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RUHR-UNIVERSITÄT BOCHUM



# **Conceptual Design and Prototype Development of a Nip Roll System for Web Transport Systems**

Bachelor Thesis 

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# CONCEPTUAL DESIGN AND PROTOTYPE DEVELOPMENT OF A NIP ROLL SYSTEM FOR WEB TRANSPORT SYSTEMS

A Thesis Presented

By

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Submitted to the department of Mechanical Engineering of the

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# ***ABSTRACT***

The web transport system is critical in web manufacturing since it enables for mass production of a variety of web products. A common use in a web transport system is the nip roll system, which is used to control tension on the web and move it through the sub-processes of the web process.

Since the last of 20<sup>th</sup> century, the development in nip roll system is getting more vigorous. There are some leading companies in USA are currently developing and optimizing nip roll systems, they are CAC (Converter Accessory Corporation), Glue Machinery Corporation, Double E Company LLC,... In the early 21<sup>st</sup> century, the moving of manufacturing to Asia led to a critical development in web manufacturing in Asian industry.

On the other hand, the cost of importing technology and items associated to the web process is increasing. Aside from the rise in import costs, the resources necessary for future development, such as overhaul and maintenance, are still not adequately optimized by present designs. It is urgently needed a nip roll system with an improved design and easy control that takes less processing time and requires low maintenance costs.

This thesis aims to create a prototype of a nip roll system for transport systems that will serve as a design reference as well as a reliable and low-cost replacement for existing web production. This paper is divided into two sections: a theoretical section with problem formulation and analysis, and a practical section with designing, assembling, and testing.

# ***ACKNOWLEDGEMENT***

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## *Error! Bookmark not defined.***NOMENCLATURE**

$F$	Force generated by one air cylinder, N
$A, B$	Reaction forces at bearing's shoulders on shaft, N
$q_0$	Line load across the shell width, N/m
$\tau_{max}$	Maximum shear stress, $N/m^2$
$\sigma_e$	Equivalent stress, $N/m^2$
$\sigma_x$	Normal stress in x-direction, $N/m^2$
$\sigma_y$	Normal stress in y-direction, $N/m^2$
$\sigma_{max}$	Maximum stress at extreme fiber of beam, $N/m^2$
$\tau_{xy}$	Shear stress in Oxy plane, $N/m^2$
$S_{yt}$	Yield strength in tension, $N/m^2$
FOS	Factor of safety, dimensionless
$F_{max}, F_{min}$	Maximum and minimum air force subjects to shaft by one pneumatic cylinder, N
$M_b$	Bending moment, Nm
$M_T$	Torsional moment (Torque), Nm
$I_y$	Second moment of area, $m^4$
$I_T$	Torsional constant, $m^4$
$I_p$	Polar second moment of inertia, $m^4$
$r_1$	Radius of idler roller's shell, m
$r_{CO_1}$	Outer radius of idler roller's shell, m

$r_{CI_1}$	Inner radius of idler roller's shell, m
$r_2$	Radius of rubber covered roller's shaft, m
$r_{CO_2}$	Outer radius of rubber roller's shell, m
$r_{CI_2}$	Inner radius of rubber roller's shell, m
$R_1$	Outer radius of idler roller's shell, m
$R_2$	Outer radius of rubber roller covering, m
$R_{eq}$	Equivalent radius, m
$a$	Distance from bearing supports to shell supports, m
$w_R$	Width of rollers, m
$d_2$	Diameter of rubber roller's shaft, m
$N_N$	Nip load or distributed load (Nip forces divided by contact width), N/m
$t$	Rubber covering thickness, m
$H_0$	Statical friction, N
$\mu_0$	Statical friction coefficient, dimensionless
$P_{avg}$	Average pressure between nip rollers, $N/m^2$
$w_c$	Deflection at center of idler roller in z-direction, m
$b$	Nominal width of contact zone, m
$b_{actual}$	Actual width of contact zone, m
$\Delta t$	Interval of time required for one particle on the rollers to rotate through angle $\theta$
$V_1$	Surface velocity of roller 1, m/s
$V_2$	Surface velocity of roller 2, m/s



$\omega$	Angular velocity, degree/s
$\theta$	Angle subtended by the nip zone in roller 2, degree
$\varepsilon$	Creep, dimensionless
$\delta$	Rubber cover indentation or engagement at center of MD contact length, m
$t_1$	Time required for roller 1 to rotate a complete rotation, m
$t_2$	Time required for roller 2 to rotate a complete rotation, m
$E_0$	Young's modulus for rubber, MPa
$\nu$	Poisson's ratio of elasticity, dimensionless
$H_{SA}$	Rubber hardness (International Rubber Hardness Degree, Shore A Durometer)



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# CHAPTER 1: Introduction

## 1.1 Background study

### 1.1.1 Introduction to nip roll system

Automated manufacturing is a major factor in today's industrialized world, allowing for the flow manufacturing of a wide range of consumer goods and products. As an outcome of mass production, a continuous, long, flexible, and thin roll of material is produced. The term "web" refers to this sort of product. Web can be manufactured in the form of rolls of materials such as paper, plastics, films, foils, textiles, wires and metals or finished products such as diapers, tissues, female hygiene products, food wraps, labels, tapes, magazines, aluminum cans, and even some electronics.

In the past, there are many various methods used to produce web, but most of them are often manual methods and their quality and quantity are still limited in manufacturing. In aid of automation, a new process called Roll-to-roll (R2R) is originated. It is a fabrication method used in manufacturing which prints, coats, embeds or laminates a variety of applications onto a substrate material while that material is fed continuously from one roll to another, shown by Figure 1. The R2R process typically consists of numbers of rollers for purposes of winding and unwinding substrate materials. That substrate is fed out from an unwind unit through one or more mid-process steps and be rewound again into another roll to form a new roll of substrate. The technique is continuous and materials can be added to or subtracted from the substrate when it travels along the web to produce a product. The terms "Roll-to-roll" are common and used predominantly since a word "web" in "web process" is often confused with the same word in "World Wide Web".

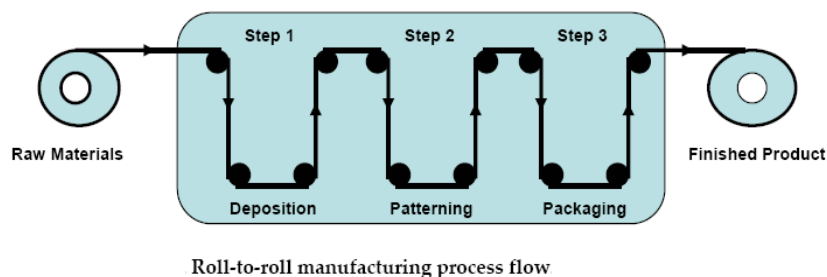


Figure 1: Roll-to-roll manufacturing process flow

Within a roll-to-roll process, a nip roll system is often placed between rolls, which generally generates and controls tension on the web. There are two common applications of nip rolls. One of them is called a drive roll which has a purpose of controlling the tension and velocity of the web, as shown in Figure 2. A rider roll in contact with a winding roll is the second application. Its function, as indicated in Figure 3, is to either drive the winding roll or exclude air trying to enter the winding roll.

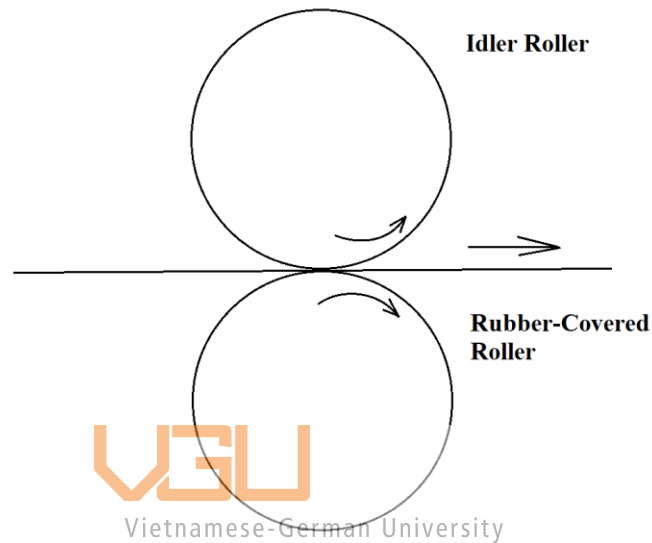


Figure 2: A Nip Roller and A Rubber Covered Roller

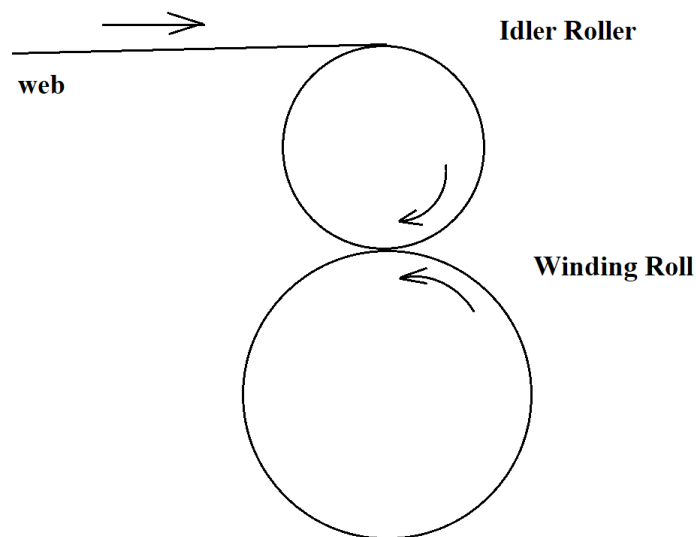


Figure 3: A Nip Roller and A Winding Roll

### 1.1.2 Review the design of nip roll system

Through design synthesis and analysis, a prototype will have a suitable geometric model for particular purposes with high degree of safety. This section provides geometric model for main machine elements and their related motions.

This thesis scopes the first application mentioned above in Section 1.1.1. Specifically, a nip roll system generally comprises 2 rollers: a Idler Roller (or Metal Roller) and a Rubber Covered Roller (or Nip Roller). The idler or idle roller is aim for support. With two bearings attached at both shaft's ends which allows the shaft to rotate freely at the same speed as the web. This roller could either move up or down for the purpose of controlling the rollers' contact area and replacing web materials and emergency reasons. A nip roller, the remain one, is usually a rubber covered roller or a roller with a flexible surface that is used to squeeze the web. The reason why this roller has to be covered by a roll of elastic material will be discuss in Section 2.1. These rollers will be supported by some bearings for the purpose of spinning around its axis. Those bearings are then connected to a body frame via some bearing housings, which transmit a parallel motion to the rollers: rotating around its axis and upward/downward translational motion. The web is controlled by two features, they are tension, and velocity. Tension is generated and controlled by air cylinders while velocity is controlled by a motor.

Two rollers have a cylinder shape which will carry loads and must be designed based on strength requirements. This analysis is discussed in the Chapter 2. The shape of bearing housings are rectangular, which take responsibility for a translational motion along the body frame. All immobile parts will have a rectangular shape for the purpose of reducing processing time.

This system is aiming at attaching to a chain of processing stages, so the design will be designed in a way that it is flexible to locate at any position. It implies that the size and the weight of the system must be considered carefully, which are determined by the selection of material and the system geometry. The traditional considerations are involved to design this prototype. Traditional considerations include strength, deflection, weight, size and shape, material properties, operating conditions, processing, cost, availability, usability, utility and life.

## 1.2 Research problem

The effectiveness of a roll-to-roll process depends on the degree of accuracy of the web tension control. Controlling tension on substrate web when it travels from a roll to another is problematic to manufacture. Throughout a whole R2R process, the web velocity must remain constant and uniform, because any variation in tensions could lead to destabilization of material applied to the substrate, which makes product unstable then.

In addition, the cost to produce a web transport system is quite high and the resources needed for future development are still not optimized, such as time to re-build the system is long, a complicated system required a high amount of components and parts which costs more than the simple one, these features disadvantageous for maintenance and overhaul as well. A solution of a nip roll system with an optimized design and simple, which consumes low cost in processing time and overhaul is a significantly needed.

## 1.3 Scope of implementation

This thesis is intended for use for web transport systems and for future developments. It has a theoretical section with formulation and problem analysis, as well as a practical section with mechanical designs and assembly.

## 1.4 Objectives

The objective of this thesis is to make a conceptual design and a prototype of the nip roll system. It includes a research on linear and non-linear materials' behaviors under loads, as well as provides options that could optimize the system for the purpose of development in the future.

## CHAPTER 2: MECHANICAL DESIGN OF NIP ROLL SYSTEM

The objective of this section is to attempt to do following things in sequence:

- Determining shapes.
- Problem formulating and computing under idealized assumptions.
- Calculating weight, geometry dimensions and tolerances.

Shapes of some parts are standard and available such as shafts and bearings, while shapes of other parts are needed to be determined such as frames, bearing's housings, etc. So, in this chapter, for rollers and bearings, their dimensions are determined only, while for the remain parts, both shapes and dimensions are needed to be determined. To determine shapes, the collection of necessary data for failure analysis must be obtained early in design process, for example, a rectangular shape is suitable for a straight movement rather than a curve one.

For purpose of logic, geometric dimensions must be calculated and determined within a practical allowed range of values obtained from formulas and several safety conditions. Hence, the section of problem formulation and computation is put first in this chapter.

### 2.1 Problem Formulation and Computation

This nip roll system is designed to be assigned to accomplish a particular task. The purpose of formulating and computing is to provide adequate evidences to build a reliable system which solve a specific task. In particular, the task is to design a nip roll system which is able to transport a web subjected by a tension force of 60N. The unwound web is fed at the velocity of  $0.2\sim 0.5\text{ m/s}$ .

#### 2.1.1 Rollers' Shaft

A rubber covered roller is composed by 3 parts: a shaft, a shell and a rubber covering, shown by Figure 4. To design a shaft, these following dimensions must be specified:

- Cross sectional radius
- Journal length
- Distance between bearing's shoulders

Forces acting on two rollers' outer surface is a force over an area of contact. Force is transmitted from the shell to the roller shaft through two shell supports, which means the line loads on shell's surface then becomes concentrated forces acting at two points on the shaft. Based on nip roller's function, the shaft is only allowed to rotate around its axis. The problem on shaft could be treated as a plane problem of a beam supported at its ends and is subjected by two external forces which are perpendicular to its axis, shown by Figure 5.

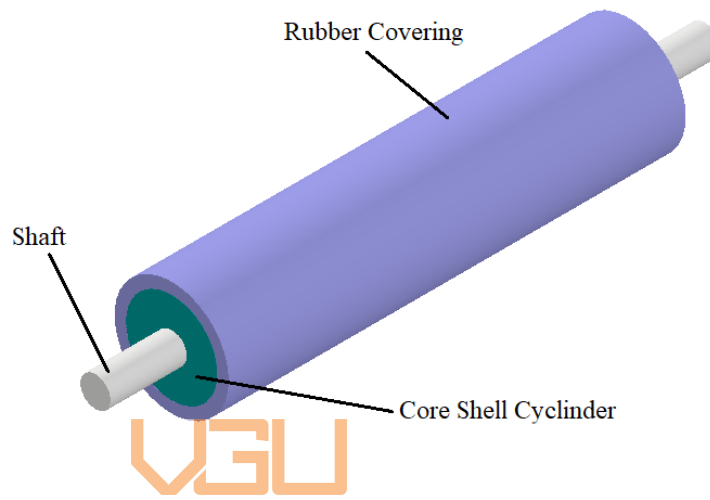


Figure 4: Rubber covering roller

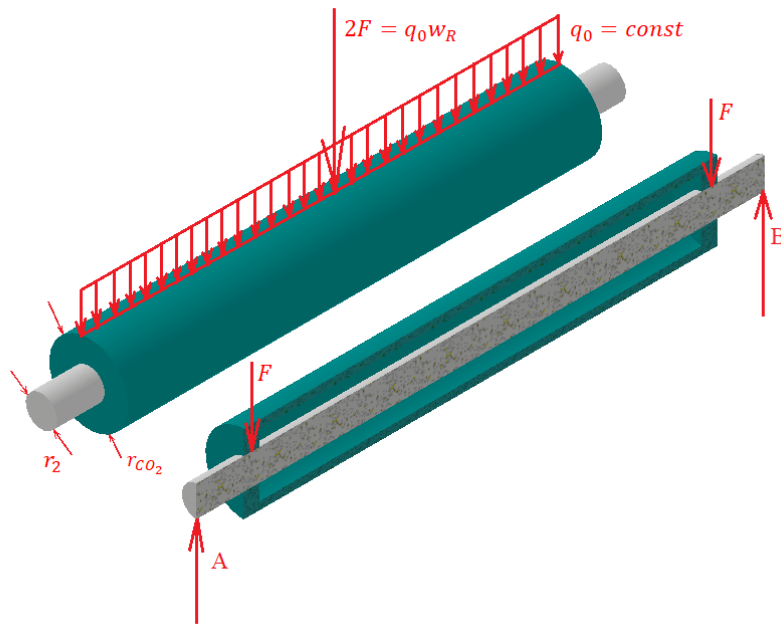


Figure 5: Shaft and shell of Rubber Covered Roller

To prevent failures, the shaft is designed on the basis of strength (strength hypotheses), which means its maximum stress and strain resistance must not exceed a certain allowable value. There are some strength hypotheses used to design a beam such as Maximum-normal-stress hypothesis, Maximum-shear-stress hypothesis and Maximum-distortion-energy-hypothesis. The key factor to choose an appropriate hypothesis is material quality. That means whether material has a quality of a ductile or brittle material.

There are many materials available for shaft design, but the most suitable one is carbon steel. By this selection, the Maximum-shear-stress hypothesis is used to derive equations as an appropriate strength hypothesis, because carbon steel has the property of a ductile material. Maximum-shear-stress hypothesis based on the assumption that failure occurs when the largest principal stress reaches a critical value, expressed by equation (2.1).

$$\tau_{max} = \frac{1}{2} \sigma_e \quad (1.1)$$

Where:

The maximum shear stress:



$$\tau_{max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \quad (1.2)$$

The equivalent stress:

$$\sigma_e = \frac{S_{yt}}{FOS} \quad (1.3)$$

The yield strength in tension  $S_{yt}$  is a characteristic material property. There is a significant difference in value of yield strength in tension between ductile and brittle materials. However, in ductile material, this value is always higher. That is the reason why ductile material is preferred to design ordinary shafts. To be more specific, JIS G3101 SS400 Structural Carbon steel is chosen material for the design, and its yield strength value equals to 245 MPa (or  $N/m^2$ ).

$$S_{yt} = 245MPa \quad (1.4)$$

One essential factor that is inevitable for structural designing is the factor of safety  $FOS$ . Safety factor measures the degree of separation between the stress and strength for a specific



failure mechanism. It is the ratio between the strength of the material and the maximum stress in the part, expressed by equation (1.3). When the stress in a specific position becomes superior to the strength of the material, the safety factor ratio becomes inferior to 1, this when there is a danger.

There are some ranges of safety factor values which are determined based on particular conditions. For example, for use with highly reliable materials where loading and environmental conditions are not severe and where weight is an important consideration, the reasonable factor of safety will be around 1.3 - 1.5. While for use with reliable materials where loading and environmental conditions are not severe, the available value will be 1.5 - 2.0. In web handling applications, the feasible value for safety factor is commonly 1.5.

$$FOS = 1.5 \quad (1.5)$$

To obtain a relation between normal stresses, shear stress and yield strength of tension expressed in equations (1.2) and (1.3), the investigation into stresses must be executed. The investigation is initially accomplished by choosing Cartesian coordinates as a reference coordinates  $Ox, y, z$ , shown in Figure 6.

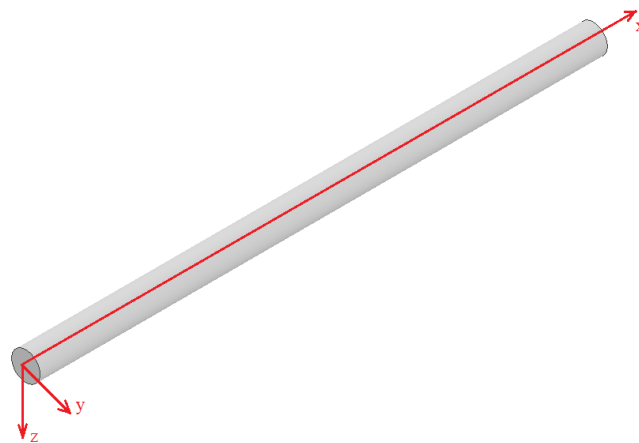


Figure 6: Shaft and Cartesian coordinates  $Ox, y, z$

The beam is subjected in  $z$ -direction. So, the bending moment in  $y$ -direction is zero:

$$\sigma_y = 0 \quad (1.6)$$

The bending stress in  $x$ -direction:

$$\sigma_x = \frac{M_b}{I_y} z \quad (1.7)$$

For circular shaft, the bending stress attains their maximum values at the extreme fiber  $z_{max} = r_2$ , the bending stress then becomes:

$$\rightarrow \sigma_x = \sigma_{max} = \frac{M_b}{I_y} r_2 \quad (1.8)$$

In order to obtain the diameter of the shaft, the bending moment  $M_b$  and torsional moment  $M_T$  of the beam must be investigated.

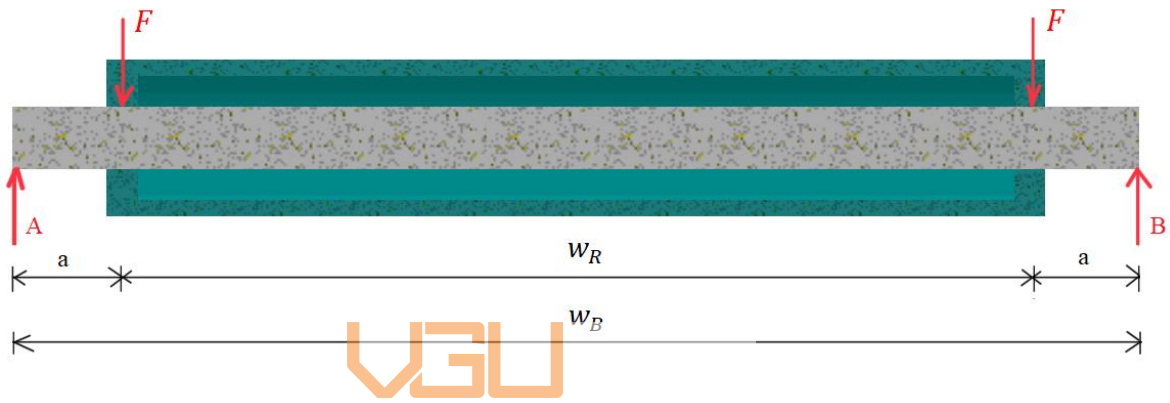


Figure 7: Forces investigation on Shaft in Oxz plane

The system is static determinacy, shown by Figure 7. Then, the support reactions can be calculated directly from the equilibrium conditions. The equilibrium conditions are given by:

$$\uparrow: \quad A + B = 2F,$$

$$\text{Moment at A: } -Fa - F(w_R + a) + B(2a + w_R) = 0 \quad \rightarrow \quad B = F$$

$$\rightarrow \quad A = F$$

There are 3 jumps in shear forces through the journal of the beam, so the beam will be investigated at 3 different sections to analyze the bending moment  $M$ , shown by the following tables.

Table 1: Section 1 investigation

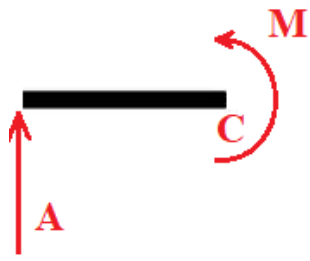
Section 1:		$0 \leq x < a$
Free body diagram		
Moment analyze	Moment at C:	$M - Ax = 0$
	$\rightarrow$	$M = Ax = Fx$

Table 2: Section 2 investigation


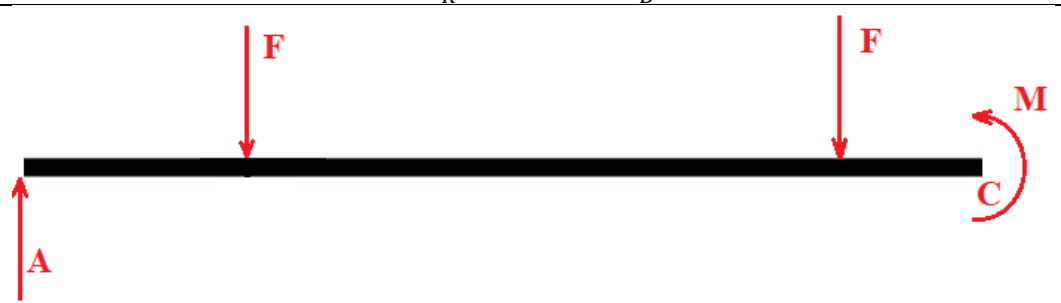
Section 2:		$a \leq x < a + w_R$
Free body diagram		
Moment analyze	Moment at C:	$M = Ax - F(x - a)$
	$\rightarrow$	$M = Fa$

Table 3: Section 3 investigation

Section 3:		$w_R + a \leq x < w_B$
Free body diagram		
Moment analyze	Moment at C:	$M - Ax + F(x - a) + F(x - a - w_R) = 0$
	$\rightarrow$	$M = F(2a + w_R - x)$

Hence, the bending moment has the maximum value at section  $w_R \leq x < w_B$  and through this section, its value is unchanged.

$$M_{max}(a \leq x < a + w_R) = M_b = Fa$$

The moment of inertia:

$$I_y = \frac{\pi r_2^4}{4} \quad (1.9)$$

The shear stress:

$$\tau_{xy} = \frac{M_T}{I_T} r \quad (1.10)$$

For circular shaft, the torsional formula attain the maximum values at the outer boundary, i.e. at  $r = r_2$ . The shear stress then becomes:

$$\tau_{max} = \frac{M_T}{I_T} r_2 \quad (1.11)$$

The torsional moment is obtained from the power of the servo.

$$M_T = \frac{60 \cdot P}{2\pi N} \quad (1.12)$$

For circular cross section, the torsional constant  $I_T$  will be equal to the polar second moment of inertia  $I_p$ .

$$I_T = I_p = \frac{\pi r_2^4}{2} \quad (1.13)$$

By substituting (1.2) - (1.13) into (1.1), we obtain a relation:

$$\sqrt{\left[\frac{16M_b}{\pi d_2^3}\right]^2 + \left[\frac{16M_T}{\pi d_2^3}\right]^2} = \frac{1}{2} \frac{S_{yt}}{FOS} \quad (1.14)$$

Which leads to the maximum force to which the system can be subjected in order to prevent failures.

$$F_{max} = \frac{1}{a} \sqrt{\left(\frac{\pi r_2^3 S_{yt}}{256 FOS}\right)^2 - \left(\frac{30 \cdot P}{\pi N}\right)^2} \quad (1.15)$$

To prevent slipping, the maximum nipping force acting on web must not exceed the static friction between contact surfaces.

$$T \leq \mu_0 F \quad (1.16)$$

Rubber does not have a single constant coefficient of friction. The coefficient depends on temperature and other variables such as dry or lubricated condition. However, for web handling purpose, to prevent slipping, the friction generated on rubber surface must be a static frictional force and the minimum value of static friction coefficient that can be found in the table in Reference [11] is:

$$\mu_0 = 0.2 \quad (1.17)$$

It is known that, the minimum tensional force on web is  $T = 60N$ , then the minimum pressing force required to prevent slipping is:

$$F_{min} = 300N \quad (1.18)$$

### 2.1.2 Rollers' Shell

To design a roller shell, these following dimensions must be specified:

- Inner radius
- Outer radius
- Width of roller



Because the overall deflection present in the system includes both deflection on the beam and deflection on the roller shell, designing the roller shell is fairly difficult. The total forces throughout the region of contact determine the average pressure on the nip roller surface. The contact area  $a$  between two rollers is equal to the width of the roller multiplied by the machine direction length of contact (a.k.a. MD footprint length). The dispersed load in terms of force per width (N/m) equals the nip forces divided by width. Figure 8 shows how the footprint length,  $b$ , is determined by roller radii and indentation or overlap of the rollers beyond tangential contact.

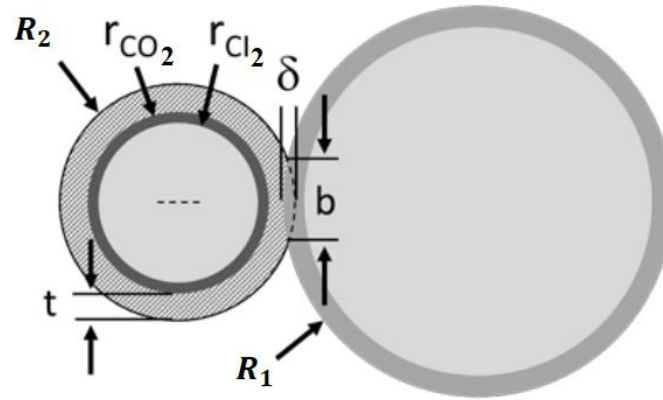


Figure 8: Cross sectional view of two rollers in contact

$$P_{avg} = \frac{2F}{w_R b} = \frac{N_N}{b} \quad (13)$$

Where

$$N_N = \frac{2F}{w_R} \quad (2.2)$$

Equations for simply supported beams with a uniform distributed load provide a first estimate for nip roller deflection. Equation (2.3) expresses the deflection of the roller's shaft with an assumption that the bearings do not transfer a moment from the shaft bending to the core shell cylinder bending. This equation implies that the increase in deflection is worse with longer shaft or journal as well.

$$w_c = \frac{5N_N w_R^4}{384EI_y} \quad (2.3)$$

The moment of inertia of the core thin-walled shell cylinder:

$$I_y = \frac{\pi(r_{CO_2}^4 - r_{Cl_2}^4)}{4} \quad (2.4)$$

Based on the bending equations, to minimize deflection of roller shafts with simple end supports, we have the following options:

- Decrease end load (force per width)
- Decrease roller width
- Increase elastic modulus of roller core

- Increase outer roller diameter
- Decrease inner roller diameter
- Decrease bearing width

### 2.1.3 Rubber Covering

To design a rubber covering, these following features must be specified:

- Covering thickness
- Covering width
- Rubber hardness

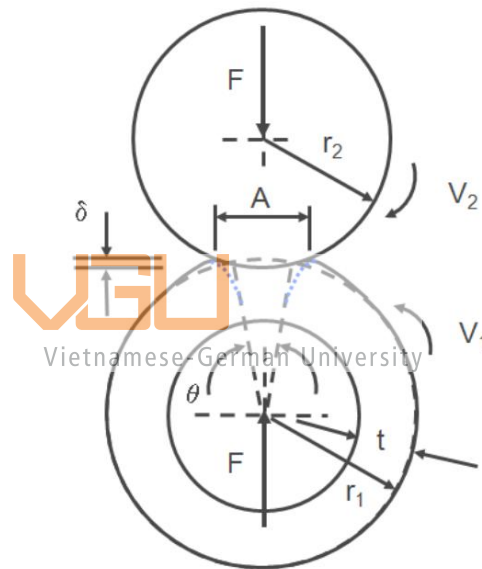


Figure 9: Cross sectional view of two rollers in contact

In this section, the introduction of indentation and creep is presented. These two features are affected by rubber covering geometry and hardness. Figure 9 illustrates a cross sectional view of an rubber covered roller (number 2) loaded against an uncovered backing roller (number 1). There is no web considered to be in the system. Existing at contact zone, each particle on the surface of each roller rotates at the same surface speed,  $V$ . After two rollers are brought into an indentation or engagement by an amount,  $\delta$ , the roller 1 still travel at the same surface speed. Therefore,  $V_1 = V$ . However,  $V_2$  will no longer be equal to  $V$ . There is a circumferential strain that develops within the contact, or nip zone. This circumferential strain is determined by the characteristics of the rubber covering on roller 2 (rubber covered

roller). The angle,  $\theta$ , is defined as the angle subtended by the nip zone in the roller 2 assuming that there is no circumferential strain in the cover. Under this assumption, the nip width is equal to  $r_2\theta$  and the interval of time required for roller 2 to rotate through this angle is found from the following:

$$\Delta t = R_2\theta / V_2 \quad (3.1)$$

In reality, the actual nip width,  $b_{actual}$ , will be different than the theoretical nip width by some nominal amount,  $\varepsilon$ , which refer to as the creep of the system:

$$b_{actual} = (1 + \varepsilon)R_2\theta \quad (3.2)$$

The creep is the average circumferential strain in the nip zone which relates to the speed of roller 2. From mass conservation, a particle on the surface of roller 1 must travel through the nip zone in the same interval of time as roller 1 rotates through  $\theta$ .

$$\Delta t = b_{actual} / V_1 = b_{actual} / V \quad (3.3)$$

Combining equations (3.1), (3.2) into (3.3) yields the surface velocity of roller 2 in terms of the creep:

$$V_2 = V / (1 + \varepsilon) \quad (3.4)$$

Equation (3.4) indicates that the actual surface velocity of roller 2 is slower than the web velocity. This equation indicates that for positive creep, the rubber covered roller will travel at a slower speed than the drive roller. Contradiction, for negative creep, the rubber covered roller will travel at a higher speed than the drive roller.

Creep is calculated from the relation between rotation times of two roller according to the following equations:

$$t_1 = \frac{2\pi(1 + \varepsilon)r_1}{V} \quad (3.5)$$

And



$$t_2 = \frac{2\pi r_2}{V} \quad (3.6)$$

Giving:

$$\varepsilon = \frac{r_2 t_1}{r_1 t_2} - 1 \quad (3.7)$$

Experiment executed in Reference [2] shows that the value of creep is significantly small and proportional to nip load and Poisson's ratio. Creep is approximately smaller than 1% of nip load. For example, the Poisson's ratio is 0.46, Nip load is 100 N/m, so the creep is around 0.5~0.8 mm.

Covering thickness is defined by outer and inner covering radius. The thickness must be larger than indentation in order to prevent failure occurred in the covering. The outer radius of the rubber covering is calculated from relation between linear and angular velocity of the shaft, expressed by the following equation.

$$R_2 = \frac{V_2}{\omega} = \frac{V_2}{\left(\frac{2\pi N}{60}\right)} = \frac{30V_2}{\pi N} \quad (3.8)$$

Rubber covering wraps over the outer surface of the roller shell, so the covering width must be at least equal to the shell width. In this prototype, the covering width is equal to the shell width.

To choose the appropriate rubber hardness for the covering, the influences of the rubber properties on the web must be analyzed. Reference [4] provides the relationship between rubber covering indentation and the force per width (Nip load) created by the rubber compression, expressed by equation (3.9). Rubber covers are elastic materials that have a stress-strain relationship, but in nip rollers this relationship is complicated by the cylinder-cylinder geometry and plane strain limits of rubber across a wide nip.

$$N_N = \frac{2(1-\nu)^2}{3(1-2\nu)(1-\nu^2)} \frac{E_0}{t} \sqrt{2R_{eq}} \delta^{3/2} \quad (3.9)$$

Equation (3.9) is based on a cylinder and a plate (base radius is infinity), but can be applied to cylinder-cylinder contact by substituting an equivalent radius, expressed by equation (3.10).

$$R_{eq} = \frac{R_1 R_2}{R_1 + R_2} \quad (3.10)$$

If equation (3.9) is obtained, indentation can be found from nip load, roller geometry, and rubber properties, expressed by equation (3.11).

$$\delta = \left( \frac{3(1-2\nu)(1-\nu^2)}{2(1-\nu)^2} \frac{N_N t}{E_0 \sqrt{2R_{eq}}} \right)^{2/3} \quad (3.11)$$

These equations require the elastic modulus (Young's modulus) and Poisson's ratio of the rubber cover. Reference [4] provides detailed information about both of these properties. In that work, it is shown that for natural and synthetic rubbers, the Young's modulus can be estimated by the Shore A (i.e.,  $H_{SA}$ ) hardness by the following expression (Reference [4], equation 1):



$$E_0 = 145.7 e^{0.0564 H_{SA}} \quad (3.12)$$

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This work also found a typical or average Poisson's ratio for various rubber:

$$\nu = 0.46 \quad (3.13)$$

## 2.2 Selections of Materials and Machine Elements

In design process, some criteria must be made to allow processing time and cost to be reduced which could be done by selecting suitable materials with low cost. In addition, the selected materials should be available in the market while its physical and geometrical properties still adequately satisfy requirements obtained from theories and calculations. However, in consequence, the total weight of the system could be heavy, which leads to difficulties in initial installation, but this can be optimized by replaced materials used for prototype by alternative materials with lighter weights.

Table 4: List of machine elements

No.	Machine element	Quantity	Material
1	Body frame	2	JIS G3101 SS400 steel
2	Top frame	1	JIS G3101 SS400 steel
3	Bottom frame	1	JIS G3101 SS400 steel
4	Supporting trusses	1	JIS G3101 SS400 steel
5	Motor mount	1	JIS G3101 SS400 steel
6	Motor coupling	1	Cast iron
7	Coupling key	1	Standard
8	Flange	2	SUS 304
9	Pneumatic cylinder	2	Standard
10	Double knuckle joint	2	Aluminum
11	Locating pin	4	SUS 304
12	Bearing	2	Standard
13	Bearing housing	2	JIS G3101 SS400 steel
14	Idler Roller/ Metal Roller	1	JIS G3101 SS400 steel (Material of Roller's shaft and shell)
15	Nip Roller/ Rubber Covered Roller	1	JIS G3101 SS400 steel (Material of Roller's shaft and shell), Durometer 70 Shore A (Rubber's hardness)
16	Servo Motor	1	Standard
17	Servo Drive	1	Standard

### 2.3 Design and Manufacture of Mechanical parts

In general, a nip roll system consists of 2 main parts, those are a mechanical part and an electrical part. A mechanical part includes a body frame, a top and a bottom support, supporting trusses, pneumatic cylinders and other mechanical elements. While the electrical part includes a servo motor and a servo drive, interface devices, power sources and other electrical components.

The system is designed in a way that all parts can be modified and developed separately without changing the entire system's structure. So, the body frame has an rectangular U-shape which has an open structure. This type of structure is advantageous to assemble and disassemble dynamic parts without relocating static ones. On top of the frame, a top support is positioned, which holds two pneumatic cylinders in place as well as keeping the two body frames parallel. The bottom support, which has the same function as the top support, is positioned directly beneath the main frames and raised by the supporting trusses.

Within the alley of the body frame, two bearing housings are placed, one movable and the other immobile. There are two main type of movements presenting in the system, which are a translational movement implemented by expansion and shortening of pneumatic cylinders and a rotational movement implemented by a servo motor and bearings.

### 2.3.1 Body Frames

Two rollers exert contact forces on web by increasing their surface's contact area. This working principle involves a sliding mechanism with one roller sliding along rails while the other remaining motionless. Basing on that conceptual idea, the body frames' overall appearance resembles a rectangular U-shape, with two slide rails attached to opposite sides of the alley.

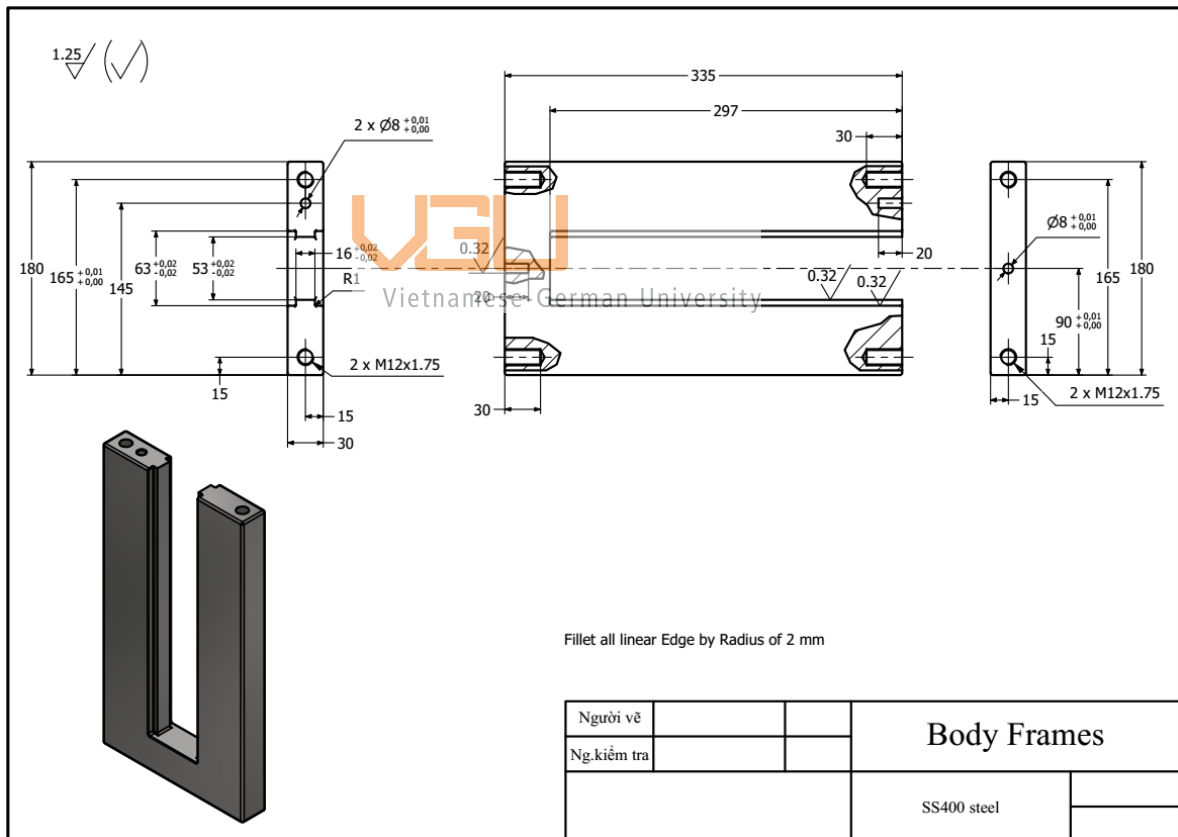


Figure 10: Body frames

Function:

- To attach supports, bearing housings and the motor mount.

- To allow bearing housings to slide vertically.

Material: JIS G3101 SS400 steel.

Volume: 1277315.419  $mm^3$

Mass: 9.96 kg

### 2.3.2 Bottom Support

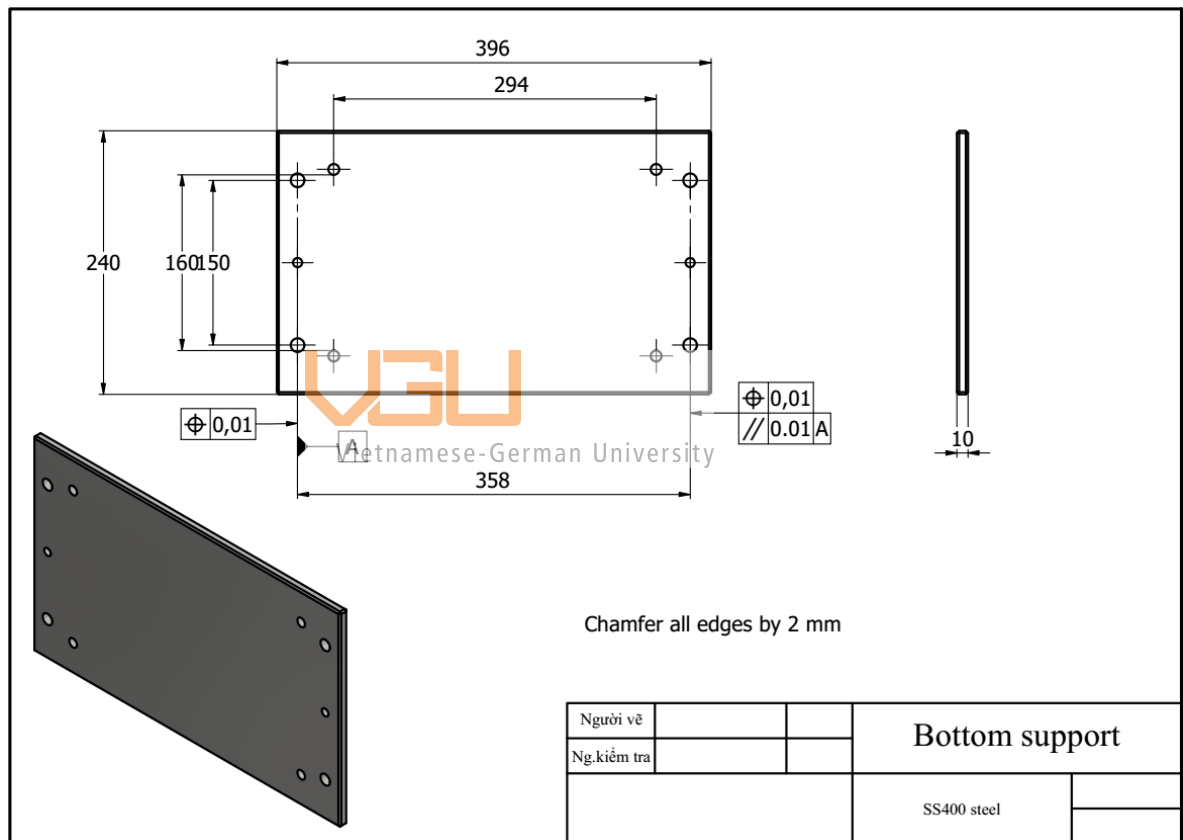


Figure 11: Bottom support

Function: To support body frames

Material: JIS G3101 SS400 steel.

Volume: 931487.805  $mm^3$ .

Mass: 7.26 kg

### 2.3.3 Supporting trusses

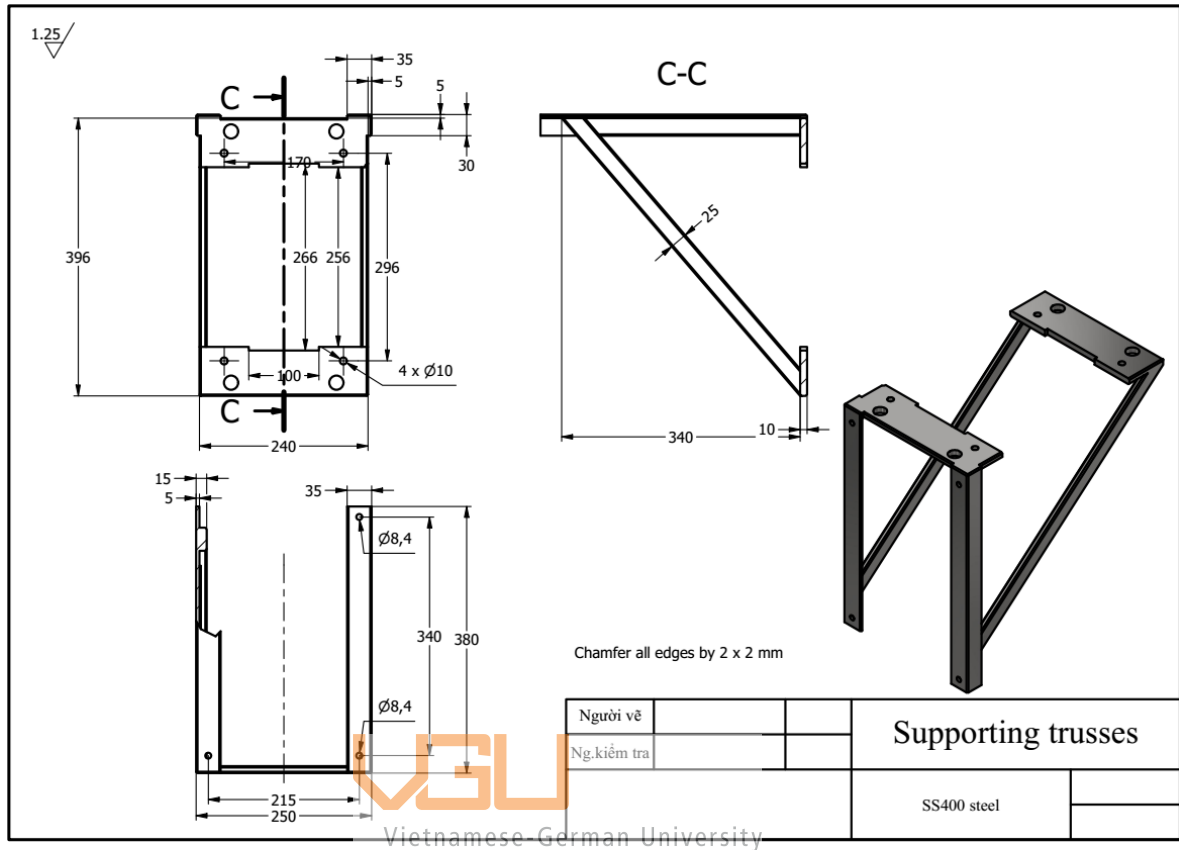


Figure 12: Supporting trusses

Function:

- To lift body frames and bottom support.
- To keep the roller axis staying along the horizontal line.

Material: JIS G3101 SS400 steel.

Volume: 779114.848  $mm^3$ .

Mass: 6 kg.

### 2.3.4 Top support

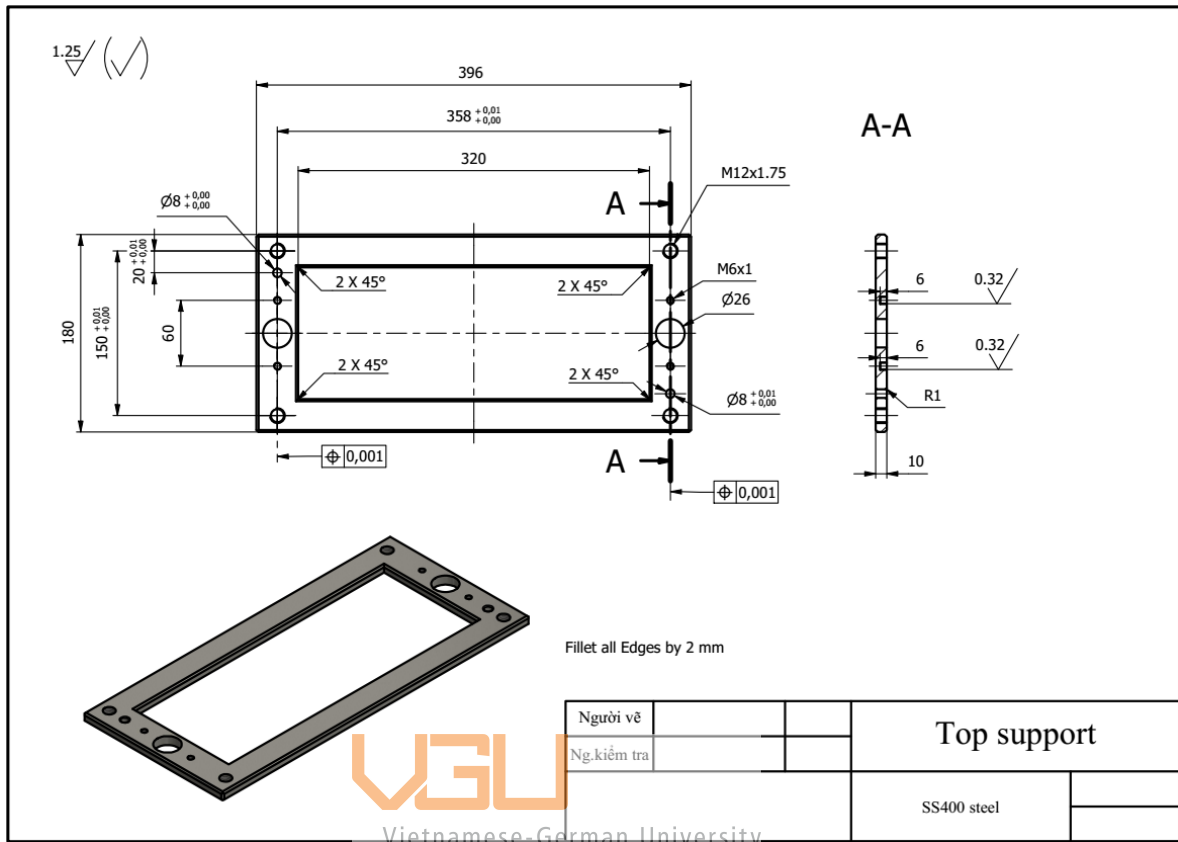


Figure 13: Top support

Function:

- To support body frames.
- To avoid body frames being deformed.
- To support pneumatic cylinders.

Material: JIS G3101 SS400 steel.

Volume: 303769.518 mm<sup>3</sup>

Mass: 2.3 kg

### 2.3.5 Rubber covered roller

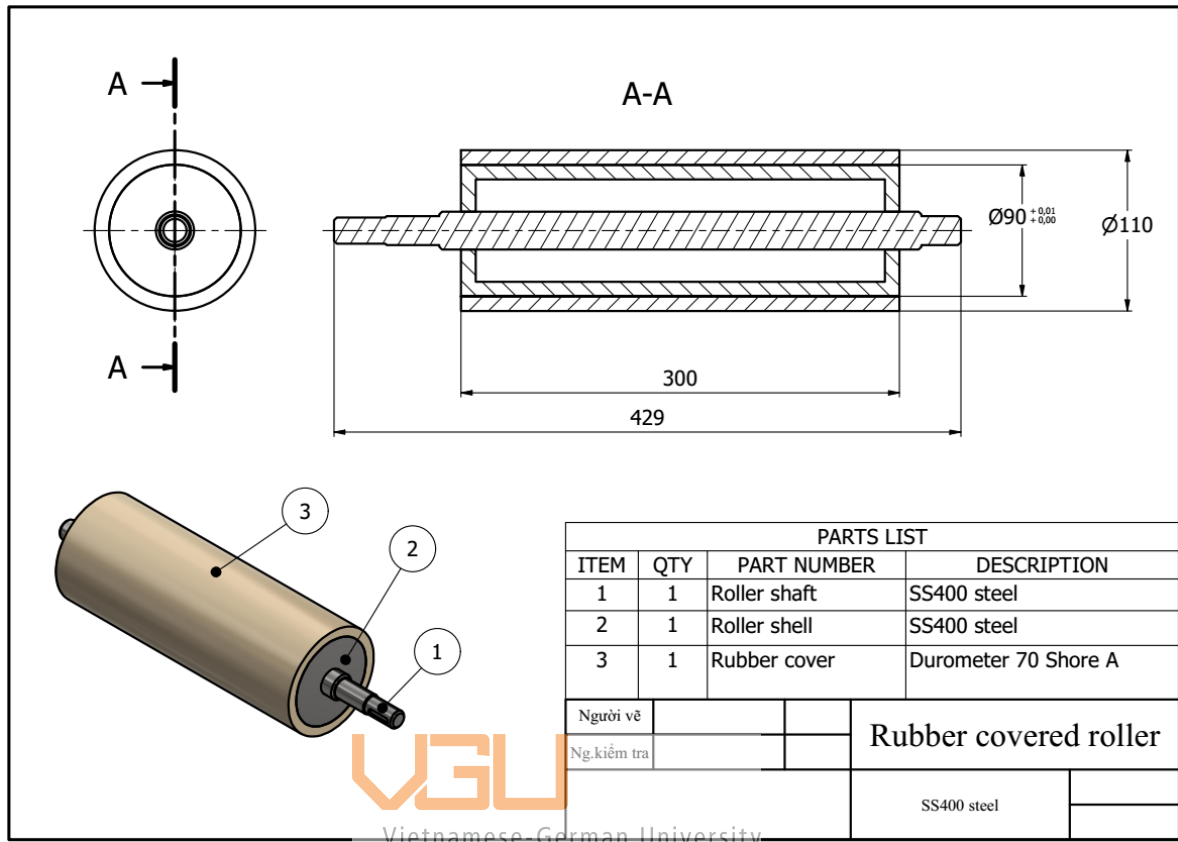


Figure 14: Rubber covered roller

Function:

- To drive the nip roller.
- To pull the web through the web transport process.
- To control tension on web.
- To control the line velocity of web.
- To reduce the deflection on rollers' shell.

Material: JIS G3101 SS400 steel.

Volume:

- Shaft:  $203324.123 \text{ mm}^3$
- Shell:  $820332.674 \text{ mm}^3$
- Rubber covering:  $942477.796 \text{ mm}^3$



Total mass: 8.6 kg

### 2.3.6 Idler roller

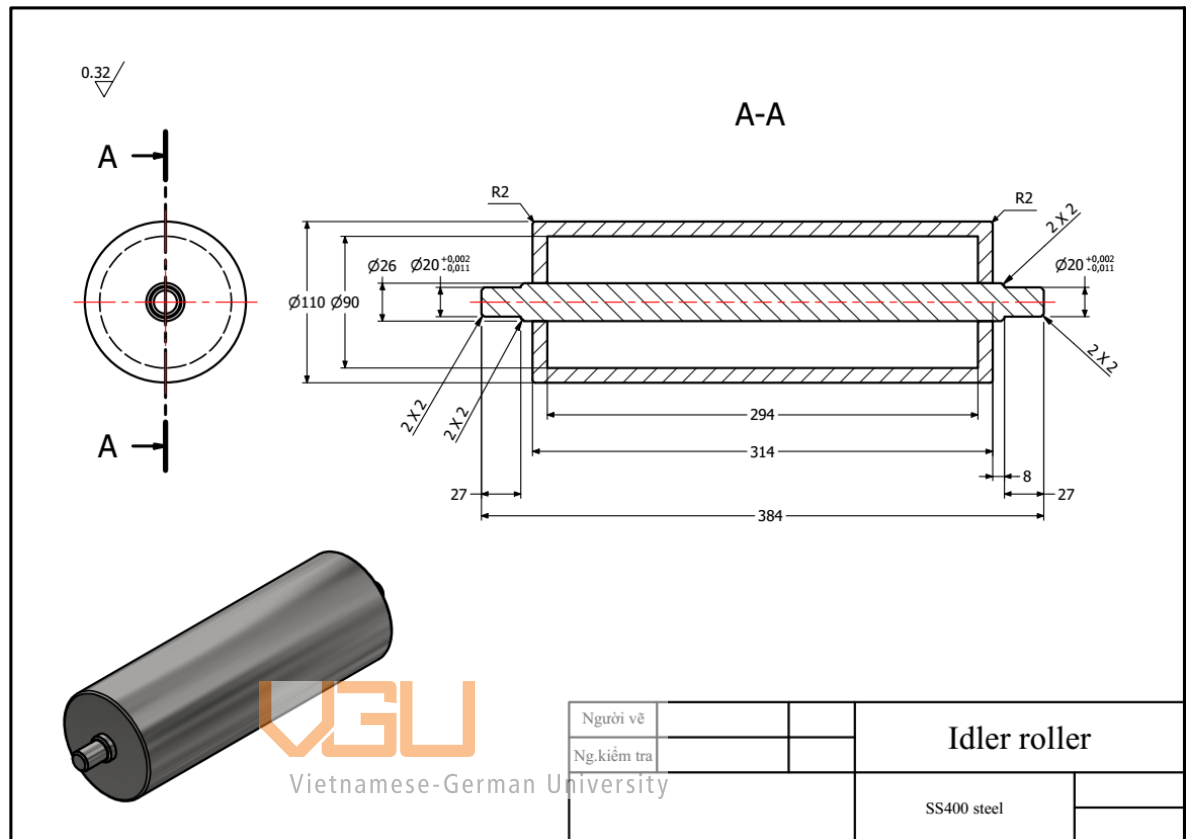


Figure 15: Idler roller

Function:

- To result in tension on web.
- To result in nip's contact pressure.

Material: JIS G3101 SS400 steel.

Volume: 1294114.223  $mm^3$ .

Mass: 10 kg

### 2.3.7 Bearings

The bearings used for the design are self-aligning ball bearings. They are sealed in order to avoid dusting. The reason why self-aligning ball bearings are selected for the design is because the bearings are insensitive to angular misalignment of the shaft relative to the housing (Figure 16), which can be caused, for example, by shaft deflection.

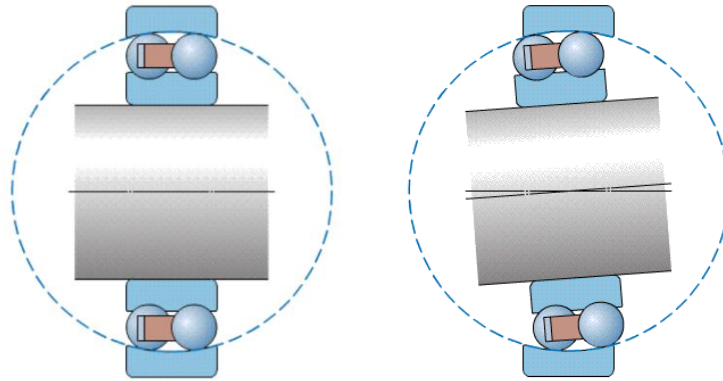


Figure 16: Angular misalignment in shaft

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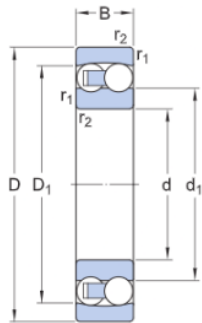
Figure 17: Self-Aligning ball bearing

Function: Allow the rubber covered roller to rotate around its axis.

Mass: 0.12 kg/ each.

Description:

Table 5: Ball bearing dimension and calculation data



DIMENSIONS

d	20 mm	Bore diameter
D	47 mm	Outside diameter
B	14 mm	Width
d1	≈28.88 mm	Shoulder diameter inner ring
D1	≈40 mm	Shoulder diameter outer ring
r1,2	min.1 mm	Chamfer dimension

CALCULATION DATA

Basic dynamic load rating	C	12.7 kN
Basic static load rating	$C_0$	3.4 kN
Fatigue load limit	$P_u$	0.176 kN
Reference speed		32 000 r/min
Limiting speed		20 000 r/min
Permissible angular misalignment	$\alpha$	2.5 °
Calculation factor	$k_f$	0.04
Calculation factor	e	0.3
Calculation factor	$Y_0$	2.2
Calculation factor	$Y_1$	2.1
Calculation factor	$Y_2$	3.3



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### 2.3.8 Bottom bearing housing

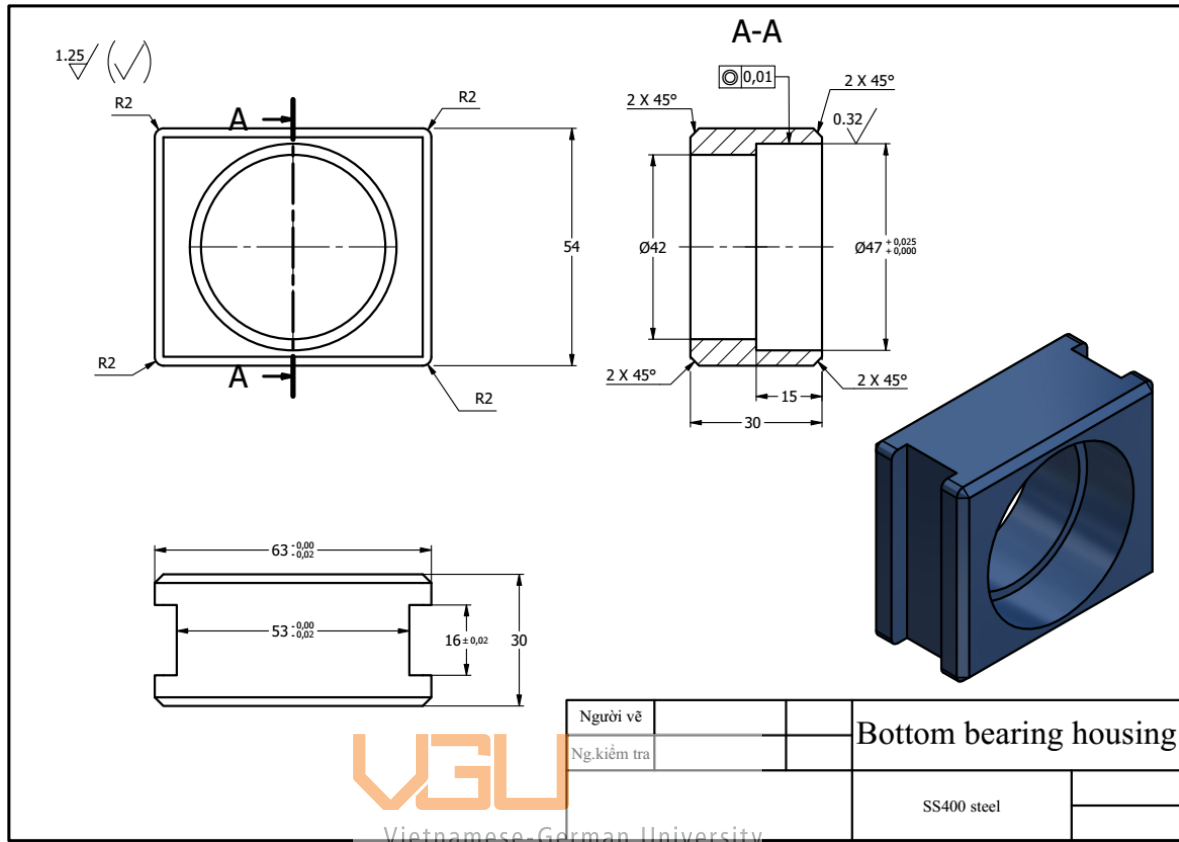


Figure 18: Bottom bearing housing

Function:

- To support bearings.
- To transmit parallel motion along body frames.

Material: JIS G3101 SS400 steel.

Volume: 45660.615  $mm^3$ .

Mass: 0.35 kg

### 2.3.9 Top bearing housing

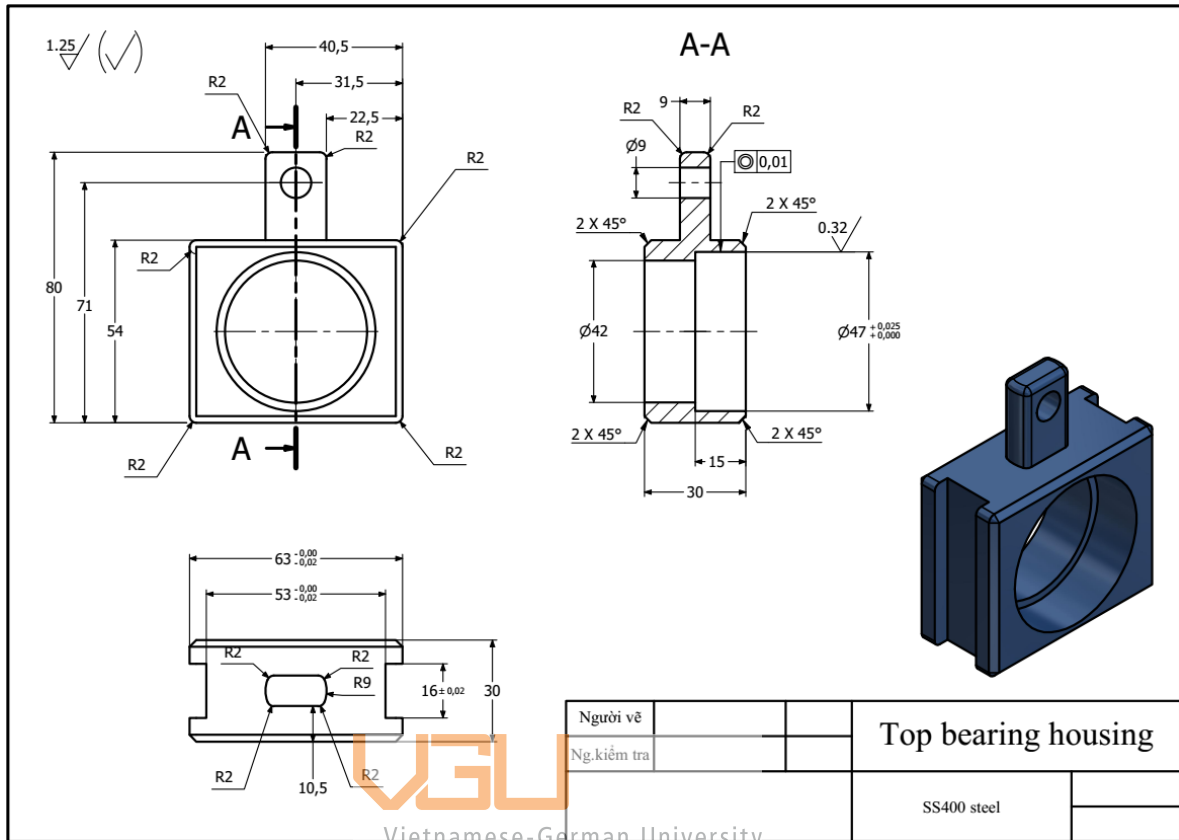


Figure 19: Top bearing housing

Function:

- To support bearings.
- To transmit parallel motion along body frames.

Material: JIS G3101 SS400 steel.

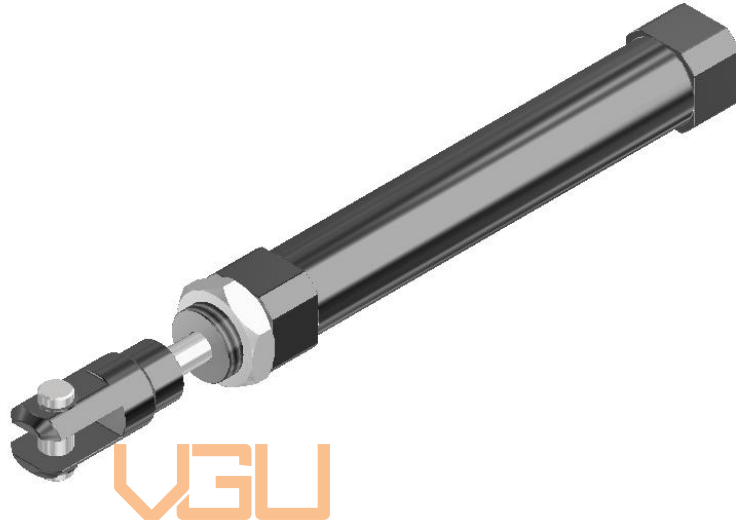
Volume: 49047.431  $mm^3$ .

Mass: 0.38 kg

### 2.3.10 Pneumatic cylinder

Function:

- To transmit contact forces to rollers.
- To push and pull idler roller moving up and down.



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Figure 20: Pneumatic cylinder

Description:

Table 6: Pneumatic cylinder's description

<b>Series</b>	CM2 (Standard type)
<b>Action</b>	Double acting
<b>Type</b>	Single rod
<b>Mounting</b>	Boss-cut/Rod flange
<b>Bore size</b>	20 mm
<b>Cylinder stroke</b>	100 mm
<b>Cushion</b>	Rubber
<b>Rod end bracket</b>	Double knuckle joint

### 2.3.11 Flange

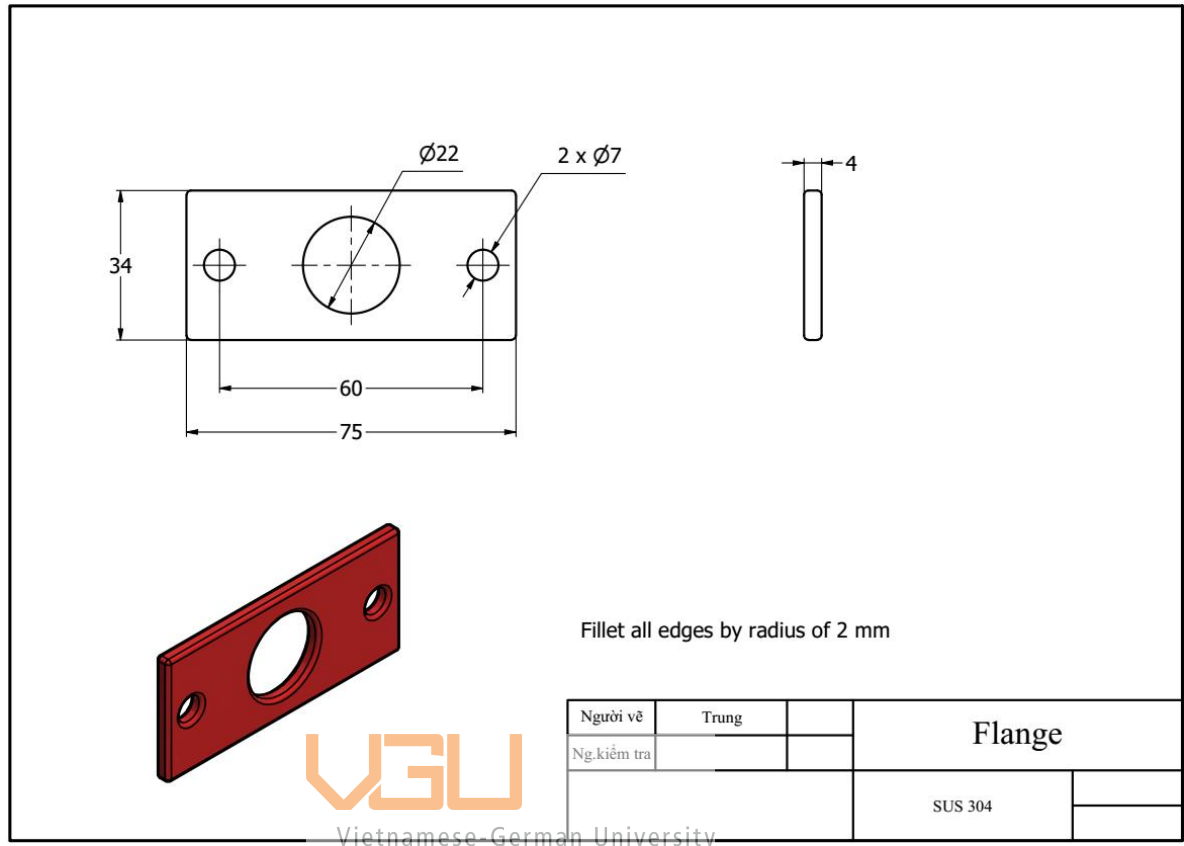


Figure 21: Flange

Function: to connect pneumatic cylinders with the top support.

Material: JIS G3101 SS400 steel.

Volume: 8225.583  $mm^3$ .

Mass: 0.066 kg

### 2.3.12 Motor mount

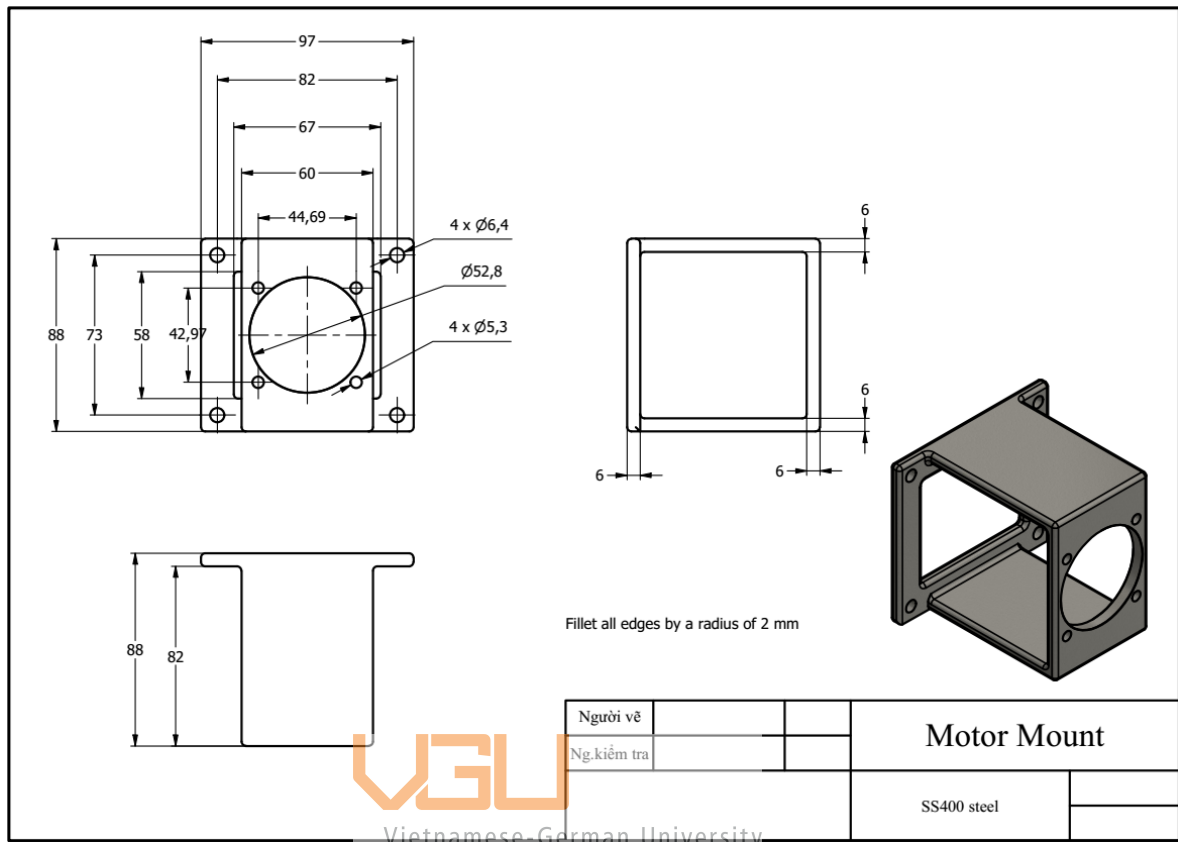


Figure 22: Motor mount

Function: To fix the servo motor to body frames.

Material: JIS G3101 SS400 steel.

Volume: 99383.247  $mm^3$ .

Mass: 0.77 kg



### 2.3.13 Locating pin

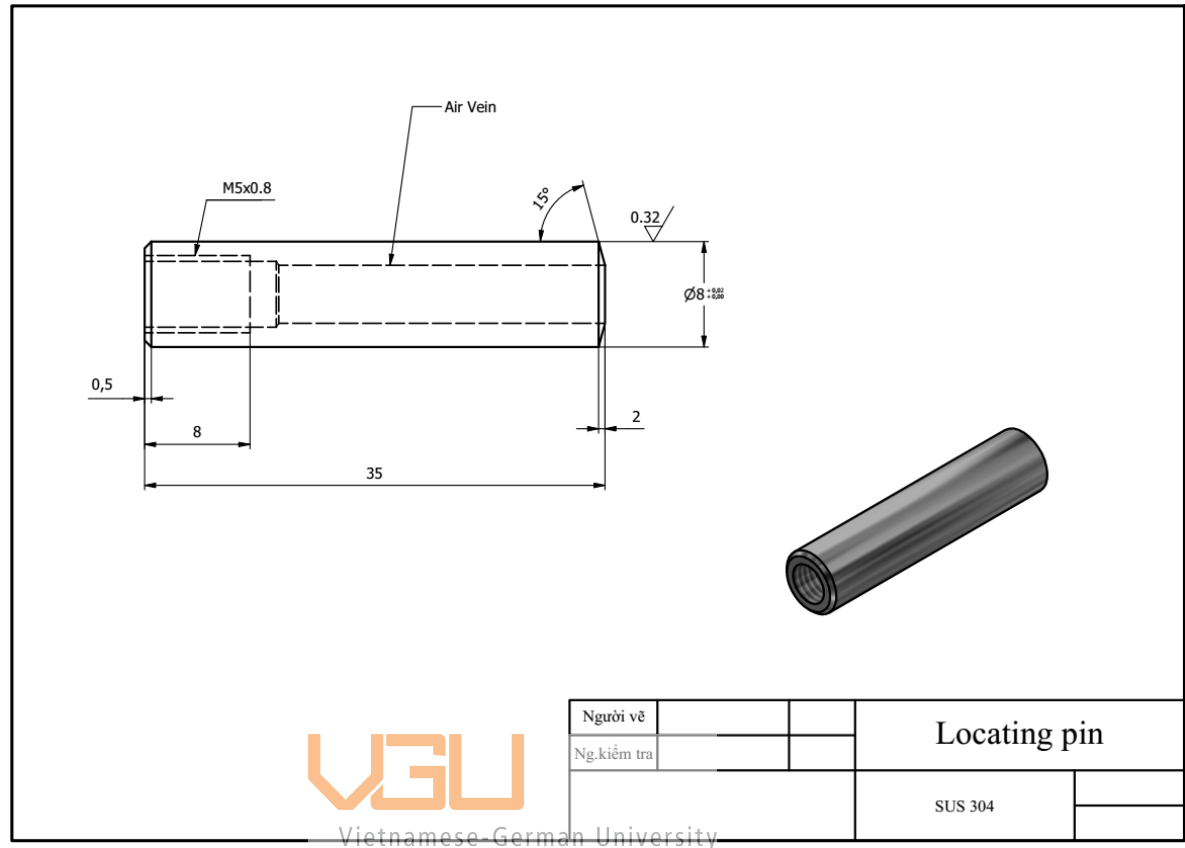


Figure 23: Locating pin

Function:

- To locate connecting's positions.
- To constrain motions of machine elements.

Material: SUS 304.

Volume: 1168.256  $mm^3$ .

Mass: 0.009 kg.

### 2.3.14 Motor coupling



Figure 24: Motor coupling

Function: Transmit power from servo motor to the roller's shaft.

Material: Cast iron.

Volume: 1168.256  $mm^3$ .

Mass: 0.008 kg.



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## 2.4 Mechanical Assembly

This section describes steps to assembly a prototype of this nip roller system. First, two body frames are located and fixed on bottom support at locating pins. shown in Figure 25, 26.

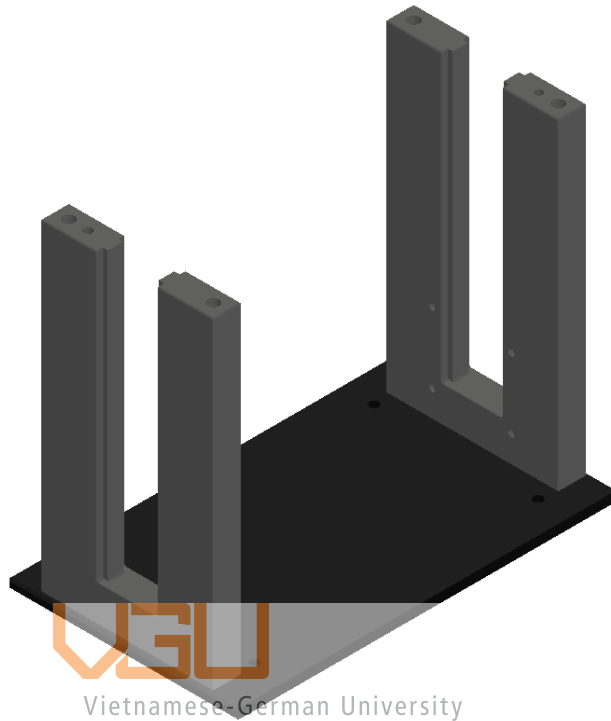


Figure 25: Body frames' installation

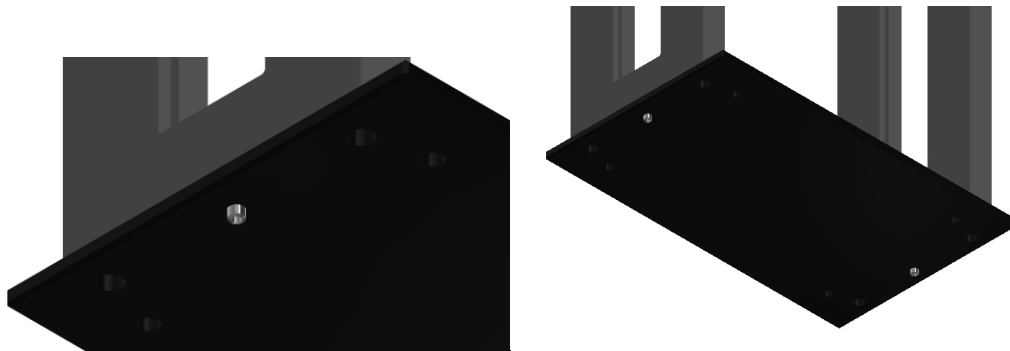


Figure 26: Body frames are located and fixed by locating pins

Bearings are attached to roller's shafts before inserting into bearing housings, shown in Figure 27, 28, 29, 30.

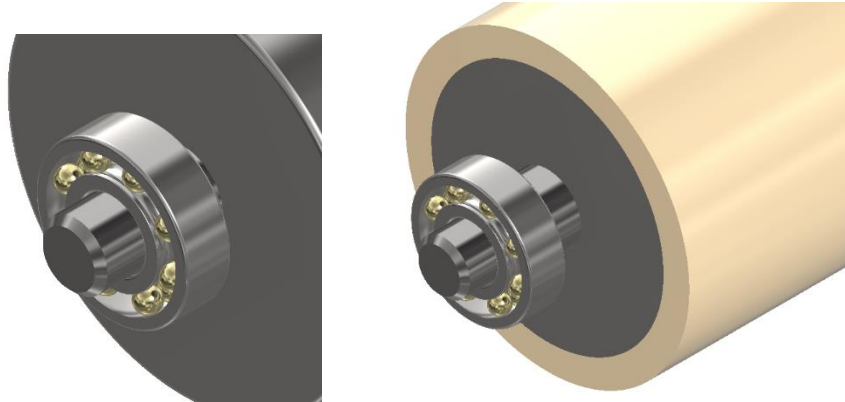


Figure 27: Bearings' installation



Figure 28: Bearings' installation (general view)

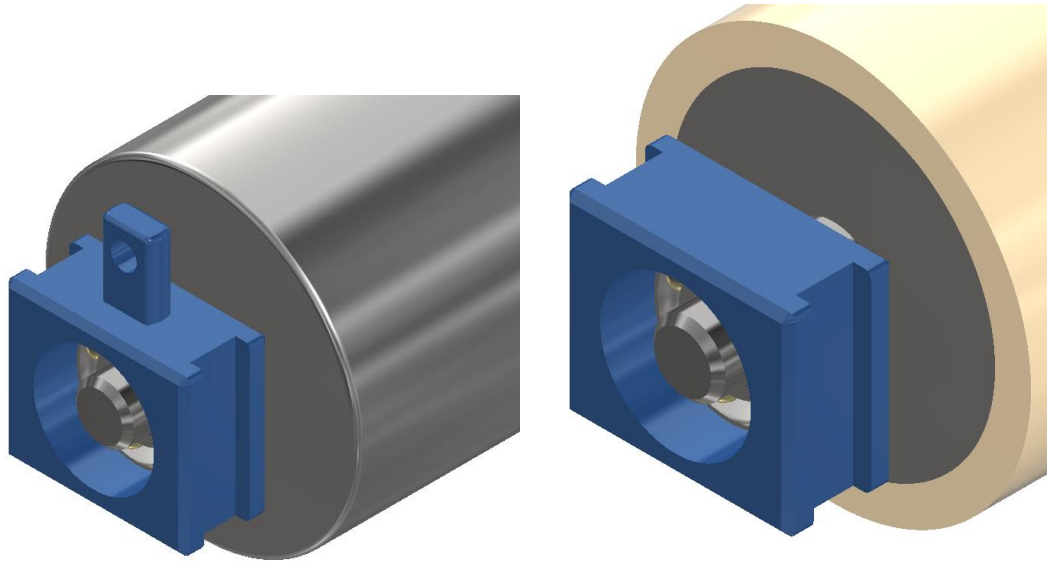


Figure 29: Bearing housings' installation



Figure 30: Bearing housings' installation (general view)

The combination of rollers, bearings and bearing housings are then inserted into the alley of body frames within where the rubber covered roller laying below the idler roller, shown in Figure 31.

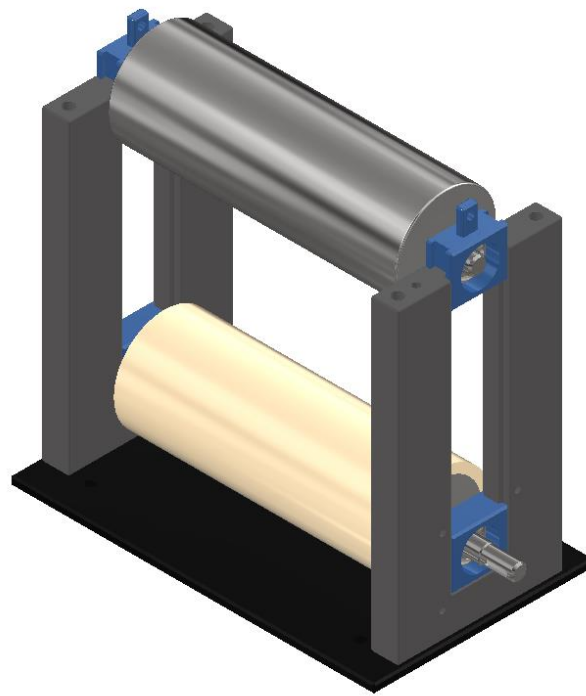


Figure 31: System after rollers' installation

After two rollers are inserted, a top support is then located and fixed on top of body frames at two locating pins' positions, shown in Figure 32.

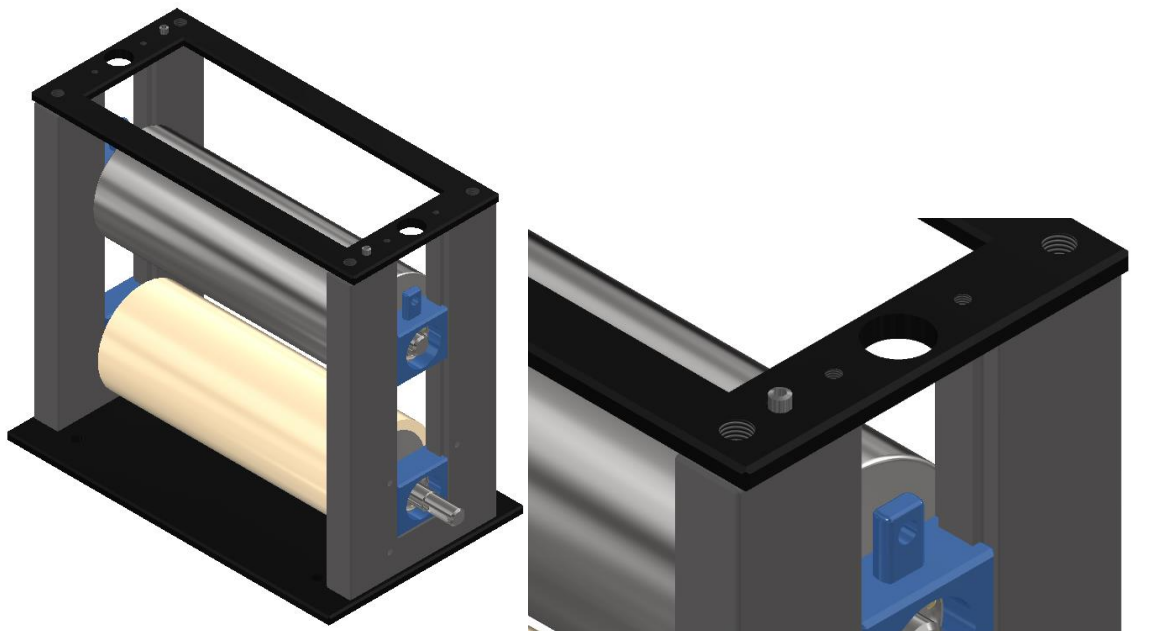


Figure 32: System after a top support's installation

Two pneumatic cylinders are attached to flanges before attaching on the top support, while double knuckle joints are connected at bearing housing's ears, shown by Figure 33, 34.

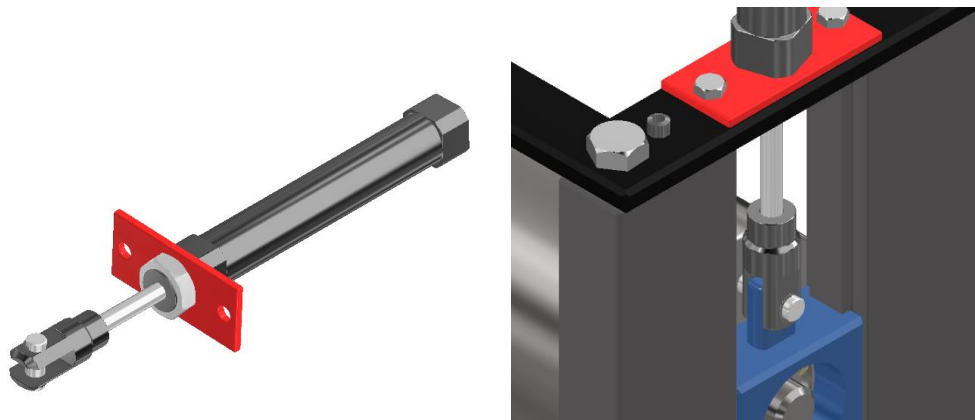


Figure 33: Pneumatic cylinders' installation



Figure 34: System after pneumatic cylinders' installation

Motor is then fixed to body frames via a motor mount. Between shafts of motor and the roller, a coupling is inserted to transmit power from motor to the shaft, shown by Figure 35, 36.

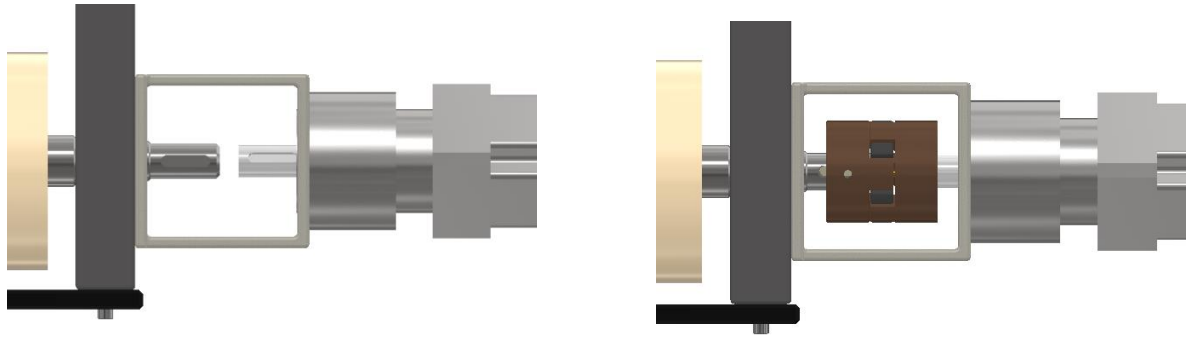


Figure 35: Before and after a motor coupling is installed

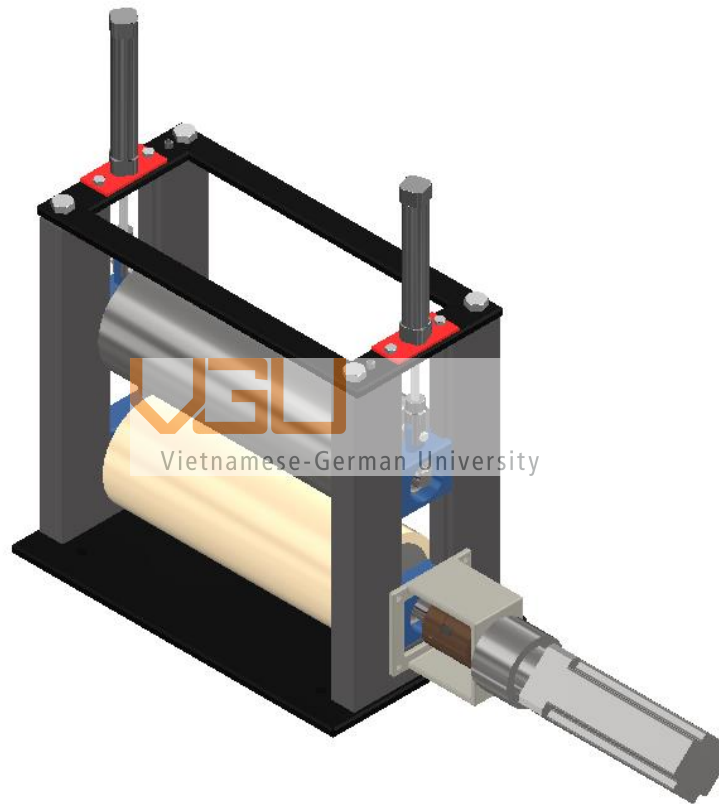
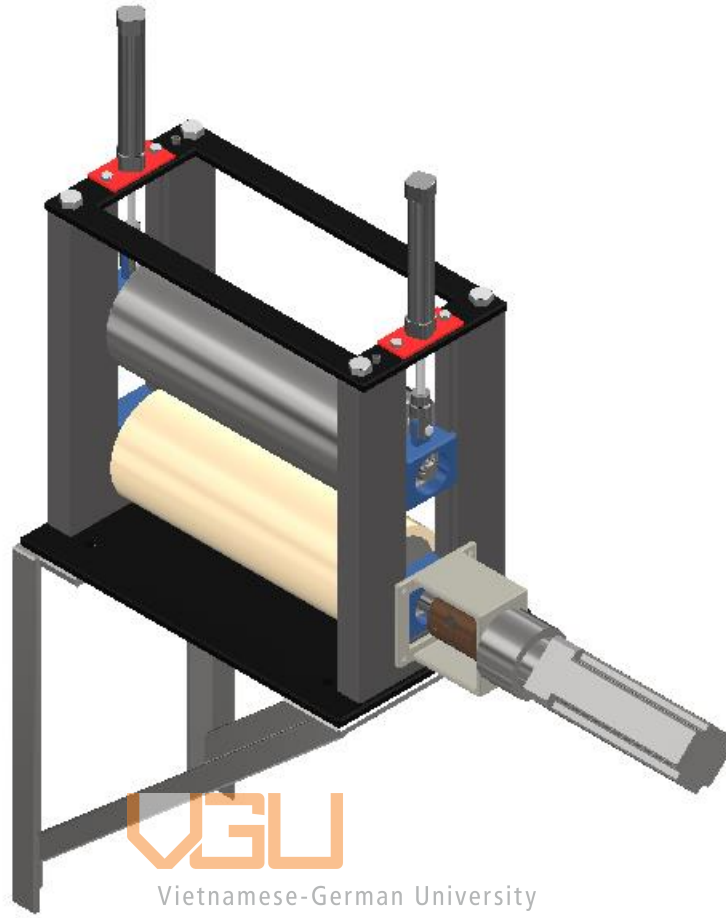


Figure 36: System after motor coupling's installation

During assembling parts to body frames, the system of trusses is installed at the web transport position. After assembling separately, the combination of frames and rollers is then fixed to the system of trusses after installing a motor, shown by Figure 37.





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Figure 37: Complete assembly

## 2.5 Cost Estimation of Mechanical Design

Table 7: Cost estimation of Mechanical Design

	Quantity	Cost per each (VND)	Total cost (VND)
Body Frame	2	1,250,000	2,500,000
Top support	1	420,000	420,000
Bottom support	1	350,000	350,000
Supporting trusses	1	1,100,000	1,100,000
Top bearing housing	2	850,000	1,700,000
Bottom bearing housing	2	700,000	1,400,000
Idler roller	1	1,480,000	1,480,000
Nip roller	Metal roller	1	1,400,000
	Rubber covering	1	400,000
Bearings	4	60,000	240,000
Pneumatic cylinders, Flange	2	700,000	1,400,000
Motor mount	1	800,000	800,000
Motor coupling	1	120,000	120,000
Total			13,310,000



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## CHAPTER 3: Electrical System Design

### 3.1 Overview of Electrical System Design

In this section, the objectives are: specifying criteria and explaining reasons of selecting motor and motor drive. Before designing, some requirements must be defined, such as the web's surface velocity, the required tension on the web, and the web's size. They have an impact on each other as well as the design's dimensions, therefore the most crucial aspect, the motor's power, must be constrained and prioritized.

### 3.2 Hardware components

#### 3.2.1 Servo Motor



Figure 38: Servo Motor

For the motor selection, there are two criteria considered to select an appropriate motor, they are: a suitable type of motor and its power. There are 7 most common types of electric motor used today, they are: AC Brushless Motors, DC Brushless Motors, DC Brushless Motors, Direct Drive, Linear Motors, Servo Motors and Stepper Motors. For the purpose of continuity, the web surface's velocity must be continuously controlled, which is able to be done by a servo motor. Servo motor is an electric motor, which is coupled to a sensor for position feedback. On displacement-controlled application, there is one of two types of sensor usually included in a servo. The specific sensor integrated in selected servo is an incremental encoder, which counts movement rather than position.

Motor's power will be transmitted to the live roll by a motor. It must be selected based on the required values of web's surface velocity, which ranges from 0.2  $m/s$  to 0.5  $m/s$ .

Web's surface velocity is proportional to the outer radius of the roller by the following expression:

$$V_2 = R_2\omega$$

The line velocity's values of the web are slightly small, so the moment of inertia of the roll will be proportionally small. For that reason, the servo motor's power for the low inertia is selected, which has a value of 3000 rpm.

The selected servo motor's information is described by the following table:

Table 8: Servo motor's description

<b>Product Name</b>	Electrical Commutation Motor
<b>Driving type</b>	AC Servo Motor
<b>Rated Input Power</b>	400W
<b>Rated Voltage/ Rated Speed</b>	220V/ 3000 rpm (used for low inertia)
<b>Encoder type</b>	Incremental encoder, 20-bit
<b>Motor Frame Size</b>	60 mm
<b>Keyway (with screw hole)</b>	Yes
<b>With Brake</b>	Yes
<b>With Oil Seal</b>	No
<b>Standard Shaft Diameter</b>	Yes

### 3.2.2 Servo Drive



Figure 39: Servo Drive

A servo drive is used to power electric servomechanisms (or servo). It does three things in sequence: receiving an input, producing an output, and then transmitting that output to a terminal. The input is frequently a time-varying voltage of current, the output is also a voltage or current with a higher amplitude, and the terminal, in this case, is the servo motor. The selection of drive is based on the Rated Input Power of the servo, which is 400 W.

The selected drive's information is described by the following table:

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Table 9: Servo drive's description

<b>Product Name</b>	AC Servo Drive
<b>Series</b>	A2
<b>Rated Output Power</b>	400 W
<b>Input Voltage and Phase</b>	220V 1-phase/ 3-phase
<b>Model Type</b>	Standard Model including Extension Port for Digital Input (CN7), Ether CAT, CAN pen, DMCNET, E-CAM

### 3.2.3 PLC



Figure 40: PLC

This is a brain of the system which controls tasks. It is equipped with different communication mechanisms such as Integral PROFINET interface and Point-to-point connection via communication modules. In this application, PROFINET interface permits communication with programming device (personal computer) and HMI device.

### 3.2.4 Power sources



Figure 41: 220VAC



Figure 42: 24VDC

There are two power sources used in this system, they are 220VAC (Volts of alternative current) and 24VDC (Volts of direct current). The 220VAC is used to generate power to the air compressor, personal computer and the servo drive, while the 24VDC supplies power to the solenoid valve and hub.

### 3.2.5 Solenoid valve



Figure 43: Solenoid valve

This valve is connected to an air compressor, a 24VDC and air ports of pneumatic cylinders. This valve has 5 ports including an air source port, two exhaust ports and two output ports. This device does two things: to supply air to pneumatic cylinders and to change air direction to compress and decompress the piston. Compressed air is directed through output ports into two different air chambers of the double-acting cylinders to move the piston back and forth. Compressed air supplied by the compressor goes into the air source port of the valve, and then directed to one of two output ports in different cases. In first case, when the solenoid is not energized, the compressed air is directed in a way that to expand the pneumatic cylinder. In the remaining case, when the solenoid is activated, the compressed air is directed to the other chamber of the cylinder to shorten the pneumatic cylinder. The solenoid receives commands from PLC to be either activated or not.

### 3.2.6 Other device and related components



Figure 44: Ethernet Switch (Hub)

Ethernet Switch an intermediate device which is powered by an 24VDC to connect PLC to terminal equipment which are a PC and HMI.



Figure 45: Personal computer

A personal computer or laptop allows users to program tasks for PLC.



Figure 46: HMI

HMI allows users to enter inputs to PLC.





Figure 47: Air compressor

Air is supplied from this air compressor to the directional control valve (solenoid valve).

### 3.3 Wiring diagram and Control Design



Figure 48: Wiring diagram

The servo drive includes three connections to three external devices, including a 220V power supply, a servo motor, and a PLC, as shown in Figure 48. Three interface connection gates can be seen in the front and top views of the servo drive in Figure 49. The Main Circuit Terminal is wired to a 220V power source, while CN1 is wired to the PLC by RS-232

Communication Cables and CN2 is wired to the servo motor via Encoder Cable.

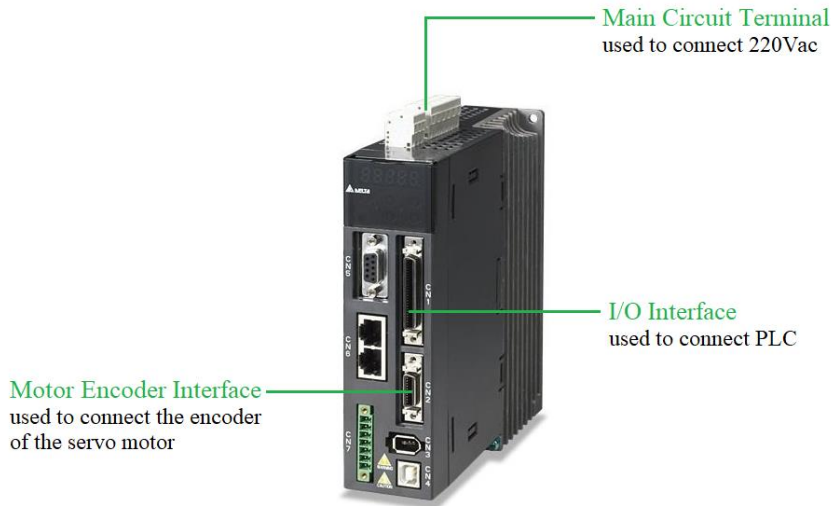


Figure 49: Servo Drive's front and top view

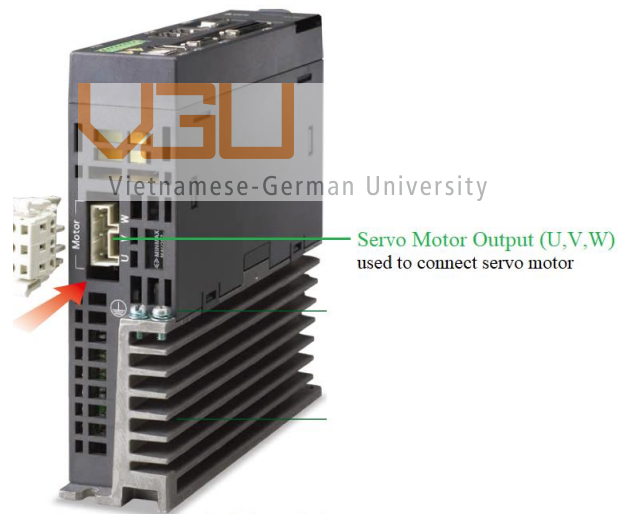


Figure 50: Servo Drive's bottom view

Profinet is a data communication protocol that is used to gather data from and operate equipment in industrial systems through a computer network. A powerful tool called TIA Portal is utilized as the programming software to develop a management and control over PLC using HMI. The following figures illustrate the interface and program syntax for this nip roll system.

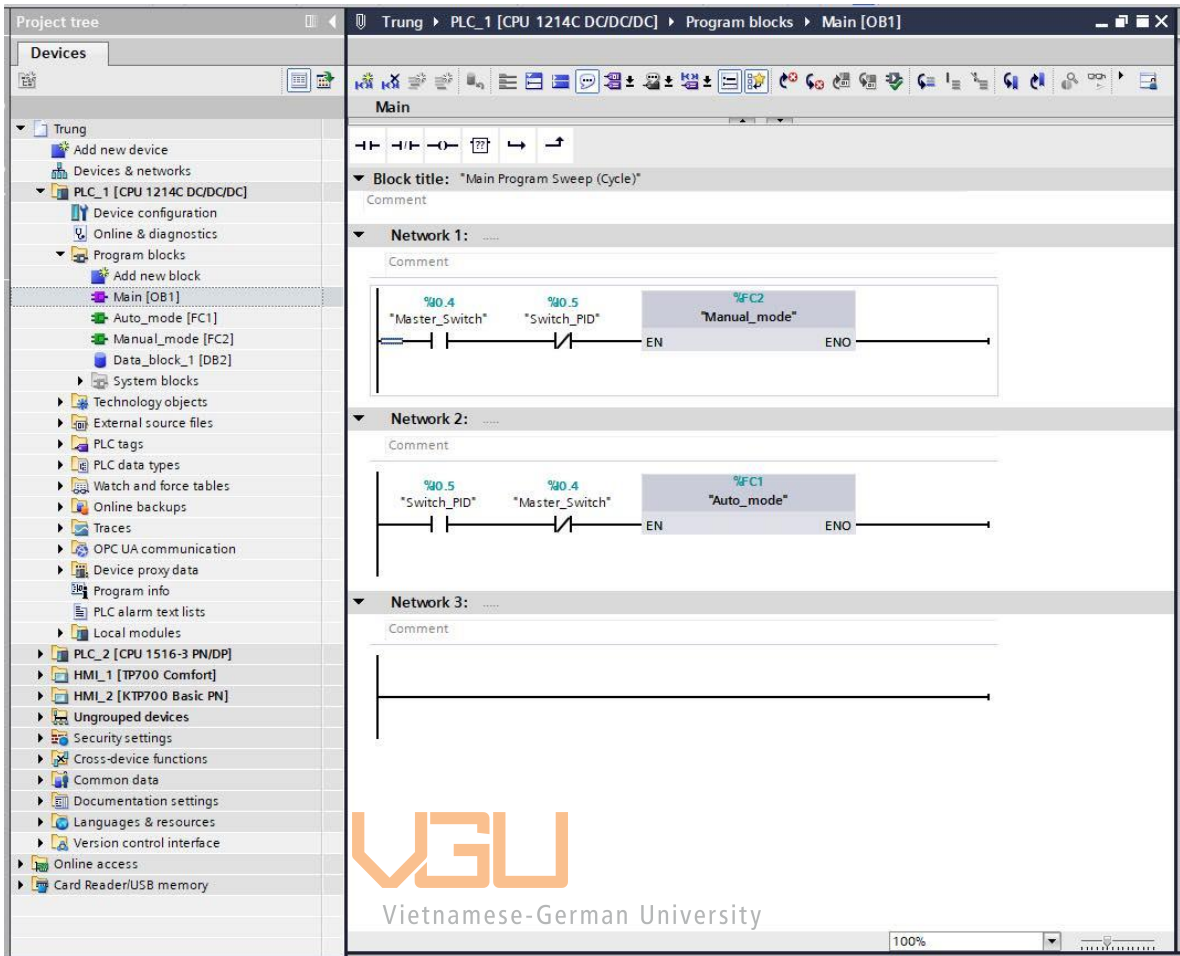


Figure 51: Main Network

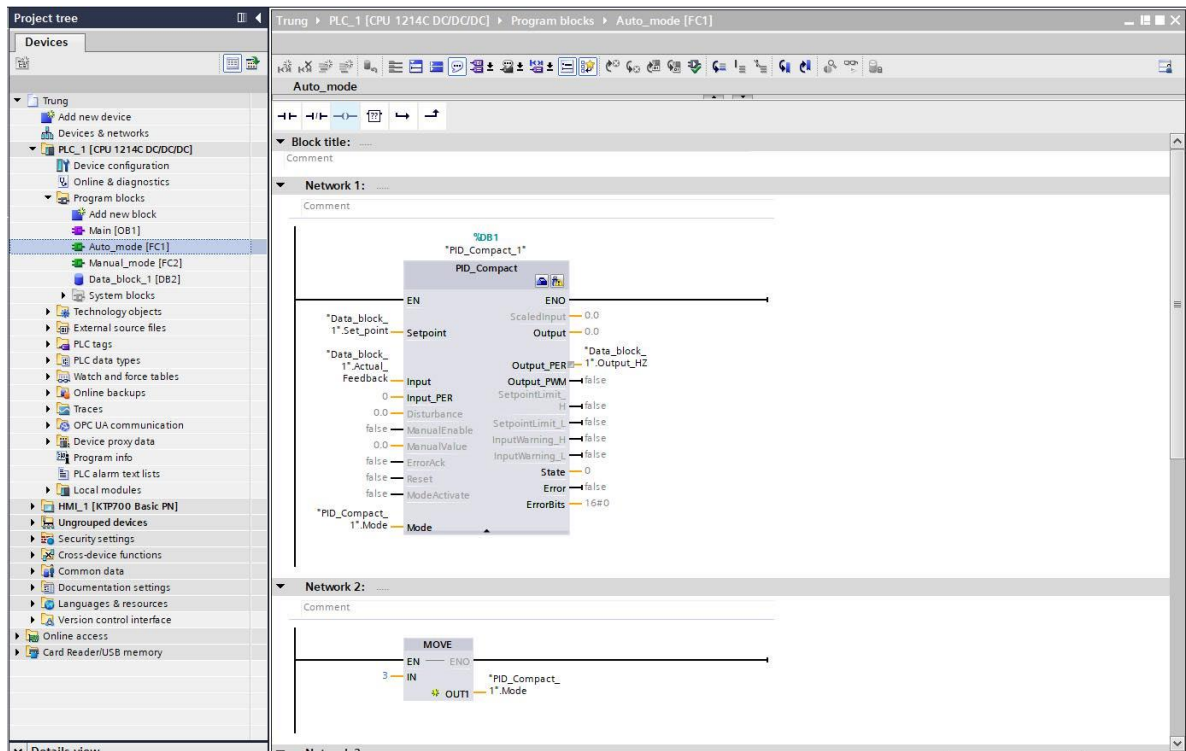


Figure 52: Work on auto function

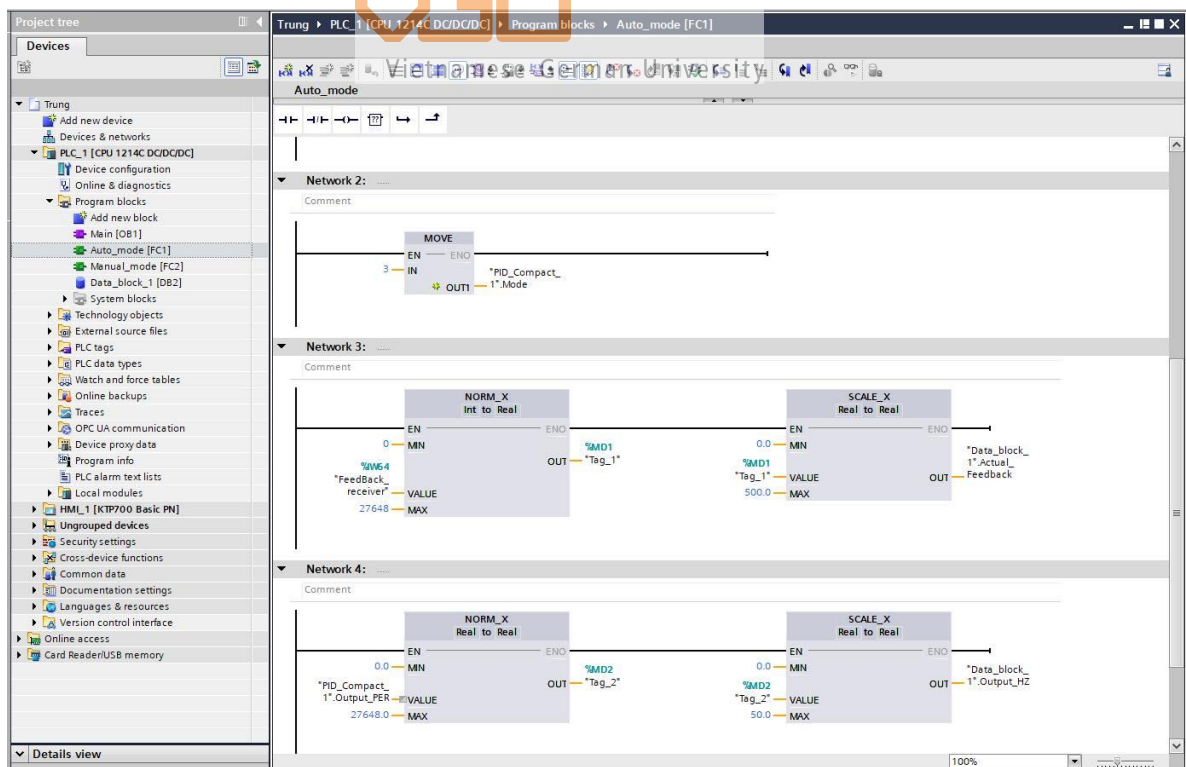


Figure 53: Work on auto function

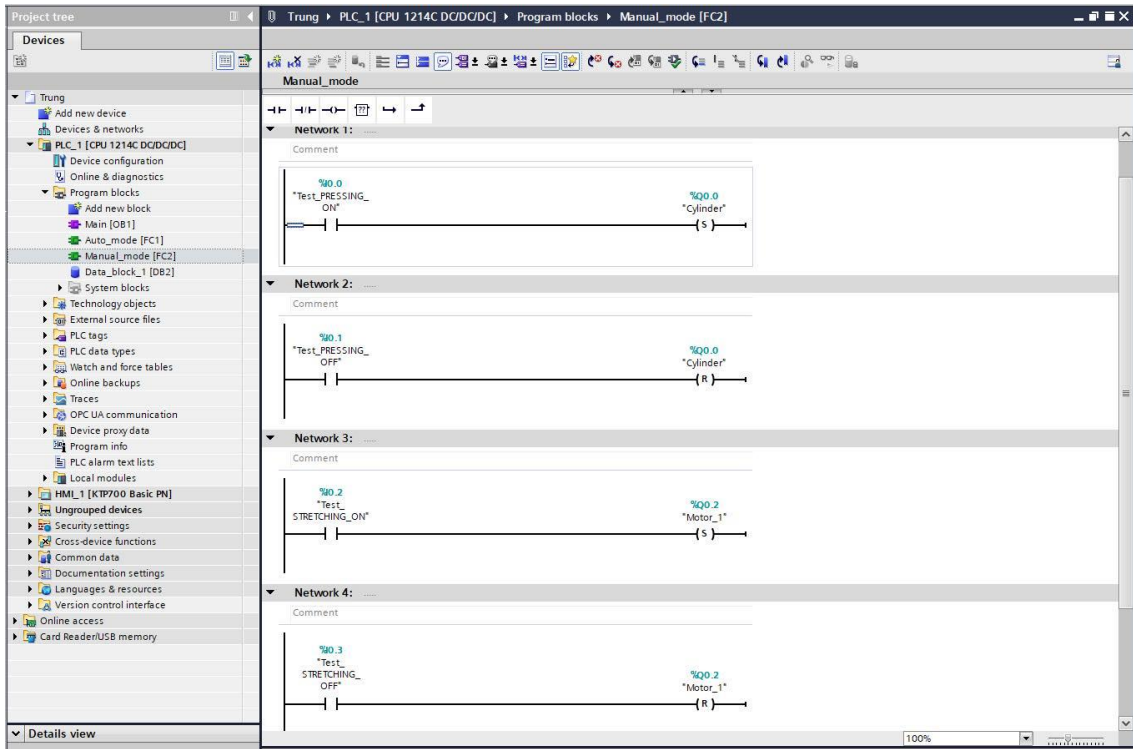


Figure 54: Test by manual function

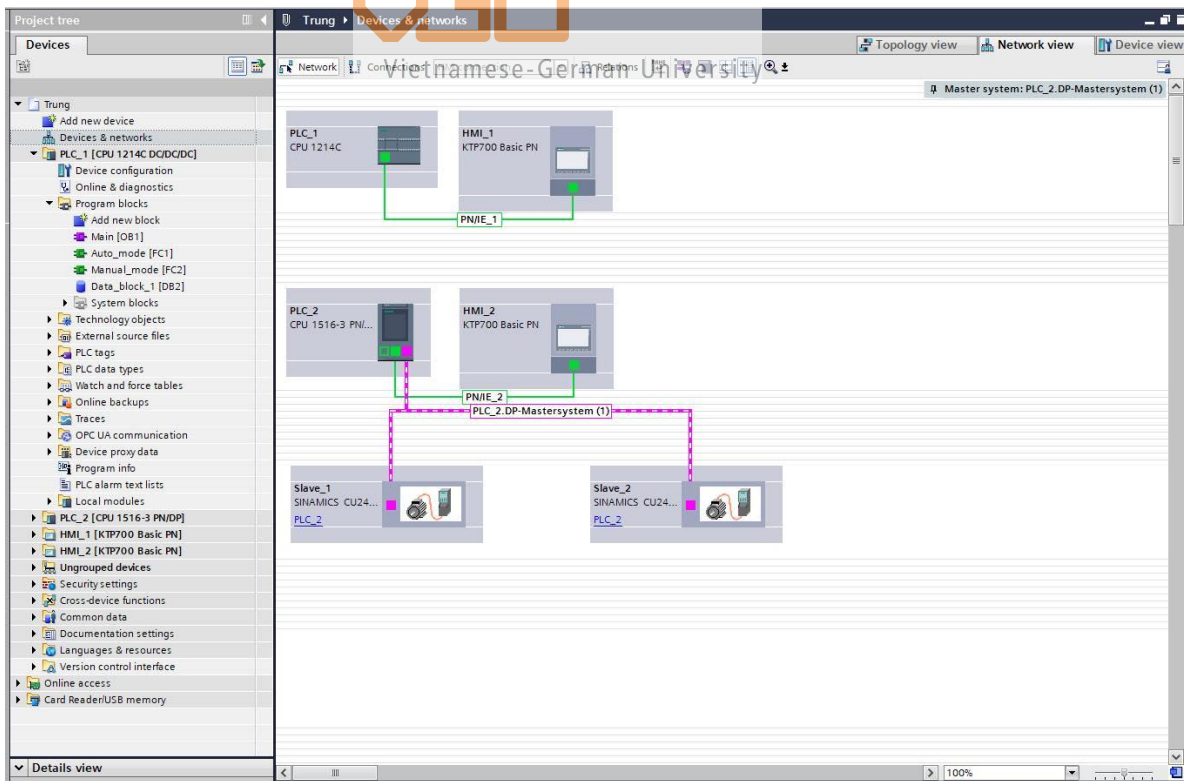


Figure 55: Device networks

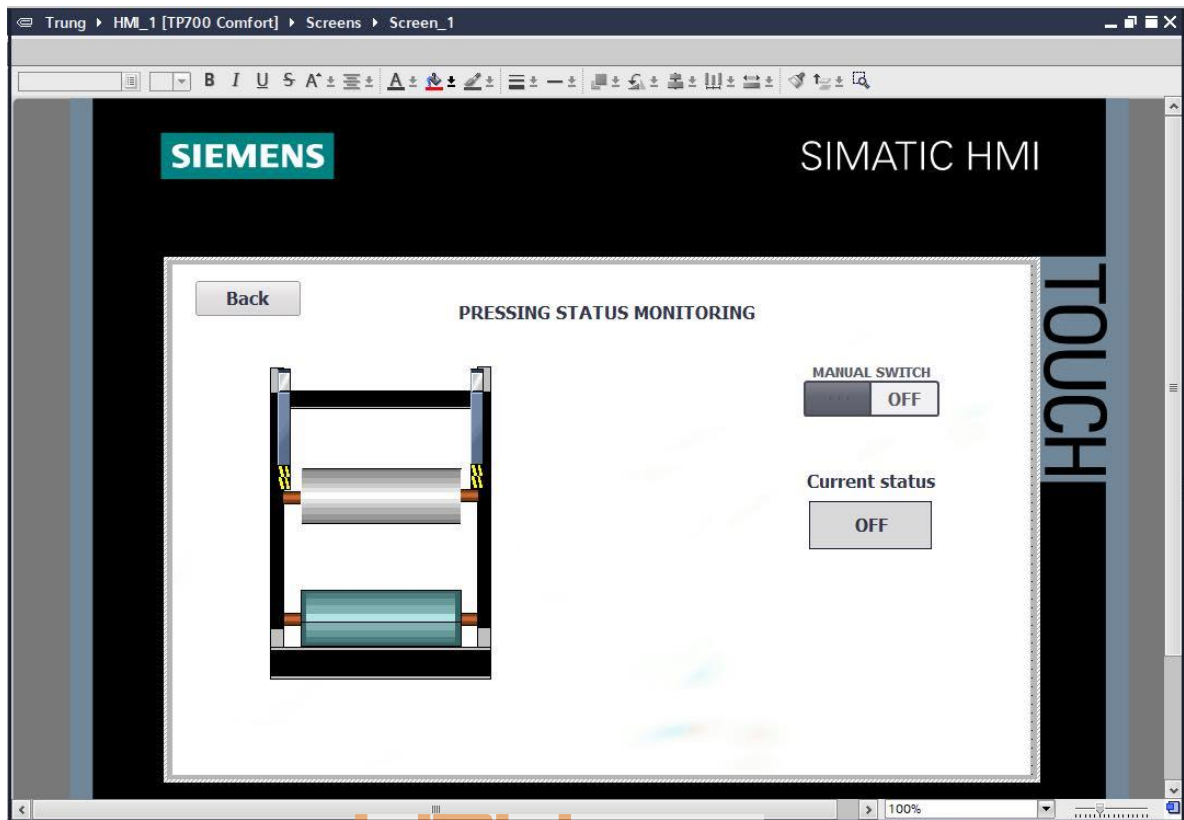


Figure 56: Status Monitor (switch OFF)  
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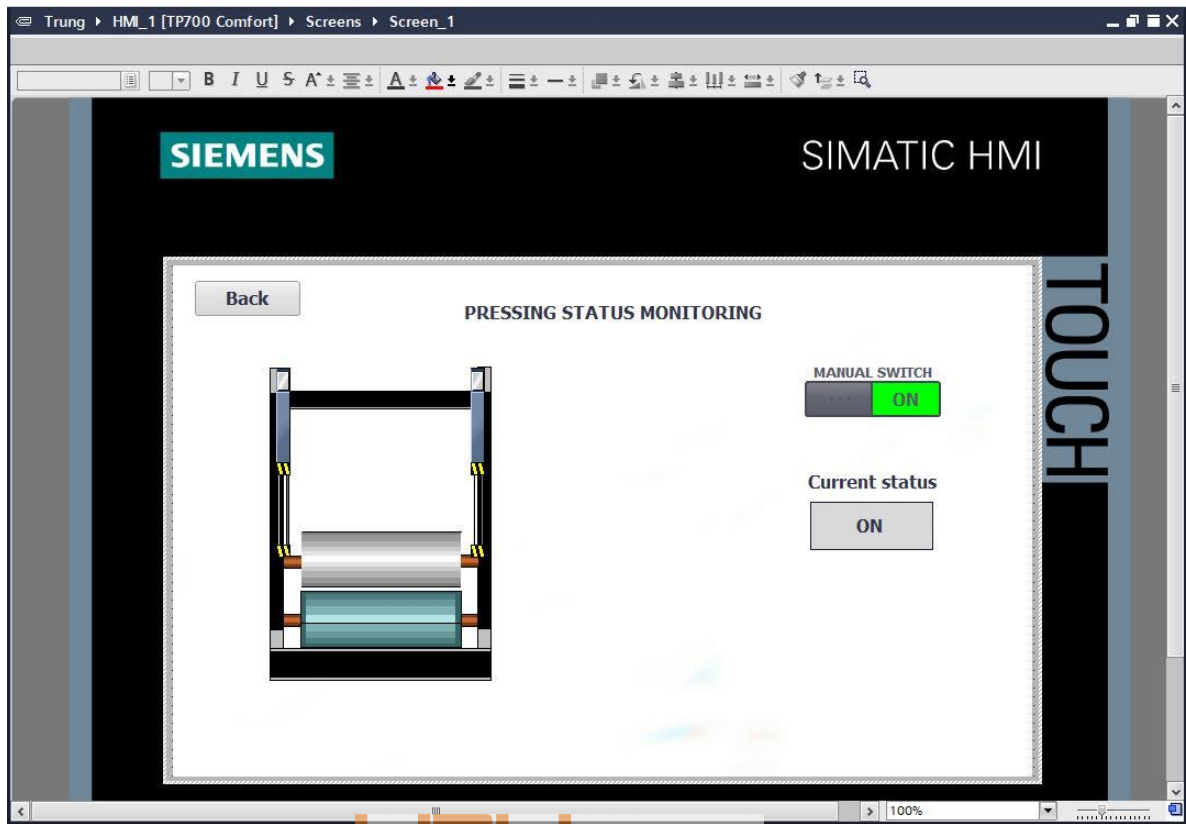


Figure 57: Status Monitor (switch ON)

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### 3.4. Prototype

In this section, the mechanical and electrical installation are presented. For the mechanical part, the prototype's sizes and dimensions is accurately identical with the CAD model. All movable parts such as bearings, bearing housings and pneumatic cylinders work normally and smoothly without getting stuck. The total mass of the prototype in reality is slightly larger than in the theoretical calculation due to variation of actual material density and to some additional parts such as bearing housing covers and attached components.

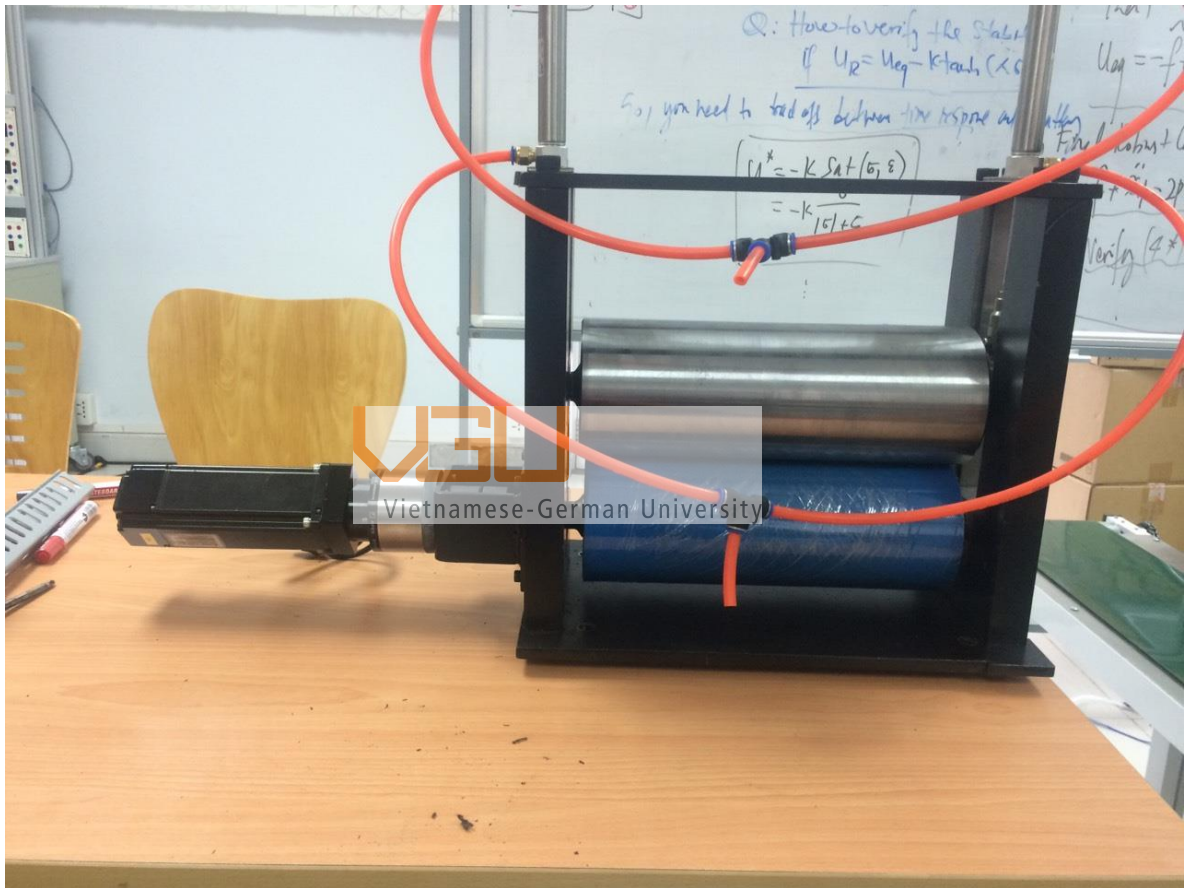


Figure 58: Prototype of mechanical part (front view)



Figure 59: Prototype of mechanical part (side view)



Figure 60: Prototype of mechanical part (side view)



Figure 61: Prototype's installation

For purpose of reliability, the specification of any product is needed, which provides necessary data for users and consumers to follow up in prior and during usage. For this prototype, the table of specification is presented below.

Table 10: Product specification

Data type	Feature		Property		
			Value	Unit	
Dimensions	Width		726	mm	
	Depth		240	mm	
	Height		531	mm	
	Weight		57.5	kg	
	Description	Shape	Rectangular		
Color		Black			
Physical parameter	Pneumatic cylinder		Maximum operating pressure	1.5	MPa
			Minimum operating pressure	0.5	MPa
			Stroke length	100	mm
			Bore size	28	mm
			Lubrication	Not required	
	Web		Width	25	mm
			Length	2000	mm
			Thickness	0.5	mm
			Material	PVC	
	Nip roller	Covering rubber	Hardness	70	Durometer (Shore A)
	Motor		Type	Servo	
			Rated Output Power	400	W
			RPM	3000	
	Ball bearings		Type	Self-aligning	
			Lubrication	Required	
			Seal	Yes	
	Operating line speed of web			0.2~0.5	m/s
Operating temperature			-10 ~ 60	°C	
Environment			Dry		
Tension in web (Net Force)			60	N	

## CHAPTER 4: Installation and Experiments

After designing mechanical elements and choosing electrical components, the system is then set up for experimental purpose to test its performance and liability. In this chapter, these following features will be clarified:

- Implementation of system setup
- Mechanism testing result
- Discussions

### 4.1 Implementation of System Setup

Before testing, some experimental tools and devices are installed. Those are a weighing meter, a load cell, and a web, shown in the following.



Figure 62: OMRON K3NV-LC1A-C2 Weighing Meter

The weighing meter receives electrical signal from a load cell and then converts it into human-readable values which are displayed directly on screen.



Figure 63: MAVIN Load Cell

Load cell is a transducer that is used to convert a force into electrical signal.

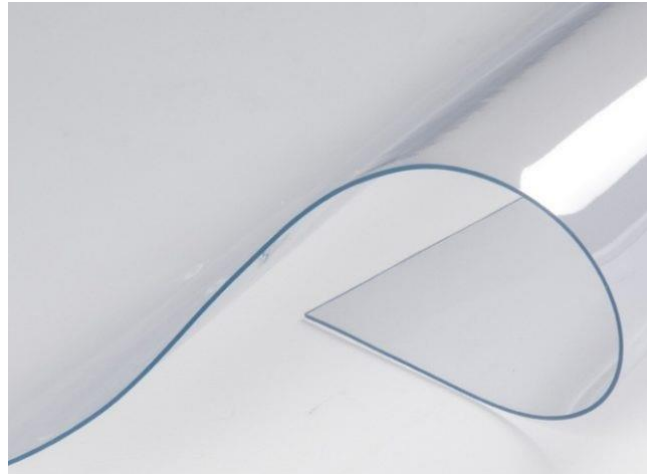


Figure 64: PVC web

To obtain the most reliable results, the web must fulfill these following criteria.

- The web's width has to be shorter than the roll width, which is 25 to 30 mm wide.
- The web's length has to be long enough for the holder be attached on, which is 2 m long.
- The web's elastic modulus must be small enough to avoid deformation under a required range of tension on web, for which PVC material is chosen as the suitable material.

Experiments were conducted in the Vietnamese-German University Automation Lab to execute experiments to test prototype's mechanism. For that purpose, a nip roller module duplicating the configuration shown in the Figure 2 was constructed to test the degree of slippage. A picture of the module is shown in Figure 52. The configuration consists of an idling upper uncovered roller and an lower roller with a rubber covering. Both are drive at a constant machine speed by means of a PVC web laying between the rollers. Nip force is controlled by two air cylinders placed on top of the frame. The test initializes with one end of the web being subjected to nip pressures in order to make a contact with two rollers. while the other end of the web is attached to a load cell. When the roller spins in the other direction, causing a reaction on the web. Early in the test, a marker is placed at the center point of contact to function as a reference for measuring the degree of slippage, shown in Figure 53.



Figure 65: Web and Rollers in contact





Figure 66: Marking on Web and Rubber Cover

The degree of slippage is determined by the degree of coincidence between two adjacent markings on both objects. Initially, a reference mark is made directly at the point where the web meets the rubber roller. The web and the roller experience a displacement on surface itself after a given time interval of rotation produced by the motor. The rotation is then delayed, allowing the next point of contact between them to be marked. To gather experimental data, the next marks are collected in the same ways.

#### 4.2 Mechanism testing result and discussion

The synchronization of bearing housing motions is achieved by the sliding mechanism. The idler roller completes all revolutions smoothly and without becoming stuck or tilted. In terms of the rotating mechanism, both rollers revolve smoothly without causing vibration in their axes, which met the criteria. For the slippage test, the collected data from experiment is illustrated by the following table.

Table 11: Slippage test data

<b>Tension on web</b>	<b>Slipping</b>
30N	No
40N	No
50N	No
60N	No

The data from the table shows that slippage did not present in system under given range of tension.

## CHAPTER 5: Conclusion and Future Work

### 5.1 Conclusion

In conclusion, the thesis project's conceptualizing, designing, processing, assembling, and testing phases have been accomplished. Through experimental tests, all mechanisms inside the system fulfill the given requirements. The mechanical framework of the prototype will now be adequately attached to the Roll-to-Roll system, which will deliver web material throughout the process. As a result of the above mechanical design, overhaul and maintenance are easier and convenient to do.

### 5.2 Future work

The mechanical design generally works and follows design process. However, to produce a perfect work, there are competing design factors which are needed to be considered.

The total mass of the mechanical prototype is quite heavy which could lead to difficulties in maintenance and early installation. The material for frames can be re-selected based on environment, budget, required strength and durability of the design.

The life span of the covering decided by the rubber layer need to be considered based on different applications. Hard rubber coverings last longer, but are less compliant (Reference [1]). In addition, decrease in diameters leads to increase in pressure on roller's shell, which causes bending and deflection under loading by time. The bending of roller's shell is inevitable in nip roll system, which can be optimized by changing the structure of the roller, thickness of both roller's shell and rubber covering (Reference [1], [6]).

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