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Design of Pin on Disc Tribometer Test Equipment Using the Hatamura Method

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ABSTRACT

This study purpose is to design a pin on disc tribometer with circular motion and reciprocating linear motion using the Hatamura method and its main components, analyze the shaft and frame, and produce working papers in the form of technical drawings and bill of materials in the manufacture of test equipment. In this study, designing several concepts of the pin on disc tribometer according to the Hatamura method and then evaluating the existing concepts and selecting concepts based on the decision matrix. Then, the concepts were designed and analyzed by manual calculations and validation using Autodesk Inventor to determine stress, deflection, as well as safety factors on the axle and frame. Based on the results of manual calculations on the shaft, the value of the safety factor was 3.234, while in the simulation using the Inventor the safety factor value of 3.4. According to the safety factor standard in this design, it can be said to be safe because the safety factor value was obtained greater than 3. The deflection value was 0.015 m and the maximum stress of 2.95 Mpa. The final result of this research was a technical document in the form of layout drawings and assembly drawings as well as detailed drawings of each component and bill of materials.

KEYWORDS: Tribometer pin on disc, Hatamura, Safety factor, Autodesk inventor.

NOMENCLATURE

μ	Coefficient of Friction
F_n	Force
Q	Wear Volume
SRR	Slide to Ratio
K	Wear
k	Constant
N	Normal Force
Fs	Friction
μs	Sliding Speed

1.0 INTRODUCTION

The friction can causes engine elements to be damaged and wear out. The definition of elements in paper that were in contact with each other. Wear is the gradual loss of material from the surface of the object in contact as a result of contact with a solid object, liquid object or gas on the surface. The wear caused a reduction in the dimensions of the engine elements [1]. To find out or measure the level of wear on the engine components that are in contact, there are several ways. One of them is with a wear test tool called a tribometer. The tribometer is a tool used to determine the wear caused by the friction of a material between two contacting surfaces [2]. The tribometer is designed with a pin on disc type, where friction between the disc and the pins. The pin is stationary and the disc rotates at a certain speed, causing friction [2]. With friction, the wear and coefficient of friction can be known.

Design is an activity in the early stages of a series in the product manufacturing process. In the design stage, important decisions are made that affect other activities, so that before a product is made, a design process is carried out, which will produce a technical drawing of the product to be made [3]. The design of this pin on disc type tribometer test tool uses design methodology Hatamura. Design methodology according to Hatamura is the steps to formulated in carrying out the design process. The steps in making the design stage are determining user requirements, making specifications, determining the functional level, conceptual level and component level [4].

The design of this pin on disc tribometer test instrument is different from other studies. The difference lies in the movement of the test and the transmission. Previous research only used one movement in the test. In the design of this test equipment, two movements are used, namely circular motion and reciprocating linear motion [5]. For circular motion using

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pins and discs. The pin rubs against a flat plate that forms a disc. As for the linear motion back and forth (reciprocating) using a plate holder. The plate holder is combined with the disc, so that back and forth motion occurs [5].

Before manufacturing, it is necessary to have a design that must comply with existing standards [6]. Therefore, in this study is designed a pin on disc tribometer test instrument with circular and alternating movements according to the Hatamura method.The final result of this research is in the form of technical documents such as layout drawings and the arrangement of components (assembly) of test equipment and detailed drawings of each element as well as a list of materials to be used (bill of material).

2.0 METHOD

The method in this research consisted of several steps, which was taken by researchers in order to gather information. The research method provided an overview of the research design, which included the procedures and steps that must be taken, research time, data sources, and problem solving methods.

2.1 Hatamura Method

One of the design methodologies commonly used in the design process is the Hatamura design methodology. The Hatamura design methodology can be seen in Figure 1. The steps of function-based design is depicted in Figure 2.

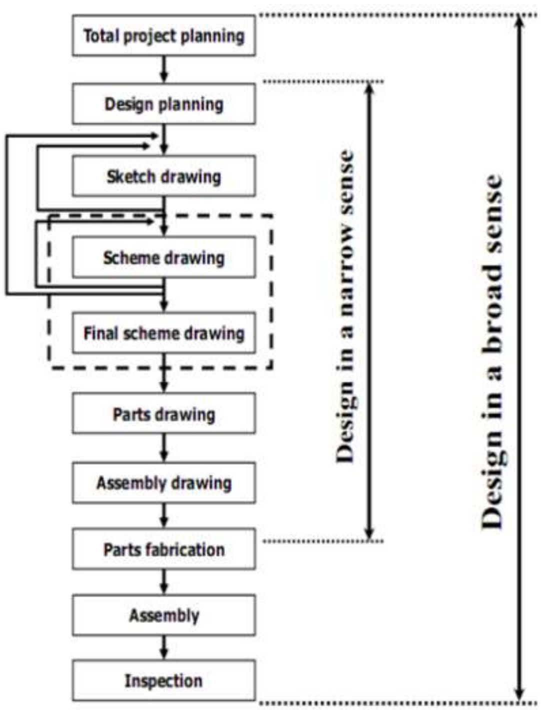

Figure : Hatamura Design Methodology [7]

In Figure 2, according to Hatamura design [7], the first step in the design stage is to determine user needs, then make specifications that will be able to meet those needs, then determine the functional level, namely the level where the things needed and the form of the system are generally represented without mentioning the realization of the concept or the physical.

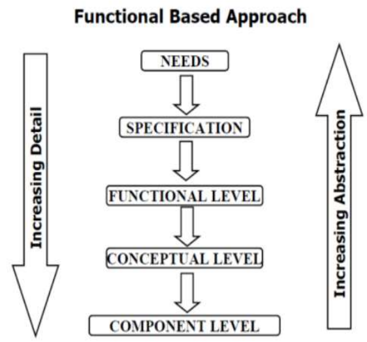

Figure 2: Stages of making a Hatamura design [7]

Then move on to the conceptual level, which is the level where the system has been modeled based on the principles of possible solutions to meet the design requirements. The last step is the component level, which is the level where the system has been modeled with specific physical properties that can realize the concept and design requirements.

2.2 Research Method

In this research was used an experimental method. The flow chart in this study can be seen in Figure 3.

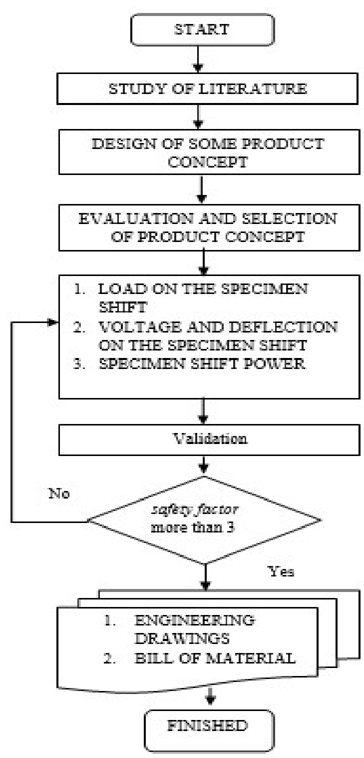

Figure 3: Research flow diagram

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2.3 Design According to the Hatamura Method

The design stages using the Hatamura method were as follows:

1. Determine design needs. The pin-on-disc tribometer test equipment was designed, which can measure the wear rate and coefficient of friction with a pin with a spherical surface area at rest of rubs against a flat plate-shaped disc while rotating [8]. The requiring criteria of design were depicted in Table 1.
2. Determine functional requirements. The function diagram of the pin on disc tribometer test tool was determined from the design requirements listed in Table 1. The resulting output was a specimen that can measure the wear rate and coefficient of friction. In this test, the input was given the electric current and test material, which can be seen in Figure 4.

Table 1: Design requirements [8]

No.	Criteria
1	Shaft rotation can be adjusted
2	Disc rotation can be controlled
3	Specimen size/dimension flexibility
5	Press load setting
6	Torque load setting
7	Anticipate shaft bending
8	Anti-slip
9	Specimen replacement speed
10	Low manufacturing cost
11	Ergonomic
12	Compact and practical tool dimensions
13	Easy to make
14	Easy to maintain

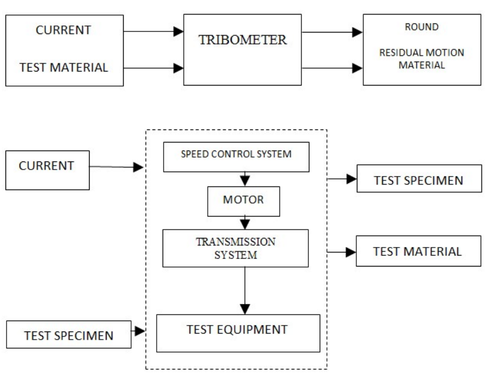

Figure 4: Function structure of the pin on disc tribometer test tool [7]

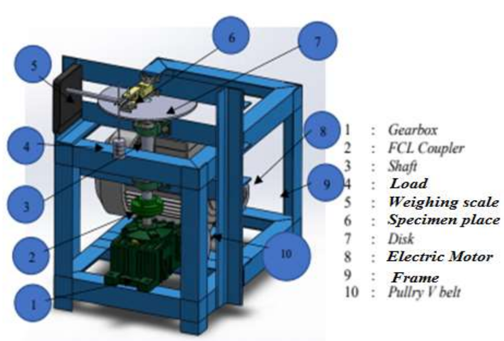

Figure 5: Concept I [9]

3. Selection mechanism. The mechanism was determined from the function diagram and then selected from the catalog of existing mechanisms. The mechanism of the pin on disc tribometer test tool was as follows [7]:
   - The rotation of the specimen shall be adjustable so that the effect of the Slide Roll Ratio (SRR) on the test can be observed.
   - Specimens can be replaced in a relatively short time.
   - Specimen dimensions (diameter and thickness) may vary.
   - Ergonomic.

2.4 Selection of the Concept of Test and Evaluation Tools

The criteria of several test instrument concepts were depicted in Table 2.

Table 2: The morphology of the test instrument concept

Criteria	Concept 1	Concept 2	Concept 3
Power Source Drive	Motor	Motor	Motor
Transmission	Pulley and Belt	Pulley and Belt	Pulley and Belt
Transmission Type	Gearbox WPO	Pulley	Pulley
Coupler	FCL Coupler	Coupler	Coupler
Disc	Available	Available	Available
Frame Feet	Steel profile L	Steel Square	Steel Square
Supporting Framework	Plate	Plate	Plate
Shaft Support	Bearing	Bearing	Bearing
Frame Splicing	Welding	Welding	Welding
Disc Position	Horizontal mount	Horizontal mount	Horizontal mount
Load Lever	Soft Breaker elevator	Square iron	Square iron
Load Laying	Bolted	Bolted	Bolted
Movement	Back-forth	Back-forth	Back-forth and circle

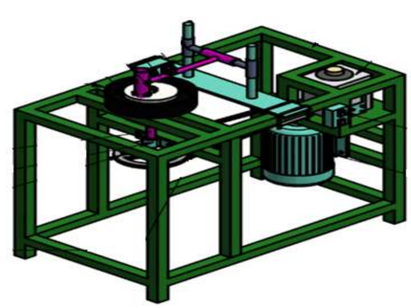

Figure 6: Concept II [10]

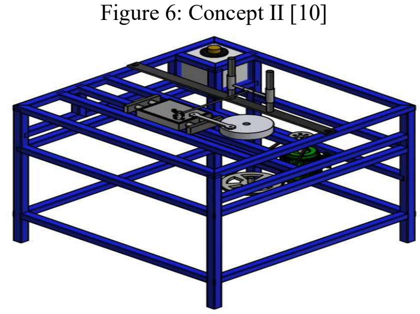

Figure 7: Concept III

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To make a decision, it is necessary to compare the assessments of each of the existing concepts. From these various concepts, subjective assessments were given as shown in Table 3. Based on the assessment in Table 3, the concept chosen from the 3 concepts were concept 3 with the highest score of 94.

Table 3: Decision matrix

No	Criteria	Wt	Consept K-I	Consept K-II	Consept K-III
1	Shaft rotation can be adjusted	10	10	10	10
2	Disc rotation can be controlled	10	10	10	10
3	Specimen size/dimension flexibility	8	4	8	8
5	Press load setting	10	7	8	9
6	Torque load setting	10	5	9	9
7	Anticipate shaft bending	9	6	8	8
8	Anti-slip	9	8	8	8
9	Specimen replacement speed	6	5	5	5
10	Low manufacturing cost	6	3	3	3
11	Ergonomic	8	2	3	3
12	Compact and practical tool dimensions	8	2	4	6
13	Easy to make	9	3	7	7
14	Easy to maintain	7	7	7	8
	Amount	110	72	90	94

2.4 Design Analysis Using Analytical Method

1. Load Analysis on Specimen Shaft

a. Load Analysis Due to V-belt stress

$$M_1 = 9555 \times \frac{P}{n}$$
$$= 9555 \times \frac{735}{1400}$$
$$= 5.016.375 \text{ Nmm}$$

$$D_1 = 101.6\, mm = 0.1016\, m$$

$$T_e = \frac{2.M_1}{d_1}$$
$$= 98.75\, N$$

$$F = \sqrt{T_1^2 + T_2^2 - 2T_1T_2(2\cos\alpha)}$$

$$N = 47.27 \text{ N}$$

b. Analysis of maximum compressive load on specimen shaft

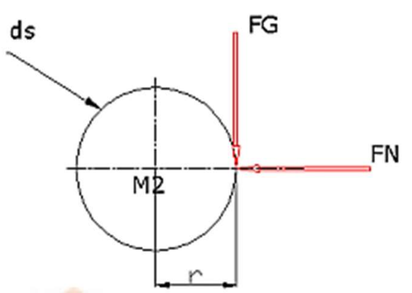

Figure 8: Analysis of load

The specimen diameter (ds) that can be tested in the design ranges from 20 mm to 50 mm, so that the maximum coefficient of static friction for a specimen diameter of 20 mm was:

$$\mu_s = \frac{2.M_2}{F_n.d_s} = 44.8$$

The oefficient of static friction at the maximum load for a specimen diameter of 50 mm was:

$$\mu_s = \frac{2.M_2}{F_n.d_s} = 17.9$$

For the coefficient of static friction greater than this range, the size of the load or the diameter of the specimen can be varied.

2. Analysis of Stress and Deflection on Shafts

a. Stress Analysis on Shaft

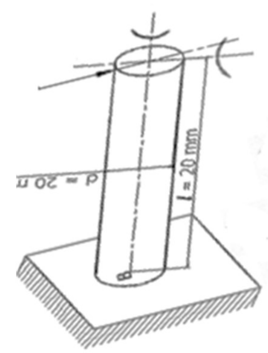

Figure 9: Load on shaft

The force acting on this support B isforce due to compressive load (F_N = 20 N) and force due to torsion (M_2 = 8,960). This compressive load FN was resulted in a shear stress that cut the shaft area in the Z axis direction and its value can be determined by the following equation:

$$M_2 = 8960$$
$$F_n = 20 \text{ N}$$

$$\tau_{F_n} = \frac{4.F_n}{3.A} \; ; \; A = \frac{\pi.d^2}{4}$$
$$\tau_{F_n} = \frac{4.F_n}{3.\left(\frac{\pi.d^2}{4}\right)} \text{ N/mm}^2$$
$$= 0.055 \text{ N/mm}^2$$

$$\sigma_{nx} = \frac{M_y.32}{\pi.d^3} \text{N/mm}^2$$

$$\tau_{M_2} = \frac{M_2.16}{\pi.d^3} \text{ N/mm}^2$$
$$= 2.92 \text{ N/mm}^2$$

$$\tau_{yz} = \sqrt{\left(\tau_{F_N}\right)^2 + \left(\tau_{M_2}\right)^2} \text{ N/mm}^2$$
$$= 2.92 \text{ N/mm}^2$$

Then the shear stress YZ is $\tau_{yz} = 2.92 \text{ N/mm}^2$

Overall the maximum shear stress (τ_{max}) that occurs at the support B can be calculated by the following equation:

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2}$$

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$$\sigma_{1,2} = \frac{\sigma_{nx}}{2} \pm \sqrt{\left(\frac{\sigma_{nx}}{2}\right)^2 + \left(\tau_{yz}\right)^2}$$

$$\sigma_2 = 0{,}65 - 2{,}95 = -2{,}3 \text{ N/mm}^2$$

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} \text{ Mpa} = 2.95 \text{ Mpa}$$

Then the maximum shear stress was obtained $\tau_{max} = 2.95$ Mpa.

b. Analysis of the deflection on the specimen shaft.
Deflection on the specimen axis due to the FN force can be calculated by the following equation (modulus of elasticity ST 37, E = 2 $x10^5$ N/mm^2):

$$\delta = \frac{F_N . l^3}{48. E. I} \qquad I = \frac{\pi}{32} d^4$$

$$\delta = \frac{2. F_N . l^3}{3. E. \pi. d^4} = \frac{2 \times 20 \times 65^3}{3 \times 2.10^5 \times 3{,}14 \times 25^4} = 0.015 \text{ µm}$$

If it was calculated with the material stiffness value, the deflection on the shaft can be calculated by the following equation:

$$\delta_1 = F \Big/ \frac{AE}{l}$$

$$\delta_1 = \frac{4. F_n . l}{\pi. d^2 . E} = \frac{4 \times 20 \times 100}{3{,}14 \times 25^2 \times 2.10^5} = 0.0204 \ \mu m$$

Meanwhile, the deflection equation for the torsional load was:

$$\theta = \frac{32. M_2 . l}{G. \pi. d^4} = 0.01662°$$

Deflection was 0.0612 > 0.01662°.

3. Analysis specimen shaft strength
This material has mechanical properties such as tensile strength (σu 330 MPa) and yield strength (σy 207 MPa). By using these materials, the shaft strength can be predicted as follows:
a. Predicted strength due to maximum shear load or MSFP (τmax).
The maximum allowable shear stress must be less than or equal to the original shear stress of the material (τallowble o). While the original shear stress can be calculated by the following equation:

$$\tau_o = \frac{\sigma_y}{\sqrt{3}} = \frac{207}{\sqrt{3}} = 119{,}5 \text{ N/mm}^2$$

The results of this calculation indicate that the shaft is very safe to withstand shear loads.

b. Predicted safety factor in design (F_k).
The prediction of the safety factor was carried out with reference to the Table 4. Therefore, the safety factor of design in this research was revealed:

$$(F_k) = F_0 \ x \ l_1 \ x \ l_2 \ x \ l_3 \ x \ S_1 \ x \ S_2 \ x \ S_3 \ x \ S_4 \ x \ S_5 = 102{,}054$$

Table 4: Value of design safety factors

Factor	Relevance in design	Value
	Consequences of Failure	
F_0	Failure	1 – 1.4
	Considering related to load estimation	
l_1	Big Load	1 – 1.6
l_2	Load application value	1.2 – 3
l_3	Load between components	1 – 1.6
	Considering the material and design	
S_1	Material Variations	1 – 1.6
S_2	Consider manufacturing	1 – 1.6
S_3	Consider operational (temperature, corrosion)	1 – 1.6
S_4	Effect of stress concentrations (Analysis of values)	-
S_5	Reliability of mathematical model	1 – 1.6
	Safety factor of design	
	$(F_k) = F_0 \ x \ l_1 \ x \ l_2 \ x \ l_3 \ x \ S_1 \ x \ S_2 \ x \ S_3 \ x \ S_4 \ x \ S_5$	

c. MPFP Normal Load Strength Prediction (σ_{max})
For ductile materials, this safety requirement can be seen from the shear stress that occurs, namely (τallowable a), while for brittle materials, this safety requirement can be seen from the maximum principal stress [11]. In this case, if using these conditions, the maximum principal stress (σnx = 1.3 MPa), tensile strength (σu = 330 MPa) and the safety factor variable wasset in maximum conditions (Fk = 102.054), so that the condition was:

$$\sigma_{max} \leq \frac{\sigma_u}{F_k}$$

$$1.3 \leq 3.234$$

Therefore, according result of safety factor of the shaft was very safe to withstand the load.

4. Mainframe Static Analysis
a. Upper mainframe analysis
Before performing an analysis on the upper main frame, the section of the trunk on the frame was calculated.

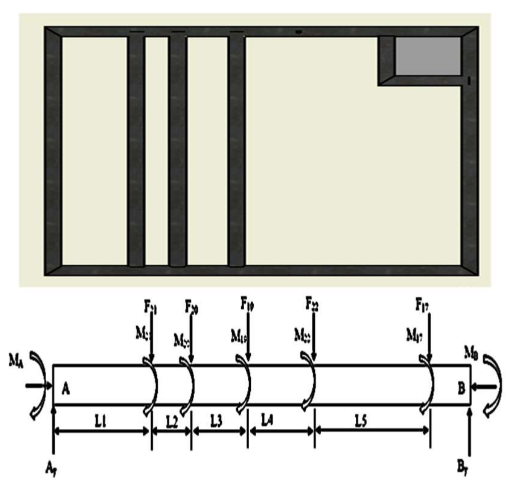

Figure 10: Main framework of test equipment

Static analysis was carried out on the top of the frame, then the working load was as follows:
F17 (Weight Regulator) = 39.24 N
F19 (Weight of Load Cell) = 17.15 N

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F20 (Disc Weight) = 22.07 N
F21 (Reciprocating Weight) = 9.81 N
F22 (Test Pin Weight) = 34.33 N
M17 = 40.8 N.mm
M19 = 17.836 N.mm
M20 = 22.95 N.mm
M21 = 10.2 N.mm
M22 = 35.37 N.mm
L1 = 220 mm L5 = 290 mm
L2 = 110 mm L = 1040 mm
L3 = 140 mm L4 = 170 mm

Table 5: Results of calculation of parts

No.	Cut area (mm)	Shear Force (N)	Formula of Moment	Moment (N.mm)
1.	0≤x<220	49.43	M = 49.43 (x)	0
2.	220≤ x<330	39.26	M = 39.62X + 2168.4	10,884.80
3.	330≤ x<470	17.55	M = 17.55X + 9474.45	15,266.00
4.	470≤ x<640	0.4	M = 0.4X + 17552.836	17,740.84
5.	640≤ x<930	33.93	M = 39559.406 − 33.93X	17,844.21
6.	930≤ x<1040	73.17	M = 76052.606 − 73.17X	8,004.51

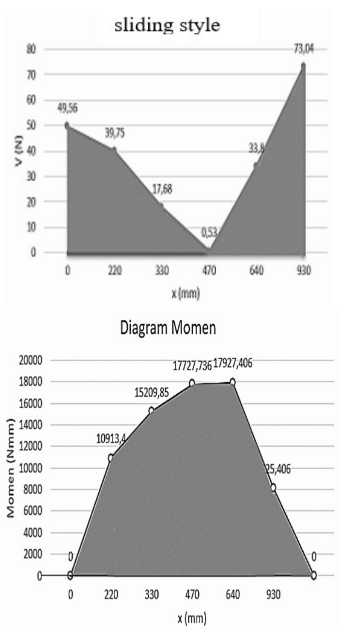

Figure 11: Moment diagram of the upper frame

b. Lower mainframe analysis

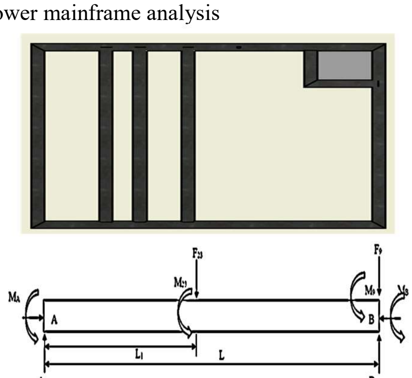
Figure 11: Main fof the bottom of the test tool

Static analysis was carried out on the top of the frame, then the working load was as follows:
F9 (Test Pin Weight) = 63.675 N
F23 (Weight of Pulley and Shaft) = 14.7 N
M9 = 66.31 N.mm
M23 = 15.288 N.mm
The distance between each load was as follows:
L = 1040 mm
L1 = 470 mm

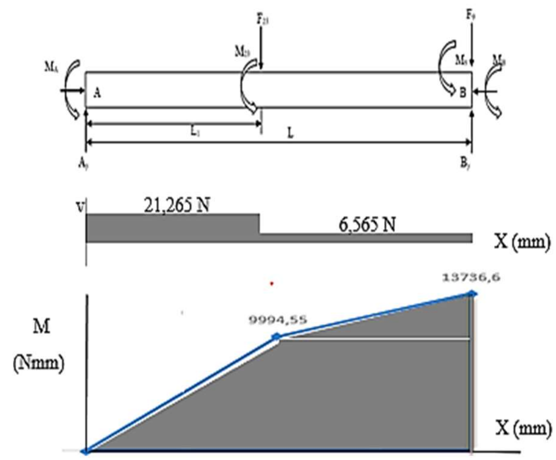

Figure 12: Shear diagram of the lower frame

3.0 RESULT AND DISCUSSION

3.1 Calculation Analysis Result

The results of the calculation analysis on the design of the pin on disc Tribometer test equipment were as follows:

1. Shaft

Based onthe results of manual calculations that have been carried out in the design, obtained the maximum stress, shear stress, deflection and safety factor. Table 6 displays the data from the calculation of the main stress, shear stress, deflection and safety factor.

Table 6: Manual calculation result

Analysis	Load	Stress (MPa) Normal	Stress (MPa) Shear	Total deformation	Safety factor
Analysis results	20 N	1.3	2.95	0.015 µm	3.23

2. Main Frame

Based on the results of manual calculations that have been carried out in the design, was obtained the maximum main stress, shear stress, Von mises and safety factor. Table 7 displays the calculation results of the maximum main stress, shear stress, Von mises and the safety factor.

Table 7: Manual calculation result

Analysis	Stress (MPa) Main Max	Stress (MPa) Shear	Von mises	Safety factor
Analysis results	37.837	0.955	66.74	7.1

3.1 Analysis Results with Autodesk Inventor Software

The results of the analysis using Autodesk Inventor were as follows:

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1. Shaft

Based on the simulation results using Autodesk Inventor software, the maximum stress, displacement, and safety factor can be seen in Table 8.

Table 8: Shaft simulation results on Autodesk Inventor software

No.	Score	Total deformation	Maximum stress	Safety factor
1	Maximum	1.65×10^{-5} mm	2.72 MPa	15
2	Minimum	0	-0.49 MPa	3.45

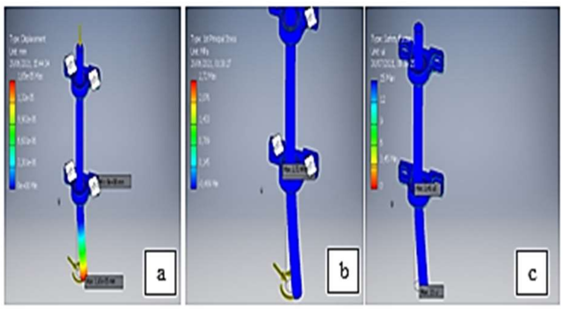

Figure 13: (a) Total deformation, (b) Maximum shear stress, (c) Safety factor

2. Main Frame

Based on the simulation results using Inventor software, the maximum stress, displacement, and safety factor can be seen in Table 9.

Table 9: Frame simulation results on Autodesk Inventor software

No.	Value	Maximum shear	Safety factor	Von Misses
1	Maximum	22.67	15	26.79
2	Minimum	-7.09	7.73	0

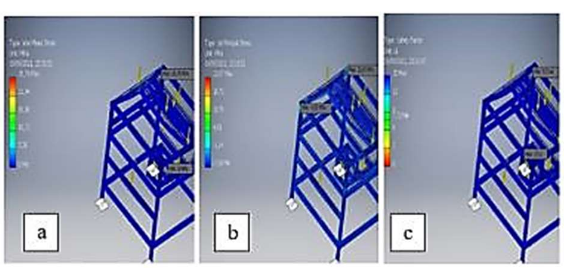

Figure 13. (a) Von Misses, (b) Main Voltage, (c) Safety Factor

4.0 CONCLUSION

In the research that has been carried out using the Hatamura method on the design of the pin on disc tribometer test equipment. The result was the pin on disc tribometer test tool that was safe to use to test the wear rate and determine the coefficient of friction with a pin diameter of at least 20 mm and a maximum of 50 mm. That was due to the safety factor desaign for the pin on disc type tribometer above 3, namely 4.04 for the frame and 3.23 for the axle. The main components of the pin on disc tribometer were the main frame, pulleys and belts, shafts, electric motors, discs, reciprocating tools, and bearings.

From the calculations that have been carried out, the dimensions of the main components, namely the main frame were made of holow steel material with a size of 1040 × 960 × 1020 mm. The shaft was made of stainless steel with a size of 25 × 440 mm; pulley with a size of 7 inches and 4 inches; belt length of 250 mm, and bearings with a size of 25 mm. From the analysis of the frame and axle carried out by calculation and validation using Autodesk Inventor, the load value on the shaft was 20 N. The maximum stress on the shaft and frame was 1.3 MPa and 22.67 Mpa. The shear stress on the shaft and frame was 2.95 MPa and 0.955 Mpa. The safety factors on the axle and frame were 3.23 and 7.01. Because the limit value of the safety factor was used 3, the design value on the frame and axle was safe to use in the design of the pin on disk type tribometer test equipment.

The results of the design of the pin on disk tribometer test equipment were in the form of technical drawings and bill of materials in accordance with ISO standards.

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