the preceding sections demonstrate considerable versatility across object shapes, sizes, and applications, and show potential relevance to the conductor removal problem. However, load capacities remain far below the requirements of conductor removal, with the highest payload soft grippers reported in the literature achieving lifting capacities on the order of a few hundred kilograms, representing a significant step forward but still far from the multi-tonne loads associated with conductor removal.

In addition to absolute load capacity, scaling introduces fundamental size-related challenges. When a characteristic length L is scaled, surface area increases with L^2 , while volume and mass increase with L^3 , placing increasing demands on structural integrity, actuation, and energy requirements. Duc and Jovanova (2021) examine these challenges in the context of scaling soft grippers for break bulk manipulation in transport engineering, identifying that while geometric scaling can extend load capacity, actuation methods such as shape memory alloys become increasingly impractical at larger scales due to the decrease in heat transfer with scale [18].

A relevant study by Dietrich and Müller (2022) further demonstrates through simulation and experiment that simple geometric scaling can enable payloads approaching 300 kilograms, suggesting that simulation-driven scaling represents a viable pathway toward higher load applications [58].

A further challenge is the subsea operating environment. Although only a limited number of soft grippers currently operate underwater, the technical challenges associated with subsea deployment are generally considered solvable, as evidenced by several of the designs reviewed in Section 2.2. Hydraulic actuation, neutrally buoyant components, and pressure-tolerant materials have all been demonstrated at relevant depths, and compared to the scaling challenge, adapting a gripper for underwater operation represents a less fundamental barrier.

Finally, the broader offshore decommissioning context introduces a further layer of operational challenges. A soft gripper for conductor removal must navigate a congested subsea environment with obstructions from jacket members, conductor guide frames, and other structural elements, while facing significant uncertainties regarding conductor surface condition, marine growth, internal configuration, and structural integrity.

Compounding these technical challenges is the requirement that any novel lifting concept must meet stringent industry certification and safety standards before offshore deployment, representing a substantial barrier independent of technical performance.

2.4 Summary

This chapter presented a targeted review of soft robotic gripping concepts relevant to the conductor removal problem, covering gripping methods, actuation strategies, materials, and a range of gripper designs. The review demonstrates that while no existing concept is directly applicable at the required scale, several designs show potential for technology transfer and provide inspiration for candidate solutions.

Scaling to conductor removal load magnitudes remains the primary challenge, with subsea deployment and offshore certification adding further operational constraints. Two strategic directions emerge from this analysis. First, concepts that exploit self-amplifying or exponential force generation mechanisms, such as the self-tightening behaviour of braided sleeves, the exponential force growth of twining grippers, and the interlocking mechanisms of jamming grippers, are particularly attractive as they offer a pathway to large holding forces without proportionally large actuator loads. Second, actuation simplicity is a critical requirement, favouring concepts compatible with conventional offshore equipment such as winches, motors, or hydraulic systems. In this regard, the definition of a compliant gripper should be expanded beyond specialised laboratory elastomers to include materials such as steel wire rope and synthetic fibre rope, which are well proven offshore, capable of conforming to irregular surfaces, and provide sufficient strength to lift conductors.

A further key insight is that the challenges specific to conductor removal should be exploited to the benefit of the gripping concept rather than treated as obstacles to be overcome. For example, the aged and degraded condition of conductors, marine growth, corrosion, surface irregularities, and repair patches, directly increase the effective friction at the contact interface and provide geometric surface features for interlocking, both of which enhance rather than hinder soft gripping. These findings, together with the load requirements and operational conditions established in Chapter 1, directly inform the concept development in Chapter 3, informing the requirements and providing inspiration for candidate concepts.

3 Conceptual Design

The design process followed in this thesis adheres to the methodological approach described by Pahl et al. (2007) in alignment with VDI Guidelines 2221 and 2222, which divides product development into four main phases: task clarification, conceptual design, embodiment design, and detail design, as illustrated in Figure 3.1 [59]. The first phase was completed in Chapters 1 and 2 through problem definition, scope clarification, and the literature study.

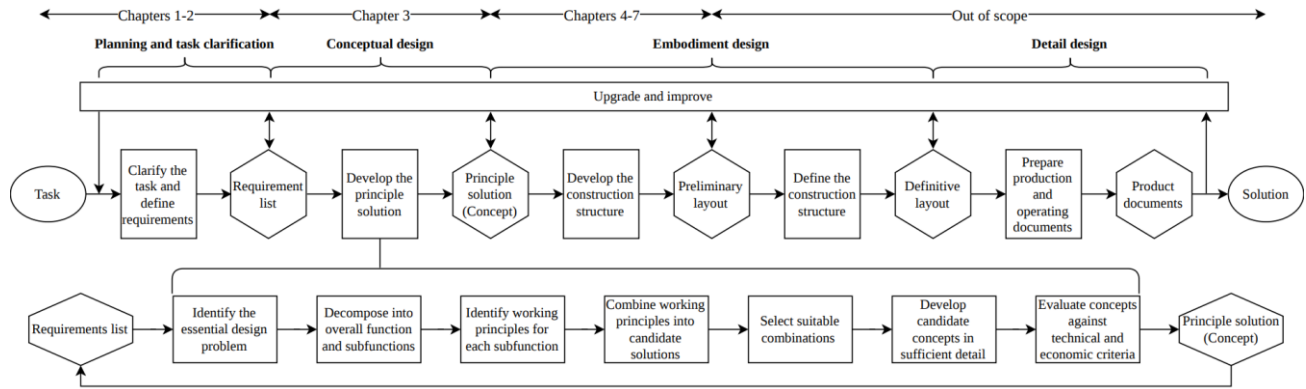


Figure 3.1: Product development phases and conceptual design breakdown, adapted from Pahl et al. (2007) [59]

This chapter addresses the second phase, conceptual design, expanded upon in the second row of Figure 3.1. Starting from a structured requirements list, the key design problem is decomposed into functions and subfunctions, for each of which working principles are identified based on the conducted research. These are subsequently combined into candidate concepts and evaluated against technical and economic criteria to arrive at a principle solution, providing an answer to SRQ1 [59].

Chapters 4 through 7 then address the embodiment design phase, developing the selected concept through analytical modelling and prototype validation. Product development is inherently iterative, and in principle this process could continue indefinitely. This thesis draws a deliberate boundary at the level of a preliminary design, defined as the depth of development necessary and sufficient to answer the three research questions. Further iterative development is proposed as future work.

3.1 Requirements List

This section presents the requirements list for the compliant gripper for conductor removal, developed through the conducted literature review, consultations with Allseas supervisors, and reference to relevant offshore lifting standards. The scenario technique was additionally applied to anticipate potential issues across the system lifecycle and its interactions with the environment and users [59].

The requirements list, presented in full in Appendix B, provides a structured overview of all technical, operational, and safety criteria for the gripper, distinguishing between functional, non-functional, and constraint requirements following the systematic approach of Pahl et al. (2007). Each requirement is classified as a demand (D) or a wish (W), where demands are non-negotiable criteria that any viable solution must satisfy, and wishes represent desirable properties that should be addressed where possible. To support traceability across the design process, each requirement is assigned a unique identifier consisting of the prefix R followed by its category and item number. These identifiers are referenced throughout the remainder of the report wherever a function, working principle, design decision, or evaluation outcome addresses a specific requirement. It should also be noted that the requirements list is a dynamic document that is updated iteratively throughout the design process as understanding of the problem develops. The following subsections discuss the requirements in detail.

3.1.1 Constraints

Constraint requirements define the fundamental boundaries within which the design must operate and are all classified as demands. In addition, a number of critical simplifications are introduced to limit the design scope and enable a feasible first design iteration, representing the most conservative assumptions applicable at this stage.

The gripper shall operate in an offshore subsea environment, with no specific environmental conditions defined for the conceptual phase and all dynamic effects captured through a dynamic amplification factor (DAF) of 1.5 applied to the conductor weight, in accordance with Allseas practice for early-stage designs (R.1.1). To support dimensioning, the North Cormorant platform, a representative fixed jacket platform and a future Allseas decommissioning project, is adopted as the reference case (R.1.2), with all relevant platform and conductor data presented in Appendix C (R.1.3).

The following operational assumptions are applied. The topside is assumed to be already removed and all preparatory work completed prior to conductor removal (R.1.4). Cutting of conductors is outside the scope of this thesis (R.1.5). The conductors and their internal casings are assumed to be pinned prior to lifting using flush pins, allowing passage through the conductor guide frames, and all inner casings are assumed to have sufficient structural capacity to be lifted while only the outer casing is gripped (R.1.6). Conductor guide frames are assumed to be either removed or present with a radial clearance of 25 mm to the conductor outer surface (R.1.7).

Regarding load distribution, the conductor shall not be lifted through a point load in tension (R.1.8), and the gripping load shall be distributed over the outer conductor surface (R.1.9). No upper limit is placed on grip force, as the conductor surface is assumed not to be damaged by the gripping mechanism. Finally, the conductor is assumed as a perfect cylinder with its centre of gravity at its geometric centre, and buoyancy effects are neglected as the gripper must be capable of holding the conductor above sea level (R.1.10).

3.1.2 Functional Requirements

Functional requirements capture the operational behaviour of the gripper and specify the essential capabilities the system must possess. Regarding core capabilities, the gripper shall distribute the grip force over the conductor surface (R.2.1), carry the entire lifting load (R.2.2), maintain grip under all operational conditions (R.2.3), conform to irregular, corroded, or damaged conductor surfaces without precise alignment (R.2.4), and allow for engagement and release of the conductor (R.2.5).

Regarding safety, the solution shall comply with relevant offshore lifting and operational standards such as DNV-ST-N001 and DNV-RP-H201 (R.3.1). It is acknowledged that complete compliance is difficult for an inherently novel concept at this stage of development, and such standards are primarily referenced for safety factors and design guidance. The gripper shall additionally be fail-safe at power loss by maintaining grip (R.3.2), prevent dropped objects (R.3.3), provide redundant gripping (R.3.4), maintain grip during conductor downending to mitigate the risk of bending or buckling (R.3.5), and maintain control of the conductor in foreseeable emergency situations (R.3.6).

Regarding operation, the gripper shall be capable of overcoming obstacles during approach and lifting, such as guide frames and jacket members (R.4.1), and shall accommodate conductor downending onto a vessel, platform, or the seabed (R.4.2). Grip verification and load monitoring are identified as requirements to be addressed in later design phases (R.4.3). Wishes for operational efficiency include limiting diver and ROV involvement (R.4.4), allowing self-deployment (R.4.5), preferring engagement of the outer conductor diameter over an internal grip as the internal configuration is largely unknown (R.4.6), and removing the conductor in a single lift or otherwise minimising the number of sectioning cuts and lifting cycles (R.4.7).

Regarding materials, the gripper shall use marine-grade materials resistant to corrosion (R.5.1) as well as to fatigue and ultraviolet degradation (R.5.2). It would additionally be beneficial to use lightweight materials to avoid adding unnecessary weight to the lifting load (R.5.3) and materials with low environmental impact given the subsea operating context (R.5.4).

The gripper shall furthermore be manufacturable with existing industrial capabilities (R.6.1), preferably using standardised methods of manufacturing and assembly (R.6.2), and shall allow for ease of maintenance and repair (R.6.3). For installation and logistics, the gripper shall include dedicated lifting points for crane or handling equipment (R.7.1), should be easily transportable and storable (R.7.2), should fit within a standard 20 or 40 ft shipping container for offshore transport (R.7.3), and should be assembled and disassembled on deck without specialised equipment (R.7.4). Finally, the gripper should minimise environmental impact during operation (R.8.1) and be reusable across multiple conductor removals (R.8.2).

3.1.3 Quality Requirements

Quality requirements are non-functional requirements that describe how and under what conditions the system functions shall be fulfilled. Additionally, they are relevant for the reliability and evaluation of candidate concepts. In this thesis, quantifying quality and cost in absolute terms is challenging for a novel concept at this stage of development. Both are therefore expressed relative to the existing cut-and-pull method as the reference case. Regarding quality, the proposed gripper should enable the removal of conductors that the cut-and-pull method cannot handle, which is also the core motivation of this thesis (R.9.1). Regarding cost, the solution should be cost-comparable to or better than the cut-and-pull method across the full lifecycle, including materials, manufacturing, assembly, and operation (R.9.2). Given the novelty of the concept, only a qualitative cost estimation is considered feasible at this stage.

3.2 Functional Structure

This section presents the functional structure of the compliant gripper for conductor removal, derived directly from the requirements list outlined in Appendix B. Following the approach of Pahl et al. (2007), the overall function of the system is decomposed into a hierarchy of subfunctions, reducing the complexity of the design problem and enabling the systematic identification of working principles. The functional structure emphasises the dependencies between functions rather than their sequential order.

The overall function is defined as the removal of offshore conductors from fixed jacket platforms, which is decomposed into four main subfunctions: move and position the gripper, grab the conductor, transport the conductor, and release the conductor.

Moving and positioning the gripper encompasses attaching the gripper to a crane or equivalent lifting equipment on a construction vessel (R.7.1), lowering it to the required depth through vertical movement, and manoeuvring it horizontally toward the conductor. Throughout this process the gripper must be capable of navigating around or through obstacles, most notably the conductor guide frames with their limited 25 mm radial clearance (R.1.7) and the surrounding jacket members that constrain access to the conductors (R.4.1).

Grabbing the conductor is broken down into deploying the gripper around the conductor, preferably engaging its outer surface given the largely unknown internal configuration (R.4.6), closing the gripper through actuation, conforming to the conductor surface to distribute the gripping load (R.2.1, R.2.4), generating the grip force through the selected gripping method so that the full lifting load can be carried (R.2.2), and verifying the grip (R.4.3). The surface conformation and grip force generation subfunctions are deliberately separated, as they may be fulfilled by different working principles and are therefore the most critical subfunctions for the morphological chart. Grip verification is identified as a subfunction for completeness but is not considered in the conceptual design phase.

Transporting the conductor requires maintaining the grip under all operational conditions, and in the event of power loss in a fail-safe manner (R.2.3, R.3.2), and lifting the conductor, meaning supporting its full weight throughout the operation (R.2.2). While transporting, the gripper must simultaneously support its own self-weight. Two major obstacles must also be overcome during this phase, namely passing the conductor guide frames as the conductor is extracted upward through the jacket (R.4.1) and passing through the splash zone, the transition region between water and air where environmental loading and dynamics are most severe and are captured at this stage through a dynamic amplification factor of 1.5 (R.1.1).

Finally, releasing the conductor encompasses positioning the gripper and conductor ready for downending (R.4.2), supporting the conductor during the downending operation to prevent bending or buckling (R.3.5), opening the gripper, and retracting it clear of the conductor surface (R.2.5).

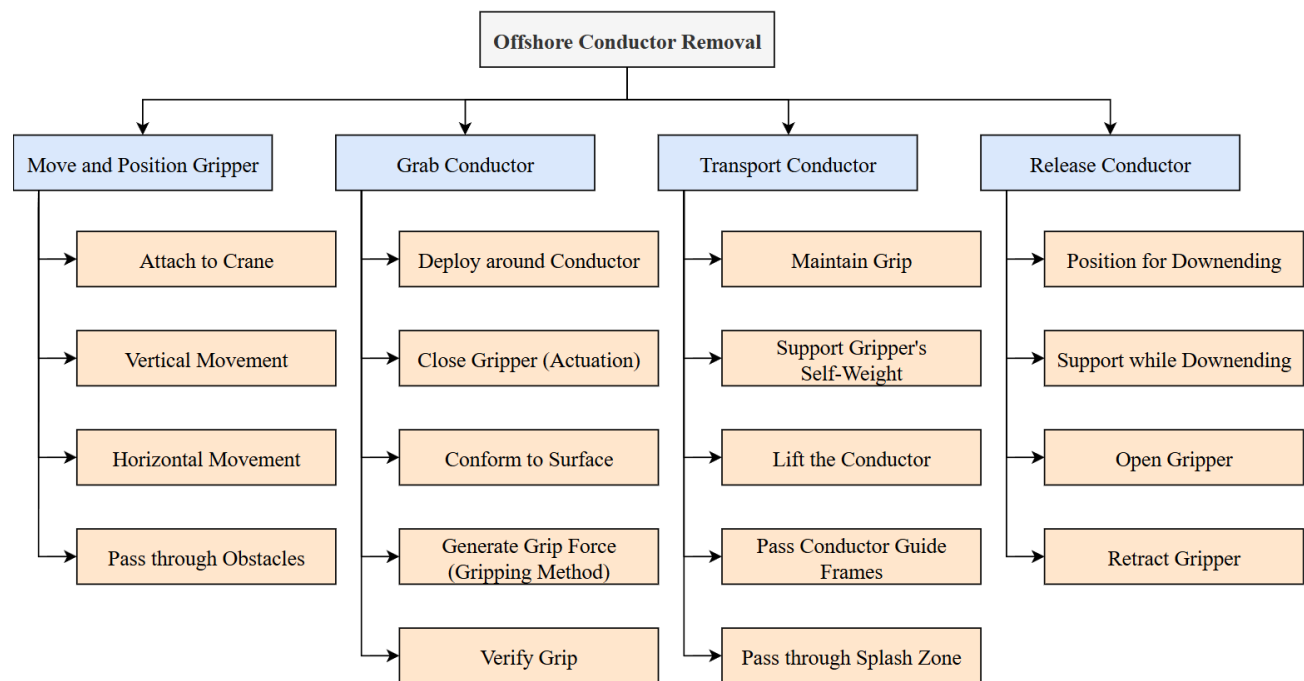


Figure 3.2: Functional structure for the conceptual design of a compliant gripper for offshore conductor removal

The functional structure is illustrated in Figure 3.2. All subfunctions can obviously be further decomposed, but the current level of detail is considered sufficient and appropriate for the conceptual design phase, with further detailing relevant for future iterations.

3.3 Morphological Chart

The solution search is based on generating alternative working principles for each subfunction identified in the functional structure and systematically combining them into complete concept solutions. Following the approach of Pahl et al. (2007), the working principles for each subfunction are derived from the conducted literature study, the background on conductor removal, and additional brainstorming. The optimal solution must satisfy all demands from the requirements list while addressing as many wishes as possible.

To systematically organise the working principles, a morphological chart is constructed, presented in Table 3.1. Each row corresponds to a subfunction and each column to a candidate working principle. Some identified subfunctions, such as grip verification, energy supply, system control, and conductor placement location, are omitted from the chart as they are considered less critical for an early conceptual design phase, though they remain relevant for later design stages.

Table 3.1: Morphological chart for the conceptual design of a compliant gripper for offshore conductor removal

Solution		1	2	3	4	5
Function						
A	Attach to Crane	Direct crane attachment	Attachment through frame			
B	Vertical Movement	Crane hoist	Self climbing on conductor	Winch assisted	Buoyancy assisted	ROV assisted
C	Horizontal Movement	Crane slew	ROV assisted	Not required		
D	Pass Through Obstacles	Through top of jacket	Through jacket side opening	Disassemble & reassemble		
E	Deploy Over Conductor Surface	Radial expansion	Radial contraction	Wrap-around	Roll-on	Mechanical closing
F	Close Gripper (Actuation)	Pneumatic	Hydraulic	Tendon-driven	Conventional drive systems	Hybrid
G	Grip Surface	Grip internal surface	Grip external surface			
H	Conform to Surface	Soft elastomer pads	Suction cups	Sleeve-like structure	Bending/wrapping fingers	Expanding/contracting membrane/bladder
I	Number of gripper units	1	2	3	>3	
J	Generate Grip Force (Gripping Method)	Friction	Form closure/ interlocking	Suction/vacuum	Magnetic adhesion	Hybrid
K	Maintain Grip	Continuous actuation	Self-locking mechanism	Self-tightening		
L	Support Gripper's Self-Weight (Stiffness Support)	Longitudinal locking (layer jamming)	Bulk locking (granular jamming)	Cable/tendon stiffening	External frame	Inflatable stiffening
M	Pass Conductor Guide Frame	Through radial gap	Frames removed prior to conductor removal	Releasing and re-grip	Serpentine crawling	Internal diameter gripper
N	Support During Downending	Gripper	Additional downending frame			
O	Release Conductor	Release actuation	Reverse actuation	Gravity-assisted	Manual release	Release mechanism
P	Retract Gripper	Crane hoist	Manual handling			

The working principles in the chart are briefly discussed below. For crane attachment, the gripper can either attach directly to the crane hook or through an intermediate frame that also supports and controls the gripper. For vertical movement, the crane hoist is the primary option, with alternatives including self-climbing on the conductor, winch assistance, ROV assistance, or buoyancy control. Horizontal movement can be achieved through crane slew or ROV assistance and may not be required if the gripper is deployed along the conductor axis. To reach the conductor, the gripper must navigate through the jacket, either from the top, through a side opening, or through disassembly and reassembly near the conductor.

For deployment over the conductor surface, the gripper can radially expand or contract, wrap around the conductor, roll onto it like a sleeve, or close through a mechanical mechanism. Actuation can be pneumatic, hydraulic, tendon-driven, through conventional mechanical drive systems, or a hybrid of these, as discussed in the literature study. The gripper can engage either the internal or external conductor surface, with the external

surface strongly preferred given the largely unknown internal configuration. The conforming element can take the form of elastomer pads, suction cups, sleeve-like structures, bending or wrapping fingers, or expandable and contracting membranes. The number of gripper units can range from one to more than three.

The gripping force can be generated through friction, form closure, vacuum, magnetic adhesion, or a combination. To maintain grip, the gripper can actuate continuously, employ a self-locking mechanism as a fail-safe, or be self-tightening such that grip increases with applied load. Supporting the gripper's own weight is addressed through longitudinal locking such as layer jamming, bulk locking such as granular jamming, cable or tendon stiffening, an external frame, or inflatable stiffening [60].

Passing the conductor guide frames can be achieved by moving through the 25 mm radial gap, removing the frames prior to conductor removal, releasing and regripping around each frame using multiple gripping points to prevent slippage, serpentine crawling around the frame in a manner analogous to how a snake navigates branches, or by using an internal diameter gripper for which the guide frames are not an obstacle. For downending, the conductor can be supported by the gripper alone or through an additional downending frame on the vessel. Release can be achieved by releasing or reversing actuation, through gravity, manually, or via a dedicated release mechanism. Finally, retraction is achieved through the crane hoist or manual handling.

3.4 Design Concepts

In this step of the conceptual design phase, working principles from the morphological chart are systematically combined into complete functional solutions. The combination process ensures that the selected working principles are logically and physically compatible, identifying which solutions integrate effectively without conflicts while fulfilling the set requirements. Inspiration for the combinations is additionally drawn from the conducted literature study, with the proposed concepts building on gripper designs identified as showing potential once scaled to the conductor removal problem. The resulting combinations of working principles for each concept are presented in Appendix D.

The goal of this section is to develop four candidate concepts to a sufficient level of detail to evaluate, compare, and select the most promising one. For each concept, basic calculations estimate the gripping capabilities based on the reference conductors established in Appendix C, alongside a qualitative assessment of workability, cost, manufacturability, durability, and failure modes. Alternative configurations for each concept are also briefly discussed, with the findings summarised in a table of advantages and disadvantages to support an informed concept selection.

3.4.1 Concept 1: Braided Constriction Sleeve

The first proposed concept is a sleeve-like gripper design based on the self-tightening mechanics of braided tubes such as the Chinese finger trap. The working principle is that the sleeve constricts radially under an applied axial load, such that the radial pressure, and therefore the friction force, increases proportionally with the load. The concept and its basic working principle are illustrated in Figure 3.3.

In the proposed configuration, the gripper consists of a braided sleeve connected at both ends to a top and bottom ring, which are in turn connected to the lifting equipment, such as a spreader bar or directly to the crane. Initially, the sleeve is held in a compressed, expanded-diameter state by tensioning the bottom ring relative to the lifting equipment, allowing the gripper to be lowered over the conductor. A fixing mechanism on the bottom ring then engages the conductor to prevent it from sliding upward as the top ring is lifted, although this fixing mechanism could be omitted if the bottom ring is sufficiently heavy to remain in place under its own weight. The tension on the bottom ring is then released, and the top ring is lifted. As the axial distance between the rings increases, the sleeve diameter reduces, generating radial pressure and consequently friction between the braid and the conductor surface. Once sufficient pretension is achieved, the conductor is cut and lifted.

After transport to the release location, the conductor is first supported, for example by a downending frame on the vessel deck or seabed. The fixing mechanism on the bottom ring is then released, and the bottom ring is pushed upward relative to the top ring to relax the braid and release the grip, either by re-tensioning and raising it through the lifting equipment or by external means such as manual handling.

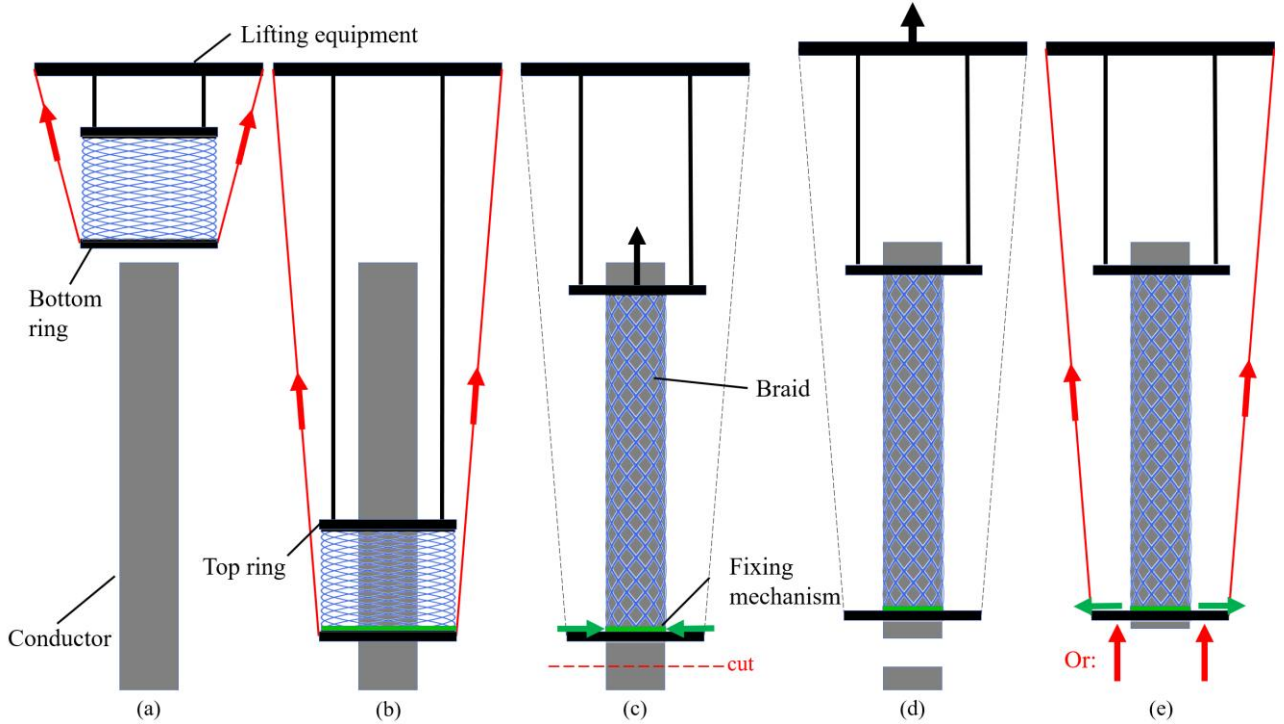


Figure 3.3: Working principle of the braided constriction sleeve concept. (a) Initial compressed state. (b) Sleeve lowered over the conductor. (c) Bottom ring fixed and top ring lifted, constricting the braid. (d) Conductor cut and lifted. (e) Release through re-tensioning or external pushing of the bottom ring upward

The calculations for this concept, as well as for the other three concepts, are documented in Appendix E, with Table 3.2 presenting a summary of the results. For the braided constriction sleeve, the procedure proposed by Lu et al. (2024) is applied, who derive the relationship between the axial force on the braid and the generated radial pressure as a function of the braid geometry [25]. This radial pressure is then compared to the radial pressure required to generate sufficient friction to lift a conductor section. It should be noted that these calculations are not intended as a validated proof that the concept would work in reality, but rather as a simple and fast first-order method to screen the concepts for feasibility at the conceptual design stage. If this concept is selected, the derivation must be redone in detail for the full-scale design and validated, as set out by this thesis.

Table 3.2: Braid parameters and key results for the braided constriction sleeve concept

Quantity	Symbol	Value	Unit
Strap width	w	20	mm
Strap thickness	t	8	mm
Strap sets	n	12	-
Gripping length	H	25	m
Number of full turns	N	14	-
Total strap length	S_{total}	1003	m
Generated radial pressure	P_{gen}	267	kPa
Required radial pressure	P_{req}	106	kPa
Lift safety factor	SF_{lift}	2.53	-
Strap tension	T_{strap}	44.1	kN
Minimum breaking load	MBL	400	kN
Strap breaking safety factor	SF_{break}	9.07	-

As seen above, the generated radial pressure exceeds the pressure required to lift the conductor by friction, resulting in a lift safety factor of approximately 2.5. The straps themselves remain well within their breaking capacity, with a safety factor of approximately 9. These results indicate that the braided constriction sleeve concept is potentially feasible for the conductor removal application at this scale, although the total strap length required, approximately 1000 m, is a practical consideration for the manufacturing and assembly of the gripper.

Additionally, the calculations above do not consider the self-forcing mechanism described by Shang et al. (2019), whereby friction between the braid and the conductor under load increases axial tension in the braid, which in turn increases the radial pressure and therefore the friction further [26]. This effect can substantially increase the gripping capability beyond what is calculated here, making the safety factors above conservative, and should be incorporated into the mechanical model if this concept is selected.

Beyond the gripping calculations, it is important to consider how well this concept fits the requirements established earlier. The braided sleeve distributes the lifting load along the conductor surface, and this distribution can be targeted toward structurally sound sections of the conductor where required. The compliant nature of the sleeve also improves the overall workability of the removal process, as it requires less precision during positioning and can accommodate larger vessel motions during deployment. The concept is furthermore lightweight, as it contains no heavy equipment or machinery, and can be compacted to fit within a 20 or 40 ft shipping container. Material and manufacturing costs are expected to be low, given that the required materials are widely available. Actuation is simple, as the entire gripping and release sequence can be controlled through the crane hoist alone, without requiring complex or underwater actuation systems. Small scale prototypes in the literature validate the underlying mechanics, and comparable designs, such as cable pulling socks, are already applied in the offshore industry. Finally, the bottom ring provides a convenient location for the integration of cutting or pinning equipment.

Table 3.3: Advantages and disadvantages of the braided constriction sleeve concept

Advantages	Disadvantages
Self-tightening grip, low risk of accidental release	Grip relies entirely on friction, coefficient unknown
Distributes load along conductor surface	Abrasion or damage to straps from damaged surface
Lightweight, compact, fits 20/40 ft container	Gripping capacity influenced by many parameters
Tolerates positioning inaccuracy and vessel motion	Braiding process complex at full scale
Simple actuation, no underwater actuation	Guide frames must be removed beforehand
Low material and manufacturing costs	Release procedure complex
Similar small-scale prototypes exist	
Compatible with cutting or pinning equipment	

The main challenge of this concept is that the grip relies entirely on friction, with the friction coefficient between the braid and the conductor surface largely unknown. As discussed in the literature study chapter, the corrosion, marine growth, and welded repair patches typically found on aged conductors could benefit this concept by increasing the effective friction coefficient, although these same features also pose a risk of abrasion and damage to the straps. A further operational challenge is that the conductor guide frames must be removed prior to conductor removal, as the gripper cannot pass around them in its current form. This results in a sequential operation in which a guide frame is cut and removed, the conductor section above it is gripped, cut, and removed, and the process is repeated level by level for all conductors. This sequence introduces the risk of conductors swinging under environmental loading once no longer supported by the frame, which could cause additional damage. However, depending on the gripping capability of the design and the specific platform configuration, removal of an entire conductor in a single lift, with only one or two guide frames removed beforehand, may also be feasible. The braiding process at this scale is expected to be complex but achievable with existing manufacturing capabilities. Finally, the release procedure remains complex and must be investigated separately. The discussed advantages and disadvantages of the concept are summarised in Table 3.3.

A number of alternative configurations of this concept could be of interest for further investigation. A heavy bottom ring could provide the axial tension required to constrict the braid without the need for a separate fixing mechanism. Such a design has not been considered in the soft robotics literature, as additional mass typically

reduces the payload capacity of a gripper. However, in offshore heavy lifting the crane capacity generally exceeds the weight of a single conductor by a large margin, so the addition of a heavy bottom ring would not meaningfully affect the overall lifting capability and is therefore worth investigating further. A further alternative is the use of steel wire rope rather than synthetic fibre rope for the braid. Its inherent bending stiffness would allow the braid to retain a set shape rather than bunching up as a slack fibre braid would, which could simplify the release process. Finally, if material and manufacturing costs allow, the braid could be designed as a sacrificial component, removing the need to resolve the release procedure entirely.

If this concept is selected, this thesis should focus on three things. First, the mechanical model should be extended to include the self-forcing mechanism and to define in more detail what happens at the bottom ring, whether through a fixing mechanism or a heavy ring. Second, a prototype should be built to validate this model and investigate the influence of braid parameters, such as the number of helix turns and gripping length, as well as the forces required to move the gripper between its open and gripping states and the resulting tension in the straps. Finally, some insight into the deployment and release procedure should be given based on the selected braid material for the full-scale design.

3.4.2 Concept 2: Torsional Constriction Sleeve

The second proposed concept shares a similar layout and working principle with the first. However, instead of deploying an already braided sleeve, the sleeve is formed directly on the conductor through the relative rotation of two rings connected by straps. This rotation winds the straps into helices around the conductor, and applying further torsional moment increases the tension in the straps and therefore the radial pressure on the conductor, allowing the load to be lifted through friction. Inspiration for this concept is drawn from similar soft grippers in the literature, such as the iris-mechanism gripper proposed by Lee et al. (2025), which similarly operates through individual bands, and the torsional sleeve grippers proposed by Shan et al. (2023) and He et al. (2025) [30], [31], [33]. The concept and its working principle are depicted in Figure 3.4.

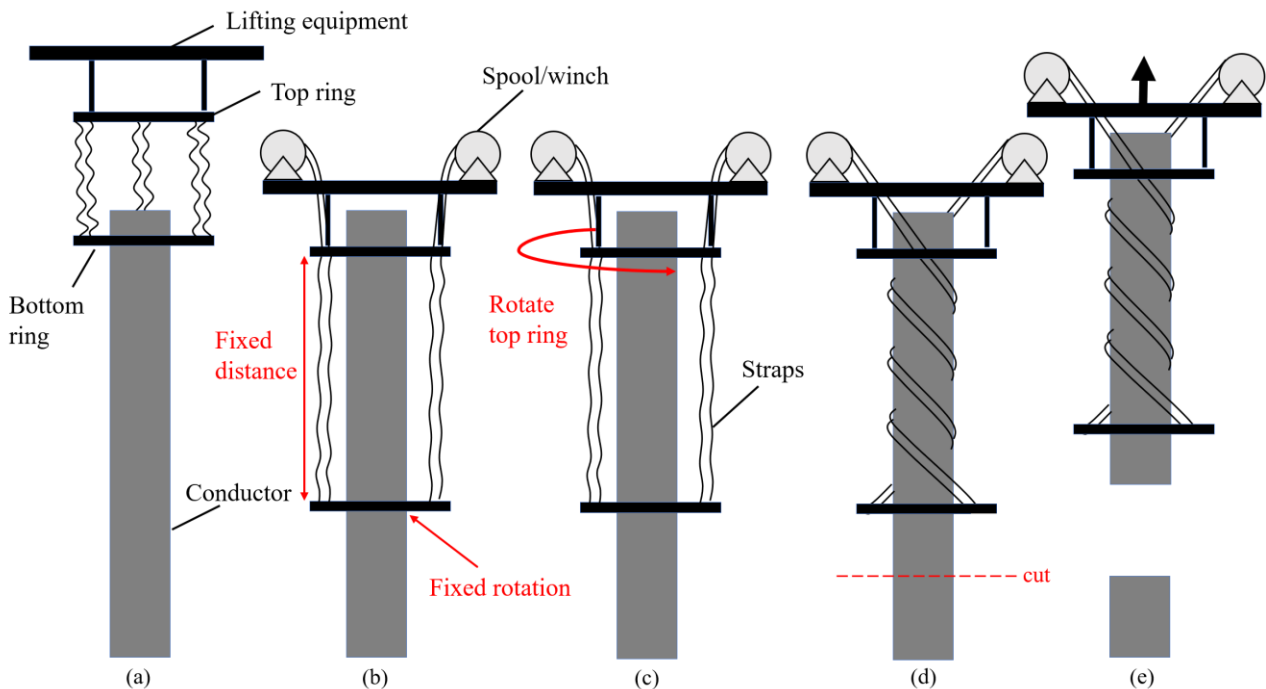


Figure 3.4: Working principle of the torsional constriction sleeve concept. (a) Gripper lowered over the conductor. (b) Bottom ring positioned and fixed to the conductor, with a fixed distance maintained between the rings. (c) Top ring rotated, winding the straps into helices. (d) Conductor cut once sufficient pretension is achieved. (e) Conductor lifted, then released by reversing the actuation

As in the first concept, a top ring is connected to the lifting equipment and to a bottom ring through a set of straps. The straps are routed through winches or spools mounted on the lifting equipment or vessel, which control the deployment of the bottom ring. Both rings are first lowered over the conductor together. The bottom ring is then lowered further to reach its desired position and, once in place, is fixed to the conductor through a fixing mechanism that locks its rotation. From this point onward, the distance between the rings is held fixed by the lifting equipment. The top ring is then rotated, releasing slack from the spools, which winds the straps into helices around the conductor. Sufficient pretension is built up through the applied torsional moment to ensure the conductor can be lifted after cutting. The spools are then locked, leaving the sleeve-like structure in tension from the torsional moment in the straps, and the conductor is cut and lifted. For release, the conductor is placed onto a downending frame or similar support on the vessel, after which the spools are unlocked and the straps are respooled to relax the tension, retract the gripper, and release the conductor.

The full concept calculations are documented in Appendix E, following the capstan methodology of Konyukhov and Shala (2021) for the tension amplification along the helical straps [61]. The parameters and key results are summarised in Table 3.4.

Table 3.4: Parameters and key results for the torsional constriction sleeve concept

Quantity	Symbol	Value	Unit
Sleeve/ring radius	r_s	500	mm
Number of straps	n_s	4	-
Gripping length	H	25	m
Number of full turns	N	6	-
Total strap length	S_{total}	115.2	m
Top ring rotation	ϕ_{ring}	2259	°
Driving torque top ring	M	5	kNm
Initial strap tension	T_0	2.5	kN
Lifting force	F_{lift}	2788	kN
Lift safety factor	SF_{lift}	4.40	-
Bottom ring tension	T_L	700	kN
Minimum breaking load	MBL	400	kN
Strap breaking safety factor	SF_{break}	0.57	-

The concept generates a total lifting force well above the conductor weight, corresponding to a lift safety factor of approximately 4.4 for the chosen parameters. The amplified tension at the bottom ring is high, at around 700 kN, which exceeds the strap breaking load for this unoptimised configuration. This is not a fundamental limitation, as the bottom ring tension can be reduced by increasing the number of straps or lowering the applied torque, but it highlights that the concept requires careful balancing of the actuation parameters. It should be noted, however, that increasing the number of straps complicates the actuation, as each strap must be actuated separately. The results nonetheless indicate that the torsional constriction sleeve is potentially feasible for the conductor removal application at this scale.

The main advantages of this concept are similar to those of the first. The sleeve formed on the conductor distributes the lifting load along the conductor surface, and the absence of a pre-formed braid improves workability and eases deployment over the conductor. Material and manufacturing costs are expected to be low, given that the required materials are widely available. The concept is furthermore lightweight and very compact in its undeployed state. The entire gripping action is controlled by a single parameter, the rotation of the top ring, and locking this rotation eliminates the risk of accidental release. Release is also notably simple, achieved through reversed actuation, and as with the first concept, similar small-scale prototypes exist in literature, validating the underlying mechanics. Finally, this concept is also compatible with the integration of pinning or cutting tools.

On the downside, this concept also relies on friction, with the friction coefficient largely unknown, although as discussed this can be beneficial. There is again a risk of damage to the straps from the conductor's surface. Furthermore, the gripping capability relies on the helix being formed consistently, which may not be reliably guaranteed in practice. The biggest challenge is the complex actuation, which must also be performed underwater. Finally, as with the first concept, the guide frames must be removed prior to conductor removal, although single-lift removal of an entire conductor may also be feasible under the right parameters. The advantages and disadvantages of the concept are summarised in Table 3.5.

Table 3.5: Advantages and disadvantages of the torsional constriction sleeve concept

Advantages	Disadvantages
Distributes load along conductor surface	Grip relies entirely on friction, coefficient unknown
Lightweight and compact in undeployed state	Abrasion or damage to straps from damaged surface
Sleeve formed on conductor, easier deployment	Consistent helix formation cannot be guaranteed
Single-parameter actuation through top ring rotation	Actuation complex, must be performed underwater
Simple release through reversed actuation	Guide frames must be removed beforehand
Low material and manufacturing costs	
Similar small-scale prototypes exist	
Compatible with cutting or pinning equipment	

Regarding alternative configurations, inspiration can be drawn from the soft gripper concepts presented in Section 2.2.1. Instead of the top ring, the bottom ring could be the one to rotate, or both rings could rotate in opposite directions. The fixation of the bottom ring could be achieved on the conductor itself, on the lifting equipment, or on the securing pin of the conductor. The bottom ring could also be moved to the top side, next to the top ring, as in some of the gripper concepts, allowing the actuation to take place on a single side. The straps could also be replaced by a continuous sleeve, with the underlying mechanics remaining unchanged. Finally, the investigation by Lee et al. (2025), in which allowing the straps to deform elastically increases the gripping capacity, is of interest and could be of benefit at full scale [30].

If this concept is selected, this thesis should focus on three things. First, the actuation must be defined in detail, as this is not trivial at full scale, along with how the rotation of the bottom ring is fixed. Second, a theoretical model with refined mechanics must be developed, as the preliminary calculations rely on simplifications that may not hold at full scale. The prototype should validate the model and provide insight into the influence of the helix and actuation parameters on the gripping capability. Finally, the tension at the strap ends and ways to reduce it should be investigated, as the preliminary calculations show that the tension amplification leads to a critical loading condition at these points.

3.4.3 Concept 3: Particle Jamming Gripper

The third concept originated from the idea of a suction cup array gripper. However, it became clear that suction alone is insufficient to lift the conductor vertically, as the contact area required under full atmospheric pressure would far exceed what is practically achievable at this scale. For this reason, the investigation was extended to jamming grippers, and in particular to concepts that combine jamming pads with rigid pinching jaws, such as those proposed by Amend and Lipson (2017), Hou et al. (2019), and Capalbo et al. (2022), which have demonstrated the capability to grasp vertical cylindrical objects [52]-[54].

The proposed concept therefore consists of a support frame carrying several jaws distributed along its length, with each finger of each jaw fitted with a jamming pad. The jaws close onto the conductor so that the pads conform to its surface, after which the pad membranes are evacuated to lock the granular material around the

conductor profile, generating grip through friction and, where possible, mechanical interlocking. The concept and its working principle are illustrated in Figure 3.5.

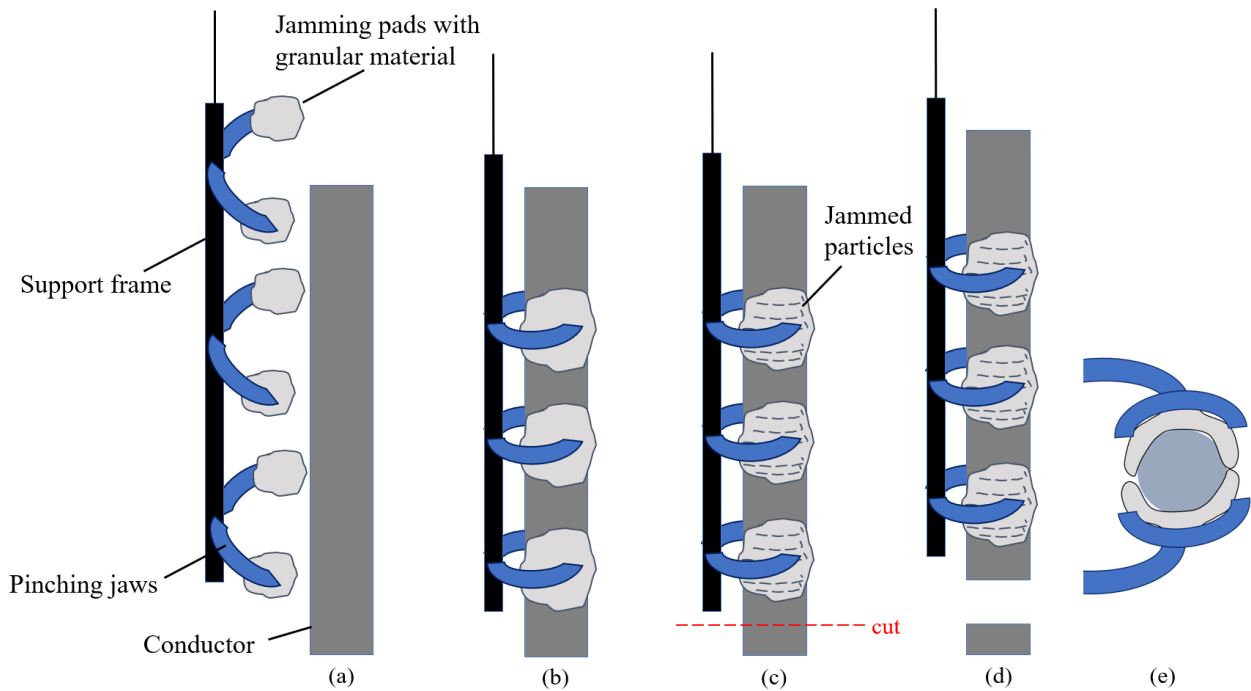


Figure 3.5: Working principle of the particle jamming gripper concept. (a) Support frame lowered alongside the conductor with jaws open. (b) Jaws closed, pressing the jamming pads against the conductor surface. (c) Pads evacuated, jamming the granular material around the conductor profile and the conductor is cut. (d) Conductor lifted with the pads locked. (e) Top view showing the two-jaw configuration with the pads conforming around the conductor cross-section

In the proposed configuration, the support frame is lowered alongside the conductor with the jaws open. The jaws then close, pressing the jamming pads against the conductor surface, including any local irregularities. The air is removed from the pads, jamming the granular material and fixing the conformed shape. With the pads locked around the conductor, the conductor is cut and lifted. For release, the pads are returned to atmospheric pressure, unjamming the granular material so that the jaws can open and retract. In contrast to the previous two concepts, this concept is potentially capable of passing the conductor guide frames without removing them beforehand. This is achieved by adding a redundant jaw pair that allows the gripper to release and regrip in stages around the guide frame, as illustrated in Figure 3.6.

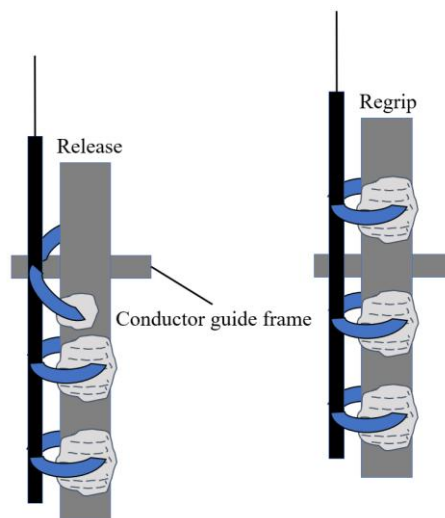


Figure 3.6: Guide frame passing through sequential release and regrip of the jaw pairs

The full concept calculations are documented in Appendix E, following the jamming holding force model of Brown et al. (2010) and its extension to a two-jaw configuration by Amend and Lipson (2017) [22], [52]. The parameters and key results are summarised in Table 3.6.

Table 3.6: Parameters and key results for the particle jamming gripper concept

Quantity	Symbol	Value	Unit
Pad/bag radius	L_b	1	m
Number of jaw pairs	N_j	6	-
Pinch force per jaw pair	F_p	500	kN
Friction from jamming squeeze per pair	$F_{F,N}$	3	kN
Friction from jaw pinch per pair	$F_{F,p}$	150	kN
Holding force per jaw pair	F_F	153	kN
Total holding force	$F_{H,tot}$	918	kN
Lift safety factor	SF_{lift}	1.45	-
Holding force during frame passing	$F'_{H,tot}$	765	kN
Lift safety factor during frame passing	SF'_{lift}	1.21	-

The results show that, for a perfect cylinder, the holding force is dominated almost entirely by the mechanical pinch of the jaws, with the jamming squeeze contributing less than 2% of the total. The jamming pads in this idealised case therefore serve mainly to conform to the surface and distribute the pinch load rather than to generate grip through the granular material itself. This is an important finding, as it means the concept as analysed reduces to a distributed friction clamp, with the granular jamming providing compliance and surface adaptation rather than gripping force.

This observation points directly to the main opportunity of the concept. As discussed in the literature study, the corrosion, repair patches, and marine growth typically found on aged conductors are advantageous for jamming grippers, as they allow the granular material to interlock with the surface and generate substantial additional holding force beyond what is calculated here. Where natural interlocking features are absent, windows could deliberately be cut into the conductor to provide them. Furthermore, the safety factor remains above unity during guide frame passing, where the conductor is held by one fewer jaw pair, confirming the feasibility of the release and regrip strategy. The number of jaw pairs or the pinch force can be increased to raise this margin further.

The main advantage of this concept is that it distributes the lifting load along the conductor surface. In contrast to the previous two concepts, it also presents a more practical way of passing the conductor guide frames without prior removal, through the release and regrip strategy, and can more readily accommodate single-lift removal of the entire conductor by adjusting the number of jaw pairs and the pinch force. The possibility of interlocking further increases the gripping capability significantly, and in scenarios with low friction the pinch force can be increased to compensate. Release is also simple, achieved by unjamming and opening the jaws, and the concept is compatible with the integration of other conductor removal equipment.

On the downside, the concept again relies on friction with an unknown coefficient, but more critically relies on the behaviour of granular material at a scale far beyond what has been demonstrated. The weight of the granular material itself must be supported by the pad membrane, which is not trivial. The membrane also poses a risk of accidental failure, which would result in immediate loss of grip. The biggest challenge is the complex actuation, which requires simultaneous hydraulic control of multiple jaws and vacuum control of the jamming pads, all performed underwater. This actuation is also inherently slow, which could make the process inefficient, particularly during the release and regrip sequence at guide frames. Finally, the material and manufacturing costs of this concept are expected to be high, as the required equipment and design approach have not previously been considered at this scale. The advantages and disadvantages of the concept are summarised in Table 3.7.

Table 3.7: Advantages and disadvantages of the particle jamming gripper concept

Advantages	Disadvantages
Distributes load along conductor surface	Grip relies entirely on friction, coefficient unknown
Can pass guide frames without prior removal	Granular material behaviour unknown at scale
Can accommodate single-lift conductor removal	Membrane must support the granular material
Interlocking significantly increases grip	Risk of accidental release if membrane fails
Pinch force compensates for low friction scenarios	Very complex actuation, also performed underwater
Simple release through unjamming and jaw opening	Slow actuation, may reduce operational efficiency
Compatible with cutting or pinning equipment	High material and manufacturing costs

Regarding alternative configurations, the size, shape, and material properties of the granular fill and the membrane represent a large design space that can be explored to identify the most suitable configuration for large-scale subsea applications. The implementation of mechanical fingers or a cage to better guide the jamming pads around the conductor during deployment is also of interest, improving load distribution and gripping capability, similar to the concept proposed by Badilla-Solórzano et al. (2024) [57]. Finally, deliberately cutting windows into the conductor to guarantee interlocking remains an attractive option.

If this concept is selected, the identified unknowns and challenges must be addressed. The thesis should therefore focus on developing a holding force model based on cylindrical rather than spherical geometry, extending the approach of Brown et al. (2010) [22]. The behaviour of the granular material at scales well beyond those previously demonstrated must also be investigated, along with the ability of the membrane to support the weight of the fill. The prototype should investigate the difference in holding force between a smooth cylinder and a cylinder with deliberately added surface features, validating the cylindrical model and providing insight into the influence of parameters such as contact angle on the gripping capability. Although the actuation is complex, the thesis should focus primarily on the fundamental gripping mechanics, with actuation left as a problem for future design iterations.

3.4.4 Concept 4: Internal Inflatable Gripper

The final concept is an internal gripper, in contrast to the external gripping approach of the previous three. This concept is somewhat at odds with several of the established requirements. Since the outer casing is the most structurally compromised, carries connections with no reliable tensile capacity, and is the primary load-bearing element to be gripped, a gripper that engages the innermost casing does not directly address these challenges. For an internal gripper to grip the outer casing from the inside, the inner casings must first be removed, which introduces additional complications. The internal configuration may be unknown, grout or cement may be present between casings preventing their individual removal, and tools capable of internally gripping and lifting inner casings of these sizes are limited at the current state of the art. For these reasons, an internal gripper is likely less widely applicable to conductor removal than the previous concepts. This is treated as a disadvantage in the subsequent evaluation rather than a reason to exclude the concept entirely, since if the concept proves superior across the other evaluation criteria, it should still be considered for applicable cases.

The concept is based on an inflatable bladder that is lowered into the conductor casing, inflated to expand against the inner surface, and then used to grip and lift the conductor through friction. Analogous solutions exist in the offshore industry in the form of inflatable packers, which are used to seal casings from escaping hydrocarbons during decommissioning and to stabilise internal tools such as cutters and inspection equipment. No such solution has been applied for removals. The concept and its working principle are illustrated in Figure 3.7.

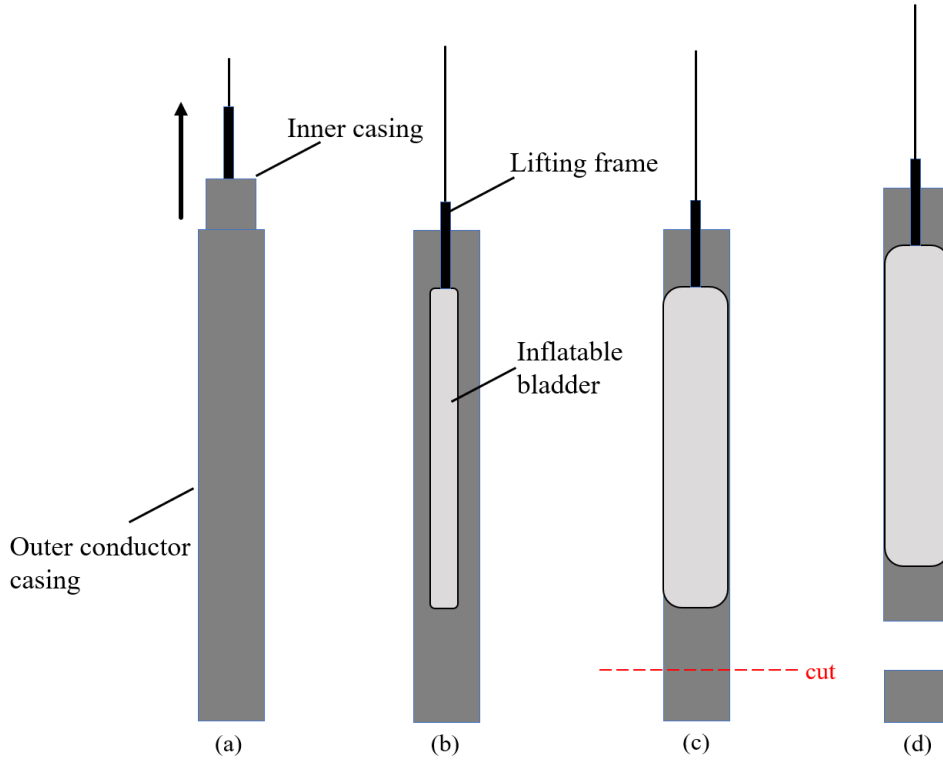


Figure 3.7: Working principle of the internal inflatable gripper concept. (a) Inner casings removed through conventional means or a smaller version of the proposed gripper. (b) Lifting frame lowered into the outer casing and bladder inflated against the inner surface. (c) Grip generated through friction and conductor cut. (d) Conductor section lifted

In the proposed configuration, the inner casings are first removed to expose the inner surface of the outer conductor casing. A lifting frame carrying the deflated bladder is lowered into the conductor. The bladder is inflated, expanding radially against the inner surface and generating a distributed normal force. Sufficient friction is then available to grip and lift the conductor after cutting. Release is achieved by deflating the bladder, after which the lifting frame can be retracted. The full concept calculations are documented in Appendix E, following the same approach applied to the previous concepts. The parameters and key results are summarised in Table 3.8.

Table 3.8: Parameters and key results for the internal inflatable gripper concept

Quantity	Symbol	Value	Unit
Outer casing weight per section	$W_{c,o}$	212	kN
Bladder contact length	H	10	m
Deployment radial clearance	c	50	mm
Contact area	$A_{contact}$	22.7	m ²
Required lift pressure	P_{lift}	31.2	kPa
Hydrostatic pressure at 110 m	P_h	1106	kPa
Required internal bladder pressure	P_b	1137	kPa
Bladder end pressure difference	ΔP_{end}	1036	kPa
Volume to be pumped	ΔV	1059	L

The results reveal a fundamental characteristic of this concept. The pressure required to generate sufficient friction to lift the conductor is modest, but the total required bladder pressure is dominated by the need to overcome the hydrostatic pressure at the gripping depth, reaching over ten times the gripping requirement. The bladder ends must furthermore withstand a significant pressure difference when lifted above the waterline, as the external hydrostatic pressure no longer acts to counterbalance the internal pressure. The volume to be pumped for hydraulic inflation is substantial but manageable with offshore pumping equipment. These results

indicate that the concept is feasible from a gripping standpoint, but that the bladder pressure and membrane integrity represent critical design challenges.

The main advantages of this concept are its simplicity and workability. The bladder conforms to the inner bore of the conductor, distributing the lifting load over the contact surface and tolerating variations in bore diameter or minor surface irregularities without precise alignment. The concept is lightweight, as no heavy equipment or complex external mechanisms are required, and the actuation is relatively simple compared to the previous concepts, consisting only of hydraulic inflation and deflation of the bladder. The concept can furthermore pass the conductor guide frames without removing them, as the bladder is entirely internal, and could potentially remove the entire outer casing in a single lift. Release is simple and achieved by deflating the bladder. Similar technology exists in the offshore industry in the form of inflatable packers, and the concept is scalable to different casing diameters. Costs are also expected to be relatively low.

A notable challenge, however, is the load transfer from the bladder membrane to the lifting frame. The friction force is generated on the flexible bladder surface, and transferring this load to the rigid lifting frame in a way that does not introduce point loads is not straightforward. One possible approach, illustrated in Figure 3.8, is to combine the bladder with a rigid internal tool that grabs the flush pin already present in the conductor, lifting the conductor in compression through the pin while the bladder provides distributed supplementary load across problematic sections such as Talon connectors. However, this hybrid approach partially defeats the thesis research objective of moving away from point loads and requires further investigation.

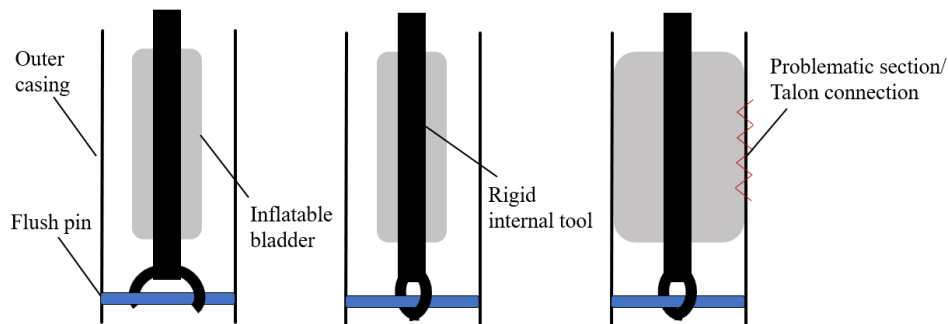


Figure 3.8: Combined rigid tool and inflatable bladder, providing distributed supplementary support across problematic sections

An additional operational challenge is the water column trapped above the bladder inside the casing, which must be vented during inflation to allow the bladder to expand. If the bladder forms an airtight seal with the casing, this venting becomes problematic. The concept also relies on friction with an unknown coefficient, and the bladder membrane is vulnerable to damage from internal casing features, which would result in immediate loss of grip. The actuation is slow, and the bladder end stress under the required internal pressure represents a significant structural challenge, as discussed above. The advantages and disadvantages of the concept are summarised in Table 3.9.

Table 3.9: Advantages and disadvantages of the internal inflatable gripper concept

Advantages	Disadvantages
Distributes load along conductor bore	Inner casings must be removed beforehand
Can pass guide frames without prior removal	Grip relies on friction, coefficient unknown
Can accommodate single-lift conductor removal	Accidental release if pressure or membrane fails
Tolerates bore variation and surface irregularities	Complex load transfer from bladder to lifting frame
Lightweight and relatively simple actuation	Water column above bladder must be vented
Simple release through deflation	Slow actuation
Relatively low cost	High bladder end pressure

Regarding alternative configurations, multiple smaller bladders distributed along the casing length could reduce the required pressure per bladder and improve load distribution. A spherical rather than cylindrical bladder shape could also be considered, as it offers more favourable stress distribution in the membrane under internal pressure. The integration of the bladder concept with a rigid internal tool, as described above, could also be an interesting direction to investigate further.

If this concept is selected, the thesis should focus on three aspects. First, the stresses on the bladder membrane and the load transfer from the bladder to the lifting frame must be investigated, as these represent the most critical unknowns. Second, a suitable bladder shape must be determined, balancing contact area, stress distribution, and ease of deployment. Finally, a prototype should be built to validate the gripping capability and investigate the influence of internal pressure and contact length on the holding force.

3.5 Concept Evaluation and Selection

To determine the most suitable concept for further development, a structured evaluation is conducted using a weighted point rating method following the approach of Pahl et al. (2007) [59] and the rating scale defined in VDI 2225-3. This method enables a systematic comparison of the four concepts against a set of weighted criteria, allowing the most balanced solution to be identified while accounting for the established requirements.

3.5.1 Evaluation Criteria and Rating Scale

The evaluation criteria were established through the requirements list, the literature study, and discussions with the Allseas supervisors. Only requirements that distinguish the concepts were included as scored criteria, while common requirements and shared design assumptions were treated as baseline conditions. Each criterion was weighted reflecting its relative importance. The criteria and their weights are presented in Table 3.10.

Table 3.10: Evaluation criteria and assigned weights

Nr.	Evaluation Criteria	Description	Weight	Related Requirements
1	Gripping capacity confidence	Confidence that sufficient gripping force is generated, given parameter uncertainty	0.15	R.2.2, R.2.3, R.1.9
2	Degree of load distribution	Size and uniformity of the contact area over the conductor surface	0.05	R.2.1, R.1.8, R.1.9
3	Vulnerability to failure modes	Number and severity of inherent failure modes	0.10	R.3.2, R.3.3, R.3.4, R.3.6
4	Complexity of operation	Number of steps, preparatory work, and degree of subsea actuation	0.12	R.4.4, R.4.5, R.1.4
5	Workability improvement	Tolerance to positioning inaccuracy and vessel motion	0.04	R.2.4, R.2.3
6	Guide frame handling	Ability to pass guide frames without prior removal	0.06	R.4.1, R.1.7
7	Full conductor removal potential	Ability to remove the conductor in a single lift	0.05	R.4.7
8	Complexity of release	Simplicity and reversibility of the release procedure	0.06	R.2.5
9	Integration with cutting equipment	Potential to integrate with conductor cutting tools	0.03	R.1.5, R.4.7
10	Scalability and prototype transferability	Confidence that prototype mechanics transfer to full scale	0.07	
11	Compactness and transportability	Transport size and ease of handling and storage	0.04	R.7.2, R.7.3
12	Manufacturability	Ease of manufacture using standard materials and processes	0.08	R.6.1, R.6.2, R.5.4
13	Cost	Estimated capital, material, and equipment cost	0.15	R.9.2
Sum			1	

The two highest weights are assigned to gripping capacity confidence and cost, each at 0.15. Gripping capacity confidence is weighted highest as it directly addresses the fundamental research question of whether a concept can generate sufficient distributed gripping force for the full-scale lifting load. Cost is weighted equally, as the concept must ultimately be economically viable to justify development and to remain competitive with the existing cut-and-pull method. Complexity of operation follows at 0.12, since the number of operational steps, the degree of subsea actuation, and the required support equipment translate directly into vessel time and operational cost. Vulnerability to failure modes is weighted at 0.10, as a concept with fewer inherent failure modes requires less engineering effort to satisfy safety requirements. Manufacturability and scalability follow at 0.08 and 0.07 respectively. The scalability criterion differs from the others in that it does not originate from a single requirement but from the research risk, central to SRQ2 and SRQ3, that the governing gripping mechanics may not transfer from the prototype to the full scale. It is included because this transferability is decisive for the value of the selected concept as a research outcome. The remaining criteria, covering load distribution, guide frame handling, full conductor removal potential, release complexity, workability, compactness, and integration with cutting equipment, are assigned lower weights between 0.03 and 0.06, as they are either secondary at this stage, difficult to differentiate between concepts, or can be compensated for by stronger performance in the more heavily weighted criteria.

Each concept is scored against each criterion using the five-point rating scale defined in VDI 2225-3, presented in Table 3.11, where a higher score reflects a closer approximation to the ideal solution [62].

Table 3.11: Rating scale according to VDI 2225-3

Approximation to the ideal solution	
Very good (ideal)	4 Points
Good	3 Points
Sufficient	2 Points
Barely Acceptable	1 Point
Unsatisfactory	0 Points

3.5.2 Evaluation

The four concepts were scored against each criterion based on the preliminary calculations, the qualitative advantages and disadvantages discussed for each concept, and the engineering judgement of the author. As the concepts are at an early conceptual stage, the scoring reflects relative rather than absolute performance and is intended to discriminate between concepts rather than to provide a definitive measure of feasibility. The weighted evaluation is presented in Table 3.12.

The reasoning behind the most influential scores is briefly summarised. On gripping capacity confidence, Concepts 1 and 2 score highest (R.2.2, R.2.3), as their self-forcing friction mechanism means the lifting load itself generates the gripping force, making them inherently less sensitive to the uncertain friction coefficient. Concepts 3 and 4 score lower, as their holding force depends respectively on interlocking, which may not always be available, and on a bladder mechanism subject to the unfavourable volume-to-area scaling at larger sizes. On complexity of operation, Concept 1 scores highest as it requires no underwater actuation and limits diver and ROV involvement (R.4.4), while Concept 4 scores lowest as it requires prior removal of the inner casings (R.1.6), which is not always possible and renders the concept inapplicable on certain platforms. On vulnerability to failure modes, Concept 1 again scores well due to its passive self-tightening grip (R.3.2), whereas Concept 3 scores lowest given its dependence on both sustained vacuum and continuous hydraulic pressure. On cost, Concepts 1 and 4 score highest due to their reliance on simple, widely available components (R.9.2), while Concept 3 scores lowest as it requires hydraulic jaws, vacuum equipment, membrane fabrication, and granular fill. On guide frame handling, Concepts 1 and 2 score zero, as neither can pass the guide frames without prior

removal (R.4.1), whereas Concepts 3 and 4 perform better through sequential regripping and internal gripping respectively. This open demand is not resolved at the conceptual stage and is therefore carried into the full-scale design rather than treated as a reason to exclude Concept 1, given its strong performance across the more heavily weighted criteria.

Table 3.12: Weighted concept evaluation

Nr.	Evaluation Criteria	Weight	C1	W. C1	C2	W. C2	C3	W. C3	C4	W. C4
1	Gripping capacity confidence	0.15	4	0.60	4	0.60	2	0.30	2	0.30
2	Degree of load distribution	0.05	3	0.15	3	0.15	3	0.15	4	0.20
3	Vulnerability to failure modes	0.10	3	0.30	2	0.20	1	0.10	2	0.20
4	Complexity of operation	0.12	2	0.24	2	0.24	1	0.12	0	0.00
5	Workability improvement	0.04	4	0.16	3	0.12	3	0.12	4	0.16
6	Guide frame handling	0.06	0	0.00	0	0.00	2	0.12	4	0.24
7	Full conductor removal potential	0.05	0	0.00	0	0.00	1	0.05	2	0.10
8	Complexity of release	0.06	1	0.06	2	0.12	3	0.18	3	0.18
9	Integration with cutting equipment	0.03	3	0.09	3	0.09	3	0.09	1	0.03
10	Scalability and prototype transferability	0.07	3	0.21	3	0.21	1	0.07	2	0.14
11	Compactness and transportability	0.04	2	0.08	3	0.12	1	0.04	2	0.08
12	Manufacturability	0.08	3	0.24	2	0.16	1	0.08	2	0.16
13	Cost	0.15	3	0.45	2	0.30	1	0.15	3	0.45
Sum		1	31	2.58	29	2.31	23	1.57	31	2.24

3.5.3 Selection and Outlook

Based on the weighted evaluation, the braided constriction sleeve (Concept 1) achieves the highest overall score of 2.58. It combines the strongest gripping capacity confidence, owing to its self-forcing mechanism, with low cost, good manufacturability, and the simplest operation of the four concepts, requiring no underwater actuation. Its weaker performance on guide frame handling and full conductor removal potential is outweighed by its strength across the more heavily weighted criteria. The torsional constriction sleeve (Concept 2) follows at 2.31, sharing the same favourable gripping mechanics but penalised by its more complex actuation. The internal inflatable gripper (Concept 4) finishes third at 2.24, performing well on load distribution and guide frame handling but constrained by its narrow applicability and unresolved load transfer. The particle jamming gripper (Concept 3) scores lowest at 1.57, primarily due to its high cost and numerous failure modes.

The braided constriction sleeve is therefore selected for further development, answering SRQ1 by identifying a compliant gripper concept suitable for offshore conductor removal based on distributed load.

The selection itself already satisfies a number of requirements. The braided sleeve distributes the lifting load over the outer conductor surface rather than through a tensile point load (R.1.8, R.1.9, R.2.1), conforms to an irregular surface on which conductor condition challenges such as corrosion raise rather than lower the effective friction (R.2.4), and engages the external diameter as preferred (R.4.6). Its self-tightening action holds the conductor passively, without continuous power and with inherent redundancy across its multiple fibre sets (R.3.2, R.3.4), while the absence of heavy machinery keeps the gripper lightweight, compact, and self-supporting for transport (R.5.3, R.7.1, R.7.2, R.7.3). The screening calculations further indicate that the concept can in principle remove conductors beyond the capability of the cut-and-pull method, the core motivation of the thesis (R.9.1), though this remains to be confirmed at full scale. These properties are the basis on which the concept was chosen and require no further demonstration.

The remaining in-scope requirements are not yet evidenced and define the work of the following chapters. The defining demand, that the gripper carry the entire lifting load (R.2.2) while maintaining grip under that load (R.2.3), is first established analytically by the mechanical model in Chapter 4, which incorporates the self-

forcing mechanism and relates the braid geometry to the gripping capacity. As this model rests on an assumed friction coefficient, the demand is only substantiated once the model is validated, which is the purpose of the prototype and experimental validation in Chapters 5 and 6. The early-phase full-scale design in Chapter 7 then addresses the requirements that operate at system level, namely the passing of the guide frames (R.4.1), the selection of marine-grade materials (R.5.1, R.5.2), and the verification of the lifting safety factors and dropped-object prevention against the demonstrated capacity (R.3.1, R.3.3). The release and downending of the conductor (R.2.5, R.4.2, R.3.5) and the manufacture and deck handling of the full-scale braid (R.6.1, R.6.2, R.7.4) are outlined at the same stage but only briefly, with their detailed development recommended for future work. The remaining requirements lie beyond the scope of the preliminary design and are proposed as future work in Chapter 8, alongside the cost comparison with current removal methods (R.9.2). The resulting mapping is given in Table 3.13.

Table 3.13: Requirements and where they are resolved

Requirement	Pr.	Resolved in
R.1.8, R.1.9, R.2.1: <i>Distributed grip, no tensile point load</i>	D	Concept selection
R.2.4: <i>Conform to irregular or corroded surface</i>	D	Concept selection
R.4.6: <i>Engage external diameter</i>	W	Concept selection
R.3.2, R.3.4: <i>Passive, fail-safe, redundant grip</i>	D	Concept selection
R.5.3, R.7.1, R.7.2, R.7.3: <i>Lightweight, compact, self-supporting</i>	D/W	Concept selection
R.2.2: <i>Carry the entire lifting load</i>	D	Model, prototype & validation
R.2.3: <i>Maintain grip under load</i>	D	Model, prototype & validation
R.4.1: <i>Pass the conductor guide frames</i>	D	Full-scale design
R.5.1, R.5.2: <i>Materials</i>	D	Full-scale design
R.3.1, R.3.3: <i>Lifting safety factors, dropped-object prevention</i>	D	Full-scale design
R.2.5, R.4.2, R.3.5: <i>Release and downending</i>	D	Outlined, detailed in future work
R.6.1, R.6.2, R.7.4: <i>Full-scale manufacture and deck handling</i>	D/W	Outlined, detailed in future work
R.9.1: <i>Remove conductors beyond cut-and-pull capability</i>	W	Outlined, detailed in future work
R.3.6, R.4.3: <i>Emergency control, grip and load monitoring</i>	D	Future work
R.4.5, R.4.7: <i>Self-deployment, single-lift removal</i>	W	Future work
R.5.4, R.6.3, R.8.1, R.8.2: <i>Environmental impact, maintenance</i>	W	Future work
R.9.2: <i>Cost-comparable to current methods</i>	W	Future work

The selection therefore satisfies the defining properties of the concept, but the central demand, that the gripper carry the entire lifting load under an uncertain friction coefficient, remains to be shown. The following chapter develops the analytical model of the braided sleeve, including the self-forcing mechanism and the possible heavy bottom ring, to establish this capacity. The prototype and experimental validation then provide the evidence, and the early-phase full-scale design addresses the remaining in-scope requirements, answering SRQ2 and SRQ3.

4 Modelling the Selected Concept

This chapter sets out to further expand upon the selected concept, the braided sleeve gripper, by describing its geometry and mechanics in greater detail. The underlying principle of this concept is a braid that experiences radial contraction under axial elongation. This radial contraction generates radial pressure between the braid and the gripped object, which is in turn converted into friction. Through this friction, the load can be carried. The load itself also acts on the braid through the friction, tending to elongate it further. This increases the axial tension, which increases the radial pressure and therefore the grip, resulting in a self-amplifying loop.

Similar gripper concepts have been developed previously. Shang et al. (2019) base their design on spring theory and background on medical stents [26], while Lu et al. (2024) examine the geometry and mechanics of the Chinese finger trap and conclude that a soft gripper can be developed from their derivations [25]. This thesis builds on this research and expands it towards a controllable braid through the two-ring configuration described in the previous chapter. A further addition is the decision to use a heavy bottom ring, with the hypothesis that the greater axial elongation and force this generates will further increase gripping capability. The reasoning is that offshore lifting offers far greater capacity than the small-scale applications typical of soft grippers, and the additional mass therefore presents no practical constraint.

As no previous work has, to the author's knowledge, added such a heavy ring to a braided gripper, the following sections derive the governing mechanics. The geometry of the braid is defined first, followed by the transition between states and the force required to move between them. The theoretical gripping model is then derived, with the chapter concluding on the estimation of maximum fibre stress and identification of the critical points.

In terms of the requirements list, this chapter addresses the central open demand identified in Section 3.5.3, namely that the gripper carry the entire lifting load (R.2.2) while maintaining grip under that load (R.2.3). As the model rests on idealised friction conditions and on the hypothesis that self-forcing develops for a heavy conductor, these requirements are established analytically rather than proven, with validation left to the following chapters.

4.1 Geometry of the Braided Sleeve

This section is divided into two subsections. The first presents a general derivation of the braid geometry and the main relationships between its parameters. The second presents the geometry relevant to the offshore removal case, introducing the radial clearance for deployment, the radius of the gripped object, and a parameter set chosen for ease of control in a full-scale design.

4.1.1 Braid Geometry

The geometry of the braid follows the derivation and notation presented for the Chinese finger trap by Lu et al. (2024) [25], deviating where necessary to remain relevant to the braided gripper concept. A number of assumptions apply throughout. The braid is assumed uniform, with constant diameter along its length and no necking pattern. Only the section lying on the cylindrical object is considered, with the segments leaving the cylinder towards the rings assumed small relative to the full braid length and therefore neglected. The braid is further assumed to form perfect helices on the cylinder when gripping, with constant pitch and braiding angle along both its length and its fibres.

The proposed braid consists of n sets of helical fibres, meaning that in total there are $2n$ individual fibres forming two opposing helices. Both helices are assumed to share the same starting point, being connected at their ends, and since only complete full turns on the cylinder are considered, the opposite end is also assumed to connect the two helices. One helix runs clockwise and the other anticlockwise, forming a simple weaving

pattern in which the fibres pass over and then under one another. The helices are traced on a cylinder of radius r , have a circular cross section of thickness d , and each fibre has a total length S . The braid forms N turns on the cylinder, the final length of the braid is set to H , and each fibre lies at a braiding angle α that remains constant along the helix, as shown in Figure 4.1 (a), where a helix with $N = 2$ turns is depicted

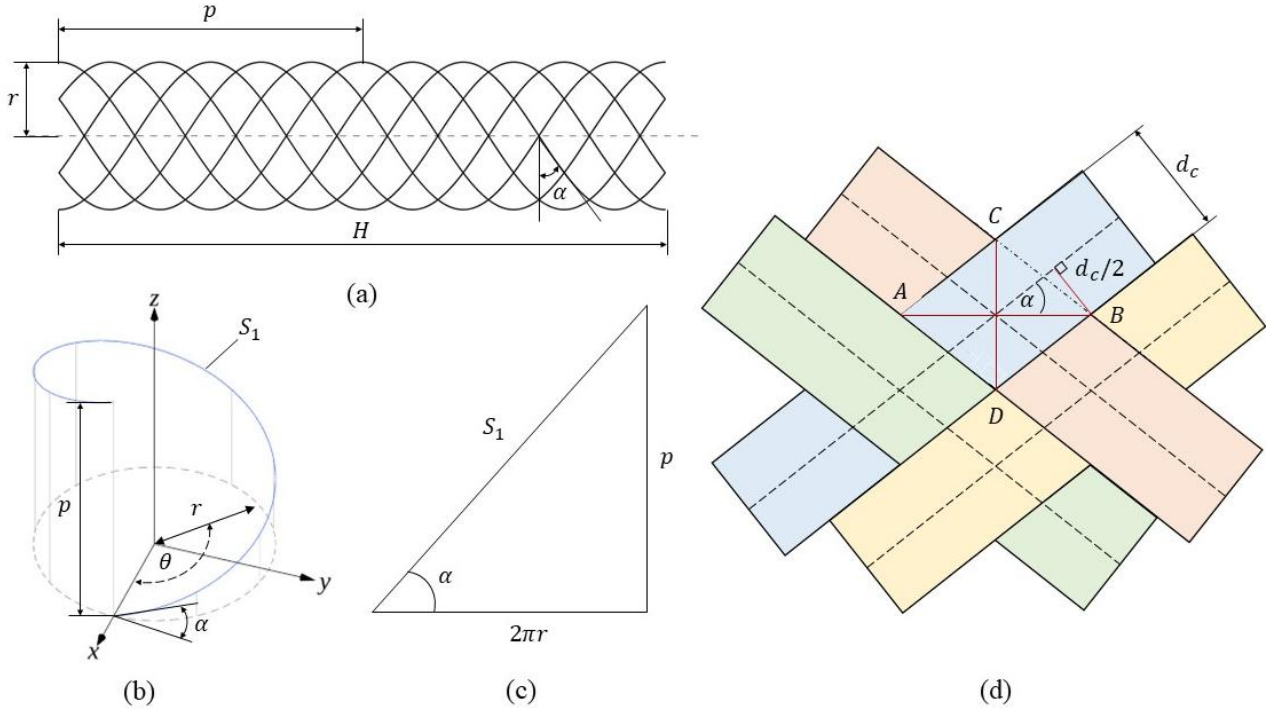


Figure 4.1: Geometry of the braided sleeve. (a) Side view of the braid with $N=2$ turns and $n=6$ fibre sets. (b) A single helical fibre over one full turn, described by the radius r , pitch p and braiding angle α , with the resulting one-turn fibre length S_1 . (c) Development of one helical turn into a right-angled triangle, relating p , r , α , and S_1 . (d) A segment of the fully compact braid at the critical thickness d_c , showing the arch AB and segment CD that form the rhombic overlaps at the weaving crossings

Figure 4.1 (b) shows an individual fibre over one turn, with $N = 1$, where the height formed by a single turn is the pitch p . The helix can be described in a Cartesian coordinate system using the radius r , pitch p , and angular position θ relative to the x -axis through the following equations:

$$x = r \cos \theta \quad (4.1)$$

$$y = r \sin \theta \quad (4.2)$$

$$z = \frac{p\theta}{2\pi} \quad (4.3)$$

The single turn helix can be unwrapped to form the triangle shown in Figure 4.1 (c). The hypotenuse of the triangle is the fibre length over one turn S_1 . The pitch p , radius r and braiding angle α are coupled through:

$$p = 2\pi r \tan \alpha \quad (4.4)$$

Applying the Pythagorean theorem, the pitch p , radius r and one turn fibre length S_1 are coupled into:

$$p^2 + (2\pi r)^2 = S_1^2 \quad (4.5)$$

The equations above describe the helix and one full turn of the fibre. The braid is, however, able to change configuration between a short and wide state and a long and slim state. To define the states that a braid of given fibre set number n and one turn fibre length S_1 can take, the extreme state must first be derived. This is the state in which the braid can no longer become shorter and wider, or taller and slimmer, as the fibres come into contact with one another, as shown in Figure 4.1 (d). This extreme case is the configuration in which the braid reaches the critical thickness d_c and becomes fully compact. As seen in the figure of the fully compact state, the braid forms n repetitions of the arch $\widehat{AB}$ along its perimeter and n repetitions of the segment $\overline{CD}$ over one pitch p :

$$n\widehat{AB} = 2\pi r \quad (4.6)$$

$$n\overline{CD} = p \quad (4.7)$$

As the fibres lie at the angle α and have a thickness of d , the lengths of the two segments are:

$$\widehat{AB} = \frac{d}{\sin \alpha} \quad (4.8)$$

$$\overline{CD} = \frac{d}{\cos \alpha} \quad (4.9)$$

Using the triangle from Figure 4.1 (c), the following holds:

$$\sin(\alpha) = \frac{p}{S_1} \quad (4.10)$$

Combining Equations (4.6), (4.8) and (4.10) gives:

$$\frac{nS_1 d}{p} = 2\pi r \quad (4.11)$$

Substituting Equation (4.11) into (4.5) gives:

$$p^4 - p^2 S_1^2 + d^2 S_1^2 n^2 = 0 \quad (4.12)$$

This is a quadratic equation in p^2 with the solutions:

$$p^2 = \frac{S_1^2 \pm \sqrt{S_1^4 - 4d^2 S_1^2 n^2}}{2} \quad (4.13)$$

Where Δ is the discriminant:

$$\Delta = S_1^4 - 4d^2 S_1^2 n^2 \quad (4.14)$$

The critical thickness d_c , at which the braid is fully compact, is determined by setting $\Delta = 0$ and is:

$$d_c = \frac{S_1}{2n} \quad (4.15)$$

Therefore, for a braid to exist under a given set of n and S_1 , the thickness of the individual fibres must satisfy the condition:

$$d < d_c \quad (4.16)$$

Finally, a general braid that does not have the critical thickness can move between two compact configurations, found by solving Equation (4.13):

$$p_{max/min} = \sqrt{\frac{S_1^2 \pm \sqrt{\Delta}}{2}} \quad (4.17)$$

From the maximum and minimum feasible pitch, the braiding angle α and the radius r can be found using the triangle of Figure 4.1 (c). These are the extreme state configurations of the braid: the shortest and widest state ($p_{min}, r_{max}, \alpha_{min}$) and the longest and slimmest state ($p_{max}, r_{min}, \alpha_{max}$).

The next step is to determine the contact area between the braid and the conductor. A set of assumptions is required to approximate this area. First, the full width of the circular cross section of diameter d is assumed to come into contact with the cylinder, as though the fibre had a rectangular cross section. Second, the contact is assumed never to be lost at the weaving points where one fibre crosses over another. Under these assumptions, the contact of $2n$ fibres over one full turn approximates a rectangular area:

$$A_{S1} = 2ndS_1 \quad (4.18)$$

The overlaps occurring at the weaving crossings must then be subtracted from this area. From Figure 4.1 (d), each overlap forms a rhombus whose diagonals are the arch $\widehat{AB}$ and the segment $\overline{CD}$. The area of one overlap is therefore:

$$A_{rhombus} = \frac{\widehat{ABCD}}{2} \quad (4.19)$$

Substituting Equations (4.8) and (4.9):

$$A_{rhombus} = \frac{d^2}{\sin(2\alpha)} \quad (4.20)$$

The total number of overlaps over one full turn is $2n^2$, where the fibre ends at the rings are assumed to form half overlaps, which is reasonable as the helices start and terminate there. The net contact area over one full turn is therefore:

$$A_1 = A_{S1} - 2n^2 A_{rhombus} \quad (4.21)$$

Substituting Equations (4.18) and (4.20):

$$A_1 = 2ndS_1 - \frac{2n^2 d^2}{\sin(2\alpha)} \quad (4.22)$$

Finally, the total contact area A and the full length of one fibre S over N turns, together with the total fibre length used in the braid S_{total} , are:

$$A = NA_1 \quad (4.23)$$

$$S = NS_1 \quad (4.24)$$

$$S_{total} = 2nS \quad (4.25)$$

It should be noted that Equation (4.23) deviates from the formulation of Lu et al. (2024), who are not consistent in the way they estimate the total contact area when deriving the radial pressure [25]. Equation (4.23) is followed throughout this work.

4.1.2 Extended Geometry for the Conductor Removal Case

This section expands the geometry with regard to the conductor removal application. The relevant parameters are first defined. The principal parameter is the conductor radius r_c . It can then be argued that controlling the manufacture of such a large braid through its braiding angle α , pitch p , and fibre segment length S_1 is impractical, particularly because the pitch and braiding angle are unlikely to remain constant in practice, as previously assumed. From this point onward, therefore, the control parameters for the braid are taken as the gripping length H and the number of full turns N . Additionally, knowing and controlling the final gripping length is more practical at large scale than controlling the braiding angle, which can be less intuitive. From these two parameters and the conductor radius r_c , all other parameters can be estimated using the equations in the previous subsection. The two further parameters under direct control are the fibre thickness d and the number of fibre sets n . The pitch then follows from the two control parameters as:

$$p = \frac{H}{N} \quad (4.26)$$

The state of the braid in contact with the cylinder is defined as the closed state. In this state the braid cannot contract further, as the gripped cylinder prevents further deformation. The radius and length of the braid in the closed state are:

$$r_{closed} = r_c \quad (4.27)$$

$$H_{closed} = H \quad (4.28)$$

For offshore deployment, a clearance to the conductor is required to ease deployment, defined as the radial clearance c . The state during deployment is defined as the open state, with radius:

$$r_{open} = r_c + c \quad (4.29)$$

Using Equations (4.26) to (4.29), together with the relationships derived in the previous subsection, the total length of the braid in its open state, H_{open} , follows from the input parameters:

$$H_{open} = N \sqrt{\left(\frac{H}{N}\right)^2 + (2\pi r_c)^2 - (2\pi(r_c + c))^2} \quad (4.30)$$

This expression gives the compact size of the gripper during deployment and is useful for handling and transport considerations in future full-scale design iterations, including the requirement that the gripper fit within a standard shipping container as per the set requirements (R.7.3). Equally useful is the height change between the open and closed states:

$$\Delta H = H_{closed} - H_{open} \quad (4.31)$$

These parameters are depicted in Figure 4.2, which shows the open and closed states with the notation used. Note that the figure is presented horizontally for clarity, whereas the actual configuration is vertical.

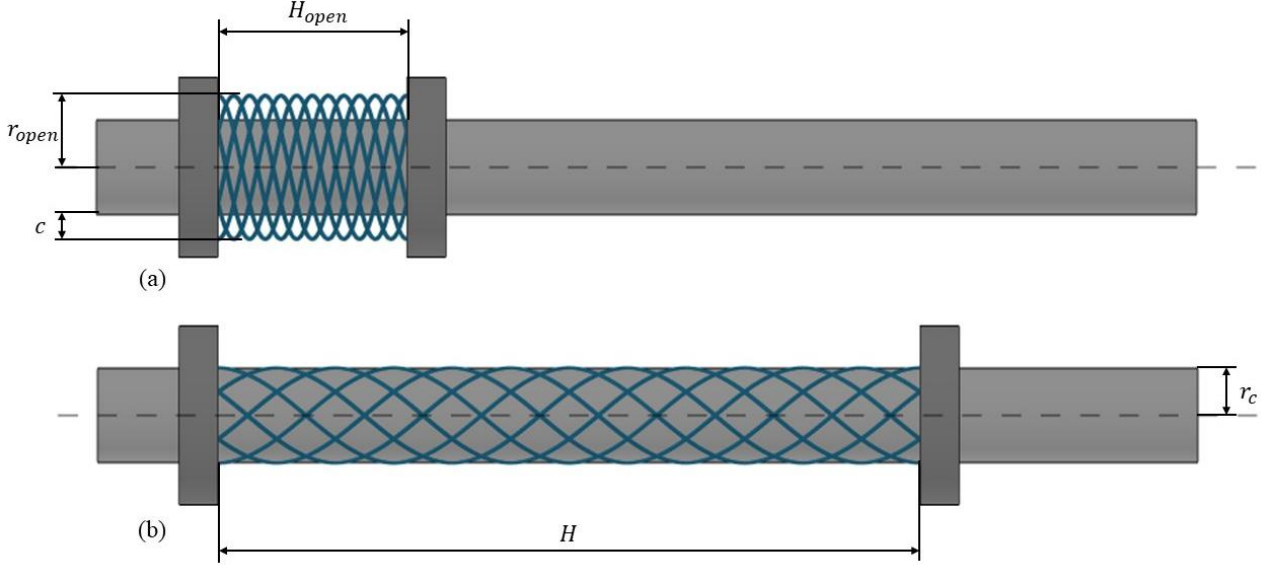


Figure 4.2: The braided sleeve in its open and closed states. (a) Open state during deployment, with the braid holding a radial clearance c over the conductor of radius r_c , and a braid length of H_{open} . (b) Closed state during gripping, with the braid lying on the conductor surface and a gripping length H

As defined in the previous section, two extreme states exist, the shortest and widest state ($p_{min}, r_{max}, \alpha_{min}$) and the longest and slimmest state ($p_{max}, r_{min}, \alpha_{max}$), beyond which the braid can no longer change its geometry. In practice, these states may not be reachable owing to material limitations. It is therefore useful to introduce a metric expressing how close the open and closed states are to these theoretical extremes, since proximity to either extreme may render the braid infeasible. The open-state ratio OSR is first defined between the shortest and widest state and the open state:

$$OSR = \frac{r_{max}}{r_{open}} \quad (4.32)$$

The closed-state ratio CSR is then defined between the closed state and the longest and slimmest state:

$$CSR = \frac{r_{closed}}{r_{min}} \quad (4.33)$$

For the braid to be feasible, both ratios must be greater than unity ($OSR \geq 1; CSR \geq 1$). To improve the likelihood of feasibility during design iterations, the ratios should be as far from unity as possible.

Finally, a valuable metric for the full-scale application is the covered area ratio C_A defined as the ratio of the total contact area A , as derived in the previous section, to the surface area of the gripped cylinder section of length H :

$$C_A = \frac{A}{2\pi r_c H} \quad (4.34)$$

This ratio quantifies how completely the grip is distributed over the conductor surface, directly reflecting the requirement that the gripping load be distributed rather than applied as a point load (R.1.9, R.2.1).

The formulas derived above support the iterative design process, as they provide ratios and metrics that allow different geometries to be generated and evaluated quickly. They can also be applied to optimisation efforts, such as maximising the deployment clearance or maximising the contact area available over the gripped section, while simultaneously satisfying the braid feasibility conditions. These derivations are less central to the gripping model, which is nonetheless developed from the input parameters defined here.

4.2 State Transitions

Building on the previous section, which described the geometry of the braid, this section presents the axial force required to move between the different geometric states, with the following section continuing on to the gripping force that results from this geometry change. To move between states, the braided sleeve must be loaded axially, either elongated towards a taller and slimmer state or compacted towards a shorter and wider one.

To derive the axial force for these transitions, the braid is treated as a combination of open-coiled helical springs with ends fixed against rotation. This methodology was first proposed by Jedwab and Clerc (1993) [63], who applied the spring theory of Wahl (1963) [64] to investigate the geometrical and mechanical properties of medical stents. A considerable body of subsequent stent research followed the same methodology, with a notable example being the study by De Beule et al. (2008) on the optimisation of braided wire stents [65]. Since the proposed soft gripper can be approximated as a scaled-up medical stent, particularly if the fibres are made of steel wire, the same methodology and assumptions can be applied here. It should be noted that Shang et al. (2019) already applied this approach to a braided gripper [26].

Let the parameter set (r_0, α_0) define the fabrication state of the gripper, where the braid is produced and woven around a mandrel of radius r_0 with the fibres following the braiding angle α_0 throughout. This state is assumed to be the equilibrium state, and any departure from it, towards either a slimmer or a wider configuration, deforms the braid and requires an axial force. Let the parameter set (r', α') define an arbitrary state to which the braid is deformed. Following the open-coiled helical spring theory of Jedwab and Clerc (1993), the axial force required for this deformation is:

$$T = 4n \left[\frac{GI_p}{K_3} \left(\frac{2 \sin \alpha'}{K_3} - K_1 \right) - \frac{EI \tan \alpha'}{K_3} \left(\frac{2 \cos \alpha'}{K_3} - K_2 \right) \right] \quad (4.35)$$

Where n is the number of fibre sets, with $2n$ fibres in total, E is the Young's modulus, G is the shear modulus, and I and I_p are the second moment of area and the polar second moment of area respectively, defined for circular cross sections of thickness d as:

$$I = \frac{\pi d^4}{64} \quad (4.36)$$

$$I_p = \frac{\pi d^4}{32} \quad (4.37)$$

The constants K_1 , K_2 and K_3 are evaluated at the initial state:

$$K_1 = \frac{\sin(2\alpha_0)}{2r_0} \quad K_2 = \frac{\cos^2 \alpha_0}{r_0} \quad K_3 = \frac{2r_0}{\cos \alpha_0} \quad (4.38)$$

It should be noted that, as identified and corrected by De Beule (2008), the original expression of Jedwab and Clerc (1993) contains a discrepancy in the form of an additional cosine term in the GI_p contribution [66]. Equation (4.35) is the corrected expression of De Beule et al. (2008), which is followed throughout this work.

The sign of the force in Equation (4.35) depends on the direction of the transition relative to the fabrication state. Moving towards a taller and slimmer state requires a positive axial force, whose positive-valued magnitude is denoted T_e , while moving towards a shorter and wider state requires a negative axial force, whose positive-valued magnitude is denoted T_c . Both transitions, starting from the common fabrication state, are illustrated in Figure 4.3.

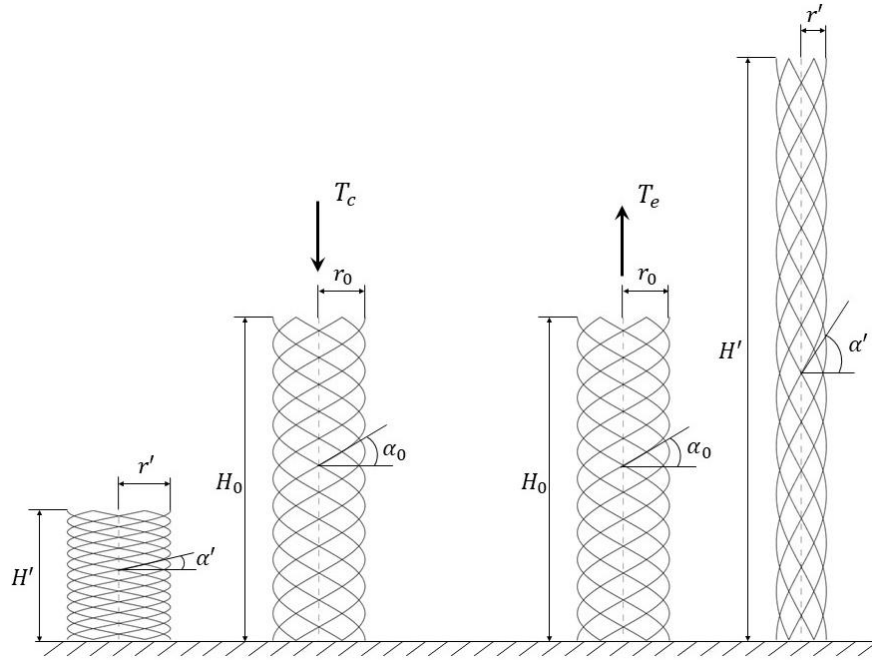


Figure 4.3: Transitions of the braided sleeve under compacting T_c and elongating T_e forces, starting from the fabrication state (r_0, α_0)

The fabrication state therefore influences the operation of the gripper. If the fabrication state is taller and slimmer than the closed state $(r_{closed}, \alpha_{closed})$, the braid must be compacted to reach the closed state. In this configuration the restoring force assists gripping, as the braid tends to return to its slimmer fabrication state and thereby increases the radial compression on the conductor. If instead the fabrication state is shorter and wider than the open state $(r_{open}, \alpha_{open})$, deployment and release are eased, as the braid always tends to return to a more compact configuration. In this case, however, the restoring force provides no positive contribution to gripping.

It is therefore useful to add a heavy bottom ring, which introduces an axial elongating force that counteracts the restoring compaction and thereby increases the grip. This design choice follows directly from the requirements established in the conceptual design. The ring provides the elongating force passively, without any active or continuous actuation, which supplies the baseline axial force from which the lifting capacity is generated (R.2.2) and reinforces the fail-safe grip established at concept selection, since the grip is sustained by weight rather than by a powered system (R.3.2).

The operational procedure and the forces acting on the two rings are illustrated in Figure 4.4. For this illustration, the fabrication state is assumed to coincide exactly with the open and deployment state, so that $r_0 = r_{open}$. This presents a reasonable assumption, as the braid can be manufactured around a mandrel at the open-state geometry and then shipped and deployed directly in the removal operation, with the braid only ever constricting inward from this state to the closed state at the conductor radius, requiring no pre-shaping or intermediate compaction step. The weights of the top ring, the fibres, and additional equipment are assumed negligible compared to the weight of the bottom ring. Friction between the braid and the cylinder is also not considered in this section, as it governs the actual gripping, which is the subject of the following section.

Figure 4.4 (a) shows the gripper hanging from the lifting equipment, with its full weight supported through the connection of the bottom ring to the lifting equipment. The top ring is also connected separately, but at this point the top ropes are slack, allowing the sleeve to remain compact. In Figure 4.4 (b), the gripper is positioned over the cylinder, around the midpoint of the intended gripping section. In Figure 4.4 (c), both rings are moved simultaneously: the top ring ropes are tensioned to pull the top ring upward, while the bottom ring ropes are gradually released to allow the bottom ring to move downward. This achieves a more controlled distribution of the braid along the cylinder. The deployment and elongation of the braid, however, introduces the compacting restoring force T_c , which acts to return the braid towards its fabrication state. The ring weight W_{ring} must therefore be large enough to overcome this restoring force. In reality, friction between the braid and the cylinder further assists by acting against the restoration. In Figure 4.4 (d), once the braid has reached the closed state, the bottom ring ropes are fully released and pretension is applied to the top ring to ensure lifting after cutting. The cylinder is then cut, the pretension is lost, and the system reaches equilibrium, as shown in Figure 4.4 (e). At this point the lifting equipment carries the full weight of the conductor section W_c , the weight of the bottom ring W_{ring} , and the weight of the remaining equipment W_{rest} , the latter being neglected in this simplification.

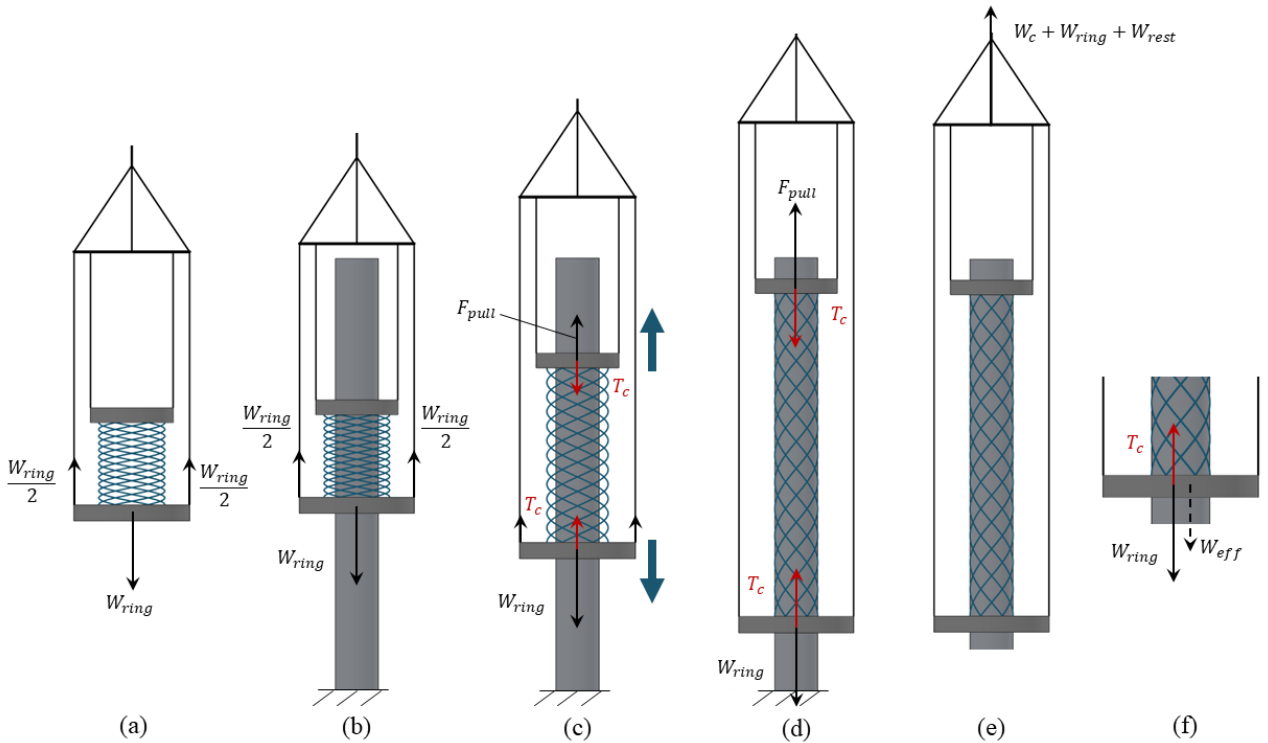


Figure 4.4: Operational sequence of the gripper where fabrication state coincides with open state. (a) Gripper hangs from lifting equipment with top ring ropes slack and sleeve compact. (b) Gripper positioned over conductor at midpoint of section. (c) Top ring pulled upward and bottom ring released downward, elongating braid against compacting restoring force T_c . (d) Closed state reached and pretension applied to top ring. (e) Conductor cut, pretension lost and system at equilibrium. (f) Free body diagram of bottom ring showing effective ring weight

Since the fabrication state coincides with the open state, which is wider than the closed state, the braid has been elongated from its equilibrium to reach the closed state. The restoring force therefore acts against this elongation and tends to compact the braid back towards its fabrication state. As shown in Figure 4.4 (f), this restoring force acts upward on the bottom ring, reducing the effective elongating force it exerts on the braid. The effective ring weight is then:

$$W_{eff} = W_{ring} - T_c \quad (4.39)$$

Where T_c is the positive-valued magnitude of the restoring force evaluated between the fabrication state (r_0, α_0) and the closed state $(r_{closed}, \alpha_{closed})$, as given by Equation (4.35).

In the alternative case, where the fabrication state is taller and slimmer than the closed state, the restoring force tends to elongate the braid with the positive-valued magnitude T_e , pushing the bottom ring further downward. In this case the effective ring weight exceeds the actual ring weight:

$$W_{eff} = W_{ring} + T_e \quad (4.40)$$

In general, the following equation holds with the sign dependent on where the initial state is in relation to the closed state:

$$W_{eff} = W_{ring} \pm T \quad (4.41)$$

It should be noted that the equilibrium of the bottom ring is not fully resolved within this section, as the friction generated between the braid and the conductor surface is deliberately excluded and treated in the following section. The quantity W_{eff} therefore represents the net axial force transmitted to the braid by the bottom ring, accounting for the restoring force arising from the assumed spring behaviour of the helices and serves as the primary input to the gripping model.

It is also important to note that the preceding analysis is a simplification of the real behaviour, based on the assumption that the braid acts as a system of helical springs. This behaviour might not always be valid and can be further simplified in certain cases. If the braid is made of synthetic fibre rope, the bending and torsional stiffness of the fibres can be assumed negligible, so that the axial force in Equation (4.35) reduces to $T \approx 0$ and therefore $W_{eff} \approx W_{ring}$. In this case, however, the negligible stiffness means the braid may not hold its form, as the ropes tend to bunch up rather than return to a defined fabrication state, which is likely to complicate deployment and would require a method to prevent bunching.

If instead the braid is made of a material with appreciable stiffness, such as steel wire rope, the restoring force is not negligible. Nevertheless, in a full-scale design where a heavy bottom ring is advantageous, the restoring force is expected to be negligible compared to the large ring weight, so that $W_{ring} \gg T$ and the simplification $W_{eff} \approx W_{ring}$ again holds, based on a reasonable selection of material and cross-section properties. The only case in which the restoring force may not be negligible is the prototype or small-scale design made from a stiff material such as steel wire, where the ring weight cannot be made large enough to dominate. This is consistent with the behaviour reported in the literature on the mechanics of medical stents.

4.3 The Theoretical Model of Gripping and Lifting

This section derives the holding capacity of the gripper after the conductor has been severed and is held by friction with the braid alone. This holding capacity is the analytical expression of the primary demand that the

gripper carry the entire lifting load (R.2.2), with the self-forcing behaviour, in which the grip increases as the load increases, underlying the maintenance of grip under that load (R.2.3). Two steps are required to derive it. First, the relationship between the axial force along the braid and the resulting radial pressure is established, as this pressure generates the friction between the conductor and the braid. Second, the tension distribution along the length of the sleeve is derived, since the friction generates additional axial force in the braid, which in turn increases the radial pressure and therefore the friction, creating an amplification loop. Applying the relevant boundary conditions then allows the maximum lifting capacity to be estimated.

4.3.1 Axial Force to Radial Pressure

Once the braid is axially deformed, it also deforms radially, following the geometry derived in Section 4.1 with the single-fibre length S held constant. This radial deformation generates a radial pressure acting between the braid and any object held within the sleeve. The aim of this subsection is to establish the relationship between the applied axial force and the resulting pressure.

Certain assumptions are adopted for this derivation. First, the helical pattern is assumed uniform, meaning that the sleeve radius and the braid angle of the fibres remain constant along the length of the sleeve. In reality, the braid follows an hourglass, or necking, pattern, as discussed and derived by Lu et al. (2024). Second, the fibres are assumed not to deform elastically, so that the external work acting on the braid is fully converted into internal energy, which is expressed as pressure on the gripped object.

These assumptions follow from the use of the virtual work principle, applied here according to the derivations of Lu et al. (2024) [25], with the correction to the contact area noted in Section 4.1 and with the parameters relevant to the full-scale design. According to the virtual work principle, the total external virtual work δW equals the total internal virtual energy δU :

$$\delta W = \delta U \quad (4.42)$$

When an axial force F acts on the braid, the pitch of the helices is elongated by δp . The external virtual work is therefore:

$$\delta W = F \delta p \quad (4.43)$$

The change in pitch δp is accompanied by a change in radius δr . When the braid elongates, becoming longer and slimmer, the pitch increases and the radius decreases, so $\delta p > 0$ and $\delta r < 0$. For contraction the opposite holds. Over a single turn of the braid ($N = 1$), the contact area is A_1 , as given by Equation (4.22). The total radial force F_r of this segment is then:

$$F_r = P A_1 \quad (4.44)$$

Where P is the radial pressure. The internal energy associated with the radial force F_r acting through the inward radial displacement δr is:

$$\delta U = -F_r \delta r = -P A_1 \delta r \quad (4.45)$$

The negative sign ensures the internal energy is positive since the radius decreases ($\delta r < 0$) as the braid constricts onto the object.

Substituting into the virtual work principle of Equation (4.42):

$$F\delta p = -PA_1\delta r \quad (4.46)$$

The force conversion factor k is introduced to express how the axial force F is converted into radial pressure P :

$$k = -\frac{\delta p}{A_1\delta r} \quad (4.47)$$

With this the pressure and axial force are coupled by:

$$P = kF \quad (4.48)$$

The relationship between the changing pitch and radius of a uniformly deforming braid, with the single-turn fibre length S_1 held constant, follows from the Pythagorean relation of Equation (4.5):

$$S_1^2 = p^2 + (2\pi r)^2 \quad (4.49)$$

Differentiating this expression with respect to r , with S_1 constant:

$$0 = 2p\frac{dp}{dr} + 8\pi^2 r \quad (4.50)$$

Simplifying, and using the relation $p = 2\pi r \tan \alpha$ from Equation (4.4):

$$\frac{dp}{dr} = -\frac{4\pi^2 r}{p} = -\frac{2\pi}{\tan \alpha} \quad (4.51)$$

Under the assumption of small deformation:

$$\frac{\delta p}{\delta r} \approx \frac{dp}{dr} = -\frac{2\pi}{\tan \alpha} \quad (4.52)$$

Substituting Equation (4.52) into Equation (4.47) gives the conversion factor for a single turn of the braid:

$$k = \frac{2\pi}{A_1 \tan \alpha} \quad (4.53)$$

Expanding this to account for the total contact area A of the braid over N turns, with $A = NA_1$:

$$k = \frac{2\pi N}{A \tan \alpha} \quad (4.54)$$

This is where the discrepancy with Lu et al. (2024) appears, with the number of turns N missing from their conversion factor. The derived factor k otherwise agrees with the derivation of Jedwab and Clerc (1993), with the exception that there the contact area is simplified as a full cylinder of area $2\pi rH$, where H is the cylinder length and the gripping length of the braid [63]. This simplification cannot be made for a braid applied at the scale of offshore conductor removal, where the actual web-like contact area of the braid is much smaller than the cylindrical approximation.

Reformulating the equation above in terms of the selected input parameters r_c , N , and H , and using the relations $\tan \alpha = p/(2\pi r_c)$ and $p = H/N$, gives:

$$k = \frac{4\pi^2 r_c N^2}{A H} \quad (4.55)$$

The expression translating axial force into radial pressure is therefore:

$$P = \frac{4\pi^2 r_c N^2}{A H} F \quad (4.56)$$

This relationship is then used in the following section, together with the tension distribution along the sleeve, to estimate the gripping capacity of the gripper.

4.3.2 Tension Distribution and Holding Capacity

As described, the working principle of the gripper is that elongation of the braid generates radial pressure, which generates friction. This friction is assumed to act purely axially, so that it adds to the total axial force in the braid, which in turn increases the radial pressure and therefore the friction. To capture this effect, the tension distribution along the sleeve must be estimated, from which the friction and the gripping capacity follow. The free body diagram of the system is presented in Figure 4.5, at the instant the conductor is severed and supported by friction at the sleeve alone. The coordinate x is taken to start at the bottom of the top ring, with the total sleeve length being H , as shown in Figure 4.5 (a).

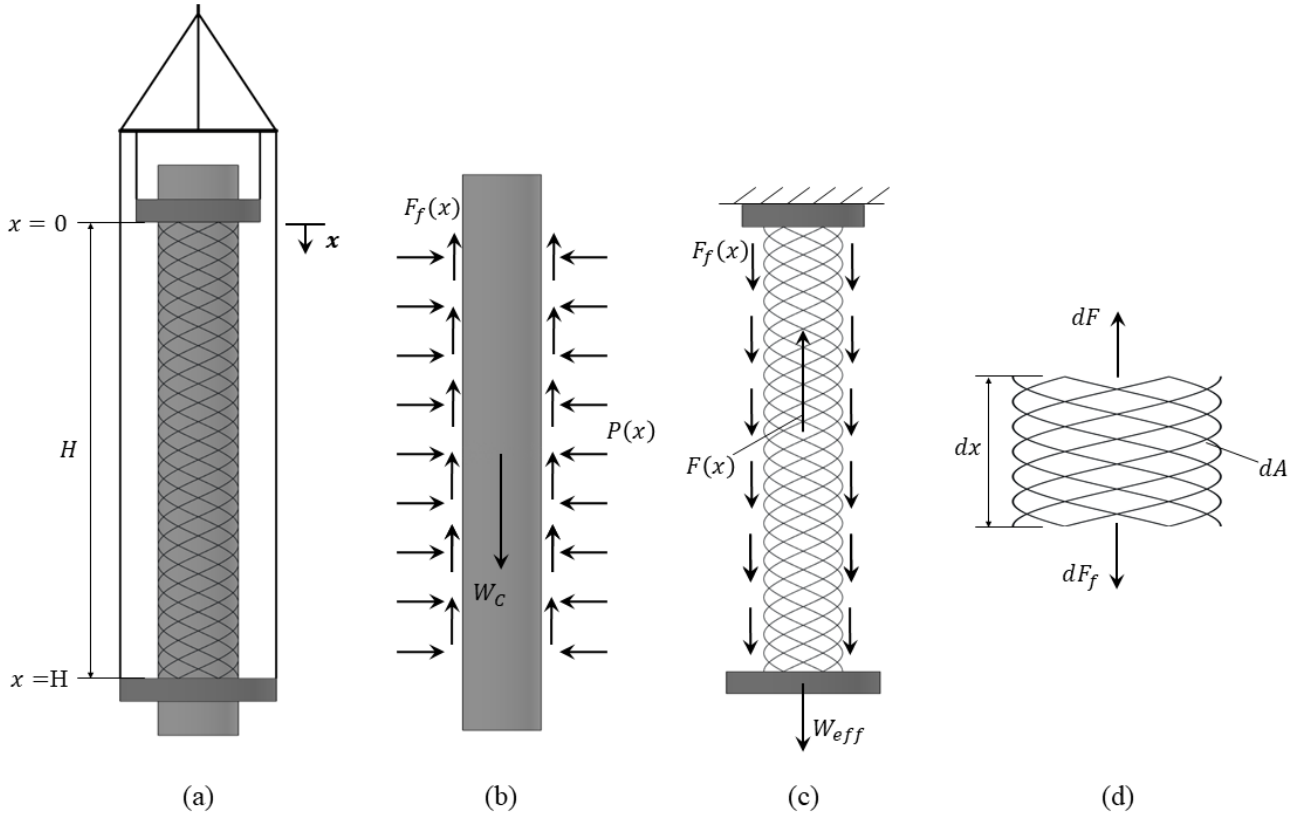


Figure 4.5: Free body diagrams at the instant the conductor is severed. (a) Full system with coordinate x and gripping length H . (b) Conductor free body diagram. (c) Braided sleeve free body diagram. (d) Infinitesimal element dx of the braided sleeve

Figure 4.5 (b) shows the free body diagram of the conductor. Its total weight W_c must be supported, and is carried by the friction distribution $F_f(x)$ which results from the pressure distribution $P(x)$ exerted by the braid.

Figure 4.5 (c) shows the free body diagram of the sleeve. The top ring is assumed fixed at the lifting equipment. The sleeve is elongated by the effective weight of the bottom ring, as described in Equation (4.41), with the sign of the restoring force T depending on the direction in which the spring-assumed helices tend to deform, either pulling the ring upward or pushing it further down. As established in Section 4.2, the braid is assumed to tend towards its fabrication state, which coincides with the open state, so the effective weight is $W_{eff} = W_{ring} - T_c$. Additional axial force comes from the friction with the conductor $F_f(x)$, acting in the direction opposite to that on the conductor. Together these produce the axial tension distribution $F(x)$ in the braid.

At this point the assumptions regarding friction are stated, since they govern the form of the result. The friction between the braid and the conductor is represented by a Coulomb idealisation, in which the local friction force is the product of a single constant coefficient of friction μ and the local normal force. This idealisation is a modelling assumption rather than a requirement, and the coefficient μ , the friction between the braid and a corroded conductor surface, is the most uncertain input to the model. This friction is further assumed to act purely in the axial direction, opposing the slipping of the conductor. The remaining assumptions are a uniform distribution of the braid, and therefore of the contact, along the cylinder, the exclusion of dynamic effects, and the neglect of the braid self-weight, as this is far smaller than the heavy ring.

The pressure distribution resulting from the tension is expressed using the relationship derived in the previous section:

$$P(x) = kF(x) \quad (4.57)$$

This can be divided into a constant pressure P_0 , caused by the axial force from the effective weight of the bottom ring W_{eff} , and a variable pressure $P_{sf}(x)$, arising through the self-forcing effect and the additional tension $F_{sf}(x)$:

$$P(x) = P_0 + P_{sf}(x) = kW_{eff} + kF_{sf}(x) = k(W_{eff} + F_{sf}(x)) = kF(x) \quad (4.58)$$

An infinitesimal element of length dx is introduced in Figure 4.5 (d). The full contact area A of the braid is spread uniformly over the gripping length H , so the contact area dA of the element is:

$$dA = \frac{A}{H} dx \quad (4.59)$$

The available friction dF_f over this element follows from the Coulomb assumption and the relations derived so far:

$$dF_f = \mu P(x) dA = \mu k F(x) \frac{A}{H} dx \quad (4.60)$$

The tension in the braid is highest at $x = 0$, where the braid carries the entire weight of the conductor together with the effective weight of the bottom ring, giving the boundary condition:

$$F(x = 0) = W_c + W_{eff} \quad (4.61)$$

The tension in the braid decreases as x increases along its length, since each element transfers part of the load it carries to the conductor through friction:

$$dF = -dF_f \quad (4.62)$$

Substituting Equation (4.60) gives the following ordinary differential equation (ODE):

$$dF = -\mu k \frac{A}{H} F(x) dx \quad (4.63)$$

The axial decay coefficient γ is introduced to simplify the expression:

$$\gamma = \mu k \frac{A}{H} \quad (4.64)$$

Substituting the coefficient k from Equation (4.55) gives:

$$\gamma = \mu \frac{4\pi^2 r_c N^2}{A H} \frac{A}{H} = \mu \frac{4\pi^2 r_c N^2}{H^2} \quad (4.65)$$

This clearly shows that γ does not depend on the actual contact area A or on the coordinate x , as expected from the Coulomb assumption, but only on the braid geometry and on the material through the friction coefficient μ . The differential equation is then:

$$\frac{dF}{F} = -\gamma dx \quad (4.66)$$

The solution of this ODE is:

$$F(x) = F(0)e^{-\gamma x} \quad (4.67)$$

Substituting the boundary condition (4.61) for $F(0)$ gives the tension distribution along the coordinate x :

$$F(x) = (W_c + W_{eff})e^{-\gamma x} \quad (4.68)$$

To estimate the maximum theoretical weight the gripper can support, a slip condition must be established. The braid tension is the sum of the constant contribution W_{eff} from the bottom ring and the self-forcing contribution $F_{sf}(x)$ built up by friction with the conductor, as in Equation (4.58). The self-forcing increment is largest at the top ring, where it reacts the full conductor weight, and decays with height as friction progressively transfers the load into the conductor. At the verge of slipping under the maximum sustainable conductor weight, this increment is fully consumed by the time the bottom of the sleeve is reached, so that $F_{sf}(H) = 0$. Were the tension at the bottom greater than W_{eff} , spare gripping capacity would remain and a heavier conductor could be held. The slip threshold therefore sits exactly at:

$$F(H) = W_{eff} \quad (4.69)$$

Substituting this into Equation (4.68):

$$W_{c,max} = W_{eff}(e^{\gamma H} - 1) \quad (4.70)$$

Finally, the dimensionless self-forcing coefficient β is introduced to simplify the expression:

$$\beta = \gamma H = \mu \frac{4\pi^2 r_c N^2}{H^2} H = \mu \frac{4\pi^2 r_c N^2}{H} \quad (4.71)$$

The final expression for the maximum weight that can be lifted is then:

$$W_{c,max} = W_{eff}(e^\beta - 1) = W_{eff} \left(e^{\mu \frac{4\pi^2 r_c N^2}{H}} - 1 \right) \quad (4.72)$$

The maximum holding capacity grows exponentially with the self-forcing coefficient β . A larger conductor radius or a larger number of turns increases β and therefore amplifies the capacity, while a longer gripping length reduces it. The capacity scales linearly with the effective ring weight W_{eff} , so the ring sets the baseline that the exponential self-forcing then multiplies.

This expression closely matches the maximum holding force derived by Shang et al. (2019) for the self-forcing braided gripper [26], the difference being the contribution of the ring introduced in this work. As noted in their research, self-forcing may not develop when gripping fragile or light objects, where the weight is insufficient to provide the additional friction required to amplify the holding capacity, which sets a minimum holding force. This case corresponds to the radial pressure remaining constant at P_0 , so that the grip reduces to the friction generated by the preload alone:

$$W_{c,min} = \mu P_0 A = \mu k W_{eff} A = \gamma H W_{eff} = \beta W_{eff} \quad (4.73)$$

The minimum and the maximum describe the two regimes of the same mechanism. When the object is too light to drive the feedback, the grip remains at the preload friction βW_{eff} , a term that scales linearly with the geometric amplification captured by β . When the object is heavy enough, the friction generated by the load itself compounds through the hypothesised self-forcing effect, and the capacity rises toward $W_{eff}(e^\beta - 1)$, which grows exponentially. This separation between the two regimes is what distinguishes self-forcing from ordinary friction, and observing it is the objective of the prototype.

Conductors are heavy objects, so the self-forcing regime is expected to govern. Whether this holds in practice is the one question the model cannot settle on its own, since it depends on the friction conditions between the braid and the conductor surface, which are unknown in advance and cannot be reproduced at prototype scale. As the exact friction behaviour at full scale cannot be captured by a scaled prototype, the objective is not to confirm the precise exponential form of Equation (4.72), but to establish whether the holding capacity amplifies with increasing β or remains close to the linear preload response of Equation (4.73). Observing this amplification across a range of β would confirm that the self-forcing mechanism develops, independently of the exact functional form it takes.

For this reason, the absolute holding capacity is not the object of validation. What the model does establish, independently of the friction conditions, is the qualitative distinction between these two regimes and the way the self-forcing coefficient β scales with the braid geometry, through its proportionality to the square of the number of turns N^2 and to the conductor radius r_c , and its inverse proportionality to the gripping length $1/H$. The demand that the gripper carry the entire lifting load (R.2.2) is therefore established here in form and scaling, and the remaining question, whether the holding capacity amplifies with β as the self-forcing mechanism predicts, is carried forward as the objective of Chapter 6.

4.4 Maximum Fibre Tension

A final consideration for scaling the gripper is to identify the weakest structural points and verify that the design is safe. The critical elements are the individual fibres, and the critical location is the top ring connection, where the braid tension reaches its maximum value $F(0) = W_c + W_{eff}$, as established in Section 4.3.2.

Two approaches exist for verifying fibre safety. The first follows the helical spring treatment of Jedwab and Clerc (1993), estimating the bending and torsional stresses in the fibre and comparing them against the yield strength of the material [63]. The second, adopted here, treats each fibre as a rope of negligible bending stiffness carrying pure axial tension. This is the more practical approach for the present application. As established earlier in this chapter, the fibres at full scale behave essentially as ropes, being either synthetic rope or steel wire rope whose restoring stiffness is overpowered by the heavy bottom ring. Such fibres are specified by manufacturers as ropes with a minimum breaking load, and the elastic properties required for a stress-based analysis are difficult to determine and rarely provided. The rope treatment also applies regardless of cross section, whether circular or rectangular, since only the cross-sectional area enters and the comparison is made against the rated breaking load rather than a computed stress.

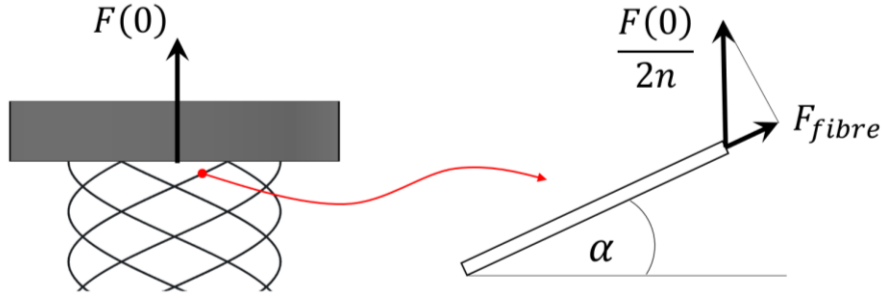


Figure 4.6: Axial tension $F(0)$ at the top of the braid and the resolution of a single fibre tension into its vertical component

As shown in Figure 4.6, the total axial tension $F(0)$ at the top of the braid is carried by the $2n$ fibres, with n the number of fibre sets. The fibres are assumed to meet the ring at the closed-state braiding angle α . The maximum tension in a single fibre is therefore:

$$F_{fibre} = \frac{F(0)}{2n \sin \alpha} = \frac{W_c + W_{eff}}{2n \sin \alpha} \quad (4.74)$$

The fibre tension increases as the braiding angle decreases, since a shallower fibre contributes a smaller axial component and must carry a correspondingly larger tension. This tension should then be compared against the minimum breaking load of the selected rope, reduced by a safety factor, as carried out for the full-scale design in Chapter 7, where it forms the basis for verifying the gripper against the offshore lifting safety factors and the prevention of dropped objects, as set in the requirement list (R.3.1, R.3.3).

4.5 Summary

This chapter developed the analytical model of the braided sleeve gripper. The geometry of the braid was defined first, relating the radius, pitch, and braiding angle, and introducing the open and closed states together with the feasibility ratios that support the design process. The transition between states was then derived, showing that the effective ring weight $W_{eff} = W_{ring} \pm T$ governs the axial force available for gripping, with the heavy bottom ring introduced to supply this force reliably at full scale. The gripping model followed, coupling the axial tension to radial pressure and friction to yield the maximum holding capacity $W_{c,max} = W_{eff}(e^\beta - 1)$, which grows exponentially with the self-forcing coefficient β , depending only on the geometry of the braided sleeve and the friction coefficient μ . Finally, the maximum fibre tension was located at the top ring connection.

For the full-scale design proposed later in this work, the restoring force is expected to be negligible relative to the heavy bottom ring, so that the effective weight reduces to $W_{eff} \approx W_{ring}$ and the transition force T drops out of the model. The full equation is retained here because it remains necessary for the prototype, where the ring cannot be made heavy enough to dominate the restoring force.

The model rests on several idealisations, including a uniform helical geometry, Coulomb friction, and the spring approximation of the braid. In terms of the requirement list, the model advances the central demand identified in Section 3.5.3 from an assertion of the concept to an analytically established relationship, quantifying the lifting capacity (R.2.2) and the self-forcing behaviour that maintains grip as the load increases (R.2.3). This establishment holds in form and scaling independently of the friction conditions between the braid and the conductor, and remains conditional only on the hypothesis that self-forcing actually develops for a heavy conductor, which requires experimental validation. The following chapters therefore present the design and fabrication of a scaled prototype, its objectives, the experiments carried out, and the discussion of the results, providing the evidence required to substantiate this hypothesis. This validation increases confidence in the full-scale design proposed afterward, through which SRQ2 and SRQ3 are answered.

5 Prototype Design

Continuing from the previous chapter, which established the holding capacity of the braided sleeve gripper in form and scaling, this chapter sets out to design a scaled prototype capable of validating the hypothesised behaviour. The objectives and validation strategy are defined first, followed by the design and fabrication of the prototype, and concluding with the experimental plan.

5.1 Objectives of the Prototype

As established in the previous chapter, the main result of the model is the hypothesis that the holding capacity of the braided sleeve grows exponentially with the self-forcing coefficient β . The prototype is not intended to predict the exact holding capacity, either at the prototype scale or at full scale, since the model depends on several idealisations carried throughout the derivation, namely the uniform helix and uniform contact assumptions, the inextensible-fibre pressure assumption, and the spring approximation of the braid. The most consequential of these is the Coulomb friction idealisation, since the friction conditions between the braid and a corroded conductor surface under marine conditions cannot be controlled in the laboratory and cannot be known in advance offshore.

The first objective is therefore to confirm that the self-forcing mechanism physically develops, meaning that the sleeve grips and holds a load exceeding what the bottom ring preload alone would sustain, and that this holding capacity amplifies with increasing β as proposed by Equation (4.72) rather than remaining linear as in Equation (4.73). The second objective is to confirm that the geometric scaling laws derived in Chapter 4 govern the physical system, so that the model can be trusted to inform a full-scale design recommendation under appropriate safety factors. This second objective is deliberately limited in scope. Because the friction coefficient is unknown and uncontrollable, the exact magnitude of the holding capacity cannot be validated, and no attempt is made to do so. What can be validated independently of this unknown is the structural form of the capacity relation and the exponents with which each geometric parameter enters it, since the friction coefficient enters the model only as a multiplicative constant and cannot alter these exponents. The model is therefore not intended as an accurate representation of the physical behaviour in full detail, but as an approximation sufficient to inform a design recommendation at an early design phase, together with recommendations for how the model could be refined in subsequent work. These objectives shape the experimental approach presented in the following subsection.

Both objectives address the requirement that the gripper carry the entire lifting load (R.2.2) and maintain grip under that load (R.2.3), identified in Section 3.5.3 as established in form and scaling but requiring experimental validation. Framed against the sub-research questions, the first objective addresses SRQ3, which concerns building and testing a prototype to demonstrate the feasibility of the selected concept, while the second supports the full-scale design recommendation of Chapter 7 and therefore SRQ2.

A secondary objective is qualitative and observational rather than measured. As multiple tests are carried out across different configurations, this presents the opportunity to observe how the prototype behaves in practice, particularly during deployment and release, informing the full-scale design and future research. The following section translates these objectives into a validation strategy designed to isolate each behaviour in turn.

5.2 Validation Strategy

After defining the objectives, the validation method and approach for the experiments must also be established, since this guides the design of the prototype. A first simplification is made regarding the braid material. As the goal of the prototype is to demonstrate the self-forcing characteristic of the gripper, less importance is placed on the material of the braid itself. Synthetic fibre rope is therefore chosen, as it captures the self-forcing

behaviour while remaining straightforward to fabricate the prototype by hand. This choice allows the restoring force to be neglected, so that $T \approx 0$ and $W_{eff} \approx W_{ring}$, as the bending and torsional stiffness of the braid can be disregarded, as explained in the previous chapter. The model used for the tests is therefore:

$$W_{slip} = W_{ring}(e^{\beta} - 1) = W_{ring} \left(e^{\mu \frac{4\pi^2 r_c N^2}{H}} - 1 \right) \quad (5.1)$$

With W_{slip} the weight of the object at the point of slip. The corresponding case in which no self-forcing behaviour develops, corresponding to Equation (4.73), reduces to:

$$W_{slip} = \beta W_{ring} = \mu \frac{4\pi^2 r_c N^2}{H} W_{ring} \quad (5.2)$$

From these expressions the central methodological difficulty is clear. The maximum holding capacity depends on the friction coefficient μ , which is highly variable for compliant polymers and cannot be measured independently of the braid mechanics itself. To separate the friction coefficient from the geometry, a geometric coefficient Γ is defined that depends only on the braid geometry:

$$\Gamma = \frac{\beta}{\mu} = \frac{4\pi^2 r_c N^2}{H} \quad (5.3)$$

This separation is what allows the validation to proceed without knowing μ , since it is the shape of the relationship between W_{slip} and Γ , rather than its absolute value, that reveals whether self-forcing is occurring. Based on this, four claims can be proposed for validation, each isolating one aspect of the theoretical model:

1. **Ring weight linearity.** The holding capacity is expected to scale linearly with the ring weight W_{ring} at constant geometry, and therefore constant Γ . Both Equations (5.1) and (5.2) predict this linearity regardless of μ , so this test is independent of the friction coefficient and serves as a baseline check of the model structure before the self-forcing behaviour itself is examined.
2. **Structural form of the holding capacity.** The central claim concerns whether the capacity amplifies with β as proposed by the self-forcing model of Equation (5.1) or remains linear as in the non-amplifying case of Equation (5.2). Exact agreement with the exponential prediction is not expected, since this would require knowledge of the friction coefficient and of the true contact mechanics, neither of which can be established. The aim instead is to demonstrate clearly that the holding capacity does not scale linearly with Γ and exhibits the amplification characteristic of self-forcing. This is tested by varying Γ over a wide range, through independent variation of N , H , and r_c while keeping the braid material and contact conditions constant. The measured capacity is then compared against both Equations (5.1) and (5.2), each evaluated over a friction coefficient range taken from literature rather than from a value obtained through a separate friction test. A dedicated friction measurement is deliberately avoided, since it would not reproduce the contact mechanics of a helical braid under radial pressure and would therefore not represent the friction conditions relevant to this application.
3. **Geometric trends.** The dependence on the individual geometric parameters is examined by testing whether the measured results follow the correct trend as N , H , and r_c are each varied individually, by holding the other parameters at a constant value. The goal is not to reproduce these trends exactly, which the unknown friction coefficient does not allow, but simply to establish, for each parameter, whether the measured capacity increases or decreases in the direction and to the degree predicted.

4. **Area independence.** The Coulomb friction idealisation underlying the model is independent of the true contact area. This implies that changing the number of rope sets n should not alter the resulting holding capacity for a constant geometry and constant Γ . A significant change in capacity would indicate an effect the model does not capture.

Finally, the braid material is varied from synthetic rope to steel wire, in order to assess a stiffer braid against the compliant one, and in particular the hypothesised spring-like restoring behaviour. This comparison is treated qualitatively rather than quantitatively, as fabricating multiple braids in steel wire is time-consuming relative to the additional information it would provide. The steel wire braid is expected to retain its fabricated form more consistently during deployment, owing to its greater stiffness. The absolute capacity of the steel configuration is not directly comparable to that of the polyamide baseline, since the friction conditions and the self-weight of the braid relative to the ring differ between the two materials. The repeatability of the steel configuration, however, is directly comparable, and provides insight into the extent to which the uniform-helix assumption of the model is satisfied in practice.

5.3 Design and Fabrication of the Prototype

This section presents the physical realisation of the prototype. The component decisions are described first, followed by the fabrication method developed to construct the braid.

5.3.1 Prototype Design

The materials and design of the prototype were kept as simple as possible, since multiple braids had to be fabricated across different parameter configurations. For the synthetic braid, a widely available 3 mm diameter polyamide braided rope was selected. For the steel wire braid, a smaller diameter of 2 mm was chosen, as the reduced diameter made hand-braiding more manageable for a stiffer material.

The two rings were 3D-printed in PLA and designed to accommodate all parameter variations across the test campaign, with sufficient thickness to withstand the applied lifting loads. Each ring has a mass of approximately 100 g. The technical drawing is presented in Appendix F. Optional control handles were also printed to allow independent control of each ring, providing a more representative demonstration of the deployment procedure. These were connected to the rings through separate ropes but were not used during the slip tests, as they complicated the test procedure.

The gripped object in all tests is a smooth PVC pipe, chosen for its consistent surface finish and availability in multiple standard diameters. A length of 1 m was selected as a more representative scale compared to similar grippers reported in the literature. The three diameters of 60, 50, and 40 mm were used as described in the experimental plan. The mass of each pipe was measured and included in the total slip mass for each test.

5.3.2 Prototype Fabrication and Assembly

The fabrication method was adapted from Kortman et al. (2025), in which a braided sleeve for an artificial muscle application was manufactured with emphasis on controlling the braiding angle locally [67]. The level of precision required in that work was not necessary here, so a simplified approach was adopted. Rather than pulling the mandrel downward with a constant weight, the mandrel was fixed to the bottom ring and the braiding disc was advanced upward along threaded rods. The disc itself is a modified version of the traditional Japanese Kumihimo braiding disc, scaled up to accommodate the larger mandrel diameters used here. The disc was 3D-printed in PLA and its technical drawing is presented in Appendix F. The PVC pipe used for the slip tests served simultaneously as the braiding mandrel.

The fabrication proceeds as follows, with each step illustrated in Figure 5.1. In Figure 5.1 (a), the ropes are knotted through the holes on the bottom ring, labelled by direction, and a small weight is attached to the free end of each rope to maintain tension during braiding. Since each pair of opposing helices shares a common starting point, two ropes are fixed at each ring position. In Figure 5.1 (b), the threaded rods are attached to the bottom ring with nuts, and a washer-and-nut assembly is placed on each rod to act as a seat for the braiding disc. In Figure 5.1 (c), the braiding disc, with its slots labelled, is positioned on the washers, and the PVC mandrel is placed through the centre of the ring and fixed in place using a cross-pattern arrangement of bolts and nuts, ensuring correct centring and a stable vertical position.

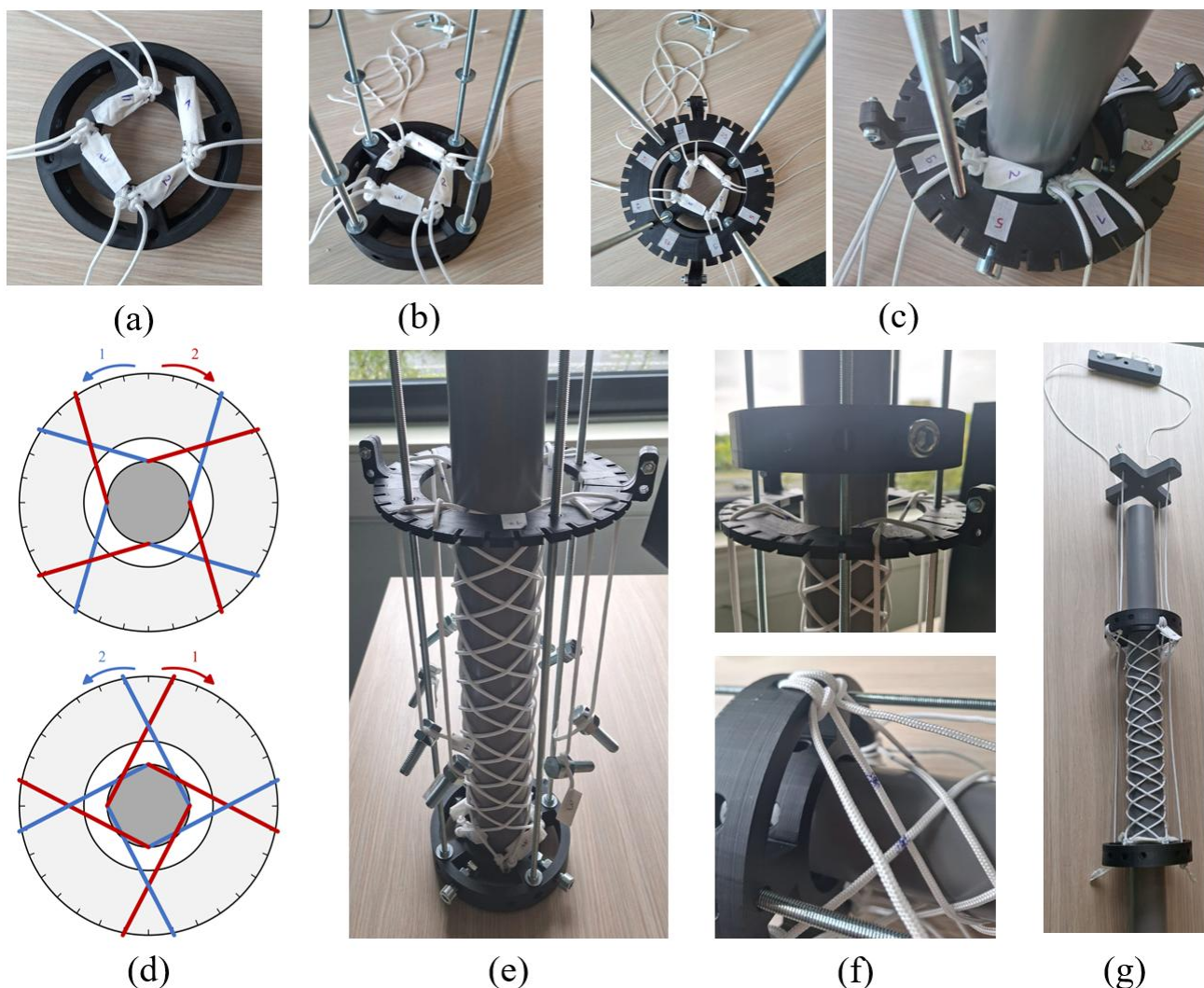


Figure 5.1: Fabrication sequence of the braided sleeve prototype. (a) Ropes knotted to the bottom ring with weights attached to maintain tension. (b) Threaded rods attached to the bottom ring with washer-and-nut seats for the braiding disc. (c) Braiding disc positioned on the seats with the PVC mandrel centred and fixed. (d) Two consecutive braiding cycles, showing the alternating crossings (e) Braiding cycles repeated until the desired length is reached. (f) Top ring positioned and fixed with ropes knotted to outer holes. (g) Completed prototype with optional control handles attached

The first braiding cycle then begins, as shown in Figure 5.1 (d). All clockwise ropes are moved one slot in the clockwise direction, then all counterclockwise ropes are moved one slot in the counterclockwise direction, crossing over the clockwise ropes. The disc height is then advanced upward by turning the nuts on the threaded rods. In the following cycle the order is reversed: the counterclockwise ropes move first, with the clockwise ropes following crossing over them. This alternation of crossings continues as the disc is raised progressively higher, replicating the Naiki Gumi braiding pattern, the same interlacing used in the Chinese finger trap and in braided medical stents. The braiding cycles are repeated, as shown in Figure 5.1 (e), until the desired braid

length is reached. For a braid of N full turns, a gripping length of H , and $2n$ individual ropes, the total number of cycles required is:

$$N_{cycles} = 2nN \quad (5.4)$$

With the disc advanced by the following increment per cycle:

$$h = \frac{H}{N_{cycles}} \quad (5.5)$$

Once all cycles are complete, the top ring is positioned and fixed on the mandrel in the same manner as the bottom ring, as shown in Figure 5.1 (f). The ropes are knotted through the outer holes of the top ring, as the inner holes are not accessible at this stage of assembly. All fabrication components are then removed. The braiding disc is designed in two halves held together with screws, allowing it to be disassembled and removed with ease. The completed prototype is shown in Figure 5.1 (g). The optional control handles are also shown connected through rope to each ring, with the cross-shaped handle connecting to the top ring and the straight handle connecting to the bottom ring, allowing independent control of each ring during deployment.



Figure 5.2: Steel wire braid fabricated using the same method, with U-bolt clamps fixing the wire ends

For the steel wire braid, the same fabrication method was followed, with the wire ends fixed using U-bolt clamps in place of knots, as shown in Figure 5.2. Braiding with steel wire is considerably more demanding than with synthetic rope, as the wire tends to escape the disc slots and is harder to control during each cycle.

5.4 Experimental Plan

This section presents the conducted tests. The test matrix defining the braid geometries tested is presented first, followed by the manufactured braids, and finally the test setup and procedure.

5.4.1 Test Matrix

This section presents the test matrix for the experiments, following the approach defined in Section 5.2, in which each braid parameter is varied independently. Each group is designed to address one of the validation claims established in that section. A baseline configuration is defined first, from which each sweep departs. The baseline

values were selected as a representative midpoint across the range of parameters to be tested. Table 5.1 presents the baseline braid properties.

Table 5.1: Baseline configuration of the prototype

Quantity	Symbol	Value	Unit
Cylinder radius	r_c	30	mm
Number turns	N	3	-
Gripping Length	H	400	mm
Number of rope sets	n	4	-
Mass of the bottom ring	m_{ring}	100	g
Weight of the bottom ring	W_{ring}	0.98	N
Braid material	3 mm polyamide rope		

From this baseline, six experimental groups are defined, each varying one governing parameter while holding all others at their baseline value. Group A varies the bottom ring weight W_{ring} by adding mass to the ring, addressing the ring weight linearity claim. Group B varies the number of turns N , requiring a new braid to be fabricated for each sub-configuration. Group C varies the gripping length H , also requiring re-fabrication. Group D varies the cylinder radius r_c , using a different mandrel and pipe for each sub-configuration, so that the mandrel radius matches the pipe radius to be gripped at each step. Together, Groups B, C, and D vary the geometric coefficient Γ and therefore address both the structural form of the holding capacity and the individual geometric trends.

Group E varies the number of rope sets n to test the area independence implied by the Coulomb friction idealisation, namely the assumption that the contact area does not influence the gripping force. Group F substitutes the polyamide rope with a 2 mm steel wire braid of otherwise identical geometry, in order to assess deployment behaviour and repeatability rather than to compare absolute capacity. Each group includes two additional sub-configurations relative to the baseline, providing sufficient information to assess trends in the model without requiring an excessive number of fabricated braids. Table 5.2 presents the full test matrix.

Table 5.2: Experimental test matrix

Group	W_{ring} [N]	N	H [mm]	r_c [mm]	n	Braid material
A	0.98, 5.20, 8.80	3	400	30	4	Polyamide
B	0.98	2, 3, 4	400	30	4	Polyamide
C	0.98	3	200, 400, 600	30	4	Polyamide
D	0.98	3	400	20, 25, 30	4	Polyamide
E	0.98	3	400	30	2, 4, 8	Polyamide
F	0.98	3	400	30	4	Steel wire

Although the spacing between sub-configuration values appears small, the exponential dependence of the gripping model on the braid parameters ensures that even modest changes in N , H , or r_c produce large changes in the predicted holding capacity. This wide resulting range in the geometric coefficient Γ is what allows both the structural form of the holding capacity and the individual geometric trends to be assessed, providing sufficient information to judge the behaviour and trends of the model.

5.4.2 Realised Prototype Configuration

This section presents the manufactured prototype configurations used across the experimental groups. Figure 5.3 shows the baseline configuration together with the method used to increase the bottom ring weight, in which additional mass is hung from hooks attached to the bottom ring.



Figure 5.3: Baseline configuration and ring weight variation. (a) Baseline: $N=3$, $H=400$ mm, $r_c=30$ mm, $n=4$, $W_{\text{ring}}=0.98$ N, polyamide rope. (b) Hooks attached to the bottom ring for additional mass

Figure 5.4 shows the prototypes corresponding to the number of turns and gripping length variations. As expected from the geometry, increasing N produces a visibly denser braid over the same gripping length, with the same effect observed when reducing H while keeping N constant.

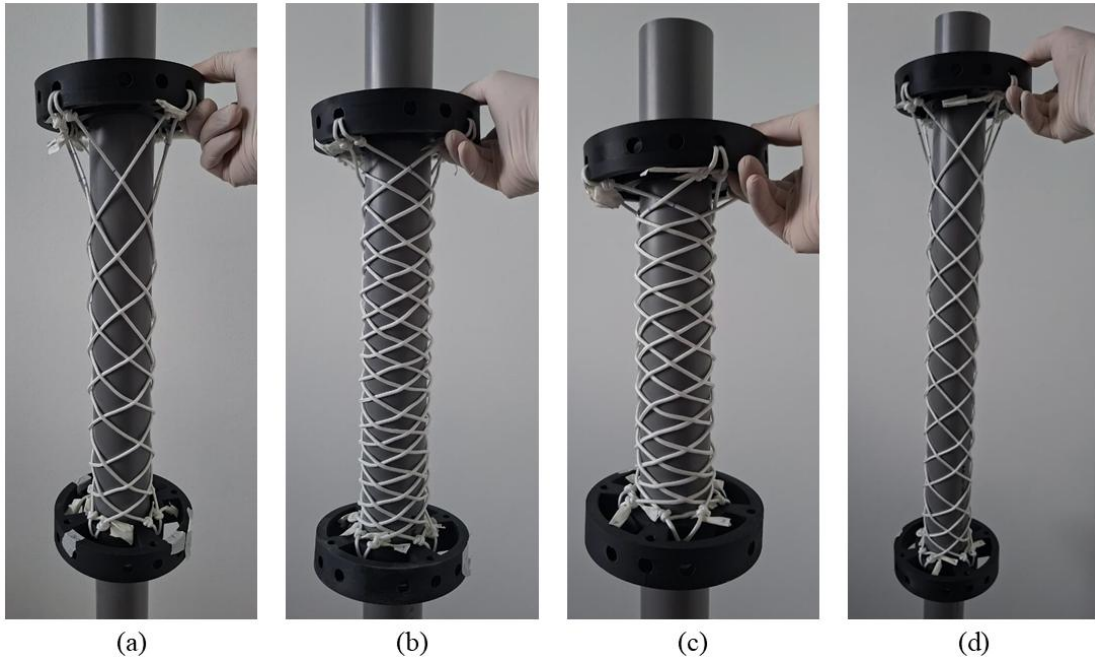


Figure 5.4: Number of turns and gripping length variation. (a) $N=2$. (b) $N=4$. (c) $H=200$ mm. (d) $H=600$ mm

Figure 5.5 shows the prototypes corresponding to the pipe radius variation, the number of rope sets variation, and the steel wire braid. The steel wire braid is visibly stiffer and holds its theoretical form more consistently than the synthetic rope braid.

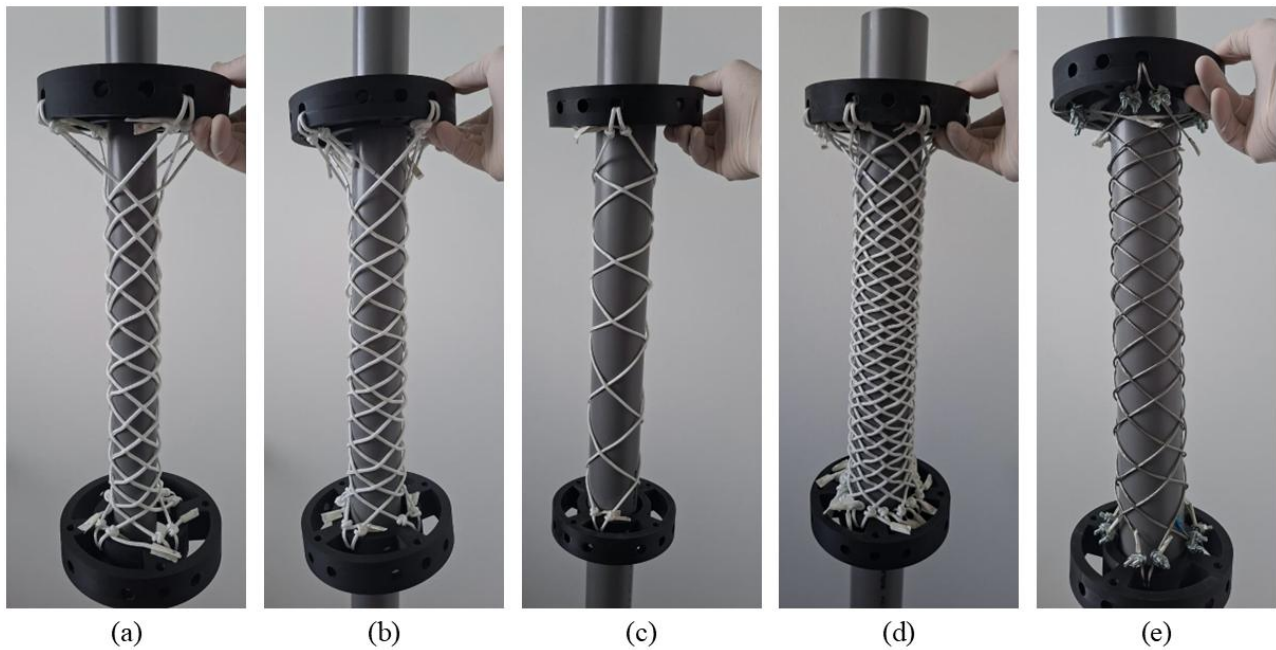


Figure 5.5: Pipe radius, number of rope sets, and material variation. (a) $r_c=20$ mm. (b) $r_c=25$ mm. (c) $n=2$. (d) $n=8$. (e) Steel wire

5.4.3 Test Procedure

The test procedure was kept simple and repeatable. A through-hole was drilled in the pipe and a bolt and nut were fixed through it, allowing a bucket to be suspended from the bolt via a carabiner. The bucket was filled incrementally with a bulk material such as gravel to find the slip point with sufficient precision.

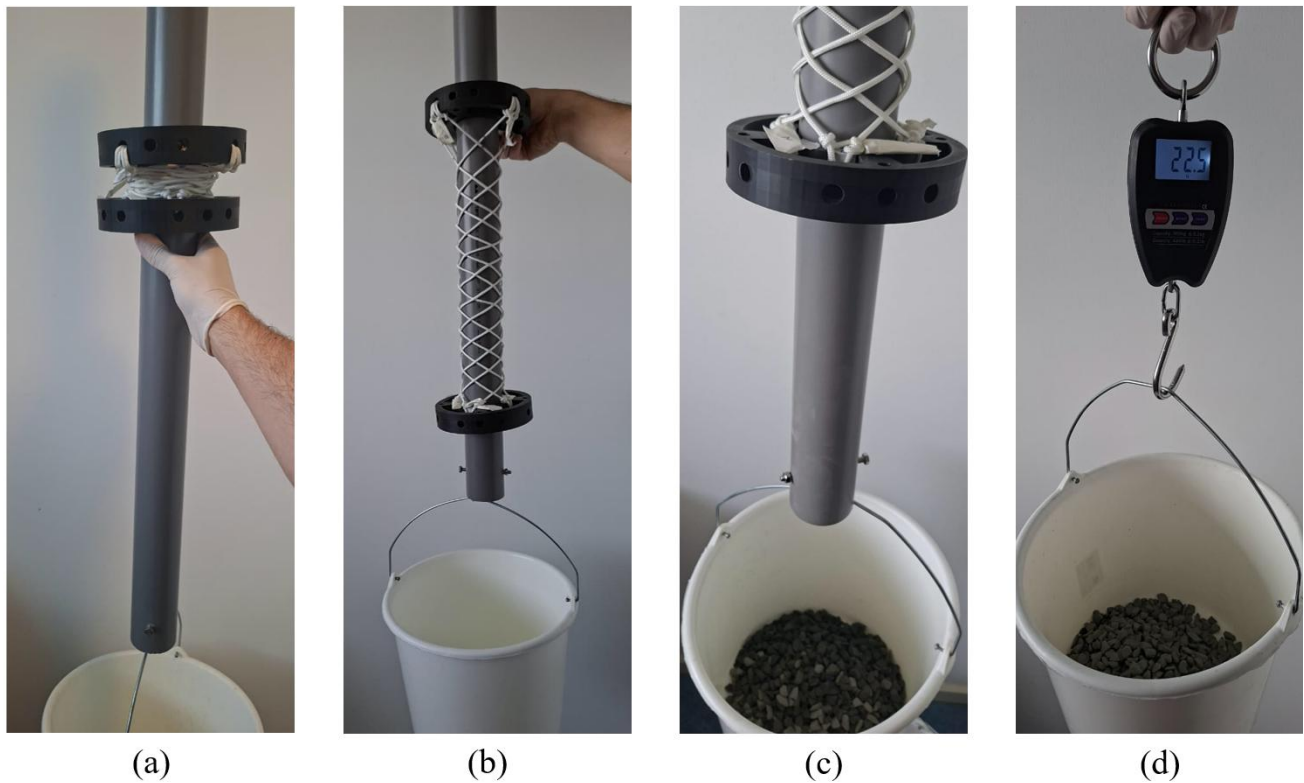


Figure 5.6: Test procedure. (a) Gripper in open state lowered over the pipe. (b) Gripper in closed state after the top ring is pulled upward and the bottom ring released. (c) Bucket filled incrementally with gravel until slip occurs. (d) Bucket removed and weighed

The procedure for a single test is as follows and is illustrated in Figure 5.6. In Figure 5.6 (a), the gripper in its open state is lowered over the pipe. The bottom ring is then released while the top ring is simultaneously pulled upward, bringing the gripper into its closed state, as shown in Figure 5.6 (b). Once deployed, the bottom ring is not touched, since pushing it downward would increase the tension in the braid and artificially increase the gripping capacity. Gravel is then filled into the bucket incrementally, as shown in Figure 5.6 (c). The filling is done slowly to avoid dynamic disturbances that could shock the system, and to allow the slip point to be observed clearly. Slip is identified optically as the first instance of relative movement between the gripper and the pipe. The pipe may slip a small amount and re-engage, but the first instance of slip is recorded for consistency. For configurations expected to hold a load exceeding a full bucket of gravel, a fixed initial weight was placed in the bucket before filling, based on the model prediction for that configuration.

Once slip occurs, the bucket is removed and weighed using a TEMPO DI SALDI BIL200 digital dynamometer, as shown in Figure 5.6 (d). The instrument operates on a load-cell principle and measures loads in the range of 0.5 kg to 200 kg with a resolution of 50 g [68]. Readings were taken directly in Newtons. The weight of the pipe and the hanging bolt assembly was added to each bucket reading to obtain the total slip load W_{slip} . Each configuration was tested five times to obtain a mean and standard deviation.

6 Experimental Validation

The analytical model developed in Chapter 4 establishes the holding capacity of the braided sleeve in both form and scaling, conditional on the hypothesis that the self-forcing mechanism develops under load. This chapter presents the experimental campaign designed to examine that hypothesis and to test the model against the behaviour of a physical prototype. As explained in Chapter 5, this validation rests on the structural form of the capacity relation and on the geometric braid parameters that govern it, and not on the friction coefficient, which is a quantity that can neither be prescribed nor measured independently in this configuration.

The chapter first presents the measured results and their statistical treatment. It then tests the four model claims established in Section 5.2 against the data, examines the back-calculated friction coefficient in order to assess whether the Coulomb assumption is defensible, reports the payload-to-self-weight ratio of the gripper, and closes with a summary and discussion of the experiments.

6.1 Experimental Results

Each configuration in the test matrix of Table 5.2 was tested five times, and the load at the point of slip was recorded for each repetition. The slip load of a single test is the sum of the weight suspended in the bucket, the weight of the pipe, and the weight of all other elements attached to the pipe.

The complete set of individual repetitions for every configuration is presented in Appendix G. For each configuration, the five repetitions are reduced to a mean and a standard deviation. The mean slip load is:

$$\overline{W}_{slip} = \frac{1}{5} \sum_{i=1}^5 W_{slip,i} \quad (6.1)$$

The scatter of the repeated measurements is expressed by the sample standard deviation, which is the appropriate metric for a small number of repeated measurements:

$$s = \sqrt{\frac{1}{4} \sum_{i=1}^5 (W_{slip,i} - \overline{W}_{slip})^2} \quad (6.2)$$

To compare scatter across configurations, the standard deviation is normalised by the mean to give the coefficient of variation (CV):

$$CV = \frac{s}{\overline{W}_{slip}} \quad (6.3)$$

The coefficient of variation is dimensionless and therefore comparable between configurations whose absolute holding capacities differ. For this reason, it is adopted as the unifying repeatability metric throughout this chapter.

Additionally, the resolution of the dynamometer is 50 g, corresponding to approximately 0.5 N, which is negligible relative to the measured loads. Slip was identified optically as the first instance of relative movement between the braid and the pipe.

Table 6.1 presents the mean slip load, sample standard deviation, and coefficient of variation for each configuration, together with the geometric coefficient Γ computed from the braid geometry. The baseline configuration is listed once and serves as the shared reference point for Groups A through F.

Table 6.1: Summary of results and statistics for all configurations

Group	N	W_{ring} [N]	H [mm]	r_c [mm]	n	Material	Γ	$\bar{W}_{slip}$ [N]	s [N]	CV [%]
Baseline	3	0.98	400	30	4	Polyamide	26.6	36.5	6.1	16.7
A	3	5.20	400	30	4	Polyamide	26.6	168.9	19.0	11.3
A	3	8.80	400	30	4	Polyamide	26.6	306.5	37.7	12.3
B	2	0.98	400	30	4	Polyamide	11.8	8.0	3.2	39.7
B	4	0.98	400	30	4	Polyamide	47.4	515.9	66.1	12.8
C	3	0.98	200	30	4	Polyamide	53.3	>741.0	-	-
C	3	0.98	600	30	4	Polyamide	17.8	25.7	12.1	46.9
D	3	0.98	400	20	4	Polyamide	17.8	12.8	0.8	6.5
D	3	0.98	400	25	4	Polyamide	22.2	24.9	2.9	11.8
E	3	0.98	400	30	2	Polyamide	26.6	24.4	8.9	36.5
E	3	0.98	400	30	8	Polyamide	26.6	46.1	3.1	6.7
F	3	0.98	400	30	4	Steel wire	26.6	88.5	1.1	1.3

As can be seen, a single configuration could not be brought to slip with the present experimental setup and methodology. The $H = 200$ mm configuration sustained the maximum applied load of 741 N, approximately 75 kg, and additional manual pulling on the pipe produced no slip. Loading was stopped at this point, as the PLA rings were observed to flex near this load. The true slip capacity of this configuration therefore exceeds 75 kg, which is the upper load that the experimental setup can resolve. This is examined further among the limitations of the experiments.

Several of the model comparisons that follow require a range for the friction coefficient between the polyamide braid and the PVC surface, since this coefficient cannot be measured independently in the present configuration. A range of 0.10 to 0.16 is adopted for this purpose, taken from the capstan-method coefficients reported by Samson Rope for synthetic fibre ropes in dry contact against a smooth surface, where the absence of any lubricant represents the condition most consistent with the present tests [69]. This range is used to bound the theoretical predictions against which the experimental results are compared in the following sections.

6.2 Validation of the Model Claims

Section 5.2 established four claims, each isolating one aspect of the analytical model and each testable independently of the unknown friction coefficient. The following subsections test each claim against the measured data and state explicitly what the data can and cannot conclude. The four claims concern the linearity of the capacity in the ring weight, the structural form of the holding capacity as a function of the geometric coefficient Γ , the trends with the individual geometric parameters, and the independence of the capacity from the contact area. A fifth comparison, between the polyamide and steel braids, is included as a check on the uniform-helix assumption underlying the model.

6.2.1 Ring-Weight Linearity

The first claim is that, at fixed geometry and therefore fixed Γ , the holding capacity scales linearly with the ring weight W_{ring} . Both the exponential self-forcing model of Equation (5.1) and the non-amplifying model of Equation (5.2) predict this linearity, since at fixed geometry the factor multiplying W_{ring} is constant regardless of which structural form governs the mechanism. The claim is therefore independent of the friction coefficient and serves as a baseline check of the model structure before the structural form itself is examined. Group A

examines this claim by adding masses of approximately 400 g and 800 g to the bottom ring, which weighs 100 g at the baseline configuration, while keeping the braid geometry fixed.

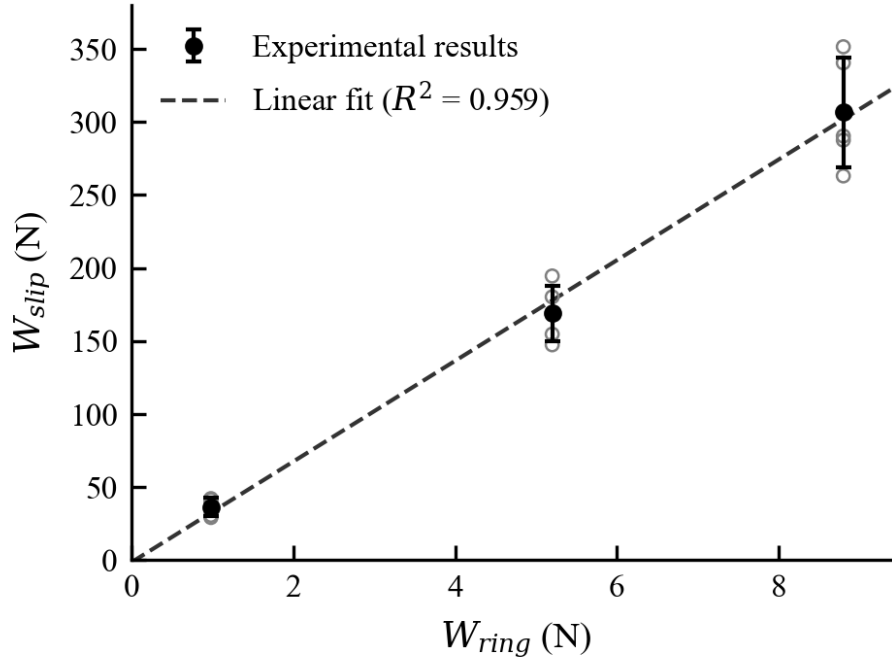


Figure 6.1: Slip load against bottom ring weight, tested at the baseline geometry ($r_c=30$ mm, $N=3$, $H=400$ mm, $n=4$) and linear fit through the individual repetitions

Figure 6.1 presents the measured slip load against the ring weight for the three Group A configurations, together with a linear regression fitted by the method of ordinary least squares. The coefficient of determination reported here is the standard output of that regression, quantifying how closely the points fall on a straight line. The line is expected to pass through the origin, since a ring weight of zero corresponds to zero preload and therefore zero capacity in both model regimes, and the fitted line obtained from the unconstrained regression confirms this expectation, passing close to the origin without being forced through it. The fit yields a coefficient of determination of $R^2 = 0.959$, and the points lie close to the fitted line across the range of ring weights tested.

The result supports the claim within the tested range. The capacity increases linearly with the ring weight, confirming that the preload sets the holding capacity in the manner the model assumes. This is a necessary condition for the model structure, and its confirmation strengthens the basis for the self-forcing test that follows. The evidence remains limited to laboratory scale and to the narrow range of ring weights achievable with the prototype. It is sufficient to support the modelling assumption and to inform an early design decision, but confirmation at larger loads is required before the linearity can be relied upon at full scale.

6.2.2 Structural Form of Holding Capacity

The second and central claim concerns the form of the relationship between the holding capacity and the geometric coefficient Γ . The non-amplifying model predicts a capacity linear in Γ , through $W_{slip} = \mu\Gamma W_{ring}$, whereas the self-forcing model predicts a capacity that grows exponentially in Γ , through $W_{slip} = W_{ring}(e^{\mu\Gamma} - 1)$. Distinguishing between these two forms is the primary objective of the campaign. Exact agreement with the exponential prediction is neither expected nor claimed, since this would require knowledge of the true friction coefficient and of the underlying contact mechanics. What can be established, independently of the friction coefficient, is whether the measured capacity follows the linear or the exponential form more closely. The claim is tested by varying Γ over a wide range, achieved through independent variation of N , H , and r_c , as established

in the test matrix of the previous section, and by comparing the measured capacity against both models, each evaluated over the friction coefficient range of $\mu \in [0.10, 0.16]$ as established from the relevant literature.

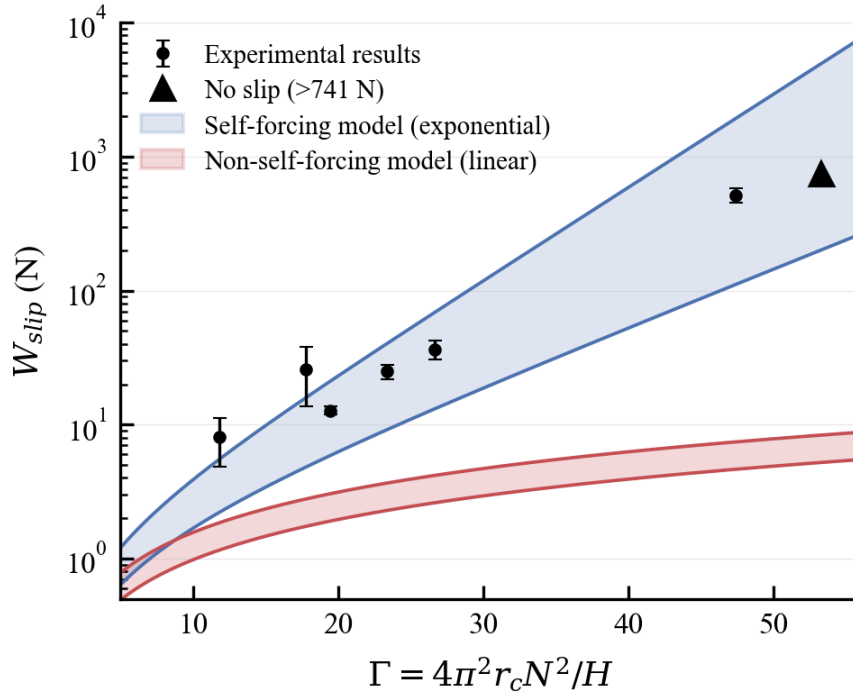


Figure 6.2: Measured slip load against the geometric coefficient Γ , compared with the self-forcing model (exponential, Equation (5.1)) and the non-self-forcing model (linear, Equation (5.2)), each evaluated over the friction coefficient range $\mu \in [0.10, 0.16]$

Figure 6.2 presents the measured slip loads against Γ , together with the linear and exponential model ranges. The measured holding capacities depart from the linear band and follow the exponential form, rising more steeply with Γ than the linear relation permits within the friction range considered. The $H = 200$ mm configuration, which sustained the maximum applied load of 741 N without slipping, supports this distinction further. At its geometric coefficient of $\Gamma \approx 53.3$, the linear model predicts slip at between approximately 5 and 9 N across the friction range, so the measured result exceeds this prediction by close to two orders of magnitude. This conclusion holds regardless of the true friction coefficient within the assumed range.

This result addresses the central demand in structural form. The holding capacity is shown to grow with the geometric amplification in the exponential manner characteristic of the self-forcing mechanism, rather than the linear manner of ordinary preload friction, indicating that the gripper carries the load through the self-forcing behaviour the model predicts. The exact form cannot be proven by this result, however. The friction coefficient remains unknown, the contact mechanics are idealised through the Coulomb assumption, and the helix is assumed uniform, so the precise functional form and its absolute magnitude cannot be confirmed from these data. A more detailed model of the contact mechanics and a more controlled experimental method would be required to establish the exact form. What the data support is that the capacity amplifies with Γ rather than scaling linearly, showing that the self-forcing mechanism develops within the tested range.

6.2.3 Geometric Trends

The third claim concerns the trends with the individual geometric parameters. The objective is not to reproduce these trends exactly, which the unknown friction coefficient does not allow, but to establish, for each of N , H , and r_c , whether the measured capacity moves in the direction and to the degree the model predicts when that parameter is varied in isolation. Figure 6.3 presents the three sweeps together, plotted against the self-forcing model of Equation (5.1) evaluated over the established friction coefficient range.

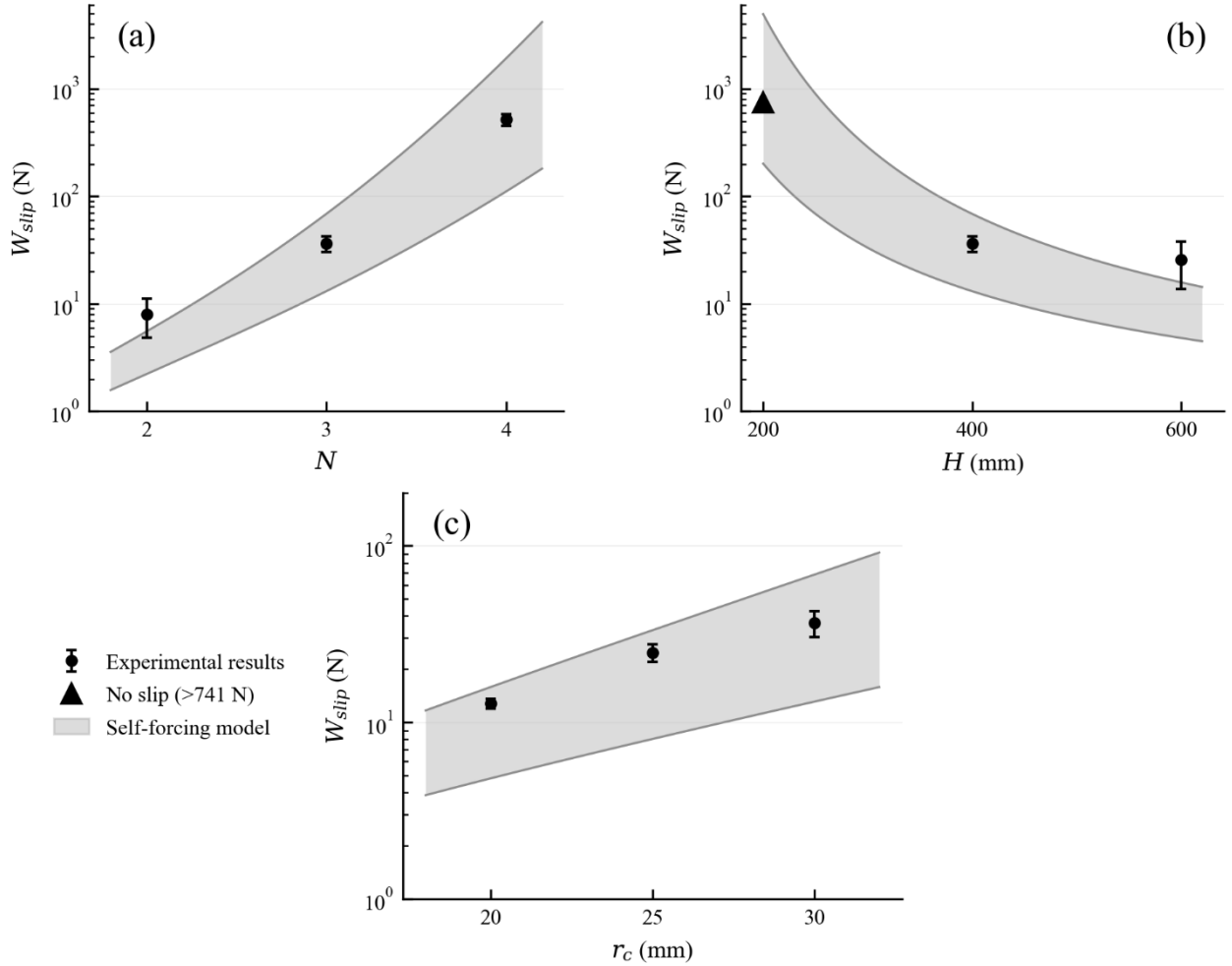


Figure 6.3: Measured slip load against each geometric parameter varied in isolation, compared with the self-forcing model evaluated over the assumed friction range. (a) Number of turns N . (b) Gripping length H . (c) Cylinder radius r_c .

As seen in Figure 6.3 (a) for the number of turns, the model predicts that Γ , and therefore the exponent, scale with N^2 . The measured capacity increases sharply between the braids at $N = 3$ and $N = 4$, consistent with the N^2 dependence, with both points falling within the model range. The $N = 2$ configuration deviates upward from the trend and shows greater scatter than the other two. A hypothesis can be proposed for this behaviour, attributed to the sparse braid formed at low N , in which fewer helical wraps are distributed over the same gripping length, leaving larger uncovered gaps between helices and allowing the individual fibres greater freedom to shift out of a uniform helical path under load. This is examined in greater detail in the material comparison Section 6.2.5, where the comparison can also be made to the stiffer steel wire rope braid.

For the gripping length, as seen in Figure 6.3 (b), the model predicts an inverse dependence, with the exponent scaling as $1/H$. The measured capacity decreases as H increases, in the predicted direction, though a definitive statement on the exponent cannot be made, as with the number of turns. The $H = 200$ mm configuration did not reach slip, while at $H = 600$ mm, greater scatter is again visible. The hypothesis proposed for the number of turns extends here, as the same number of helices is now formed over a greater distance, again producing a sparse braid.

For the cylinder radius in Figure 6.3 (c), the model predicts that the exponent scales with r_c . The measured capacity increases with r_c in the predicted direction, with all three points falling within the model range, though the points do not track the curvature of the model range closely. A possible explanation for this is a geometric inconsistency. When the radius is reduced while the rings and their connection points are held fixed, the braid

forms an hourglass-like pattern, so that the length over which it actually contacts the pipe is shorter than the ring-to-ring distance that was set as H . Figure 6.4 illustrates this difference in the developed braid form for the $r_c = 30$ mm and $r_c = 20$ mm configurations.

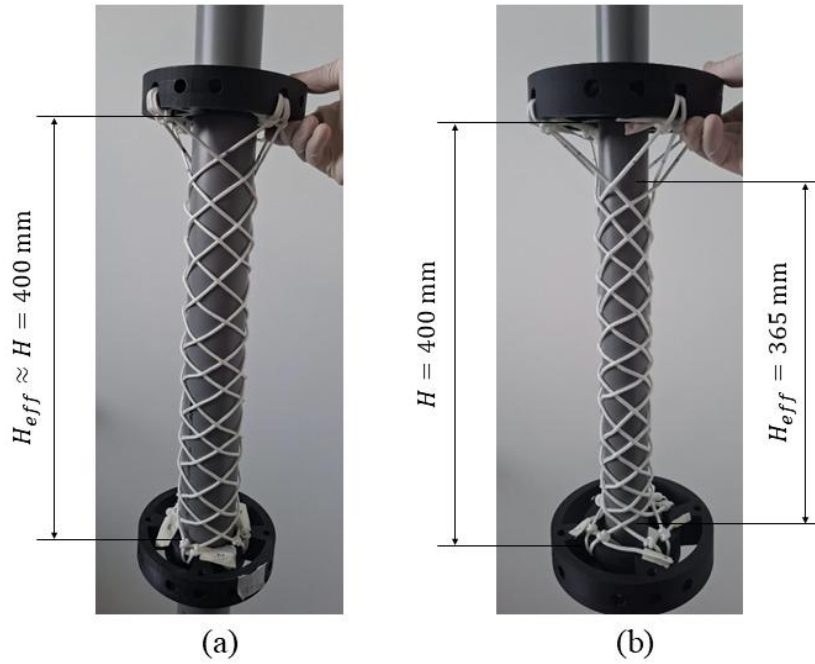


Figure 6.4: Necking pattern of the braid formed at reduced cylinder radius, with the ring-to-ring distance set to $H=400$ mm in both configurations. (a) $r_c=30$ mm. (b) $r_c=20$ mm

This hourglass-like form, or necking pattern, was presented and modelled by Lu et al. (2024) [25], which the present model neglects through its uniform-helix and uniform-deformation assumptions and should be investigated for the gripper in future work. This is particularly relevant given that the effective gripping length has a considerable impact on the final holding capacity through the $1/H$ dependence seen in Figure 6.3 (b). To account for this in the present work, the effective contact length H_{eff} was measured directly for the reduced-radius configurations, giving 365 mm and 380 mm at the 20 mm and 25 mm cylinder radii respectively, against the full 400 mm at baseline. This correction is not large enough to visibly reshape the trend, but it is not negligible either, and it is carried into the friction coefficient identification, where its effect becomes clearer. For consistency, all friction calculations in the remainder of this chapter that involve the radius configurations are evaluated using these measured effective lengths rather than the nominal gripping length. As with the previous sweeps, the radius trend is supported in direction, but no stronger claim can be made from the sweep alone. The larger-radius configuration is also hypothesised to form a sparser braid, and the increased scatter observed as the radius increases is consistent with the same sparse-braid effect proposed above.

6.2.4 Area Independence

The fourth claim follows from the Coulomb idealisation, under which the holding capacity is independent of the true contact area. A change in the number of fibre sets n at fixed geometry and fixed Γ should therefore not alter the capacity. Group E tests this by varying n while holding all other parameters at their baseline values.

Figure 6.5 presents the measured slip load against the number of rope sets. The capacity is not constant, and it increases as the number of fibre sets increases. This departure from the prediction cannot be attributed with certainty to a failure of the area-independence assumption, since a further effect the model does not capture offers a plausible alternative explanation, namely the self-weight of the rope. Both possibilities remain open and warrant further investigation.

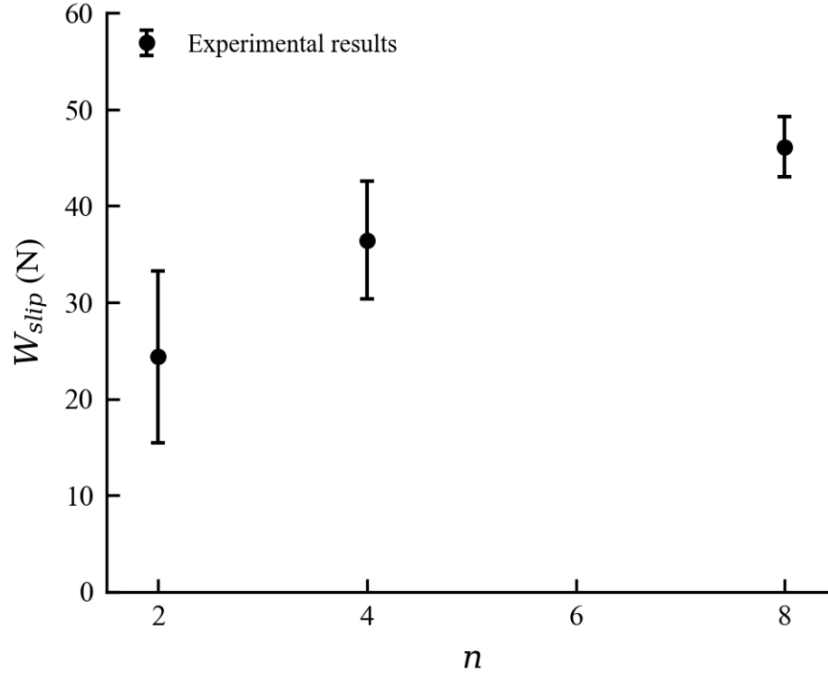


Figure 6.5: Measured slip load against the number of fibre sets n , giving $2n$ fibres in total

The self-weight explanation follows from a limitation already present in the analytical model, which neglects the weight of the braid relative to the ring. This approximation is reasonable at full scale, where the ring is substantially heavier than the braid, but becomes questionable at prototype scale, where the ring weighs only 100 g. As the number of fibre sets increases, the added mass of the additional fibres contributes directly to the axial pull on the braid, in the same manner as an increase in ring weight. Group A established how sensitive the measured capacity is to changes of this kind, with additions of 400 g and 800 g to a 100 g ring producing substantial increases in slip load, indicating that even modest additional mass at the ring measurably affects the result. This raises the possibility that the added mass of the extra fibre sets in Group E has a similar effect on the axial preload.

Confirming this explanation is not possible with the present model, since it does not carry the braid self-weight as a term, and would require a refined formulation that includes this weight explicitly, proposed as future work. The sparser configurations within this sweep, at the lower values of n , also exhibit greater scatter than the denser ones. This is consistent with the sparse-braid effect observed across several of the geometric sweeps, addressed in more detail through the material comparison in the following section.

6.2.5 Material Comparison

Group F substitutes the polyamide rope with a steel wire braid of otherwise identical geometry, and therefore identical Γ . The comparison is treated as a check on repeatability and on the uniform-helix assumption rather than on absolute capacity, since the friction pair and the ratio of braid weight to ring weight both differ between the two materials, so the absolute holding capacities are not directly comparable.

Figure 6.6 presents the measured slip load of the braids with the different materials at the shared geometry. The clearest difference is in the scatter, which is substantially smaller for the steel braid. A plausible explanation is that the greater bending stiffness of the steel wire allows it to hold its helical form more consistently, satisfying the uniform-helix assumption of the model more closely than the compliant polyamide braid.

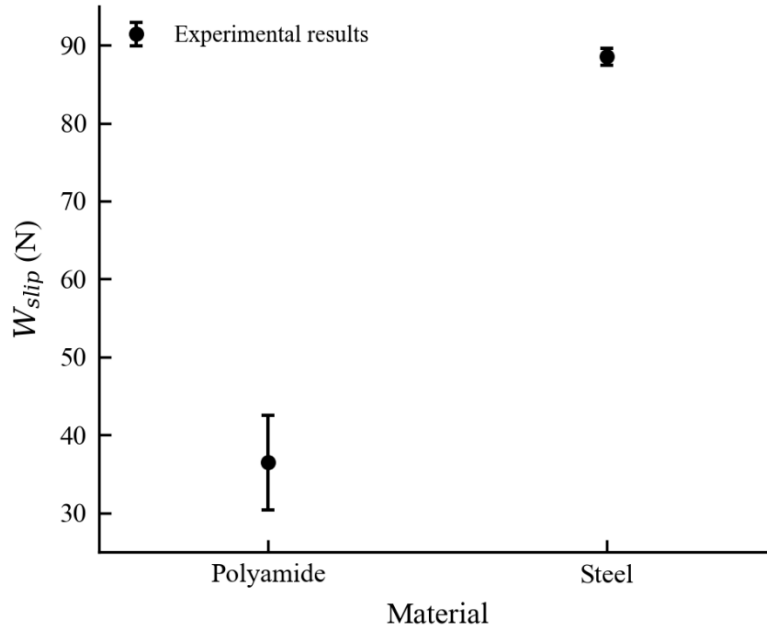


Figure 6.6: Measured slip load for the polyamide and steel wire rope braids at shared geometry

This observation may also offer an explanation for the elevated scatter noted across the sparse polyamide configurations in the preceding subsections. Where the braid is sparse, at low N , high H , high r_c , or low n , the helices do not form uniformly, as shown in Figure 6.7 (a).

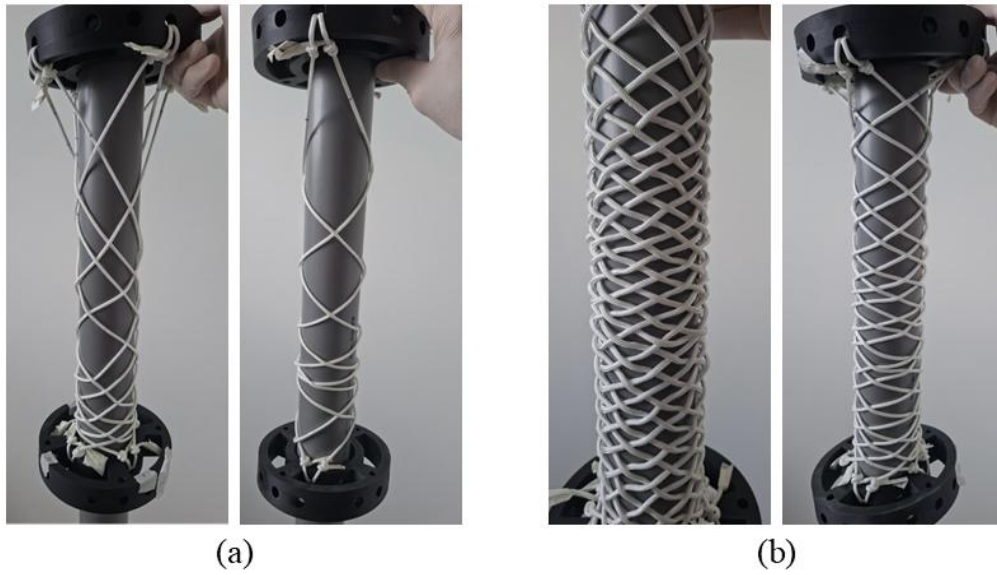


Figure 6.7: Helix inconsistencies at the braid. (a) Sparse braid examples. (b) Dense braid examples

These inconsistencies are assumed to create local regions of higher and lower grip along the braid, producing scatter in the measured capacity, whereas a denser and more uniform braid, as shown in Figure 6.7 (b), produces fewer such inconsistencies and therefore more repeatable results with a lower coefficient of variation. The steel braid, holding its form through its bending stiffness, does not develop these inconsistencies and shows the lowest scatter of any configuration tested. The coefficients of variation of all configurations are compared in Figure 6.8. It is emphasised that this reasoning is a hypothesis, and that a more accurate model would be required to verify the link between helix uniformity and scatter.

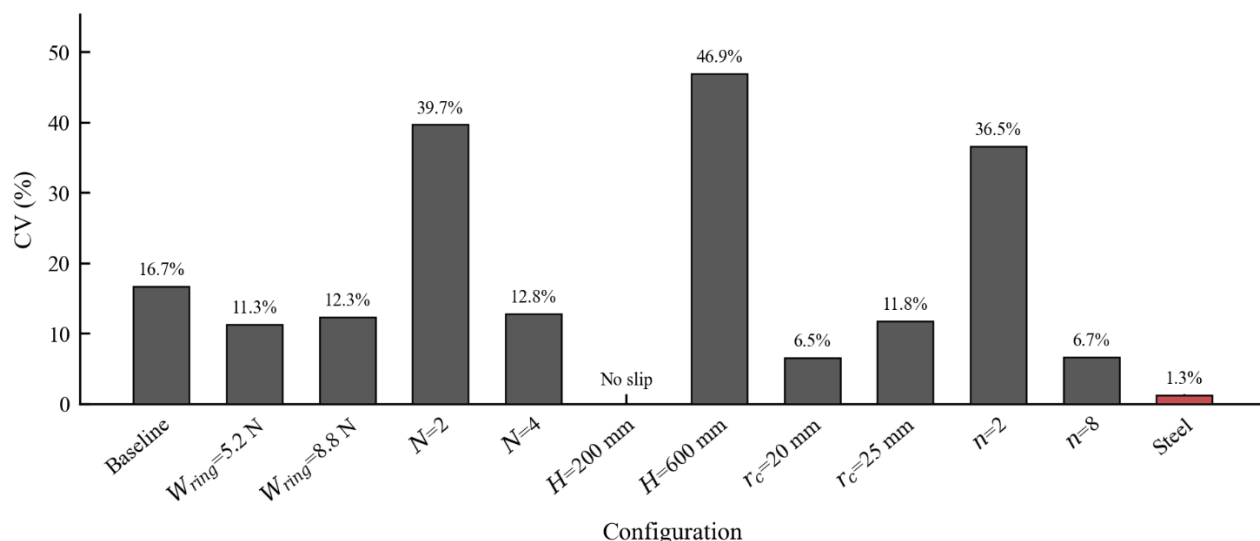


Figure 6.8: Coefficient of variation CV for all tested configurations

Beyond repeatability, the steel braid showed a practical advantage during handling. Owing to its bending stiffness, it retained its form when removed from the pipe, whereas the compliant polyamide braid bunched together and lost its shape, as shown in Figure 6.9. For the same reason, the steel braid was easier to deploy over the pipe and held a more consistent open state geometry.

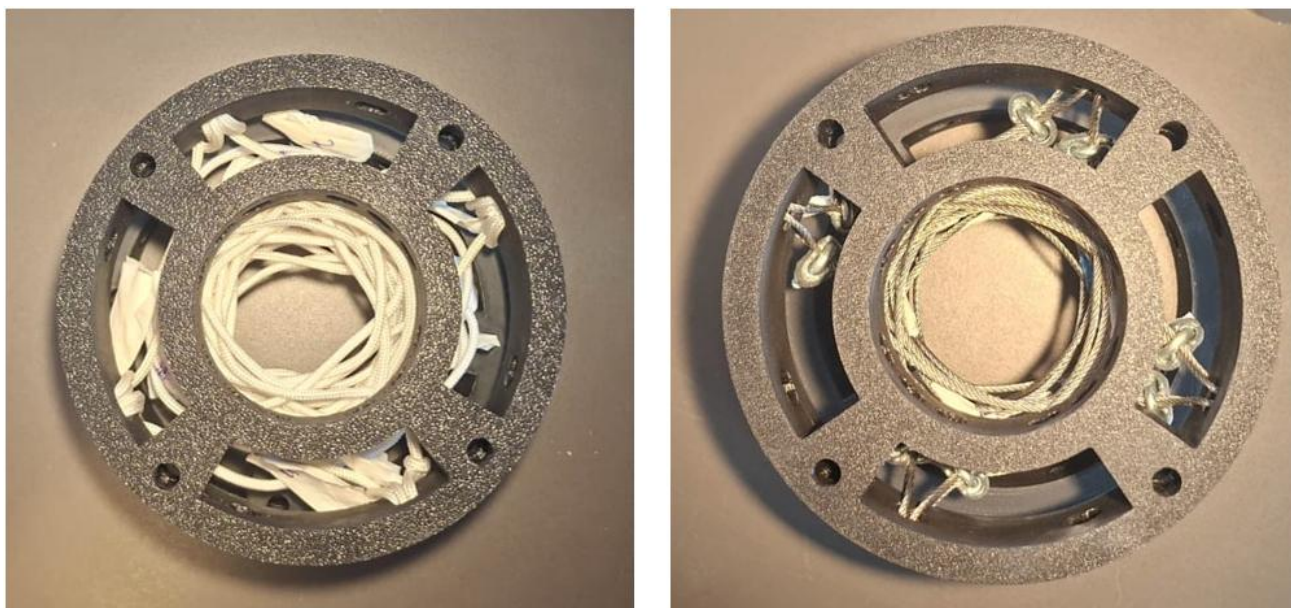


Figure 6.9: The two braids after removal, with the polyamide braid bunched up and the steel wire braid retaining its form better

This represents a definitive advantage for a full-scale design, and in particular for offshore deployment, where controlled and accurate positioning of the gripper is essential. It should be noted, however, that in both materials the braid must be prevented from losing its form during deployment and must maintain a constant open state radius with sufficient clearance over the conductor. These observations inform the material selection and deployment discussion of the full-scale design in Chapter 7. Finally, the greater self-weight of the steel braid also contributes to the axial pull, producing a tighter grip, a further consideration carried into the discussion.

6.3 Friction Coefficient Identification

With the structural form and the geometric trends established, a representative friction coefficient can now be identified from the measured data. Inverting the slip load relation of Equation (5.1) expresses the coefficient in terms of the measured slip load and the known braid geometry through the geometric coefficient Γ :

$$\mu = \frac{1}{\Gamma} \ln \left(\frac{W_{slip}}{W_{ring}} + 1 \right) \quad (6.4)$$

The identification is applied only to a calibration subset of well-formed braids, excluding the configurations whose deviations were attributed to effects the model does not capture. The $H = 200$ mm configuration is excluded first, as its slip was never captured. The sparse $H = 600$ mm and $N = 2$ configurations are excluded for their non-uniform helices, and the $n = 2$ and $n = 8$ configurations are excluded for the rope self-weight effect that the model does not represent. The steel braid is treated separately as a distinct friction pair, while the radius configurations are retained and evaluated with their measured effective gripping lengths. Table 6.2 presents the estimated friction coefficient for the resulting set of consistent configurations.

Table 6.2: Estimated friction coefficient across the consistent measurement configurations

Configuration	Γ	$\bar{W}_{slip}$ [N]	μ
Baseline	26.6	36.5	0.137
$W_{ring} = 5.20$ N	26.6	168.9	0.132
$W_{ring} = 8.80$ N	26.6	306.5	0.134
$N = 4$	47.4	515.9	0.132
$r_c = 20$ mm ($H_{eff} = 365$ mm)	19.5	12.8	0.136
$r_c = 25$ mm ($H_{eff} = 380$ mm)	23.4	24.9	0.140

Across these consistent configurations, the mean coefficient remains close to constant at $\mu = 0.135$, with a coefficient of variation of approximately 2.3%. This low variation is the central result of the identification, as it shows that a single coefficient reproduces the measured capacity across configurations of widely differing geometry. The steel braid, identified separately, gives a mean of $\mu = 0.169$.

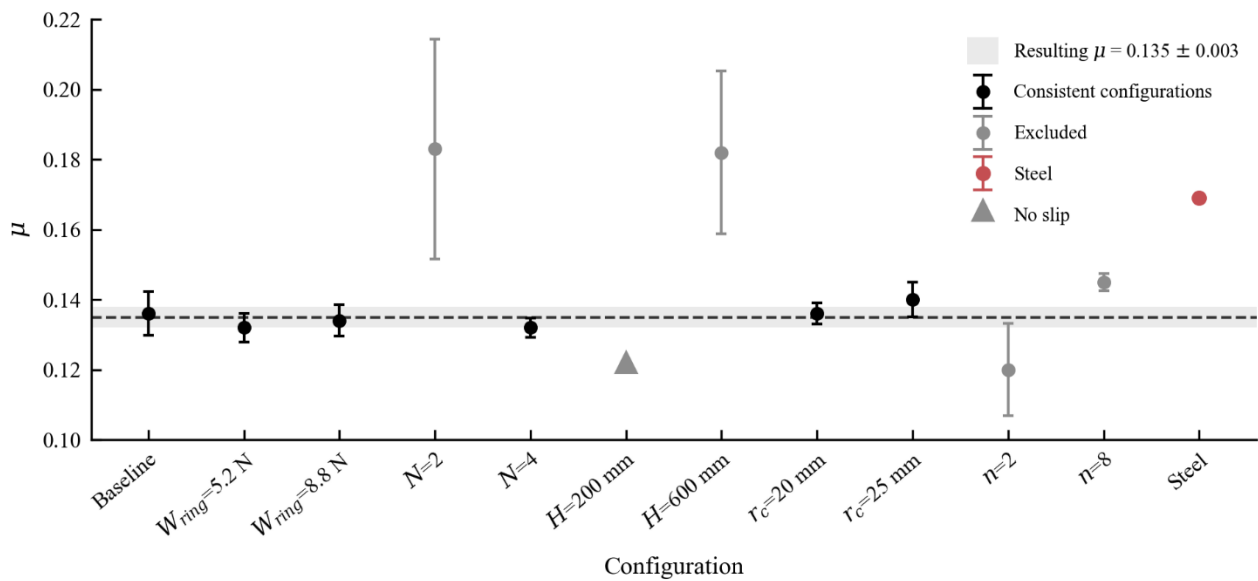


Figure 6.10: Estimated friction coefficient for all tested configurations, with the resulting mean computed from the consistent configurations

Figure 6.10 presents the estimated friction coefficients across all configurations, together with the mean and the standard deviation estimated from the consistent configurations alone. The figure shows the difference in consistency and scatter between the sparse and dense braids, as well as the systematic rise in the estimated coefficient with the number of fibre sets, which reflects the increase in holding capacity due to the increase of the braid's self-weight.

Finally, this consistency across the included configurations is a positive indicator that the Coulomb idealisation and the assumed contact mechanics hold within the tested range, since a single coefficient reconciles the measured capacity across every well-formed configuration. This should not be taken as definitive proof, however, as the number of configurations is small and the identification rests on the same simplified contact model.

6.4 Payload-to-Self-Weight Ratio

A metric commonly reported for soft grippers is the payload-to-self-weight ratio, expressing the holding capacity relative to the combined weight of the two rings and the braid. It provides a simple measure of structural efficiency and situates the present concept against the wider soft gripping literature. The ratio cannot be carried directly to the full-scale case, since the self-weight grows with the cube of the characteristic length while the holding capacity does not increase in the same proportion, as examined in Section 2.3. It remains a useful indication nonetheless, and is of particular interest for this concept, which is mechanically simple and passively actuated and therefore more attractive to scale than the more complex grippers found in the literature.

The self-weight of the gripper varies only modestly across the polyamide braids, so the ratio is governed almost entirely by the holding capacities. Three representative configurations are therefore sufficient, namely the polyamide baseline, the steel braid at the same geometry, and the no slip $H = 200$ mm configuration, as presented in Table 6.3.

Table 6.3: Payload-to-self-weight ratio for representative configurations

Configuration	$W_{gripper}$ [N]	$\bar{W}_{slip}$ [N]	R_{pw}
Baseline	2.23	36.5	16
Steel wire (same geometry)	3.28	88.5	27
$H = 200$ mm (no slip)	2.20	>741	>337

The ratio ranges from approximately 16:1 at the baseline to at least 337:1 for the most efficient configuration. At identical geometry the steel braid reaches the higher ratio of approximately 27:1 despite its greater self-weight, as its higher friction and the additional axial pull from its own weight raise the capacity by more than the added weight. The highest recorded value of at least 337:1 is a lower bound only, as the true slip point of the $H = 200$ mm configuration could not be reached before the experimental setup reached its limit.

These ratios confirm that a substantial holding capacity is achieved at a small fraction of the gripper self-weight, an attractive result that indicates the concept scales favourably towards a comparatively lightweight solution capable of lifting loads of several tonnes. This efficiency supports the requirement for a lightweight gripper (R.5.3) and motivates the lightweight full-scale design.

6.5 Summary and Discussion

The experiments examined whether the self-forcing mechanism predicted in Chapter 4 appears in a real prototype, and whether the geometric scaling laws hold. On the points that could be tested, the results are consistent with the model. The holding capacity rose linearly with the bottom ring weight at fixed geometry, which confirms that the preload sets the baseline capacity. The capacity then increased sharply with the

geometric coefficient Γ , exceeding what preload friction alone would produce, which is the behaviour the self-forcing mechanism produces. The geometric trends also followed the predicted directions, with the capacity rising as the number of turns or the radius increased and falling as the gripping length increased.

The friction coefficient found from the consistent configurations remained close to $\mu = 0.135$ and fell within the range reported in the literature. A single value fitting configurations of quite different geometry supports the Coulomb idealisation within the tested range, although it does not constitute proof. The steel braid gave more repeatable results than the polyamide braid and held its form better during handling and deployment, which suggests the uniform-helix assumption is met more closely by a stiffer braid. The payload-to-self-weight ratio ranged from approximately 16:1 at the baseline configuration tested to more than 337:1 in the most efficient case. Taken together, the results show that the mechanism develops at laboratory scale for a heavy enough load and that the capacity grows with the braid geometry in the direction the model predicts. They do not establish the exact functional form or the true holding capacity, since both depend on the real contact mechanics, which are not known.

Two deviations from the model appeared in the results. The first was observed at the sparser braids, where the measured capacity did not follow the prediction as closely and the scatter was large. This is likely caused by the helices forming imperfectly in these configurations, due to contact mechanics that the model does not capture. The second is that the capacity also rose with the number of rope sets, which the model does not predict. This is attributed to the self-weight of the additional fibres, which the model neglects relative to the much heavier ring. Both indicate limitations of the current model rather than in the mechanism itself, and both can be addressed in future work by adding more detailed contact mechanics and the rope self-weight to the model.

The results are subject to a number of limitations. The test procedure was manual and therefore operator-dependent, and the dynamometer offered limited precision at the smallest loads. The most significant limitation concerns the identification of slip. Slip was determined visually, as the first observable movement between the braid and the pipe, which is difficult to distinguish from the small settling and flexing of the braid as it conforms onto the surface under load. A load cell recording the full loading history would allow slip to be identified from the load trace rather than by eye, and this improvement is recommended in Chapter 8. The maximum load that could be reached was governed by the manual holding capacity of the operator and by the strength of the printed rings, rather than by the gripper itself, so the $H = 200$ mm configuration never reached slip and was recorded as a lower bound at 741 N.

A further observation is reported for completeness. Initial tests returned unusually high capacities, which was traced to dust from the granular material used as added weight reaching the contact interface and raising the effective friction. The pipes were cleaned, the affected braids were remade, and the tests were repeated to obtain the results presented here. This behaviour illustrates the sensitivity of the mechanism to the condition of the contact surface, and it supports the argument that the rough surface of an aged conductor is more likely to raise the effective friction than to lower it.

The remaining limitations concern scale and scope. Only two sub-configurations were tested per group beyond the baseline, which is sufficient to establish the direction of each trend but not its exact form. The prototype also remains far from full-scale conditions, in particular the friction pair, marine growth and corrosion, submersion, and the magnitude of the load, so the friction coefficient identified here functions as a check on the model rather than a value to be transferred to full scale. Testing at larger scale is therefore required to determine whether the same behaviour transfers to full-scale conditions. The braiding was also performed by hand, which is slow and, in the case of the steel wire, difficult to control, and this constrained the number of configurations that could be produced. These points are carried forward as future work in Chapter 8.

In summary, the campaign supports the aspects of the model that can be examined without knowledge of the friction coefficient, and it does not fully validate the exact functional form or the absolute capacity. What it does establish is that the self-forcing mechanism physically develops for a sufficiently heavy load, that the capacity grows with the braid geometry in the direction the model predicts, and that a single friction coefficient fits the consistent configurations within the tested range. This is sufficient to support an informed design decision and to proceed to the full-scale design in Chapter 7. In terms of the requirements, the demand that the gripper carry the full lifting load (R.2.2) and maintain grip under that load (R.2.3) is now supported within the tested range. SRQ3 is thereby answered, and the feasibility of the concept is demonstrated at laboratory scale.

7 Full-Scale Design and Recommendations

The preceding chapters established the analytical model of the braided sleeve gripper and found support for its central prediction, that the holding capacity increases with the geometric coefficient Γ through the self-forcing mechanism, within the range tested on the prototype. This chapter applies the model to the North Cormorant reference case in order to produce an early-phase full-scale design and thereby answer SRQ2. The design is developed under deliberately conservative assumptions and against the offshore lifting safety factors, and each design choice is presented alongside the requirement it addresses, following the requirements list established in Chapter 3. The chapter first defines the design inputs and their justification, then presents a feasibility design space and its sensitivity to friction, selects a design point, outlines an exemplary operational procedure, and finally identifies the design elements to be addressed in future design iterations.

7.1 Design Inputs and Selections

The full-scale design is developed for the North Cormorant conductor, whose properties are established in Appendix C. The outer casing has a radius of $r_c = 381$ mm, and the governing load case corresponds to the worst-case fully grouted 29.75 m section. Following the lifting-weight estimation of Appendix C, in which the net section weight is amplified by a contingency factor of 1.10 and a dynamic amplification factor of 1.50 to account for unknown additional weight and for the dynamic effects of the lift, the design lifting weight per section is 64.6 t, corresponding to a conductor weight of $W_c = 634$ kN.

A heavy bottom ring supplies the passive axial force from which the self-forcing capacity is generated. The holding capacity scales directly with the weight of the bottom ring through the self-forcing relation derived in Chapter 4, $W_{c,max} = W_{ring}(e^\beta - 1)$. A heavier ring therefore raises the baseline capacity, at the cost of adding to the gripper self-weight and to the mass to be lifted. A ring mass of 10 t is adopted, giving an effective ring weight of $W_{eff} \approx W_{ring} = 98$ kN. This value is heavy enough to render the restoring force of the braid negligible, so that the effective weight reduces to the ring weight as established in Sections 4.2 and 4.5, yet it remains small relative to the 64.6 t conductor section. It is fixed here as a first estimate for the early-phase design. The ring provides this force without any active or continuous actuation, which supplies the baseline capacity (R.2.2) and sustains the fail-safe grip identified at concept selection (R.3.2). The ring outer radius is bounded by the inter-conductor spacing of the platform, which is 1528 mm between rows and 1678 mm between columns measured from the outer conductor surfaces as per Appendix C.

A radial clearance of $c = 0.1$ m is adopted between the conductor surface and the braid in its open deployment state, large enough for the gripper to be lowered over the conductor without snagging on marine growth, corrosion, or positioning misalignment, and small enough that the open-state geometry remains achievable. This value provides a practical allowance for marine growth and subsea positioning tolerance, and it is comfortably accommodated by the available spacing. It can be refined at later stages of the design, once further testing under more representative conditions has been conducted, and a more detailed ring geometry is also left to the future design iterations.

The braid is specified in steel wire rope, following the material comparison of Section 6.2.5, in which the steel braid retained its helical form, eased deployment and release, and produced more repeatable results than the compliant polyamide braid, with its greater self-weight further assisting the grip. Steel wire rope is additionally a marine-grade material established in the offshore industry and resistant to corrosion, fatigue, and ultraviolet degradation (R.5.1, R.5.2), and its favourable strength-to-weight characteristics contribute to a lightweight gripper (R.5.3). A rope diameter of $d = 28$ mm is selected, which has a minimum breaking load of 438 kN according to supplier data [70]. The ropes are arranged in $n = 8$ rope sets, corresponding to 16 individual ropes.

These values are not optimised but are chosen as an initial feasible combination that carries the top-ring tension with the required strength margin. They therefore present a workable starting point for the early-phase design and should be refined in a more detailed design. The material selection is likewise a preferred indication rather than a proven requirement, since the material comparison was qualitative and conducted at prototype scale.

The design friction coefficient is set conservatively to $\mu = 0.10$ for the steel wire rope coming in contact with the steel conductor casing. In reality the effective friction is likely to be higher, since an aged conductor presents welded patches, corrosion, marine growth, and damaged regions, the same features that make it difficult to remove by traditional methods. Additionally, some degree of interlocking or form closure is also expected to develop at these surfaces. However, as the true coefficient is unknown, the design cannot rely on this benefit in advance, and therefore the sensitivity of the design to the friction coefficient is examined in Section 7.3.

The gripper is dimensioned against two safety factors, both set to 5, which represent a standard value applied to early-stage designs in offshore lifting standards. A lift safety factor $SF_{lift} = 5$ requires the holding capacity to reach five times the design lifting weight, and a breaking safety factor $SF_{break} = 5$ limits the maximum rope tension to one fifth of the minimum breaking load (R.3.1, R.3.3). Table 7.1 summarises the design inputs.

Table 7.1: Design inputs for the North Cormorant full-scale gripper

Quantity	Symbol	Value	Unit
Conductor outer casing radius	r_c	381	mm
Design lifting weight per section	W_c	634	kN
Bottom ring mass	m_{ring}	10	t
Bottom ring weight	W_{ring}	98	kN
Radial deployment clearance	c	0.1	m
Steel wire rope diameter	d	28	mm
Minimum breaking load	MBL	438	kN
Number of rope sets	n	8	-
Design friction coefficient	μ	0.10	-
Lift safety factor	SF_{lift}	5	-
Breaking safety factor	SF_{break}	5	-

With these inputs fixed, the only remaining open parameters are the number of turns N and the gripping length H . These two parameters together define the geometry of the braid and are the most important choices in the design. The following section addresses their selection and identifies the feasible region in which they may lie.

7.2 Feasibility Design Space

With the inputs fixed, the feasible design space is expressed in the plane of the two remaining control parameters, the number of turns N and the gripping length H . Two boundaries delimit the feasible region. The first follows from the lift safety factor and the self-forcing relation derived in Chapter 4:

$$W_{c,max} = W_{ring} \left(e^{\mu \frac{4\pi^2 r_c N^2}{H}} - 1 \right) \geq SF_{lift} W_c \quad (7.1)$$

The second follows from the maximum fibre tension at the top ring, also derived in Chapter 4, which is limited by the breaking safety factor:

$$F_{fibre} = \frac{W_c + W_{ring}}{2n \sin \alpha} \leq \frac{MBL}{SF_{break}} \quad (7.2)$$

The feasible region is the set of (N, H) pairs for which both conditions hold simultaneously and is presented in Figure 7.1 together with the two boundaries at safety factors of 1 and 5, and the combinations of N and H that reproduce two of the geometric coefficients tested at prototype scale, namely the baseline $\Gamma = 27$ and the most efficient configuration $\Gamma = 53$. It should be noted that every combination of N and H along a given Γ curve produces the same holding capacity under the derived theoretical model.

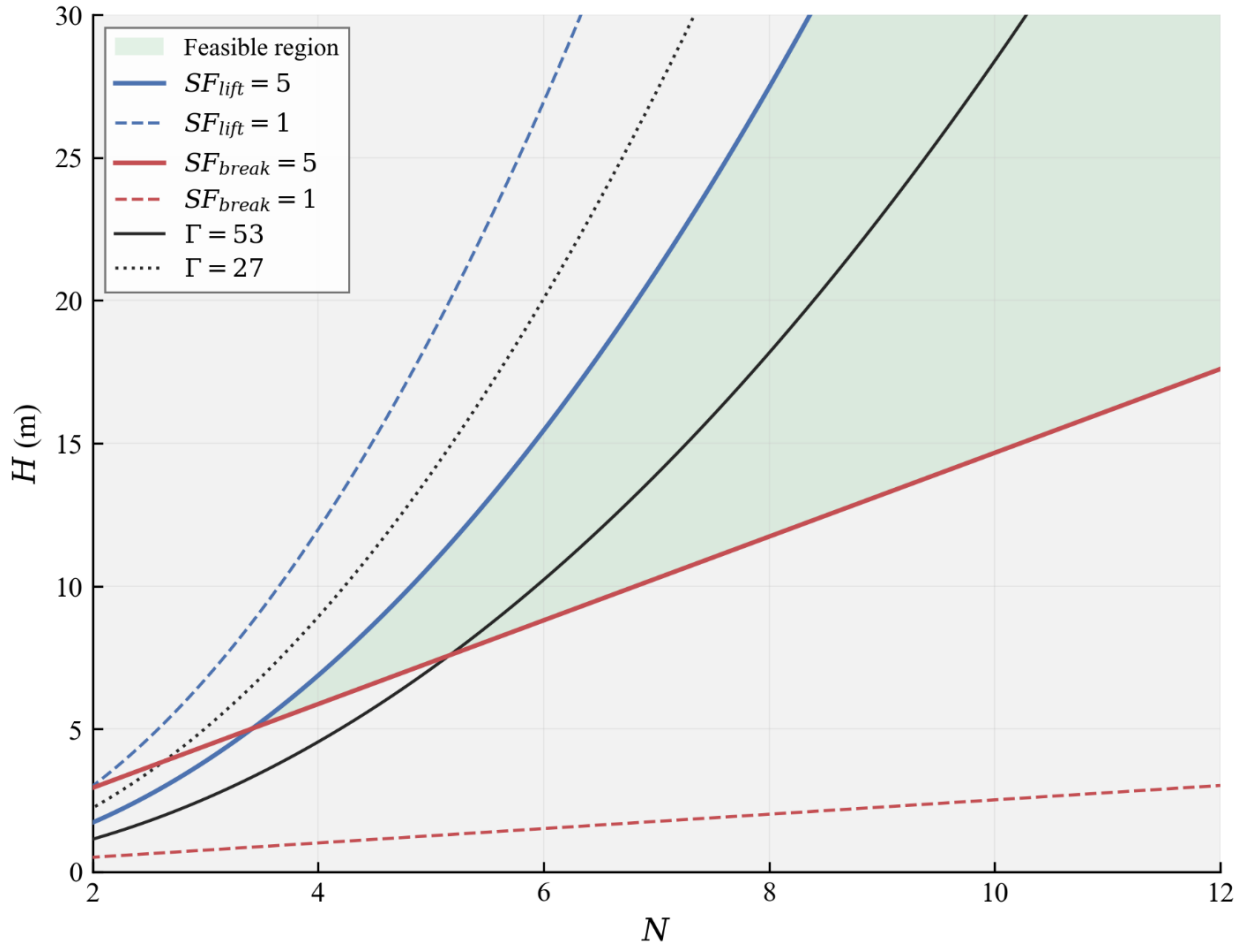


Figure 7.1: Feasible design space in the (N, H) plane, showing the slip and rope breaking safety factor boundaries at values of 5 and 1, together with the $\Gamma=27$ and $\Gamma=53$ geometric coefficient curves

The shaded feasible region opens only once the number of turns exceeds roughly four, widening steadily thereafter as further turns raise the holding capacity faster than they raise the fibre tension. A key observation is that the slip boundary, evaluated at the design safety factor of five, corresponds to a geometric coefficient of $\Gamma \approx 35$, which falls inside the prototype-tested range spanned by the baseline and most efficient prototype configurations, $\Gamma \approx 27$ to $\Gamma \approx 53$, shown by the two grey curves. The feasible region therefore rests on the range of behaviour that the experiments established, rather than on an extrapolation of the model into untested territory. The rope-break boundary forms the lower limit of the region, since a denser braid shortens the pitch, lowers the braiding angle, and raises the fibre tension beyond the rope capacity.

The position of the slip boundary depends directly on the friction coefficient through the self-forcing coefficient β , whereas the rope-break boundary depends only on the geometry and the rope strength and is therefore independent of friction. As the true full-scale friction cannot be known in advance, the extent to which the feasible region expands or contracts with the friction coefficient must be examined before a design point is selected. The following section addresses this sensitivity.

7.3 Sensitivity to Friction

The friction coefficient carries the largest uncertainty of all the design inputs, since it cannot be measured in advance for the true full-scale contact pair. Its influence on the feasible design space is therefore examined directly. Figure 7.2 presents the feasible region for three coefficients, a pessimistic $\mu = 0.05$, the conservative design value of $\mu = 0.10$, and a more likely in-service value of $\mu = 0.20$ that reflects the raised friction expected from corrosion and marine growth. The rope breaking boundary is unaffected across the three cases, since it depends only on the geometry and the rope strength, and friction therefore acts entirely through the position of the slip boundary.

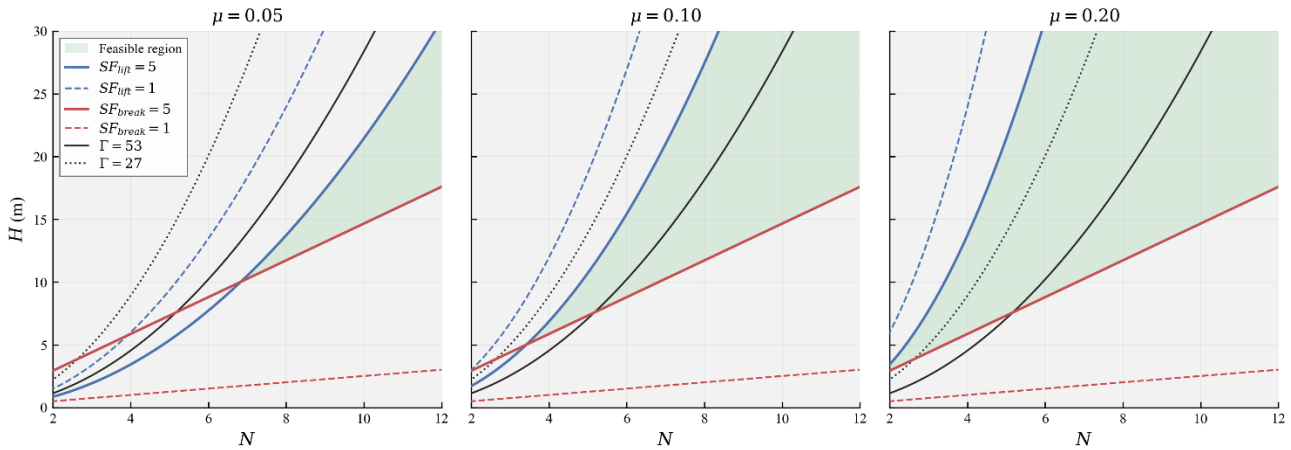


Figure 7.2: Feasible design space in the (N, H) plane for $\mu=0.05$, 0.10 , and 0.20 , showing the feasible region expanding as friction increases

The feasible region expands as the coefficient increases. At the design value of $\mu = 0.10$ the slip boundary sits at a geometric coefficient of $\Gamma \approx 35$, inside the prototype-tested range, and the region opens beyond roughly four turns. At $\mu = 0.20$ the boundary falls to $\Gamma \approx 18$ and the region opens almost immediately, whereas only in the pessimistic case of $\mu = 0.05$ does the region contract sharply, with the boundary rising to $\Gamma \approx 70$, above the tested maximum of 53. In this case a feasible design would require an extrapolation beyond the tested parameter range.

A key takeaway is that the design remains feasible across the range of coefficients realistically expected for a corroded steel contact pair, for which $\mu = 0.10$ is already a conservative estimate. The pessimistic case of $\mu = 0.05$ is included only to bound the sensitivity, and it falls below any value the aged conductor surface is likely to present. Since the condition of such a surface instead suggests a higher effective friction, the real design space is expected to be wider than the conservative case on which the design point is based.

7.4 Design Point Selection

Within the feasible region the choice of a working configuration is a trade-off rather than a single optimum, since every point that satisfies both boundaries is a valid design that serves a different engineering goal. Reducing the number of turns and the gripping length moves the design towards the slip boundary and lowers the quantity of rope and the height of the gripper, which favours material economy and a compact tool. Increasing them moves the design deeper into the feasible region and spreads the grip over a longer contact length, which favours load distribution and reduces the local contact pressure on the conductor, at the cost of more material and a taller gripper. The final choice therefore depends on what is prioritised for a given conductor and platform, whether that is material economy, distribution of the gripping load, the overall size of the gripper, or the chosen removal strategy.

A configuration of $N = 7$ turns and $H = 14$ m is selected for the reference case, placing the design on the $\Gamma \approx 53$ line of Figure 7.1. This point coincides with the most efficient configuration examined at prototype scale, so the design rests directly on validated behaviour rather than on an extrapolation of the model. At this geometry the geometric coefficient is $\Gamma \approx 52.6$ and the self-forcing coefficient is $\beta \approx 5.26$, which through the self-forcing relation of Chapter 4 gives a holding capacity of approximately 18800 kN. This corresponds to a lift safety factor of about 30 against the design lifting weight, far above the required value of five. This large margin follows directly from the exponential nature of the self-forcing mechanism, and it confirms that slip is not the constraint that governs the design. The design is instead governed by the requirement to remain within the validated range of the geometric coefficient and by the strength of the rope at the top ring. At the selected point the braiding angle is approximately 39.9° , the maximum rope tension is approximately 71.4 kN, and the breaking safety factor is approximately 6.1, which satisfies the strength requirement with margin (R.3.1, R.3.3).

From these two control parameters the remaining braid quantities follow through the geometric relations derived in Chapter 4. The total rope length required to fabricate the braid is approximately 349 m, and the total contact area between the braid and the conductor is approximately 9.07 m^2 . Relative to the gripped surface, this gives a covered area ratio of approximately 0.27, indicating that the grip is distributed across roughly a quarter of the conductor surface over the gripping length and confirming that the load is spread rather than applied as a point load (R.1.9, R.2.1). For deployment, the braid contracts to an open-state radius of $r_{open} = 481$ mm over a shortened open-state length of $H_{open} \approx 5.4$ m, a height change of $\Delta H \approx 8.6$ m between the open and closed states. The compact open-state length is comfortably transportable and supports the requirement that the gripper should fit within standard offshore transport dimensions (R.7.3).

Table 7.2: Design point and resulting quantities for the North Cormorant full-scale gripper

Quantity	Symbol	Value	Unit
Number of turns	N	7	-
Gripping length	H	14	m
Pitch	p	2.0	m
Braiding angle	α	39.9	deg
Geometric coefficient	Γ	52.6	-
Self-forcing coefficient	β	5.26	-
Holding capacity	$W_{c,max}$	18849	kN
Lift safety factor	SF_{lift}	29.7	-
Maximum rope tension	F_{fibre}	71.4	kN
Breaking safety factor	SF_{break}	6.1	-
Total rope length	S_{total}	349	m
Total contact area	A	9.07	m^2
Covered area ratio	C_A	0.27	-
Open-state radius	r_{open}	481	mm
Open-state length	H_{open}	5.4	m
Height change	ΔH	8.6	m
Open-state ratio	OSR	1.03	-
Closed-state ratio	CSR	10.7	-
Estimated braid mass	m_{braid}	1.05	t

A closer inspection of the feasibility ratios reveals a point that must be carried into the next design iteration. The closed-state ratio is comfortable at $CSR \approx 10.7$, confirming that the gripping state lies far from the slimmest attainable geometry. The open-state ratio, however, is only $OSR \approx 1.03$, meaning that the chosen deployment clearance of $c = 0.1$ m brings the open state close to the widest geometry the braid can physically reach before its fibres come into contact. The design therefore remains feasible, but with little margin on the open state, and

this indicates that the deployment clearance, the rope diameter, or the number of rope sets should be revisited to open this margin in a refined design.

Finally, the self-weight of the braid is estimated. Taking a representative linear mass of approximately 3 kg/m for the 28 mm steel wire rope, the 349 m of rope corresponds to a braid mass of approximately 1.05 t. This mass is deliberately excluded from the holding-capacity model, which neglects the braid weight relative to the ring, so its effect is conservative. In practice it adds to the axial pull on the braid in the same manner as the bottom ring and therefore increases the grip, consistent with the observation in Section 6.2.4 that the greater self-weight of braids with more ropes increased the holding capacity of the gripper. At roughly one tenth of the 10 t ring mass, the braid contributes a significant additional preload that the model does not credit, further supporting the conservative basis of the design and the requirement for a lightweight gripper (R.5.3). The design point and its resulting quantities are summarised in Table 7.2.

7.5 Exemplary Operational Procedure

The full-scale operation follows the gripping sequence derived in Section 4.2, scaled here to the North Cormorant reference conductor. An exemplary sequence is presented in Figure 7.3, rendered as a three-dimensional visualisation of the full-scale gripper on the conductor field, with each stage of the operation shown in turn.

The sequence begins with the removal of the first conductor guide frame, shown in Figure 7.3 (a). The gripper cannot pass the guide frames while gripping the conductor, so the frames must be removed beforehand. This is done in stages, first removing the uppermost frame, then cutting and removing all the conductor sections above the next frame, then removing that frame and continuing the removal downward level by level. The exact procedure must still be defined in detail, and further possibilities exist. Two frames could be removed at a time, allowing a greater section of conductor to be lifted in one operation, or several frames could be removed and the entire conductor extracted in a single lift. The latter, however, depends on both the actual holding capacity of the gripper and the structural integrity of the conductor, since gripping only a top section requires the sections below to be lifted with it, which cannot be guaranteed in every removal case.

Following the first guide frame removal, the braided sleeve gripper is lowered over the conductor in its compact open state, suspended from the lifting equipment through the bottom ring, as shown in Figure 7.3 (b). In Figure 7.3 (c), the top ring is raised while the bottom ring is lowered, which elongates the braid along the conductor and constricts it onto the surface under the effective weight of the bottom ring. A pretension is then applied at the top ring to establish the grip before cutting, the conductor is severed below the gripper, and the section is lifted clear, as shown in Figure 7.3 (d).

The section is transported to the release location and set down onto a downending frame on the vessel, as shown in Figure 7.3 (e). This is one of several possibilities for handling the conductor at release, inspired in part by the down-end frame developed by Heerema Marine Contractors. The frame remains nearly vertical while conductor sections are positioned onto it, holding up to twelve sections, after which it reclines in a controlled manner to the horizontal so that a crane can lift the entire cradle and store it on deck, handling the cut sections safely and in batches [71].

With the section held by the frame, the grip is then released and the gripper retracted clear of the conductor, as shown in Figure 7.3 (f). This cycle is repeated across multiple conductors, progressively filling the downending frame with sections. Once the frame is full, it reclines from the vertical to a horizontal position, laying the collected sections down together in a controlled manner, as shown in Figure 7.3 (g).

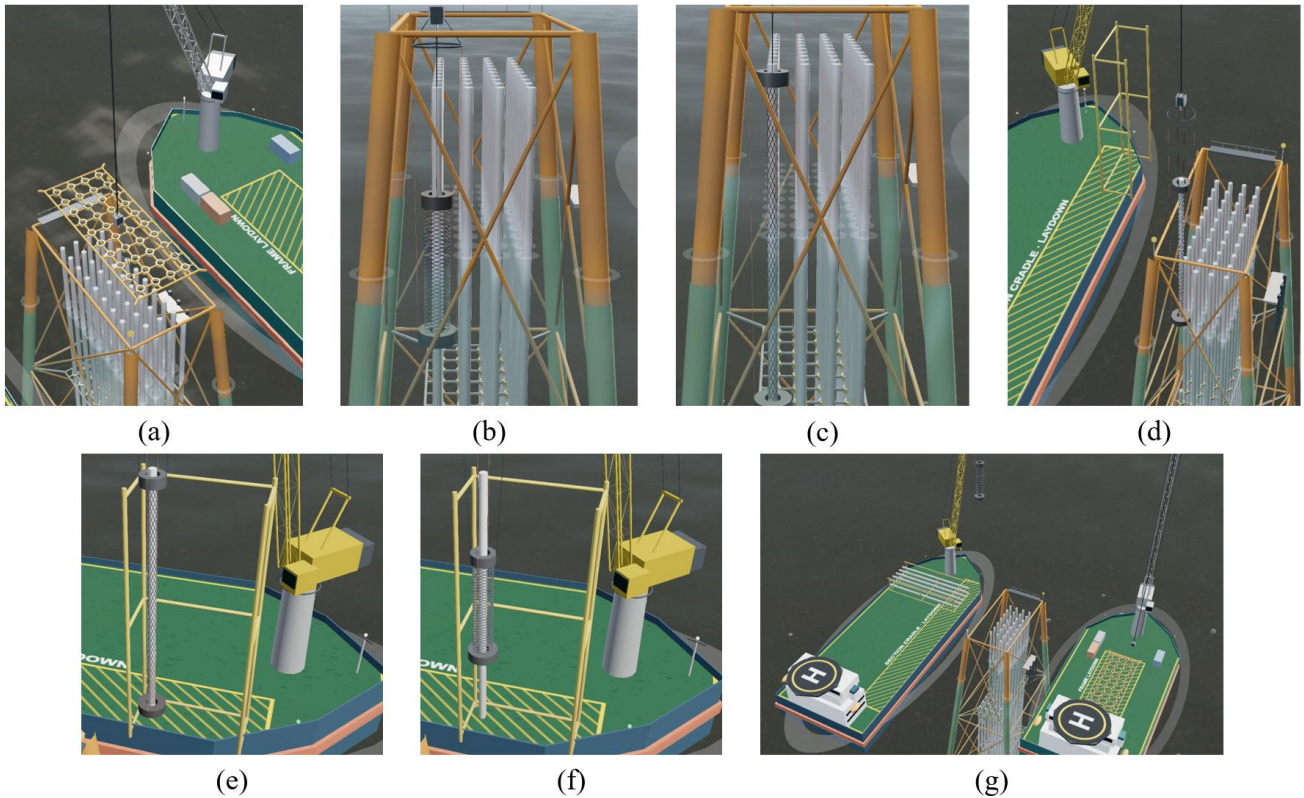


Figure 7.3: Exemplary operational sequence of the full-scale gripper on the North Cormorant conductors. (a) Removal of the first guide frame. (b) Gripper lowered over the conductor in its open state. (c) Top ring raised and bottom ring lowered, constricting the braid onto the conductor. (d) Conductor cut below the gripper and the section lifted clear. (e) Section set down onto the downending frame. (f) Grip released and gripper retracted. (g) Downending frame filled with conductor sections and reclined on the vessel

Three stages of this sequence require further treatment, as none is fully resolved by the concept in its current form. The first is the deployment of the braid onto the conductor, shown in Figure 7.3 (c), for which further possibilities exist beyond the method presented here. The bottom ring could be held fixed while the top ring is pulled upward, or the top ring held fixed while the bottom ring is released downward, rather than moving both simultaneously as proposed here. The different advantages and challenges of each method cannot be addressed within the scope of this thesis and should be investigated as part of future work. The simultaneous method proposed here is the most demanding in terms of actuation, but it offers the potential to distribute the braid more consistently along the conductor from a predetermined starting position.

The second is the release, shown in Figure 7.3 (f). Two direct routes exist to relax the grip and retract the gripper as set by the requirement list (R.2.5, R.4.2). The bottom ring can be pulled upward relative to the top ring, or the top ring can be released, both reducing the axial distance between the rings so that the braid relaxes and the radial pressure falls away. In practice the ropes may catch on the corroded and irregular conductor surface as the braid relaxes, which prevents a clean release and represents the principal difficulty of this phase. The downending frame supports this step by securing the section in its vertical orientation, so that it cannot drop or swing as the grip is relaxed. The frame may additionally carry elements that assist the release by driving the bottom ring upward, either through a fixed element against which the ring is pushed as the conductor is lowered, so that the ring is held in place and the grip relaxes as the conductor moves upward relative to it, or through an active mechanism that drives the ring upward directly.

The third is the handling of the guide frames, shown in Figure 7.3 (a). As described above, the frames must be removed ahead of the gripper, which introduces the risk that a conductor becomes free to move under environmental loading once its supporting frame is removed. This is acknowledged here as an operational consideration to be addressed alongside the full removal strategy rather than resolved within the gripper design

(R.4.1). Deployment and release are the two problems that remain open at the current development level of the gripper and both are carried forward as the central subjects of the following section.

7.6 Recommendations for Future Design Iterations

The full-scale design of the preceding sections establishes a feasible configuration for the North Cormorant reference case, but several of its elements are fixed as first estimates and a number of practical problems remain open. This section sets out the recommendations for the next design iteration, ordered by priority, and identifies the specific engineering work that would advance the concept from an early-phase design toward a deployable gripping and lifting tool.

The single most important next step is testing at a larger and more representative scale. The prototype campaign of Chapter 6 supports the model in form and scaling, but it does so at laboratory scale, with a friction pair, a braid-to-ring weight ratio, a surface condition, a helix formation, and a load magnitude that all differ from full scale. The design presented here is deliberately confined to the range of the geometric coefficient established on the prototype, with the selected point at $\Gamma \approx 52.6$ lying within the tested range, yet the absolute holding capacity and the true friction coefficient of the full-scale contact pair remain unconfirmed. Remaining within the tested range of the geometric coefficient bounds the extrapolation of the model in form but not in scale, and it does not guarantee that the self-forcing mechanism reproduces the same behaviour at full scale. Testing at a larger and more representative scale is therefore the necessary next step, both to confirm that the self-forcing behaviour transfers to full-scale conductors and to establish the friction of the full-scale contact pair.

A more detailed mechanical design, including finite-element analysis of the rings and the braid, a dynamic analysis of the lift, and detailed fabrication drawings, is not justified before this behaviour is confirmed, since it would refine a design, whose governing mechanism has only been confirmed at prototype scale.

Once the behaviour is confirmed, the main design effort should be directed at the deployment of the gripper, which is the principal unresolved practical problem and the most direct obstacle to the handling and installation requirements (R.6.1, R.7.4). As established in the operational procedure of the previous section, the braided sleeve must be lowered over the conductor while holding a constant open-state radius with sufficient clearance, and it must be prevented from bunching or losing its form during deployment. The tight open-state ratio of the selected design, at $OSR \approx 1.03$, makes this the binding constraint of the entire configuration, since the open state already lies close to the widest geometry the braid can physically reach. A reliable method for handling the braid and maintaining its shape is therefore the most valuable subject for the next iteration, as the mechanism itself has been shown to work while its deployment has not.

One promising approach is an anti-bunching support sleeve, in which an inner mandrel holds the braid at a fixed open-state radius during deployment. The sleeve could be fixed to the bottom ring, so that it remains in place as the gripper is lowered while the braid passes over it and closes onto the smaller conductor radius below. A drawback of this arrangement is that no contact is established between the braid and the conductor over the length occupied by the sleeve, which reduces the effective gripping length and therefore the holding capacity of the gripper. The proposed sleeve and the associated deployment procedure are shown in Figure 7.4 on the prototype with the steel wire braid, where the support sleeve is formed from a section of cut PVC pipe fixed to the ring by bolts.



Figure 7.4: Anti-bunching support sleeve concept for controlled deployment, shown on the steel wire prototype

Alongside the deployment, the release should be developed further according to the requirement list (R.2.5, R.4.2, R.3.5). The downending frame outlined in the previous section provides one route, in which the frame secures the section in its vertical orientation and may additionally drive the bottom ring upward to relax the grip, either through a fixed element against which the ring is pushed as the conductor is lowered or through an active mechanism. A sacrificial braid that is not recovered offers a simpler alternative that removes the release problem entirely at the cost of the braid material, since the braid is cut away and left rather than relaxed and retrieved from the corroded conductor surface. A cost analysis should therefore be carried out to determine whether sacrificing the braid, while reusing the rings, is economically viable across a removal campaign, comparing the recurring cost of the rope material against the development and operation of a reliable release mechanism.

This analysis also feeds directly into the still-open cost comparison with the current cut-and-pull method (R.9.2). The detailed design of the rings, including their material, cross-section, mass, lifting points, and their connection to the braid, is likewise carried to a later stage, once the deployment method and the full-scale behaviour are established (R.6.1, R.7.1), together with the interfaces to the lifting equipment and the possibility of integrating cutting and pinning equipment on the bottom ring.

A final direction is optimisation. The design point selected here is a workable rather than an optimal configuration, chosen by engineering judgement to sit on the most efficient geometric coefficient examined at prototype scale while satisfying both feasibility boundaries. The feasibility metrics derived in Chapter 4, in particular the open-state and closed-state ratios and the covered area ratio, provide the basis for a formal optimisation study once the model has been refined and larger-scale data are available. Such a study could maximise the open-state ratio, which is the binding constraint of the present design, minimise the total rope length and therefore the material cost, or minimise the overall gripper height, in each case subject to the slip and breaking safety factors. The geometric relations of the model already permit rapid evaluation of any candidate geometry, which makes the design well suited to such an approach.

In summary, the overarching priority across these recommendations is experimental validation at a larger scale, without which the remaining refinements, from the deployment mechanism to the ring embodiment design and any optimisation efforts, cannot be reliably grounded.

7.7 Summary and Discussion

This chapter applied the analytical model, validated in form and scaling in Chapter 6, to the North Cormorant reference conductor in order to produce an early-phase full-scale design. Working under deliberately conservative assumptions and against the offshore lifting safety factors, and beginning from a heavy bottom ring that supplies the passive axial preload, the design arrived at a braid of $N = 7$ turns over a gripping length of $H = 14$ m, corresponding to a geometric coefficient of $\Gamma \approx 53$. This point coincides with the most efficient configuration examined at prototype scale, so the design rests on behaviour that the experiments established rather than on an extrapolation of the model beyond its tested range.

A key outcome of the design is that slip is not the constraint that governs it. The exponential nature of the self-forcing mechanism produces a holding capacity of approximately 18800 kN and a corresponding lift safety factor of about 30, far above the required value of 5. The design is instead governed by the strength of the rope at the top ring connection, where the maximum fibre tension gives a breaking safety factor of approximately 6, and by the tight open-state ratio of $OSR \approx 1.03$, at which the braid in its open deployment state lies close to the widest geometry it can physically reach. These two factors, the rope breaking strength and the open-state ratio, therefore identify the deployment of the braid and the strength of its connections as the elements requiring most attention in a refined design. Beyond the gripping capacity, the design also addressed the system-level requirements carried into this chapter, namely the passing of the guide frames through their prior removal (R.4.1), the selection of a marine-grade steel wire rope resistant to corrosion, fatigue, and ultraviolet degradation (R.5.1, R.5.2), and the verification of the lifting and breaking safety factors against the demonstrated capacity (R.3.1, R.3.3). The remaining in-scope requirements, principally the deployment and release of the braid, were outlined in the operational procedure and identified in the previous section as the central subjects for the next design iteration.

This is, to the author's knowledge, the first compliant gripping concept dimensioned for offshore conductor removal at the multi-tonne scale. The design presented here is not yet ready for implementation but demonstrates that the mechanism scales to the loads of conductor removal under conservative assumptions, accompanied by a structured set of recommendations for the work that must follow. On this basis it answers SRQ2, establishing what a feasible full-scale design requires and confirming that the selected concept is scalable to full-load conductor removal. Finally, it provides a solid foundation for the further development of the concept, whose broader implications and future direction are discussed in the concluding chapter.

8 Conclusion and Future Work

This thesis set out to address a persistent bottleneck in offshore well decommissioning, the removal of conductors from fixed jacket platforms. Current removal practice relies on the cut-and-pull method, in which the entire lifting load is transferred through a tensile point load at the top of the conductor. The method is least effective on the conductors that are hardest to remove, since their aged and corroded sections and connections cannot sustain the concentrated load, and no existing approach distributes the lifting load over the conductor surface. The research objective was therefore to propose a compliant gripping mechanism that distributes the lifting load over the surface of the conductor, and to demonstrate that such a mechanism can scale to the load magnitudes of conductor removal, a regime far beyond the small-scale applications to which soft robotic grippers have so far been confined. This chapter summarises the work carried out, presents the answers to the research questions, and sets out the future work required to advance the concept toward offshore deployment.

8.1 Summary and Results

This section summarises the research and its principal results by returning to the research questions defined in Section 1.2. The three sub-research questions are addressed in turn, each restated together with the answer that the corresponding part of the work provides, before the main research question is answered on the basis of their combined outcome.

SRQ1: Which compliant gripping concept is most suitable for conductor removal?

A literature study into soft robotic gripping methods, actuation strategies, materials, and load-bearing capabilities identified the candidate technologies, and a structured conceptual design process following the methodical design model of Pahl et al. (2007) translated these into four concepts, a braided constriction sleeve, a torsional constriction sleeve, a particle jamming gripper, and an internal inflatable gripper. A weighted evaluation against the requirements established in Chapter 3 selected the braided constriction sleeve as the most suitable concept. Its defining property is the self-forcing mechanism, in which the weight of the conductor itself tightens the braid around the conductor surface, so that a heavier load produces a stronger grip. The concept requires no underwater actuation and holds the load passively, with inherent redundancy across its rope sets, while remaining lightweight and inexpensive relative to the alternatives. The braided constriction sleeve therefore answers SRQ1, chosen for its self-forcing grip and its simple, passive, and low-cost operation.

SRQ2: How does the selected concept scale to full-load conductor removal and what does a feasible full-scale design require?

The need for a full-scale design motivated the development of the analytical model in Chapter 4, which establishes the governing mechanics of the braided sleeve. The model relates the geometry of the braid to its holding capacity through the relation $W_{c,max} = W_{eff}(e^\beta - 1)$ in which the self-forcing coefficient $\beta = \mu\Gamma$ separates the friction coefficient μ from the geometric coefficient $\Gamma = 4\pi^2 r_c N^2 / H$. The holding capacity therefore grows exponentially with the braid geometry, which is the property that makes the concept scalable, and it is supplied passively by the effective weight of a heavy bottom ring. Once the model had been experimentally supported, it was applied to the North Cormorant reference conductor in Chapter 7 under deliberately conservative assumptions, including a design friction coefficient of $\mu = 0.10$ and offshore lifting safety factors of five. The resulting design employs a steel wire braid of $N = 7$ turns over a gripping length of $H = 14$ m, preloaded by a 10 t bottom ring, at a geometric coefficient of $\Gamma \approx 52.6$, which lies within the range of geometric coefficients examined at prototype scale, with the caveat that the upper end of this range yielded a lower bound on capacity rather than a measured slip point. At this point the holding capacity reaches approximately 18800 kN against a design lifting weight of 634 kN, a lift safety factor of about 30, which

establishes that slip does not govern the design. The governing constraints are instead the rope strength at the top ring, satisfied with a breaking safety factor of approximately 6, and the open-state ratio of $OSR \approx 1.03$, which identifies the deployment of the braid as the binding practical constraint. An exemplary operational procedure covering deployment, gripping, lifting, sectioning, and downending was defined, and the remaining design elements were structured into prioritised recommendations. SRQ2 is therefore answered as follows. The concept scales to the full-load conductor removal through the exponential self-forcing relation, and a feasible full-scale design requires a geometry within the experimentally supported range, a passive preload from a heavy bottom ring, and sufficient rope strength at the top ring connection, with a reliable deployment method as the decisive remaining requirement.

SRQ3: How can a scale prototype be designed and tested to demonstrate the feasibility of the selected concept?

Because the friction coefficient of the contact pair cannot be measured independently, a validation strategy was developed that tests the model without relying on a fitted friction value. Ten prototype braids were fabricated and tested across six experimental groups, each varying one governing parameter, with five repetitions per configuration. The results supported every claim of the model that could be examined within the tested range. The holding capacity rose linearly with the bottom ring weight at fixed geometry, confirming that the passive preload sets the baseline capacity. The capacity then grew far more steeply with the geometric coefficient than ordinary friction alone would allow, discriminating the exponential structural form of the model from a linear alternative, with the $H = 200$ mm configuration sustaining 741 N without slip and acting as the decisive case. The geometric trends followed the predicted directions in every group, and a single friction coefficient of approximately $\mu = 0.135$, lying inside the literature range, fitted the consistent configurations across geometries that differ substantially. The steel wire braid produced consistently more repeatable results than the polyamide braid and held its helical form during handling, identifying steel wire rope as the preferred material for the full-scale design. The payload-to-self-weight ratio reached values above 337:1 in the most efficient configuration, demonstrating the exceptional structural efficiency of the mechanism. Two deviations were observed, elevated scatter at sparse braids and a capacity increase with the number of rope sets, and both were traced to idealisations of the model rather than to the mechanism itself. SRQ3 is therefore answered through a validation strategy realised as a parametric prototype campaign, which demonstrated the self-forcing mechanism physically and confirmed the predicted form and scaling of the model at laboratory scale.

MRQ: How can a compliant gripping mechanism be designed to distribute lifting loads over the surface of conductors during removal from fixed offshore jacket platforms?

Taken together, the three sub-questions answer the main research question. A compliant gripping mechanism can be designed to distribute lifting loads over the surface of conductors by employing a braided sleeve gripper, in which a passive preload and the self-forcing mechanism convert the weight of the conductor into a distributed frictional grip whose capacity grows exponentially with the braid geometry. To the author's knowledge, this constitutes the first compliant gripper designed, modelled, experimentally supported, and dimensioned for loads of this magnitude. Soft robotic gripping has so far been demonstrated at payloads of a few hundred kilograms, whereas the present work carries the underlying principles to a design load of several tonnes, extending the reach of the field and showing that compliant gripping is a viable principle for heavy offshore operations.

Returning to the core quality requirement of the thesis, the ability to remove conductors that the cut-and-pull method cannot handle (R.9.1), the concept addresses the underlying failure mode in principle. By distributing the lifting load over a large gripping length instead of concentrating it at a single cross-section, the gripper removes the dependence on the structural integrity of exactly those damaged sections and connections that cause the cut-and-pull method to fail. Whether this translates into successful removals in practice remains to be

confirmed, since it depends on the actual friction and the actual deployment condition realised at full scale, both of which lie beyond what the present work can evidence.

The conclusions above should be read within the limitations of the work. The experimental support for the model was obtained at laboratory scale, on clean and dry polyamide and steel contact pairs whose friction may differ substantially from that of a marine-grown, wet conductor surface. The deployment of the braid, identified as the binding practical constraint, was observed but not resolved, and the model retains the idealisations of a uniform helix, weightless ropes, and a purely static lift. These limitations do not undermine the demonstrated mechanism, but they define the boundary between what this thesis has shown and what the future work must still establish.

The applicability of the mechanism extends beyond the conductor removal case for which it was developed. The gripper engages any sufficiently stiff cylindrical body through distributed friction alone, and its capacity scales with the weight of the object it carries. A natural next application lies in the offshore wind sector, where the installation and, increasingly, the removal of monopiles involve comparable lifting loads on thin-walled tubulars that are sensitive to point and clamping loads. The same principle extends to the handling of large-diameter piles and pipes in other offshore and civil settings. Because the analytical model requires only the geometry, the weight, and the friction pair as inputs, each of these applications can be assessed directly with the tools presented here. The concept is not yet ready for implementation, but the analytical model, the validation methodology, and the full-scale feasibility assessment together provide the foundation on which these developments can be built.

8.2 Future Work and Recommendations

The recommendations for future work follow the same structure as the research itself and concern the refinement of the analytical model, the extension of the experimental campaign, and the further development of the full-scale design.

The analytical model should first be refined by removing the idealisations that the experiments exposed. The self-weight of the ropes should be included as a distributed contribution to the axial preload, since the experiments showed that additional rope sets raise the holding capacity through their weight, an effect the current model neglects next to the much heavier ring. The contact mechanics deserve particular attention, as the assumption of a uniform helix in continuous line contact broke down at the sparse braids, where imperfect helix formation produced both unpredictable capacity and elevated scatter. A refined contact model should capture the directionality of friction along and across the rope, through which the actual braid shape and the local variation of the radial contact pressure can also be represented. The generalised rope-on-surface contact theory of Konyukhov and Schweizerhof (2015), which extends the Euler-Eytelwein relation to helical ropes on rough surfaces with orthotropic friction, provides an analytical foundation for this refinement and is the appropriate starting point for continuing the modelling work [72]. For the submerged full-scale case, buoyancy should be introduced, since it reduces the effective weight of both the ring and the conductor, and the dynamic effects of an offshore lift, presently absorbed into a single amplification factor, should be modelled explicitly. Following these improvements, finite element modelling would provide numerical validation of the analytical relations and would resolve the ring and rope stresses that the analytical model cannot capture. The curve-to-surface contact algorithms and the corresponding benchmark problems developed by Konyukhov and Shala (2021), which are verified directly against the generalised Euler-Eytelwein solutions underlying the present model, offer a validated framework for such simulations [61].

Beyond refinement, the model can be extended to alternative braid architectures. Triaxial braids, which add axial ropes or rods to the helical rope sets as examined by Ochola et al. (2021), could hold the open-state form of the braid during deployment and thereby relax the binding open-state constraint identified in Chapter 7, while

also increasing the holding capacity [73]. Looped rope ends in place of the individual helix terminations, following the research approach of Shanahan et al. (2017), could raise the gripping capacity further while also reducing the fibre tension [74].

The experimental campaign should be extended in three steps. The first is an improved setup, replacing the manual loading and the visual slip identification with a controlled loading mechanism and a load cell recording the full load history, so that slip is identified from the load trace rather than by eye. Given the complexity of the contact mechanics, however, even a load cell may not provide a fully objective slip criterion, but it would improve on the optical method. The second step is a repetition of the full test matrix in steel wire rope. The material comparison identified steel wire as the more consistent material and Chapter 7 specifies it for the full-scale design, so repeating the matrix would ground the validated behaviour in the material that would actually be deployed. The stiffer wire is expected to form more consistent helices around the cylinder and thereby to reduce the scatter across all configurations. Such a campaign would additionally allow the braid to be examined as a system of coupled helical springs, testing whether neglecting this spring behaviour in the presence of a heavy bottom ring is a valid simplification. The third step is testing at a larger and more representative scale, on heavier pipes with realistic surface conditions and, ultimately, submerged, since the friction pair, the surface condition, and the load magnitude all differ between the laboratory and the offshore reference case. As established in Section 7.6, this larger-scale validation is the overarching priority, without which the remaining refinements cannot be reliably grounded. Such a campaign would also serve the design directly. It would inform the open design points, including the dimensioning of the rings, the interface between the rings and the braid, and the interfaces of the rings to the lifting equipment, and it would force the development of a practical fabrication method for the braid at scale. Larger-scale testing therefore feeds back into the requirements established in Chapter 3 and provides input for the subsequent design iterations. Finally, the deployment and release behaviour should be quantified rather than observed, measuring the forces and tolerances involved in lowering the braid over the conductor and in relaxing the grip after downending according to the requirement list (R.2.5, R.4.2, R.3.5).

The final direction concerns the full-scale design itself, where two open fronts stand out. The first is deployment, identified in Chapter 7 as the binding practical constraint. The braid must be held reliably in its open form while it is lowered over a conductor that may be damaged or obstructed. The deployment aid proposed in Section 7.6 should therefore be developed and tested with priority. The second is fabrication, since a 14 m braid of 28 mm steel wire rope around a mandrel of almost 1 m diameter exceeds the capabilities of conventional braiding machines. A purpose-built tool that winds the rope sets around a conductor-diameter mandrel will therefore need to be developed following the requirement list (R.6.1, R.6.2, R.7.4). Beyond these design tasks, the certification of the gripper represents an unavoidable hurdle. Offshore lifting standards such as DNV-ST-N001 presume rigid lifting appliances with deterministic load paths, and no established class rules exist for a compliant device whose capacity depends on friction. Certification will therefore require a technology qualification process, for instance following DNV-RP-A203, in which the failure modes of the braid are systematically identified and the holding capacity is demonstrated statistically across representative conditions. The large safety margins of the present design provide a favourable starting position, but the qualification effort should not be underestimated.

Taken together, the future work outlined above defines the path from the early-phase design presented in this thesis toward a deployable conductor removal tool. The obstacles that remain concern the maturation of the design rather than the working principle, which the experiments supported within the tested range and the full-scale assessment showed to hold with substantial margins under conservative assumptions. The analytical model, the validation methodology, and the design framework developed in this thesis provide the foundation for this continued development, both for conductor removal and for the wider class of heavy tubular lifting operations to which the concept may extend.

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AI Statement

For this MSc thesis, generative AI was employed to:

- Create parts of the Python code used to generate the figures presenting the measured experimental data
- Create parts of the code used to generate the three-dimensional visualisation of the full-scale operational sequence
- Improve the grammar, style, layout, and spelling of the text

Generative AI was not employed to generate technical content, to perform the modelling or analysis, to interpret the experimental results, or to formulate the conclusions presented in this work.

All output has been reviewed and corrected by the author, who remains fully responsible for the content, structure, arguments, and conclusions of this report.

Appendices

A Scientific Research Paper

This appendix contains the scientific research paper that accompanies this report.

Self-Forcing Braided Sleeve Gripper: A Soft Robotics Approach to Offshore Removal of Cylindrical Structures

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ABSTRACT

The decommissioning of offshore oil and gas infrastructure requires the removal of large cylindrical steel structures whose aged and degraded condition often prevents lifting through conventional point loads. This paper presents a braided sleeve gripper that distributes the lifting load over the outer surface of a cylindrical structure through friction. The gripper consists of a braided sleeve connected to a top ring at the lifting equipment and a heavy bottom ring, whose weight passively elongates the braid and constricts it onto the structure. An analytical model is derived that couples the braid geometry to the holding capacity, predicting a linear dependence on the ring weight and an exponential dependence on a self-forcing coefficient formed by the friction coefficient, the cylinder radius, the number of braid turns, and the gripping length. A fabrication method for two-ring braided sleeves is presented, and a laboratory prototype is tested across twelve configurations. The measured slip loads scale linearly with the ring weight and amplify with the geometric coefficient in the manner characteristic of self-forcing, and each geometric parameter follows the direction predicted by the model. The most tightly braided configuration sustained more than 75 kg without slipping, so that its payload-to-self-weight ratio reached at least 337:1 against 16:1 at the baseline configuration. The braided sleeve gripper shows strong potential for offshore heavy-lift removals and for handling large cylindrical structures across the offshore and civil sectors.

1. Introduction

As offshore infrastructure reaches the end of its service life, the removal of ageing structures has become a growing task for the offshore industry. A large share of this work concerns the removal of large-diameter cylindrical steel elements, including piles, caissons, risers, and tubular members, which are typically severed into sections and lifted by cranes or dedicated tools.

These structures are challenging lifting objects. After decades of service they present corrosion, fatigue damage, marine growth, and welded repairs, their internal configuration and structural integrity are often uncertain, and documentation is frequently incomplete. Threaded or degraded connections may not sustain tensile loads, and no two structures are identical, so tooling developed for one case does not always transfer to the next. Current removal methods lift such sections through point loads, either in tension from above or in compression from below. Point loading concentrates the lifting force at a single cross-section, which can be insufficient for severely corroded or damaged structures, and compression lifts are inherently unstable. A method that distributes the lifting load over a large part of the outer surface would reduce local stresses, improve stability, and tolerate the variability between structures.

Soft robotics offers a starting point for such a method. Compliant grippers conform to irregular or variable shapes without detailed geometric information and distribute contact forces over large areas. For heavy cylindrical objects offshore, friction and form closure are the most relevant force-generation principles, because the surface features that hinder conventional tooling, such as corrosion, marine growth, and repair patches, increase the effective friction at

the contact interface and therefore assist a friction-based soft grip, while simultaneously providing interfaces for geometric interlocking. Soft gripping concepts potentially applicable to cylindrical offshore objects can be grouped into four categories. Sleeve-like grippers constrict radially onto the object, through braids loaded in axial tension [1–5], through relative rotation of rings or membranes [6–10], or through inflation of a soft membrane [11–13]. Wrapping and envelopment grippers coil arms, fingers, or filaments around the object, including in subsea conditions [14–20]. Suction-based grippers are applied industrially to the horizontal lifting of cylindrical structures, and soft suction grippers have been adapted to curved, irregular, and underwater surfaces [21–24]. Granular jamming grippers combine friction, suction, and geometric interlocking [25, 26] and have been demonstrated both at heavier scales [27] and underwater [28], with two-pad configurations for cylindrical objects [29, 30] and reinforcements through soft particles or added pinching [31–33].

Across all categories, the highest reported payloads remain on the order of a few hundred kilograms [34], far below the multi-tonne loads of offshore removal. Scaling is fundamentally unfavourable, because the contact area grows with the square of the characteristic length while the gripper volume, and with it the self-weight, grows with the cube [35], requiring additional measures for structural support, energy supply, and cooling. Subsea deployment is a lesser barrier, as hydraulic actuation, neutrally buoyant components, and pressure-tolerant materials have been demonstrated at depth [14, 28], although not at the scale of offshore removals. From these considerations follow the requirements for a soft gripping mechanism suited to offshore removal.

These are low self-weight, simple actuation, preferably through methods already established offshore, self-forcing behaviour in which the load itself generates and reinforces the grip, and a cost comparable to or below that of current removal tooling. A mechanism meeting these requirements would make removals more efficient and would extend removal to degraded structures that cannot be lifted with conventional rigid tooling. Among the categories above, braided sleeves and tubes show the greatest potential for offshore deployment. They exhibit the self-forcing behaviour [1, 2], their actuation is simple, achievable with standard lifting equipment, and they can be constructed from synthetic fibre rope or steel wire rope, materials whose cost remains low when scaled to offshore dimensions.

This paper presents a braided sleeve gripper that uses the self-forcing principle and can be scaled to offshore removal applications. The gripper consists of a braided sleeve connected at its ends to a top ring, suspended from the lifting equipment, and a heavy bottom ring. The sleeve is lowered over the structure in an expanded state. As the top ring is raised and the bottom ring released, the weight of the bottom ring elongates the braid, which contracts radially onto the surface and generates radial pressure and friction. Once the section is severed, its weight adds axial tension to the braid through the friction at the contact interface, which raises the radial pressure and therefore the friction further. This self-amplifying loop, termed self-forcing by Shang et al. [2], means that the grip strengthens as the load increases.

The contributions of this paper are threefold. First, an analytical model is derived that relates the braid geometry and the bottom ring weight to the holding capacity, extending the braid mechanics of Lu et al. [1] and the self-forcing analysis of Shang et al. [2] with a two-ring configuration and a passive gravity preload. Second, a fabrication method for two-ring braided sleeves is presented and used to build ten braids, which are tested in twelve configurations. Third, the claims of the derived model are tested and validated experimentally. To the authors' knowledge, this is one of the first applications of soft robotic gripping proposed at this scale, and the first proposed for offshore heavy-lift removal operations. The remainder of the paper derives the model (Section 2), describes the prototype and the experiments (Sections 3 and 4), presents the results (Section 5), and closes with conclusions and future work (Section 6).

2. Analytical model

This section derives the holding capacity of the braided sleeve gripper. The geometry of the braid is defined first, followed by the axial force required to change its configuration, the coupling of axial tension to radial pressure and friction, and the resulting holding capacity and maximum fibre tension. The derivation follows the braid geometry of Lu et al. [1] and the self-forcing analysis of Shang et al. [2], and extends them with the two-ring configuration and the passive preload supplied by a heavy bottom ring. Offshore lifting capacity typically exceeds the weight of a single removed

section by a wide margin, so additional ring mass carries no practical penalty, in contrast with the payload-driven designs common in soft robotics.

2.1. Braid geometry

The braid consists of n sets of helical fibres, giving $2n$ fibres in total, half wound clockwise and half anticlockwise, interlaced in a plain weaving pattern. Each fibre is a helix of radius r , pitch p , circular cross-section of thickness d , and constant braiding angle α , forming N full turns over the gripping length H , as shown in Fig. 1(a). The braid is assumed uniform, with constant radius and braiding angle along its length, and only the section lying on the cylinder is considered. A single fibre over one full turn, whose height is the pitch p , is shown in Fig. 1(b). From the development of one helical turn into a right-angled triangle (Fig. 1(c)), the pitch, radius, and braiding angle are coupled as follows:

$$p = 2\pi r \tan \alpha \quad (1)$$

The fibre length over one turn S_1 follows from the Pythagorean theorem:

$$p^2 + (2\pi r)^2 = S_1^2 \quad (2)$$

The braid can change configuration between a short and wide state and a long and slim state. To define the states available to a braid of given n and S_1 , the extreme state is derived first. This is the state in which the braid can no longer become shorter and wider, or longer and slimmer, as adjacent fibres come into contact and the braid becomes fully compact, as shown in Fig. 1(d). In this configuration the braid forms n repetitions of the arch AB along its perimeter and n repetitions of the segment CD over one pitch:

$$n AB = 2\pi r \quad n CD = p \quad (3)$$

Since the fibres lie at the braiding angle α and have thickness d , the lengths of the two segments are:

$$AB = \frac{d}{\sin \alpha} \quad CD = \frac{d}{\cos \alpha} \quad (4)$$

From the triangle of Fig. 1(c), $\sin \alpha = p/S_1$, so that the perimeter condition of Eq. (3) becomes:

$$\frac{ndS_1}{p} = 2\pi r \quad (5)$$

Substituting Eq. (5) into Eq. (2) eliminates the radius and yields a quadratic equation in p^2 :

$$p^4 - p^2 S_1^2 + d^2 S_1^2 n^2 = 0 \quad (6)$$

with the discriminant $\Delta = S_1^4 - 4d^2 S_1^2 n^2$. The critical thickness d_c , at which the braid is fully compact and a single configuration exists, follows from $\Delta = 0$:

$$d_c = \frac{S_1}{2n} \quad (7)$$

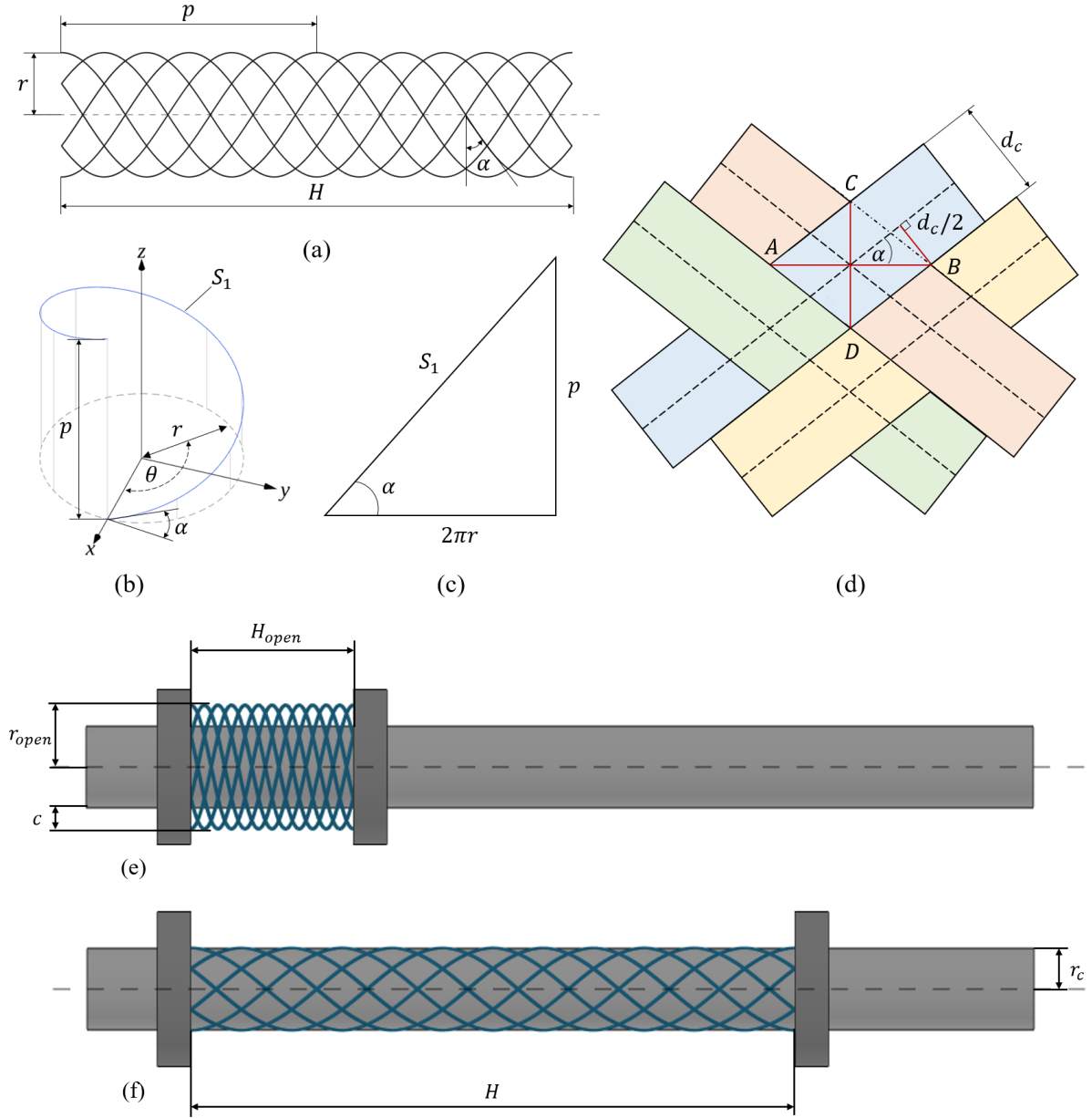


Fig. 1. Geometry of the braided sleeve. (a) Side view of the braid with $N = 2$ turns and $n = 6$ fibre sets. (b) A single helical fibre over one full turn, described by the radius r , pitch p , and braiding angle α , with the resulting one-turn fibre length S_1 . (c) Development of one helical turn into a right-angled triangle relating p , r , α , and S_1 . (d) A segment of the fully compact braid at the critical thickness d_c , showing the arch AB and segment CD that form the rhombic overlaps at the weaving crossings. (e) Open state during deployment, drawn horizontally for clarity, with the braid holding a radial clearance c over the cylinder of radius r_c and a braid length H_{open} . (f) Closed state during gripping, with the braid lying on the cylinder surface over the gripping length H .

A deformable braid therefore requires $d < d_c$. For $d < d_c$ the braid can move between two compact extremes, whose pitch follows from the solutions of Eq. (6):

$$p_{\max/\min} = \sqrt{\frac{S_1^2 \pm \sqrt{\Delta}}{2}} \quad (8)$$

These correspond to the extreme states of the braid, namely the longest and slimmest state $(p_{\max}, r_{\min}, \alpha_{\max})$ and the shortest and widest state $(p_{\min}, r_{\max}, \alpha_{\min})$.

The contact area between the braid and the cylinder is needed for the pressure derivation. Two assumptions are

made, that the full width of each fibre contacts the cylinder, as though the cross-section were rectangular, and that contact is not lost at the weaving crossings. Under these assumptions the $2n$ fibres cover a rectangular strip A_{S_1} over one full turn:

$$A_{S_1} = 2ndS_1 \quad (9)$$

The overlaps occurring at the weaving crossings must be subtracted from this area. Each overlap forms a rhombus whose diagonals are the arch AB and the segment CD

(Fig. 1(d)), with area:

$$A_{\text{rhombus}} = \frac{AB \cdot CD}{2} = \frac{d^2}{\sin 2\alpha} \quad (10)$$

The crossings over one full turn number $2n^2$, where the fibre ends at the rings are taken as half overlaps since the helices start and terminate there. The net contact area over one turn A_1 is therefore:

$$A_1 = A_{S_1} - 2n^2 A_{\text{rhombus}} = 2ndS_1 - \frac{2n^2 d^2}{\sin 2\alpha} \quad (11)$$

The total contact area A , the full length of one fibre S , and the total fibre length used in the braid S_{total} over N turns are:

$$A = NA_1 \quad S = NS_1 \quad S_{\text{total}} = 2nS \quad (12)$$

Equations (11) and (12) deviate from Lu et al. [1], who are not consistent in the way the total contact area enters the radial pressure. The forms above are used throughout this work.

2.2. Open and closed states

Controlling the manufacture and operation of a large braid through its braiding angle is impractical. The braid is therefore specified through two control parameters, the gripping length H and the number of turns N , together with the cylinder radius r_c , from which all remaining parameters follow through Eqs. (1) to (12), with the pitch given by:

$$p = \frac{H}{N} \quad (13)$$

The state in which the braid lies on the cylinder is the closed state, with $r_{\text{closed}} = r_c$ and length $H_{\text{closed}} = H$. For deployment, the braid holds a radial clearance c over the surface, defining the open state with $r_{\text{open}} = r_c + c$. Both states are shown in Fig. 1(e) and (f). Since the fibre length is conserved between states, the braid length in the open state follows from Eqs. (2) and (13):

$$H_{\text{open}} = N \sqrt{\left(\frac{H}{N}\right)^2 + (2\pi r_c)^2 - (2\pi(r_c + c))^2} \quad (14)$$

This expression gives the compact size of the gripper during deployment, relevant for handling and transport, and the height change between the states is $\Delta H = H_{\text{closed}} - H_{\text{open}}$.

The extreme states of Eq. (8) may not be reachable in practice owing to material limitations, and proximity to either extreme may render a braid infeasible. Two ratios express how close the open and closed states are to these extremes. The open state ratio (OSR) is defined between the shortest and widest state and the open state, and the closed state ratio (CSR) between the closed state and the longest and slimmest state:

$$\text{OSR} = \frac{r_{\text{max}}}{r_{\text{open}}} \quad \text{CSR} = \frac{r_{\text{closed}}}{r_{\text{min}}} \quad (15)$$

A feasible braid requires both ratios to be greater than unity, and design iterations should place them as far from unity as

possible. Finally, the area cover ratio C_A is defined between the total contact area A of Eq. (12) and the surface of the cylinder section over the gripping length:

$$C_A = \frac{A}{2\pi r_c H} \quad (16)$$

This ratio quantifies how completely the grip is distributed over the cylinder surface, in contrast with the point loads of conventional removal methods.

2.3. State transitions and effective ring weight

Moving the braid between states requires an axial force. To estimate it, the braid is treated as a combination of open-coiled helical springs with ends fixed against rotation, following the methodology introduced by Jedwab and Clerc [36] for braided medical stents on the basis of the spring theory of Wahl [37], applied to a braided gripper by Shang et al. [2]. Let (r_0, α_0) denote the fabrication state, in which the braid is woven around a mandrel and which is taken as the equilibrium state. Deforming the braid to an arbitrary state with braiding angle α' requires an axial force calculated as follows:

$$T = 4n \left[\frac{GI_p}{K_3} \left(\frac{2 \sin \alpha'}{K_3} - K_1 \right) - \frac{EI \tan \alpha'}{K_3} \left(\frac{2 \cos \alpha'}{K_3} - K_2 \right) \right] \quad (17)$$

where E and G are the Young's and shear moduli of the fibre. For a circular cross-section, the area moments of inertia are:

$$I = \frac{\pi d^4}{64} \quad I_p = \frac{\pi d^4}{32} \quad (18)$$

The constants are evaluated at the fabrication state:

$$K_1 = \frac{\sin 2\alpha_0}{2r_0} \quad K_2 = \frac{\cos^2 \alpha_0}{r_0} \quad K_3 = \frac{2r_0}{\cos \alpha_0} \quad (19)$$

Equation (17) is the corrected expression of De Beule et al. [38]. The discrepancy in the original formulation of Jedwab and Clerc [36], an additional cosine term in the torsional contribution, was identified by De Beule [39]. The sign of T depends on the direction of the transition relative to the fabrication state. Moving towards a longer and slimmer state requires a positive axial force, whose positive-valued magnitude is denoted T_e , while moving towards a shorter and wider state requires a negative axial force, whose positive-valued magnitude is denoted T_c . Both transitions, starting from the common fabrication state, are illustrated in Fig. 2.

The fabrication state therefore influences the operation of the gripper. If the fabrication state is longer and slimmer than the closed state, the braid must be compacted to reach the closed state, and the restoring force assists gripping, as the braid tends to return to its slimmer fabrication state and thereby increases the radial compression on the cylinder. If instead the fabrication state is shorter and wider than the open state, deployment and release are eased, as the braid

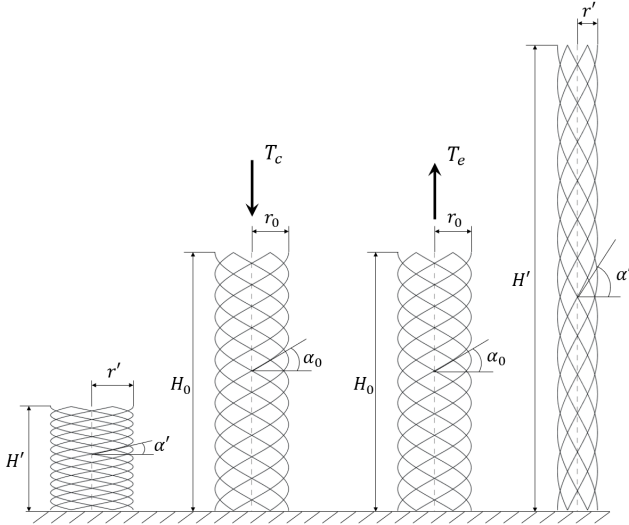


Fig. 2. Transitions of the braided sleeve under compacting T_c and elongating T_e forces, starting from the fabrication state (r_0, α_0).

always tends to return to a more compact configuration, but the restoring force provides no positive contribution to gripping. The heavy bottom ring introduces an axial elongating force that counteracts the restoring compaction and thereby increases the grip. The ring provides this force passively, without active or continuous actuation, and the grip is sustained by weight rather than by a powered system.

The operational procedure and the forces acting on the two rings are illustrated in Fig. 3. For this illustration the fabrication state is assumed to coincide with the open state, $r_0 = r_{\text{open}}$, a practical choice since the braid can be woven around a mandrel at the deployment geometry and then shipped and deployed directly, with the braid only ever constricting inward from this state to the closed state. The weights of the top ring, the fibres, and additional equipment are assumed negligible compared to the weight of the bottom ring, and friction between the braid and the cylinder is not considered here, as it governs the actual gripping treated in the following section. In Fig. 3(a) the gripper hangs from the lifting equipment, with its full weight supported through the connection of the bottom ring and with the top ring ropes slack, allowing the sleeve to remain compact. In Fig. 3(b) the gripper is positioned over the cylinder, around the midpoint of the intended gripping section. In Fig. 3(c) both rings are moved simultaneously, with the top ring ropes tensioned to pull the top ring upward and the bottom ring ropes gradually released to lower the bottom ring, distributing the braid along the cylinder in a controlled manner. The elongation of the braid introduces the compacting restoring force T_c , which acts to return the braid towards its fabrication state, so the ring weight W_{ring} must be large enough to overcome it. In practice, friction between the braid and the cylinder further assists by acting against the restoration. In Fig. 3(d) the braid has reached the closed state, the bottom ring ropes are fully released, and pretension is applied to the top ring to ensure lifting after cutting. The section is then severed, the pretension is lost, and the system reaches the equilibrium of

Fig. 3(e), in which the lifting equipment carries the weight of the severed section and of the bottom ring.

Since the fabrication state coincides with the open state, which is wider than the closed state, the braid has been elongated from its equilibrium to reach the closed state. The restoring force therefore acts upward on the bottom ring, as shown in the free body diagram of Fig. 3(f), reducing the effective elongating force the ring exerts on the braid:

$$W_{\text{eff}} = W_{\text{ring}} - T_c \quad (20)$$

In the alternative case, where the fabrication state is slimmer than the closed state, the restoring force tends to elongate the braid and pushes the bottom ring further downward, so the effective ring weight exceeds the actual ring weight. In general, the effective ring weight is:

$$W_{\text{eff}} = W_{\text{ring}} \pm T \quad (21)$$

with T the positive-valued magnitude of the restoring force evaluated between the fabrication state and the closed state through Eq. (17). The quantity W_{eff} is the net axial force transmitted to the braid by the bottom ring and is the primary input to the gripping model.

Two limiting cases simplify Eq. (21). For a braid of synthetic fibre rope, the bending and torsional stiffness of the fibres is negligible, $T \approx 0$, and $W_{\text{eff}} \approx W_{\text{ring}}$. The braid then may not hold its form when unloaded, as the ropes tend to bunch up rather than return to a defined fabrication state, which complicates deployment. For a stiff braid, such as steel wire rope at full scale, the ring weight can be selected such that $W_{\text{ring}} \gg T$ and the same simplification holds. The full expression is retained for small-scale prototypes of stiff material, where the ring cannot be made heavy enough to dominate the restoring force.

2.4. Holding capacity

The holding capacity is derived in two steps. The conversion of axial force into radial pressure is established first. The tension distribution along the sleeve, which results from the friction feedback, is then derived.

2.4.1. Axial force to radial pressure

Axial deformation of the braid produces radial deformation, which presses the braid onto the cylinder. The relationship between the axial force F and the radial pressure P follows from the virtual work principle, applied as by Lu et al. [1] under the assumptions of a uniform helix and inextensible fibres, so that the external work converts fully into pressure on the gripped object. Equating the external work of the axial force through a pitch change δp to the internal energy of the radial force through the radial displacement δr over one turn gives the following relation:

$$F \delta p = -P A_1 \delta r \quad (22)$$

where the negative sign reflects that the radius decreases as the braid elongates. The pressure and axial force are then coupled by the force conversion factor k :

$$P = kF \quad k = -\frac{\delta p}{A_1 \delta r} \quad (23)$$

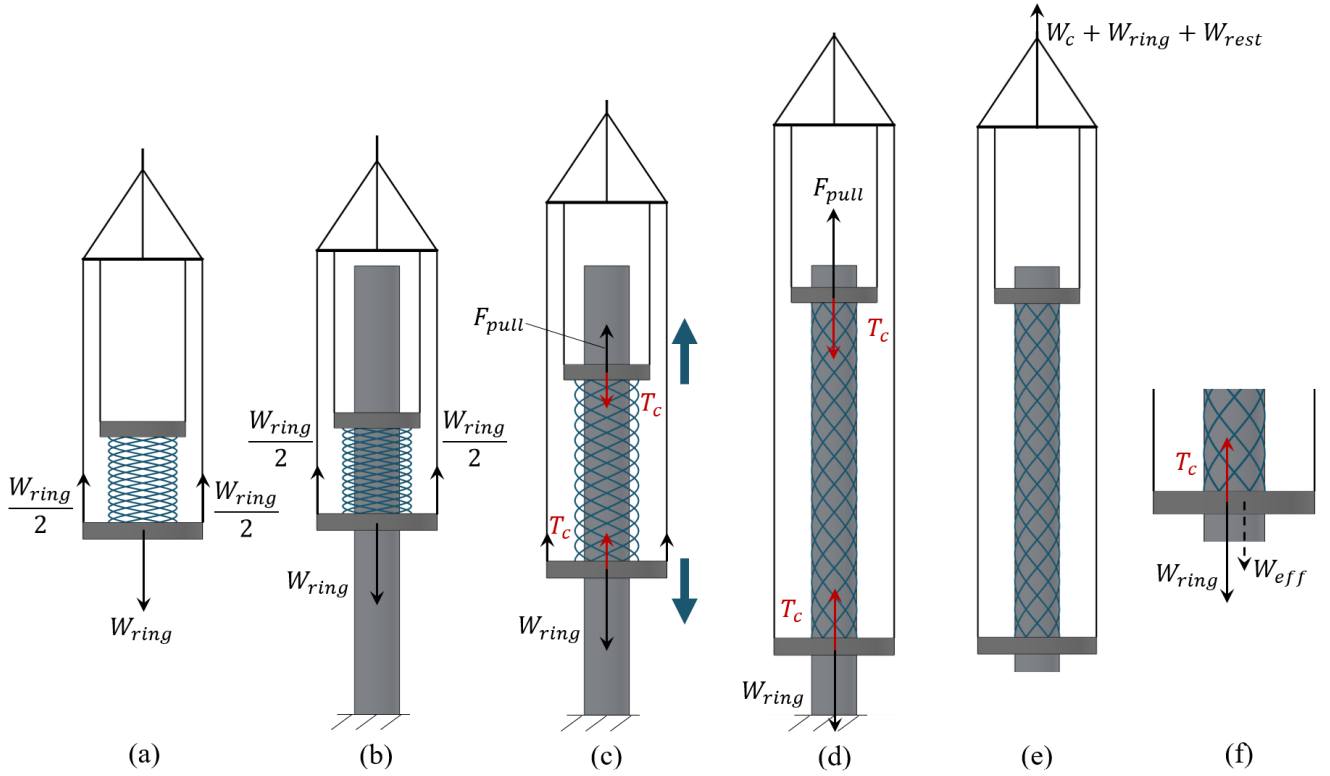


Fig. 3. Operational sequence of the gripper for a fabrication state coinciding with the open state. (a) Gripper hangs from the lifting equipment with the top ring ropes slack and the sleeve compact. (b) Gripper positioned over the cylinder at the midpoint of the intended section. (c) Top ring pulled upward and bottom ring released downward, elongating the braid against the compacting restoring force T_c . (d) Closed state reached and pretension applied to the top ring. (e) Section severed, pretension lost, and the system at equilibrium. (f) Free body diagram of the bottom ring showing the effective ring weight.

Differentiating Eq. (2) at constant S_1 yields:

$$\frac{dp}{dr} = -\frac{4\pi^2 r}{p} = -\frac{2\pi}{\tan \alpha} \quad (24)$$

Substituting Eq. (24) into Eq. (23) for small deformations, with the total contact area $A = NA_1$ and the control parameters of Section 2.2, gives the conversion factor:

$$k = \frac{2\pi N}{A \tan \alpha} = \frac{4\pi^2 r_c N^2}{AH} \quad (25)$$

The number of turns N is absent from the conversion factor of Lu et al. [1]. Equation (25) otherwise agrees with the derivation of Jedwab and Clerc [36], where, however, the contact area is simplified to a full cylinder. That simplification is not acceptable at the scale considered here, where the web-like contact area of the braid is much smaller than the full cylinder surface.

2.4.2. Tension distribution and slip condition

Friction converts the radial pressure into axial holding force, and the friction itself adds tension to the braid, which raises the pressure further. Fig. 4 shows the free body diagrams at the instant the section is severed and supported by friction alone. The full system is shown in Fig. 4(a), with the coordinate x running from the top ring ($x = 0$) to the bottom of the sleeve ($x = H$). The cylinder carries the

friction distribution $F_f(x)$ and the pressure distribution $P(x)$ (Fig. 4(b)), which react on the braided sleeve (Fig. 4(c)).

The friction between the braid and the cylinder is represented by a Coulomb idealisation with a single constant coefficient μ , acting purely in the axial direction. The braid and its contact are assumed uniformly distributed along the cylinder, dynamic effects are excluded, and the braid self-weight is neglected against the heavy ring. The coefficient μ , describing contact between the braid and an aged, corroded steel surface, is the most uncertain input of the model.

The contact area of an infinitesimal element dx of the sleeve (Fig. 4(d)) is $(A/H) dx$, so the available friction on the element is expressed as follows:

$$dF_f = \mu P(x) \frac{A}{H} dx = \mu k F(x) \frac{A}{H} dx \quad (26)$$

At the top of the braid the tension carries the full cylinder weight together with the effective ring weight, $F(0) = W_c + W_{eff}$, and each element transfers part of its load to the cylinder through friction, $dF = -dF_f$. Substituting Eq. (26) and introducing the decay coefficient:

$$\gamma = \frac{\mu k A}{H} = \mu \frac{4\pi^2 r_c N^2}{H^2} \quad (27)$$

gives a first-order differential equation whose solution is the tension distribution:

$$F(x) = (W_c + W_{eff}) e^{-\gamma x} \quad (28)$$

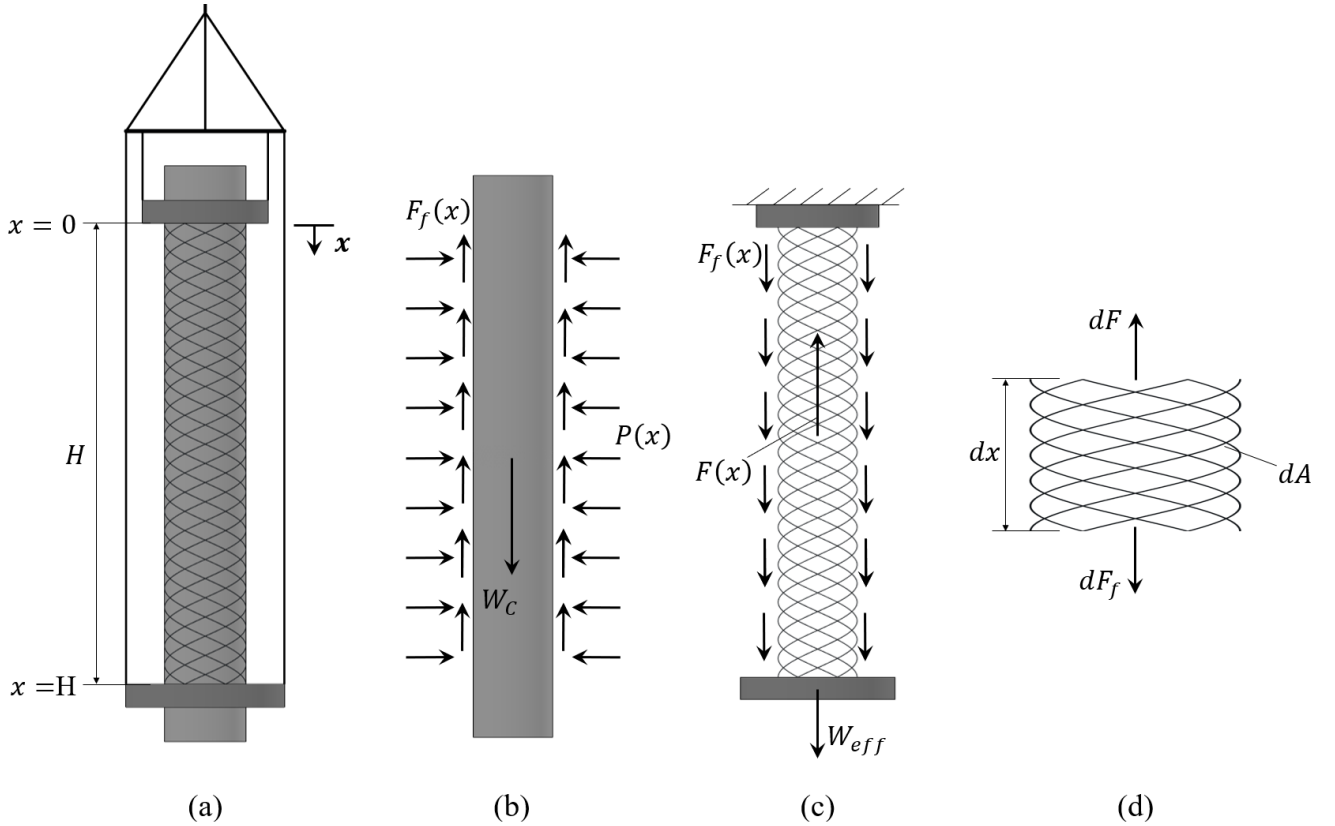


Fig. 4. Free body diagrams at the instant the section is severed. (a) Full system with coordinate x and gripping length H . (b) Cylinder free body diagram with the friction distribution $F_f(x)$ and pressure distribution $P(x)$. (c) Braided sleeve free body diagram. (d) Infinitesimal element dx of the sleeve.

Note that γ does not depend on the actual contact area, as expected from the Coulomb assumption, but only on the braid geometry and the friction coefficient.

At the verge of slipping under the maximum sustainable cylinder weight, the self-forcing increment of the tension is fully consumed at the bottom of the sleeve, so the slip threshold is $F(H) = W_{eff}$. Substituting this boundary condition into Eq. (28) yields the maximum holding capacity:

$$W_{c,max} = W_{eff} (e^\beta - 1) \quad (29)$$

where β is the dimensionless self-forcing coefficient:

$$\beta = \gamma H = \mu \frac{4\pi^2 r_c N^2}{H} \quad (30)$$

The capacity scales linearly with the effective ring weight, which sets the baseline, and exponentially with β , where a larger cylinder radius or number of turns amplifies the capacity while a longer gripping length reduces it. Equation (29) closely matches the maximum holding force of Shang et al. [2], the difference being the ring contribution introduced here.

Self-forcing may not develop for light objects, whose weight is insufficient to feed the friction amplification [2]. In that regime the radial pressure remains at the preload value and the grip reduces to the following minimum:

$$W_{c,min} = \mu k W_{eff} A = \beta W_{eff} \quad (31)$$

which grows only linearly with the geometric amplification. The separation between Eqs. (29) and (31) distinguishes self-forcing from ordinary preload friction. Heavy structures are expected to operate in the self-forcing regime, but whether the mechanism develops cannot be settled by the model alone, since it depends on friction conditions that are unknown in advance. The experimental objective is therefore not to confirm the exact exponential form of Eq. (29), but to establish whether the capacity amplifies with β or remains close to the linear response of Eq. (31), together with the scaling of β with N^2 , r_c , and $1/H$, which is independent of the friction coefficient.

2.5. Maximum fibre tension

The critical structural location is the top ring connection, where the braid tension reaches $F(0) = W_c + W_{eff}$. Treating each fibre as a rope of negligible bending stiffness carrying pure axial tension, which is appropriate for synthetic and steel wire rope specified by a minimum breaking load, and resolving the fibre tension at the closed-state braiding angle α , the maximum tension in a single fibre is calculated as follows:

$$F_{fibre} = \frac{W_c + W_{eff}}{2n \sin \alpha} \quad (32)$$

The fibre tension increases as the braiding angle decreases, since a shallower fibre contributes a smaller vertical component. Equation (32) is compared against the rated minimum

breaking load of the selected rope, reduced by the applicable safety factor, when dimensioning a full-scale design.

3. Prototype design and fabrication

Building on the model, this section presents the scaled prototype used to examine the hypothesised behaviour. The objectives and validation strategy are defined first, followed by the design of the prototype and the fabrication method developed to construct the braids.

3.1. Objectives and validation strategy

The prototype is not intended to predict the absolute holding capacity, which depends on the unknown friction coefficient and on idealised contact mechanics. Its first objective is to confirm that the self-forcing mechanism physically develops, meaning that the measured capacity amplifies with the braid geometry rather than remaining linear. Its second objective is to confirm the geometric scaling of the model, so that the model can inform a full-scale design under appropriate safety factors.

Synthetic fibre rope was selected for the braid, which makes the restoring force negligible ($T \approx 0$, $W_{\text{eff}} \approx W_{\text{ring}}$) and fabrication by hand practical. For the self-forcing regime, the model used for the tests then reduces to the following form:

$$W_{\text{slip}} = W_{\text{ring}} (e^{\mu\Gamma} - 1) \quad (33)$$

and for the non-amplifying regime:

$$W_{\text{slip}} = \mu \Gamma W_{\text{ring}} \quad (34)$$

where W_{slip} is the load at slip. The friction coefficient is separated from the geometry by defining the geometric coefficient:

$$\Gamma = \frac{\beta}{\mu} = \frac{4\pi^2 r_c N^2}{H} \quad (35)$$

The shape of the relationship between W_{slip} and Γ reveals whether self-forcing occurs, independently of its absolute value. Four claims are tested, each isolating one aspect of the model. The first is ring-weight linearity. At fixed geometry both regimes predict capacity proportional to W_{ring} , providing a friction-independent check of the model structure. The second is the structural form of the capacity. Varying Γ over a wide range through N , H , and r_c distinguishes the exponential from the linear form, with both regimes evaluated over a friction coefficient range taken from literature. The third is the geometric trends, where each of N , H , and r_c is varied in isolation to test the direction and degree of its predicted influence. The fourth is area independence. Under the Coulomb idealisation, changing the number of fibre sets n at fixed geometry should not alter the capacity. In addition, the braid material is varied from synthetic rope to steel wire as a qualitative check on the uniform-helix assumption and on deployment behaviour.

3.2. Prototype design

The prototype was kept as simple as possible, since a separate braid had to be fabricated for every geometric configuration. The synthetic braids use a widely available polyamide braided rope of 3 mm diameter. The steel braid uses 2 mm steel wire, whose smaller diameter kept hand-braiding manageable. The two rings were 3D-printed in polylactic acid (PLA), dimensioned to accommodate all parameter variations, each with a mass of approximately 100 g. Optional control handles, connected to each ring through separate ropes, allow independent control of the rings during deployment. They were not used during the slip tests, where they complicated the procedure. The gripped object in all tests is a smooth polyvinyl chloride (PVC) pipe of 1 m length, selected for its consistent surface finish and availability in the standard outer radii of 20, 25, and 30 mm. The measured pipe mass is included in the slip load of each test.

3.3. Fabrication

The fabrication method was adapted from Kortman et al. [40], who manufactured braided sleeves for artificial muscles with local control of the braiding angle. The precision required there was not necessary here, and a simplified procedure was developed. The braiding disc is a scaled-up, 3D-printed version of the traditional Japanese Kumihimo disc, advanced upward along threaded rods fixed to the bottom ring, with the PVC pipe serving as braiding mandrel so that the fabrication state coincides with the gripped geometry. The procedure is illustrated in Fig. 5. The ropes are knotted through the holes of the bottom ring, with a small weight on each free end to maintain tension (Fig. 5(a)). The threaded rods are attached to the bottom ring with washer-and-nut assemblies acting as seats for the disc (Fig. 5(b)), the disc is positioned on the seats, and the mandrel is centred through the ring and fixed (Fig. 5(c)). In each braiding cycle, all clockwise ropes are moved one slot clockwise, then all anticlockwise ropes one slot anticlockwise, crossing over them. In the following cycle the order is reversed, so that the crossings alternate, reproducing the hollow round braid (Naiki Gumi) pattern also found in the Chinese finger trap and in braided stents. Two consecutive cycles, showing the alternating crossings, are shown in Fig. 5(d). After each cycle the disc is raised by a fixed increment, and the cycles are repeated until the target length is reached (Fig. 5(e)).

Table 1

Baseline configuration of the prototype.

Quantity	Symbol	Value	Unit
Cylinder radius	r_c	30	mm
Number of turns	N	3	–
Gripping length	H	400	mm
Number of fibre sets	n	4	–
Bottom ring mass	m_{ring}	100	g
Bottom ring weight	W_{ring}	0.98	N
Braid material	3 mm polyamide rope		

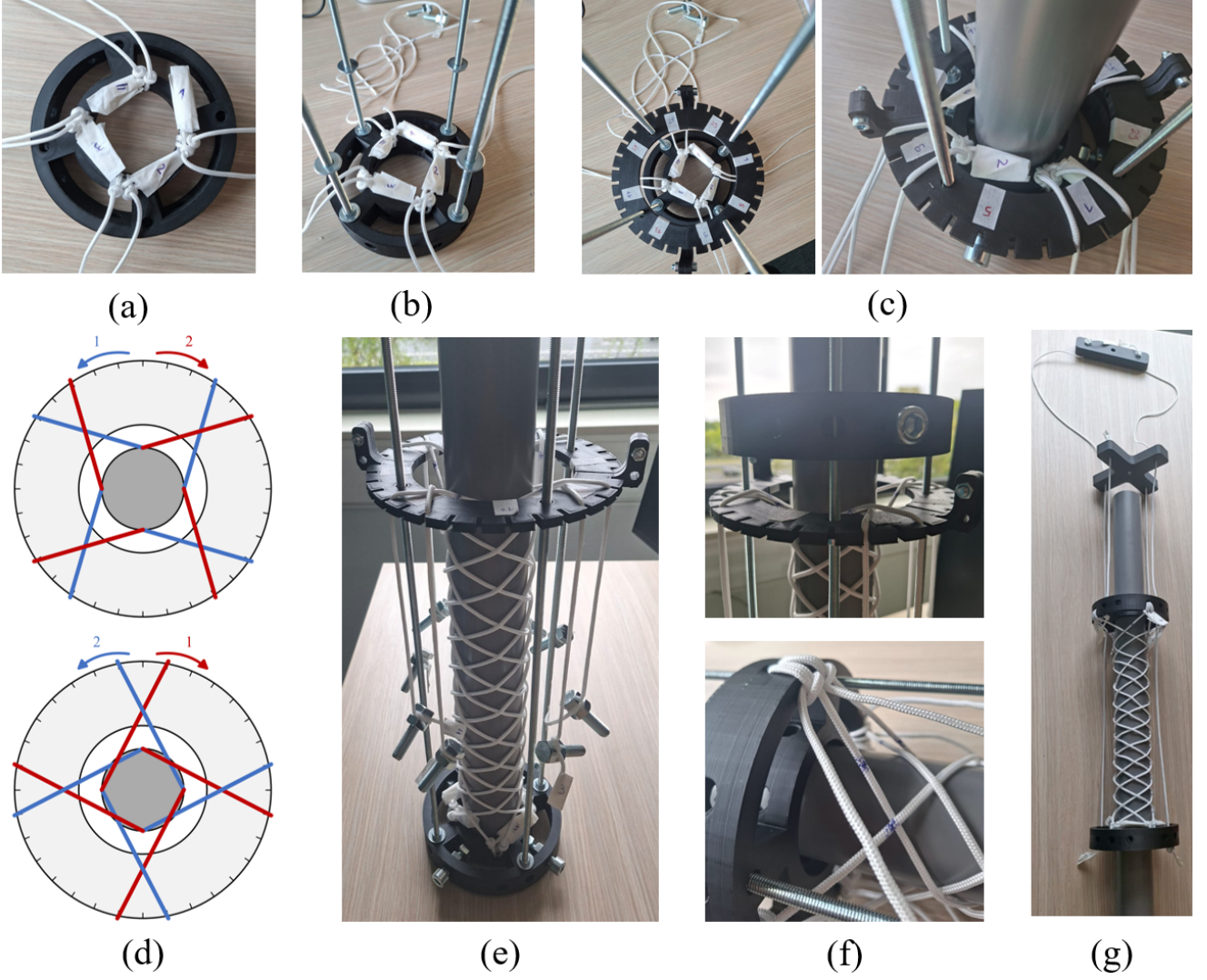


Fig. 5. Fabrication of the braided sleeve prototype. (a) Ropes knotted to the bottom ring with weights maintaining tension. (b) Threaded rods attached to the bottom ring with washer-and-nut seats for the braiding disc. (c) Braiding disc positioned on the seats with the PVC mandrel centred and fixed. (d) Two consecutive braiding cycles, showing the alternating crossings. (e) Braiding cycles repeated until the target length is reached. (f) Top ring positioned and fixed with the ropes knotted to its outer holes. (g) Completed prototype with the optional control handles attached.

For a braid of N turns, gripping length H , and $2n$ ropes, the number of cycles and the height increment per cycle are given by:

$$N_{\text{cycles}} = 2nN \quad h = \frac{H}{N_{\text{cycles}}} \quad (36)$$

Once all cycles are complete, the top ring is fixed on the mandrel and the ropes are knotted through its outer holes (Fig. 5(f)). The two-piece disc is then unscrewed and removed. The completed prototype, with the optional control handles attached, is shown in Fig. 5(g). For the steel braid, the wire ends are fixed with U-bolt clamps instead of knots, and the braiding itself is considerably more demanding, as the wire tends to escape the disc slots during the cycles.

4. Experiments

This section presents the experimental campaign carried out with the prototype. The test matrix defining the braid configurations is presented first, followed by the test procedure and the statistical treatment of the measurements.

4.1. Test matrix

A baseline configuration was defined as a representative midpoint of the parameter ranges, given in Table 1. From this baseline, six groups were defined, each varying one governing parameter while holding the others fixed, as listed in Table 2. Group A varies the bottom ring weight by hanging additional masses of approximately 400 and 800 g from hooks on the ring, addressing ring-weight linearity. Groups B, C, and D vary N , H , and r_c respectively, each requiring a newly fabricated braid (and, for Group D,

Table 2

Experimental test matrix. Each group varies one parameter with respect to the baseline.

Group	W_{ring} (N)	N	H (mm)	r_c (mm)	n	Material
A	0.98, 5.20, 8.80	3	400	30	4	Polyamide
B	0.98	2, 3, 4	400	30	4	Polyamide
C	0.98	3	200, 400, 600	30	4	Polyamide
D	0.98	3	400	20, 25, 30	4	Polyamide
E	0.98	3	400	30	2, 4, 8	Polyamide
F	0.98	3	400	30	4	Steel wire

**Fig. 6.** Fabricated configurations of the test matrix. (a) Baseline. (b) $N = 2$. (c) $N = 4$. (d) $H = 200$ mm. (e) $H = 600$ mm. (f) $r_c = 20$ mm. (g) $r_c = 25$ mm. (h) $n = 2$. (i) $n = 8$. (j) Steel wire.

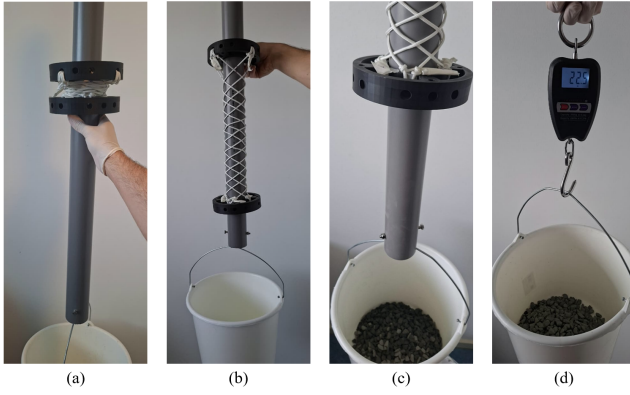


Fig. 7. Test procedure. (a) Gripper in its open state lowered over the pipe. (b) Gripper deployed in its closed state on the pipe with the empty bucket suspended. (c) Bucket filled incrementally with gravel until slip occurs. (d) Bucket removed and weighed with the hanging dynamometer.

a matching mandrel and pipe), and together address the structural form of the capacity and the individual geometric trends. Group E varies the number of fibre sets n to test area independence, and Group F substitutes the polyamide rope with steel wire at otherwise identical geometry. Although the spacing between parameter values appears small, the exponential dependence of the model on the geometry produces a wide range of predicted capacities across the matrix. The fabricated configurations are shown in Fig. 6, with the baseline in Fig. 6(a). Figs. 6(b) and (c) show the turn sweep ($N = 2$ and 4), Figs. 6(d) and (e) the length sweep ($H = 200$ and 600 mm), Figs. 6(f) and (g) the radius sweep ($r_c = 20$ and 25 mm), Figs. 6(h) and (i) the fibre set sweep ($n = 2$ and 8), and Fig. 6(j) the steel wire braid. Increasing N or n , or reducing H or r_c , visibly densifies the braid, and the steel braid holds its form more consistently than the polyamide one.

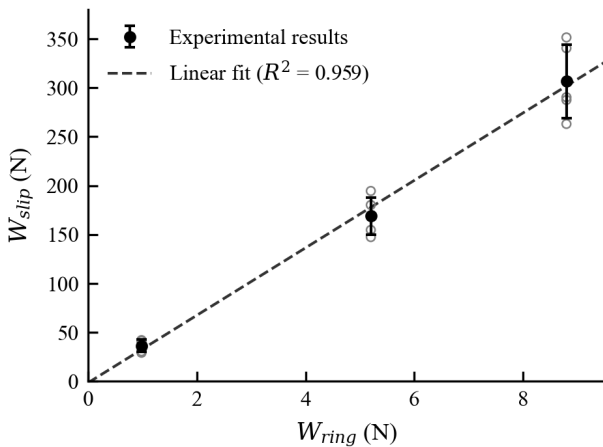


Fig. 8. Slip load against bottom ring weight at the baseline geometry ($r_c = 30$ mm, $N = 3$, $H = 400$ mm, $n = 4$), with a linear fit through the individual repetitions.

4.2. Test procedure

The test procedure is illustrated in Fig. 7. The gripper in its open state was lowered over the pipe (Fig. 7(a)). The bottom ring was then released while the top ring was pulled upward, bringing the braid into its closed state, with an empty bucket suspended from a bolt fixed through the pipe wall (Fig. 7(b)). Once deployed, the bottom ring was not touched, since pushing it downward would artificially increase the braid tension. The bucket was filled incrementally with gravel, slowly, to avoid dynamic disturbances, until slip occurred (Fig. 7(c)). Slip was identified optically as the first instance of relative movement between the braid and the pipe. The pipe could re-engage after a small slip, but the first instance was recorded for consistency. For configurations expected to exceed a full bucket, a fixed initial weight based on the model prediction was placed in the bucket before filling. After slip, the bucket was removed and weighed with a digital hanging dynamometer (BIL200, Tempo di Saldi, Fig. 7(d)) with a range of 0.5 to 200 kg and a resolution of 50 g, corresponding to approximately 0.5 N, negligible relative to the measured loads. The weight of the pipe and hanging hardware was added to obtain the total slip load W_{slip} . Each configuration was tested five times. The repetitions were reduced to a mean, a sample standard deviation s , and a coefficient of variation $CV = s/\bar{W}_{\text{slip}}$, which is dimensionless and therefore comparable across configurations of different capacity.

5. Results and discussion

This section first reports the measured slip loads and their statistics. The four model claims of Section 3.1 are then tested against the data, followed by the material comparison, the identification of the friction coefficient, the payload-to-self-weight ratio, and the limitations of the experiments.

5.1. Measured slip loads

Table 3 summarises the mean slip load, standard deviation, and coefficient of variation for all configurations, together with the geometric coefficient Γ computed from the braid geometry. One configuration could not be brought to slip. The $H = 200$ mm braid sustained the maximum applied load of 741 N, approximately 75 kg, with additional manual pulling producing no slip. Loading was stopped at this point because the PLA rings were observed to flex, so the true capacity of this configuration exceeds the maximum load the setup can apply. Several of the model comparisons below require a friction coefficient range for the polyamide–PVC pair, which cannot be measured independently in this configuration. A range of $\mu \in [0.10, 0.16]$ is adopted from the capstan-method coefficients reported for synthetic fibre ropes in dry contact against a smooth surface [41], the condition most consistent with the present tests.

5.2. Ring-weight linearity

The first claim is that, at fixed geometry, the capacity scales linearly with the ring weight. Both model regimes predict this regardless of μ . Fig. 8 presents the measured

Table 3

Summary of results and statistics for all configurations. Each entry is the mean of five repetitions.

Group	W_{ring} (N)	N	H (mm)	r_c (mm)	n	Material	Γ	$\bar{W}_{\text{slip}}$ (N)	s (N)	CV (%)
Baseline	0.98	3	400	30	4	Polyamide	26.6	36.5	6.1	16.7
A	5.20	3	400	30	4	Polyamide	26.6	168.9	19.0	11.3
A	8.80	3	400	30	4	Polyamide	26.6	306.5	37.7	12.3
B	0.98	2	400	30	4	Polyamide	11.8	8.0	3.2	39.7
B	0.98	4	400	30	4	Polyamide	47.4	515.9	66.1	12.8
C	0.98	3	200	30	4	Polyamide	53.3	>741.0	—	—
C	0.98	3	600	30	4	Polyamide	17.8	25.7	12.1	46.9
D	0.98	3	400	20	4	Polyamide	17.8	12.8	0.8	6.5
D	0.98	3	400	25	4	Polyamide	22.2	24.9	2.9	11.8
E	0.98	3	400	30	2	Polyamide	26.6	24.4	8.9	36.5
E	0.98	3	400	30	8	Polyamide	26.6	46.1	3.1	6.7
F	0.98	3	400	30	4	Steel wire	26.6	88.5	1.1	1.3

slip load against the ring weight for Group A, together with an ordinary least squares fit through the individual repetitions. The fit yields a coefficient of determination of $R^2 = 0.959$, and the unconstrained regression passes close to the origin, as expected since zero ring weight corresponds to zero preload in both regimes. The result confirms that the preload sets the holding capacity in the manner the model assumes, within the range of ring weights achievable with the prototype.

5.3. Structural form of the holding capacity

The central claim concerns the form of the relationship between capacity and the geometric coefficient Γ , which is linear for ordinary preload friction, Eq. (34), and exponential for self-forcing, Eq. (33). Fig. 9 presents the measured slip loads of the baseline and Groups B, C, and D, all at $W_{\text{ring}} = 0.98$ N, against Γ , with both model forms evaluated over $\mu \in [0.10, 0.16]$. The measurements depart from the linear band and follow the exponential form, rising more steeply with Γ than the linear relation permits within the friction range considered. The $H = 200$ mm configuration reinforces the distinction. At $\Gamma \approx 53.3$, the linear model predicts slip between approximately 5 and 9 N across the friction range, while the measured lower bound of 741 N exceeds this prediction by close to two orders of magnitude. This conclusion holds regardless of the true friction coefficient within the assumed range. The exact functional form cannot be proven by these data, since the friction coefficient remains unknown and the contact mechanics are idealised. What the data establish is that the capacity amplifies with Γ rather than scaling linearly, so the self-forcing mechanism develops within the tested range.

5.4. Geometric trends

Fig. 10 presents the three geometric sweeps against the self-forcing model evaluated over the assumed friction range. For the number of turns (Fig. 10(a)), the model predicts that the exponent scales with N^2 . The capacity increases sharply between $N = 3$ and $N = 4$, consistent with this dependence, with both points inside the model band. The $N = 2$ configuration deviates upward and scatters more

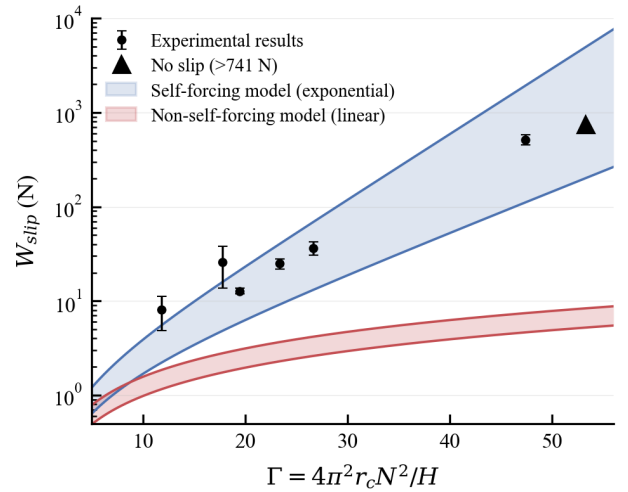


Fig. 9. Measured slip load against the geometric coefficient Γ , compared with the self-forcing model (exponential, Eq. (33)) and the non-amplifying model (linear, Eq. (34)), each evaluated over the friction coefficient range $\mu \in [0.10, 0.16]$. Only the baseline and Groups B, C, and D, all at $W_{\text{ring}} = 0.98$ N, are shown.

widely. A plausible explanation is the sparse braid formed at low N , in which fewer helical wraps cover the same length, leaving larger gaps and allowing individual fibres to shift out of a uniform helical path under load. For the gripping length (Fig. 10(b)), the capacity decreases as H increases, in the predicted direction, with the $H = 200$ mm point entering as a lower bound and the sparse $H = 600$ mm braid again showing wide scatter. For the cylinder radius (Fig. 10(c)), the capacity increases with r_c in the predicted direction, with all three points inside the model band, though the points do not track the curvature of the band closely.

A geometric inconsistency contributes to the radius trend. When the radius is reduced while the ring positions are fixed, the braid forms an hourglass-like necking pattern, so the length over which it contacts the pipe is shorter than the nominal ring-to-ring distance H , as shown in Fig. 11(b) for $r_c = 20$ mm against the baseline radius of Fig. 11(a). This pattern was modelled for the Chinese finger trap by

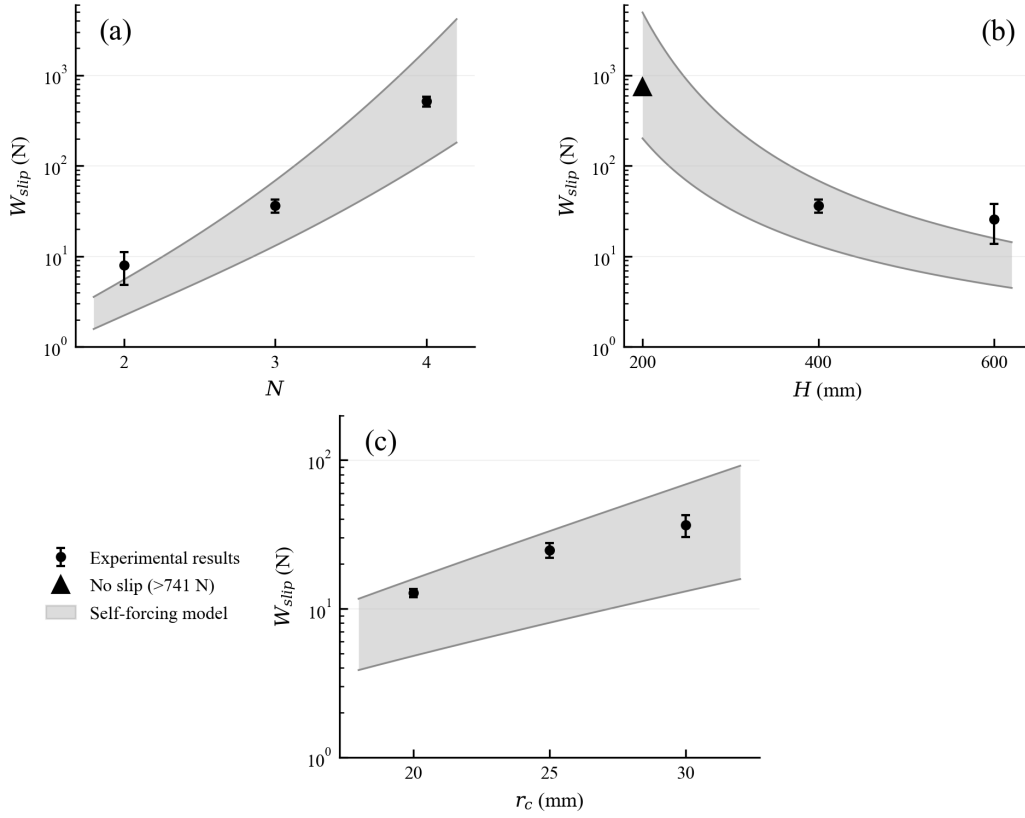


Fig. 10. Measured slip load against each geometric parameter varied in isolation, compared with the self-forcing model evaluated over the assumed friction range. (a) Number of turns N . (b) Gripping length H . (c) Cylinder radius r_c .

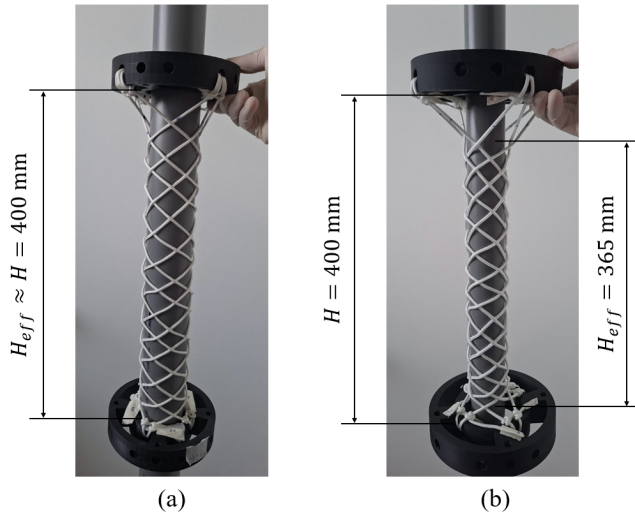


Fig. 11. Necking of the braid at reduced cylinder radius, with the ring-to-ring distance set to $H = 400$ mm in both cases. (a) $r_c = 30$ mm, where the contact length approaches the nominal length. (b) $r_c = 20$ mm, where the effective contact length reduces to 365 mm.

Lu et al. [1] and is neglected here through the uniform-helix assumption. The effective contact length H_{eff} was measured directly for the reduced-radius configurations, giving 365 mm and 380 mm at $r_c = 20$ and 25 mm against the nominal 400 mm. The correction is too small to reshape

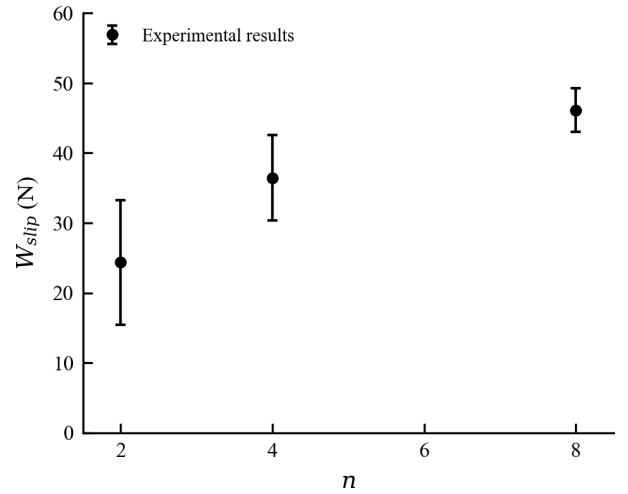


Fig. 12. Measured slip load against the number of fibre sets n , giving $2n$ fibres in total, at otherwise fixed baseline geometry.

the trend visibly, but it is carried into the friction coefficient identification below, where all radius configurations are evaluated with their measured effective lengths.

5.5. Area independence

Under the Coulomb idealisation the capacity is independent of the true contact area, so changing the number of

fibre sets n at fixed geometry should not alter it. Fig. 12 shows that the measured capacity instead increases with n . This departure cannot be attributed with certainty to a failure of the area-independence assumption, because the self-weight of the rope offers a plausible alternative explanation. The model neglects the braid weight against the ring, an approximation that is reasonable at full scale but questionable at prototype scale, where the ring weighs only 100 g. The added mass of extra fibres contributes to the axial pull in the same way as added ring weight, and Group A established how sensitive the capacity is to such additions. Distinguishing the two explanations requires a model that carries the braid self-weight explicitly, which is left to future work. The sparser configurations in this sweep also show the wider scatter observed across the other sparse braids.

5.6. Material comparison

Group F substitutes the polyamide rope with steel wire at identical geometry and therefore identical Γ . The comparison addresses repeatability and the uniform-helix assumption. Absolute capacity is not comparable between the two materials, since the friction pair and the ratio of braid weight to ring weight differ. The clearest difference, shown in Fig. 13, is the far smaller scatter of the steel braid, with a coefficient of variation of 1.3% against 16.7% for the polyamide baseline. The greater bending stiffness of the steel wire holds the helical form more consistently, satisfying the uniform-helix assumption more closely. This observation extends to the sparse polyamide configurations, where the helices form non-uniformly (Fig. 14(a)) and the measured capacities scatter widely. A denser braid forms fewer such inconsistencies (Fig. 14(b)) and produces more repeatable results, as the coefficients of variation in Table 3 show across all configurations. The link between helix uniformity and scatter is a hypothesis at this stage and would require a more detailed model to verify.

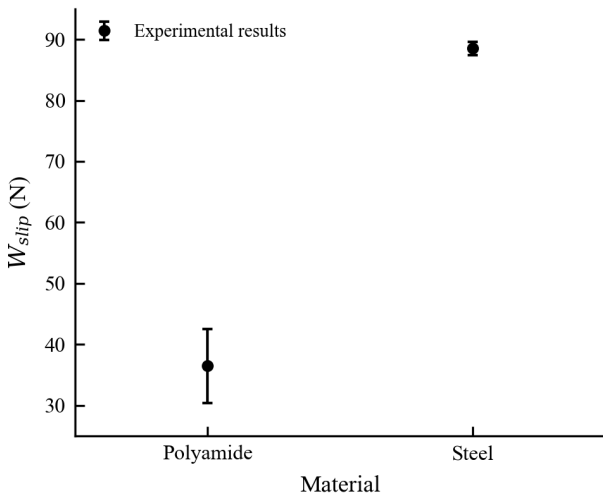


Fig. 13. Measured slip load for the polyamide and steel wire braids at shared geometry ($r_c = 30$ mm, $N = 3$, $H = 400$ mm, $n = 4$, $W_{\text{ring}} = 0.98$ N).

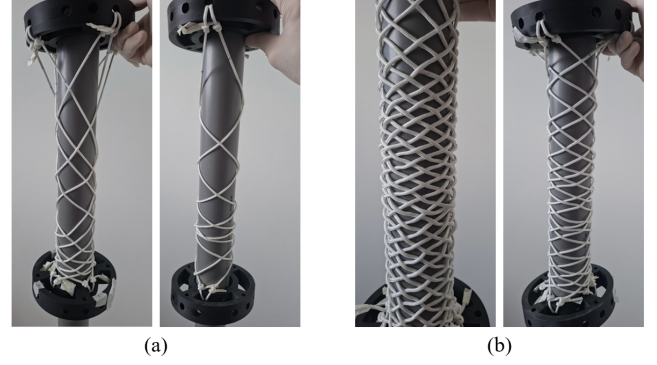


Fig. 14. Helix formation in the fabricated braids. (a) Sparse braids, in which the helices deviate from a uniform path. (b) Dense braids, which form consistent helices.

Beyond repeatability, the steel braid retained its form when removed from the pipe, whereas the polyamide braid bunched together and lost its shape, as shown in Fig. 15. The steel braid was correspondingly easier to deploy and held a more consistent open-state geometry. This is a practical advantage for offshore deployment, where controlled positioning is essential, and it informs the material selection for a full-scale design. The greater self-weight of the steel braid also contributes to the axial pull, tightening the grip.

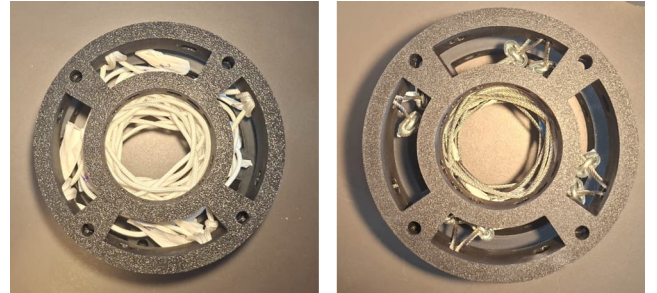


Fig. 15. The two braids after removal from the pipe, with the polyamide braid bunched together and the steel wire braid retaining its open-state form.

5.7. Friction coefficient identification

With the structural form and trends established, a representative friction coefficient can be identified from the data by inverting Eq. (33):

$$\mu = \frac{1}{\Gamma} \ln \left(\frac{W_{\text{slip}}}{W_{\text{ring}}} + 1 \right) \quad (37)$$

The identification is applied to a calibration subset of consistent configurations, excluding those whose deviations were attributed to effects the model does not capture. These are the $H = 200$ mm braid, whose slip was never reached, the sparse $H = 600$ mm and $N = 2$ braids, and the $n = 2$ and $n = 8$ braids affected by the rope self-weight. The radius configurations are evaluated with their measured effective lengths of 365 mm and 380 mm rather than the nominal 400 mm listed in Table 3, and the steel braid

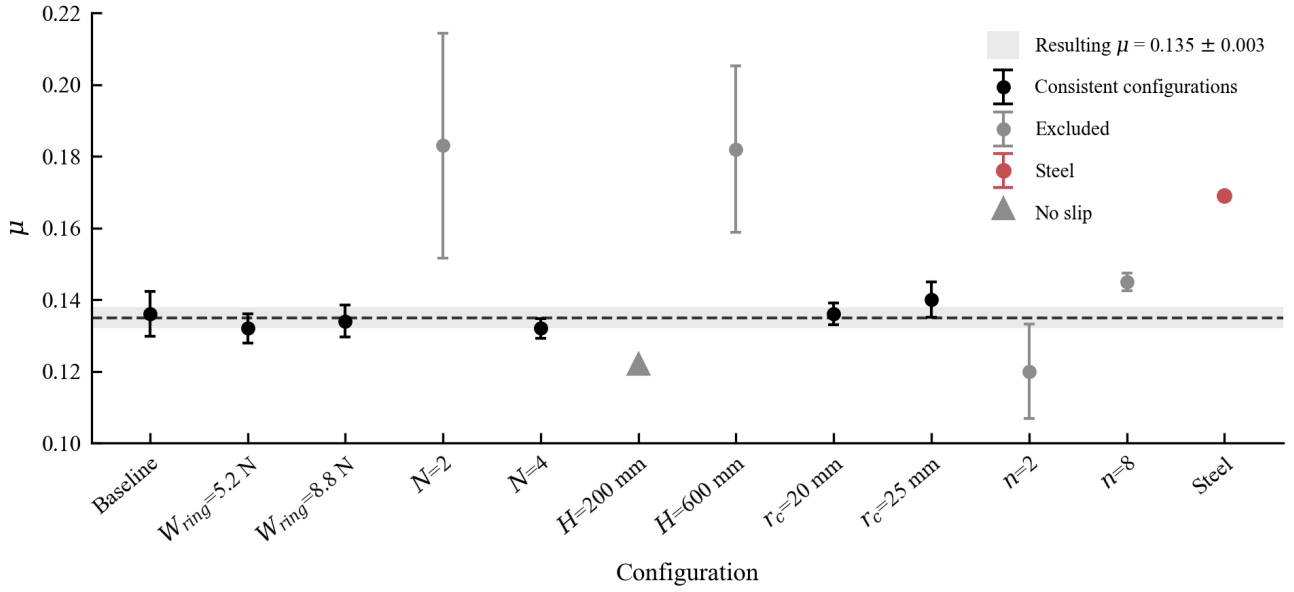


Fig. 16. Estimated friction coefficient for all tested configurations, with the mean and standard deviation computed from the consistent configurations alone.

is treated separately as a distinct friction pair. Across the consistent configurations the coefficient remains close to $\mu = 0.135$, with a coefficient of variation of approximately 2.3%, inside the literature range adopted earlier. The steel braid gives $\mu = 0.169$. Fig. 16 presents the estimates for all configurations, showing the systematic rise with the number of fibre sets that reflects the braid self-weight effect. That a single coefficient reconciles configurations of widely differing geometry supports the Coulomb idealisation within the tested range, though the number of configurations is small and the identification rests on the same simplified contact model.

5.8. Payload-to-self-weight ratio

The payload-to-self-weight ratio, the holding capacity relative to the combined weight of the rings and braid, situates the concept against the soft gripping literature. The ratio does not transfer directly to full scale, since self-weight grows with the cube of the characteristic length while capacity does not grow in the same proportion, but it indicates structural efficiency. The self-weight varies only modestly across the polyamide braids, so three representative configurations suffice. The baseline, with a gripper weight of 2.23 N, reaches a ratio of approximately 16:1. The steel braid at identical geometry reaches approximately 27:1 despite its greater self-weight of 3.28 N, as its higher friction and the additional axial pull of its own weight raise the capacity by more than the added weight. The $H = 200 \text{ mm}$ configuration, with a gripper weight of 2.20 N, reaches at least 337:1, a lower bound because slip was never reached. A substantial capacity is therefore achieved at a small fraction of the gripper self-weight, which supports scaling toward a comparatively lightweight tool for multi-tonne loads.

5.9. Limitations

The results are subject to several limitations. The procedure was manual and therefore operator-dependent, and the most consequential step, the identification of slip, was visual. The first observable movement between braid and pipe is difficult to distinguish from the settling of the braid as it conforms under load. A load cell recording the full loading history would allow slip to be identified from the load trace instead. The maximum applicable load was governed by the manual loading method and by the strength of the printed rings, while the gripper itself had not reached its capacity, so the $H = 200 \text{ mm}$ configuration is recorded only as a lower bound. Only two sub-configurations per group beyond the baseline were tested, sufficient to establish the direction of each trend but not its exact form. The prototype also remains far from offshore conditions in its friction pair, surface state, submersion, and load magnitude, so the identified friction coefficient checks the model and should not be transferred to full scale.

One further observation is reported for completeness. Initial tests returned unusually high capacities, traced to dust from the gravel reaching the contact interface and raising the effective friction. The pipes were cleaned, the affected braids remade, and the tests repeated. This sensitivity to surface condition supports the expectation that the roughened surface of an aged offshore structure is more likely to raise the effective friction than to lower it.

6. Conclusions

This paper presented a braided sleeve gripper for the offshore removal of cylindrical structures, in which a heavy bottom ring passively elongates a braided sleeve onto the structure and the lifted load itself amplifies the grip through

the self-forcing mechanism. An analytical model was derived that couples the braid geometry to the holding capacity, predicting a capacity linear in the effective ring weight and exponential in the self-forcing coefficient $\beta = 4\pi^2\mu r_c N^2/H$, together with the maximum fibre tension at the top ring. A fabrication method for two-ring braided sleeves was developed, and a laboratory prototype was tested in twelve configurations spanning the ring weight, the number of turns, the gripping length, the cylinder radius, the number of fibre sets, and the braid material.

The experiments support the aspects of the model that can be examined without knowledge of the friction coefficient. The capacity scaled linearly with the ring weight ($R^2 = 0.959$), amplified with the geometric coefficient Γ in the manner characteristic of self-forcing rather than of ordinary preload friction, and followed the predicted direction of each geometric trend. A single back-calculated friction coefficient of $\mu = 0.135$, with a coefficient of variation of 2.3%, reconciled all consistent configurations, supporting the Coulomb idealisation within the tested range. The payload-to-self-weight ratio was 16:1 at the baseline and reached at least 337:1, and the stiffer steel wire braid produced the most repeatable results and the most practical deployment behaviour. Two deviations from the model were observed, at sparse braids and with increasing fibre count. Both point to refinements of the model, and neither indicates a failure of the mechanism. Within the tested range, the results demonstrate that a passively actuated braided sleeve can grip and hold a cylindrical object with a capacity that grows with the load, which is the property required for lifting structures whose condition prevents point loading. To the authors' knowledge this is one of the first applications of soft robotic gripping principles at this scale, and the first proposed for offshore heavy-lift removal operations.

The applicability of the mechanism extends beyond offshore decommissioning. The gripper engages any sufficiently stiff cylindrical body through distributed friction, and its capacity grows with the weight of the object it carries. Comparable lifting operations arise in the offshore wind sector, where the installation and removal of monopiles involve heavy thin-walled tubulars that are sensitive to point and clamping loads, and in the handling of large-diameter piles and pipes in other offshore and civil settings. Since the model requires only the geometry, the weight, and the friction pair as inputs, each of these applications can be assessed directly with the relations derived here.

Future work is identified in three directions. The first concerns the model. The braid self-weight should enter the formulation explicitly, since it acts as a distributed preload and became measurable at prototype scale. The uniform-helix assumption should be replaced by contact mechanics that capture the necking pattern and the behaviour of sparse braids, including the directionality of friction along and across the fibres. The generalised rope-on-surface contact theory of Konyukhov and Schweizerhof [42], which extends the Euler–Eytelwein relation to ropes on rough surfaces with orthotropic friction, is a suitable starting point. For

submerged operation, buoyancy reduces the effective weight of both the ring and the gripped section and should be introduced, and the dynamic effects of an offshore lift should be modelled explicitly. Finite element modelling with curve-to-surface contact algorithms, for which benchmarks verified against the generalised Euler–Eytelwein solutions exist [43], would then resolve the ring and fibre stresses that the analytical model cannot capture. The braid architecture itself can also be extended. Triaxial braids, which add axial elements to the helical sets [44], could hold the open-state form during deployment, and looped fibre ends, examined for braided stents by Shanahan et al. [45], could raise the capacity while reducing the fibre tension.

The second direction concerns the prototype and its testing. A controlled loading mechanism and a load cell recording the full load history would replace visual slip detection with an identification from the load trace, and a stiffer test frame and rings would extend the applicable load range beyond the 741 N limit of the present setup. Repeating the test matrix in steel wire rope would ground the observed behaviour in the material identified as the more consistent of the two, and would allow the braid to be examined as a system of coupled helical springs, for which the restoring force term of the model becomes relevant. More sub-configurations per parameter would allow the exponents of the scaling to be estimated, where the present data support only the direction of each trend, and automated braiding would remove the variability of hand fabrication. Testing at larger scale against representative surfaces, including corroded steel, marine growth, and submerged conditions, would establish friction coefficients relevant to operation, and the deployment and release of the braid should be quantified rather than only observed.

The third direction concerns the path toward offshore application. The mechanics validated here should be demonstrated at intermediate scale under multi-tonne loads with steel wire rope braids. Deployment is the principal practical constraint. The braid must hold its open-state geometry while it is lowered over a structure that may be damaged or obstructed, and the release of the braid after set-down, whether through re-tensioning of the bottom ring, external handling, or a sacrificial braid, remains to be developed. Fabricating braids at this scale exceeds the capabilities of conventional braiding machines and requires a purpose-built winding tool. Finally, offshore lifting standards presume rigid lifting appliances with deterministic load paths, so a compliant device whose capacity depends on friction would require a technology qualification process in which the failure modes are identified and the holding capacity is demonstrated statistically across representative conditions.

Altogether, this work shows that the self-forcing braided sleeve can lift heavy cylindrical structures whose degraded condition rules out conventional point loading, distributing the load through friction while its grip strengthens with the weight it carries. Scaling the mechanism to multi-tonne offshore removals, together with the deployment and release of the braid in the field, remains the focus of future work.

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B Requirements List

TU Delft Allseas		Requirements List for Soft Robotic Gripper for Offshore Conductor Removal		Page 1 of 4
Change	Pr.	ID	Requirements	
Constraints				
4/2/2026		1	System Environment	
	D	1.1	Shall operate in offshore subsea environment (no specific environmental conditions defined for conceptual phase, DAF = 1.5)	
	D	1.2	Based on North Cormorant platform as reference: depth, jacket dimensions, guide frame configuration and conductor spacing (Appendix C)	
	D	1.3	Conductor specifications (diameter, internal configuration, grouting, length, weight, marine growth) based on North Cormorant platform (Appendix C)	
	D	1.4	Topside assumed removed and all preparatory work completed before conductor removal	
	D	1.5	Conductor cutting is outside scope (assumed pre-cut using diamond wire saw, abrasive waterjet cutting, or equivalent method)	
	D	1.6	Inner casings are pinned using (flush) pins, securing all casings and assumed to have sufficient structural capacity to carry their full weight	
	D	1.7	Conductor guide frames either removed or remain in place with 25 mm radial clearance to conductor outer surface	
	D	1.8	Conductors shall not be lifted through a point load in tension	
	D	1.9	Gripping load shall be distributed along outer conductor surface with no upper limit on grip force (surface assumed not to be damaged by gripping)	
	D	1.10	Conductors assumed symmetric with centre of gravity at geometric centre (no deviation from the cylindrical shape) and buoyancy is neglected	
Functional Requirements				
22/1/2026		2	General Performance & Capabilities	
	D	2.1	Shall distribute grip force over the conductor surface	
Replaces version dated: 4/2/2026		Pr. D W	Priority Demand Wish	Version: Date: Responsible: 4 9/3/2026 Christos Dielas

TU Delft Allseas		Requirements List for Soft Robotic Gripper for Offshore Conductor Removal		Page 2 of 4
Change	Pr.	ID	Requirements	
	D	2.2	Shall carry the entire lifting load	
	D	2.3	Shall maintain grip under all conditions (environmental loads, vessel motions, etc.)	
	D	2.4	Shall conform to irregular, corroded, or damaged surfaces without precise alignment	
	D	2.5	Shall allow engagement and release of the conductor	
9/3/2026		3	Safety	
	D	3.1	Shall comply with offshore lifting standards (DNV-ST-N001, DNVGL-ST-0378, DNV-RP-H201, DNV-RP-N102, etc.)	
	D	3.2	Shall be fail-safe on power loss (maintain grip)	
	D	3.3	Shall prevent dropped objects	
	D	3.4	Shall provide redundant gripping	
	D	3.5	Shall maintain grip during downending (bending/buckling risk mitigation)	
	D	3.6	Shall mitigate emergency situations while maintaining control of the conductor	
9/3/2026		4	Operation & Control	
	D	4.1	Shall overcome obstacles during approach and lifting (guide frames, jacket members, etc.)	
	D	4.2	Shall accommodate downending of conductor (onto vessel, platform, or seabed)	
	D	4.3	Shall provide grip verification and load monitoring (to be detailed in later design phases)	
Replaces version dated: 4/2/2026		Pr. D W	Priority Demand Wish	Version: Date: Responsible: 4 9/3/2026 Christos Dielas

TU Delft Allseas		Requirements List for Soft Robotic Gripper for Offshore Conductor Removal		Page 3 of 4
Change	Pr.	ID	Requirements	
	W	4.4	Should limit diver and ROV involvement	
	W	4.5	Should allow self-deployment	
	W	4.6	Should engage outer diameter (preferred over internal grip)	
	W	4.7	Should remove conductor in one go or minimise sectioning cuts and lifting cycles	
22/1/2026		5	Materials	
	D	5.1	Shall use marine-grade, corrosion-resistant materials	
	D	5.2	Shall resist fatigue and UV degradation	
	W	5.3	Should use lightweight materials	
	W	5.4	Should use materials with low environmental impact	
22/1/2026		6	Manufacturing, Assembly & Maintenance	
	D	6.1	Shall be manufacturable with existing industrial capabilities	
	W	6.2	Should use standardised methods of manufacturing and assembly	
	W	6.3	Should allow ease of maintenance	
22/1/2026		7	Installation & Logistics	
	D	7.1	Shall include lifting points for crane or handling equipment	
Replaces version dated: 4/2/2026		Pr. D W	Priority Demand Wish	Version: 4 Date: 9/3/2026 Responsible: Christos Dielas

TU Delft Allseas		Requirements List for Soft Robotic Gripper for Offshore Conductor Removal		Page 4 of 4
Change	Pr.	ID	Requirements	
	W	7.2	Should be easily transportable and storable	
	W	7.3	Should fit within a standard 20 ft or 40 ft shipping container for offshore transport	
	W	7.4	Should be assembled and disassembled on deck without specialised equipment	
22/1/2026		8	Environment & Sustainability	
	W	8.1	Should minimise environmental impact during operation	
	W	8.2	Should be reusable across multiple conductor removals	
Quality Requirements				
22/1/2026		9	Quality & Costs	
	W	9.1	Should enable removal of conductors that the cut-and-pull method cannot handle	
	W	9.2	Should be cost-comparable or better than the cut-and-pull method (materials, manufacturing, assembly, and operation)	
Replaces version dated: 4/2/2026		Pr. D W	Priority Demand Wish	Version: 4 Date: 9/3/2026 Responsible: Christos Dielas

C North Cormorant Platform and Conductors

This appendix provides relevant information on the North Cormorant offshore platform and its conductors, used as the reference case for concept calculations and dimensioning and full-scale design recommendations throughout this thesis. The North Cormorant platform is selected as it represents a representative fixed jacket platform, and its decommissioning is a future project of Allseas. The platform information and dimensions are drawn from publicly available operator publications, while conductor dimensions and specifications are sourced from internal Allseas documentation.

C.1 North Cormorant Platform

The North Cormorant platform is located on the UK continental shelf in the northern North Sea. The platform is supported by an eight-legged steel jacket structure at a water depth of 161 m below Lowest Astronomical Tide (LAT). The planned decommissioning scope involves cutting and removing the platform at -116 m LAT [75], as illustrated in Figure C.1, which also shows an elevation map of the jacket.

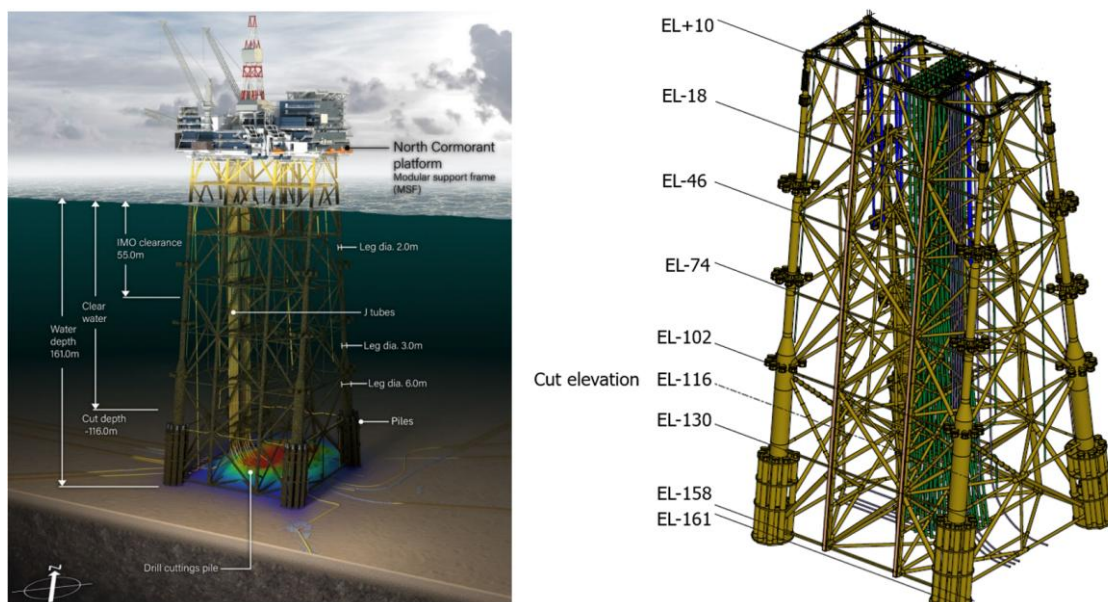


Figure C.1: North Cormorant decommissioning schematic [75] (left) and jacket frame elevation map [76] (right)

The current Allseas planning involves removing the topside in a single lift using the Pioneering Spirit, leaving the jacket standing at +11 m LAT with the conductors reaching +13 m LAT. Conductors are planned to be removed at -106 m LAT rather than -116 m LAT, with the remaining stub removed in a later campaign, giving a total removable conductor length of 119 meters. As visible in Figure C.1, seven conductor guide frames are present at elevations of +10, -18, -46, -74, -102, -130, and -158 m LAT. Conductor spacing within the frames is presented in the following section.

C.2 North Cormorant Conductors

The North Cormorant platform has 40 conductors, as shown in Figure C.2. Their internal configurations vary considerably across the platform. All conductors share a 30-inch outer casing, while the internals vary. For simplicity, the most common internal configuration is adopted, consisting of two inner casings of 20-inch and 13 3/8-inch outer diameter respectively. The annuli formed between the casings are referred to as the C annulus (innermost) and D annulus (outermost). While the wall thickness of the casings is not known, a common thickness of 3/4 inch is assumed throughout. A number of conductors also feature Talon connectors, indicated by the suffix -T in the figure, which have no reliable tensile capacity and therefore cannot be relied upon for

lifting. Additionally, several outer conductors are curved, but these are excluded from the scope of this thesis. As shown in Figure C.2, the spacing between conductor rows is 1528 mm and between conductor columns is 1678 mm, measured from the outer conductor surfaces [76].

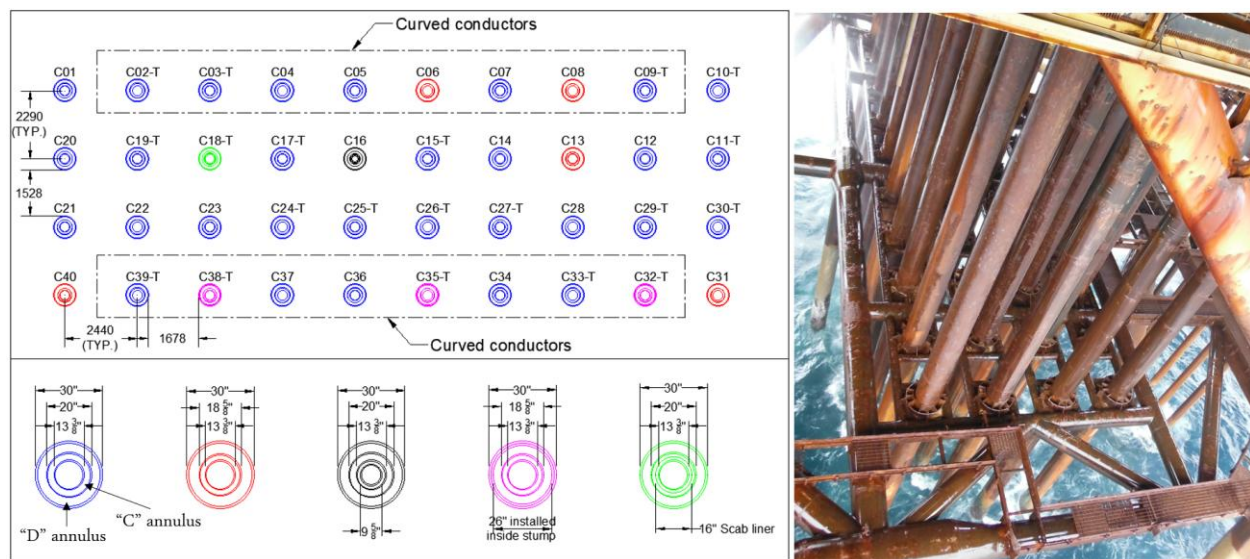


Figure C.2: North Cormorant conductor layout [76]

Important for the calculation of concepts is the mass estimation of the conductors. To establish a conservative design basis, the worst case must be determined, corresponding to both the C and D annuli being fully grouted, although this is not always the case in practice. Based on an estimate by Allseas, a conductor with three casings and fully grouted annuli weighs 1222.7 kg/m, giving a worst-case weight of 145.5 tonnes for a 119 m conductor. Allseas further estimates 11 tonnes of marine growth per conductor, resulting in a total net weight of 156.6 tonnes. To determine the actual lift weight, Allseas applies a method based on offshore lifting standards suitable for early design phases. First, a contingency factor of 1.10 is applied to account for unknown additional weight. Second, a dynamic amplification factor (DAF) is applied to account for dynamic effects during the lift, including hoisting speed variation, vessel and crane motions, and environmental loading on the lifted object. While the determination of the DAF should in principle be based on a dynamic analysis for each phase of the lift, including the subsea lift and the passage through the splash zone, a conservative DAF of 1.5 is adopted for the early design phase. Applying both factors to the net weight gives a total lifting weight of 258.2 tonnes. Assuming the conductor is removed in four sections of 29.75 m each, each section weighs 64.6 tonnes in the worst case. Table C.1 summarises the conductor dimensions and the assumptions adopted above.

Table C.1: North Cormorant conductor dimensions and weight assumptions

Quantity	Value
Outer casing diameter	30 in (762 mm)
Intermediate casing diameter	20 in (508 mm)
Inner casing diameter	13 3/8 in (340 mm)
Casing wall thickness	3/4 in (19.05 mm)
Conductor cut length	119 m
Mass per unit length (C and D annuli grouted)	1222.7 kg/m
Conductor net weight	145.5 t
Marine growth weight	11 t
Weight contingency factor	1.1
Dynamic amplification factor (DAF)	1.5
Total lifting weight	258.2 t
Length per section (4 sections)	29.75 m
Total lifting weight per section	64.6 t

D Solution Combinations into Concepts

This appendix presents the combination of working principles from the morphological chart into the four concepts developed in Chapter 3.4.

Table D.1: Solution combinations

Solution		1	2	3	4	5
Function						
A	Attach to Crane	Direct crane attachment	Attachment through frame			
B	Vertical Movement	Crane hoist	Self climbing on conductor	Winch assisted	Buoyancy assisted	ROV assisted
C	Horizontal Movement	Crane slew	ROV assisted	Not required		
D	Pass Through Obstacles	Through top of jacket	Through jacket side opening	Disassemble & reassemble		
E	Deploy Over Conductor Surface	Radial expansion	Radial contraction	Wrap-around	Roll-on	Mechanical closing
F	Close Gripper (Actuation)	Pneumatic	Hydraulic	Tendon-driven	Conventional drive systems	Hybrid
G	Grip Surface	Grip internal surface	Grip external surface			
H	Conform to Surface	Soft elastomer pads	Suction cups	Sleeve-like structure	Bending/wrapping fingers	Expanding/contracting membrane/bladder
I	Number of gripper units	1	2	3	>3	
J	Generate Grip Force (Gripping Method)	Friction	Form closure/interlocking	Suction/vacuum	Magnetic adhesion	Hybrid
K	Maintain Grip	Continuous actuation	Self-locking mechanism	Self-tightening		
L	Support Gripper's Self-Weight (Stiffness Support)	Longitudinal locking (layer jamming)	Bulk locking (granular jamming)	Cable/tendon stiffening	External frame	Inflatable stiffening
M	Pass Conductor Guide Frame	Through radial gap	Frames removed prior to conductor removal	Releasing and re-grip	Serpentine crawling	Internal diameter gripper
N	Support During Downending	Gripper	Additional downending frame			
O	Release Conductor	Release actuation	Reverse actuation	Gravity-assisted	Manual release	Release mechanism
P	Retract Gripper	Crane hoist	Manual handling			

The following are the selected concepts for further investigation:

- **Concept 1:** Braided Constriction Sleeve (*A1-B1-C3-D1-E2-F4-G2-H3-I1-J1-K3-L3-M2-N2-O3-P1*)
- **Concept 2:** Torsional Constriction Sleeve (*A1-B1-C3-D1-E3-F4-G2-H3-I4-J1-K2-L3-M2-N2-O2-P1*)
- **Concept 3:** Particle Jamming Gripper (*A2-B1-C1-D1-E5-F5-G2-H5-I4-J5-K1-L2-M3-N1-O1-P1*)
- **Concept 4:** Internal Inflatable Gripper (*A1-B1-C3-D1-E1-F1-G1-H5-I1-J1-K1-L5-M5-N1-O1-P1*)

E Concept Calculations

This appendix documents the calculations conducted for each of the four proposed concepts. These calculations are not intended as validated proof that a given concept would work in reality, but rather as a simple and fast first-order method to screen the concepts for feasibility at the conceptual design stage.

Table E.1: Conductor section properties used in the concept calculations

Quantity	Symbol	Value	Unit
Conductor outer casing diameter	d_c	762	mm
Conductor outer casing radius	r_c	381	mm
Conductor inner casing diameter	$d_{c,i}$	340	mm
Conductor section length	L_c	29.75	m
Conductor outer surface area	A_c	71.2	m ²
Conductor mass	m_c	64.6	t
Gravitational acceleration	g	9.81	m/s ²
Conductor weight	W_c	634	kN
Friction coefficient	μ	0.3	-

All concepts are evaluated based on lifting the conductor section described in Table E.1. Since all concepts rely on friction to grip the conductor, a friction coefficient of 0.3 is adopted consistently across all calculations to keep the results comparable.

E.1 Concept 1 Calculations: Braided Constriction Sleeve

The following calculations follow the derivations of Lu et al. (2024) to determine the radial pressure generated by the braid on the conductor. First, the braid parameters are defined, presented in Table E.2. To remain consistent with the Chinese finger trap concept, rectangular straps are chosen.

Table E.2: Braid parameters for the braided constriction sleeve concept

Quantity	Symbol	Value	Unit
Strap width	w	20	mm
Strap thickness	t	8	mm
Strap sets	n	12	-
Gripping length	H	25	m
Number of full turns	N	14	-
Minimum breaking load	MBL	400	kN

Dyneema is selected as the strap material, which for the chosen strap width typically offers a minimum breaking load (MBL) of 400 kN. Twelve strap sets are selected, corresponding to 24 individual straps, arranged over a gripping length of 25 m with the braid forming 14 full helical turns around the conductor surface. It should be noted that these are arbitrary but realistic values used for these early-stage calculations.

The following assumptions are adopted. The braid deforms uniformly, maintaining the same diameter along the full gripping length with no localised necking pattern. The strap material is assumed inextensible, neglecting elastic deformation and energy storage.

The pitch p of the helices is first determined:

$$p = \frac{H}{N} = 1.79 \text{ m} \quad (\text{E.1})$$

When the helix is unwrapped, a right-angled triangle is formed, with S_1 being the strap length over one full turn and α the braid angle, as depicted in Figure E.1.

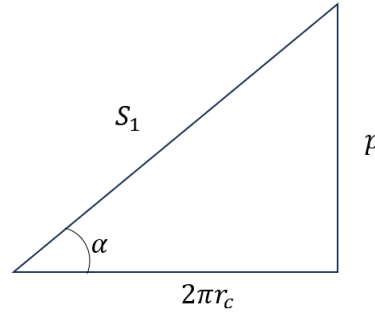


Figure E.1: Unwrapped geometry of one braid helix

The braid angle and strap segment length are then:

$$\alpha = \arctan\left(\frac{p}{2\pi r_c}\right) = 36.7^\circ \quad (\text{E.2})$$

$$S_1 = \sqrt{p^2 + 4\pi^2 r_c^2} = 2.99 \text{ m} \quad (\text{E.3})$$

The force conversion factor k is then determined according to Lu et al. (2024):

$$k = -\pi \cot \alpha / \left(wNS_1n - \frac{n(n-1)w^2}{\sin(2\alpha)} \right) = -0.422 \text{ m}^{-2} \quad (\text{E.4})$$

Assuming that the total axial force F equals the weight of the conductor W_c in order for it to get lifted the generated radial pressure P_{gen} is:

$$P_{gen} = |kF| = |kW_c| = 267 \text{ kPa} \quad (\text{E.5})$$

To compare this with the required radial pressure P_{req} , the actual contact area between the braid and the conductor is determined according to Lu et al. (2024):

$$A_{contact} = 2wNS_1n - \frac{2n^2w^2}{\sin(2\alpha)} = 19.95 \text{ m}^2 \quad (\text{E.6})$$

The required normal force $F_{N,req}$ to lift the conductor is estimated based on the Coulomb friction assumption:

$$F_{N,req} = \frac{W_c}{\mu} = 2110 \text{ kN} \quad (\text{E.7})$$

The required radial pressure is then:

$$P_{req} = \frac{F_{N,req}}{A_{contact}} = 105.8 \text{ kPa} \quad (\text{E.8})$$

Comparing this with the generated radial pressure gives the safety factor for lifting:

$$SF_{lift} = \frac{P_{gen}}{P_{req}} = 2.53 \quad (\text{E.9})$$

Furthermore, the actual force on the straps T_{strap} is determined to check their integrity under load. The full conductor weight W_c is carried by $2n$ straps oriented at the angle α . The strap force is then:

$$T_{strap} = \frac{W_c}{2n \sin \alpha} = 44.1 \text{ kN} \quad (\text{E.10})$$

This leads to a safety factor against strap breaking of:

$$SF_{break} = \frac{MBL}{T_{strap}} = 9.07 \quad (\text{E.11})$$

Finally, it is also useful to estimate the total strap length necessary to realise the braid:

$$S_{total} = 2nNS_1 = 1003 \text{ m} \quad (\text{E.12})$$

E.2 Concept 2 Calculations: Torsional Constriction Sleeve

The following calculations estimate whether the torsional constriction sleeve can generate sufficient friction to lift a conductor section. The gripping force is determined using a rope analysis method, following the approach applied to the band-wrapping gripper based on the iris mechanism by Lee et al. (2025). The tension amplification along the helical strap is described using the generalised Euler–Eytelwein, or capstan, relation derived by Konyukhov and Shala (2021), who show that a helix forms a geodesic on a cylinder, for which the tension ratio reduces to a closed form depending only on the friction coefficient, the helix curvature, and the contact length [61]. The starting parameters are presented in Table E.3. As with the previous concept, these are arbitrary but realistic values used for first-order screening.

Table E.3: Parameters for the torsional constriction sleeve concept

Quantity	Symbol	Value	Unit
Sleeve/ring radius	r_s	500	mm
Gripping length	H	25	m
Number of full turns	N	6	-
Number of straps	n_s	4	-
Driving torque on top ring	M	5	kNm
Minimum breaking load	MBL	400	kN

The following assumptions are adopted. The straps form a helix on the conductor with constant curvature and torsion, lying tangentially against the conductor surface at every point. The strap material is assumed inextensible, neglecting elastic deformation and energy storage. The non-contacting strap length between the rings and the conductor is neglected, and the friction force is assumed to act in the lifting direction, representing the most favourable first-order estimate.

Dyneema is again chosen for the straps, with the same minimum breaking load as in the first concept. The ring (or sleeve) radius r_s is purposefully chosen larger than the conductor radius to ease deployment. A similar gripping length H is adopted, with fewer straps than the first concept given the more complex actuation. The torque M is applied to the top ring to wrap the straps around the conductor and tension them.

The number of full turns must first be related to the required rotation of the top ring and to the total wrapping angle on the conductor. For this the methodology of Lee et al. (2025) is adapted, neglecting the strap ends that are not in contact with the conductor. Figure E.2, also adapted from the same source, illustrates the geometry. It represents a top view of the conductor and ring system, showing the strap end connections, one to the rotating top ring and one to the fixed bottom ring.

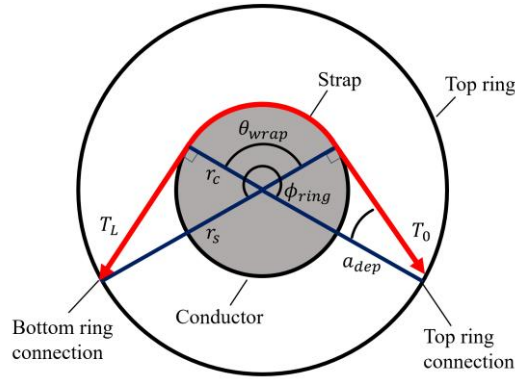


Figure E.2: Top view of the conductor and ring system for the torsional constriction sleeve, adapted from Lee et al. (2025)

The top ring has rotated through an angle ϕ_{ring} , and the strap contacts the conductor over the wrap angle θ_{wrap} . These angles relate to the number of ring rotations N_{ring} and the number of helix turns on the conductor N through:

$$N_{ring} = \frac{\phi_{ring}}{2\pi} \quad (E.13)$$

$$N = \frac{\theta_{wrap}}{2\pi} \quad (E.14)$$

The strap leaves the conductor tangentially towards the rings, with a departure angle:

$$\alpha_{dep} = \arccos\left(\frac{r_c}{r_s}\right) = 40.4^\circ \quad (E.15)$$

Accounting for the tangential lead-in at both strap ends, the geometry in Figure E.2 gives the coupling relation:

$$\theta_{wrap} = \phi_{ring} - \pi + 2\alpha_{dep} \quad (E.16)$$

Substituting the turn definitions yields the relationship between the rotation of the top ring and the number of helix turns formed on the conductor:

$$N_{ring} = N + \frac{1}{2} - \frac{\alpha_{dep}}{\pi} = 6.28 \quad (E.17)$$

Corresponding to a top ring rotation of:

$$\phi_{ring} = 2\pi N_{ring} = 2259^\circ \quad (E.18)$$

As in the first concept, the pitch p of the helix and the arc length l_{strap} of one strap over the gripping length are:

$$p = \frac{H}{N} = 4.17 \text{ m} \quad (E.19)$$

$$l_{strap} = N\sqrt{p^2 + (2\pi r_c)^2} = 28.8 \text{ m} \quad (E.20)$$

The strap length stored in each spool prior to rotation is therefore:

$$l_{spool} = l_{strap} - H = 3.8 \text{ m} \quad (\text{E.21})$$

And the total strap length required is:

$$S_{total} = n_s l_{strap} = 115.2 \text{ m} \quad (\text{E.22})$$

The constant curvature of the helix on the cylindrical conductor is:

$$\kappa = \frac{r_c}{r_c^2 + (p/2\pi)^2} = 0.65 \text{ m}^{-1} \quad (\text{E.23})$$

The driving torque M applied to the top ring, shared by n_s straps acting at radius r_s , generates an initial strap tension at the top ring connection:

$$T_0 = \frac{M}{n_s r_s} = 2.5 \text{ kN} \quad (\text{E.24})$$

As each strap wraps helically around the conductor under friction, its tension is amplified along its length according to the capstan relation of Konyukhov and Shala (2021). Integrating the resulting normal pressure along the strap gives the total radial normal force per strap:

$$F_N = \frac{T_L - T_0}{\mu} = \frac{T_0}{\mu} (e^{\mu \kappa l_{strap}} - 1) = 2323 \text{ kN} \quad (\text{E.25})$$

The total friction force available to lift the conductor is then:

$$F_{lift} = n_s \mu F_N = 2788 \text{ kN} \quad (\text{E.26})$$

Comparing this with the conductor weight gives the lift safety factor:

$$SF_{lift} = \frac{F_{lift}}{W_c} = 4.40 \quad (\text{E.27})$$

Finally, the fully amplified tension at the bottom ring, which the strap and ring must withstand, is:

$$T_L = T_0 e^{\mu \kappa l_{strap}} = 700 \text{ kN} \quad (\text{E.28})$$

Giving a safety factor against strap breaking of:

$$SF_{break} = \frac{MBL}{T_L} = 0.57 \quad (\text{E.29})$$

E.3 Concept 3 Calculations: Particle Jamming Gripper

The following calculations estimate whether the particle jamming gripper can generate sufficient friction to lift a conductor section. The holding force model follows Brown et al. (2010) and its extension to a two-jaw configuration by Amend and Lipson (2017). The starting parameters are presented in Table E.4. As with the previous concepts, these are arbitrary but realistic values used for first-order screening. The granular material parameters, including strain, bending modulus, and applied jamming stress, are taken directly from Amend and Lipson (2017).

Table E.4: Parameters for the particle jamming gripper concept

Quantity	Symbol	Value	Unit
Bag radius	L_b	1	m
Number of jaw pairs	N_j	6	-
Pinch force per jaw pair	F_p	500	kN
Applied jamming stress	σ_j	50	kPa
Material strain	ϵ	0.0013	-
Bending modulus	E	7400	kPa

The following assumptions are adopted. The interlocking contribution is set to zero, as the conductor is assumed a perfect cylinder. The suction contribution is set to zero, as the corroded and irregular conductor surface is assumed incapable of forming an airtight seal. The contact between each jamming pad and the conductor is approximated as two parallel lines, as described below.

Following the model of Brown et al. (2010), the total holding force F_H per jaw pair is the sum of a friction, an interlocking, and a suction contribution:

$$F_H = F_F + F_I + F_S \quad (\text{E.30})$$

Under the stated assumptions the interlocking term F_I and suction term F_S vanish, and the holding force reduces to the friction contribution F_F alone. Following the two-jaw extension of Amend and Lipson (2017), this friction contribution consists of the friction generated by the normal force from the jamming stress of the pad F_N and the friction generated by the mechanical pinch force of the jaw F_P :

$$F_F = 2\mu F_N(\sin \theta - \cos \theta) + \mu F_P \quad (\text{E.31})$$

Where θ is the contact angle between the pad and the conductor surface. The normal force from the jamming squeeze depends on the contact area between the pad and the conductor. The contact width d of the pad is estimated following Brown et al. (2010):

$$d = \sqrt{2\epsilon r_c(L_b - r_c)} = 24.8 \text{ mm} \quad (\text{E.32})$$

The original model of Brown et al. (2010) describes a spherical object, for which the contact forms an elliptical region. For the cylindrical conductor this contact is approximated as two parallel lines of width d and length equal to the bag diameter $2L_b$, as illustrated in Figure E.3.

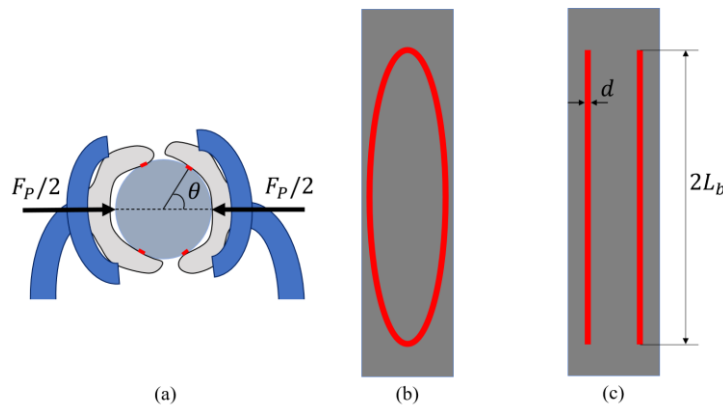


Figure E.3: Contact between a jamming pad and the conductor. (a) Top view with each jaw applying half of the pinch force. (b) The actual contact region between the pad and the cylinder forms an ellipse. (c) Simplification of the elliptical contact into two parallel lines

The contact area per pad is therefore:

$$A_j = 2(d \cdot 2L_b) = 0.1 \text{ m}^2 \quad (\text{E.33})$$

The normal force generated by the jamming stress σ_j acting over this contact area at contact angle θ is:

$$F_N = \sigma_j A \sin \theta \quad (\text{E.34})$$

The friction contribution from the jaw pinch force is:

$$F_{F,P} = \mu F_P = 150 \text{ kN} \quad (\text{E.35})$$

It should be noted that the jamming friction term is only positive above a critical contact angle, below which the pad interface will slip regardless of the applied normal force. This critical angle is:

$$\theta_c = \arctan\left(\frac{1}{\mu}\right) = 73.3^\circ \quad (\text{E.36})$$

Evaluating the friction contribution at the maximum contact angle of $\theta_{max} = 90^\circ$ gives the holding force per jaw pair:

$$F_F = 2\mu F_N (\sin \theta_{max} - \cos \theta_{max}) + \mu F_p = 153 \text{ kN} \quad (\text{E.37})$$

The total holding force of the system with N_j jaw pairs is then:

$$F_{H,tot} = N_j F_F = 918 \text{ kN} \quad (\text{E.38})$$

Comparing this with the conductor weight gives the lift safety factor:

$$SF_{lift} = \frac{F_{H,tot}}{W_c} = 1.45 \quad (\text{E.39})$$

During passage of a conductor guide frame, one jaw pair releases and regrips in turn, so that the conductor is momentarily supported by only $N_j - 1$ jaw pairs. The holding force in this configuration is:

$$F'_{H,tot} = (N_j - 1) F_F = 765 \text{ kN} \quad (\text{E.40})$$

Giving a reduced lift safety factor of:

$$SF'_{lift} = \frac{F'_{H,tot}}{W_c} = 1.21 \quad (\text{E.41})$$

The safety factor remains above unity even with one jaw pair released, confirming that the conductor can be held throughout the frame passing sequence, although with a reduced margin. This margin can be increased by adding further jaw pairs or increasing the pinch force.

E.4 Concept 4 Calculations: Internal Inflatable Gripper

As this concept grips and lifts only the outer casing, with the inner casings assumed already removed, the weight of the section is re-estimated, following the same approach as Appendix C but accounting for the outer casing alone. Table E.5 summarises this estimation together with the starting parameters relevant to the gripper.

Table E.5: Parameters for the internal inflatable gripper concept

Quantity	Symbol	Value	Unit
Conductor outer casing outer radius	r_c	381	mm
Casing thickness	t	19	mm
Conductor outer casing inner radius	$r_{c,i}$	362	mm
Steel density	ρ	7850	kg/m ³
Casing length	L_c	119	m
Outer casing net mass	$m_{c,o}$	41.4	t
Outer casing lifting mass (marine growth, contingency, DAF)	$m_{c,o,lift}$	86.5	t
Total lifting weight per section (4 sections)	$W_{c,o}$	212	kN
Water density	ρ_w	1025	kg/m ³
Gripping length	H	10	m
Deployment radial clearance	c	50	mm
Max depth relevant for gripping	D	110	m

The following assumptions are adopted. The bladder is assumed to inflate into uniform contact with the inner surface of the outer casing over its full contact length. The grip is generated purely through friction, following the Coulomb assumption used for the previous concepts.

The contact area between the inflated bladder and the inner surface of the outer casing is:

$$A_{contact} = 2\pi r_{c,i}H = 22.7 \text{ m}^2 \quad (\text{E.42})$$

The normal force required to lift the conductor section by friction is:

$$F_N = \frac{W_{c,o}}{\mu} = 707 \text{ kN} \quad (\text{E.43})$$

The internal bladder pressure required to generate this normal force over the contact area is therefore:

$$P_{lift} = \frac{F_N}{A_{contact}} = 31.2 \text{ kPa} \quad (\text{E.44})$$

However, the bladder must additionally overcome the external hydrostatic pressure acting on it at the gripping depth, since the internal pressure must exceed the surrounding water pressure before any net contact pressure develops. At the maximum gripping depth, the hydrostatic pressure is:

$$P_h = \rho_w g D = 1106 \text{ kPa} \quad (\text{E.45})$$

The total required internal bladder pressure is then:

$$P_b = P_{lift} + P_h = 1137 \text{ kPa} \quad (\text{E.46})$$

The required internal pressure is therefore dominated almost entirely by the need to oppose the hydrostatic pressure rather than by the gripping requirement itself.

The stress in the bladder membrane is then considered. Along the cylindrical sides of the bladder the internal pressure is reacted directly by the casing wall, so no hoop stress develops. At the top and bottom ends, however, once the gripper is lifted above the waterline there is no external pressure to react against, and the bladder ends must withstand the full pressure difference:

$$\Delta P_{end} = P_b - P_{atm} = 1036 \text{ kPa} \quad (\text{E.47})$$

Finally, the volume that must be pumped to inflate the bladder is estimated, with the volume change illustrated in cross section in Figure E.4.

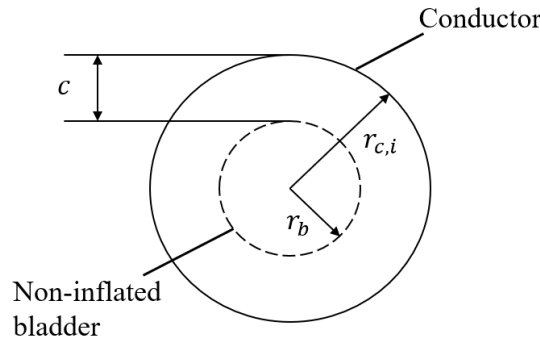


Figure E.4: Cross-sectional view of the bladder deployment

With a radial clearance c between the deflated bladder and the casing inner surface to allow deployment, the initial bladder radius is:

$$r_b = r_{c,i} - c = 312 \text{ mm} \quad (\text{E.48})$$

The volume change required to inflate the bladder from its initial radius to full contact with the casing is therefore:

$$\Delta V = \pi(r_{c,i}^2 - r_b^2)H = 1.06 \text{ m}^3 \quad (\text{E.49})$$

If an incompressible fluid is used to inflate the bladder, this volume change equals the volume to be pumped, approximately 1059 L. The use of pneumatic means to inflate the bladder is not considered, as at these pressures pneumatic actuation becomes complex and dangerous.

F Technical Drawings

This appendix presents the technical drawings of the prototype rings and braiding disc used for fabrication. The designs shown represent one of many viable configurations, selected for simplicity and ease of manufacture rather than as the definitive solution.

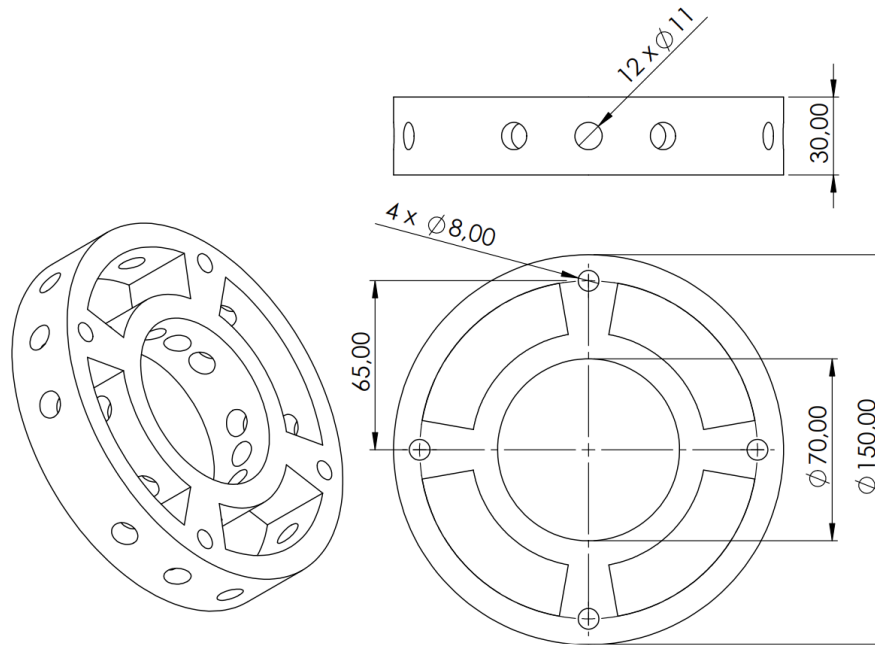


Figure F.1: Ring technical drawing

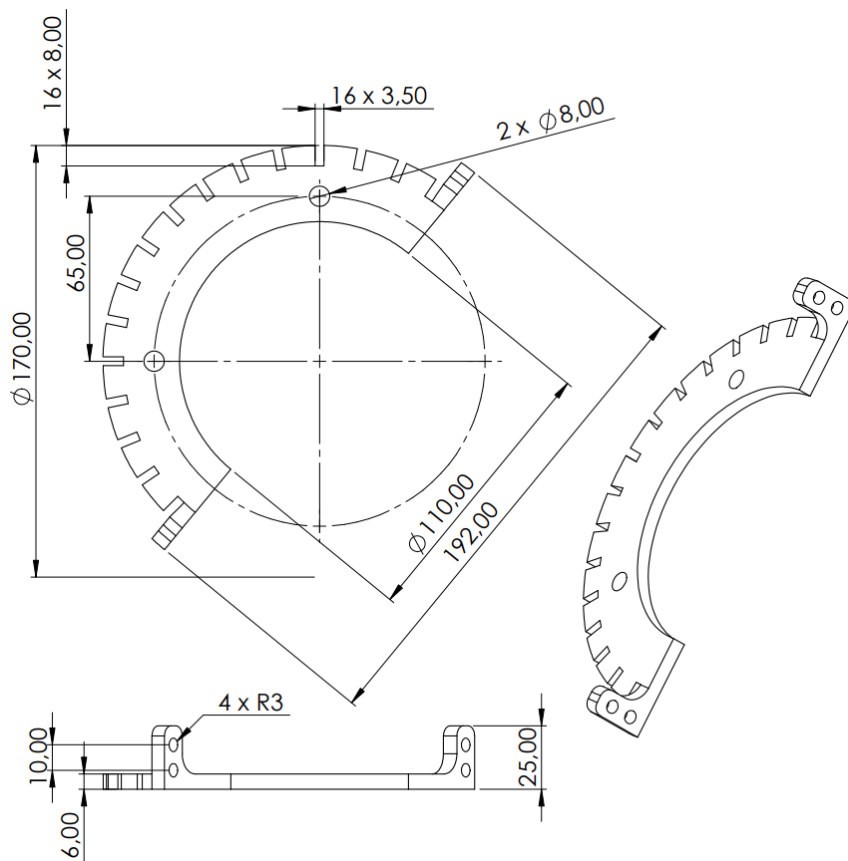


Figure F.2: Braiding disc half technical drawing

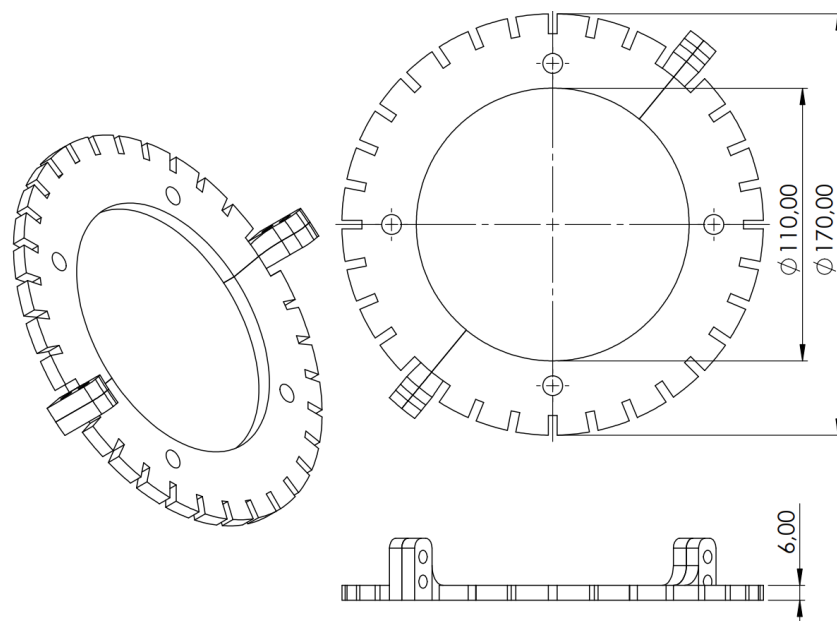


Figure F.3: Braiding disc technical drawing

G Individual Repetition Results

The following table presents the individual repetition results for all configurations, from which the means, standard deviations, and coefficients of variation were calculated.

Table G.1: Individual repetition results for all configurations

Group	Configuration	$W_{slip,1}$ [N]	$W_{slip,2}$ [N]	$W_{slip,3}$ [N]	$W_{slip,4}$ [N]	$W_{slip,5}$ [N]
Baseline	Baseline	41.3	39.0	30.7	29.2	42.1
A	$W_{ring} = 5.20$ N	180.2	147.4	154.7	167.9	194.5
A	$W_{ring} = 8.80$ N	263.0	290.3	340.4	287.5	351.4
B	$N = 2$	10.1	12.0	5.2	4.5	8.4
B	$N = 4$	460.0	539.2	432.4	567.3	580.7
C	$H = 200$ mm	Held the maximum applied load of 741.0 N with no observed slip				
C	$H = 600$ mm	18.0	45.7	23.4	26.7	14.9
D	$r_c = 20$ mm	13.0	12.3	12.0	14.1	12.4
D	$r_c = 25$ mm	28.2	24.5	27.3	23.2	21.1
E	$n = 2$	28.0	23.4	18.4	37.5	14.6
E	$n = 8$	50.1	43.2	47.8	42.9	46.7
F	Steel wire	88.4	90.4	88.0	87.4	88.4