Politecnico di Torino

Automotive Engineering – Autonomous and Connected Vehicles
a.a. 2023/2024
Graduation Session October 2024

Automated Design of Jointed Mechanical Structures

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Abstract

This project develops an application for the automated design of jointed mechanical structures. A three-step process is followed for its development, starting with literature review that includes theory regarding joint mechanical structures and the standards, ECSS and Eurocode, incorporated onto the software. This is followed by a schematization of the algorithm using Excel to enable having a quick and usable reference to have during the development of the software. This is finalized by the development of the software using Python and some of its specialized libraries for the proper construction of the algorithm and the graphical user interface. The result is a first version of the software capable of reproducing the results of the Excel schematization that provides a specialized interface with customization capabilities for further developments and an easier implementation of new dedicated functions relating to usability and expansion of the scope of the capabilities of the software.

Acknowledgements

This thesis has been possible to many people that have encouraged and supported me through the highs and lows of this path, and I dedicate it to them for all they have done.

First, I thank the Politecnico di Torino and Capgemini Engineering for the opportunity to have the experience of developing this thesis. With special thanks to my supervisors, professor Cristiana Delprete, Ferdinando Liquori, and Giovanni Pesare for seeing my potential to carry out the development of this research project and giving me their full support during this time.

To my family, for your unconditional love and support in every decision I have made and that has led me towards this path of life. The tools you have given me have helped me to grow and reach the person I have become and continue to develop to be.

To Carla, for being my friend and support since the very beginning of this journey that started almost three years ago. For being there for me when I needed to vent out and gain courage to face the challenges presented to me.

To Dhruv, for being a great friend and classmate, helping me through all the projects and exams we had to take, and for being there on those walks back home after nights of fun. Making every moment more enjoyable,

To Santiago, for your friendship and giving me the tools to look at life in a more relaxed way, letting me know that in the end everything falls into place and the outcome is always on the bright side.

To Edoardo, Ivan, and everyone else on the 4th floor of Capgemini, for making me feel welcome and a real part of the team, and for believing in my capabilities as an engineer and for making every second of the working hours enjoyable even if the work and projects pile up.

To everyone else who has been part of this journey, those unconditional friends from Mexico, those who are still in Torino, and those who are now in other places. For making this part of my life so enjoyable and allowing me to learn and grow as a person with you.

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List of Acronyms and Abbreviations

A	Gross cross-sectional area of the fastener
A_0	Smallest cross-section of fastener shank
A_3	Cross-sectional area at minor diameter of fastener thread
a_l	Edge distance (measured from hole axis to edge of plate)
A_{nt}	Net area subjected to tension
A_{nv}	Net area subjected to shear
A_s	Stress area
A_{sm}	Stiffness area
$B_{p,Rd}$	Punching shear resistance per bolt
d	Nominal fastener diameter; outside diameter of the thread
d_0	Diameter at smallest cross-section of fastener shank
d_2	Pitch diameter
d_3	Minor diameter of thread; for the true thread
D_h	Nominal diameter of hole in plate
d_s	Diameter used for stress calculation
d_{sha}	Shank diameter; for necked-down fasteners
d_{sm}	Diameter used for stiffness calculation
d_{uh}	Effective diameter of friction under head or nut
$d_{uh,brg}$	Outer diameter of bearing area; either under head or under nut
e_1	End distance from the center of a fastener hole to the adjacent edge of any part, measured in the direction of load transfer
e_2	Edge distance from the center of a fastener hole to the adjacent edge of any part, measured at right angles to the direction of load transfer
E_b	Young's modulus of fastener
F	Externally applied shear load
F_A	Axial force transmitted by joint

$F_{b,Rd}$	Design bearing resistance per bolt
$F_{b,Rd,res}$	Design bearing resistance of the resin of an injection bolt
$f_{b,res}$	Bearing strength of the resin
F_K	Clamping force acting at the interstice between two plates
$F_{p,Cd}$	Design preload of bolts
F_Q	Shear force transmitted by joint
$F_{r,max}$	Plate residual compression at maximum preload
$F_{s,Rd}$	Design slip resistance per bolt at the ultimate limit state
$F_{s,Rd,ser}$	Design slip resistance per bolt at the serviceability limit state
$F_{t,Ed}$	Design tensile force per bolt for the ultimate limit state
$F_{t,Ed,ser}$	Design tensile force per bolt at the serviceability limit state
$F_{t,Rd}$	Design tension resistance per bolt
F_{th}	Axial force transmitted by the thread
f_u	Ultimate tensile strength of the fastener
F_V	Nominal preload
$F_{v,Ed}$	Design shear force per bolt for the ultimate limit state
$F_{v,Rd}$	Design shear resistance per bolt
f_y	Yield strength of the fastener
F_Z	Maximum preload loss due to relaxation
$F_{\Delta T-}$	Minimum (most negative) increments in the fastener load due to thermo-elastic effects
$F_{\Delta T+}$	Maximum (most positive) increments in the fastener load due to thermo-elastic effects
GUI	Graphic User Interface
h	Height of the basic thread
JADIES	Joint Analysis Design Interactive Efficiency Software
k_2	Factor depending on bolt type
k_l	Factor defined in Table 29

K_R	Reduction factor for ultimate strength of the material in the net tension section
k_s	Factor defined in Table 24
$k_{s,res}$	Factor defined in Table 32
k_t	Factor defined in Table 31
L_{eng}	Engaged length of the thread
M_{app}	Peak torque applied to the fastener or nut during tightening
M_{jd}	Design moment resistance of a joint
M_{nom}	Nominal applied torque
MoS	Margin of safety
M_P	Prevailing torque
p	Pitch of thread
p_1	Spacing between the center of fasteners in a line in the direction of the load transfer
p_2	Spacing measured perpendicular to the load transfer direction between adjacent lines of fasteners
R_A	Utilization ratio due to the nominal preload
R_S	Shear strength ratio
sf_{gap}	Gapping safety factor
sf_{ult}	Ultimate strength safety factor
sf_y	Yield safety factor
T	Externally applied axial load
X	Number of shear planes
ε	Uncertainty factor of tightening method given in Table 17
λ	Under-head bearing angle
σ	Bending stress
σ_{br}	Bearing stress
σ_{uh}	Bearing stress under-head

Φ	Basic force ratio
Φ_n	Force ratio of a concentric joint including the loading plane factor effect
ω	Torque wrench accuracy

1. Scope and Introduction

1.1. Purpose & Scope

The purpose of this project is the development of software capable of calculating all margins of safety related to a joint according to the procedures specified in ECSS and Eurocode. The software is given the name: JADIES, which means Joint Analysis Design Interactive Efficiency Software. The documentation presented covers an introduction to the most common mechanical fasteners and modes of load, but not all of them fall under the scope of the work done.

The scope of the work:

- ECSS and Eurocode
- Removable fasteners (bolts and screws), which may or may not include locking devices (nuts, washers, and inserts).
- Concentrically loaded joints.

The concepts and formulas related to eccentrically loaded joints, and joints with rivets and pins are mentioned in this document but are not present in the first version of the software as they fall out of scope. They are to be added to the software in a further version of it.

1.2. Introduction

This section provides an overview of the standards ECSS and Eurocode, and their respective methodology. Including a full description of concepts that are out of the scope of the project, to have a full understanding of the literature and have a reference for future developments, as mentioned in section 6.2.

1.2.1.ECSS

The European Cooperation for Space Standardization (ECSS) is a collaboration the European Space Agency (ESA), the European space industry represented by Eurospace, and several space agencies, to develop and maintain a single set of standards for use in all European space activities. For this project we are concerned with the Threaded Fasteners

Handbook [9], which establishes specifications for the design of joints with threaded fasteners for space applications.

The methodology of ECSS divides the joints into five main categories:

- Concentric Axially Loaded Joints: for this type of joint the line of action of the applied load on the joint is parallel to and coincident with the longitudinal axis of the fastener.
- Eccentric Axially Loaded Joints: for this type of joint the line of the applied load on the joint is parallel to the longitudinal axis of the fastener, but not coincident with it. This results in a prying action that occurs between the clamped parts of the joint such that bending loads are introduced under the bolt head and in the shank.
- Shear Loaded Joints: for this type of joint the line of action of the applied load on the joints is in the plane of the clamped parts immediately adjacent to the fastener, and therefore normal to the longitudinal axis of the fastener. This type of joint is further subdivided into friction-grip and bearing joints.
- Combined Loaded Joints: for this type of joint more than one system of loads acts on the joint relative to the axis of the fastener. This type of joint is further subdivided into friction-grip and bearing joints.
- Low Duty Joints: for this type of joint, it is subjected to loads that can fit into one of the previously mentioned categories, however they form an independent category since they have small external loading with respect to the fastener strength.

1.2.2.Eurocode

The Eurocode is a series of ten standards used in the design of buildings and other civil engineering works. For this project we are concerned with the standard EN-1993-1-8:2005 [10], which establishes specifications for the design of joints for steel structures.

The standard contemplates connections made with various kinds of fasteners that include bolts, rivets, and pins. Bolts are classified according to the metric standard and joints are divided into two types: connections subject to shear, which is further subdivided into three categories, and connections subject to tension, which is subdivided into two categories.

The bolted connections subjected to shear [10] are designed as one of the following categories:

- Category A: Bearing type
    - Bolts from class 4.6 up to and including class 10.9 should be used.
    - There are no requirements for preloading or special provisions for contact surfaces.
    - The design ultimate shear load should not exceed the design shear resistance nor the design bearing resistance.
- Category B: Slip-resistant at serviceability limit state
    - Preloaded bolts of classes 8.8 and 10.9 should be used.
    - Slip should not occur at the serviceability limit state.
    - The design serviceability shear load should not exceed the design slip resistance.
    - The design ultimate shear load should not exceed the design shear resistance nor the design bearing resistance.
- Category C: Slip-resistant at ultimate limit state
    - Preloaded bolts of classes 8.8 and 10.9 should be used.
    - Slip should not occur at the ultimate limit state.
    - The design ultimate shear load should not exceed the design slip resistance nor the design bearing resistance.
    - Additionally for a connection in tension, the design plastic resistance of the net cross-section at the bolt holes should be checked at the ultimate limit state.

The bolted connections subjected to tension [10] are designed as one of the following categories:

- Category D: Non-preloaded
    - Bolts from class 4.6 up to and including class 10.9 should be used.
    - There is no requirement for preloading.

- This category should not be used if the connections are frequently subjected to variations of the tensile loading.
- An exception can be made for connections designed to resist normal wind loads.

- Category E: Preloaded
    - Preloaded bolts from classes 8.8 and 10.9 with control tightening should be used.

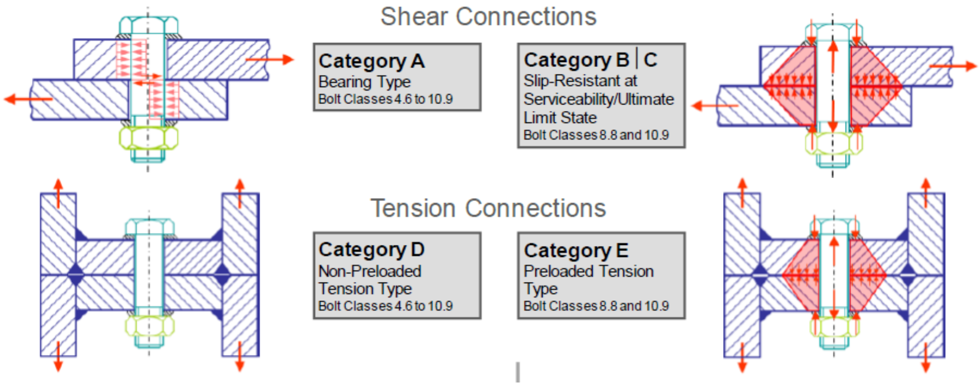

Figure 1 - Eurocode Connection Categories

2. General Design Information

This section presents the definition of the basic concepts pertaining to this project, including concepts used throughout the document and the general definitions of the most common types of fasteners and their parts considered under the scope of this project.

2.1. Definition of Concepts

- Clamp load: the tensile force applied to a fastener.
- Clamped parts: these are the regions of a joint that are compressed by the fastener.
- Fastener: any threaded component used to clamp parts creating a mechanical joint, such as bolts or screws.
- Plate: the clamped part that transmits loads from a threaded fastener to other regions of the structure.
- Prevailing torque: it is the torque provided by the locking device. This torque needs to be overcome before the threaded fastener can be loosened.
- Serviceability limit state: it is the state of design beyond which a joint loses operational serviceability for the actual service load to which it is subjected.
- Ultimate limit state: it is the state of design beyond which a joint fails due to being subjected to loads that exceed the ultimate strength of at least one of its components.

2.2. Mechanical Fasteners and Locking Devices

This section presents a description of the most common mechanical fasteners, which includes the standard definition of their geometric characteristics, the more often seen forms, and their applications.

2.2.1. Threads

A thread is a helical groove on the external surface of a shaft or the internal surface of a hole. The helix converts the rotational movement between the nut and bolt into linear movement along the center axis of them. The angled contact surface creates a wedging action when tightened, which generates a tension in the bolt and a resulting clamp force that holds the clamped parts together.

The clamping force also generates friction between the mating surfaces. This friction prevents the movement in the threads and keeps the fastener from backing off and loosening. The helix angle of a thread is set so that the friction forces, generated during tightening, should always be greater than the restoring force working against the wedging action [12].

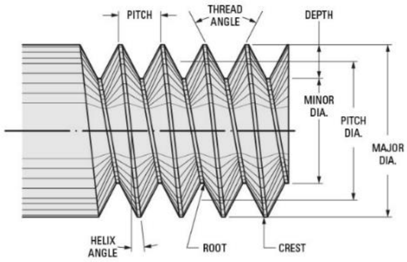

Figure 2 - Thread Geometry

Figure 2 presents the geometry of a screw thread, highlighting the main characteristics considered when choosing a threaded fastener. The definition of each is presented below:

- The pitch is the distance between contiguous thread forms measured parallel to the thread axis.
- The major diameter is the largest diameter of the thread.
- The minor or root diameter is the smallest diameter of the thread.
- The pitch or effective diameter is the theoretical diameter between the major and minor diameters.
- The depth is the radial distance between the crest and the root of the thread.
- The crest is the topmost part of the thread.
- The root is the bottom part of the thread.
- The thread angle is the angle between the sides of the thread measured in an axial plane.
- The helix angle is the angle between
- The lead, not shown in the figure, is the distance the fastener moves parallel to its axis in a single turn.

There exist six main types of thread forms: Sharp V threads, Whitworth threads, Buttress threads, Square threads, ACME threads, and Worm threads. These are illustrated in Figure 3.

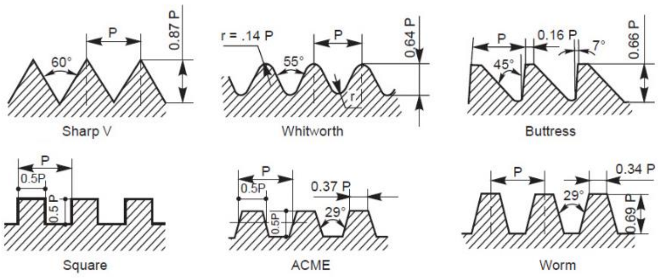

Figure 3 - Thread Forms

The variations in thread geometry are given by the different application each type is used for:

- Sharp V threads are used for fastening applications for frictional connections, such as screw connections with anti-fatigue shafts, and tension anchors.
- Whitworth threads are used for applications such as general brass work, gas fittings, brass tubing, and antenna bases.
- Buttress threads are used for vertical lifting applications where the load is supported on the upper flank of the screw thread, such as lifting screw for heavy lifting equipment, power screws in machines with high loads, and transmission threads in turning and milling machines.
- Square threads are used for applications where high-power transmission efficiency and high-load capacity are necessary, such as leadscrews and jackscrews.
- ACME threads are used for power transmission and load-carrying applications, such as leads screws in split nuts, jacks, vices, and machine equipment.
- Worm threads are used for the transfer of force and motion between two perpendicular shafts, such as speed reducers in small motors, automatic door systems, and adjustment systems for musical instruments.

There exist two basic types of thread, which include the coarse and the fine thread. The coarse thread has a larger pitch, meaning it has fewer threads per axial distance. It also has a larger thread form relative to the diameter of the screw. The main advantages of coarse threads are a greater resistance to stripping and cross threading due to their greater flank engagement, and a faster installation as they require fewer turns per unit length. They are most common in industrial applications, including construction, aviation, and military.

The fine thread has a smaller pitch, meaning it has more threads per axial distance. It also has a smaller thread form relative to the diameter of the screw. The main advantages of fine threads are that they are stronger due to a larger stress area for the same diameter of the thread, they are less likely to loosen due to vibrations as they have a smaller helix angle, allowance for finer adjustment, and they develop a larger preload with less tightening torque. They are most common in industrial applications, for machine parts that have limited space, such as hydraulic systems and mechanical transmission parts.

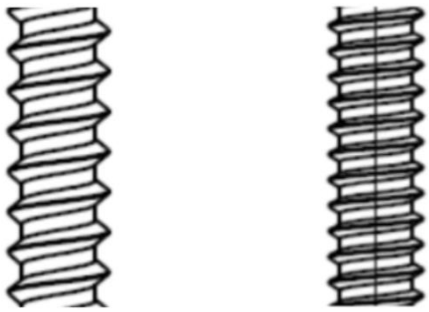

Figure 4 - Coarse Thread (Left) and Fine Thread (Right)

2.2.2. Locking Devices

A nut is a device with an internally threaded hole with its primary functions being the application of tension to the shaft of the bolt and compression to the clamped parts [12].

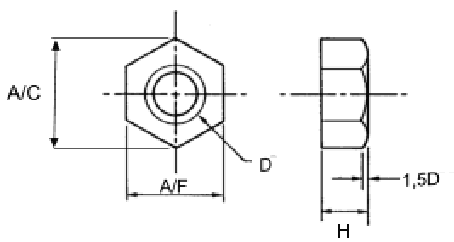

Figure 5 - Nut Schematic

The nut is characterized by its thread characteristics, described in section 2.2.1, and outer dimensional characteristics shown in Figure 5 and described below:

- D is the nominal hole diameter.
- A/C is the external diameter across corners.
- A/F is the external diameter across flats.
- H is the height.

An insert is a part with a threaded hole that is embedded in a weaker material to increase the loads that can be transmitted through the joint. It is normally a metallic part [12].

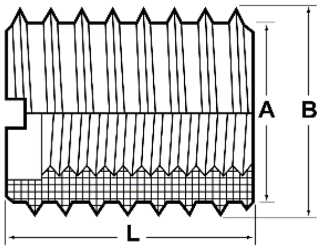

Figure 6 - Insert Schematic

2.2.3.Washers

A washer is a thin plate with a hole in the middle that is commonly used to distribute the load of a threaded fastener, such as a bolt or nut [12].

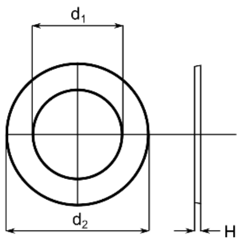

Figure 7 - Washer Schematic

The washer is described by the fastener size and three main dimensions of it:

- d_1 is the internal or hole diameter.
- d_2 is the external diameter.
- H is the thickness of the washer.

2.2.4.Bolts

A bolt is a threaded mechanical fastener used to non-permanently join two parts. It is used together with nuts and washers for a more effective fastening and distribution of the load [12].

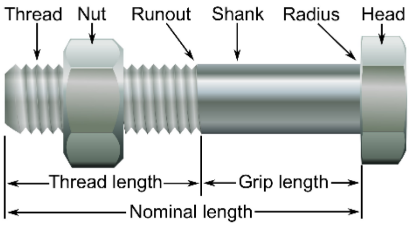

Figure 8 - Bolt Schematic

A bolt consists of two parts:

- Head: it is the upper portion of the bolt; it may have different shapes depending on the purpose for which the bolt is required, the most common shapes are described in section 2.2.4.1.
- Shank: it is the cylindrical portion of the bolt. The tail end of the shank is threaded to a sufficient length to allow for the nut to be engaged on it, also the end of the shank is chamfered to allow for an easy insertion in a hole and nut.

2.2.4.1. Types of Bolt Heads

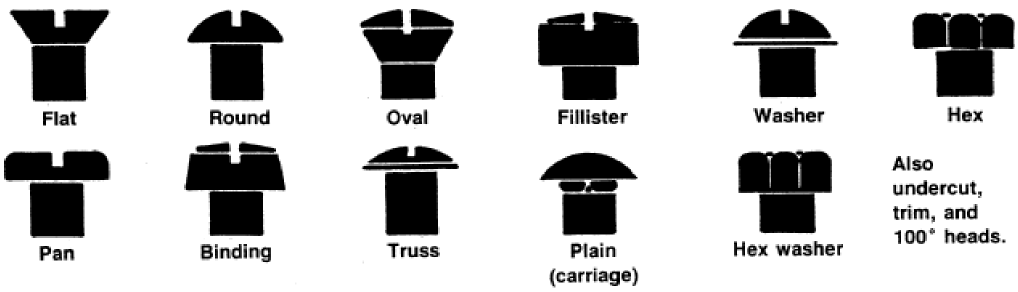

Figure 9 - Bolt Heads

- A flat bolt head is a counter shank head with a flat top that is used in areas of limited clearance or when a low-profile fastener is desired.
- A round bolt head features a domed head that is used in decorative applications.

- An oval bolt head is a counter shank head with a rounded head top that is used when there is a need to minimize the risk of snagging or damaging the material being joined.
- A hex bolt head features a hexagonal top that is used in a wide variety of applications, including automotive, construction, and machinery. They are easy to grip with a wrench or socket, and the shape allows for the application of a high torque.
- A pan bolt head is a slightly rounded head with a short vertical side that is used when there is a need to fasten metal materials that need a smooth surface and an increased bearing surface.
- A truss bolt head is an extra-wide head with a rounded top that is used when there is a need to fasten materials to a curved surface, due to its wide diameter that distributes the load evenly.
- A hex washer bolt head features a hexagonal top with a built-in round washer at the bottom, the washer is used to increase the bearing surface and ensure safety. It is used for materials that need large holes.
- A slotted hex washer bolt head features a hexagonal top with a built-in round washer and slot, the slot allows the bolt to be tightened or loosened with a screwdriver. It is used in applications that need frequent adjustments.
- A socket cap bolt head features a small cylindrical head with a socket driver. It is used in applications that require high torque application and a low-profile fastener is desired, such as the automotive industry.
- A button bolt head features a round top that sits flush with the material surface. It is used for decorative applications.

2.2.4.2. Injection Bolts

An injection bolt is a bolt in which the cavity produced by the clearance between the bolt and the wall of the hole is filled up with a two-component resin [12]. It may be used in shear connections where slip is not allowed as an alternative to fitted bolts and for high strength friction grip bolts.

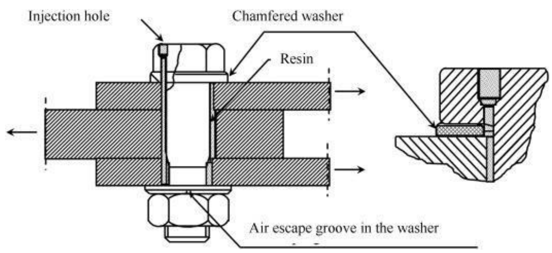

Figure 10 - Injection Bolt Schematic

The bolt has an injection hole through which the resin is added into the cavity allowed by the clearance between the bolt and the wall of the hole. The nut-side washer has an air escape groove that allows the exit of the displaced air from the above cavity, and the bolt head-side washer is chamfered for two main reasons: to prevent it from digging into the surface it is being applied to, and to reduce the amount of stress on the washer itself, which helps in prolonging its lifespan.

2.2.5.Screws

A screw is a threaded mechanical fastener that is like a bolt but differentiated by the fact that the entire length of the shaft of the screw is fully threaded. This fact leads to a differentiation in application with the bolt passing through an unthreaded hole and secured with a nut, while a screw passes through a threaded hole in one of the clamped parts. Due to this, the screw relies only on friction at the threads to remain securely fastened [12].

There are two main types of screws: machine screws and self-tapping screws, which are detailed in the following subsections.

2.2.5.1. Machine Screws

Machine screws have a standard thread on a parallel shaft, they are designed to be screwed into a tapped hole in a part. There are many types of screws, depending on their application and head shape, the five most common are presented below:

- Hex screw: it is the most common type; it is remarkably like a hex bolt but without the unthreaded portion of the shank.
- Countersunk head screw: a tapered head designed to be countersunk so that it lies flat with the surface of a part.

- Cap head screw: a barrel shaped head with a socket, designed to fit into a counterbored hole.
- Pan head screw: a domed head fitted with a socket.
- Grub screw: a type of set screw which has its entire outer surface threaded, with no head, allowing it to be screwed all the way through a hole to lock against a part below the hole.

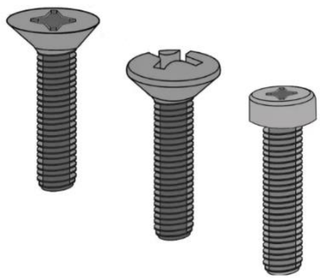

Figure 11 - Machine Screws

2.2.5.2. Self-tapping Screws

Self-tapping screws are designed to cut their own thread in sheet metal, wood, or plastic. They often have a sharp pointed end and sharp threads on a tapered shaft, some may have a section of unthreaded shank like a bolt. Common types include wood screws and self-tapping screws.

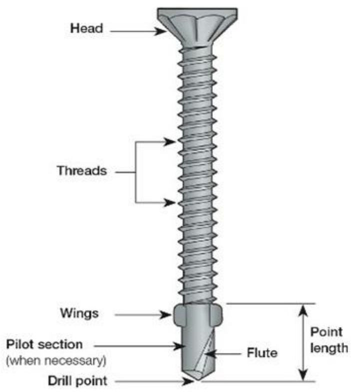

Figure 12 - Self-tapping Screws

2.2.6. Rivets

A rivet is a permanent mechanical fastener. Prior to installation it consists of a smooth cylindrical shaft with a head on one end, the end opposite to the head is called the tail. During installation, the rivet is placed on a hole and the tail is deformed, so that it expands to about 1.5 times the original shaft diameter, holding the rivet in place [12].
As the is effectively a head on each end of the installed rivet, it can support tension loads. However, it is more capable of supporting shear loads.

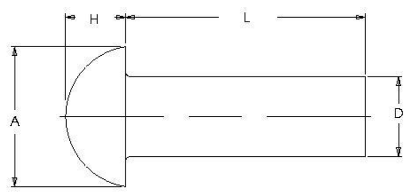

Figure 13 - Rivet Schematic

The rivet is described by four main dimensions:

- L is the length of the shaft.
- H is the height of the head.
- D is the diameter of the shaft prior to installation.
- A is the diameter of the head.

2.2.7.Pins

A pin is an unthreaded mechanical fastener that is designed to be inserted through preformed holes [12]. The most common types of pins are:

- Dowel pin: a simple solid cylinder that is inserted through a hole. Typically, it has a slight interference with the hole so that the compression of the pin and the resulting friction holds the pin firmly in place. They also have a chamfered end to ease insertion.
- Slotted pin: a pin formed from sheet material rolled into a partial cylinder with a chamfer at each end. The chamfer allows the pin to be forced into a hole with a smaller diameter than the relaxed pin diameter. The gap allows the pin to compress to fit into the hole. The sprung nature of the pin holds it securely in the hole.

- Coiled or spiral roll pin: it is like a slotted pin, but the sheet material is coiled by more than one complete revolution, therefore it is more heavy duty.
- Grooved pin: a solid pin with three grooves swaged along its length or a part of it. It has higher elasticity than a dowel pin and it is stronger than a spring pin. It is also driven to a tight hole.
- Split pin: a pin that is bent in half so that both ends may be inserted through the same hole. It is typically manufactured from a half-circular profile so that the two ends together form a circle which fits into the hole. The bent end is formed into an enlarged end and the double end can be bent outwards to prevent the pin from being removed.
- Cotter: a wedged or tapered pin that is driven into a hole. The tapered nature means that it is compressed as it is driven into the hole, causing friction which prevents it from loosening.

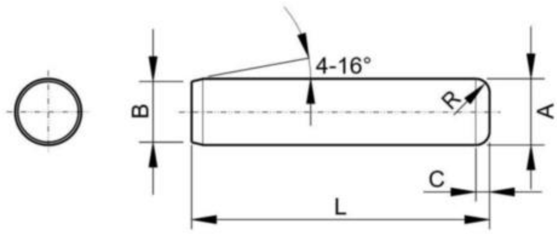

Figure 14 - Dowel Pin Schematic

3. General Joint Information

This section presents a definition for friction and bearing joints, and their behavior under load. It also covers the different failure modes that these joints can incur in, with a description of each, as an introduction to the concepts covered by the ECSS and Eurocode methodologies presented in sections 4.1 and 4.2.

Failure Mode – Friction Joint	**Section**
THREAD FAILURE DURING TIGHTENING	3.1.1.1
THREAD PULL-OUT DUE TO SHEAR	3.1.1.2
CRUSHING OF PLATE	3.1.1.3
PLATE SEPARATION (Slippage & Gapping)	3.1.1.4
FASTENER HEAD SEPARATION (Slippage head)	3.1.1.6
GAPPING (min. preload + exceeded screw load due to head separation)	3.1.1.6
FASTENER BENDING FAILURE (due to under head load)	3.1.1.6.1
COMBINED LOADS ON FASTENER (due to exceeded screw load)	3.1.1.6.2

Table 1 - Modes of Failure of Friction Joints

Failure Mode – Bearing Joint	**Section**
SHEAR FAILURE OF FASTENER	3.2.1.1
TENSION FAILURE OF FASTENER	3.2.1.2
COMBINED FAILURE OF FASTENER (Shear + Tension)	3.2.1.3
NET TENSION SECTION FAILURE OF PLATE	3.2.1.4
BEARING FAILURE OF PLATE	3.2.1.5
SHEAR-OUT FAILURE OF PLATE	3.2.1.6

Table 2 - Modes of Failure of Bearing Joints

3.1. Friction Joint

A friction joint or friction-grip joint is a joint that uses the friction generated between the clamped parts to transmit the load [2]. The amount of force that can be transmitted by the means of friction force is proportional to the clamp load of the fastener. The strength of the joint comes from the mechanical properties of the fasteners, the materials used in the structures, and the interlocking fastener threads with mating threads, and their preload distribution in the two clamped parts.

It has the main advantage that, if slip does not occur, the fastener only feels the tensile preload in the case of symmetric joints. In the unsymmetrical case of a single shearing surface, the deformation of the joint creates a small tension load in the fasteners, which is normally negligible [9].

Friction joints have no free play, making them preferable for uses cases where limited movement and wear is required. For friction joints, it is possible to go from slippage into bearing. These applications include bridges, buildings, aircraft, vehicles, and industrial equipment.

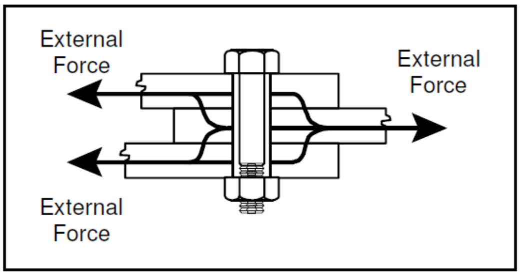

Figure 15 - Loading of Friction Joint

3.1.1. Failure Modes

During their service life, friction-grip joints can undergo several modes of failure can occur due to distinct factors, which include inherent design issues and external factors such as environmental conditions. The modes of failure for friction joints that fall under the scope of this work are presented in the following subsections.

3.1.1.1. Thread Failure during Tightening

This mode of failure occurs when the tightening torque exceeds the maximum stress that the thread profile of the bolt can sustain. It can lead to different types of failures, including thread pull-out (3.1.1.2), head snapping, or thread damage, which damage the integrity of the fastener and can result in eventual catastrophic failure.

To prevent this failure mode, it is important to consider four factors:

- The material of the fastener, as the torque that can be applied to the fastener, is dependent on the properties of the material it is made of.
- The material of the clamped parts, for the same reasons it is important to consider the material of the fastener, the properties of the material of the clamped parts determine the amount of load they will tolerate prior to any damaging.
- The drive of the fastener, as different shapes of the driving slot result in a different allowable tightening torque. Some of the most common drive styles are listed below in order of the most to the least in terms of torque-taking ability and resistance to stripping:
    - Star or torx
    - Internal hex
    - Robertson or square
    - Pozidrive
    - Phillips
    - Slotted
- The type of thread, as from the two types of thread (coarse and fine), the fine thread can withstand more torque. For more details on this refer to section 2.2.1.

Figure 16 - Thread Failure during Tightening

3.1.1.2. Shear Thread Pull-Out

This mode of failure occurs when the engaged length of the thread is insufficient, leading to a pull-out of the thread due to an excessive shear load [5]. There are two subcategories under this mode of failure, as a distinction is made between the failure of female threads and the failure of male threads. A description for both cases is presented in the following subsections.

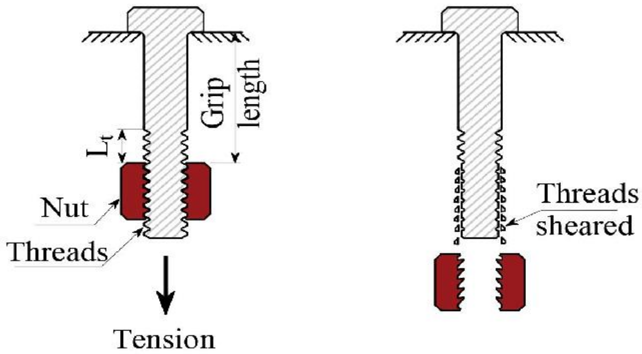

Figure 17 - Thread Pull-Out due to Shear

3.1.1.2.1.Failure of Female Threads

The failure of a female thread is caused when the shear strength of the material of the thread is exceeded by the applied load. In the case that the locking device providing the female thread is too weak for the application, a proposed solution is the use of threaded inserts of higher strength is recommended. This allows for a better distribution of the load and avoidance of a possible catastrophic failure.

3.1.1.2.2.Failure of Male Threads

The failure of a male thread is a less common phenomenon as, by norm, the fastener should have a higher strength than the locking device used in the joint. In case thread pull-out of the fastener appears, this is likely to be caused by the incorrect selection of the fastener or poor material properties, which may have been affected by external factors such as thermal fluctuation.

3.1.1.3. Crushing of Plate

This mode of failure occurs when an excessive compression load (or excessive preload) is applied to the fastener, causing the under head or under nut surface to crush the plate, leaving a dent and mark on the material. This phenomenon can cause the loss of mechanical properties of the plate due to deformation or failure of the joint due to loosening as a gap might be created due to the crushing of the plate.

3.1.1.4. Plate Separation due to Slippage

This kind of failure occurs due to the combination of two factors. The first is the loss of the friction that causes the clamped parts to stay together, this commonly occurs through a transversal load that exceeds the resistance provided by the friction, causing slippage. The second is the influence of the tension load on the fastener that causes gaping, because of the slippage phenomenon. The combination of these two actions causes the separation of the plate.

3.1.1.5. Plate Separation due to Gapping

The phenomenon of gaping occurs when the friction contact between the clamped parts is overcome and the compression force of the joint weakens. This can be caused by an externally applied tension load or variations in material properties, manufacturing tolerances, or installation errors. This leads to a physical gap or a separation of the joint in the direction parallel to the axis of the fastener. It is undesirable behavior as it may lead to permanent failure of the joint.

3.1.1.6. Fastener Head Separation due to Slippage

The phenomenon of slippage occurs when the clamping load provided by the fastener is insufficient to prevent the movement of clamped parts. It can reduce the preload and lead to failure of the connection, and it often results in the deformation of the joint, which can impact on the performance and service life.

Due to the fastener head separation failure the bolt is subject to bending load and induces the extra tensile load on it. So, when this failure has been considered, is necessary calculate the margin of safety of the bending to the bolt and re-calculated the margin of safety of gapping and combined load (tension + shear).

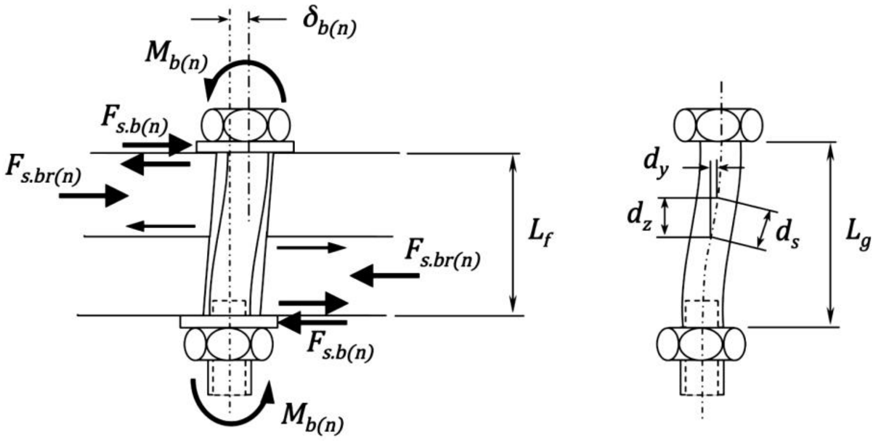

Figure 18 – Joint In-Plane loads reacted into a bolt

3.1.1.6.1.Fastener Bending

This kind of failure occurs when the fastener is subjected to bending moments due to a force being applied at a perpendicular angle to its axis [5]. This can happen due to several factors, such as misalignment of the clamped parts, offset loading, or over-tightening of the fasteners. The bending moments created due to these conditions lead to an increased stress concentration at the weakest point of the fastener, resulting in fracture or fatigue crack initiation.

3.1.1.6.2.Combined Failure (Tension + Shear) of Fastener

This mode of a failure occurs through a combination of the phenomena explained in sections 3.1.1.6 and 3.1.1.6.1.

3.2. Bearing Joint

A bearing joint or bearing-type joint is a joint that depends on the contact between the fastener and the clamped parts to transmit the load [2]. In this type of joint, the fastener is tightened only enough to keep the members of the joint in full contact, making the effects of friction between the clamped parts negligible. In a bearing joint, the fasteners are subjected to shear, tension, and the clamped parts to bearing stresses.

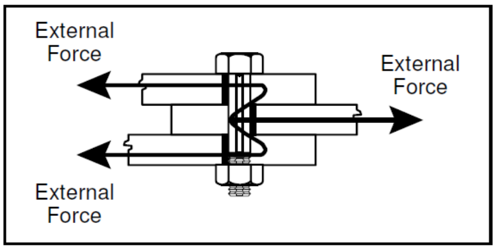

Figure 19 - Loading of Bearing Joint

3.2.1.Failure Modes

During their service life, bearing-type joints can undergo several modes of failure can occur due to varied factors, which include inherent design issues and external factors such as environmental conditions. The modes of failure for bearing joints that fall under the scope of this work are presented in the following subsections.

3.2.1.1. Shear Failure of Fastener

This kind of failure is characterized by the tearing of bolts when the joint is subjected to loads perpendicular to the bolt axis [5]. Bolts are built to resist a certain level of shear load, but if the load exceeds the shear strength of the bolt, it will result in the failure of the fastener in the direction of the load, as seen in Figure 20.

This failure mode is primarily seen in short joints subjected to heavy loads, such as those in the construction of bridges, crane gantries, and roof trusses.

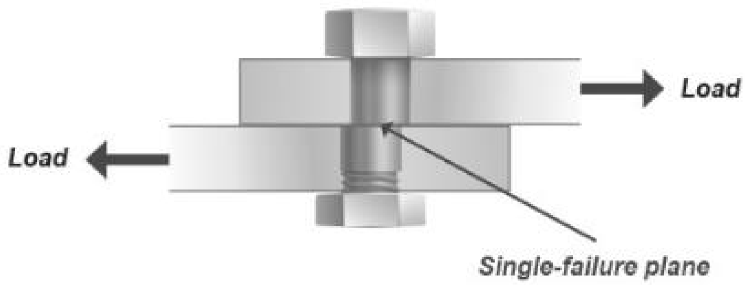

Figure 20 - Shear Failure of Fastener

3.2.1.2. Tension Failure of Fastener

This kind of failure occurs when the axial load of the bolt exceeds its yield strength (or ultimate strength), resulting in its fracture and disconnection of the clamped parts [5].

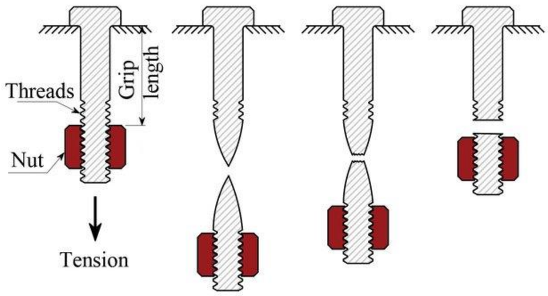

Figure 21 - Tension Failure of Fastener

3.2.1.3. Combined Failure of Fastener

This mode of failure presents itself when there is a combination of shear failure of the fastener, shown in section 3.2.1.1, and tension failure of the fastener, shown in section 3.2.1.2.

3.2.1.4. Net Tension Failure of Plate Section

This mode of failure occurs in two scenarios: when the width of the plate, excluding the diameter of the hole, is insufficient or when the tensile strength of the plate is less than the tensile load applied. If any of these scenarios is true, it will result in catastrophic failure of the joint through a fracture of the clamped part along the axis of the hole in the direction to the closest edge.

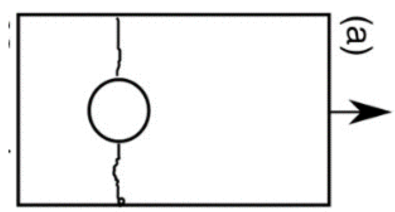

Figure 22 - Net Tension Failure of Plate

3.2.1.5. Bearing Failure of Plate

This kind of failure occurs when the applied load exceeds the capacity of the plate, resulting in plastic deformation of the bolt and plate interface around the bolt hole and creating a hole that is larger than the shank of the bolt [5]. This failure mode is like shear failure and cannot be distinguished just by visual inspection, so it is critical to understand the service and loading conditions that lead to it.

This failure mode is primarily seen in cases where the plate is too thin and improperly stiffened or bolted with oversize bolts.

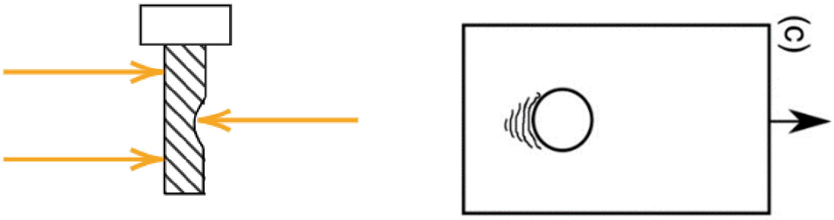

Figure 23 - Bearing Failure of Fastener and Plate

3.2.1.6. Shear-Out Failure of Plate

This kind of failure occurs when inadequately sized plates are used and are susceptible to stretching under heavy loads [5]. As a result of this overextension, the bolt holes become oversized and lead to substantial localized forces and eventually leading to failure. This failure can also occur if the bolt spacing is inadequate or if the structural components such as angles have lower thicknesses.

Shear-out failure occurs when the hole for the fastener is too close to the edge of the clamped part, or when the thickness of the material is insufficient to support the applied load, causing the clamped part to deform and create a vertical crack. It is most common in through-thickness tear-out, which tears along the thickness direction perpendicular to the load.

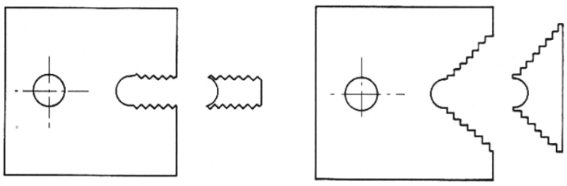

Figure 24 - Shear Failure of Plate

4. Procedure

4.1. ECSS

4.1.1. Overview

Table 3 reports the flowchart for the ECSS procedure, following subsections present the details for each step, regarding the procedure for each and the parameters required.

PROPERTIES OF THE PARTS (Bolt, washer, insert, nut, thread, plate)
↓
MATERIALS OF THE PARTS (Screw, washer, insert, nut, plate)
↓
FRICTION COEFFICIENTS (bolt head, bolt thread, plate)
↓
JOINT STIFFNESS PROPERTIES (bolt VS plate)
↓
ALLOWALBLE LOADS ON PARTS (bolt, plate)
↓
LOADS ON THE PARTS (gasket, bolt, plate, external load)
↓
STRESS OF THE PARTS (bolt, plate)
↓
SAFETY FACTORS
↓
CORRECTION FACTORS
↓
PRELOAD CALCULATION & TORQUE DISTRIBUTION
↓
MS CALCULATION

Table 3 – ECSS procedure

INPUT
CALCULATION
INPUT / CALCULATION

4.1.1.1. Properties of the Parts

The ECSS procedure requires the input of properties of the parts interacting in the joint, this includes the fastener, the plates, the locking device if any is present, and the washers if any are present. These properties are detailed in the following subsections.

4.1.1.1.1.Fastener Properties

The ECSS procedure considers several important fastener dimensions [9], which are shown and described in the following images and tables.

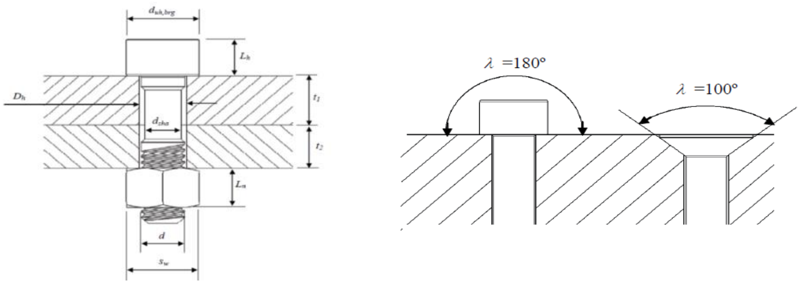

Figure 25 - Fastener Geometry and Underhead Bearing Angle

Variable	Definition	Notes
p	Pitch of thread	
h	Height of the basic thread	
d	Nominal fastener diameter	Outside diameter of thread
d_0	Diameter at smallest cross-section of fastener shank	
d_3	Minor diameter of thread	For the true thread, not the basic profile
d_{sha}	Shank diameter	For necked-down fasteners
l_{sha}	Shank length	For necked-down fasteners
$d_{uh,brg}$	Outer diameter of bearing area	Either under a head or under a nut
A_0	Smallest cross-section of fastener shank	
A_3	Cross-sectional area at minor diameter of fastener thread	
λ	Under-head bearing angle	Always 100° for countersunk aerospace standard fasteners
s_w	Size of wrench	Across flats dimension of either a head or a nut

Table 4 - Fastener Geometric Characteristics

It is also important to consider the geometry of the thread [9].

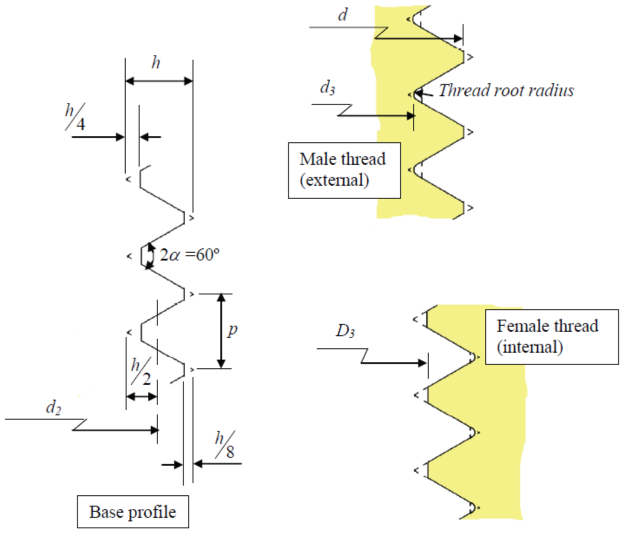

Figure 26 - Thread Geometry

Variable	Notation	Equation
Pitch diameter	d_2	$= d - 0.64952\ p$
Minor diameter	d_3	$= d - 1.22687\ p$
Diameter used for stress calculation	d_s	$= 0.5\ (d_2 + d_3)$ For M type thread
		= see stress area below For MJ type thread
Diameter used for stiffness calculation	d_{sm}	$= d_3$
Effective diameter of friction under head or nut	d_{uh}	$= 0.5\ (D_{head} + D_h)$
Stress area	A_s	$= 0.25\ \pi\ d_s^2$ For standard fasteners, M type thread $= 0.25\ \pi\ (d_3)^2\ [2-(d_3/d_2)^2]$ For standard fasteners, MJ type thread
		$= 0.25\ \pi\ d_0^2$ For reduced shank fasteners
Stiffness area	A_{sm}	$= 0.25\ \pi\ d_{sm}^2$
Smallest cross-section of fastener shank	A_0	$= 0.25\ \pi\ d_0^2$

Table 5 - Thread Geometric Characteristics

4.1.1.1.2. Plate Properties

The ECSS procedure requires geometric characteristics of the plates as inputs, considering two separate ones, that in contact with the fastener head and the second, also known as the clamped part, which may be in contact with the nut in case there is one present. The main characteristics of the plates are presented in Table 6.

Variable	Definition	Notes
D_h	Nominal diameter of hole in plate	
t	Plate thickness	This is to be specified for both plates
a	Minimum shear-out length	The distance from the center of the fastener to the edge of the plate
X	Number of shear force transmitting interfaces	
A_{net_plate}	Area of the plate	This area ignores the presence of holes and considers 2d as the edge distance
K_r	Maximum reduction factor of the ultimate strength of the material of the plate	

Table 6 - Plate Geometric Characteristics

4.1.1.1.3. Locking Device Properties

The locking devices considered by the ECSS procedure are the nuts and inserts, described in section 2.2.2, for which four different inputs are required, presented in Table 7.

Variable	Definition
D_{nut}	Outer diameter of the locking device
H	Height of the locking device
-	Minimum prevailing torque
-	Maximum prevailing torque

Table 7 - Locking Device Properties

4.1.1.1.4. Washer Properties

The inclusion of washers in the joint, as that of a locking device, is optional. In case that it is included, some information regarding its presence and geometry are to be specified, these are presented in Table 8.

Variable	Definition	Notes
-	Position of the washer	Indicate if the washer is located under the fastener head or under the nut. It is possible to have both.
$D_{wash,ext}$	External diameter of the washer	
$D_{wash,int}$	Internal diameter of the washer	
H_{wash}	Height of the washer	
-	Minimum prevailing torque	
.	Maximum prevailing torque	

Table 8 - Washer Properties

4.1.1.2. Material

The material of the parts and their properties is the second input requested for the analysis using the ECSS procedure. The most common materials and their properties are retrieved from [9] and [11], and those considered for the analysis, and for which parts, are reported in Table 9.

Variable	Unit	Property	Fastener	Locking Device	Plate
E	MPa	Young's Modulus	X	X	X
Fty	MPa	Yield Tensile Strength	X	X	X
Ftu	MPa	Ultimate Tensile Strength	X	X	X
Fsy	MPa	Yield Shear Strength	X	X	
Fsu	MPa	Ultimate Shear Strength	X	X	X
σbry	MPa	Yield Bearing Strength (e/d = 1.5)			X
σbru	MPa	Ultimate Bearing Strength (e/d = 1.5)			X
σbry	MPa	Yield Bearing Strength (e/d = 2.0)			X
σbru	MPa	Ultimate Bearing Strength (e/d = 2.0)			X
ν	-	Poisson ratio	X		X

Table 9 – Material Properties

4.1.1.3. Friction Coefficients

The friction coefficients to consider for the ECSS methodology are illustrated in Figure 27, which include the friction at the clamped interface, the friction under the bolt head, and the friction at the threaded interface. The values can be specified by the user or retrieved from the database according to the material and its coating of the contact surfaces.

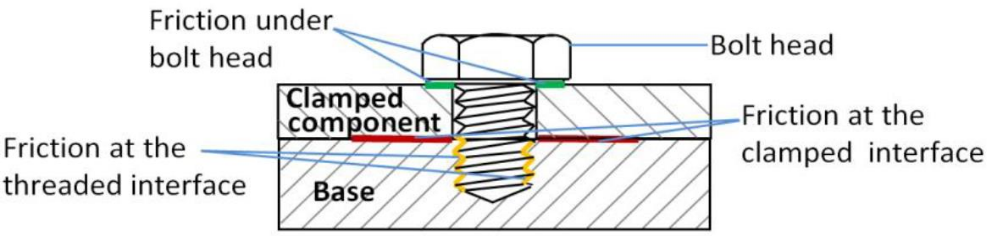

Figure 27 - Friction Coefficients Schematic

4.1.1.4. Joint Stiffness Properties

The joint stiffness refers to the total stiffness provided by all the parts of the joint, which include the fastener, the plates, and the locking device, if it is present. The calculation of these properties is important for fasteners that are subjected to axial load, and allows to define the behavior of each component, and of the joint, with respect to this load.

The stiffness of the joint is often represented as a set of resistances in series-parallel, as seen in Figure 28, the first set of resistances in series are those representing each of the plates, which are in parallel with a second set of resistance of series, which represent the fastener and the locking device.

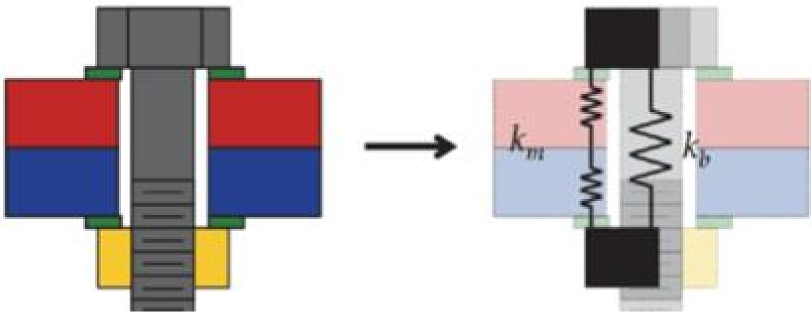

Figure 28 - Joint Stiffness Schematic

The process for the stiffness calculation for each of the parts is outlined in Table 10, which is detailed in the following subsections.

Fastener stiffness calculation	Parameters (bolt head, engaged thread, shank, engaged shank, locking device)
↓	
Plate stiffness calculation	Cone, Sleeve
↓	
RATIO CALCULATION	

Table 10 - Procedure for Joint Stiffness Calculation

4.1.1.4.1.Fastener Stiffness

The characteristics of the fastener stiffness are obtained from the different parts of the fastener that are in the same axis to it, including the locking device, and divided according to their behavior in the joint, as seen in Table 11.

Variable	Description	Unit	Formula
L_{h,sub_fact}	Factor regarding the length for deformations within the fastener head	-	$\begin{cases} 0.5, head = Hexagonal \\ 0.4 \end{cases}$
L_{eng,sub_fact}	Factor regarding the length for deformations within the engaged shank	-	$\begin{cases} 0.4, if\ nut \\ 0.33 \end{cases}$
L_{n,sub_fact}	Factor regarding the length for the deformations within the locking device	-	$\begin{cases} 0.4, if\ d_{nut} > 0 \\ "NA" \end{cases}$
R_{bolt}	Resilience of the bolt head	mm/N	$\frac{L_{h,sub\_fact} * d}{E_b * A_{nom}}$
$R_{eng,shank}$	Resilience of the bolt engaged shank area	mm/N	$\frac{L_{eng,sub\_fact} * d}{E_b * A_{sm}}$
R_{shank}	Resilience of the bolt shank area	mm/N	$\begin{cases} \frac{l_{shank}}{E_b * A_{shank}} \\ "NA", if\ A_{shank} = 0 \end{cases}$

$K_{locking\_device}$	Resilience of the locking device	mm/N	$\begin{cases} \frac{L_{n,sub\_fact} * d}{E_{nut} * A_{nom}}, if\ d_{nut} > 0 \\ "NA" \end{cases}$
R	Resilience of upper and lower engaged thread	mm/N	$\begin{cases} \frac{t}{A_{sm} * E_b}, if\ R_{shank} = "NA" \\ \frac{t - l_{shank}}{A_{sm} * E_b} \end{cases}$
K_b	Fastener stiffness	N/mm	$\frac{1}{R_{bolt} + R_{eng,shank} + R_{shank} + K_{locking\_device} + R}$

Table 11 - Fastener Stiffness Variables

Figure 29 shows the free body diagram for the interaction of the fastener stiffness with the components of the joint, including any locking device, if present, and the interaction at the contact surfaces.

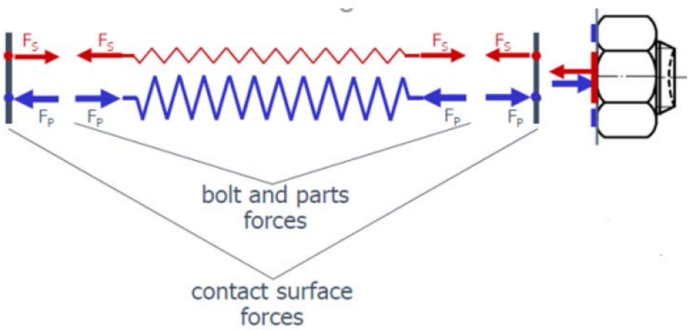

Figure 29 - Fastener Stiffness

4.1.1.4.2.Plate Stiffness

The plate stiffness properties are obtained through the effects of the compression zone generated by the loads created by the components in the joint. The compression zone obtains a distinct shape and characteristics depending on the characteristics of the interaction between the fastener, and its elements, with the plates. A more in-depth description of these cases is presented below.

The variables that describe the plate stiffness are presented in Table 12.

Variable	Description	Unit	Formula
L_c	Clamped part length	mm	$\begin{cases} t + H_{washer,head} + H_{washer,nut}, if\ NUT \\ t_{1,min} + H_{washer,nut} + (\min(d, t_{2,min}))/2 \end{cases}$
D_{avail}	Available diameter for the compression zone	mm	$2 * (2 * d)$
$D_{uh,brg}$	Underhead bearing diameter	mm	$\begin{cases} D_{wash,ext,head}, if\ D_{wash,ext,head} > 0 \\ D_{head} \end{cases}$
x	Non-dimensional compression cone parameters	-	$\frac{L_c}{D_{uh,brg}}$
y			$\frac{D_{avail}}{D_{uh,brg}}$
$\tan(\Phi)$	Tangent of compression cone half angle	-	$\begin{cases} 1.295 - 0.246 * \ln(x) + 0.94 * \ln(y), if\ INSERT \\ 0.362 + 0.032 * \ln\left(\frac{x}{2}\right) + 0.153 * \ln(y), if\ NUT \\ 0.348 + 0.013 * \ln(x) + 0.193 * \ln(y) \end{cases}$
w	Compression zone parameter	-	$\begin{cases} 1, if\ NUT \\ 2 \end{cases}$
D_{lim}	Limit diameter of the compression zone	mm	$D_{uh,brg} + (w * L_c * \tan(\Phi))$
$K_{Davail>Dlim}$	Fully developed compression zone stiffness	N/mm	$\frac{1}{\frac{2 * \ln\left[\frac{(D_{uh,brg} + d) * (D_{lim} - d)}{(D_{uhbrg} - d) * (D_{lim} + d}\right]}{w * E_{plate1} * \pi * d * \tan(\Phi)}}$
$K_{Dw<Davail<Dlim}$	Partially developed compression zone stiffness	N/mm	See equation 7.6.11 in [9]
L_{cone}	Length of cone	mm	$\frac{D_{avail} - D_{head}}{2 * \tan(\Phi)}$

L_{sleeve}	Length of sleeve	mm	$L_c - \left(\frac{2 * L_{cone}}{w}\right), if\ L_{sleeve} > 0$
R_{cone}	Resilience of cone	mm/N	$\frac{\ln\left[\frac{(D_{uh,brg} + d) * (D_{uh,brg} + 2 * L_{cone} * \tan(\Phi) - d)}{(D_{uh,brg} - d) * (D_{uh,brg} + 2 * L_{cone} * \tan(\Phi) + d)}\right]}{E_b * d * \pi * \tan(\Phi)}$
R_{sleeve}	Resilience of sleeve	mm/N	$\frac{4 * L_{sleeve}}{E_{plate2} * \pi * (D_{avail}^2 - d^2)}$
$K_{Dw<Davail<Dlim}$ (different material)	Partially developed compression zone stiffness (for plates with different materials)	N/mm	$\frac{1}{\frac{\ln\left[\frac{(D_{uh,brg} + d) * (D_{uh,brg} + 2 * L_{cone} * \tan(\Phi) - d)}{(D_{uh,brg} - d) * (D_{uh,brg} + 2 * L_{cone} * \tan(\Phi) + d)}\right]}{E_b * d * \pi * \tan(\Phi)}}$

Table 12 - Plate Stiffness Variables

Figure 30 shows the schematic of the parameters that define the plate stiffness, mentioned in Table 12. The schematic illustrates the behavior of the compression cone and when it takes the form of a cone or a sleeve.

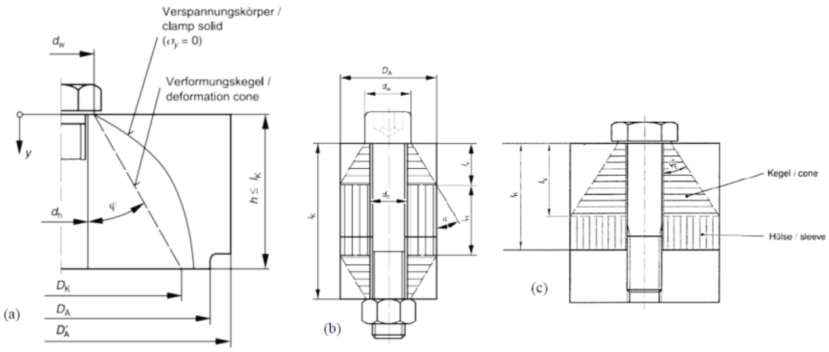

Figure 30 - Plate Stiffness

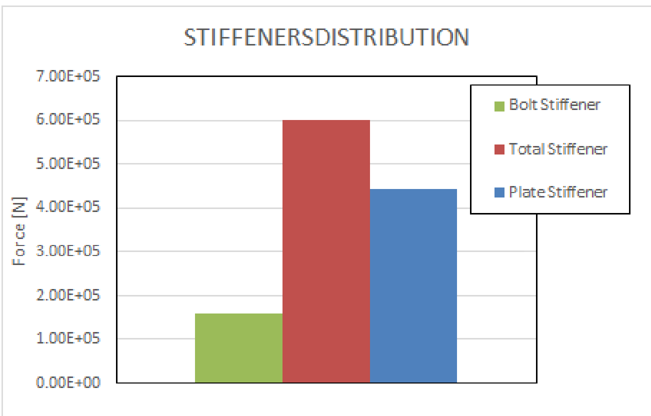

Figure 31 - Stiffness Distribution Plot

4.1.1.5. Allowable Loads on Parts

The calculation of the allowable loads on the parts of the joint permits the obtention of the theoretical limit of them according to the characteristics provided and results in the proper calculation of the margins of safety.

The methodology considers six allowable loads, presented in Table 13, that cover the calculation of the margins of safety for the kinds of failure indicated in Table 14.

Variable	Description	Unit	Formula
$F_{all,yield}$	Yield Allowable Tension Load on Screw	N	$A_s * F_{ty}$
$F_{all,ult}$	Ultimate Allowable Tension Load on Screw	N	$A_s * F_{tu}$
$F_{shear,ult}$	Ultimate Allowable Shear Load	N	$A_s * F_{su}$
T_{all}	Allowable Transverse Load	N	$\mu_c * F_{r,min}$
$F_{ult,th,n}$	Critical Load on Male Thread	N	$F_{su} * A_{th,n} * c_2 * c_1$
$F_{ult,th,n2}$	Critical Load on Female Thread	N	$\begin{cases} F_{su,nut} * A_{th,n} * c_2 * c_1, if\ NUT\ or\ INSERT \\ F_{su,plate2} * A_{th,n} * c_2 * c_1 \end{cases}$

Table 13 - Allowable Loads

Item	Kind of failure		
Bolt	Axial	Shear	Combined
Plate	Slippage	Gapping	

Table 14 – Kinds of Failure

4.1.1.6. Loads on Parts

The loads on the fastener and the plates are influenced by the load plane factor. This factor relates to the deformation of the joint caused by the preload [9] and it is used to define the plane where the loads are located, which is considered with a value of 0.5, as seen in Figure 32.

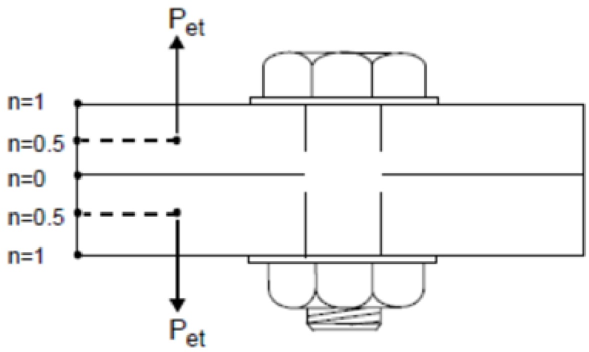

Figure 32 - Load Plane Factor

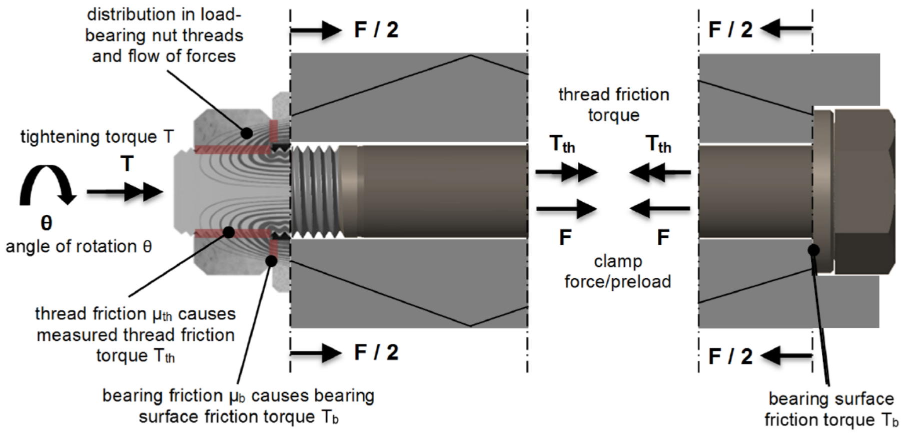

Figure 33 - Loads on Joint

In Figure 33, a schematic of the loading of the joint is shown, considering the different reaction loads to the preload and the external loads.

4.1.1.6.1.Loads on Fastener

The fastener is subjected to a tension load that considers two factors, the first being the deformation caused by the preload and the second being the additional deformation caused by the external load. The load is calculated at both the yield and ultimate strength levels for the fastener.

Yield: $\boldsymbol{F_{max,yield} = F_{v,max} + (n * f * sf_{yield} * T)}$

Ultimate: $\boldsymbol{F_{max,ult} = F_{v,max} + (n * f * sf_{ult} * T)}$

4.1.1.6.2. Loads on Plate

The plate is subjected to compression caused by the loads applied to the fastener. It is also subjected to a moment due to the friction under the fastener head.

The minimum residual compression allows to check for the presence of the slippage phenomenon and is given by:

$$\boldsymbol{F_{r,min} = F_{v,min} - (1 - f * n) * T}$$

The maximum residual compression allows to check for the crushing of plate and is given by:

$$\boldsymbol{F_{r,max} = \begin{cases} \dfrac{F_{v,max}}{2 - (1 - f * n)} * T \text{ } if \text{ } NUT \\ F_{v,max} - (1 - f * n) * T \end{cases}}$$

The minimum moment absorbed by the friction under the fastener head is given by:

$$\boldsymbol{M_{uh,min} = \frac{d_{uh}}{2} * \left(F_{v,max} - F_{t+}\right) * \mu_{uh,min}}$$

4.1.1.7. Stress of Parts

4.1.1.7.1. Fastener Stress

The ability of the fastener to sustain the tightening torque should be checked, this is done by calculating the stress by means of the Von Mises equation, which divides the torque into two components: a tension caused by the increasing preload and a shear stress caused by the application of the torque.

$$\sigma_{v.m.} = \sqrt{\sigma_{V,max}^2 + 3\tau_{max}^2}$$

where $\sigma_{V,max}$ is the maximum axial stress in the fastener due to preloading, and τ_{max} is the maximum shear stress due to remaining torsion in the fastener. These stresses are given by the following equations:

$$\sigma_{V,max} = \frac{F_{V,max} + F_{\Delta T^+}}{A_0}$$

$$\tau_{max} = \frac{M_{app,max} - M_{uh,min}}{P}$$

where the minimum under-head frictional torque is given by:

$$M_{uh,min} = \frac{d_{uh}}{2} \cdot \left(F_{V,max} - F_{\Delta T^+}\right) \cdot \mu_{uh,min}$$

and the polar section modulus is:

$$P = \frac{\pi \cdot d_0^3}{16}$$

After the shear stress in the fastener exceeds its yield point a state of total plastic deformation is reached, whereby the torsional stresses through the cross section are constant. To include this effect, the polar section modulus is corrected to:

$$\mathrm{P}' = \frac{\pi \cdot d_0^3}{12}$$

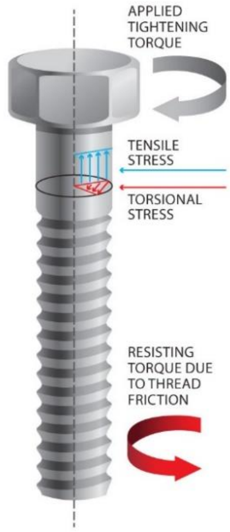

Figure 34 - Stresses on Fastener

4.1.1.7.2. Plate Stress

The loading on the plate causes an axial stress on the material. The methodology considers the stress on plate one as that caused by the load created by the fastener, and the stress on plate two as that caused by the load created by the nut.

Variable	Description	Unit	Formula
$\sigma_{v,plate1}$	Tensile stress on Plate 1, with or without washer	MPa	$\frac{F_{r,max}}{Area}$
$\sigma_{v,plate2}$	Tensile stress on Plate 2, with or without washer (only for nut presence)	MPa	$\begin{cases} \frac{F_{r,max}}{Area}, if\ NUT \\ NA \end{cases}$

Table 15 - Plate Stress Formulas

The calculation of the contact area depends on the presence or absence of a washer, either under the head for plate one or under the nut for plate two. If there is a washer present the contact area is that of the washer and if not, the contact area is that of the fastener head.

4.1.1.8. Safety Factors

Verification approach	Test type	Test factor	Safety factor for yield strength	Safety factor for ultimate strength	Safety factor for joint separation or gapping	
					Safety critical	Other
Analysis Only	NA	NA	1.25	2.0	1.4	1.2
Analysis and proto-type test	Qualification	1.4	1.0	1.4	1.4	1.2
Analysis and proto-flight test	Acceptance or proof	1.2	1.0	1.4	1.4	1.2

Table 16 - Safety Factors for ECSS[1]

To the safety factors specified above, a joint fitting factor with a value of 1.15 should be applied if a case where the joint load distribution is difficult to accurately predict.

To determine the safety factor to be used, according to Table 16, the following cases of joint separation should be taken into consideration:

- If the joint separation could lead to catastrophic failure: $\boldsymbol{sf} = \mathbf{max}(\mathbf{1.2};\ \boldsymbol{sf_y})$
- If the joint separation does not lead to a catastrophic failure: $\boldsymbol{sf} = \mathbf{max}(\mathbf{1.4};\ \boldsymbol{sf_{ult}})$

4.1.1.9. Correction Factors

The uncertainty factor ε is given according to the following table:

Method of preload application	Uncertainty factor (ε)
Torque measurement of unlubricated bolts	± 0.35
Torque measurement of lubricated bolts	± 0.25
Hydraulic fasteners	± 0.15
Preload indicating washers	± 0.10
Ultrasonic measurement devices	± 0.10
Bolt elongation measurement	± 0.05
Instrumented bolts	± 0.05

Table 17 - Uncertainty Factor

[1] Safety factors for yield strength and ultimate strength are to be taken as reference and a different one can be specified.

This is followed by the definition of the maximum and minimum bounds for the applied torque, which are given by the following:

$$\boldsymbol{M_{app,max} = (1 + \omega) M_{nom}}$$

$$\boldsymbol{M_{app,min} = (1 - \omega) M_{nom}}$$

where M_{nom} is the nominal applied torque and ω is the torque wrench accuracy, which is in the range ±5% to ±15%.

4.1.1.10. Preload Calculation & Torque distribution

For the ECSS procedure it is important to consider the applied torque to the fastener during installation, which is given by the following equation:

$$\boldsymbol{M_{app} = F_V \left[\frac{p}{\pi} + d_2 \frac{\mu_{th}}{\cos\frac{\theta}{2}} + \mu_{uh} \frac{d_{uh}}{2\sin(\lambda/2)} \right] + M_P}$$

The equation above can be separated into the following terms:

$\boldsymbol{F_V \left[\frac{p}{2\pi}\right]}$ representing the torque absorbed in stretching the fastener.

$\boldsymbol{F_V \left[\frac{d_2 \mu_{th}}{2\cos\frac{\theta}{2}}\right]}$ representing the torque due to friction in the threaded interface.

$\boldsymbol{F_V \left[\frac{d_{uh}\mu_{uh}}{2\sin(\lambda/2)}\right]}$ representing the torque due to friction under the nut or head.

$\boldsymbol{M_P}$ representing the prevailing torque due to the locking device.

A typical distribution of the torque between these components is:

- 10% Stretching of the bolt
- 30% Absorbed by friction in threads
- 50% Absorbed by friction under head of bolt or nut
- 10% Prevailing torque of a self-locking thread

Next, the maximum and minimum preloads are found using the following equations, which are derived from the previous equations and modified to include thermal and embedding effects:

$$F_{V,max} = \frac{(1+\varepsilon)\left(M_{app,max} - M_{P,min}\right)}{\frac{1}{2}d_2\left(\tan\phi + \frac{\mu_{th,min}}{\cos\frac{\theta}{2}}\right) + \frac{1}{2}d_{uh}\mu_{uh,min}} + F_{\Delta T^+}$$

$$F_{V,min} = \frac{(1-\varepsilon)\left(M_{app,min} - M_{P,max}\right)}{\frac{1}{2}d_2\left(\tan\phi + \frac{\mu_{th,max}}{\cos\frac{\theta}{2}}\right) + \frac{1}{2}d_{uh}\mu_{uh,max}} + F_{\Delta T^-} - F_Z$$

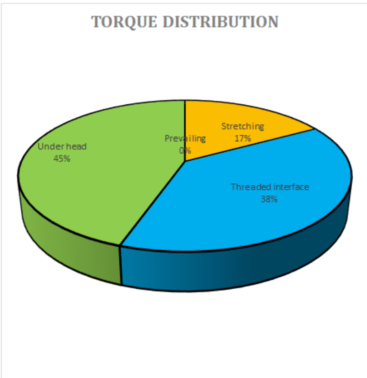

Figure 35 - Torque Distribution Plot

4.1.1.11. Margin of Safety Calculation

This section presents an overview of the methodology implemented by the ECSS standard for the design of joints, and the calculation of the margins of safety corresponding to each type of failure that might present on each type of joint.

Relevant failure mode description	**Relevant joint categories**	
	Friction grip	**Bearing**
Combined Stresses in fastener during tightening @ LL	YES	NO
Combined Stresses in fastener during tightening @ UL	YES	NO
Shear thread pull-out Male @ UL	YES	NO
Shear thread pull-out Female @ UL	YES	NO
Crushing of plate1 @ LL - (e/d = 2.0)	YES	YES
Crushing of plate1 @ UL - (e/d = 2.0)	YES	YES
Crushing of plate2 @ LL - (e/d = 2.0) (only for nut presence)	YES	YES
Crushing of plate2 @ UL - (e/d = 2.0) (only for nut presence)	YES	YES
Gapping (Min Preload)	YES	NO
Slippage (Min Preload)	YES	NO
Slippage Recovery (AVG Preload) **(*)**	YES	NO
Slippage (Under head)	YES	NO
Bending Bolt @ LL (due to load under head)	YES	YES
Bending Bolt @ UL (due to load under head)	YES	NO
Gapping (Min Preload + exceeded screw load)	YES	NO
Combined loads on fastener @ LL (exceeded screw load)	NO	YES
Combined loads on fastener @ UL (exceeded screw load)	NO	YES
Shear load in fastener @ LL	NO	YES

Shear load in fastener @ UL	NO	YES
Tension Load in Fastener @ LL (External Load Only)	NO	YES
Tension Load in Fastener @ UL (External Load Only)	NO	YES
Tension Load in Fastener @ LL (Preload + External Load)	NO	YES
Tension Load in Fastener @ UL (Preload + External Load)	NO	YES
Combined loads on fastener @ LL	NO	YES
Combined loads on fastener @ UL	NO	YES
Net Tension Section failure of Plate 1 @ UL	NO	YES
Net Tension Section failure of Plate 1 @ UL	NO	YES
Bearing Plate 1 strength @ LL - (e/d = 1.5)	NO	YES
Bearing Plate 1 strength @ UL - (e/d = 1.5)	NO	YES
Bearing Plate 2 strength @ LL - (e/d = 1.5)	NO	YES
Bearing Plate 2 strength @ UL - (e/d = 1.5)	NO	YES
Plate one shear-out @ UL	NO	YES
Plate two shear-out @ UL	NO	YES

Table 18 - Margins of Safety from ECSS Procedure[2]

[2] (*) for ECSS procedure if the plates are in slippage condition the AVG preload can be used to a recovery assumption. For MS calculation detail see sections 4.1.2, 4.1.3, and 4.1.4.

4.1.2. Friction Joints – Margins of Safety

4.1.2.1. Combined Stresses in Fastener during Tightening

The margins of safety on tightening are related to the stress on the fastener, specified in section 4.1.1.7, and are given by the following equations:

Yield: $\boldsymbol{MoS_{ti,y} = \frac{\sigma_y}{\sigma_{v.m.}} - 1}$

Ultimate: $\boldsymbol{MoS_{ti,ult} = \frac{\sigma_{ult}}{\sigma_{v.m.}} - 1}$

No factors of safety are used in these equations, since all uncertainty sources are already included in $F_{V,max}$. If the MoS for yield is positive, the MoS for the ultimate is always positive.

4.1.2.2. Shear thread Pull-Out

The margin of safety on shear pull-out of the thread under the external load is given by:

$$\boldsymbol{MoS_{th,A} = \frac{F_{th,crit}}{F_A \cdot sf_{ult}} - 1}$$

and for the overall load is given by:

$$\boldsymbol{MoS_{th,tot} = \frac{F_{th,crit}}{F_{V,max} + \Phi \cdot F_A \cdot sf_{ult}} - 1}$$

where $F_{th,crit}$ is the critical thread failure load calculated for male and female thread.

4.1.2.3. Crushing of Plate

The margins of safety on plate crushing are given by the following equations:

Yield: $\boldsymbol{MoS_{crush,y} = \frac{\sigma_{br,y}}{\sigma_{uh,max} \cdot sf_y} - 1}$

Ultimate: $\boldsymbol{MoS_{crush,ult} = \frac{\sigma_{br,ult}}{\sigma_{uh,max} \cdot sf_{ult}} - 1}$

where $\sigma_{uh,max}$ is the maximum compressive stress that occurs under the head or nut for either the overall load or the external load.

4.1.2.4. Gapping

The margin of safety for joint separation is given by:

$$\boldsymbol{MoS_{gap} = \frac{F_{V,min} - F_{K,req}}{(1 - \Phi) \cdot F_A \cdot sf_{sep}}}$$

where sf_{sep} is the safety factor defined for separation as seen in Table 16 and Φ is the force ratio defined by the following equation:

$$\boldsymbol{\Phi = \frac{\Delta F_{b,A}}{F_A}}$$

4.1.2.5. Slippage

The margin of safety for slipping is calculated considering the lowest possible preload $F_{V,min}$ and the safety factor for ultimate loads. It is given by:

$$\boldsymbol{MoS_{slip} = \frac{\left(F_{V,min} - \left(1 - \Phi_{e,n}\right) \cdot F_A\right) \cdot \mu_s \cdot x}{F_Q \cdot sf_{ult}}}$$

Where X is Number of shear force transmitting interfaces. If the MoS for slipping is positive, the margin of safety on separation defined in section 6.1.3.2 is also positive. The net tension section of each plate should also have sufficient strength to carry the shear load, for details on this, see section 6.1.4.8.

If the MoS for slipping is below zero, the joint should be reanalyzed as a bearing joint. Alternatively, if the joint has multiple fasteners, MoS_{slip} could be recalculated with the average preload.

4.1.3. Bearing Joints – Margins of Safety

4.1.3.1. Combined Stresses in Fastener during Tightening

Refer to section 4.1.2.1.

4.1.3.2. Crushing of Plate

Refer to section 4.1.2.3.

4.1.3.3. Shear Failure of Fastener

The margin of safety for shear failure of the fastener, for a pure shear load, is given by:

Yield: $\boldsymbol{MoS_{Q,y} = \frac{\tau_y \cdot A_s}{F_Q \cdot sf_y} - 1}$

Ultimate: $\boldsymbol{MoS_{Q,ult} = \frac{\tau_{ult} \cdot A_s}{F_Q \cdot sf_{ult}} - 1}$

4.1.3.4. Axial Stress in Fastener

The margin of safety due to the axial stress in the fastener is distinguished between two cases. The first case involves the axial stress caused by the effect of the external load, and it is given by:

Yield: $\boldsymbol{MoS_{ax,y} = \frac{F_{all,y}}{|T| * sf_y} - 1}$

Ultimate: $\boldsymbol{MoS_{ax,ult} = \frac{F_{all,ult}}{|T| * sf_{ult}} - 1}$

The second case involves the axial stress caused by the combined effect of the external load and the preload, and it is given by:

Yield: $\boldsymbol{MoS_{ax-pre,y} = \frac{F_{max,y}}{sf_y} - 1}$

Ultimate: $\boldsymbol{MoS_{ax-pre,ult} = \frac{F_{max,ult}}{sf_{ult}} - 1}$

4.1.3.5. Combined Fastener Failure

For the case of combined shear and axial loads, it is important to make a verification for the combined stresses. This verification can be done either for when the thread is in the shear plane, which is given by:

Yield: $\boldsymbol{R_{comb,y} = \sqrt{R_{A,y}^2 + R_{S,y}^2} \leq 1}$

Ultimate: $\boldsymbol{R_{comb,ult} = \sqrt{R_{A,ult}^2 + R_{S,ult}^2} \leq 1}$

Or for when the shaft is in the shear plane, which is given by:

Yield: $\boldsymbol{R_{comb,y} = R_{A,y}^2 + R_{S,y}^2 \leq 1}$

Ultimate: $\boldsymbol{R_{comb,ult} = R_{A,ult}^2 + R_{S,ult}^2 \leq 1}$

The utilization ratios for the strength of the fastener, used in the verifications above, are given by:

$$\boldsymbol{R_{A,y} = \frac{\left(F_{V,max} + \Phi_{e,n} \cdot F_A \cdot sf_y\right)}{\sigma_y \cdot A_s}}$$

$$\boldsymbol{R_{A,ult} = \frac{\left(F_{V,max} + \Phi_{e,n} \cdot F_A \cdot sf_{ult}\right)}{\sigma_y \cdot A_s}}$$

$$\boldsymbol{R_{S,y} = \frac{F_Q \cdot sf_y}{\tau_y \cdot A_s}}$$

$$\boldsymbol{R_{S,ult} = \frac{F_Q \cdot sf_{ult}}{\tau_{ult} \cdot A_s}}$$

The margins of safety for combined axial and shear forces acting on the fastener are given by:

Yield: $\boldsymbol{MoS_{comb,y} = \frac{1}{R_{comb,y}} - 1}$

Ultimate: $\boldsymbol{MoS_{comb,ult} = \frac{1}{R_{comb,ult}} - 1}$

4.1.3.6. Net Tension Section Failure of Plate

The margin of safety on net tension section failure is given by:

$$\boldsymbol{MoS_{Q,net} = \frac{K_R \cdot \sigma_{ult} \cdot A_{net,min}}{F_Q \cdot sf_{ult}} - 1}$$

4.1.3.7. Bearing Failure of Plate

The hole bearing phenomenon occurs after slipping of one or more of the fasteners. First, an elastic bearing stress is developed in the plate material adjacent to the hole, peaking at the line of contact. This is followed by a plastic elongation of the hole due to an increase in the shear force. For most cases, the stress distribution can be given by the following equation:

$$\boldsymbol{\sigma_{br}} = \frac{\boldsymbol{F_Q}}{\boldsymbol{d \cdot t}}$$

This value should be compared to the allowable values for yield and ultimate bearing strength of the material. The margins of safety on hole bearing strength are given by:

Yield: $\boldsymbol{MoS_{br,y}} = \frac{\sigma_{br,y} \cdot d \cdot t}{F_Q \cdot sf_y} - \mathbf{1}$

Ultimate: $\boldsymbol{MoS_{br,ult}} = \frac{\sigma_{br,ult} \cdot d \cdot t}{F_Q \cdot sf_{ult}} - \mathbf{1}$

4.1.3.8. Shear-Out Failure of Plate

Shear-out or tear-out failure can occur when the fasteners are close to the edge of the plate. These modes of failure do not need to be considered for fasteners that are more than *2d* from the plate edge.

To calculate the shear stress on the plate, the minimum shear-out length is used. Following the notation in Figure 36, this length is given by:

$$\boldsymbol{2a_l = |ef| + |gh|}$$

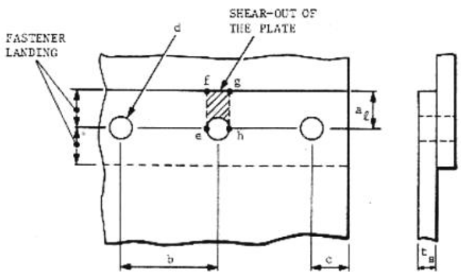

Figure 36 - Geometry for Shear-Out Calculation

For each fastener within *2d* from an edge, the following condition must be satisfied by the design:

$$\boldsymbol{\tau = \frac{F_Q}{2 \cdot a_l \cdot t} < \tau_{ult}}$$

This leads to the following margin of safety on shear-out:

$$\boldsymbol{MoS_{SO} = \frac{2 \cdot \tau_{ult} \cdot a_l \cdot t}{F_Q \cdot sf_{ult}} - 1}$$

4.1.4.Fastener Head Separation – Margin of Safety

The ECSS methodology presents mixed joints that have characteristics of both friction joints and bearing joints. The margins of safety for these are calculated separately, as they occur in the cases where the fastener head separates from the plate [25] due to either slippage under the bolt head, bending of the bolt, gapping, or a combination of loads where the load of the bolt is exceeded. The equations for the calculation of the margins of safety mentioned are presented in the following subsections.

4.1.4.1. Slippage under head

The margin of safety for the slippage under the bolt head is given by:

$$\boldsymbol{MoS_{slip,uh} = \frac{T_{all,uh}}{F_{under,bolt} * sf_{slippage}} - 1}$$

4.1.4.2. Bending of Bolt

The margin of safety for the bending of the bolt is considered for both at yield and ultimate limits. This phenomenon occurs due to the load under the fastener head.

Yield: $\boldsymbol{MoS_{bend,y} = \frac{F_{ty,b}}{\sigma * sf_y}}$

Ultimate: $\boldsymbol{MoS_{bend,ult} = \frac{F_{tu,b}}{\sigma * sf_{ult}}}$

4.1.4.3. Gapping

The phenomenon of gapping for the combined case is considered for the combination of the minimum preload and the exceeded load on the fastener. The margin of safety is given by:

$$\boldsymbol{MoS_{gap} = \frac{F_{v,min} - F_{k,req}}{\left((1 - f) * |T + F_b|\right) * sf_{gapping}} - 1}$$

4.1.4.4. Combined loads in bolt

The phenomenon of slippage due to the combined loads on the fastener appears upon exceeding its load. The margin of safety is calculated for both yield and ultimate strength.

Yield: $\boldsymbol{MoS_{comb,y}} = \frac{1}{\sqrt{(\frac{F_{max,yield}+F_b}{A_s*F_{ty}})^2+(\frac{F*sf_{yield}}{A_s*F_{sy}})^2}} - \mathbf{1}$

Ultimate: $\boldsymbol{MoS_{comb,ult}} = \frac{1}{\sqrt{(\frac{F_{max,ult}+F_b}{A_s*F_{tu}})^2+(\frac{F*sf_{ult}}{A_s*F_{sy}})^2}} - \mathbf{1}$

4.2. Eurocode

4.2.1. Overview

Table 19 reports the flowchart for the Eurocode procedure, following subsections present the details for each step, regarding the procedure for each and the parameters required.

PROPERTIES OF THE PARTS (bolt, thread, plate)
↓
MATERIALS OF THE PARTS (bolt and plate)
↓
FRICTION COEFFICIENTS (bolt head, bolt thread, plate)
↓
ALLOWALBLE LOADS ON PARTS (bolt, plate)
↓
LOADS ON PARTS (external load)
↓
SAFETY FACTORS
↓
CORRECTION FACTORS
↓
PRELOAD CALCULATION
↓
MS CALCULATION

Table 19 - Eurocode Procedure

INPUT
CALCULATION
INPUT / CALCULATION

4.2.1.1. Properties of the Parts

The parts considered by the Eurocode methodology are the fasteners and the plates, note that compared to the ECSS, this procedure does not consider any locking devices or washers in the joint.

The properties required are those specified in 4.1.1.1.1 for the fastener and 4.1.1.1.2 for the plate.

The Eurocode procedure not only considers bolts and screws as fasteners, but also pins, which have specific design parameters specified in subsection 4.2.1.1.1.

4.2.1.1.1. Design of Pins

The design requirements presented in the following sections are for solid circular pins.

In the case that the pin is meant to be replaceable, the contact bearing stress should satisfy the following requirement:

$$\sigma_{h,Ed} \leq f_{h,Rd}$$

where:

$$\sigma_{h,Ed} = 0.591 \cdot \sqrt{\frac{E \cdot F_{b,Ed,ser} \cdot (d_0 - d)}{d^2 \cdot t}}$$

$$f_{h,Rd} = \frac{2.5 \cdot f_y}{\gamma_{M6,ser}}$$

4.2.1.2. Materials

The Eurocode methodology considers several types of fasteners to define the permissible load according to the load case.

The procedure considers only fasteners made with the steel grades presented in Table 20 [10].

Bolt class	4.6	4.8	5.6	5.8	6.8	8.8	10.9
f_{yb} (N/mm^2)	240	320	300	400	480	640	900
f_{ub} (N/mm^2)	400	400	500	500	600	800	1000

Table 20 - Yield and Ultimate Strength by Bolt Class

4.2.1.3. Friction Coefficients

See section 4.1.1.3.

4.2.1.4. Allowable Loads on Parts

The allowable loads on parts considered by the Eurocode cover the different failure cases for the fasteners and plate. A detailed view on each of them and the formula for their calculation is presented in Table 21.

Variable	Description	Unit	Formula
$B_{p,Ed}$	Crushing allowable @ LL (head side)	N	$\frac{0.6 * \pi * D_{head} * t_{1,min} * F_{ty,1}}{\gamma_{M2}}$
$B_{p,Rd}$	Crushing allowable @ UL (head side	N	$\frac{0.6 * \pi * D_{head} * t_{1,min} * F_{tu,1}}{\gamma_{M2}}$
$B_{p,Ed,nut}$	Crushing allowable @ LL (nut side)	N	$\frac{0.6 * \pi * D_{nut} * t_{2,min} * F_{ty,2}}{\gamma_{M2}}$
$B_{p,Rd,nut}$	Crushing allowable @ UL (nut side)	N	$\frac{0.6 * \pi * D_{nut} * t_{2,min} * F_{tu,2}}{\gamma_{M2}}$
$F_{v,Ed}$	Allowable shear load on fastener @ LL	N	$\begin{cases} \frac{\alpha_v * F_{ty} * A_s}{\gamma_{M2}} * \beta_p \\ \frac{\alpha_v * F_{ty} * A_{nom}}{\gamma_{M2}} * \beta_p \end{cases}$

$F_{v,Rd}$	Allowable shear load on fastener @ UL	N	$\begin{cases} \frac{\alpha_v * F_{tu} * A_s}{\gamma_{M2}} * \beta_p \\ \frac{\alpha_v * F_{tu} * A_{nom}}{\gamma_{M2}} * \beta_p \end{cases}$
$F_{b,Ed}$	Allowable bearing load on plate @ LL	N	$\frac{k_l * \alpha_b * F_{ty} * d * t}{\gamma_{M2}} * BF$
$F_{b,Rd}$	Allowable bearing load on plate @ UL	N	$\frac{k_l * \alpha_b * F_{tu} * d * t}{\gamma_{M2}} * BF$
$F_{t,Ed,ser}$	Allowable tensile load on fastener @ LL	N	$\frac{k_2 * F_{ty} * A_s}{\gamma_{M3,ser}}$
$F_{t,Ed} = F_{t,Rd}$	Allowable tensile load on fastener @ UL	N	$\frac{k_2 * F_{tu} * A_s}{\gamma_{M2}}$
$F_{s,Rd}$	Allowable shear load in slippage connection @ LL	N	$\frac{k_s * X * \mu_c}{\gamma_{M3}} * F_{p,C}$
$F_{s,Rd,ser}$	Allowable shear load in slippage connection @ UL	N	$\frac{k_s * X * \mu_c}{\gamma_{M3,ser}} * F_{p,C}$
$F_{s,Rd,ser,comb}$	Combined shear & tension loads in slippage connection @ LL	N	$\frac{k_s * X * \mu_c * (F_{p,C} - 0.8 * F_{t,Ed,ser})}{\gamma_{M3,ser}}$
$F_{s,Rd,comb}$	Combined shear & tension loads in slippage connection @ UL	N	$\frac{k_s * X * \mu_c * (F_{p,C} - 0.8 * F_{t,Ed})}{\gamma_{M3}}$

Table 21 - Eurocode Allowable Loads

4.2.1.5. Loads on Parts

From section 4.1.1.6, the only loads considered for the Eurocode methodology are the external loads.

4.2.1.6. Safety Factors

Verification	Fastener	Safety Factor	Value

Preload	Bolts	γ_{M7}	1.1
Shear resistance per shear plane	Bolts	γ_{M2}	1.25
	Rivets		
Bearing resistance	Bolts	γ_{M2}	1.25
	Rivets		
Tension resistance	Bolts	γ_{M2}	1.25
	Rivets		
Punching shear resistance	Bolts	γ_{M2}	1.25
Combined tension and shear	Bolts	(*)	See 4.2.3.3.1
	Rivets		
Bearing resistance of the resin	Injection bolt	γ_{M4}	1.0
Preload of slip-resistant connections	Bolt at ultimate limit state	γ_{M3}	1.25
	Bolt at serviceability limit state	$\gamma_{M3,ser}$	1.1
Combined tension and shear of slip-resistant connections	Bolt at ultimate limit state	γ_{M3}	1.25
	Bolt at serviceability limit state	$\gamma_{M3,ser}$	1.1
Block tearing	Bolt group subject to concentric loading	γ_{M0}	1.0
		γ_{M2}	1.25
	Bolt group subject to eccentric loading	γ_{M0}	1.0
		γ_{M2}	1.25
Shear resistance	Pin	γ_{M2}	1.25
Bearing resistance	Plate	γ_{M0}	1.0
	Pin		
	Replaceable pin	$\gamma_{M6,ser}$	
Bending resistance	Pin	γ_{M0}	1.0
	Replaceable pin	$\gamma_{M6,ser}$	
Combined shear and bending resistance	Pin	(*)	See 4.2.3.3.2

Table 22 - Safety Factors for Eurocode[3]

4.2.1.7. Correction Factors

The Eurocode methodology provides several correction factors to be applied to different situations related to the geometry of the parts or the loading conditions of the joint.

[3] (*) There is no safety factor for the case of combined loads, an inequality is used to perform this verification, which is presented in the section specified.

The factors, there definition, and calculation are presented in sections 4.2.2 and 4.2.3 within the context of the calculation of their corresponding margin of safety.

4.2.1.8. Preload Calculation

The design preload and preload of bolts is calculated using the following equation:

Design preload: $\boldsymbol{F_{p,Cd}} = \frac{\mathbf{0.7} * \boldsymbol{f_{ub}} * \boldsymbol{A_s}}{\boldsymbol{\gamma_{M7}}}$

Preload: $\boldsymbol{F_{p,C}} = \mathbf{0.7} * \boldsymbol{f_{ub}} * \boldsymbol{A_s}$

The f_{ub} is the nominal value of the ultimate tensile strength of the bolt, which depends on the bolt class and is given in Table 20.

4.2.1.9. Margin of Safety Calculation

It is important to note that this methodology is used for the calculation of the allowable or permissible load, referred to as resistance. To obtain the margin of safety for each mode of failure, the generic equation of margin of safety is applied, which is given by:

$$\boldsymbol{MoS} = \frac{\boldsymbol{Allowable\ Load}}{\boldsymbol{Limit\ Load \cdot sf}} - \mathbf{1}$$

The allowable load is dependent on each mode of failure, while the limit load and safety factor are dependent on the material of the evaluated component and the mode of failure. The values chosen for each specific margin of safety correspond to the load case and the part entering failure. The margins of safety are calculated within the application and presented to the user as outputs upon request.

Relevant failure mode description	Relevant Joint Categories	
	Friction Grip	**Bearing**
Crushing Plate - Preload + Tension @ LL (Bolt Side)	YES	NO
Crushing Plate - Preload + Tension @ UL (Bolt Side)	YES	NO
Crushing Plate - Preload + Tension @ LL (Nut Side)	YES	NO
Crushing Plate - Preload + Tension @ UL (Nut side)	YES	NO
Slippage Shear @ LL	YES	NO
Slippage Shear @ UL	YES	NO
Slippage Shear + Tension @ LL	YES	NO
Slippage Shear + Tension @ UL	YES	NO
Shear load in fastener @ LL	NO	YES
Shear load in fastener @ UL	NO	YES
Tension load in fastener @ LL	NO	YES
Tension load in fastener @ UL	NO	YES
Combined Loads on Fastener @ LL	NO	YES
Combined Loads on Fastener @ UL	NO	YES
Bearing Load @ LL	NO	YES
Bearing Load @ UL	NO	YES

Table 23 - Margins of Safety from Eurocode Procedure

4.2.2. Friction Joints – Margins of Safety

4.2.2.1. Plate Preload and Tension

The allowable punching for a bolt or nut is calculated using the following equation:

$$\boldsymbol{B_{p,Rd}} = \frac{\boldsymbol{0.6 \cdot \pi \cdot d_m \cdot t_p \cdot f_u}}{\boldsymbol{\gamma_{M2}}}$$

The margin of safety at yield is given by:

$$\boldsymbol{MoS_{ti,y}} = \frac{\boldsymbol{B_{p,Ed}}}{(|\boldsymbol{T}| + \boldsymbol{F_{p,C}}) * \boldsymbol{sf_y}} - \boldsymbol{1}$$

The margin of safety at ultimate strength is given by:

$$\boldsymbol{MoS_{ti,ult}} = \frac{\boldsymbol{B_{p,Rd}}}{(|\boldsymbol{T}| + \boldsymbol{F_{p,C}}) * \boldsymbol{sf_{ult}}} - \boldsymbol{1}$$

4.2.2.2. Slippage Shear

The design slip resistance of preloaded bolts of classes 8.8 and 10.9 is calculated using the following equations:

- For ultimate limit state: $\boldsymbol{F_{s,Rd}} = \frac{\boldsymbol{k_s \cdot X \cdot \mu}}{\boldsymbol{\gamma_{M3}}} \cdot \boldsymbol{F_{p,C}}$
- For serviceability limit state: $\boldsymbol{F_{s,Rd,ser}} = \frac{\boldsymbol{k_s \cdot X \cdot \mu}}{\boldsymbol{\gamma_{M3,ser}}} \cdot \boldsymbol{F_{p,C}}$

Where the value of X is the number of friction planes, k_s is given in Table 24, and μ is the slip factor obtained either by specific tests for the friction surface or by the values given in Table 25.

Description	k_s
Bolts in normal holes	1.0
Bolts in either oversized holes or short slotted holes with the axis of the slot perpendicular to the direction of the load transfer	0.85
Bolts in long slotted holes with the axis of the slot perpendicular to the direction of the load transfer	0.7
Bolts in short-slotted holes with the axis of the slot parallel to the direction of the load transfer	0.76
Bolts in long slotted holes with the axis of the slot parallel to the direction of the load transfer	0.63

Table 24 - k_s Factor

Class of friction surfaces	Slip factor (μ)
A	0.5
B	0.4
C	0.3
D	0.2

Table 25 - Slip Factor[4]

The margin of safety at yield is given by:

$$MoS_{slip,y} = \frac{F_{s,Rd} * X}{F * sf_{slip}} - 1$$

The margin of safety at ultimate strength is given by:

$$MoS_{slip,ult} = \frac{F_{s,Rd,ser} * X}{F * sf_{slip}} - 1$$

4.2.2.3. Slippage Shear and Tension

If a slip-resistant connection is subjected to an applied tensile force additionally to the shear force tending to produce slip, the design slip resistance is calculated using one of the following equations:

- For a category B connection: $F_{s,Rd,ser,comb} = \frac{k_s \cdot X \cdot \mu \cdot (F_{p,C} - 0.8 \cdot F_{t,Ed,ser})}{\gamma_{M3,ser}}$
- For a category C connection: $F_{s,Rd,comb} = \frac{k_s \cdot X \cdot \mu \cdot (F_{p,C} - 0.8 \cdot F_{t,Ed})}{\gamma_{M3}}$

If, in a moment connection, a contact force on the compression side counterbalances the applied tensile force, no reduction in slip resistance is required.

The margin of safety for category B connections is given by:

$$MoS_{slip,comb,B} = \frac{F_{s,Rd,ser,comb}}{(F + T) * sf_{slip}} - 1$$

The margin of safety for category C connection is given by:

$$MoS_{slip,comb,C} = \frac{F_{s,Rd,comb}}{(F + T) * sf_{slip}} - 1$$

[4] For a detailed description of the classes of friction surfaces, refer to Table 17 of EN-1090-2:2018.

4.2.3. Bearing Joints – Margins of Safety

4.2.3.1. Shear Failure of Fastener

4.2.3.1.1.Bolts

The allowable shear load per shear plane for a bolt is calculated using the following equation:

$$F_{v,Rd} = \frac{\alpha_v \cdot f_{ub} \cdot A}{\gamma_{M2}}$$

The equation above distinguishes two distinct cases for values to use for the area and the constant coefficient α_v, which are described in Table 26.

<table>
<tr><th>Case</th><th>Bolt Classes</th><th>Area (A)</th><th>α_v</th></tr>
<tr><td rowspan="2">Shear plane passes through the threaded portion of the bolt</td><td>4.6
5.6
8.8</td><td rowspan="2">Tensile stress area (A_s)</td><td>0.6</td></tr>
<tr><td>4.8
5.8
6.8
10.9</td><td>0.5</td></tr>
<tr><td>Shear plane passes through the unthreaded portion of the bolt</td><td>-</td><td>Area of the gross cross-section of the bolt</td><td>0.6</td></tr>
</table>

Table 26 - α_v Factor

The procedure provides specification for exceptions when calculating the shear resistance of bolts [10], these are:

- For bolts with cut threads, for which the threads do not comply with EN-1090, the values to be used according to Table 26 should be multiplied by a factor of 0.85.
- The design shear resistance formula presented should be used only for bolts used in holes with nominal clearance.
- Where bolts pass through packing of total thickness t_p greater than eight times the nominal diameter d, the calculated shear resistance should be multiplied by a reduction factor β_p given by:

$$\beta_p = \frac{9d}{8d + 3t_p} \; but \; \beta_p \leq 1$$

- For double shear connections with packing on both sides of the splice, t_p should be taken as thickness of the thicker packing.

The margin of safety at yield is given by:

$$MoS_{Q,y} = \frac{1}{\sqrt{(\frac{sf_y * F}{F_{V,Ed}})^2}} - 1$$

The margin of safety at ultimate strength is given by:

$$MoS_{Q,ult} = \frac{1}{\sqrt{(\frac{sf_{ult} * F}{F_{V,Rd}})^2}} - 1$$

4.2.3.1.2.Rivets

The allowable shear load per shear plane for a rivet is calculated using the following equation:

$$F_{v,Rd} = \frac{0.6 \cdot f_{ur} \cdot A_0}{\gamma_{M2}}$$

4.2.3.1.3.Pins

The allowable shear load for a pin should satisfy the following requirement:

$$F_{v,Rd} = \frac{0.6 \cdot A \cdot f_{up}}{\gamma_{M2}} \geq F_{v,Ed}$$

4.2.3.2. Tensile Failure of Fastener

4.2.3.2.1.Bolts

The allowable tensile load for a bolt is calculated using the following equation:

$$F_{t,Rd} = \frac{k_2 \cdot f_{ub} \cdot A_s}{\gamma_{M2}}$$

Criteria	Value
Countersunk Bolt	0.63
Other	0.90

Table 27 - k_2 Factor

The margin of safety at yield is given by:

$$\boldsymbol{MoS_{T,y}} = \frac{\boldsymbol{F_{t,Ed}}}{(|\boldsymbol{T}| + \boldsymbol{F_{p,C}}) * \boldsymbol{sf_y}} - \mathbf{1}$$

The margin of safety at ultimate strength is given by:

$$\boldsymbol{MoS_{T,ult}} = \frac{\boldsymbol{F_{t,Rd}}}{(|\boldsymbol{T}| + \boldsymbol{F_{p,C}}) * \boldsymbol{sf_{ult}}} - \mathbf{1}$$

4.2.3.2.2.Rivets

The allowable tensile load for a rivet is calculated using the following equation:

$$\boldsymbol{F_{t,Rd}} = \frac{\mathbf{0.6} \cdot \boldsymbol{f_{ur}} \cdot \boldsymbol{A_0}}{\boldsymbol{\gamma_{M2}}}$$

4.2.3.3. Combined Fastener Failure

4.2.3.3.1.Bolts and Rivets

The yield margin of safety for the combined shear and tension loads case on a fastener is given by the following equation:

$$\boldsymbol{MoS_{comb,y}} = \frac{\mathbf{1}}{\sqrt{\left(\frac{\boldsymbol{F_{t,Ed}}}{\boldsymbol{A_s} * \boldsymbol{F_{ty}}}\right)^2 + \left(\frac{\boldsymbol{F} * \boldsymbol{sf_y}}{\boldsymbol{F_{v,Ed}}}\right)^2}} - \mathbf{1}$$

The ultimate margin of safety for the combined shear and tension loads case on a fastener is given by the following equation:

$$\boldsymbol{MoS_{comb,ult}} = \frac{\mathbf{1}}{\sqrt{\left(\frac{\boldsymbol{F_{t,Rd}}}{\boldsymbol{A_s} * \boldsymbol{F_{tu}}}\right)^2 + \left(\frac{\boldsymbol{F} * \boldsymbol{sf_{ult}}}{\boldsymbol{F_{v,Rd}}}\right)^2}} - \mathbf{1}$$

4.2.3.3.2.Pins

For the combined bending and shear loads for a pin, it should satisfy the following requirement:

$$\left[\frac{\boldsymbol{M_{Ed}}}{\boldsymbol{M_{Rd}}}\right]^2 + \left[\frac{\boldsymbol{F_{v,Ed}}}{\boldsymbol{F_{v,Rd}}}\right]^2 \leq \mathbf{1}$$

4.2.3.4. Bearing Failure of Fastener

4.2.3.4.1.Bolts and Rivets

The allowable bearing load for a bolt or rivet is calculated using the following equation:

$$F_{b,Rd} = \frac{k_l \cdot \alpha_b \cdot f_u \cdot d \cdot t}{\gamma_{M2}}$$

The coefficient α_b is given in the direction of the load transfer and by the smallest of either α_d ; $\frac{f_{ub}}{f_u}$; 1.0, where α_d is given by one of the following two cases:

For end bolts	$\alpha_d = \frac{e_1}{3 \cdot d_0}$
For inner bolts	$\alpha_d = \frac{p_1}{3 \cdot d_0} - \frac{1}{4}$

Table 28 - α_d Factor

The coefficient k_l is given in the direction perpendicular to the load transfer by the smallest value corresponding to one of the following two cases:

For end bolts	$k_l = 2.8\frac{e_2}{d_0} - 1.7$
	$k_l = 1.4\frac{p_2}{d_0}$
	$k_l = 2.5$
For inner bolts	$k_l = 1.4\frac{p_2}{d_0} - 1.7$
	$k_l = 2.5$

Table 29 - k_l Factor

The standard specifies some cases where the allowable bearing load is to be multiplied by an adjustment factor. These are presented in Table 30.

Case	**Adjustment Factor**
Oversized holes	0.8
Slotted holes, where the longitudinal axis of the holes is perpendicular to the direction of the force transfer	0.6
Countersunk bolt	Plate thickness *t* is equal to the thickness of connected plate minus half the depth of the countersinking

Table 30 - Adjustment Factor

The margin of safety at yield is given by:

$$MoS_{br,y} = \frac{F_{b,Ed}}{F * sf_y} - 1$$

The margin of safety at ultimate strength is given by:

$$MoS_{br,ult} = \frac{F_{b,Rd}}{F * sf_{ult}} - 1$$

4.2.3.4.2.Injection Bolts

An injection bolt can be used as an alternative to ordinary bolts and rivets for category A, B, and C connections specified in section 1.2.2.

The allowable bearing load for the resin of an injection bolt is calculated using the following equation:

$$F_{b,Rd,res} = \frac{k_t \cdot k_{s,res} \cdot d \cdot t_{b,res} \cdot \beta \cdot f_{b,res}}{\gamma_{M4}}$$

Where the values for the coefficients are defined in Table 31, Table 32, and Table 33.

Case	k_t
Serviceability limit state (long duration)	1.0
Ultimate limit state	1.2

Table 31 - k_t Factor

Case	$k_{s,res}$
Holes with normal clearance	1.0
Oversized holes	$1.0 - 0.1 \cdot m$
m is the difference (in mm) between the normal and oversized hole dimensions. For short-slotted holes:	

Table 32 - $k_{s,res}$ Factor

t_1/t_2	β	$t_{b,res}$
≥ 2.0	1.0	$2t_2 \leq 1.5d$
$1.0 < t_1/t_2 < 2.0$	$1.66 - 0.33 \cdot (t_1/t_2)$	$t_1 \leq 1.5d$
≤ 1.0	1.33	$t_1 \leq 1.5d$

Table 33 - β and $t_{b,res}$ Factors[5][6]

[5] β is the coefficient depending on the thickness ratio of the connected plates.
[6] $t_{b,res}$ is the effective bearing thickness of the resin.

The determination of the bearing strength of the resin ($f_{b,res}$) is determined according to the procedure specified in Annex K.6 specified in the specification EN-1090-2:2008 [4].

4.2.3.4.3.Pins

The allowable bearing load for a plate and pin connection should satisfy the following requirement:

$$\boldsymbol{F_{b,Rd} = \frac{1.5 \cdot t \cdot d \cdot f_y}{\gamma_{M0}} \geq F_{b,Ed}}$$

If the pin is intended to be replaceable, it should also satisfy the following requirement:

$$\boldsymbol{F_{b,Rd,ser} = \frac{0.6 \cdot t \cdot d \cdot f_y}{\gamma_{M6,ser}} \geq F_{b,Ed,ser}}$$

4.2.3.5. Bending Failure of Fastener

4.2.3.5.1.Pins

The allowable bending load for a pin should satisfy the following requirement:

$$\boldsymbol{M_{Rd} = \frac{1.5 \cdot W_{cf} \cdot f_{yp}}{\gamma_{M0}} \geq M_{Ed}}$$

If the pin is intended to be replaceable, it should also satisfy the following requirement:

$$\boldsymbol{M_{Rd,ser} = \frac{0.8 \cdot W_{cf} \cdot f_{yp}}{\gamma_{M6,ser}} \geq M_{Ed,ser}}$$

5. Software Development

The development of the software is divided into three large scale phases: the software preparation, the development of the algorithm, and the development of the graphic user interface. The steps taken on each of these phases are detailed in the following sections.

The algorithm development and GUI development stages, presented in sections 5.2 and 5.3 is divided into two phases, the initial development phase, and the feedback development phase. The initial development consisted in replicating the behavior of the schematization, see section 5.1.1 and an initial proposal for the GUI, making it functional with the minimum of necessary functions for an improved user experience.

The functional prototype and a user guide were then submitted for a user test phase, where two people that will interact with the tool during production used it during a period of four weeks, where they tried several use cases and noted any bugs or errors encountered, as well as any suggestions for improvements, changes, and further developments, for both the algorithm and the user interface. Some of the feedback received was implemented during the feedback development phase, with a focus on those changes that were assigned as critical for production, while the rest will be developed in the future, as mentioned in section 6.2.

5.1. Software Preparation

The software preparation phase consists of three stages that involve the schematization of the methodologies, ECSS and Eurocode, the definition of the database, and the definition of the software architecture.

5.1.1. Methodology Schematization

The methodology schematization stage is the first in the preparation of the software phase, in this phase the theory behind the ECSS and Eurocode methodologies, explained in sections 4.1 and 4.2 is developed in an Excel with macros. This is done to fully understand the use of them and being able to interact with working algorithms prior to coding.

5.1.2. Database Definition

The database plays an important role in the operation of the software, as it requires to access data related to components and material properties. The requirements for the database contents are outlined by the methodology schematization presented in section 5.1.1.
The decision is to create a single internal SQL database using the SQLite3 module, as outlined in section 5.2.1, with the structured presented in the diagram in Figure 37.

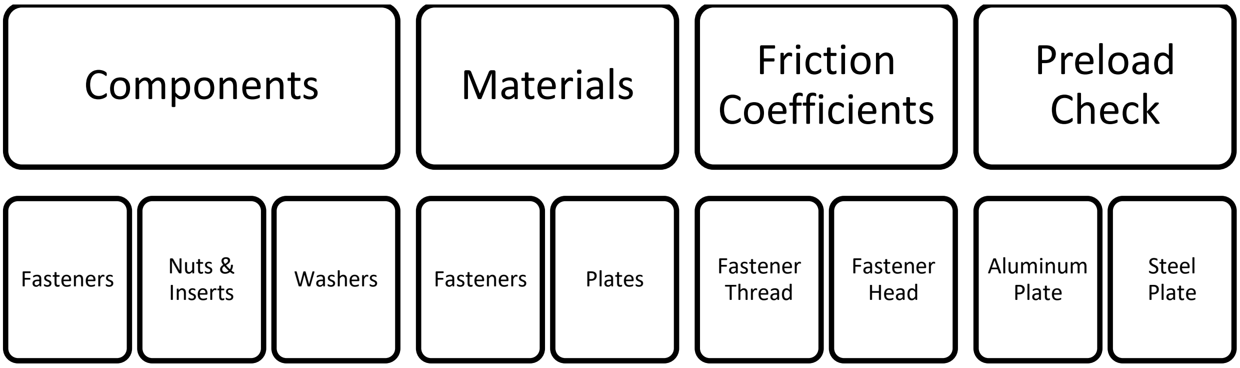

Figure 37 - Database Structure

There are four categories of table structures that are divided into components, materials, friction coefficients, and preload check. This structure allows us to define the required properties for each dataset in the database. Within these categories there is a further division onto the different tables contained within each one of them, with small variations between one table from the other depending on requirements.

5.1.3. Software Architecture

The software architecture is defined using the schematization done in Excel and the database structure defined. This is done to properly understand how the software should connect between its different functions to perform the required calculations and access the database as needed. This process also allows us to understand the interaction between the algorithm and the graphic user interface.

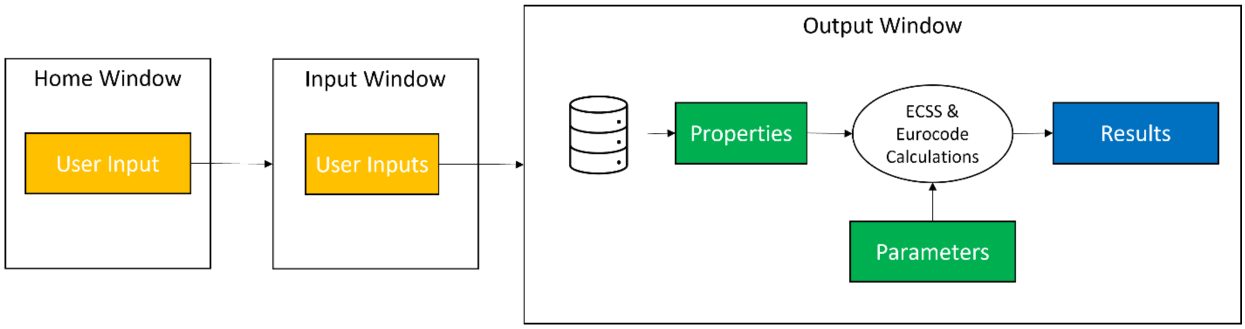

Figure 38 - Software Architecture

Figure 38 shows a schematic of the architecture of the software. Three main blocks encapsulate the graphic, each of them representing one of the windows of the software, which are detailed in sections 5.2 and 5.3. Each block contains the general logic of each class, interactions within the algorithm and the user interface. The yellow rectangles with the text "User Input" represent inputs that come directly from the user, which are typed or selected in the GUI. The green rectangles represent values obtained directly by the algorithm and that are set based on specific conditions according to the input or the methodologies. The blue rectangle with the text "Results" represents the output presented to the user via the GUI.

As seen in the diagram, each window interacts with those consecutive to it and some inputs interact directly with the algorithm or the database to obtain parameters required for the calculations. More detailed diagrams for each class are presented in subsections 5.2.2.1, 5.2.2.2, and 5.2.2.3

5.2. Algorithm Development

Figure 39 - Python Logo

The development of the software is done using the Python programming language. The selection of this language is due to its combination of simplicity, flexibility, and performance. It allows to develop the logic and GUI interface in a single platform, which can easily work in any OS as required, this is due to the vast array of libraries that work with the language to perform and integrate different types of operations, as shown in section 5.2.1.

5.2.1. Python Libraries and Modules

For the development of the software several python libraries and modules were used, these are specified in Table 34.

Library	Description
Math	A built-in module in the Python 3 standard library that provides standard mathematical constants and functions.
Openpyxl	A Python library to read/write Excel 2010 xlsx/xlsm/xltx/xltm files.
OS	A built-in module in the Python 3 standard library that provides functions for interacting with the operating system.
Pandas	A Python library used for working with data sets. It has functions for analyzing, cleaning, exploring, and manipulating data.
PyQt6	A comprehensive set of Python bindings for Qt v6. Qt is a set of cross-platform C++ libraries that implement high-level APIs for accessing many aspects of modern desktop and mobile systems.
SQLite3	A built-in module in the Python 3 standard library based in C-language implements a small, fast, self-contained, high-reliability, full-featured, SQL database engine.
Sys	A built-in module in the Python 3 standard library that provides various functions and variables that are used to manipulate different parts of the Python runtime environment.

Table 34 - Python Libraries

5.2.2. Software Structure

The software is divided onto three classes, related to the three graphic user interface windows displayed, which are described in section 5.3. In Table 35 the functions used in each class are shown, each of them is described in detail in the following subsections. All classes include an "__init__" method that serves to initialize the GUI and the variables required throughout the entire class.

Class	Functions
HomeWindow	open_input_window
InputWindow	set_tab_info create_tabs get_combo_box_items add_row delete_row copy_row help_message open_file check_rows next_button_clicked back_button_clicked open_output_window restart_app update_tab_text_format
OutputWindow	set_tab_info create_tabs restart_app export_data calculate

Table 35 - Software Structure

5.2.2.1. HomeWindow Class

The HomeWindow class provides the logic for displaying the initial screen of the software, and the logic for the setup of the InputWindow interface and for opening the interface for it. The diagram in Figure 40 shows the interactions within the class and the call to the InputWindow class.

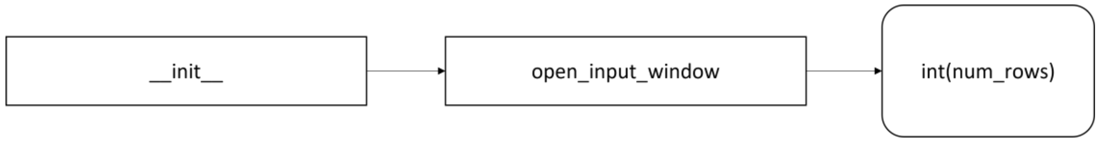

Figure 40 - HomeWindow Class Diagram

5.2.2.2. InputWindow Class

The InputWindow class provides the logic to allow the user to input the information regarding the joint, and for the creation of the GUI and its modification. The functions in the class can be divided into two categories: GUI creation and interaction with the GUI.

GUI creation functions include:

- set_tab_info: This method creates a list of dictionaries that contains the relevant information for the GUI, this information is the name of the tab, the number of columns for the table in the tab, the headers for such columns, the editor type for each column, and the image present in the tab.
- create_tabs: This method receives the list of dictionaries created in set_tab_info and uses it to create the tab interface according to the information contained in it. It also creates the button for the user interaction with the GUI and calls the get_combo_box_items function for the columns with this editor type.
- get_combo_box_items: This method is called by create_tabs when a column has the editor type combo box, and it receives the column header label to identify which options are to be set for the combo box.

Interaction with the GUI functions include:

- add_row: This method allows the user to add an empty row to the table in the current tab below the selected row. It is called upon clicking the “Add Row” button on the GUI.

- delete_row: This method allows the user to delete the selected row of the table in the current tab. It is called upon clicking the "Delete Row" button on the GUI.
- copy_row: This method allows the user to create a copy of the selected row, the copy is placed below the selected row. It is called upon clicking the "Copy Row" button on the GUI.
- help_message: This method allows the user to visualize help messages for the corresponding tab, these messages consist of explanations of the columns and expected ranges of values for each, as well as an indication of any default values set if the cell is left blank. The messages appear on a pop-up message box and the method is called upon clicking the "Help" button on the GUI.
- open_file: This method allows the user to open an auxiliary file with the properties of components, which include fasteners, nuts, inserts, and washers, according to standards. It is called upon clicking the "Standards" button on the GUI.
- check_rows: This method performs a verification to ensure that all the tabs have the same number of rows, as they must coincide to do the calculations properly, as each row correspond to a fastener and the information from all tabs is required for all fasteners. It is called from the next_button_clicked method and it returns either True or False depending on if the number of rows in all tabs matches or not.
- next_button_clicked: This method allows the user to navigate from one tab onto the next one, as it does this navigation in the GUI, it also consolidates the input values of the tab from which the method is called. It is called upon clicking the "Next" button on the GUI, when clicked from the last tab, it calls the check_rows method and depending on the result from it, it either triggers a warning or calls the open_output_window method.
- back_button_clicked: This method allows the user to navigate from one tab to the previous one, as it does this navigation in the GUI, it also clears the consolidated values from the target tab, allowing the user to edit them if needed. It is called upon clicking the "Back" button on the GUI.

- restart_app: This method allows the user to restart the software. It is called upon clicking the "Restart" button on the GUI.
- update_tab_text_format: This method changes the format of the tab text by making bold the text of the active. It is called upon changing tabs, either via the "Next" button or by clicking directly on one.

The diagram in Figure 41 shows the relationship and interactions within the methods in the class. Orange arrows indicate the methods called upon the user clicking a button on the GUI.

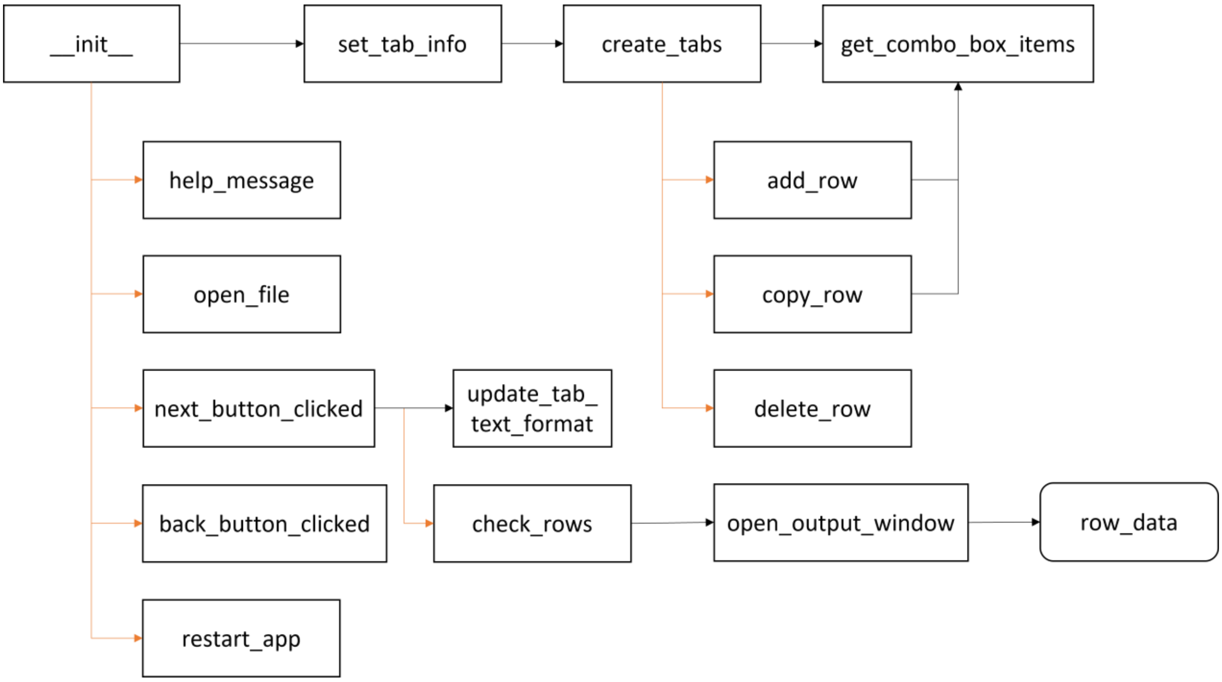

Figure 41 - InputWindow Class Diagram

5.2.2.3. OutputWindow Class

The OutputWindow class provides the logic to perform the calculations and displaying the results on the GUI. The functions in the class can be divided into three categories: GUI creation, data manipulation, and interaction with the GUI.

GUI creation functions include:

- set_tab_info: This method creates a list of dictionaries that contains the relevant information for the GUI, this information is the name of the tab, the number of columns for the table in the tab, the headers for such columns, and the image present in the tab.

- create_tabs: This method receives the list of dictionaries created in set_tab_info and uses it to create the tab interface according to the information contained in it.

Data manipulation functions include:

- calculate: This method receives the values from the InputWindow to process them to perform the calculations according to the ECSS and Eurocode methodologies. The method first accesses the database to retrieve the required data and then proceeds to perform the entire set of calculations for all the fasteners, storing the results in lists that are later used to properly display them in the GUI.
- export_data: This method allows the user to export the data of the current tab to an Excel file, allowing them to share the results of the analysis and perform further operations that are out of scope for it. It is called upon clicking the "Export to Excel" button on the GUI.

Interaction with the GUI functions include:

- restart_app: This method allows the user to restart the software. It is called upon clicking the "Restart" button on the GUI.

The diagram in Figure 42 shows the relationship and interactions within the methods in the class. Orange arrows indicate the methods called upon the user clicking a button on the GUI.

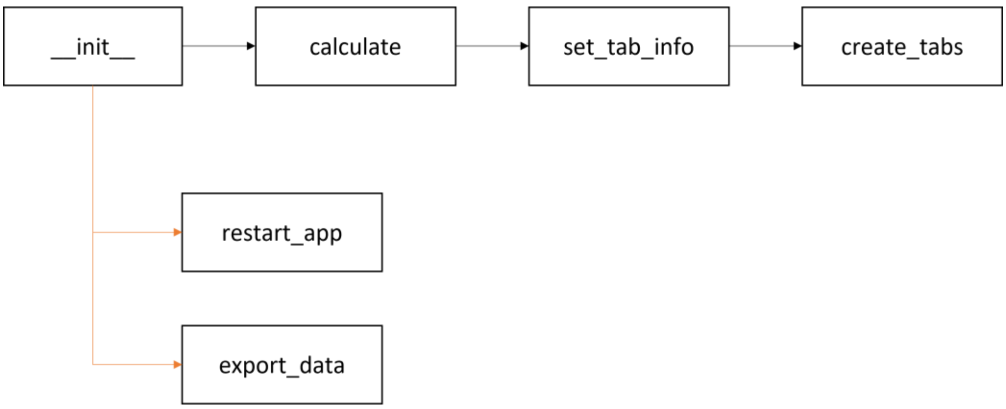

Figure 42 - OutputWindow Class Diagram

5.3. Graphic User Interface Development

Figure 43 - PyQt Logo

The GUI is developed directly in the Python environment using the PyQt6 library. A brief definition of the PyQt6 library is given in Table 34. This library was chosen due to its high capabilities regarding the building of customized GUIs. The flexibility is a consequence of its vast number of modules that are combinable to create the desired interface and its connection with the algorithm, while using the same coding structure as it, as it is done with the Python language.

The GUI is divided into three main sections or windows to match the classes of the algorithm presented in section 5.2.2.

5.3.1. Home Window Interface

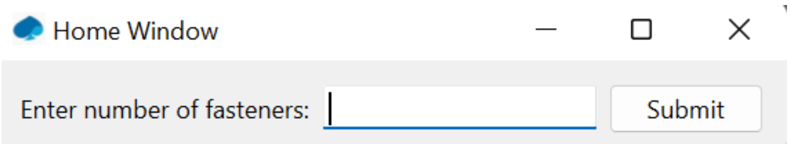

Figure 44 - Home Window Screenshot

The Home Window consists of a basic interface with a prompt, an input field, and a button. This window allows the InputWindow interface to have an initial setup. The window prompts the user to enter the number of fasteners for which the analysis will be performed, and it takes an integer as an input.

After clicking the "Submit" button, the Input Window, described in section 5.3.2, opens with the number of rows equal to the input in Home Window.

The number of rows is not absolute and can be modified directly on the Input Window, the Home Window serves only as an initializer for the interface.

5.3.2. Input Window Interface

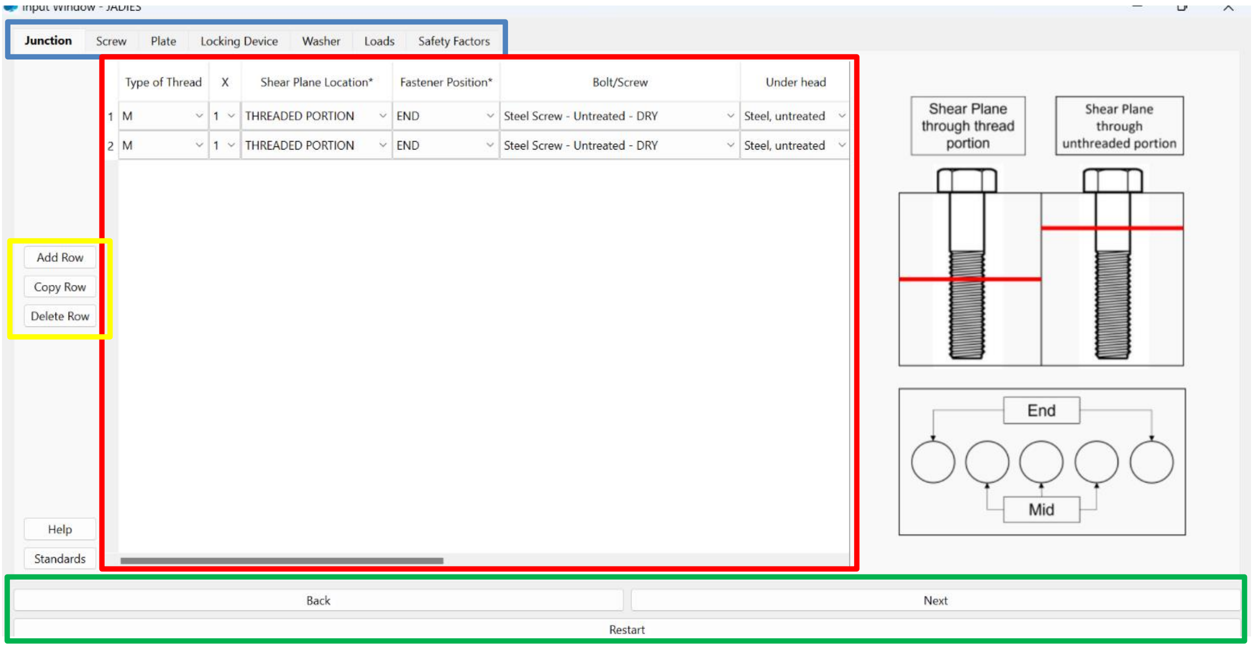

Figure 45 - Input Window Screenshot

The Input Window has a main interface conformed by seven tabs, highlighted with a blue rectangle in Figure 45, corresponding to the input categories. Each tab has the same structure with four subsections.

The main section of the page is the input area, highlighted by a red rectangle, this section presents the necessary input fields to the user. Each row represents a different joint, this applies across all tabs, so the first row on each tab corresponds to the same joint, and the same pattern follows for each row corresponding to a distinct joint. To the right of this section, an aid image is presented to the user. To the left of the input area, highlighted by a yellow rectangle, are three buttons that allow for modification of the input area. At the bottom of the screen there is another set of three buttons, highlighted by a green rectangle, which allow for an interaction with the main interface and application. The functions connected to the buttons in the input window interface are explained in section 5.2.2.2.

5.3.3. Output Window Interface

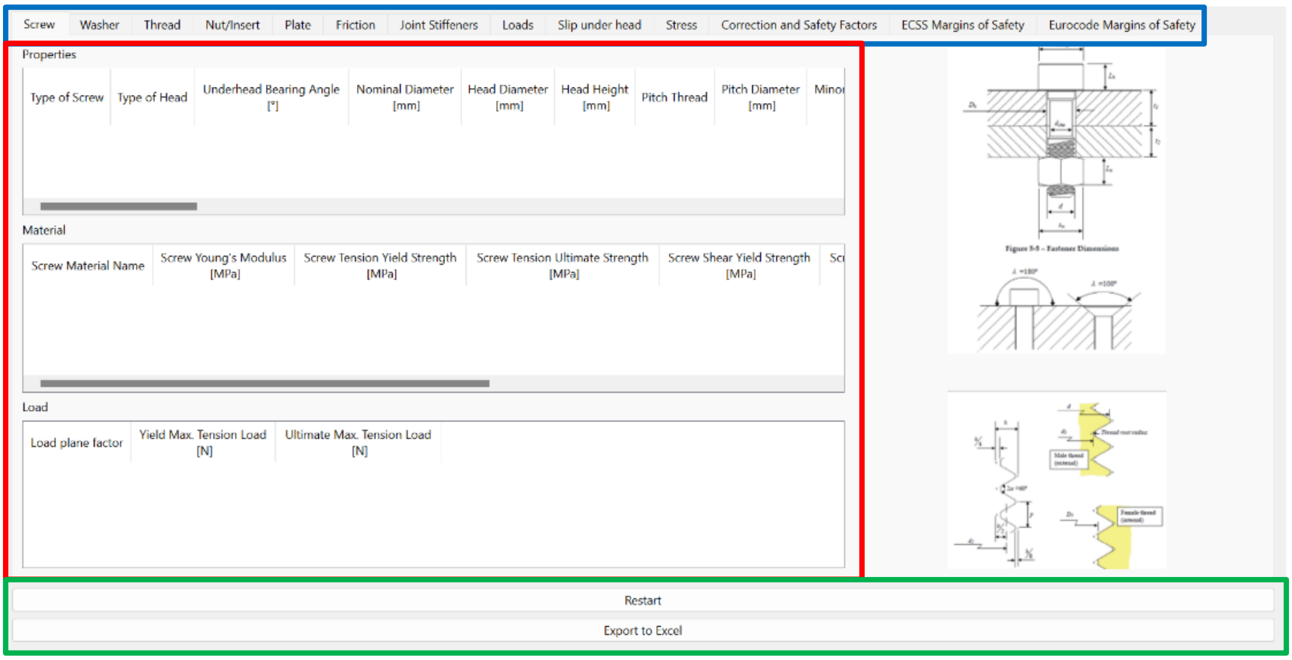

Figure 46 - Output Window Screenshot

The Output Window has a main interface conformed by thirteen tabs, highlighted with a blue rectangle in Figure 46, corresponding to the input categories. Each tab has the same structure with three subsections.

The main section is the display area, highlighted by a red rectangle, this section presents the results of the analysis conducted on the joints. Each row of results represents a different joint, with the same logic and order as in the input window. To the right of this area, as in the input window, there is an aid image presented to the user. At the bottom of the screen there is a set of two buttons, highlighted by a green rectangle, which allow for an interaction with the application. The functions connected to the buttons in the output window interface are explained in section 5.2.2.3.

6. Conclusions and Future Developments

6.1. Conclusion

The result of this project was the development of near-to-release software that can replace the current solution implemented in production for the automation of the process of designing jointed mechanical structures.

The current pre-release version of the software is a direct substitute for the current production tool, in terms of functionality, but provides a customized interface tailored to the use of the application.

The development and testing of the software opened the door for the creation of a dedicated software that will improve efficiency of day-to-day processes and will allow the implementation of custom functionalities that will provide a more in-depth insight about the joints of mechanical structures that will, in turn, increase the scope of what can be offered to the users and customers.

6.2. Future Developments

As with any software application, JADIES will be in constant development to fix any overseen issues prior to the release of the first version. Along with this, new functionalities will be added to increase the scope of its capabilities, some of which are already accounted for but are not essential for the day-to-day use of it, and others that will come to mind with the passing of time. Some of these future developments are presented below.

The future developments can be separated into two categories, the first being related to the scope of the algorithm and the second being improvements to the usability and friendliness of the interface.

6.2.1. Algorithm Developments

As discussed in section 1.1, this project considers the ECSS and Eurocode methodologies for threaded fasteners and, as seen in section 2.2, there are unthreaded fasteners that fall outside of the defined scope.

Further developments could consider adding these types of fasteners to the scope of the software. In addition, there are other methodologies that are used in the industry and that can be useful, such as the VDI standard that is widely used in the German industry.

The developments mentioned above cover an expansion of the current scope regarding the quantification of the efficiency of mechanical joints. However, there is a second path for further developments that involves the addition of other appendices of the ECSS standard, which is the most widely used.

The appendices considered for this development include the ECSS 32-22A that relates to inserts and the ECSS 32-21A that relates to adhesive bonding.

Other developments related to the algorithm of JADIES that fall outside the scope of the methodologies provided by the standards is the inclusion of Finite Element Method (FEM) calculations to provide a more in-depth verification of the results obtained by the software.

6.2.2. Interface Usability Developments

The current capabilities of the GUI are described in section 5.3. Developments regarding the improvement of the interface include:

- Improvements to the visuals of the interface, which include the addition of colors and highlights to active and key tabs or cells and the improvement of aid images.
- Addition of functionalities that do not interfere with the algorithm related to the methodologies:
    - Allowing the user to add data to the database, as it is complicated to have a database that includes all the components that might appear, as well as for the development team to keep the database fully up to date.
    - Improvements to the export function:
        - Allow the user to choose the save location.
        - Allow the user to export the inputs.
        - Better formatting for the export file.

- Allow the user to choose the key margins of safety for each calculation run, so that the software displays the fastener with the minimum value for the selected margins of safety.

All these functionalities and others that could come up in the future are not currently present in the software due to constraints of time and them not being essential for day-to-day operation.

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