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Portable Coffin Lowering Device for COVID-19 Corpse Handling: Education, Design Process, and Strength Analysis

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ABSTRACTS

With the growing rate of the COVID-19 case that has not decreased significantly, healthcare workers are required to work extra both in the hospital and for the corpses' funeral process. Since the corpse condition is different from a normal corpse, the treatment must also be different. The objective of this paper is to propose a portable device as an education tool that can ease the burial process of COVID-19 corpse. This device can be operated by only two people, and the load is bearable because there is a worm gear system that helps the pulley system. Other supporting features will be explained in this paper. Theoretical strength analyses are conducted under static and fatigue loading. From the static case, based on the Maximum Shear Stress failure criterion, the safety factors on the critical parts are 1.28 and 5.67. Then, fatigue strength analysis is conducted to provide life prediction for this device. As a result, the fatigue safety factors are found to be greater than 1. These findings indicate that the proposed design is appropriate for real-world use and can be further studied for manufacturing.

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1. INTRODUCTION

The World Health Organization (WHO) in China was alerted on December 31, 2019, that many patients with an undisclosed form of pneumonia have been found in Wuhan, China (Beltrán *et al.*, 2021). This earlier unknown phenomenon was finally identified as a newly discovered infectious disease called Coronavirus Disease (COVID-19). COVID-19 has quickly spread across the world since the first confirmed case in Wuhan in late 2019. Soon on March 11, 2020 the pandemic was declared by the World Health Organization (WHO) (Jimenez-Sotomayor *et al.*, 2020; Beltrán *et al.*, 2021). COVID-19 has spread to almost all countries in the world, including Indonesia. On March 2, 2020, President Joko Widodo confirmed that there are two Indonesians have tested positive exposed by COVID-19, making them as the first two confirmed cases in Indonesia. Since the first case confirmed, there were other cases of COVID-19 in Indonesia, the number of people infected with it has significantly increased. **Figure 1** shows data on confirmed positive cases of COVID-19 in Indonesia since the first cases in March 2020.

COVID-19 pandemic has continued to increase, adding to the long list of people who have been infected and died. The front-line contributors to this pandemic are healthcare workers (Tosepu *et al.*, 2021; Widjaja *et al.*, 2020). Many researchers have reported the ways how to fight against COVID-19 pandemic (Machmud and Minghat, 2020; Putra and Abidin, 2020; Anggraeni, 2020; Razon, 2020; Hamidah *et al.*, 2020; Hashim *et al.*, 2020; Dirgantari *et al.*, 2020; Mulyanti *et al.*, 2020; Sangsawang, 2020; Nasution and Nandiyanto).

This pandemic would undoubtedly increase the level of tension experienced by healthcare workers, who will have to deal with personal protective equipment as well as the fear of infecting themselves and other family members (Wong *et al.*, 2020). From March 2020 to January 2021, 647 health care workers died due to COVID-19. It is not only because of being exposed to the COVID-19 virus, but few of them are also suffering because they must work extra while people are recommended to reduce outside home activities (Ketphan *et al.*, 2020; Tosepu *et al.*, 2021). Healthcare workers not only work directly in the hospital, but some also have to work to help bury people with COVID-19 who need special treatment in the funeral process.

According to health protocols, patients with COVID-19 who have died must be covered with a shroud/plastic material (not permeable to water) and buried with a coffin made of wood or other material that is not easily contaminated to reduce further transmission. The number of deaths that getting bigger makes the overload works also add more burden for the healthcare workers. **Figure 2** shows a statistical graph for the number of death cases from COVID-19. This vast numbers of death, which requires healthcare workers to bury many corpses every day, will trigger infirmity to their bodies, and it must be something difficult for them to deal with every day. Besides, referring to the typical condition of the coffin with the body inside, approximately the total weight will be around 200 kilograms. Therefore, there is a need for devices that can help healthcare workers to carry out funerals more easily and without requiring a lot of energy from many people compared to conventional methods using ropes (Oktaviandri & Paramasivam, 2020).

The aim of this work is to introduce a device that can help the burial process of a COVID-19 coffin and does not require many people in the operational process. As the feature, this device should be able to withstand a max load of 200 Kg. Also, this device should be able to be operated by two operators. The design specifications are based on qualitative data collection from references, papers, journals, and other online media. Finally, the structural strength of the critical components is evaluated under static and dynamics loading condition by theoretical approach. Discussion on the factor of safety as well as the service life will be provided.

2. METHODOLOGY

The data for design specifications of the device are obtained through references, papers, journals, books, and other online media. The Engineering Drawing method are used to make a rough design during proposed designs stage. After the sketch is made and finalized based on the design specifications obtained from the data collection, it is then processed to produce a 3D design using SolidWorks. The last thing is to analyze the device that we have designed using Mechanical Behavior of Engineering Materials and Engineering Component Design knowledge to ensure this design is marketable.

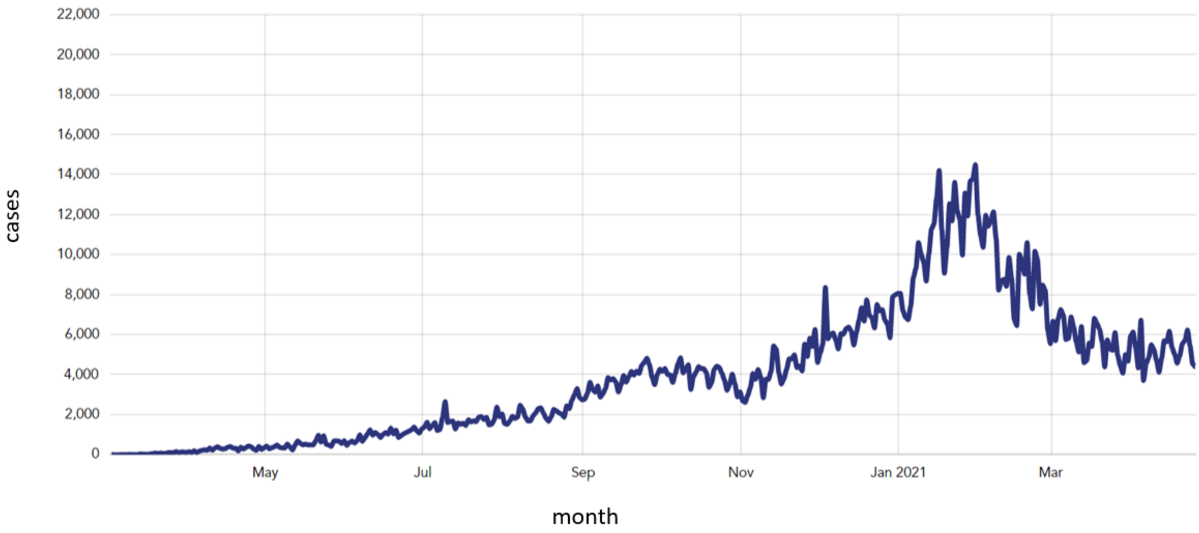

Figure 1. Graph of COVID-19 Development

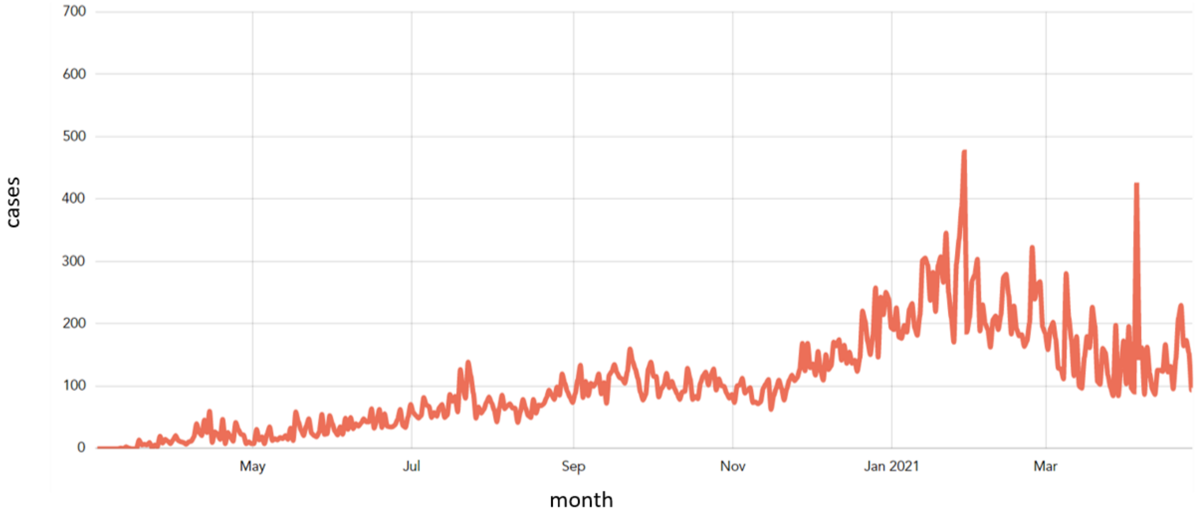

Figure 2. Graph of COVID-19 Death Cases

2.1. Design Concept

In order to accomplish the objective, the design must meet four requirements: (1) it can withstand a load of 200 kg at maximum, (2) it can be operated by two operators only, (3) it does not take a long time to operate, and (4) it must be convenient and/or portable to put in an ambulance. The proposed design has two key features that can achieve those objectives. There are modular and foldable base, as a place to put coffin, and a worm gearbox system to ease the process of lifting the coffin before it is lowered into the grave hole.

When lowering the coffin into the grave hole, the coffin is first attached by a rope to the pulley and worm gear system. Then, the modular and foldable center must be opened so as not to interfere with the lowering process of the coffin. In this section, the base in the middle must be removed from the bolt so that the base can be divided into two sides. Then fold it down so it does not hang down too much. To obtain this function, the base consists of bolts, nuts, and rotational joints. **Figure 3** shows the design of the modular and foldable base, and **Figure 4** shows its mechanism when it is being folded.

During the process of lowering the coffin to the grave hole, the heavy load will be challenging for the operator and require many people to operate it. For that, the device needs assistance other than the pulley system. The system that can help is a worm gearbox system. This gearbox will ease the process of turning the pulley to lift and lower the coffin. Worm gearboxes are preferable to standard gearboxes because the gear ratio used can be larger. If using this worm gear, the handle force used for the gear ratio is 1.32 kg.

This device is made to fit into an ambulance, so the dimension is adjusted to the ambulance in general. In this work, the ISUZU NHR 55 is used as a benchmark. The overall height of this device must not exceed the height of the ambulance. Also, the height of the legs must fit the height of the ambulance's trunk so that the legs can be folded automatically when the legs and trunk collide. Furthermore, the width is also adjusted to the conditions in the ambulance, where there are still several other things in it. **Figure 5** shows the overall dimensions of the device.

Figure 3. Modular and foldable coffin base

Figure 4. Folded coffin base

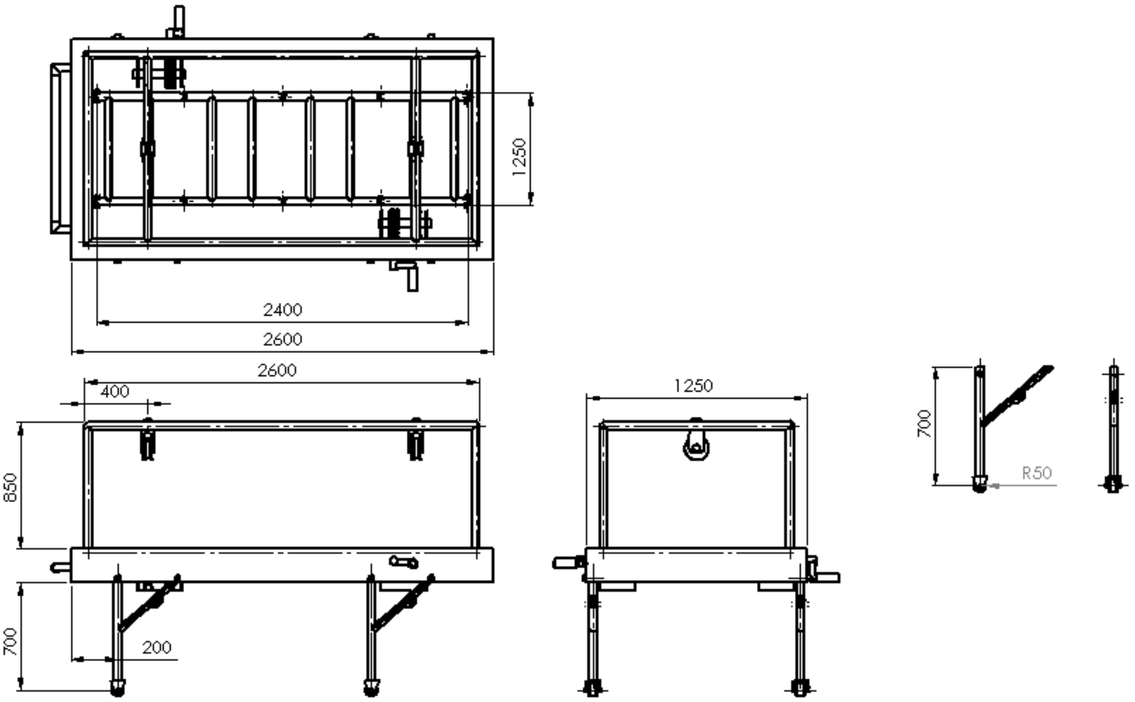

Figure 5. The device's overall dimensions

2.2. Synthesis Procedure

Before the final design is made, there are three sketched designs made to fulfill the design requirements. Basically, those designs are almost the same in term of their features; the difference is only on the placement of some parts and their mechanism. The final design selection is taken after considering the mobility and strength of this device.

The first proposed design was almost the same as the final design; the difference was the pulley holder. If the pulley has two shafts and the direction is vertical in the final design, then there is only one shaft in the first proposed design, and the direction is horizontal. This design was not chosen because it will produce a significant bending moment in the pulley system.

The second proposed design that is currently selected is the final design. The third proposed design is almost the same as the final design; the difference was the direction of the legs when it is folded. In the proposed design 1 and 2, the legs will be folded in the same direction. While in the proposed design 3, the legs will be folded in different directions and will meet in the middle. This design was not chosen because it would complicate the process of getting this device into the ambulance.

2.3. Evaluation Procedure

The device is evaluated theoretically on the main critical parts which are the base and the shaft of the pulley (Zulaikah *et al.*, 2020). These two parts are critical because the load is concentrated on those parts while the device is being used. To perform stress analysis and to evaluate the value safety factor, the assumptions is made as shown on **Table 1.** Based on the table, $\mathrm{m_C}$ is the maximum mass of the coffin that the device can afford while $\mathrm{W_C}$ is the weight of the coffin; $\mathrm{m_G}$ is the mass of the gear used to help the lifting process while $\mathrm{W_G}$ is the weight of the gear with the mass of the gear is 3 kg; g (acceleration due to gravity), is the acceleration on the body caused by the gravitational field that centralize toward the earth; $\mathrm{D_o}$ and $\mathrm{D_i}$ stands for the outer and inner diameter of the shaft because the hollow cylinder is used in this case; D is the diameter of the base part in which the solid cylinder is used.

For the service life prediction, the fatigue analysis is done by using the assumptions shown on **Table 2.** The material used is stainless steel 316L with cold drawn surface finish.

3. RESULT AND DISCUSSION

3.1. Free Body Diagram Analysis

The first analysis is done for the free body diagram due to the external force. The device deals with two different cases. First, when the coffin is placed on the device base, not connected to the pulley yet, as shown on **Figure 6**. Meanwhile, the second case is when the coffin is lifted using the pulley as shown on **Figure 7**.

The applied or external forces of the two cases are the coffin and the gear box. Therefore, the applied and reaction forces are drawn as shown on **Figures 6** and **7**, respectively. By applying the static equilibrium analysis, the reaction forces of case 1 and case 2 are obtained and it is shown on **Tables 3** and **4**, respectively.

At the inner base of case 1, where the coffin is located, there is a modular part that is critical because the load is concentrated on that point. To obtain the force, the partition analysis which is half of the inner base is used. The FBD is shown on **Figure 8**. It is using distributed force because the load of 1962 N from the coffin is distributed on that area. The result of reaction forces as shown on **Table 5**.

Table 1. Parameters for stress and safety factor analysis

$\mathrm{m_C}$ [kg]	$\mathrm{W_C}$ [N]	$\mathrm{m_G}$ [kg]	$\mathrm{W_G}$ [kg]	$\mathrm{g[m/s^2]}$	$\mathrm{D_o}$ [m]	$\mathrm{D_i}$ [m]	D [m]
200	1962	3	29.43	9.81	0.05	0.03	0.05

Table 2. Parameters for fatigue analysis

Properties	**Stainless steel 316L**
S_{ut}	485 MPa
S_{yt}	170 MPa
F_{max}	1962 N
F_{min}	0 N
Surface finish	Cold drawn
Working temperature	20°C or 68°F
reliability	90 %

Table 3. Reaction forces of case 1

Parameters	**Force [N]**
A_y	596.9
B_y	1394.53

Table 4. Reaction forces of case 2

Parameters	Force [N]
A_y	596.9
B_y	1394.53

Table 5. Reaction forces of base part

Parameters	Force [N]
O_y	445.91
E_y	535.09

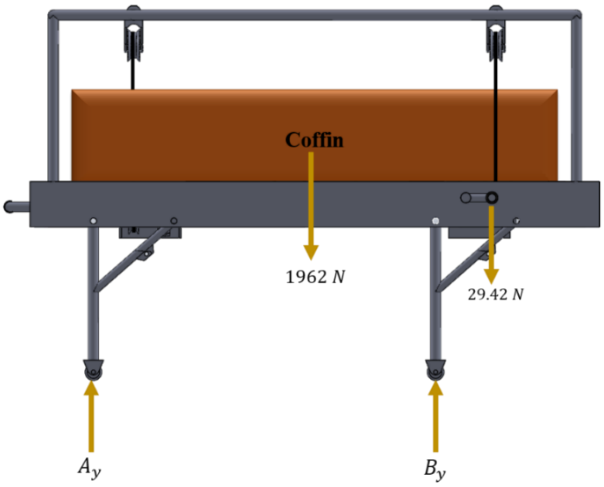

Figure 6. Case 1: Coffin is put on the base.

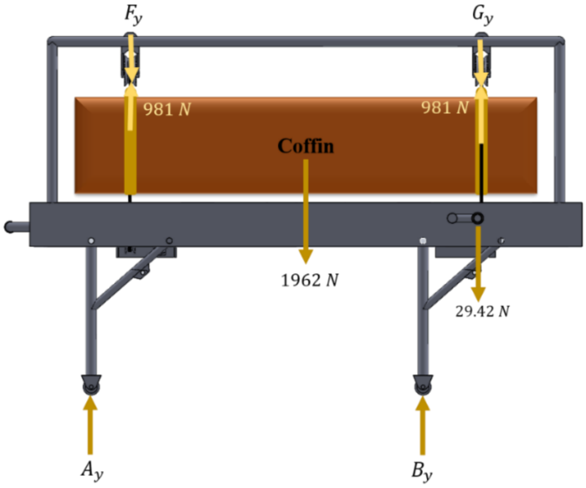

Figure 7. Case 2: Coffin is hanged by the pulleys.

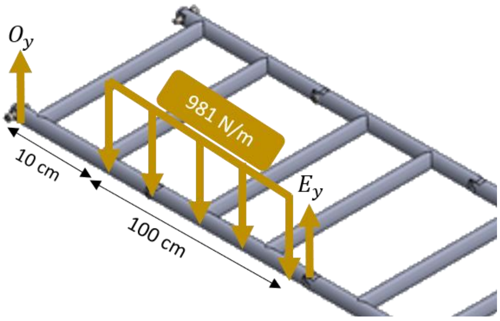

Figure 8. Inner base part

The other critical part is located at the shaft of case 2 where the pulley is located because once the coffin is lifted, the one that will support the coffin is the shaft. Therefore, the FBD analysis is important in this part. The FBD and reaction forces of the pulley shaft are shown on **Figure 9** and **Table 6**, respectively. The 981 N is the weight felt by each shaft.

3.2. Stress and Safety Factor Analysis

The second analysis is the Stress analysis to evaluate the frame strength. In this work the stress analysis is divided into two parts. One is when the coffin is sitting at the modular base (case 1) and the other when it is being lifted (case 2). In this part, the stress analysis was done for the whole structure as well as the inner beam and shaft of pulley component (critical points). For the shaft of the pulley, its length is defined to be 125cm. **Figure 10** show the 3D for the shaft of pulley. Once the stress analysis is completed, the safety factor for each component can be obtained.

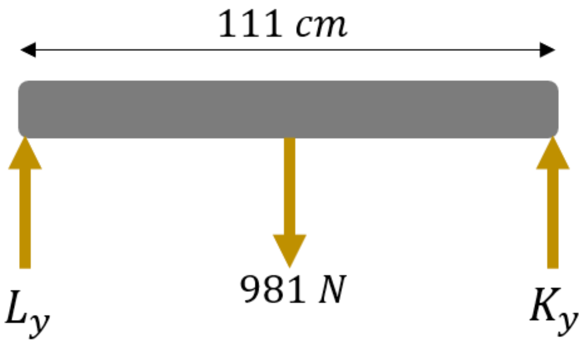

Figure 9. Pulley shaft

Table 6. Reaction forces on pulley shaft

Parameters	Force [N]
L_y	490.5
E_y	490.5

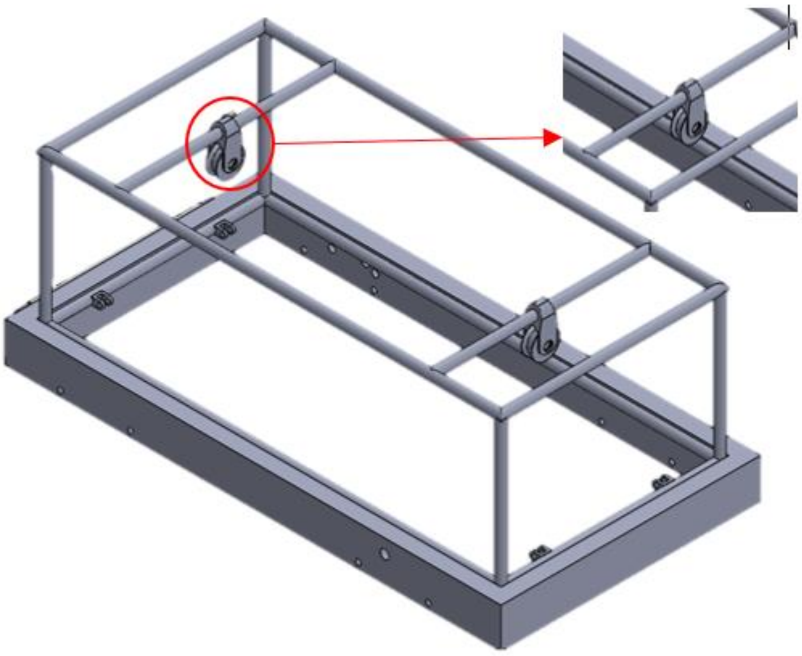

Figure 10. Shaft of the pulley

The stress analysis for all components was done theoretically using the following equations.

$$\tau_{\max(\text{shaft hollow cylinder})} = \frac{VQ}{It} = \frac{\frac{4}{3}V(R_o^3 - R_i^3)}{\pi(R_o^4 - R_i^4)(R_o - R_i)} \quad (1)$$

$$\tau_{\text{max(whole structure case 1 and 2)}} = \frac{VQ}{It} = \frac{3V(4b'+h')}{2th(6b'+h')} \qquad (2)$$

$$\sigma_{\text{max(shaft hollow cylinder)}} = \frac{M_{\text{max}}}{S_{\text{hollow cylinder}}} \qquad (3)$$

$$\sigma_{\text{max(whole structure case 1 and 2)}} = \frac{M_{\text{max}}}{S_{\text{channel}}} \qquad (4)$$

$$S_{\text{hollow cylinder}} = \frac{\pi\left(D_o^4 - D_i^4\right)}{32(D_o)} \qquad (5)$$

$$S_{\text{channel}} = \frac{BH^2}{6} - \frac{bh^3}{6H} \qquad (6)$$

$$\tau_{\text{max(modular part)}} = \frac{4V_{\text{max}}}{3A} \qquad (7)$$

$$\sigma_{\text{max(modular part)}} = \frac{M_{\text{max}}}{S_{\text{solidcircle}}} \qquad (8)$$

$$S_{\text{solidcircle}} = \frac{Ar}{4} \qquad (9)$$

$$n = \frac{S_y}{\sigma_{\text{max}}} \qquad (10)$$

$$\sigma_{\text{max,min}} = \frac{\sigma_x+\sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + \tau_{xy}^2} \qquad (11)$$

where the symbol definition is shown in the following

$\tau_{\text{max(shaft hollow cylinder)}}$ = maximum shear stress of hollow cylinder shaft
$\tau_{\text{max(whole structure case 1 and 2)}}$ = maximum shear stress for whole structure case 1 and 2
$\sigma_{\text{max(whole structure case 1 and 2)}}$ = maximum stress of hollow cylinder shaft
$S_{\text{hollow cylinder}}$ = section modulus of hollow cylinder shaft
$S_{\text{rectangular}}$ = section modulus of rectangular solid
$\tau_{\text{max(Inner base)}}$ = maximum shear stress of the inner base
$\sigma_{\text{max(Inner base)}}$ = maximum stress of inner base
S_{channel} = section modulus of channel beam (c beam)
D_o = Outer diameter of hollow cylinder
D_i = inner diameter of hollow cylinder
R_o = outer radius of hollow cylinder
R_i = inner radius of hollow cylinder
M_{max} = maximum bending moment
V_{max} = maximum shear force
A = area of solid circle
r = radius of solid circle
n = safety factor

With the dimensions known, the shear stress as well as the maximum normal stress can be obtained. The calculations for shear and the maximum normal stress for the shaft of hollow cylinder is using Eqs. (1), (3), and (5) whiles, for the Inner base will be using Eqs. (7), (8), and (9). Furthermore, the maximum normal stress and shear stress for the whole structure at both case 1 and case 2 can be calculated using the Eqs. (2), (4), and (6). Lastly, the principal stress for the critical points can be obtained using Eq. 11. The value for M_{max} and V_{max} is obtained from the SFD and BMD of their respective cases. From this, the data can be processed to obtain the principal stresses. Lastly, using Maximum Shear Stress (MSS) Theory, the safety factor is obtained, as shown on **Table 7**. Component Inner base was found to be the most critical component, although its safety factor is bigger than 1, which is 1.28. From **Table 7**, since the safety factor is above one, it can be concluded that every component is safe.

Table 7. Maximum bending moment, maximum shear force, maximum and minimum normal stress, and safety factor of every component

Part	M_{max} [Nm]	V_{max} [N]	σ_{max} [MPa]	σ_{min} [MPa]	n
Case 1	537.21	1365	8.04	0	5.67
Case 2	303.13	1010.43	1010.43	0	1.28
Pulley shaft	318.83	450	29.85	0	21.14
Inner beam	145	535.09	54.7	0	37.44

3.3. Fatigue and Life Expectancy Analysis

In fatigue analysis, endurance limit is the important parameter. In this work, the material that was chosen was the cold worked stainless steel 316L. Stainless steel 316L has an ultimate strength of 485 MPa and thus the endurance limit is simply half of that, $Se' = 242.5$ MPa. Then the pure endurance limit is affected by the Marin factors. To find the endurance limit or Se, simply multiply all the Marin factor with the endurance or Se'. However, we need to find the value first for each of the Marin factors. The value for each Marin factor is shown at **Table 8.** After knowing the value of all the factor simply multiply all of them with the value of the endurance strength. The calculation can be seen in the below.

$$S_e = k_a k_b k_c k_d k_e k_f S'_e = (0.8759)(1.04)(1)(0.897)(1)(242.5) = 198.15\ \mathrm{MPa}$$

The next step is to find σ_m and σ_a. These values are important to find the fatigue factor of safety using Goodman Criteria and ASME – Elliptic Criteria. To find the value of σ_m and σ_a, the formula is as shown below.

$$\sigma_a = \frac{k_f(\sigma_{max} - \sigma_{min})}{2} \qquad (12)$$

$$\sigma_m = \frac{k_f(\sigma_{max} + \sigma_{min})}{2} \qquad (13)$$

The after determining the value for σ_m and σ_a, we can find the fatigue factor of safety using Goodman Criteria and ASME – Elliptic Criteria. The formula and calculations for each criterion is as shown in the below.

$$\text{Goodman Criteria: } \frac{1}{nf} = \frac{\sigma_a}{Se} + \frac{\sigma_m}{S_{ut}} \qquad (14)$$

$$\text{ASME} - \text{Elliptic Criteria: nf} = \sqrt{\frac{1}{\left(\frac{\sigma_a}{Se}\right)^2 + \left(\frac{\sigma_m}{S_y}\right)^2}} \qquad (15)$$

The result for σ_m and σ_a as well as the life expectancy for each part can be seen through **Table 9**. From **Table 9**, it can be concluded that all of the components have a factor of safety of above 1, $nf > 1$. Since the factor of safety is above 1, it means that all of the components have infinitely life and is thus safe to use.

Table 8. Marin factors value

Marin Factor	Value
Surface Factor $(\mathbf{k_a})$	0.8759
Size Factor $(\mathbf{k_b})$	1.04
Loading Factor $(\mathbf{k_c})$	1
Temperature Factor $(\mathbf{k_d})$	1
Reliability Factor $(\mathbf{k_e})$	0.897
Miscellaneous Factor $(\mathbf{k_f})$	1

Table 9. Value for alternating and midrange stress and factor safety (Goodman and ASME)

Part	σ_a [MPa]	σ_m [MPa]	nf (Goodman)	nf (ASME)
Case 1	4.02	4.02	34.99	32.09
Case 2	2.27	2.27	61.98	56.84
Inner Base	27.35	27.35	5.14	4.72
Pulley	14.925	14.925	9.43	8.64

4. CONCLUSION

A primary design portable coffin lowering device for COVID-19 Corpse handling has been presented for educational purpose as well as for real application. To evaluate the frame's strength, stress analysis under static and fatigue loading conditions are done on the critical parts, which are the base part and pulley shaft. Based on the Maximum Shear Stress failure theory, the safety factor under static loading condition is confirmed to be at minimum number of 1.28. Then, for fatigue analysis, the fatigue safety factor is also found to be greater than one. This means that the proposed design is appropriate and safety from failure under static loading and has an infinite service life. In addition, the design process presented in the paper can be referred as a learning material for Machine Design Course.

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6. AUTHORS' NOTE

The author declares that the design idea originally came from the members. The authors declare that there is no conflict of interest regarding the publication of this article. Authors confirmed that the paper was free of plagiarism

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