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MechEng
Mechanical Engineering
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DESIGN AND FABRICATION OF A MECHANICAL ACTIVATED SHAFT ROLLER

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Declaration of authorship

I, Lam Quang Nghia, hereby state that this thesis is the result of my independent efforts. I have properly acknowledged all the assistance received during its creation. Any ideas or information obtained from external sources have been clearly attributed.

As of now, this thesis has not been submitted to any other evaluating body in a comparable format, nor has it been published.



Abstract

The expanding market for converters necessitates the need for flexible and efficient roll-to-roll manufacturing processes. Expanding shafts play a vital role in the packaging, manufacturing, printing, and converting industries by providing stable attachment and support at the entry and exit of roll-to-roll machines. In Vietnam, airshafts, which are activated by air, are commonly used.

However, air leaks in these shafts pose significant challenges, leading to machine malfunction. Another drawback of air shafts is their reliance on a pneumatic system. This thesis aims to address the drawbacks of airshafts by designing and fabricating a mechanical activated shaft roller, also known as a decoiler shaft.



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1. Introduction

1.1. Back ground study

The need for converters will increase as the converting market expands. Manufacturers will be pushed ever further by difficult tasks, and flexibility will be the key. In its most basic essence, Roll-to-Roll manufacturing involves a continuous procedure where a flexible substrate is unwound onto a production line and subsequently subjected to material deposition, cleaning, patterning, or other modifications. This step-by-step progression gradually constructs the desired structures on the substrate[1]. Expanding shaft is one of the most common components used at the entry and exit of a roll-to-roll machine in the packing, manufacturing, printing, and converting industries. These shafts are designed to expand after being put into the reel core and activated to provide stable and firm attachment. In most factories and prototype roll-to-roll machines in Vietnam, these expanding shafts are activated by air introduced in the shaft. In this case, these shafts are called airshafts. In the business community, air shaft is also referred to by a number of other names, including air expandable shaft, pneumatic expanding shafts, and air expanding shaft.

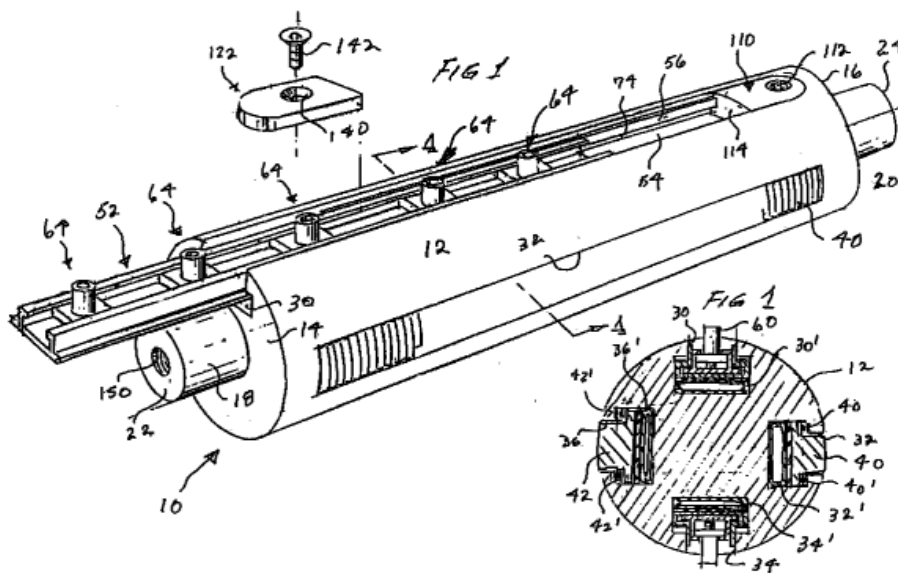


Figure 1. 1 pneumatic expanding shaft[2]

1.2. Introduction to problem

Despite some advantages of airshafts like swift insertion and extraction from the reels, and fast mounting and dismounting onto machinery, there are also some drawbacks. Air leaks are a problem that frequently occurs. When this issue arises, the machine might not function properly. There are essentially four causes of air expansion shaft leakage[3] (2). The first is that the sealing rings are separated by a space. The second case is that the air nozzle/valve leaks. This is due to a damaged or improperly placed air nozzle. To fix the issue, the air nozzle must first be replaced and then installed. The third one is that the interior inflatable rubber tube cracked and leaking. This happens as a result of the rubber bladder wearing out after prolonged use. The airbag must be replaced with a new rubber tube once it has aged and deflated inside the inflatable shaft and developed an air leakage phenomenon. The fourth reason is that the pressure ring is loose. To fix this issue, just fasten the pressure ring. Besides the shaft damage, compressed air is also needed to operate the shaft. An air compressor is required or industrial air compressor if airshafts are used in large scale. As a result, it serves as an excellent catalyst for the research and creation of web directing systems, which may subsequently be applied to the creation and introduction of products in the future. As a result, it serves as an excellent motivator for the research and development of Manual Expansion Decoiler, which may subsequently be applied to the manufacture and deployment of the devices in the future.

1.3. Objectives

This decoiler is a prototype that has the function of an expanding shaft placed at the entry and exit of a roll-to-roll machine. Other than the air-expanding shaft which is activated by air, this decoiler can expand and hold paper reel firmly manually by rotating the nut at one end. This thesis' main objective is to design and fabrication a decoiler, which is low cost and without any disadvantages of an airshaft like air leakage and facility requirements.

This prototype consists of four main components. The first component is the central shaft that is situated at the center. There is a hole at one end of the shaft to connect with the drive shaft by taper pin. The second component is four outer beams, their function is to expand and hold the paper core. The third component is the nut that can slide back and forth at the end of the central shaft to adjust expansion of the four outer beams. And finally, there are twelve bars to unite central shaft, outer beams and nut together.



1.4. Research problem

Dimensions is one of the constrain requirements need to be met. Because the Decoiler, in operation, is put into the reel core, the smaller diameter of the device is, the more materials that can be wound onto a reel. Small dimensions are quite challenging as it needs more precision, the cost is higher thereby. Moreover, the device is required to contain purely mechanical parts, no electric or pneumatic components is allowed to be used. Additionally, if the decoiler is considered as a shaft, this shaft is supported at only one end. So, bending is inevitable. Bending too much can lead to web twist and fatigue failure of the device. The design should be simple for machining and does not require complicated machining processes. In addition, it must strong enough to bear the load without much deflection.

2. Product Design Specifications

Product design specifications specify the parameters within which problem-solving approaches must follow. They are a collection of detailed, quantifiable requirements on the desired outcome of product development. The PDS considers the customer's needs and establishes quantifiable standards without imposing a particular design strategy to satisfy those requirements. By defining measurable specifications, the PDS takes into account client needs without imposing a particular design strategy to satisfy them. In order to generate as many possible designs as possible, the PDS is built prior to the concept generation stage in order to guide the ideas in the appropriate direction without limiting inventiveness.

Followings are some fundamental requirements of the devices:

- Dimension: be able to hold the reel with the width of at least 150 mm and inner core diameter from 60 to 200 mm (the smaller diameter the better).
- Strength: be strong enough to be supported at one end only end without much deflection.
- Function: be able to hold the reel with the web tension of 100N. Have the ability to keep expanding in order not to lose friction with the paper reel during the operation of the roll-to-roll machine.

3. Conceptual Design

The conceptual design phase represents an initial stage within the design process. It shows how the components are organized. There were two concepts generated and discussed. Components of these concepts are represented by blocks for visual illustration.

- The first concept, temporarily called the expanding shaft, consists of a 100 mm hollow tube with an 80 mm inner diameter. On this tube, there are three slots arranged radially symmetric. Each slot then will hold a lug which is supposed to expand outward to hold paper reel. A core shaft is design to slide on the interior of the tube. Below each lug are two cam followers, their function is to follow the cam profile one the core so that the lug can move in or out. The inner core is driven by thread.
- The second concept is in the can be called a decoiler because of its shape and working mechanism. Backbone of the device is the central shaft which has all other components attached on it. Four outer beams are supported by eight bars and pushed to expand by the nut via four bars.

4. Concept selection



4.1. Design selection criteria

Selection criteria indicate the crucial needs for a high-quality product and are features for determining the optimum overall designs. Design selection criteria refer to the factors or considerations that are used to evaluate and choose a particular design option or solution. These criteria are typically established based on the specific requirements, goals, and constraints of a project or problem. Design selection criteria help in making informed decisions and selecting the most suitable design among multiple alternatives. There are three criteria, as described below:

- **Rigidity:**
A mechanical design that displays less deformation when subjected to load and moment is considered to have a high level of rigidity. The concept that has fewer components and these components have larger dimensions tends to be rigid and good at enduring forces. The first concept looks more rigid as the ϕ tube is more optimum to resist forces and moments compare to central shaft of concept 2.

- **Adaptability:**

An ability to be adjusted to hold different reel cores diameter. This feature is very important because it give the operators more convenient. The operators do not need to change another device for different reel core diameter. Additionally, this feature also saves customers money because they do not need to buy different devices for different core diameters as long as the core diameters are within range of the device. Concept decoiler more than adequately satisfies this need, whereas concept expanding shaft does not. The first concept can only support the core with the same diameter as its shaft diameter or just a slightly bigger.

- **Ease of fabrication and maintenance**

For fabrication, a design with less complexity and low manufacturing cost should be preferred. For maintenance, a better design should have parts which have some similarities and can be interchanged. Devices with inexpensive maintenance techniques typically have lower total costs.

Regarding this requirement, the uncoiled concept has more advantage than concept expanding shaft because of the simplicity of each part. Three linear cam of concept 2 is very tough to machine. It even requires 5-axis milling machine and therefore the cost will be quite high. It even requires a 5-axis milling machine. The cost, therefore, will be quite high compared to just using a 3-axis milling machine. Considering the significance of this criterion in terms of cost savings, manufacturing efficiency, and long-term maintenance benefits, it becomes crucial to prioritize designs that excel in ease of fabrication and maintenance. By focusing on designs that are straightforward to produce and maintain, businesses can optimize their production processes, minimize expenses, and ensure smooth operations throughout the lifecycle of the product. Among the four selection criteria mentioned, the third criterion, ease of fabrication and maintenance, appears to be the most important.

4.2. Final concept selection

Based on the selection criteria provided, it is clear that concept 2, or the second concept, is the preferred choice. While the first concept may have an advantage in terms of rigidity due to its optimized ϕ tube, concept 2 excels in the criteria of adaptability and ease of fabrication and maintenance, which are deemed to be more important. Concept 2 offers the ability to adjust and accommodate different reel core diameters, providing convenience for operators and cost savings for customers. On the other hand, the first concept can only support cores with the same diameter as its shaft or slightly larger, limiting its adaptability. In terms of ease of fabrication and maintenance, concept 2 surpasses the first concept due to the simplicity of its parts. The three linear cams in concept 2 are challenging to machine and require a 5-axis milling machine, resulting in higher costs compared to the simpler parts of concept 1 that can be manufactured with a 3-axis milling machine. Considering the significance assigned to the ease of fabrication and maintenance criterion, which emphasizes cost savings, manufacturing efficiency, and long-term maintenance benefits, concept 2 emerges as the more suitable choice. By selecting concept 2, businesses can achieve a balance between rigidity and adaptability while prioritizing cost-effectiveness, streamlined manufacturing processes, and simplified maintenance procedures. Therefore, based on the provided selection criteria, concept 2 is the recommended option for further development and implementation.

5. Design process

5.1. Design methodology

After the concept is selected, the design process follows the sequence of geometry selection, stress calculation, and material selection, with dimension alteration as needed to meet the required strength criteria. This approach is commonly used in engineering and design disciplines, particularly in mechanical engineering and structural engineering.

This design method is broken down step-by-step as follows:

1. Geometry selection: The first step in the design process is to determine the geometry or shape of the component or structure being designed. This involves selecting appropriate dimensions, shapes, and configurations based on the desired function, performance requirements, and other design constraints.

2. Stress calculation: Once the geometry is selected, stress analysis techniques can be used to calculate the internal stresses and strains that will occur in the component or structure under the anticipated loads and operating conditions. This typically involves using engineering principles and mathematical modeling to analyze the structural behavior and determine the stress distribution throughout the component or structure.
3. Material selection: After stress calculation, suitable materials can be considered for the design. The selection of materials is based on their mechanical properties, such as strength, stiffness, toughness, and other relevant characteristics. The materials should be able to withstand the anticipated loads and stresses without experiencing excessive deformation or failure.
4. Material strength evaluation: Once the material options are narrowed down, the calculated stresses can be compared to the strength properties of the materials to determine if they are adequate for the design requirements. If the calculated stresses exceed the material's strength limits, it may be necessary to reconsider the material selection or modify the design.
5. Dimension alteration: If the selected material does not provide the required strength, one option is to alter the dimensions of the component or structure. This may involve adjusting the geometry, size, or proportions of the design to reduce stress concentrations, increase cross-sectional areas, or otherwise improve the structural integrity.
6. Iterative process: The design process is often iterative, involving multiple iterations of geometry selection, stress calculation, material selection, and dimension alteration until the design meets the desired strength criteria and other performance requirements.

5.2. Dimensions and materials selection

With the aim for four outer beams to expand from 95 mm to 120 mm and for the decoiler to support the reel with of 150 mm , some crucial dimensions are initially determined. Central shaft is 290 mm in length and has a diameter of 16 mm . The length of each bar is 40 mm . This length is the distance between the two holes on of the bar. These holes are to connect

the bars to the hubs and outer beams. The distance of the two hub is 147 mm and the distance from the nearer hub to central shaft end is 35 mm .

The central shaft is the component that bears the most load compared to other components. Other components, such as bars, beams, nut, and hubs, are not supposed to bear as much load as the central shaft. So, the material for these components is not necessary to be as strong as that of the central shaft, but it needs to be lighter to reduce the weight of the device.

C45 steel is selected for central shaft because of the following criteria:

- **Strength and hardness:** C45 steel is known for its high strength and hardness. Its tensile strength is around $570\text{-}700\text{ MPa}$, and its hardness is from 170HB to 210 HB [4]. It has a good combination of toughness and wear resistance, making it a good choice for applications where the material will be subjected to high stress or wear.
- **Machinability:** C45 steel has good machinability, meaning it can be easily shaped and worked with conventional machining processes like drilling, milling, and turning. This makes it a popular choice for manufacturing applications where precision and consistency are important.
- **Cost-effectiveness:** C45 steel is relatively inexpensive compared to many other high-strength materials, making it a cost-effective option for many industrial applications.

Al 6061 is chosen as the material for the other components. Al 6061 is an aluminum alloy with a tensile strength of around 276 MPa and a hardness of around 95 HRB [5]. It is commonly used in applications that require lightweight materials with good corrosion resistance. Moreover, Aluminum 6061 is widely available and can be found in various forms, such as sheets, plates, bars, and tubing, making it easy to source and work with. In addition, this alloy is easy to machine, and can be drilled, cut, and formed with relative ease. This makes it a popular choice for manufacturing parts and components that require precision and accuracy.

5.3. Central shaft

Designing process of central shaft is shown in this section. A mechanical shaft is a rotating component that transmits power and rotational motion. It typically has a circular cross-section and can be solid or hollow. Power is transmitted through a machine through components installed on the shaft, including gears, pulleys (sheaves), flywheels, clutches, and sprockets[6]. This central shaft is designed so that on one end, it can be connected to a drive shaft; in the middle section, two hubs can be mounted; and a lead screw on the other end for the nut to be mounted and adjust the expansion.

In this case, force is transmitted from paper tension to drive shaft follows the sequence: paper, beam, bars, hubs, central shaft and drive shaft. Figure illustrates how the force flow from the web tension to the drive shaft.

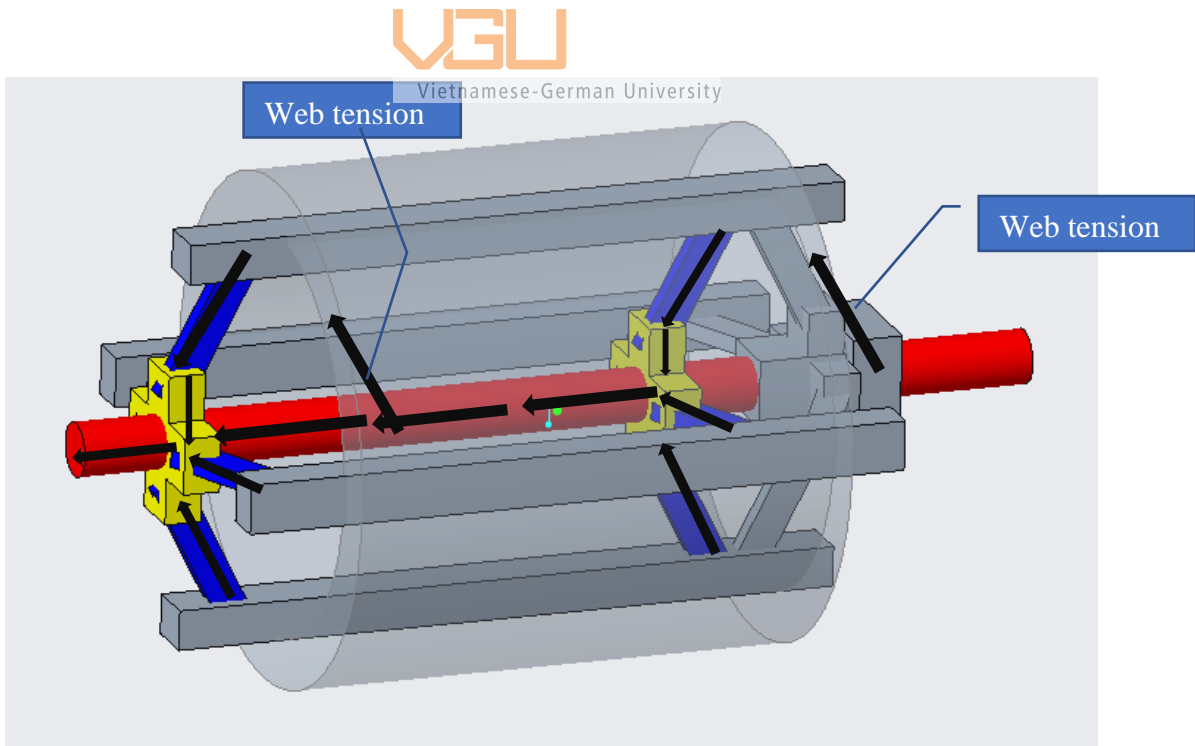


Figure 5. 1 design of the decoiler

Assuming the deflection of the shaft is very small and the reel is situated in the middle of the bearing bars, the two hubs is now transmitting the same forces and moment to the central shaft. For simplicity, the two are displayed in dishes shape with diameter of 200 mm, this is the maximum diameter allowed in this machine. 50 N and 5000 Nm are force and moment each hub transmits in the worst-case scenario. Both of the forces acting on the dishes are transverse to the axis of the shaft, which will cause the shaft to bend normally. The bending stress shifts from compressive to tensile to compressive to tensile as the shaft spins. The shaft then experiences a completely reversing stress.

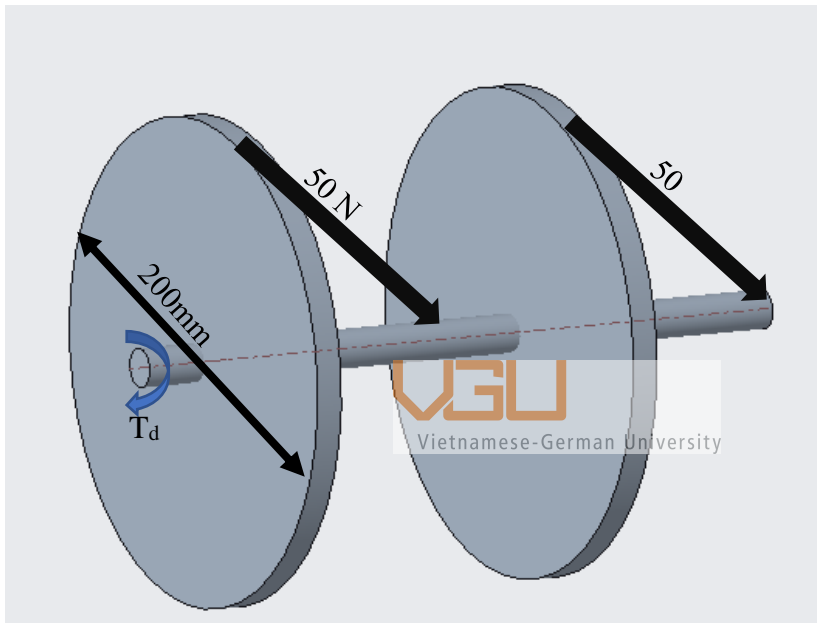


Figure 5. 2 Torque diagram of central shaft

Dish torque on one dish (T) = web tension at each dish times the radius ($r=100\text{N}$).

$$T = (50\text{N}) \times (100\text{mm}) = 5000\text{Nm} \quad (1)$$

Two external forces on two dishes are on the same direction and there is only one reaction force at the left end. Consequently, the reaction force and the two forces on the pulley must lie on the same plane.

When drawing shear force and bending moment diagrams, there are standard sign conventions that should be followed to ensure consistency and accuracy[7]. These

conventions are based on the assumed positive direction for shear forces and bending moments at a given section of a beam. Below are the sign conventions used when sketching shear force and bending moment diagrams:

1. Shear force diagram:

- The positive direction of shear force is upward.
- The negative direction of shear force is downward.
- The shear force changes sign across a point of concentrated load or support.

2. Bending moment diagram:

- The positive direction of bending moment is clockwise on the compression side of the beam and counterclockwise on the tension side of the beam.
- The negative direction of bending moment is counterclockwise on the compression side of the beam and clockwise on the tension side of the beam.
- The bending moment changes sign at a point of zero shear force.



Free body diagram of central shaft:

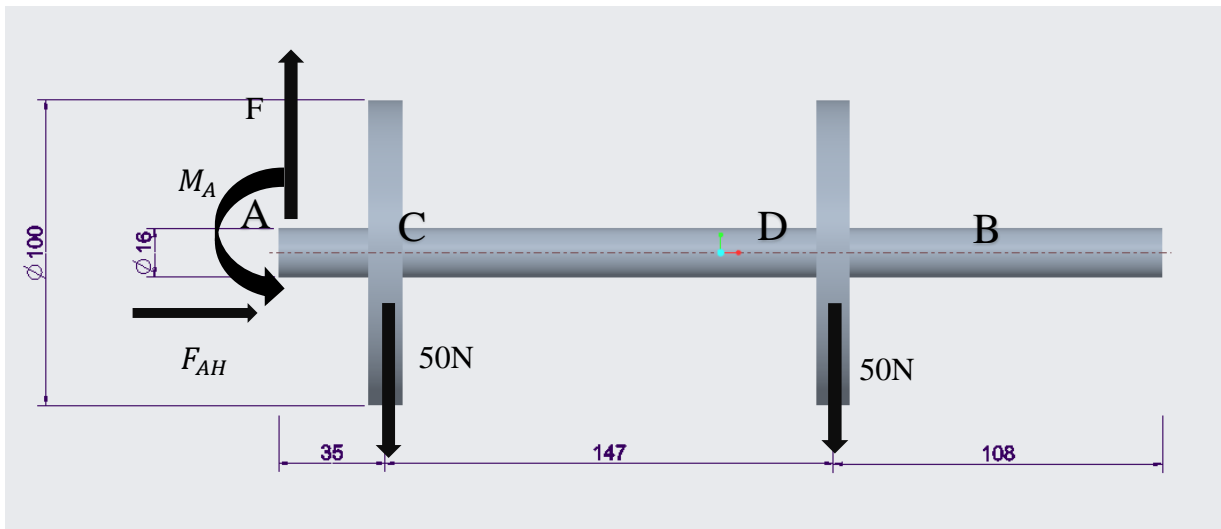


Figure 5. 3 Free body diagram of central shaft

The central shaft is rotating at a constant speed, it is in a state of dynamic equilibrium. In this state, the total of all the forces and moments exerted on the shaft must be equal to zero, just as in a static state of equilibrium.

Utilize the equations of equilibrium to find the unidentified reaction forces acting at Point A. The forces acting in the horizontal direction must add up to 0. Since F_{AH} is the sole force acting in the horizontal direction, it must possess a value of 0:

$$\sum F_x = 0 \quad (2)$$

$$F_{AH} = 0 \quad (3)$$

The total of the vertical forces must balance out to 0:

$$\sum F_y = 0 \quad (4)$$



$$F_A - 50N - 50N = 0 \quad (5)$$

$$F_A = 100N \quad (6)$$

Moreover, the sum of the moments around any given point should be equal to 0. Calculation takes place at point A to solve for moment at A:

$$M_A - 35mm \times 50N - 182mm \times 50N = 0 \quad (7)$$

Then, moment at A is

$$M_A = 10.85Nmm = 0.01085Nm \quad (8)$$

The free body diagram may now be revised because all of the external loads pulling on the central shaft have been identified.

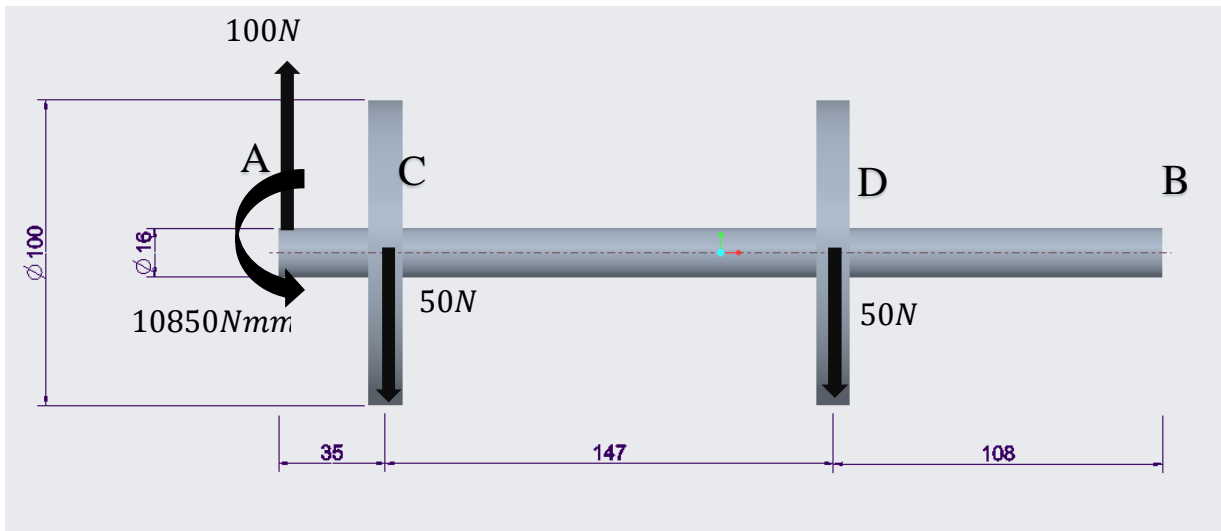


Figure 5. 4 Free body diagram of central shaft with moment and reaction force at A have been calculated

Section DB does not bear any external load so shear forces and bending moment are not calculated at this section.

The diagrams are obtained by cutting the shaft at arbitrary point x in section A-C and constructing free-body diagrams for the pieces to the left and right of the cut. The value x could be from 0 mm to 3 mm.

$$V(x) = 100N \quad (9)$$

$$M(x) = V \times x + C = 100x + c \text{ (C is constant)} \quad (10)$$

Base one boundary condition, moment at A is $10850Nmm$: $M(0) = -10850 Nmm$. The value of C is then -10850 . As a result:

$$M(x) = -10850 + 100x \quad (11)$$

Substitute $x = 35$ to find the moment at point C:

$$M_c = M(35) = -10850Nmm + 100N \times 35mm = -7350Nmm \quad (12)$$

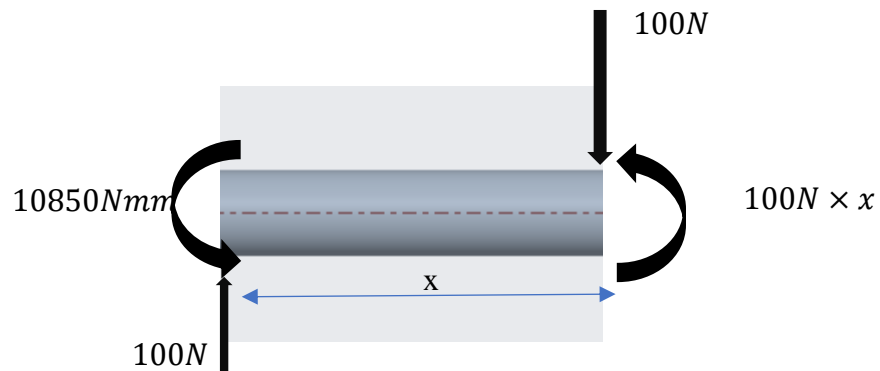


Figure 5. 5 Diagram depicting the free body forces at an imaginary section located immediately to the right of the support

This process can be repeated for the section C-D. In this section, x run from 35 to 182.

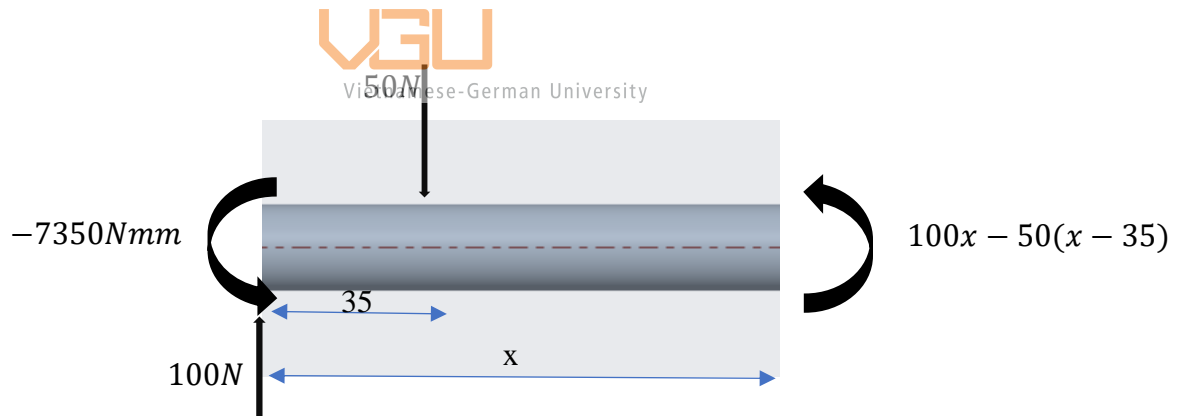


Figure 5. 6 Diagram illustrating a hypothetical section located at a greater distance to the right

$$V(x) = 100 - 50 = 50N \quad (13)$$

$$M(x) = -10850 + 100x - 50(x - 35) \quad (14)$$

$$M(x) = -7350 + 50(x - 35) \quad (15)$$

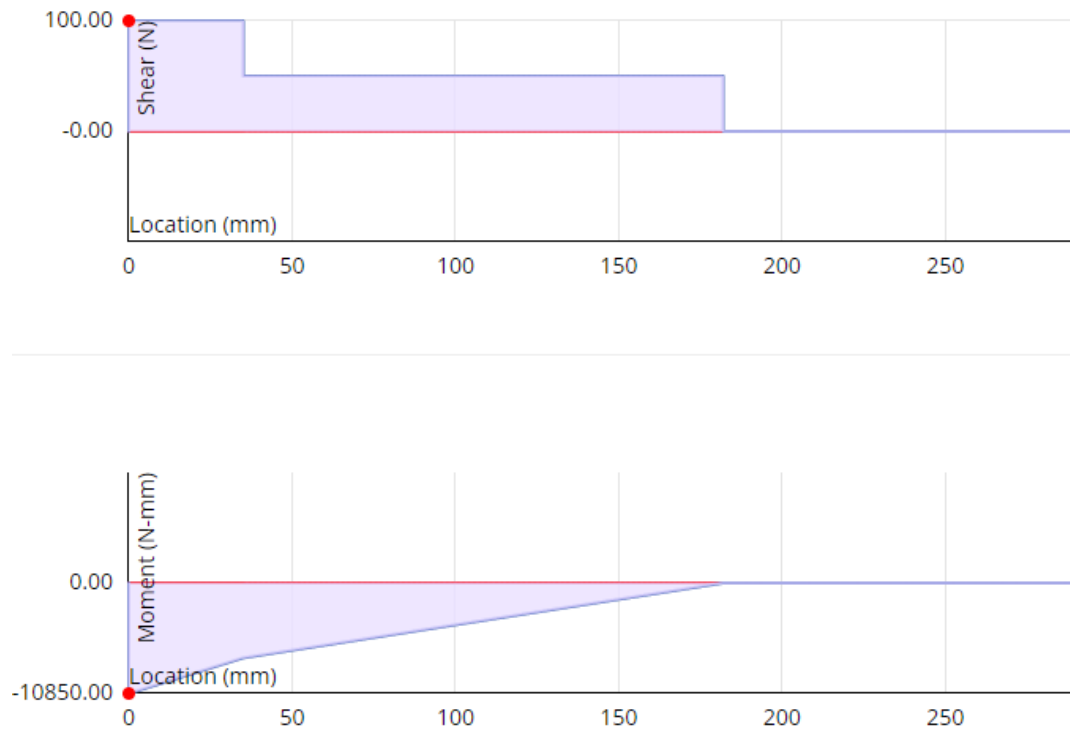



 Figure 5. 7 Shear force V and Bending moment M
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5.3.1. Connection feature

The type of joining the central shaft to the drive shaft is based on the existing configuration of the drive shaft of the unwinding and rewinding in the laboratory. The diameter of the segment is 14 mm, and the length of this segment is 19 mm. Additionally, there is a hole perpendicular to and intersecting the drive shaft axis. With these dimensions of the of the available shaft, one end of central shaft can be design like a hub with outer diameter and inner diameter is equal to the diameter of the drive shaft. Taking advantage of the 4mm hole, a radial pin is used in this shaft hub connection.



Figure 5. 8 Connecting section of drive shaft



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The capacity of a coiled spring pin to prevent hole damage is one of the main advantages of utilizing one to fasten a hub or gear to a shaft. The capability of the coiled pin to tolerate broader hole tolerances than any other press-fit pin is another advantage. This results in decreased assembly manufacturing costs overall. There are some considerations in designing shaft and hub[8]:

1. A shaft's hole shouldn't be larger than one-third of the shaft's diameter. Standard duty pins are advised for shafts made of mild steel and nonferrous metals. The hole on the shaft is 4 mm, it is less than $\frac{1}{4}$ the diameter of the shaft then, this requirement is met.
2. It is advised that the hub's wall thickness be at least 1.5 multiply by the size of the pin's diameter. Under different circumstances, the durability of the hub won't match the pin's shear strength (Figure 5.9). The hub's wall thickness and the amount of material surrounding the pin both grow. The diameter of the drive shaft is 4 mm then the outer diameter of the hub should be at least 26 mm. To satisfy this requirement, the outer hub's diameter is chosen to be 30 mm and the length is 28.85 mm.

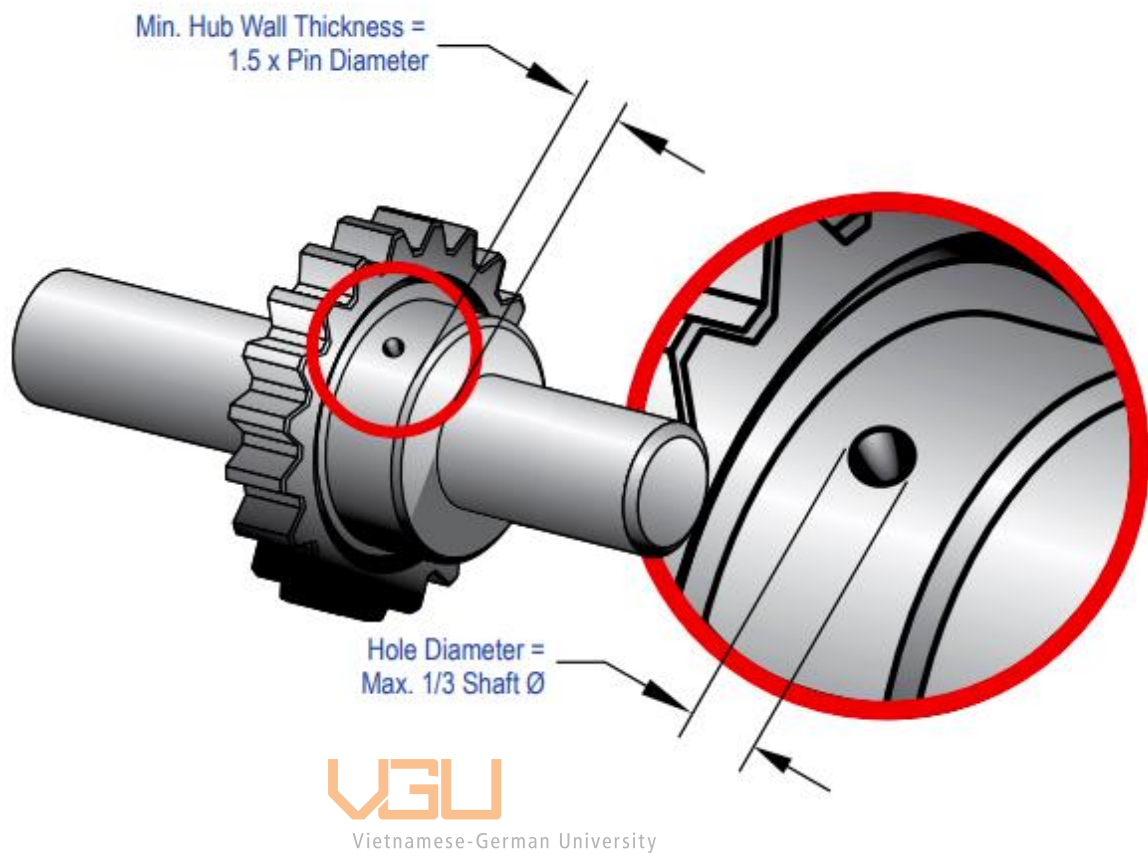
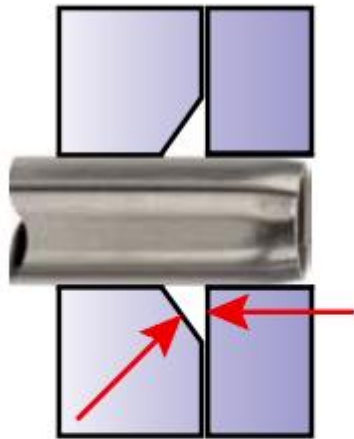


Figure 5. 9 Min hub wall thickness and hole diameter [8]

3. To prevent the pin from moving within the holes, the diameter of the holes through the shaft and hub should be precisely aligned. To prevent movement of the parts relative to one another, it is advised that the difference between the hole sizes in the hub and shaft should not surpass a tolerance of 0.05mm. Alternatively, the pin will experience dynamic forces, which might cause the assembly to be significantly impacted by a very minor change in velocity. The hole on the hub will be equal in diameter to the hole in the shaft to prevent the moment of the pin.
4. To avoid stress concentration and make sure there is enough material around the pin to sustain the applied forces, the hole should be centered in both the shaft and the hub.
5. If the holes cannot be perfectly aligned, it is advised to divide the tolerance between the shaft and hub. The component with the longest engagement length should receive the bigger half of the tolerance, while the other component should receive the smaller half.

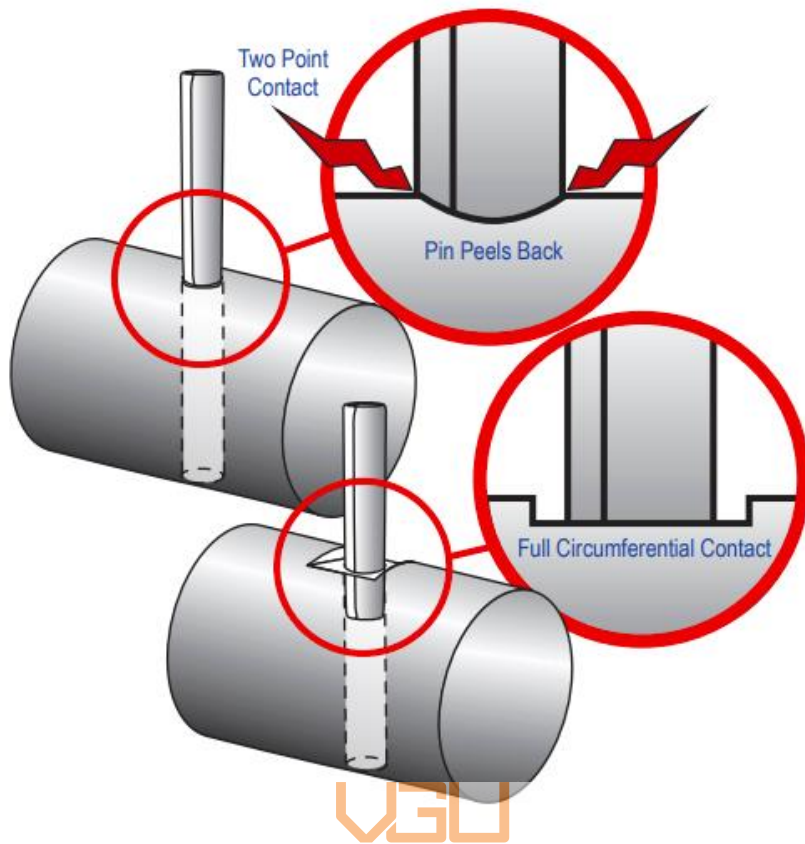
6. On-hole countersinks are not advised. Additionally, the shaft's outer diameter (OD) and the collar's inner diameter (ID) should be constructed so that their respective shear plane separations (OD-ID) do not exceed 0.13 millimeters (.005"). In both situations, there is an unsupported length of pin in a potential area for torque. This might result in a bending moment, reducing the pin's lifespan (Figure 5.10).



A countersink increases the distance between the shear planes. This can put the pin in bending and thus reduce the strength of the pin.

Figure 5. 10 Countersinks on the holes[8]

7. Two points of contact are created when a cylindrical pin is inserted into a hole on the surface of another cylindrical object. As a result, only two spots on the circumference experience compression. A flat should be put on the outside of the hole to improve the contact area between the perimeter and the pin and to facilitate installation. (Figure 5.7).



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Figure 5. 11 Contact surface between the periphery of the hole and pin[8]

To make sure this shaft-hub connection does not yield on the first cycle, the minimum strength which each component must provide should be computed. Following are equations to determine those values:

Pin shearing stress:

$$\tau = K_{sb} \frac{4 \cdot T}{\pi \cdot d^2 \cdot D} \quad [MPa, psi] \quad (16)$$

Pressure on shaft:

$$p_1 = K_{sb} \frac{6 \cdot T}{d \cdot D^2} \quad (17)$$

Pressure on hub:

$$p_2 = K_{sb} \cdot \frac{4 \cdot T}{d \cdot (D_1^2 - D^2)} \quad [MPa, psi] \quad (18)$$

Shaft torsional stress:

$$\tau = K_{sb} \cdot \frac{16 \cdot T}{\pi \cdot D^3 \cdot \left(1 - 0.9 \frac{d}{D}\right)} \quad [MPa, psi] \quad (19)$$

in which formulas T = torque in newton-millimeters ($N \cdot mm$); τ = shear stress in newtons per millimeter square (N/mm^2); d^2 and D denote dimensions shown[9]; and K_{sb} is the service factor. A service factor of one is typically chosen when the equipment is operated under well-controlled and predictable conditions, and the likelihood of unexpected overloads or other variations in operating conditions is low. This roll-to-roll machine is placed in laboratory so the conditions like temperature and humidity, etc.... are well-controlled. The service factor, therefore, is chosen to be one.

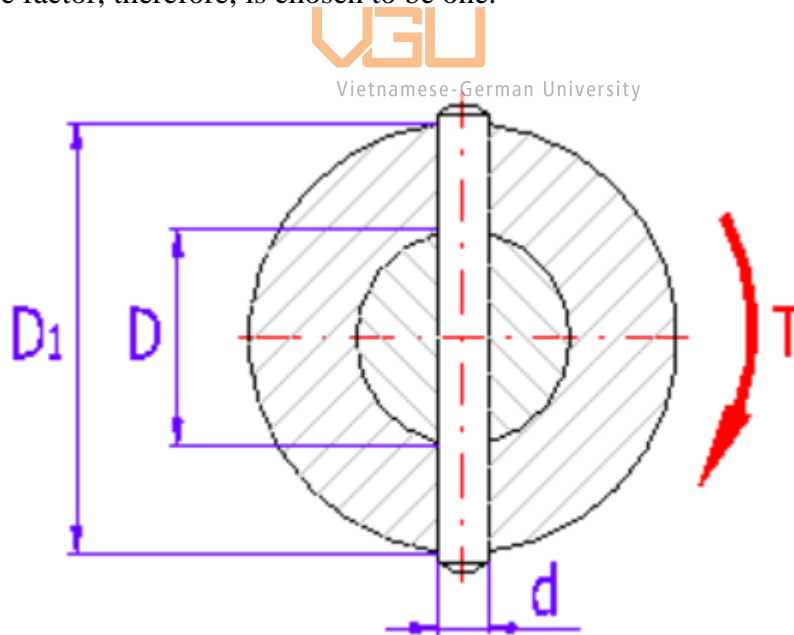


Figure 5. 12 Cross section of radial pin for shaft-hub connection[9]

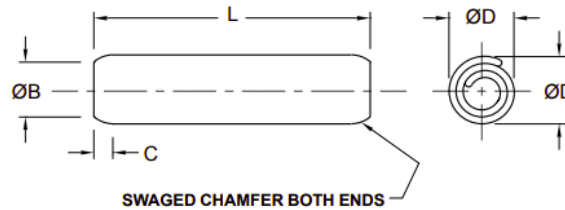
Substitute the subsequent parameters to four equations about to get the minimum strength required of each component in the hub-shaft connection:

- $T = 10000N \cdot mm$ (To be conservative, maximum torque is used in this calculation)
- $d = 4 mm$
- $D_1 = 30 mm$
- $D = 14 mm$
- $K_{sb} = 1$

The values of Pin shearing stress: pin shear stress, pressure on shaft, pressure on hub and shaft torsional stress are $57 N/mm^2$, $77 N/mm^2$, $14,2 N/mm^2$ and $25 N/mm^2$ respectively.

SPIROL's Coiled Pin is considered for this shaft-hub connection. The Coiled Pin, a meticulously engineered fastener, offers three different "duties" to cater to diverse host materials and application requirements. This allows designers to select the optimal combination of strength, flexibility, and diameter. The Coiled Pin uniformly disperses both static and dynamic loads across its cross section, avoiding stress concentration at specific points. The orientation of the external force does not impact the material's pliability and resistance to shear forces, eliminating the need for orientation during assembly to optimize performance. In dynamic assemblies where impact loading and wear can cause failures, Coiled Pins are specifically designed to retain their flexibility after setup and actively contribute to the assembly. They effectively dampen shock/impact loads and vibrations, preventing damage to the holes and extending the overall lifespan of the assembly.

A light-duty coiled pin with a diameter and length of 4 mm and 30 mm, respectively, is chosen based on the pin's predicted needed shear stress.



NOMINAL DIAMETER		1.5	2	2.5	3	3.5	4	5	6	8
DIAMETER ØD	MAX.	1.75	2.28	2.82	3.35	3.87	4.45	5.50	6.55	8.65
	MIN.	1.62	2.13	2.65	3.15	3.67	4.20	5.20	6.25	8.30
CHAMFER	B DIA. MAX.	1.40	1.90	2.40	2.90	3.40	3.90	4.85	5.85	7.80
	C LENGTH REF.	0.50	0.70	0.70	0.90	1.00	1.10	1.30	1.50	2.00
RECOMMENDED HOLE SIZE	MAX.	1.60	2.10	2.60	3.10	3.62	4.12	5.12	6.15	8.15
	MIN.	1.50	2.00	2.50	3.00	3.50	4.00	5.00	6.00	8.00

Note: All dimensions apply prior to plating.

MINIMUM DOUBLE SHEAR STRENGTH kN

NOMINAL DIAMETER		1.5	2	2.5	3	3.5	4	5	6	8
CARBON ALLOY STEEL CHROME STAINLESS		0.80	1.50	2.30	3.30	4.50	5.70	9	13	23
NICKEL STAINLESS		0.65	1.10	1.80	2.50	3.40	4.40	7	10	18

Note: Shear tests performed in accordance ASME B18.8.3M and ISO 8749.

Table 5.1 Shear strength of coiled pins



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In table 5.1 the value of double shear strength is shown. In essence, the pin's resistance to shearing forces is determined by the maximum force it can bear before fracturing when the force is exerted at a right angle to the pin's axis. Pins can undergo shearing in various orientations, resulting in distinct outcomes. For instance, a pin that fractures under single shear conditions would lead to the formation of two distinct pieces, whereas a pin that experiences failure under double shear conditions would yield three separate pieces as a result.

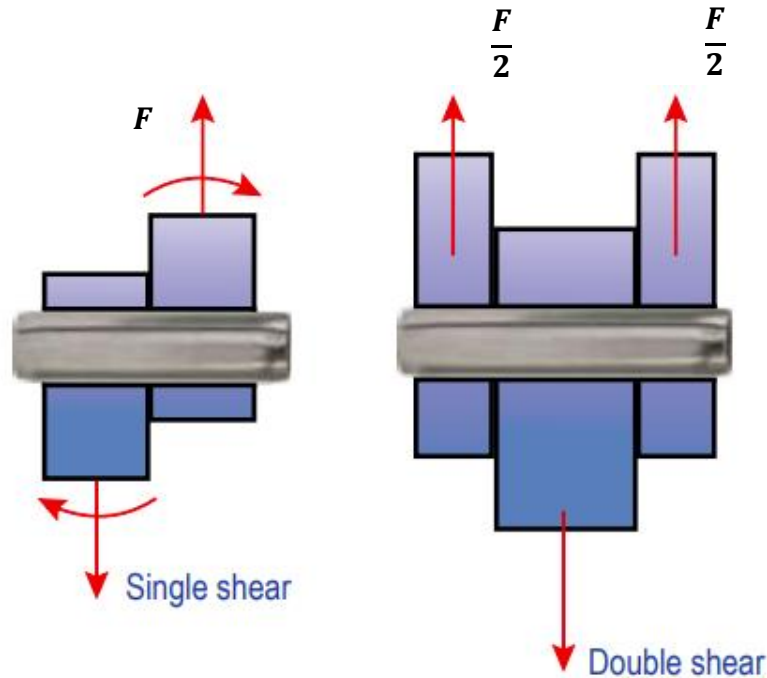


Figure 5. 13 Double shear and single shear[8]

As the figure (5.13) shows, the pin in this shaft-hub connection is considered to bears double shear during operation. Equation for double shear is

$$\tau_{ave} = \frac{F}{2 \cdot A} = \frac{F}{2 \cdot \pi \cdot r^2} \quad [MPa] \quad (20)$$

Substitute the value of F given in figure 5.13 to equation (20) to find the minimum shear strength of a Spirol Light Duty Coiled Pin made of nickel stainless and a diameter of 4 mm.

$$\tau_{ave} = \frac{F}{2 \cdot A} = \frac{F}{2 \cdot \pi \cdot r^2} = \frac{4400}{2 \cdot \pi \cdot 2^2} = 175 \text{ N/mm}^2 \quad (21)$$

This is about three times the shear stress requirement of the pin. So, a Spirol Light Duty Coiled Pin made of nickel stainless and a diameter of 4 mm is selected for this shaft hub connection.

The hub diameter is 30mm and shaft diameter is 16mm. At the point $x = 28.85\text{mm}$ where the diameter changes, the stress distribution in the shaft also changes abruptly, leading to a

localized increase in stress known as stress concentration. It is crucial to calculate and check the stress concentration at the junction of the different diameters in the step-shaft to ensure that the shaft can withstand the intended load without failures such as unexpected deformation, fracture, or even catastrophic failure of the shaft, which can cause severe damage to machinery or pose a safety hazard to personnel.

Equations (22) define the stress concentration factors, which relate the maximum stress at a discontinuity to the nominal stress. These factors pertain to theory or geometry and are symbolized by K_t .

$$K_t = \frac{\sigma_{max}}{\sigma_0} \quad (22)$$

Where σ_{max} and σ_0 are maximum stress and nominal stress, respectively.

Certain materials display a diminished response to the presence of notches, indicating that a lower value of K_t may be utilized for them. Consequently, the maximal effective stress in fatigue for these materials is:



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$$\sigma_{max} = K_f \sigma_0 \quad (23)$$

K_f denotes a decreased value of K_t , and σ_0 represents the nominal stress. K_f is commonly referred to as the fatigue stress-concentration factor, denoted by the subscript "f." It is useful to conceptualize K_f as a stress-concentration factor that has been reduced from K_t , owing to a lowered sensitivity to notches.

To calculate stress concentration for reversed bending stress, K_f is needed to be found. The reduced stress concentration factor (K_f) can be estimated using the Neuber equation. The Neuber equation is an empirical equation that relates the equivalent stress and strain in a highly stressed region to the nominal stress and strain in the surrounding material:

$$K_f = 1 + \frac{K_t - 1}{1 + \sqrt{a/r}} \quad (24)$$

where \sqrt{a} is defined as the Neuber constant and is a material constant and r is the notch radius. \sqrt{a} can be calculated through this equation:

$$\sqrt{a} = 0.246 - 3.08(10^{-3})S_{ut} + 1.51(10^{-5})S_{ut}^2 - 2.67(10^{-8})S_{ut}^3 \quad (25)$$

In order to determine the geometric stress-concentration factor K_t , consult figure 5.14 using the values $\frac{D}{d} = \frac{30}{16} = 1.875$ and $\frac{r}{d} = \frac{1}{16} = 0.0625$, which correspond to ratios of diameters and radii, respectively. The value of K_t can be read from the chart as 2.1.

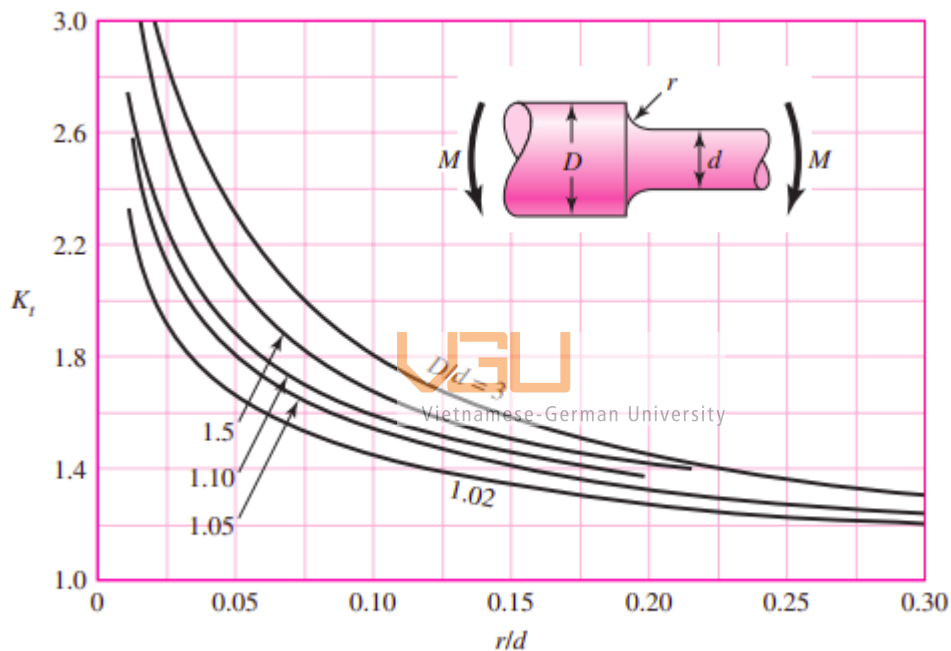


Figure 5. 14 Round shaft with shoulder fillet in bending[9,p.1028]

S_{ut} of AISI 1045 Steel or C45 steel is 565 MPa. Replacing the value of $S_{ut} = 565/6.89 = 82$ *kpsi* into equation (25) gives $\sqrt{a} = 0.0802\sqrt{in} = 0.04\sqrt{mm}$. Substituting this into equation (24) gives

$$K_f = 1 + \frac{K_t - 1}{1 + \sqrt{a/r}} = 1 + \frac{2.1 - 1}{1 + 0.04/\sqrt{1}} = 2.05 \quad (26)$$

The bending stress at location $x=28.85 \text{ mm}$ needs to be estimated next. The moment of bending is

$$M_{x=35} = F_A x = 100 \cdot 0.02885 = 2.885 \text{ N} \cdot \text{m} \quad (27)$$

The section modulus is $\frac{I}{c} = \frac{\pi d^3}{32} = \frac{\pi 16^3}{32} = 402.12 \text{ mm}^3 = 4.0212(10^{-7}) \text{ m}^3$ just to the right of $x=28.85 \text{ mm}$. Considering an infinite life, the reversing bending stress is

$$\sigma_{rev} = K_f \frac{M_B}{I/c} = 2.05 \frac{2.885}{4.0212(10^{-7})} = 14707674.33 \text{ Pa} = 14.707 \text{ MPa} \quad (28)$$

With a yield strength of 310 MPa, C45 steel is 21 times stronger than the calculated reversing bending stress. This suggests that the shaft made of this material should be able to endure the applied stress without succumbing to permanent deformation or failure.

5.3.2. Power Screw

A power screw is a mechanical apparatus employed to transform rotary motion into linear motion, or conversely, linear motion into rotary motion. It consists of a threaded screw and a mating nut, where the screw typically has a helical thread form and the nut is designed to engage with the screw's threads. When the screw is rotated, the nut translates along the length of the screw, producing linear motion. Power screws are widely used in various applications, including heavy machinery, robotics, aerospace, and automotive industries. They can be classified based on different criteria such as the type of thread form, the type of load transmission, and the direction of motion. Power screws can provide high mechanical advantage, precision positioning, and high load-carrying capacity, making them suitable for a wide range of applications [11].

5.3.2.1. Thread Forms, Terminology, and Standards

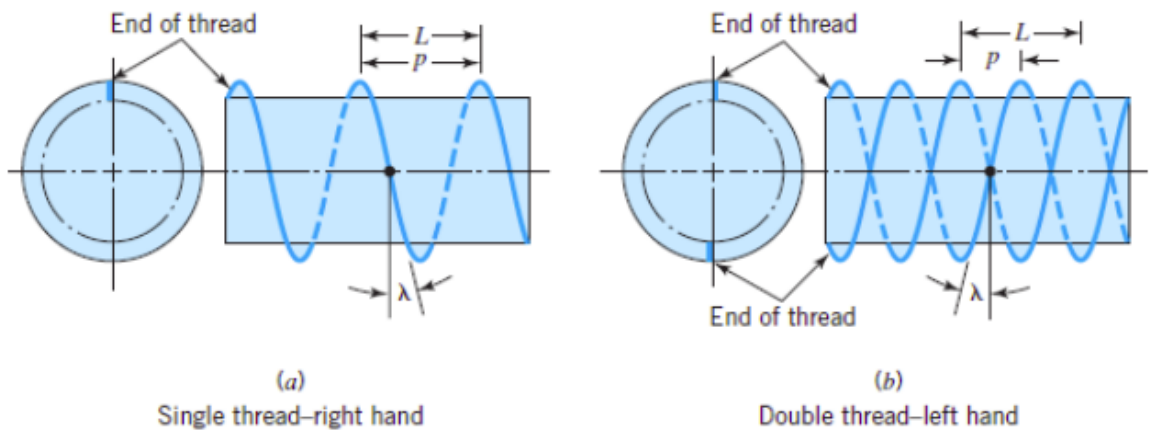


Figure 5. 15 Helical threads of pitch p , lead L , and lead angle λ [10,p.412]

The diagram in Figure (5.15) illustrates how a helical thread is wrapped around a cylinder, which is commonly used for screw-type fasteners, power screws, and worms. The main terminology used for power screws is as follows:

- Pitch (p), which is the distance between each corresponding point on adjacent threads measured parallel to the screw axis;
- Lead (L), which is the distance that the nut advances parallel to the screw axis in one revolution of the screw; The lead of a single threaded screw is equal to its pitch, whereas a double threaded screw has a lead that is twice the pitch, and this pattern continues for screws with more threads.
- Nominal diameter (d), which is the largest diameter of the screw and also known as the major diameter.
- Core diameter (d_r): which is the smallest diameter of the screw thread and also called the minor diameter;
- Lead angle (λ): which is the angle formed between the helix of the thread and a plane perpendicular to the screw axis. The helix angle is dependent on the lead and mean diameter of the screw.

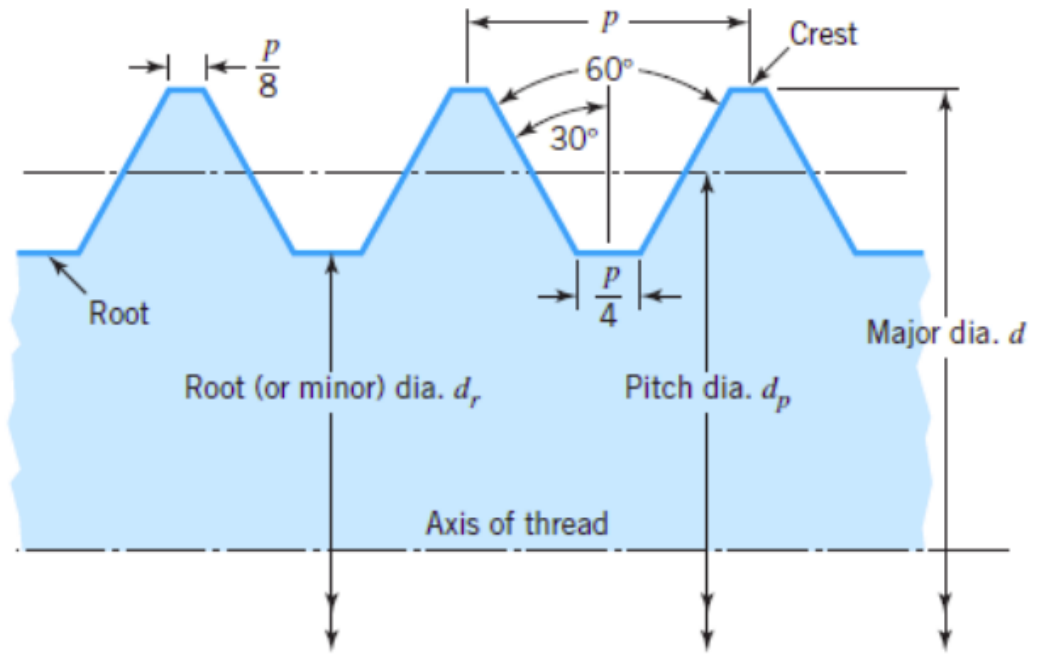


Figure 5. 16 The basic profile of the external thread[10,p.412]

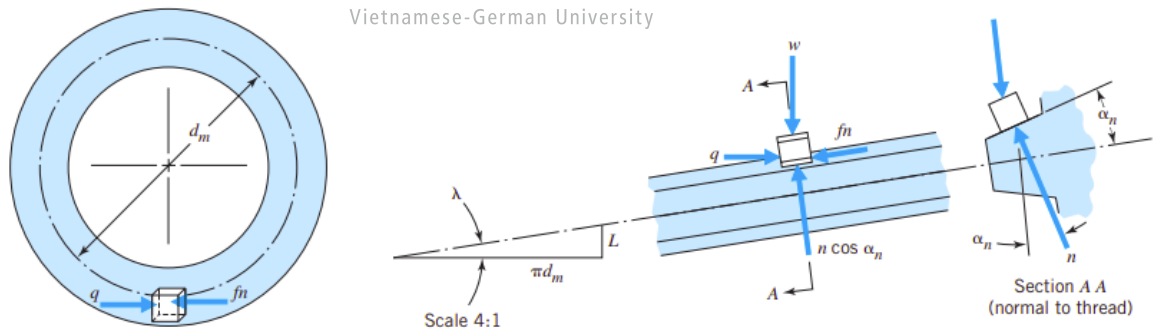


Figure 5. 17 Screw thread forces[10,p.417]

The figure 5.16 depicts the typical design of screw threads used in fasteners, which is generally consistent for both the Unified (inch series) and ISO (International Standards Organization, metric) thread systems. Tables 5.2 provides the standard sizes for the ISO system. The term "pitch diameter" (d_p) refers to the diameter of a cylinder within a flawless thread, where the widths of the thread and groove are equal. The indicated stress area is determined by calculating the mean value of the pitch diameter and the root diameter.

Nominal Diameter d (mm)	Coarse Threads			Fine Threads		
	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm ²)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm ²)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Table 5. 2 Fundamental Measurements of ISO Metric Screw Threads [10,p.414]

Figure 5.18 displays the standard thread forms utilized for power screws. Although Acme threads are the oldest type, they are still widely used. The Acme stub is sometimes preferred due to the ease of heat-treating. The square thread is more efficient than Acme threads but is not commonly used as it is challenging to manufacture the 0° thread angle and cannot be used with a split nut. The modified square thread with a 5° thread angle partially addresses these issues. The buttress thread is occasionally used to resist large axial forces in one direction by carrying the load on the face with the 7° thread angle.

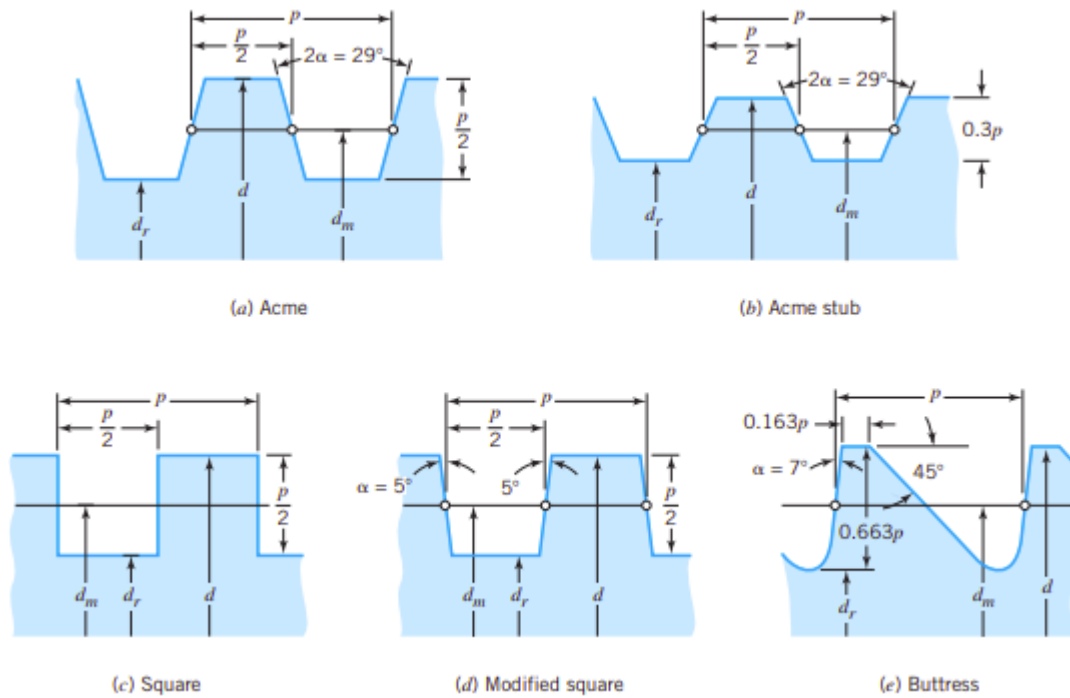


Figure 5.18 Power screw thread forms [10,p.415]

5.3.2.2. The mechanics of power screws



Relation of lead angle and lead: The lead angle and lead are related through the tangent function, as shown in figure 5.17. The tangent of the lead angle is equal to the lead divided by the circumference of the mean diameter of the thread contact. That is,

$$\tan \lambda = \frac{L}{\pi \cdot d_m} \quad (29)$$

Figure 5.17 displays a small block that represents a section of the nut and is subjected to the load w (A fraction of the complete axial load W), normal force n , friction force ($f \cdot n$), and tangential force q . It is noteworthy that the multiplication of force q and $\frac{d_m}{2}$ represents the torque applied to the nut segment. The sum of the tangential forces (horizontal forces) acting on the block gives:

$$\sum F_t = 0: \quad q - n(f \cos \lambda + \cos \alpha_n \cdot \sin \lambda) = 0 \quad (30)$$

Summing all axial forces (vertical forces) gives:

$$\sum F_a = 0: \quad w + n(f \sin \lambda - \cos \alpha_n \cdot \cos \lambda) = 0 \quad (31)$$

The normal force can be determined from equation (31) as:

$$n = \frac{w}{\cos \alpha_n \cos \lambda - f \sin \lambda} \quad (32)$$

Combining Equations. (30), (31) and (32) we have

$$q = w \frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda} \quad (33)$$



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The amount of rotational force, or torque, that corresponds to the force, q , is calculated as q multiplied by half of the mean diameter of thread contact, d_m . When the small block is taken as a representative element of a nut thread, the equations for the entire contact surface of the thread remain unchanged, except for the substitution of variables. In this scenario, the variables q , w , and n are replaced by Q , W , and N , respectively. These new variables represent the overall tangential, vertical, and normal loads applied to the thread. Consequently, the equation for calculating the torque required to lift the load W is as follows:

$$T = Q \frac{d_m}{2} = \frac{W \cdot d_m}{2} \cdot \frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda} \quad (34)$$

In power-screw applications, it's typical to need an additional torque component. If the screw is being loaded in an axial direction, a thrust or collar bearing is necessary between the rotating and stationary parts to handle the axial load. The thrust collar depicted in figure 5.19 is an example where the load is believed to be focused at the average collar diameter, d_c .

Assuming that f_c represents the friction coefficient of the collar, the necessary torque is $\frac{Wf_c d_c}{2}$.

For the general scenario of lifting a load W , the complete torque needed is:

$$T = \frac{W d_m f \pi d_m + L}{2 \pi d_m - f L} + \frac{W f_c d_c}{2} \quad (35)$$

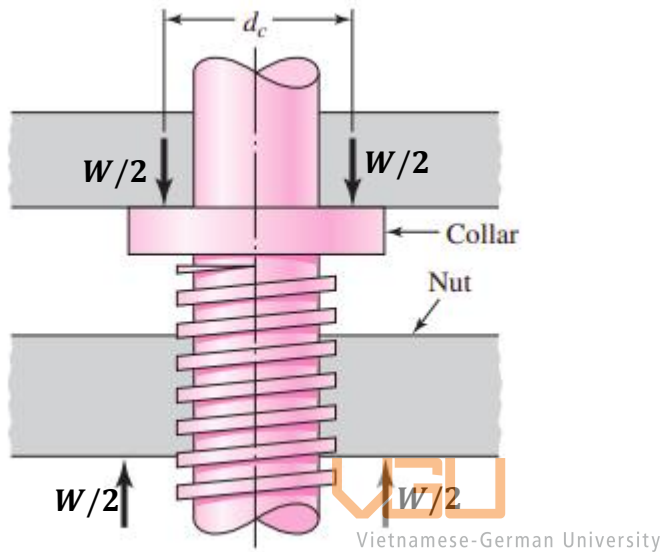


Figure 5. 19 thrust collar has frictional diameter d_c [11, p. 147]

The analysis that came before focused on lifting a weight or rotating a member in a way that worked against the weight. However, the study for bringing down a weight (or rotating a structural element "under the influence of the load") is identical, except that the directions of q and $(f \times n)$ (as shown in Figure 5.17) are flipped. Generally, the cumulative torque necessary for lowering the weight W is:

$$T = \frac{W d_m f \pi d_m - L}{2 \pi d_m + f L} + \frac{W f_c d_c}{2} \quad (36)$$

A self-locking screw requires an intentional application of torque necessary for bringing the load down, whereas an overhauling screw has minimal friction, allowing the load to descend on its own. In simpler terms, to prevent the load from descending, a negative external torque

must be exerted. If the friction of the collar is negligible, equation (36) indicates that a screw is self-locking when

$$f \geq \frac{L \cos \alpha_n}{\pi d_m} \quad (37)$$

, provided that T is greater than or equal to zero.

5.3.2.3. Values of Thread Angle in the Perpendicular Perspective

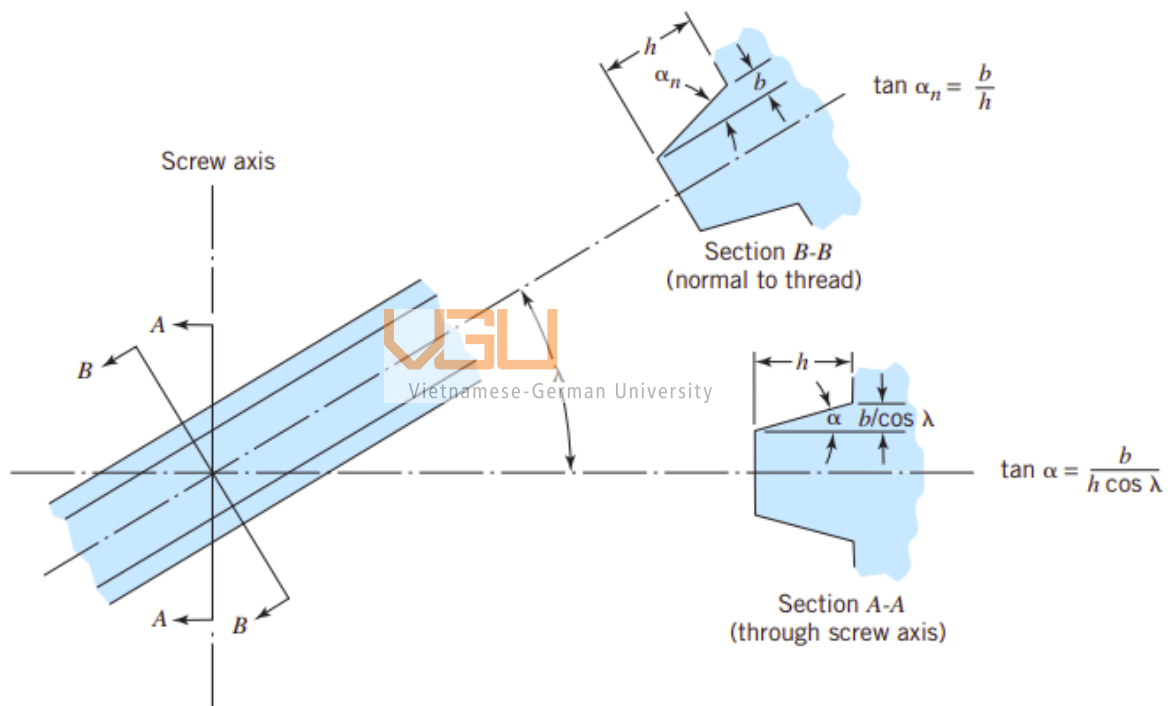


Figure 5.20 Contrasting Thread Angles Assessed in Axial and Perpendicular Orientations (α and α_n) [10, p.420]

Figure 5.20 provides a graphical representation of the inclination of the thread, quantified as the thread angle, which has been measured in the perpendicular plane (α_n , specified in the previous equations) and in the parallel plane (α , conventionally indicated and illustrated in figure 5.18). Consequent to the analysis of Figure 10.7, it can be inferred that

$$\tan \alpha_n = \tan \alpha \cos \lambda \quad (38)$$

In the case of helix angles that are relatively small, it is commonly accepted to approximate the cosine of λ as a value of one.

5.3.2.4. Design of power screw

In this section, thread form, number of threads, pitch, nominal and minor diameter will be determined. Multi-start threads also offer faster feed rates than single-start threads. This is because the thread pitch of a multi-start thread is larger than that of a single-start thread with the same lead. The larger thread pitch means that the nut will travel a greater distance for each revolution of the screw, resulting in a faster feed rate. However, one disadvantage of multi-start threads is their reduced self-locking ability. Self-locking is a critical requirement of this application because it ensures that the load, in this case, they are the four beams, remains in the desired position. So single-start thread is chosen.

Buttress threads have a sloping flank on one side and a perpendicular flank on the other. This asymmetrical shape provides higher load-bearing capacity and greater resistance to axial forces in one direction. However, buttress threads are more difficult to manufacture, and this can lead to higher costs. Acme thread form is selected due to its simplicity, ease of manufacturing, and relatively low cost. Acme threads have a symmetric shape, which means they can be easily cut on a lathe or with a thread rolling machine. Additionally, standard thread forms can provide sufficient axial force when properly designed, manufactured, and assembled.

The shaft section on which the nut is held is determined to be 14 mm in diameter. With shaft diameter of 14 mm, from table 5.2, pitch and minor diameter are determined. Coarse threads are selected because a greater pitch distance, allowing them to be inserted and tightened more quickly than fine threads. Additionally, Coarse threads are also more resistant to damage from debris or corrosion. The pitch, p , and minor diameter, d_r , are then decided to be 2 mm and 11.6 mm respectively.

Self-locking ability of the thread is needed to be check to make sure the power screw can maintain its position. From figure 5.16, $d_m = d - \frac{p}{2} = 13 \text{ mm}$. From equation (29) and because of single-start thread, $L=p=2 \text{ mm}$, λ is calculated as follow:

$$\lambda = \tan^{-1}\left(\frac{L}{\pi d_m}\right) = \tan^{-1}\left(\frac{2}{\pi \cdot 13}\right) = 2.8^\circ \quad (39)$$

Equation 38 is used to find α_n :

$$\alpha_n = \tan^{-1}(\tan \alpha \cos \lambda) = \tan^{-1}(\tan 14.5^\circ \cos 2.8^\circ) = 14.48^\circ \quad (40)$$

Use equation (37) to check for self-locking ability:

$$f \geq \frac{L \cos \alpha_n}{\pi d_m} \quad (37)$$

$$\text{Right hand side} = \frac{2 \cos 14.48^\circ}{\pi \cdot 15} = 0.04 \quad (41)$$



The coefficient of friction between steel and aluminum, can vary depending on a number of factors, such as the surface finish, lubrication, temperature, and pressure. A specific value for the friction coefficient between C45 steel and aluminum 6061 is not found. However, static friction coefficient between steel and aluminum is 0.61. This value can be used as a reference for C45 steel and aluminum 6061 as C45 steel is a type of steel and Aluminum 6061 is a type of aluminum alloy. The value 0.61 indicates that the thread exhibits self-locking capability, as it exceeds the required value by a factor of 15.

5.4. Hubs-shaft connection

In this prototype, there are three hubs, two of which are rigidly attached to the central shaft to support the four outer bars and transmit forces from the web to the central shaft. There exist multiple techniques to affix features to a shaft, contingent upon the particular specifications of the task. Below are a few prevalent approaches.

- Keyway and key: A keyway is a slot machined into the shaft, and a key is a piece of metal that fits into the slot. This method provides a positive, non-slip connection between the shaft and the attached feature.
- Spline: A spline is a series of ridges or teeth machined onto the shaft and the attached feature. The teeth interlock, providing a positive, non-slip connection that can handle high torque loads.
- Threaded connection: The shaft and features can be threaded, with a nut or other fastener used to secure the features in place.
- Press fit: The feature can be pressed onto the shaft, using a hydraulic press or other tool to apply force.
- Adhesive bonding: An adhesive can be used to bond the feature to the shaft. This method can provide a strong, permanent connection, but may require special preparation of the surfaces and curing time.
- Welding: the feature can be welded directly to the shaft. This method requires careful preparation of the surfaces and may not be suitable for all materials or applications.

Key and keyway or splines method is not suitable for this application because there is not enough space available on the hub to accommodate for key and keyway or splines. Furthermore, these hubs are subjected to axial loads, the key and keyway or splines method may not provide sufficient axial retention. Thread connections may not be suitable for attaching certain features to a shaft because they are not able to transmit torque effectively. Also, the stress concentration at the base of the threads can cause premature failure due to fatigue. Therefore, this method may not be feasible. If press fit method is used in this design configuration, the portion of the shaft to hold one hub must be smaller in diameter to let another hub to slide through, which reduces the strength of the shaft and causes stress concentration. Besides, press fit connections can be difficult to disassemble, which can make maintenance or repair more challenging. Consequently, this type of connection is not a good option. About welding connection, if a welded joint fails, it can be difficult to repair or replace. Welding requires specialized equipment and expertise, which can make repairs time-consuming and expensive. Thus, this type of connection is deemed inappropriate for the intended purpose.

Adhesive bonding is often the preferred method for attaching features to a shaft due to its numerous advantages over other methods. Firstly, it provides a strong and durable bond by forming a continuous layer between the feature and shaft, resulting in a large contact area. Secondly, it distributes load and stress evenly over the entire bonded area, reducing the likelihood of stress concentrations that can occur with mechanical fasteners. Thirdly, it can replace heavier attachment methods, such as welding or mechanical fasteners, leading to weight savings. Fourthly, it can improve aesthetics by creating a cleaner appearance. Lastly, it's versatile and can be used with a wide range of materials, including dissimilar materials, which are difficult to join with other methods. In conclusion, adhesive bonding is a reliable and versatile attachment method that offers a high strength bond, uniform stress distribution, weight reduction, improved aesthetics, and versatility.

LOCTITE 638[18] has been specifically formulated for the purpose of bonding cylindrical fitting parts, especially in cases where the bond gaps may be as small as 0.25 mm and where maximum strength is required at room temperature. This adhesive product solidifies when it is restricted in the situation where metal surfaces touch without air in between that fit closely together, thereby preventing loosening and leakage from shock and vibration. Typical applications that benefit from this adhesive include securing locking bushings and sleeves onto housings and shafts. LOCTITE 638 has excellent curing properties and can be used on both active metals (such as mild steel) and passive substrates (like stainless steel and plated surfaces). It is also resistant to high temperatures and oil, and can withstand minor surface contaminations from various types of oils, including cutting, lubrication, anti-corrosion, and protection fluids. According to the manufacturer's datasheet, the shear strength of Loctite 638 can vary depending on the curing time and environmental conditions. After 15 minutes of curing time at 22 °C, the shear strength of Loctite 638 on steel can reach up to 13.5 N/mm² (1960 psi). After 24 hours of curing time at room temperature, the maximum shear strength on steel increases to 24 N/mm² (3500 psi). Finally, after 7 days of curing time at 22 °C, the maximum shear strength on steel is reported to reach 29 N/mm² (4200 psi). Loctite 638 also exhibits a significant advantage due to its capacity for disassembly. The adhesive can be removed utilizing standard hand tools, which facilitates ease of maintenance and repair. Additionally, in cases where the bond proves resistant to hand tools, the application of localized heat to the assembly, up to approximately 250 °C, softens the adhesive bond,

allowing for easy disassembly. This feature proves particularly beneficial in applications where access to the components is limited or disassembly is required for repair or maintenance.

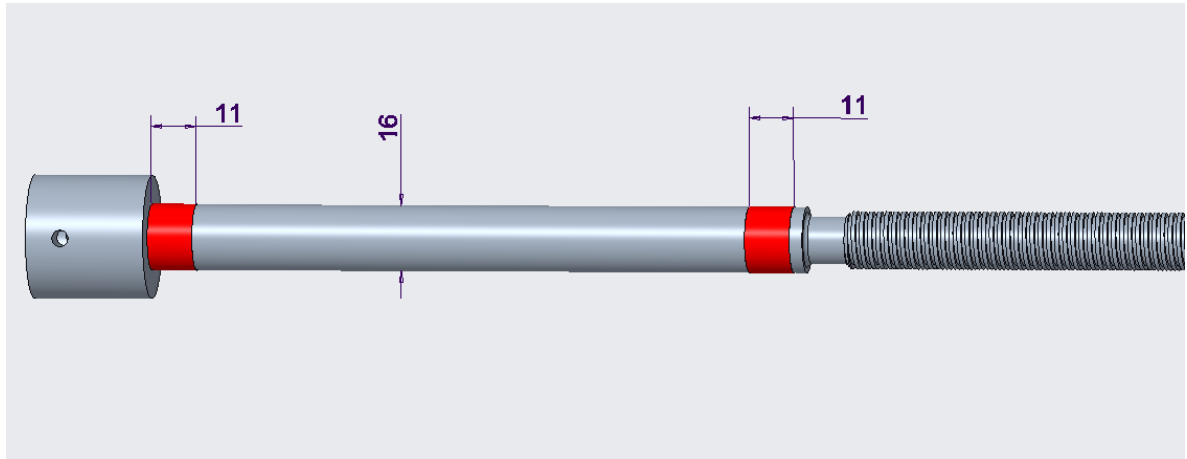


Figure 5. 21 Sections where the hubs are mounted

The necessary shear stress on the surface that makes contact with the hub is evaluated and compared against the shear stress that can be provided by Loctite 638. The region shaded in red indicates the location for the installation of two hubs in figure 5.20. In order to compute the shear stress of an adhesive bond within a shaft-hub connection, knowledge of the applied force and the bond area is requisite. In this model in which the bond takes on the form of two red cylindrical layers linking the hub to the shaft, the bond area can be determined by way of the following formula:

$$A = 2 \cdot \pi \cdot D \cdot L = 2 \cdot \pi \cdot 16 \cdot 11 = 1105.8 \text{ mm}^2 \quad (42)$$

where A denotes the bond area, D refers to the bond diameter, and L represents the bond length.

To evaluate the shear stress, the following equation would be utilized:

$$\tau = \frac{F}{A} = \frac{125N}{1105.8\text{mm}^2} = 0.11N/\text{mm}^2 \quad (43)$$

where τ signifies the shear stress, and F represents the applied force acting on the bond. The force acting on the bond is determined through the equation:

$$F = \frac{T}{r} = \frac{1000 \text{ Nmm}}{8 \text{ mm}} = 125 \text{ N} \quad (44)$$

where F is the force applied to the bond, T denotes the torque being transmitted through the connection, and r represents the radius of the bond.

Based on the manufacturer's datasheet and the given computed shear stress requirement of $\tau = 0.11 \text{ N/mm}^2$, it can be concluded that Loctite 638 is more than capable of sustaining the expected stresses within the given shaft-hub connection. The datasheet indicates that the shear strength of Loctite 638 on steel can reach up to 13.5 N/mm^2 (1960 psi) after just 15 minutes of curing at 22°C , which is well above the computed shear stress requirement of 0.11 N/mm^2 .

6. Detail design and drawings and cost estimation



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6.1. Introduction

Within Chapter 5 of the current study, the computation and selection of crucial dimensions crucial for the system's durability in bearing the necessary loads are outlined. The intention behind this chapter is to expound upon the remaining dimensions that will enable cost-effective manufacturing, replacement, and maintenance, all while concurrently reducing the overall manufacturing time. A comprehensive exposition is provided within this chapter, encompassing material quantities and intricate drawings specific to each individual component. Furthermore, visual representations in the form of photographs are included to enhance clarity and serve as visual references. Furthermore, this chapter encompasses cost estimation, offering valuable insights into the financial considerations associated with the implementation of the proposed system.

6.2. Central shaft

Thread relief is added between the threaded portion and the 16 mm-diameter shaft portion. Thread relief, which is also referred to as undercutting, is a machining technique used to create clearance or space around a threaded hole or shaft. The purpose of thread relief is to allow threading tools, such as taps and dies, to reach the full depth of the threads without encountering any obstruction, thereby ensuring more accurate and consistent threads.

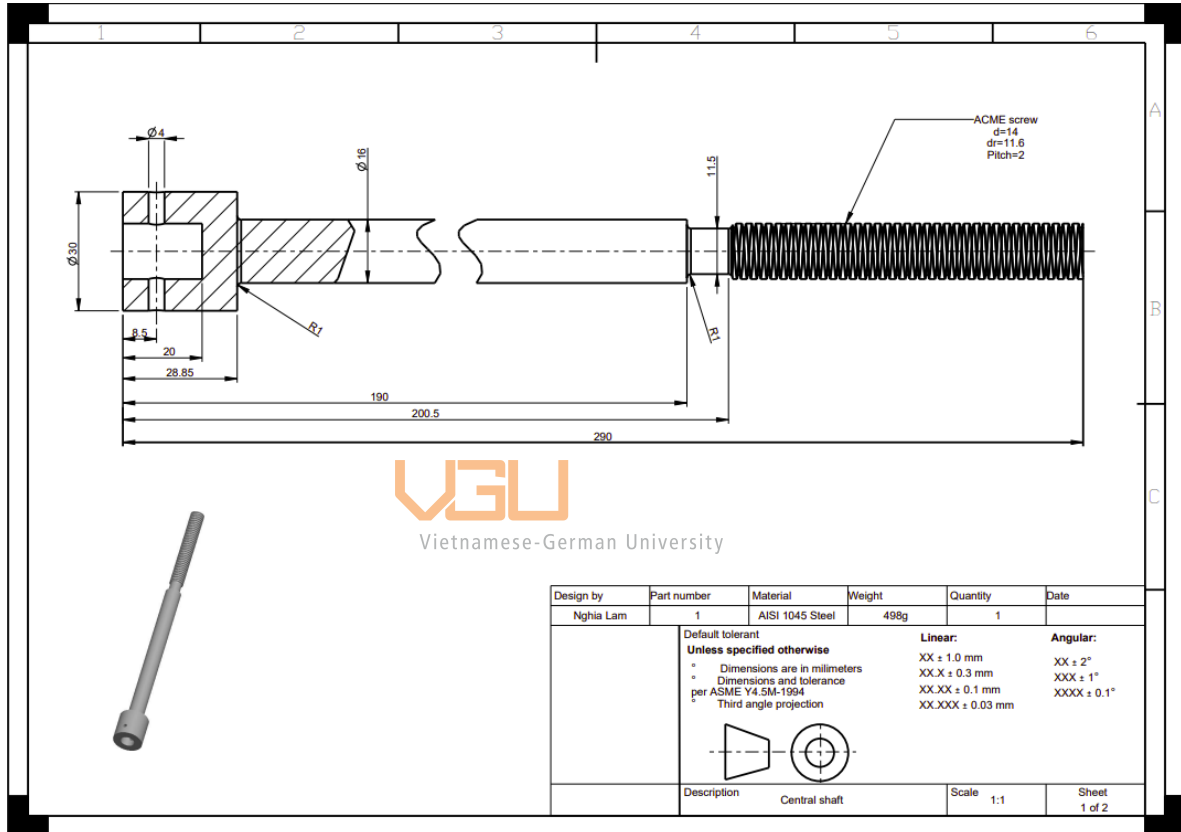


Figure 6. 1 Drawing of central shaft



Figure 6. 2 Central shaft

6.3. Hub 1



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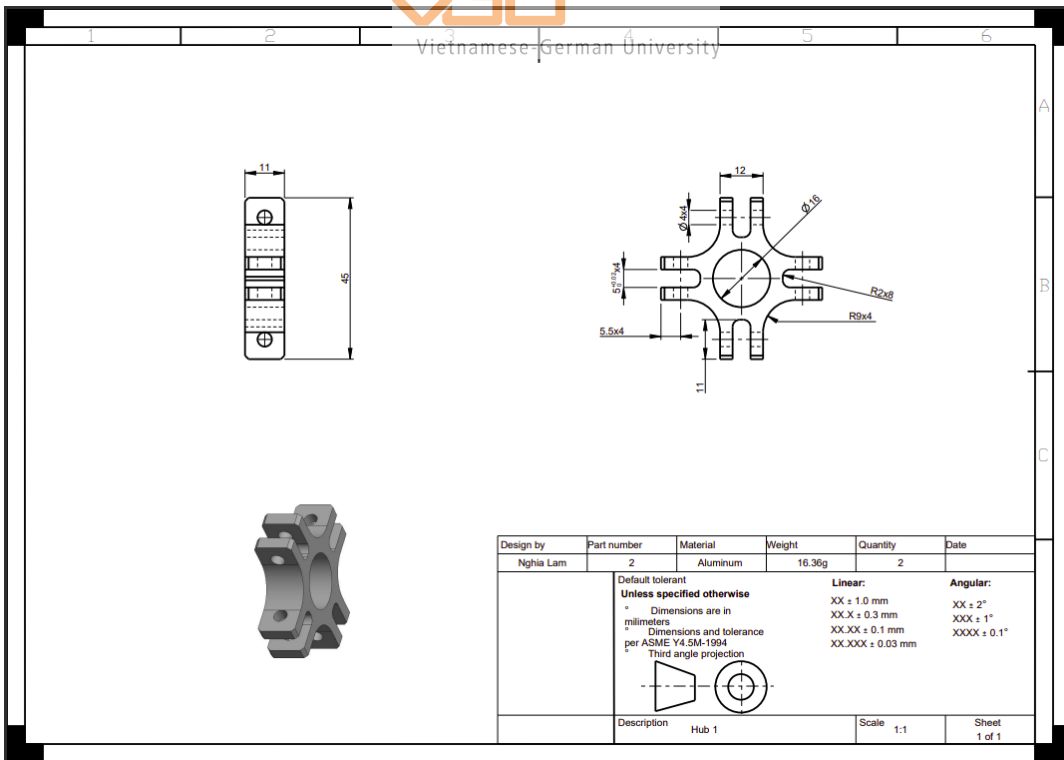


Figure 6. 3 Drawing of hub 1

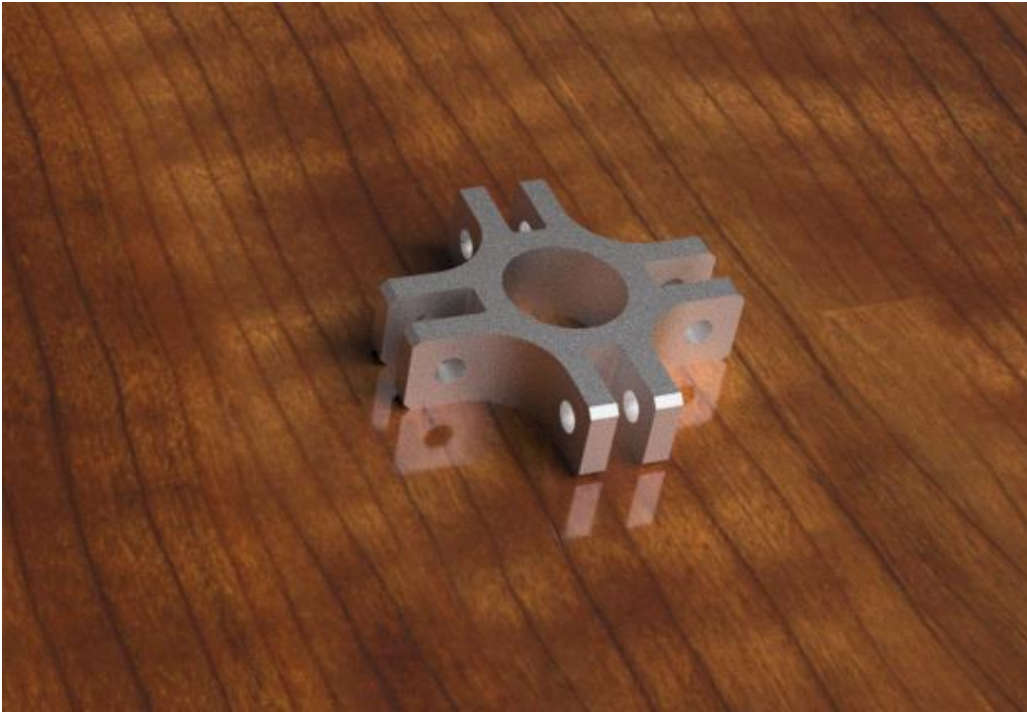


Figure 6. 4 Hub 1

6.4. Hub 2

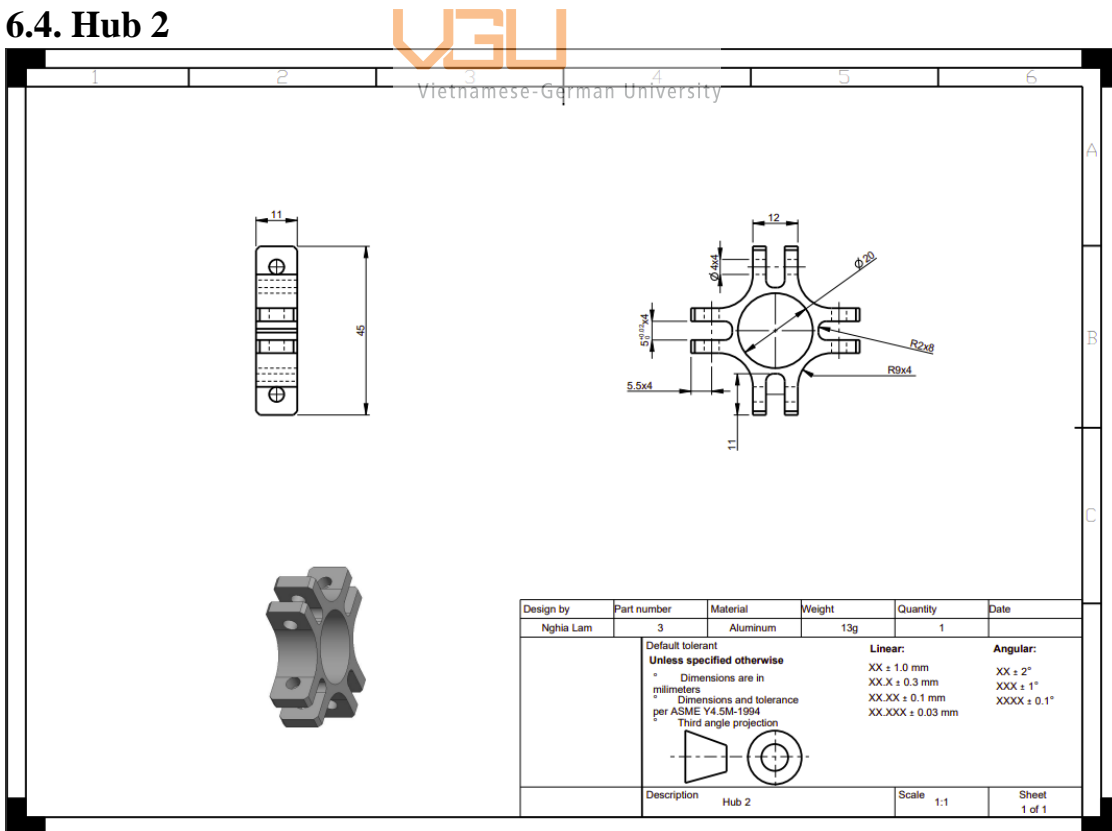


Figure 6. 5 Drawing of hub 2



Figure 6. 6 Hub 2

6.5. Bars

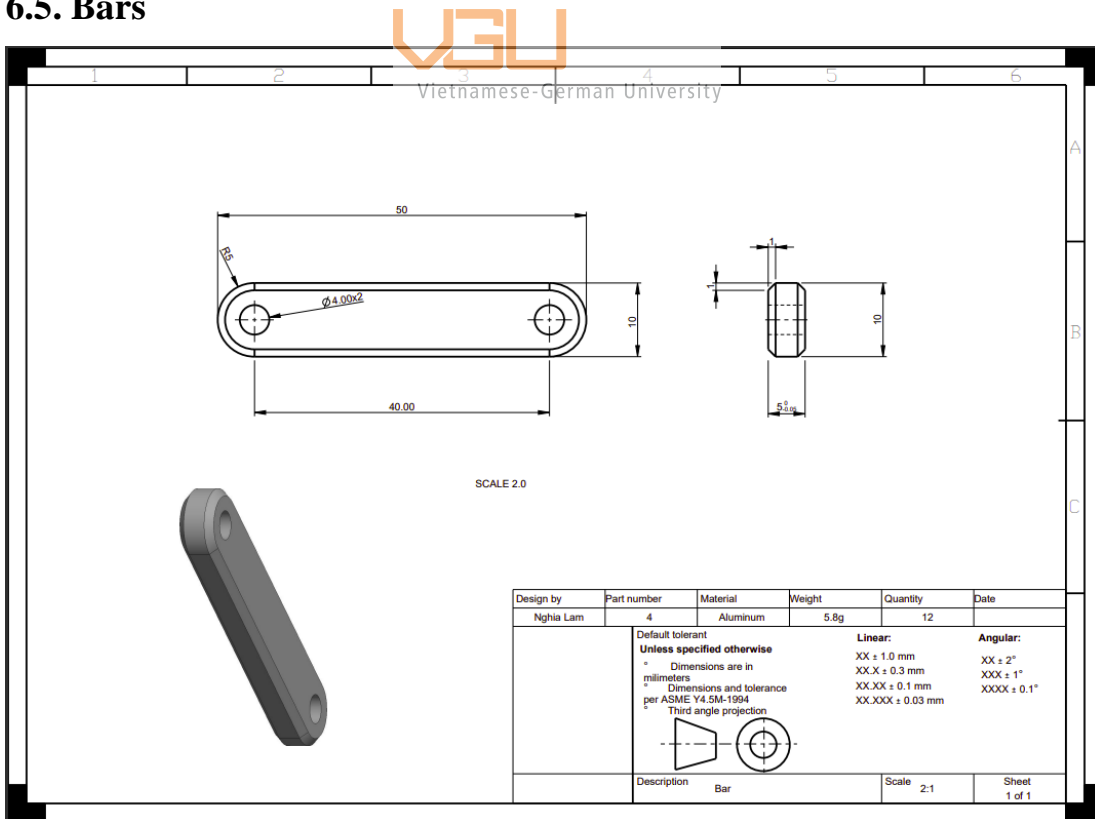


Figure 6. 7 Drawing of a bar



Figure 6. 8 Bar

6.6. Nut

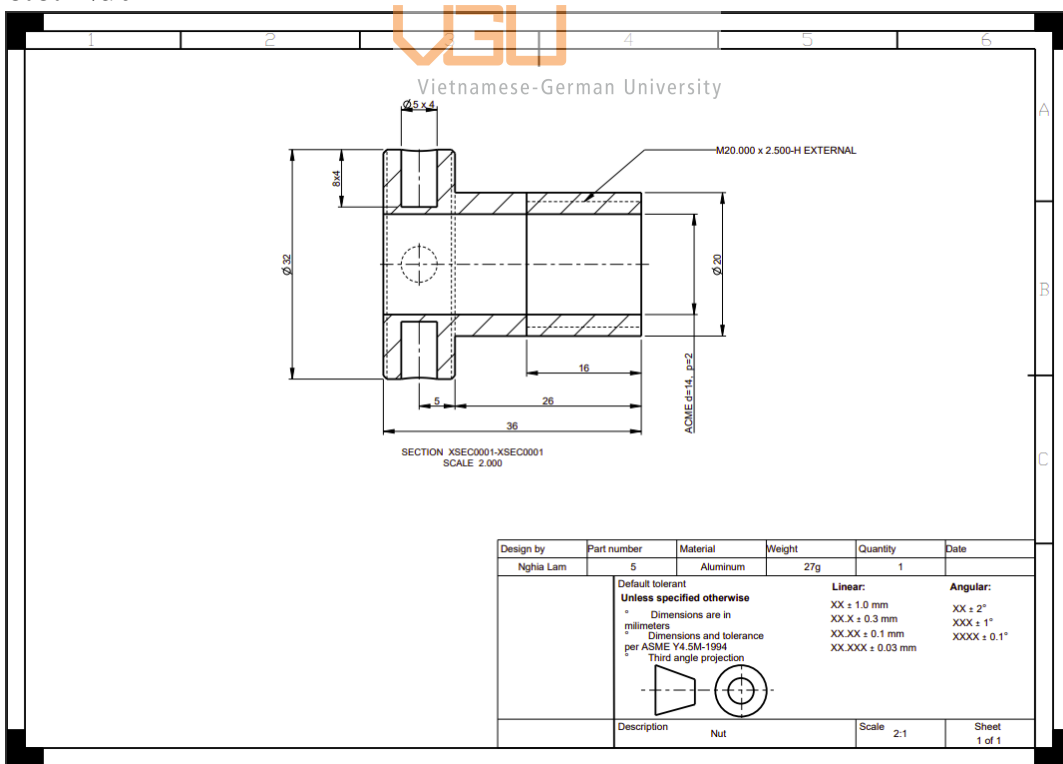


Figure 6. 9 Drawing of nut



Figure 6. 10 Nut

6.7. Beams

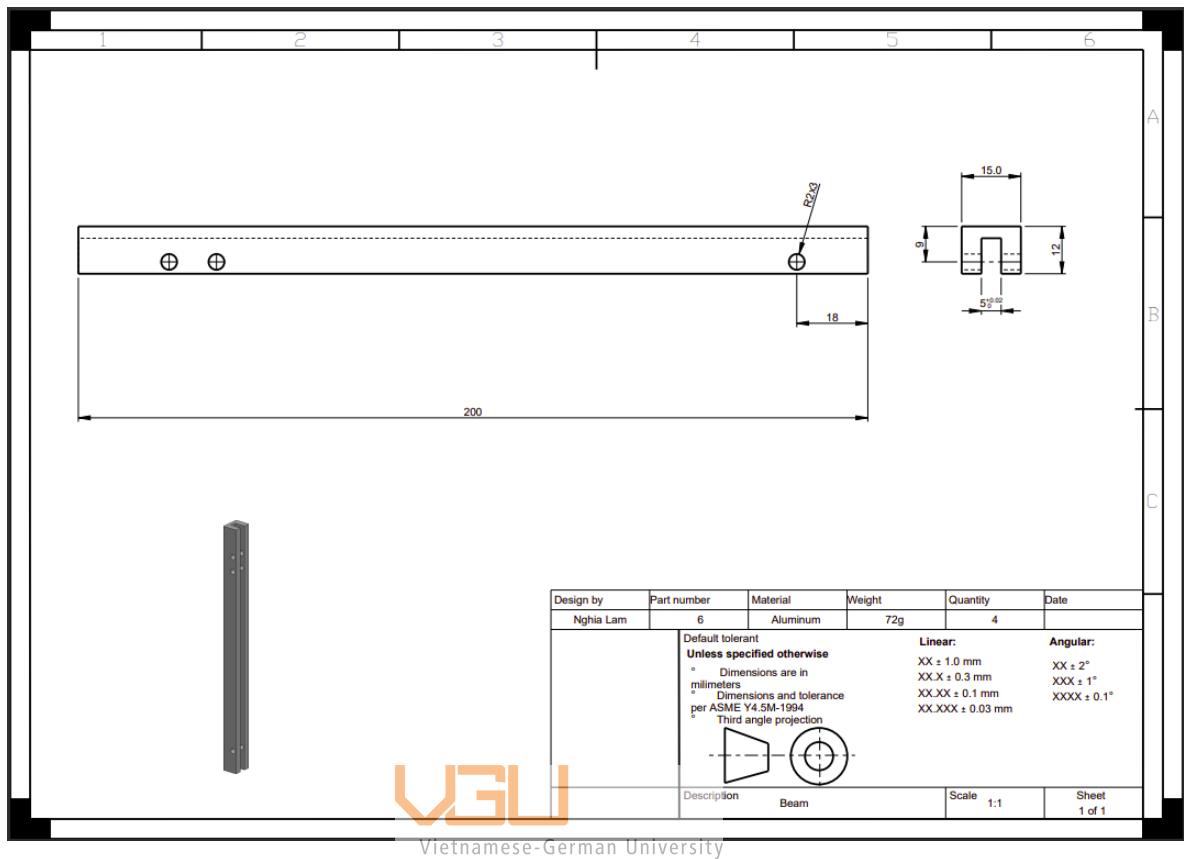


Figure 6. 11 Drawing of a beam

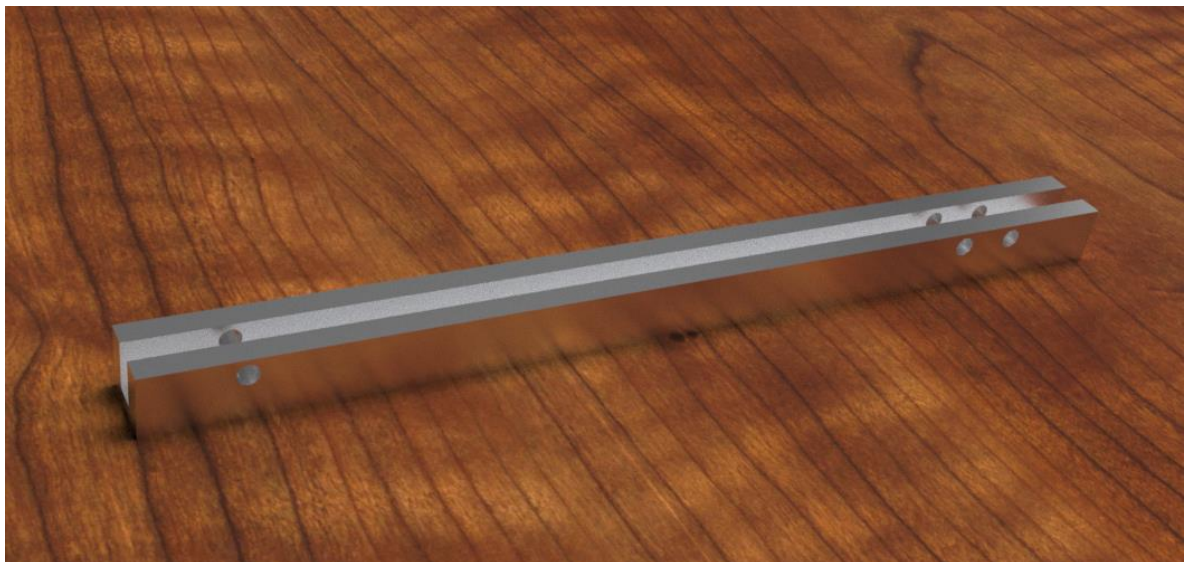


Figure 6. 12 Beam

6.8. Locknut

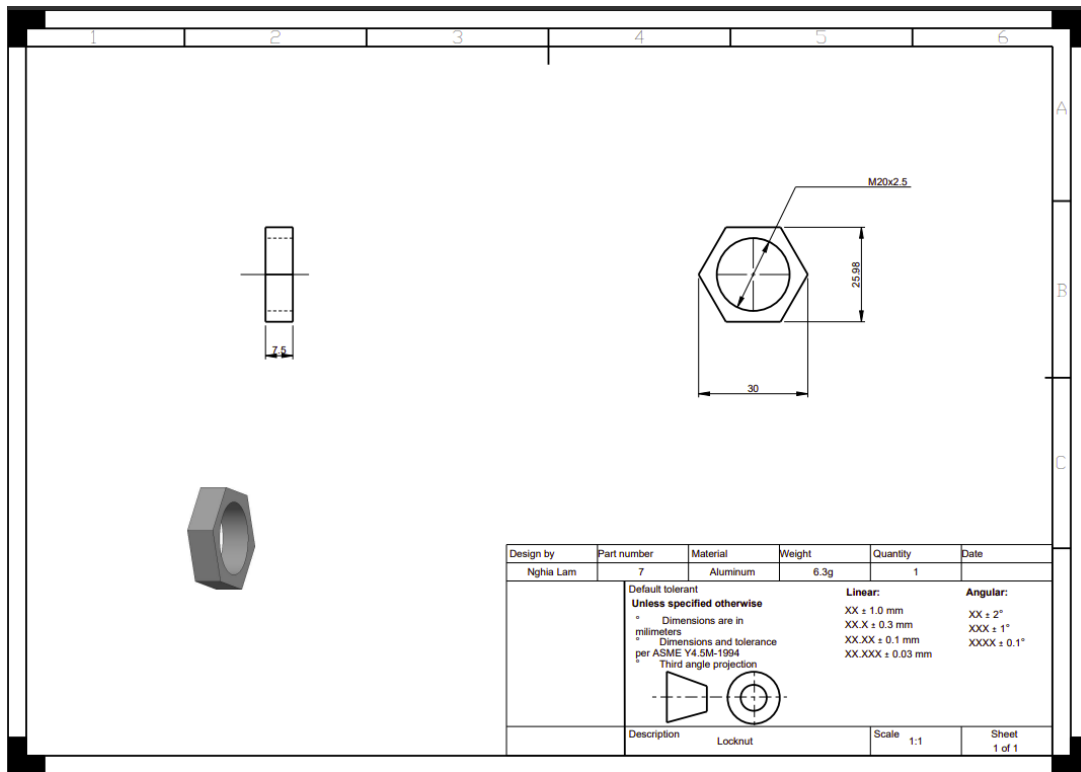


Figure 6. 13 Drawing of locknut



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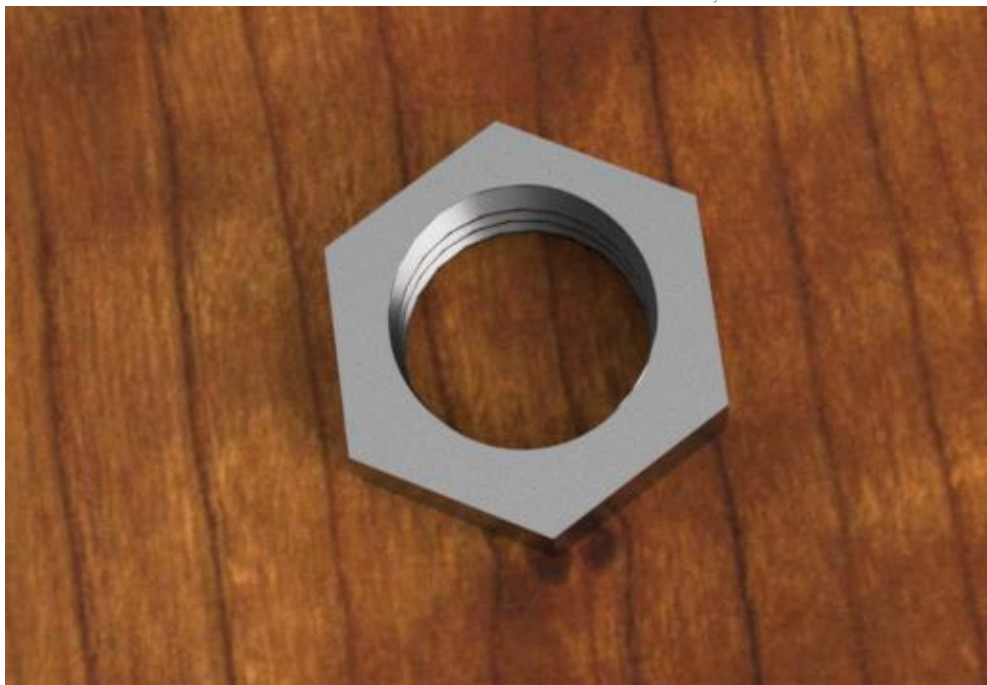


Figure 6. 14 Locknut

6.9. Cost estimation

No.	Machine element	Quantity	Material	Cost per each (VND)	Total cost (VND)
1	Central shaft	1	AISI 1045 Steel	200,000	200,000
2	Hub 1	2	6061 Aluminum	180,000	360,000
3	Hub 2	1	6061 Aluminum	180,000	180,000
4	Bar	12	6061 Aluminum	60,000	720,000
5	Nut	1	6061 Aluminum	150,000	150,000
6	Beam	4	6061 Aluminum	120,000	480,000
7	Locknut	1	6061 Aluminum	20,000	20,000
8	Misc. (Bolts, Nuts, Pin)		Standard	50,000	50,000
	Total				2,160,000

Table 6. 1 List of mechanical components and cost

7. Assembly



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This section provides instructions on how to assemble individual components and outlines the steps involved in putting together a prototype of the decoiler shaft.

First, the two hubs are fixed onto the central shaft using Loctite 638 adhesive in such a way that the distance between the two hubs is 147 mm, and one face of the left hub is coincident with the shoulder of the central shaft.

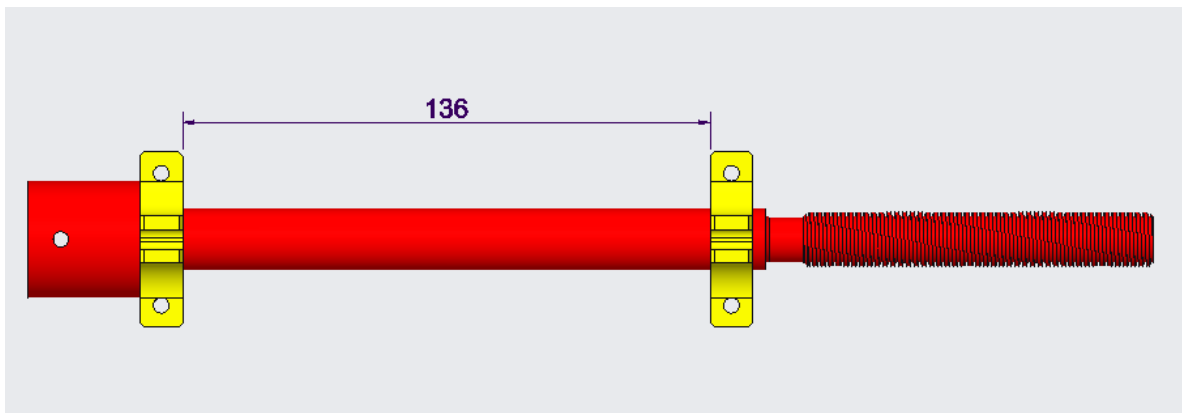


Figure 7. 1 Two hub 1 are in stalled

Eight bars is then installed on two hubs by standard m4 hex head bolts. To reduce clutter, the picture below shows only 1 bar installed in the hub.

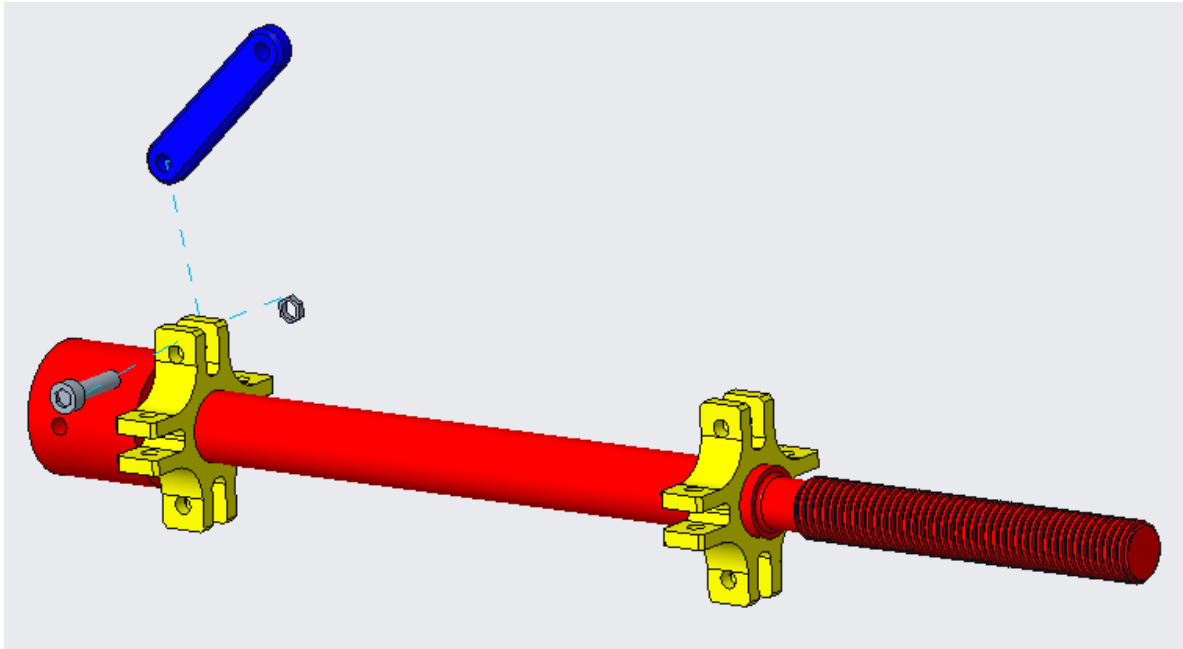


Figure 7. 2 Installation of one bar

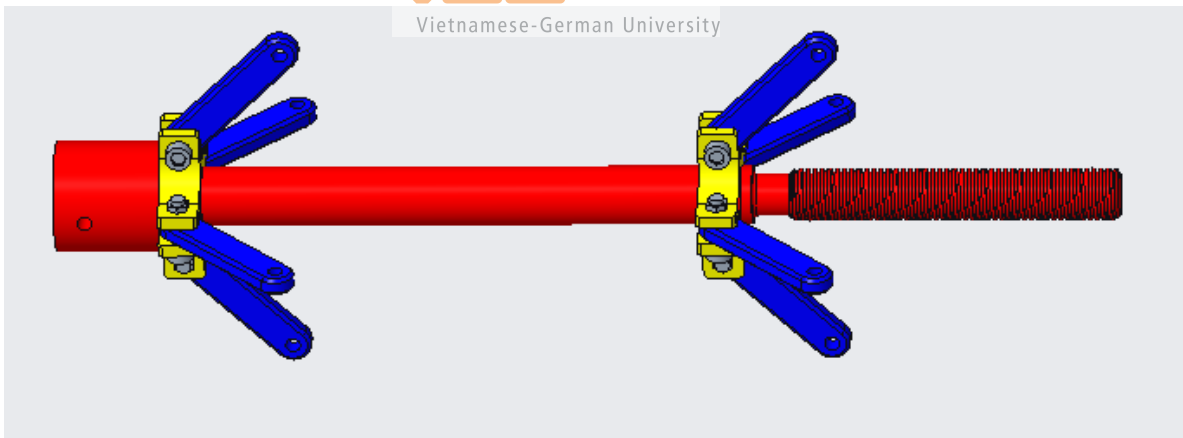


Figure 7. 3 Eight bars are installed

Four beams are installed in the prototype, each beam is mounted on 2 bars as shown

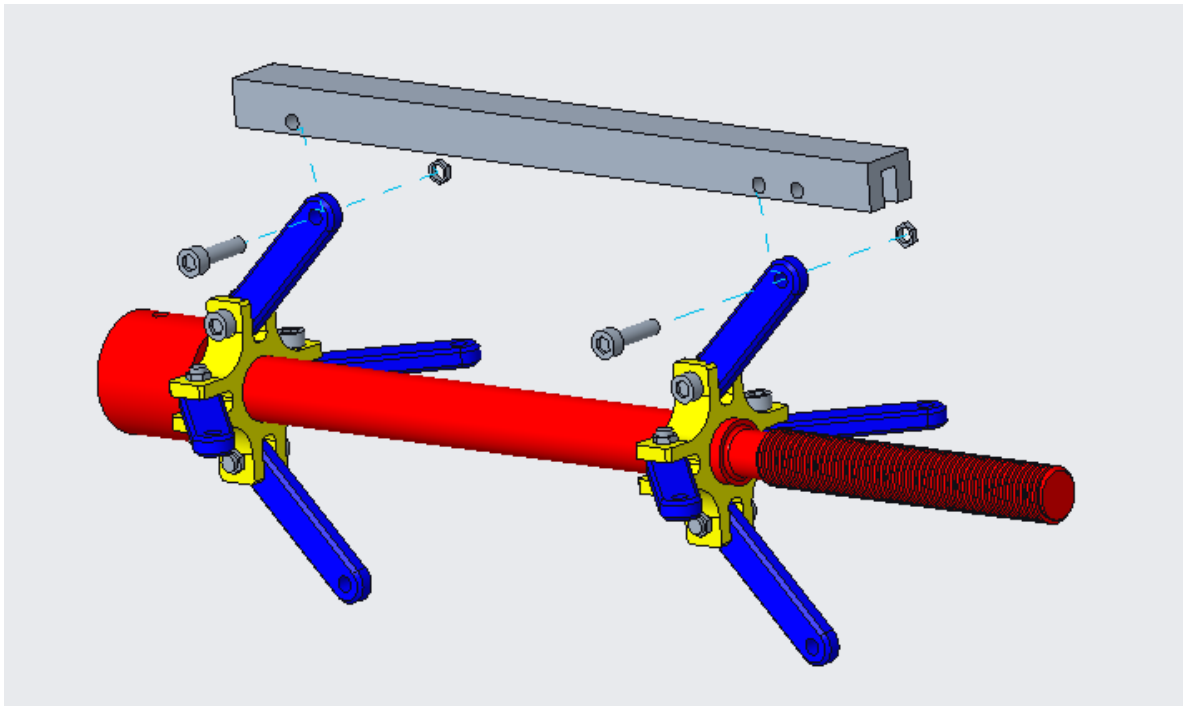


Figure 7. 4 Installation of one beam

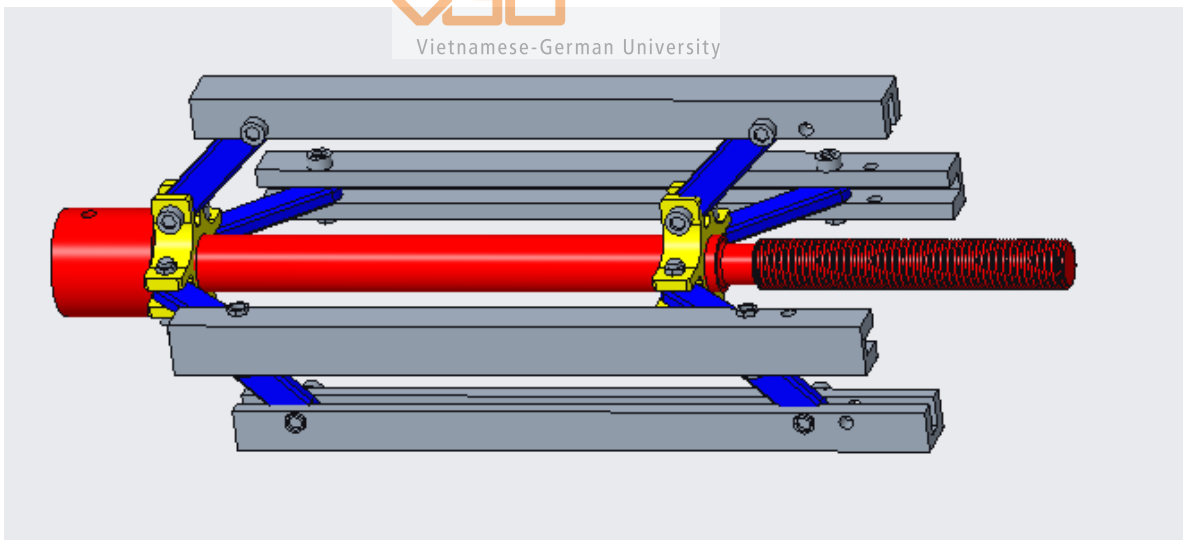


Figure 7. 5 Four beam are installed

The hub 2 is affixed to the nut via a locknut, following the prescribed sequence in figure 7.7. Caution must be exercised in order to avoid excessive tightening, which would impede rotation of hub 2 about the nut.

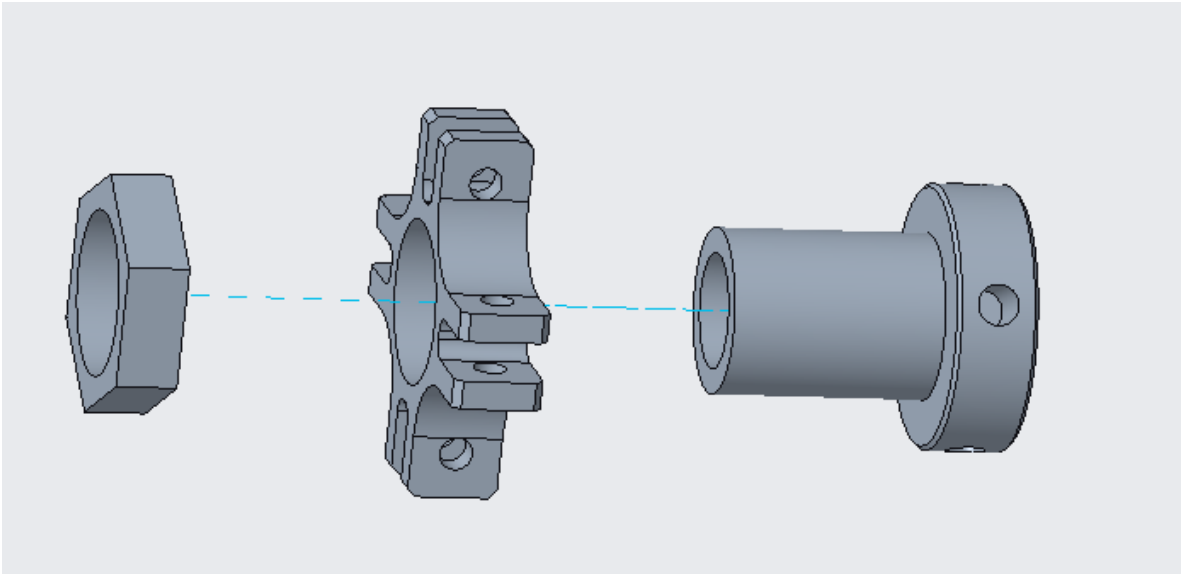


Figure 7. 6 locknut hub 2 and nut assembly

Subsequently, the four bars are introduced into the Nut Assembly, with the installation of a single bar being presented for the purpose of facile observation.

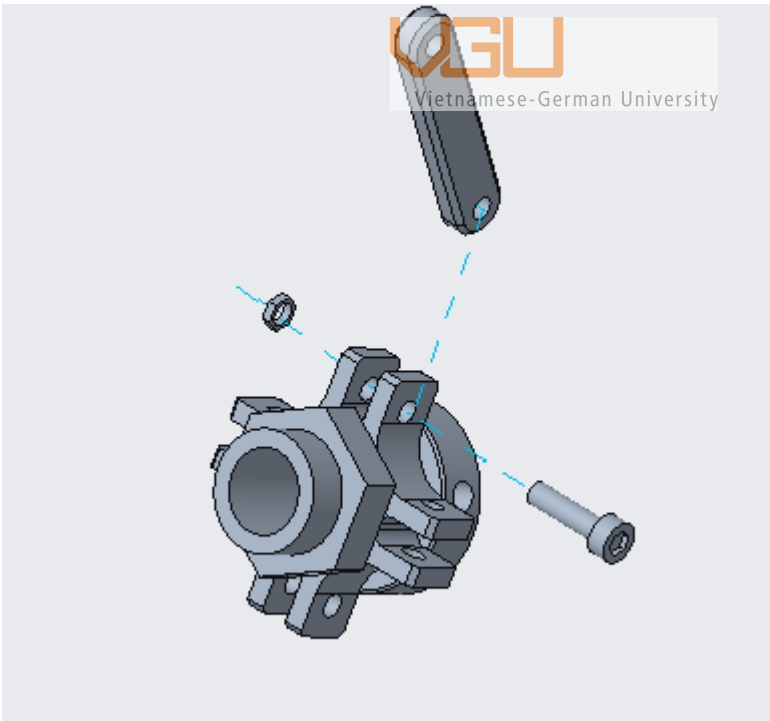


Figure 7. 7 One bar is installed to hub 2

The Nut Assembly is later fastened to the central shaft by means of the lead screw mechanism, while the bars are secured to the beam via hex head bolts.

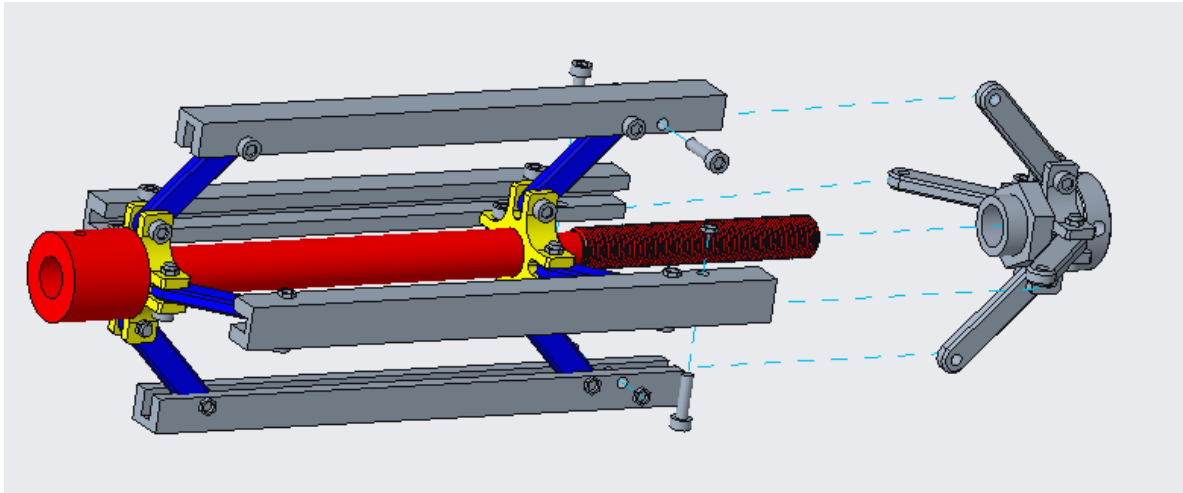


Figure 7. 8 Installation of nut assembly to central shaft

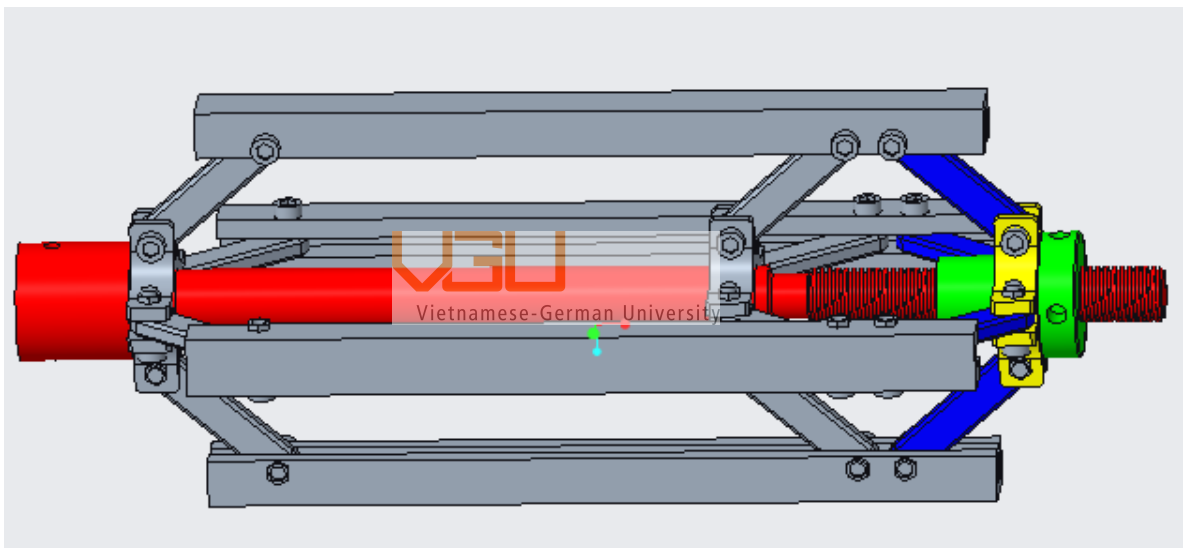


Figure 7. 9 Complete assembly

8. Finite Element Analysis

8.1. Why Finite Element Analysis is needed?

Although hand calculations are a good place to start with shaft design, finite element analysis (FEA) is usually needed to verify the hand calculations and obtain a more precise understanding of how the shaft behaves under different loading conditions. Hand calculations rely on simplified assumptions about the shaft's shape and loading, which may not account

for all the complexities of the real-world situation. FEA, on the other hand, uses computer-based models to replicate the actual shape and loading of the shaft, providing more detailed and accurate results. FEA can accommodate more complex shapes and loading conditions, such as uneven stress distributions, heat effects, and dynamic loading, which are difficult or impossible to take into account using hand calculations. Additionally, FEA can assist in optimizing the design of the shaft by evaluating various materials, sizes, and shapes. It can also pinpoint parts of the shaft that may be vulnerable to deformation or failure, allowing for design changes to avoid potential issues. So, while hand calculations are a good place to start, FEA is often necessary to ensure that the shaft design is dependable, efficient, and safe[13].

8.2. Finite Element Analysis and its application

Finite element analysis (FEA) involves using mathematical calculations, simulations, and models to forecast and comprehend the potential behavior of an object under different physical situations. Engineers use FEA to pinpoint any weaknesses in their prototype designs and improve them accordingly. To conduct FEA, engineers utilize a computational tool known as the finite element method (FEM), which divides the structure of an object into a variety of segments, or "elements," and then reassembles these elements at specific points called "nodes." By doing so, FEM generates a collection of mathematical equations that engineers leverage to carry finite element analysis and gain a more in-depth understanding of how the object will behave.

Often, the behavior of a product under various physical conditions, like how it handles pressure, heat, and structural load, is represented by means of partial differential equations (PDEs). Finite element analysis (FEA) has been recognized as a computerized method to solve both linear and nonlinear partial differential equations (PDEs). Nonetheless, it's worth noting that FEA exclusively offers an estimation of the outcome, and it relies on numerical methods to calculate the actual results of partial differential equations.

FEA has numerous applications across various fields, including engineering, physics, and materials science. It is used to analyze the structural integrity of buildings, bridges, aircraft, and other mechanical components under different loading conditions, simulate heat transfer and temperature distribution in complex systems such as engines, power plants, and electronic devices, model fluid flow patterns in systems such as pipelines, chemical reactors,

and combustion engines, and simulate the behavior of electromagnetic fields in devices such as motors, transformers, and antennas. By providing insights into the behavior of complex systems, FEA enables engineers and scientists to make informed decisions that can improve the efficiency, reliability, and safety of products and processes[14].

8.3. Simulation setup

Model of a 200mm-diameter and 150mm-length paper roll is made. One the paper roll surface, a groove is made for the forces which mimics the paper tension to be placed on. The decoiler shaft is fixed at one end. A force of 100N is inserted on the paper roll and the gravittional accelerration for the system is set to be $9.8m/s^2$.

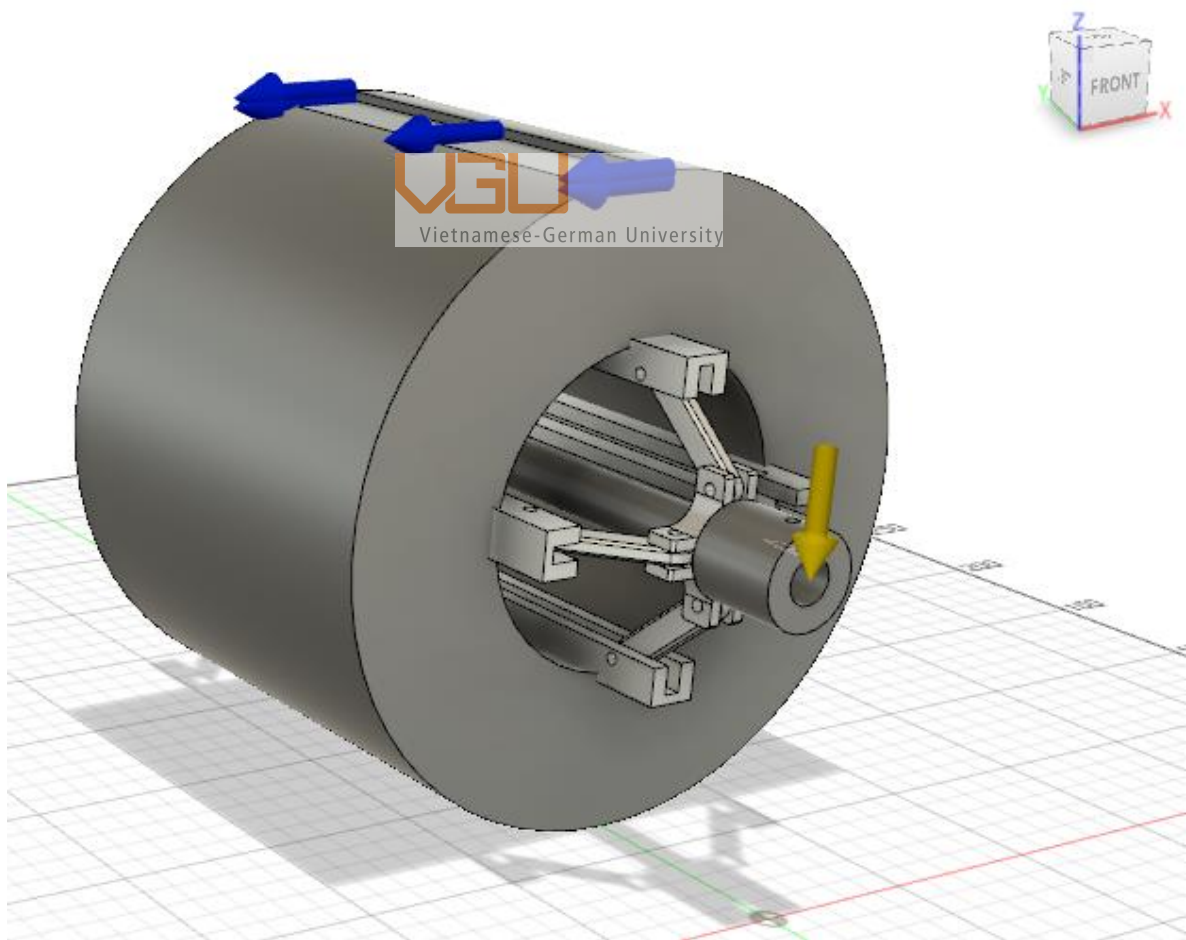


Figure 8. 1 Simulation setup

8.4. Von Mises Stress

The von Mises stress is a parameter that facilitates the determination of whether a particular material will experience yielding or fracturing. This parameter is typically applied to ductile substances like metals. The von Mises yield principle specifies that if the von Mises stress of a material subjected to a load is equal to or exceeds the maximum stress the material can withstand without permanent deformation of the same material under simple tension, then the material will undergo yielding[15].

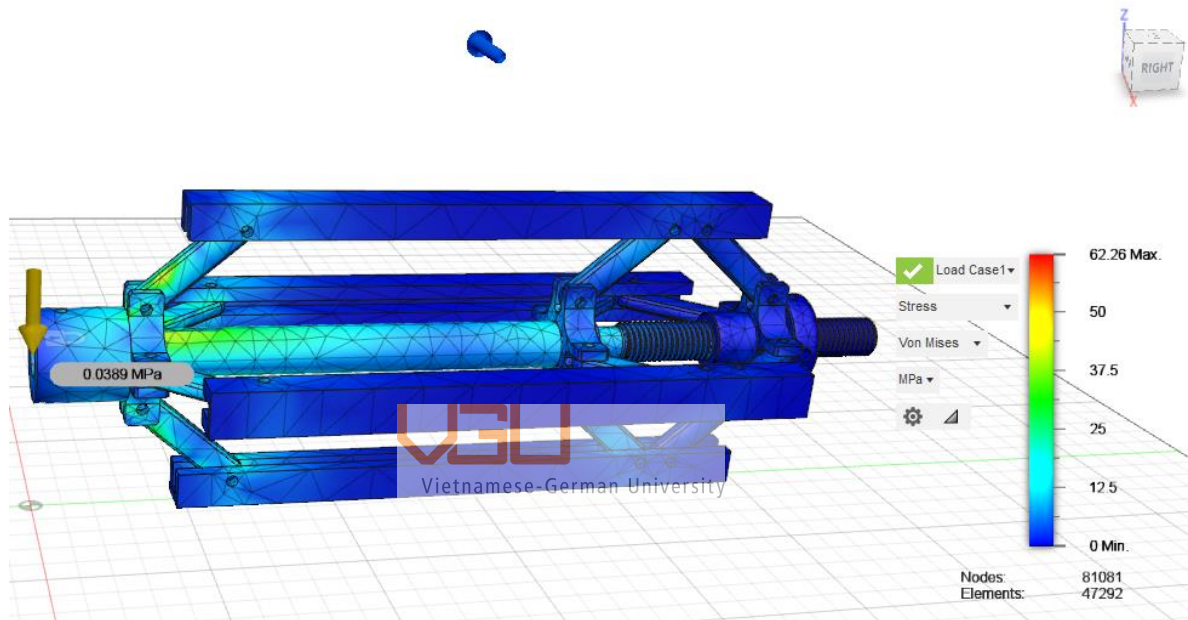


Figure 8. 2 Von Mises stress analysis

To make it easier to observe, the paper roll model has been hidden. The figure reveals that the highest stress value is 62.26MPa at the edge corner of the hub on the left. It is apparent that upon application of a 100N load, the area of highest stress is located to the left of the decoiling mechanism. Moving progressively towards the right of the mechanism results in a reduction in the magnitude of the stress experienced. By conducting Von Mises Stress Analysis, the design engineer can pin point the vulnerable areas in the current design.

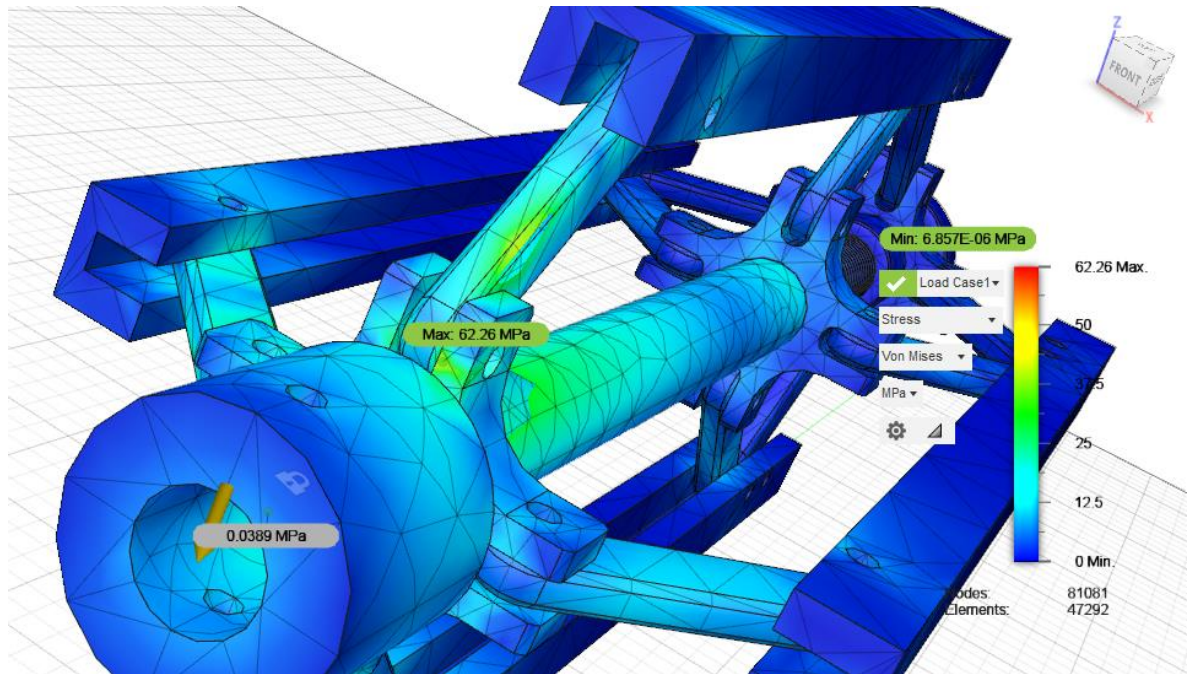


Figure 8. 3 Point of highest stress

8.5. Displacement

The principal objective of implementing the finite element method that is based on displacement is to evaluate the displacement distribution that occurs in a solid under the influence of external loads. To provide a clearer explanation, one can imagine a solid undergoing deformation due to external loads, where the displacement of every point within the solid in response to the applied force[16].

In this particular scenario, a load of 100N is acting on the roller. Observing the left-hand side, it is evident that there is no displacement when the load is applied. Conversely, on the left-hand side, there is a slight movement in the direction of the force applied of 0.28 mm.

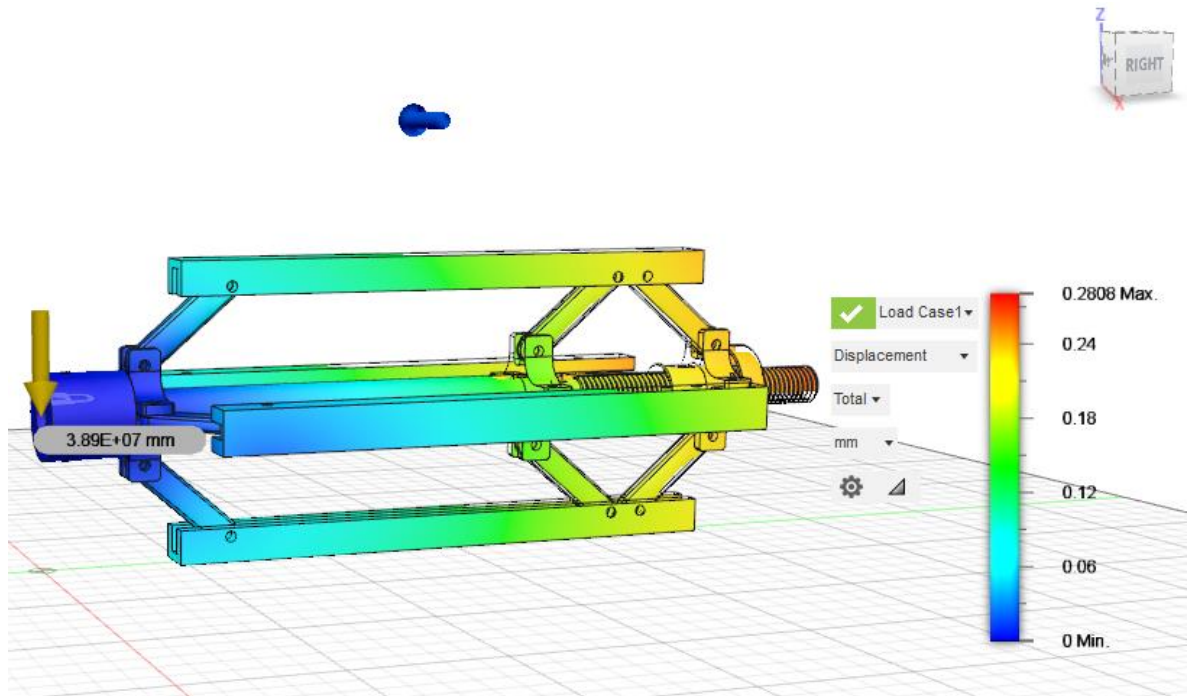


Figure 8. 4 Displacement analysis

8.6. Safety factor

The safety factor is a straightforward concept. It is the measure of the strength of the material compared to the maximum stress present in a particular component. If the stress in a given region exceeds the strength of the material, then the safety factor drops below 1, indicating that the component is at risk. Essentially, this means that in certain regions of the component, the stress levels exceed the material's strength. If the stress levels remain much lower than the material strength, the safety factor stays above 1, and the component is considered "safe"[17].

Based on the results obtained from the Finite Element Analysis, the safety factor values range from a maximum of 8+ to a minimum of 4.4. The area with the highest stress levels was identified as the least safe region. These findings provide evidence that the system is very safe, with all safety factor values indicating that the component can bear the applied loads without risk of failure.

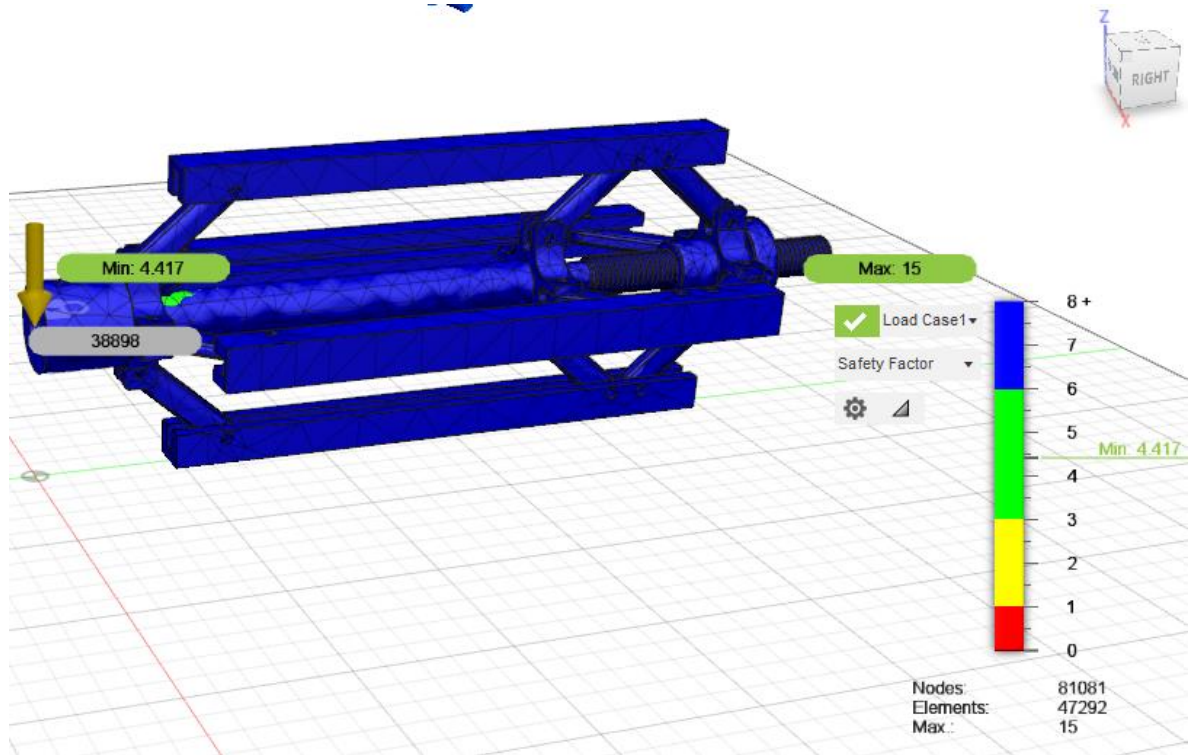


Figure 8. 5 Safety factor analysis

9. Testing results



9.1. Paper Reel Holding Test

The purpose of this test is to evaluate the performance of a decoiler in securely holding a paper reel and ensuring even contact between the mechanical components and the paper surface. The addition of twelve rubber pieces (figure 9.1) to the outer beam of the decoiler serves to enhance the friction coefficients between the mechanical components and the paper reel during the unwinding process. This modification is aimed at improving the overall grip and stability of the decoiler, ensuring a more secure and reliable hold on the paper reel.

Figure 9.2 and Figure 9.3, captured from opposing ends of the decoiler shaft, reveals a noteworthy observation. It is evident that the four beams of the decoiler expand uniformly, resulting in a well-distributed grip on the reel core.

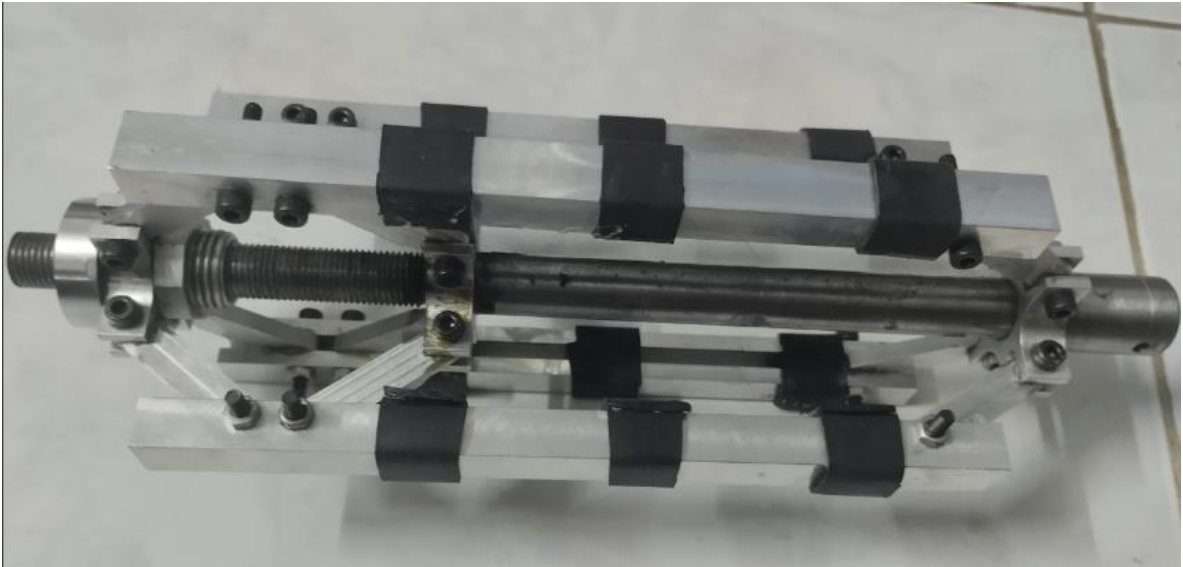


Figure 9. 1 twelve rubber pieces are attached on four beams



Figure 9. 2 Photo from one end 1



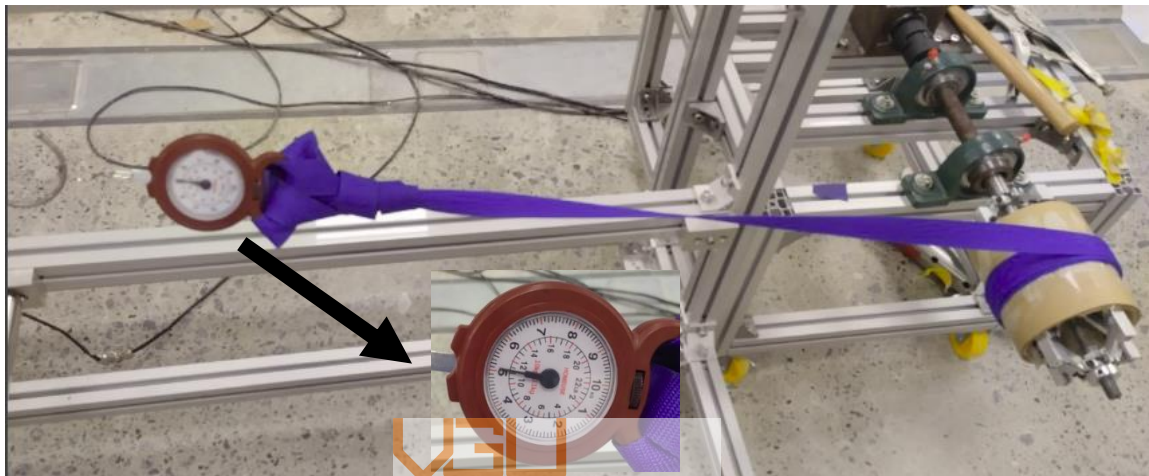
Figure 9. 3 Photo from one end 2

9.2. Tension Stability Test

The test aims to evaluate the performance of a decoiler in securely holding a paper reel under different web tension conditions. This test allows for the evaluation of the decoiler's capacity to sustain a uniform tension level and mitigate slippage or misalignment occurrences during the unwinding procedure. It holds significant importance in industries where precise tension control plays a pivotal role in attaining superior outcomes during the processing of paper reels, thereby ensuring a high standard of quality.

The procedure involves mounting the decoiler onto a drive shaft and placing a paper reel onto the decoiler. The decoiler is then adjusted to expand and firmly hold the paper reel in place. Subsequently, the web tension is created by pulling the web with weights of 2kg, 4kg, and

7kg. Throughout this process, marks are made on the decoiler and the paper reel to indicate their initial alignment. The primary objective of this test is to observe any changes in the relative position of the marks on the decoiler and the paper reel under different web tension levels. If the marks show significant shifts or misalignment, it indicates that the decoiler is unable to hold the corresponding tension securely. On the other hand, if the marks remain relatively stationary and maintain their original alignment, it suggests that the decoiler is capable of effectively holding the paper reel at the tested tension levels.



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Figure 9. 4 Testing at tension 5kg or 50N

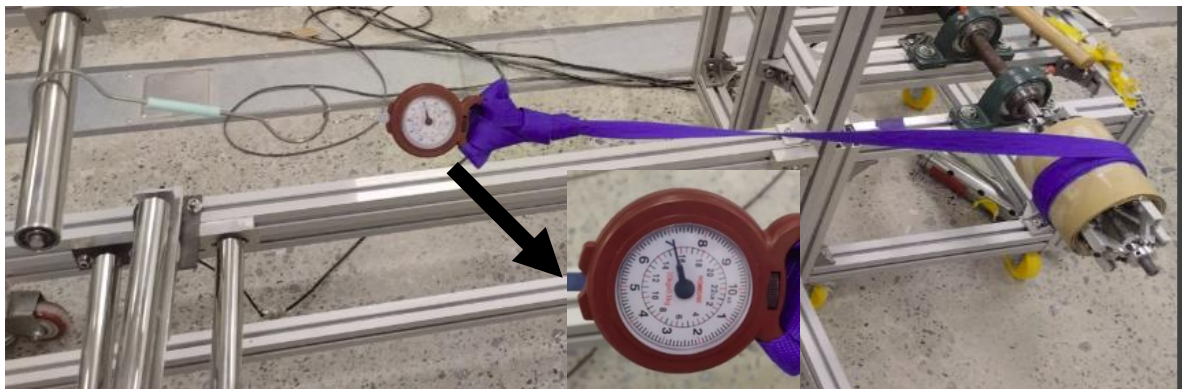


Figure 9. 5 Testing at tension 7kg or 70N

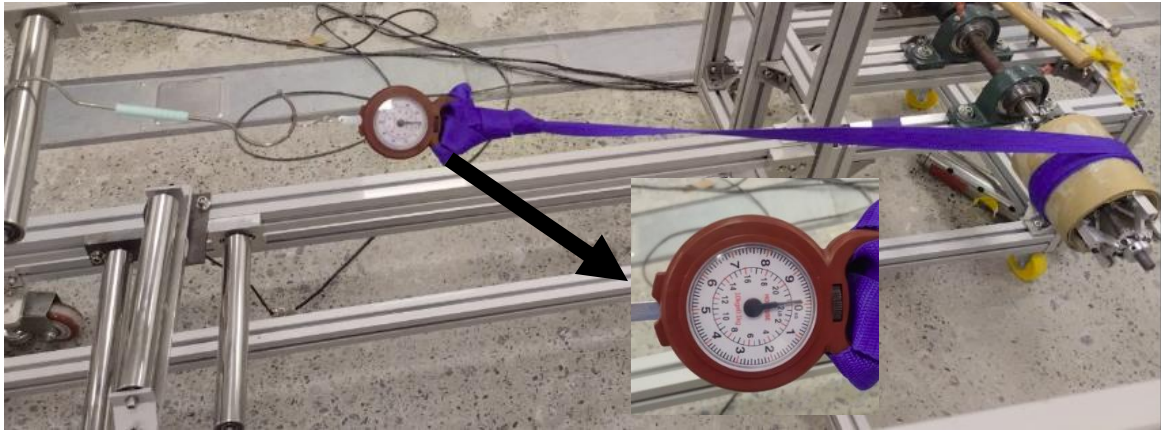


Figure 9. 6 Testing at tension 10kg or 100N



Figure 9. 7 Marks are made on the shaft and the paper reel

The results show that when the decoiler shaft is set to expand and grip the paper reel core strong enough, it can hold the core at the tension of 100N and the marks are still in the original position relative to each other.

10. Conclusion

10.1. Conclusion

In conclusion, all the stages of the thesis project, including ideation, creation, construction, and evaluation, have been accomplished effectively. The outcomes of the practical assessments demonstrate that each component within the system fulfills the necessary criteria, although there is room for enhancing precision and aesthetics. The mechanical structure of the prototype can now be connected to the Roll-to-Roll system in a suitable manner. This system can be installed to securely hold the reel core in both the rewinding and unwinding units of the overall system.

10.2. Future improvement

Overall, the conceptual design and manufacturing function effectively in accordance with the specified criteria. Nevertheless, further studies can be conducted to transform this design into a marketable product:

- Fatigue analysis should be carried out to ensure that the prototype archives a finite life or longest possible lifespan.
- The first concept in chapter 3 has a good potential for future development if cost-effective methods for producing the core can be discovered.
- Additional stress analysis should be conducted to optimize the geometry and dimensions of the component, aiming to reduce material and manufacturing costs.
- To support smaller diameter further researches to increase the strength of the material should be carried out.

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