

Components Design of Cube Cutter for Fruit and Meat: Strength Analyses Under Static and Fatigue Conditions

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Abstrak

Persiapan makanan atau *food preparation* adalah tahap yang memakan banyak waktu dan tenaga dalam memasak. Dalam tahap persiapan, ada banyak sekali bahan makanan yang perlu dipotong menjadi kotak dan stik. Selain itu, kebanyakan perusahaan Usaha Mikro Kecil Menengah (UMKM) makanan diharuskan melayani banyak pelanggan dengan jumlah pekerja yang terbatas, mengakibatkan para pekerja kelelahan bahkan sebelum proses memasak makanan dimulai. Oleh karena itu, penelitian ini bertujuan untuk membuat desain sebuah alat pemotong yang dapat memotong makanan menjadi bentuk kotak dan stik. Desain ini juga dilengkapi dengan analisis kekuatan yang bertujuan untuk mengevaluasi kekuatan alat pemotong ini bila dipakai pada beban maksimal 50 N. Analisis teoretikal ini dilaksanakan dengan beban statis dan lelah. Dari hasil analisis, alat pemotong ini memiliki faktor keamanan diatas 3,7 untuk beban statik dan 2,14 untuk beban lelah. Penelitian ini diharapkan dapat menjadi referensi dalam pembuatan alat pemotong yang dapat membantu usaha kecil dan menengah di Indonesia.

Kata kunci: *Alat pemotong, pemotong kubus, desain struktur, analisis kekuatan, beban statik dan lelah.*

Abstract

Food preparation is one of the longest and the most time consuming part in cooking a food. In the preparation phase, the food ingredients are often needed to be cut into cube shape or stick shape. Besides that, most of the small food industry needs to serve a lot of customers while having limited number of workers, causing the workers to be tired even before the cooking process begin. For this reason, this paper proposes a design idea of a cutting tool that can cut food ingredients into cube and stick shape. This design is also completed with the strength analysis to evaluate the components' safety for applied force of 50 N. Furthermore, the strength analysis was done by a theoretical approach of static and fatigue loading. According to the analysis, the cutting tool has a safety of more than 3.7 for static loading and 2.14 for fatigue loading. This paper is expected to be used as a reference to create a cutting tool that can help workers in Indonesia.

Keywords: *Cutting tool, cube cutter, structural design, strength analysis, static and fatigue load.*

1. INTRODUCTION

In 2019, Indonesia has 65.47 million total of Small and Medium Enterprise (UMKM), and the highest number was held by food industry UMKM (Kementerian Koperasi dan UKM Republik Indonesia, 2019). Furthermore, many of the food industry UMKM still require government assistance in terms of social help (Maharani & Jaeni, 2021). Other than that, a lot of food industry UMKM are not able to employ enough worker to work in the business which causes the workers to work more than the appropriate hour.

A worker that works in a kitchen usually deal a lot with excessive workload (Murray-Gibbons & Gibbons, 2007). This is due to a lot of the workers need to cook and prepare the food ingredients. Furthermore, food preparation is also considered as one of the most energy consuming activities in the food industry (Carlsson-Kanyama & Faist, 2000). Therefore, a lot of company in the world has started to create cutting tools that are able to simplify the food preparation process.

There are a lot of cutting tools that has been created to help the food preparation. One of the most interesting cutting tools is known as the cube cutter. However, these cube cutters that was created by the companies have some drawbacks. The first type of cube cutter that was made is only able to cut the ingredients into cubes after the food ingredients is cut into a plate shape (GEFU, 2019). While the second cube cutter is still just a knife that was arranged in a net form (BOSCH, 2014). Lastly, the third cube cutter is a cutter that can cut in both x and y axis using a link and manual mechanism, but this cutter is limited to cutting vegetables and fruit only (Starfrit, 2019).

By this reason, this paper aims to design a cube cutter capable of directly cutting fruit, vegetables, and meat into cube and stick shapes. Furthermore, the designed cube cutter will only require the user to apply 2.5 kg of force to cut vegetables and fruits while 5 kg of force is needed to cut meat due to the targeted user being housewives and chefs that are needed to cut a lot of food ingredients in a day (Ko & Lu, 2021). Besides that, all the components are analyzed theoretically under static and dynamic loading, while the life prediction of the design was determined by using dynamic loading and fatigue analysis. Lastly, the cube cutter is designed to be able to be disassembled to make it easier to store the cube cutter when it is not being used.

2. METHODOLOGY

2.1. Qualitative Research

A survey with a total of 50 participants were done to define the problems that exist. These survey participants are selected based on the targeted customer of the product which are fruit salad seller, “rujak” seller, satay seller, housewives, and small food vendor. From the survey that were conducted, it was found that these participants do not know about the existence of a product that can cut food ingredient directly into cube shape or stick shape. Other than that, the participants are all excited to have a cube cutter. Lastly, some of the participants also said that with cube cutter that can cut precisely, the business will be able to increase the profit that were made from selling the food.

2.2. Design Concept

The design that of the cube cutter needs to fulfill several requirements (Oktaviandri & Kian, n.d.). The requirements are obtained from the interviews done to the participants, which consist of: the design is limited to 3 kg in weight and 30 x 30 x 30 cm in dimension; the product is able to be disassembled and easy to store; the steps in cutting the food ingredients are not more than 3; the material of the product needs to be food-compatible; and the product only require 2 to 5 kg of force to use due to the fact that the user uses the product in a long period of time.

2.3. Material Selection

The material for the body of the cube cutter should be light, moderately strong, and also food compatible. Based on the needs, Polypropylene (PP) is the most suitable option for the body material. The reason is because polypropylene is known for its properties of (Shubhra et al., 2011): i) lightweight, ii) moderately strong, iii) food compatible, and iv) easily mass produced. Other than that, Polypropylene is also suitable because the body will not receive a high stress.

The material that is used for the rods of the actuator should also be light, food compatible, and strong. Based on the needs, 304 grade stainless steel is the most suitable material for the rods of the actuator. This type of stainless steel is known for its (Huang et al., 2019): i) lightweight, ii) strong, iii) high wear resistance, iv) food compatible, and v) corrosion resistance. Furthermore, the rods will have a hollow cylindrical shape to reduce the weight of the whole product and to ensure there is no material that is being wasted. Lastly, the 304 grade stainless steel has a yielding strength of 205 MPa (Metals4U, 2019) and ultimate tensile strength of 515 MPa (AZO Materials, 2001).

The material that is used for the knife of the design should be light, food compatible, strong, and high wear resistance. Based on these needs, 420 grade stainless steel is the most suitable material for the knife. The reason is because the 420 grade stainless steel is already being used a lot as a knife material in the market (Banuta & Tarquini, 2010). Other than that, the 420 grade stainless steel is also known for its properties of (Alam et al., 2020): i) lightweight, ii) good wear resistance, iii) high corrosion resistance, and iv) easily sharpen.

2.4. Design Components and Final Specifications

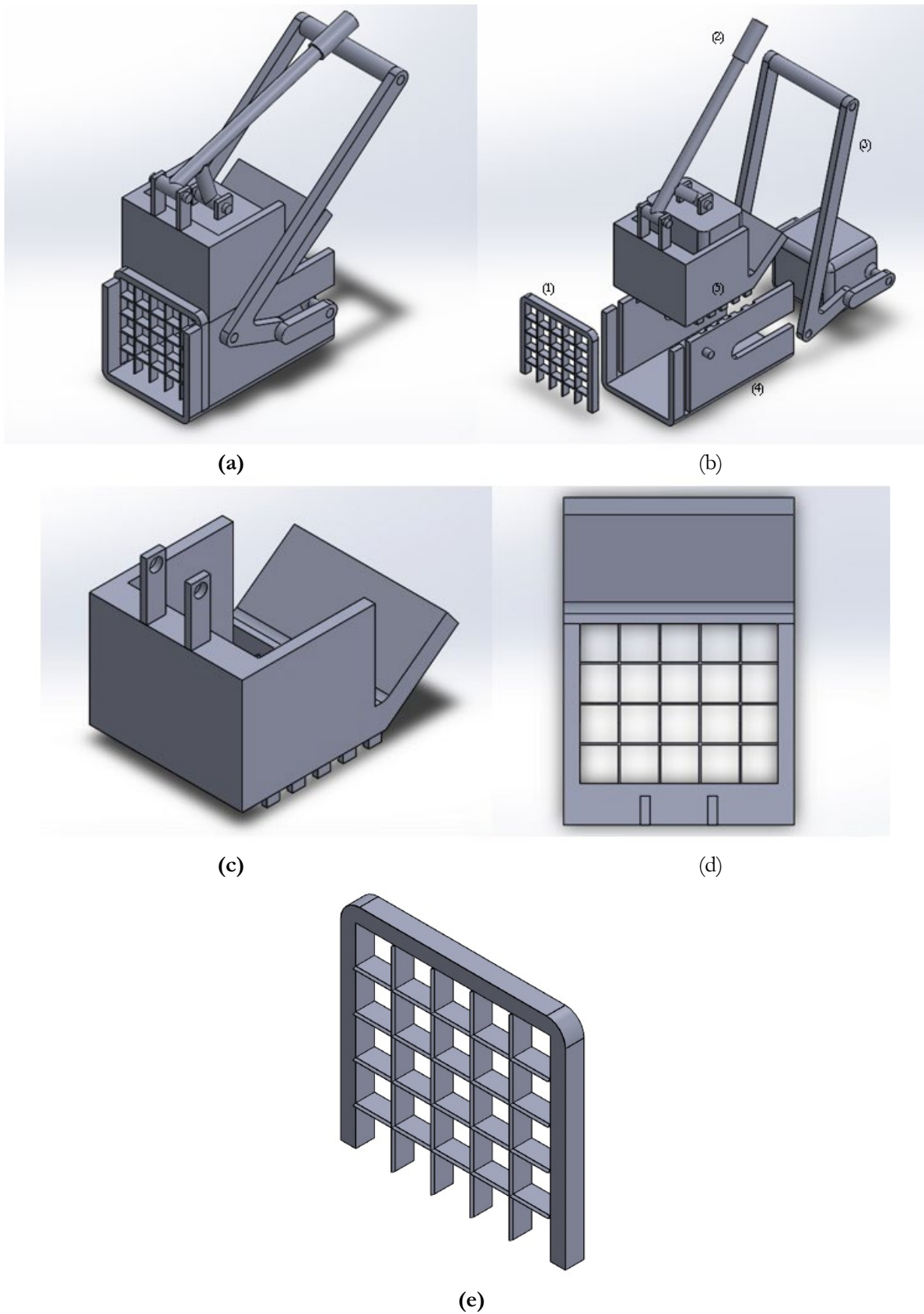


Figure 1. Isometric View of The Proposed Cube Cutter (a) Assembled Condition (b) Exploded View (c) Top Cutter Isometric View (d) Top Cutter Top View.(e) Front Cutter Isometric View

Table 1. Names For The Components

Number	Name
1	Front Cutter
2	Top Pusher
3	Bottom Pusher
4	Body
5	Top Cutter

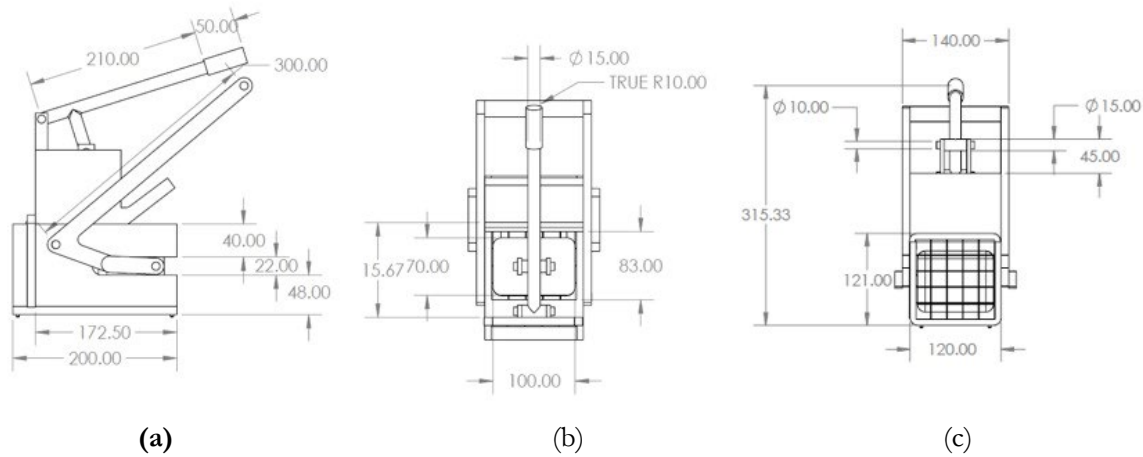


Figure 2. Dimension Of The Proposed Cube Cutter (a) Side View (b) Top View (c) Front View (all dimensions in mm).

The proposed design assembled view can be seen in Figure 1(a). As it can be seen, the designed product will have two different cutters which are the top cutter (Figure 1(c) and 1(d)) and the front cutter (Figure 1(e)). Other than that, the proposed design uses the clamping mechanism for the top pusher and a link mechanism for the bottom pusher. These two mechanisms are believed to be able to reduce the needed applied force. The proposed design will also be able to be disassembled into four different parts which can be seen in Figure 1(b). This disassembled feature is added to make the user easily store the cube cutter when it is not being used. Lastly, the proposed product knife is removable from the main body so the user can change the knife when it is dull.

The dimensions of the proposed product can be seen in Figure 2. As shown, the dimension of the proposed product does not exceed 30 x 30 x 30 cm. Figure 2 also shows the side view, top view, and front view of the product. Lastly, holder for both of the pusher is going to have a tubular shape with diameter of 2 cm to ensure that the user feel comfortable in using the product.

Table 2 shows the final specification of the proposed product. And these specifications are determined based on the problem that has been defined.

Table 2. Finalized Model's Specification

No.	Variable	Value	Unit
1	Dimension	20 × 12 × 31.5	cm
2	Weight	2.5	kg
3	Cube Per Cut Size	20 × 20 × 20 ± 0.05	cm
4	Specimen Type	Vegetables, Fruits, and Meat	-
5	Specimen Size	11 × 8 × 9	cm
6	Force Needed to Cut Fruit	25	N
7	Force Needed to Cut Meat	50	N

2.5. Working Mechanism

The proposed product need three simple steps to cut the specimen into cube shape. These are the three simple steps: 1) put the specimen into the top cutter, 2) push the top cutter using the clamp mechanism, and 3) pull down the lever of the bottom pusher. Other than that, the user can also create a stick shape by taking off the top pusher followed by putting the specimen into the bottom pusher and pulling of the bottom pusher down.

2.6. Static Analysis

The static analysis is assumed to have 25 and 50N of external force applied to the system in the static condition. The objective of the static analysis is to find the cutting force and static safety factor of the most critical part of the tool which is the top pusher. The static analysis calculation can be obtained from the Free Body Diagram (FBD) and applying the equation of equilibrium to the structure. The detailed dimension of the pushers can be seen in Figure 3 and 4. For further analyses the following equations are utilized to find the safety factor of the product (Hardiyasanti *et al.*, 2021; Zulaikah *et al.*, 2020):

$$\tau = \frac{F}{A} \text{ or } \tau = \frac{4V}{3A} \quad (1)$$

$$\sigma = \frac{F}{A} \text{ or } \sigma = \frac{Mc}{I} \quad (2)$$

$$I = \frac{\pi}{4} (r_i^4 - r_o^4) \quad (3)$$

$$\sigma_{A,B} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \quad (4)$$

$$n = \frac{S_y}{\sigma'} \quad (5)$$

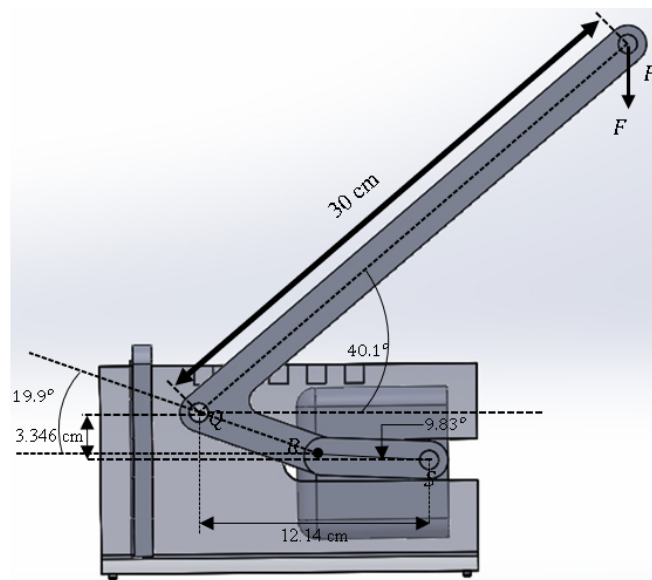


Figure 3. Real Product Drawing of The Top Pusher

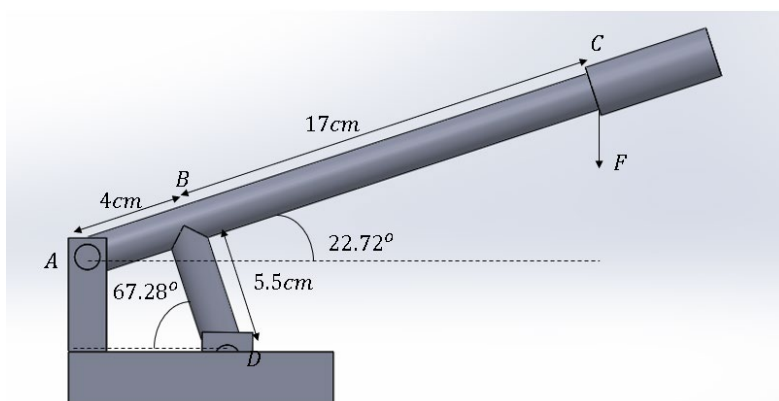


Figure 4. Real Product Drawing of The Bottom Pusher

2.7. Fatigue Analysis

The fatigue analysis is performed to find the fatigue factor of safety of the product (Budiman et al., 2020; Putra & Iswanto, 2020). From the fatigue safety factor, the number of cycles the product has can be determined and the life span of the product can be predicted. The fatigue analysis will also be done at the most critical part of the product which is the top pusher using the fluctuating load of 0 to 50 N. For the fatigue analysis, equations (6) to (14) are utilized, as shown below:

$$S_e' = \begin{cases} 0.5S_{ut} & S_{ut} \leq 1400 \text{ MPa} \\ 700 \text{ MPa} & S_{ut} > 1400 \text{ MPa} \end{cases} \quad (6)$$

$$S_e = k_a k_b k_c k_d k_e k_f S_e' \quad (7)$$

$$d_e = 0.307d \quad (8)$$

$$\sqrt{a} = 0.246 - 3.08(10^{-3})S_{ut} + 1.51(10^{-5})S_{ut}^2 - 2.67(10^{-8})S_{ut}^3 \quad (9)$$

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} \quad (10)$$

$$K_f = 1 + q(k_t - 1) \quad (11)$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad (12)$$

$$\sigma_a = \left| \frac{\sigma_{max} - \sigma_{min}}{2} \right| \quad (13)$$

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} = \frac{1}{n_f} \quad (\text{Soderberg criteria}) \quad (14)$$

2.8. Nomenclature

Table 3 below represents the nomenclature of this paper.

Table 3. Nomenclature Used In This Paper

Symbol	Description
V	Shear force (N)
F	Axial force (N)
F_f	Friction force (N)
F_{cut}	Cutting force (N)
W	Weight (N)
N	Normal force (N)
M	Bending moment (N.m)
A	Area (m)
r_i	Inner radius (m)
r_o	Outer radius(m)
τ	Shear stress (Pa)
σ	Normal stress (Pa)
I	Moment of inertia (m ⁴)
σ_A	Principal stress 1 (Pa)
σ_B	Principal stress 2 (Pa)
n_s	Static factor of safety
S_e	Endurance strength (MPa)
S_e'	Ideal endurance strength (MPa)
S_{ut}	Ultimate tensile strength (MPa)
S_y	Yield strength (MPa)
k_a	Surface condition modification factor
k_b	Size modification factor

k_c	Load modification factor
k_e	Reliability factor
k_f	Miscellaneous-effect modification factor
k_t	Stress-concentrated factor
K_f	Fatigue-concentrated factor
σ_a	Amplitude component (MPa)
σ_m	Midrange component (MPa)
σ_{min}	Minimum stress (MPa)
σ_{max}	Maximum stress (MPa)
n_f	Fatigue factor of safety

3. RESULT AND DISCUSSION

In analyzing the strength of the product, static and fatigue analyses are performed. In the static analysis, the whole component of the product is analyzed to determine the most critical part or the part that is under the highest force. From the specification of the product, the force needed to cut fruit is 25 N and to cut meat is 50 N. However, in analyzing the most critical part from the product, only the system with 50N of applied force that is used.

3.1. Result

3.1.1. Strength Under Static Loading

The static analysis is conducted by using 25 N and 50 N of applied force (F) to the FBD. The FBD of the system can be seen in Figures 5 and 6.

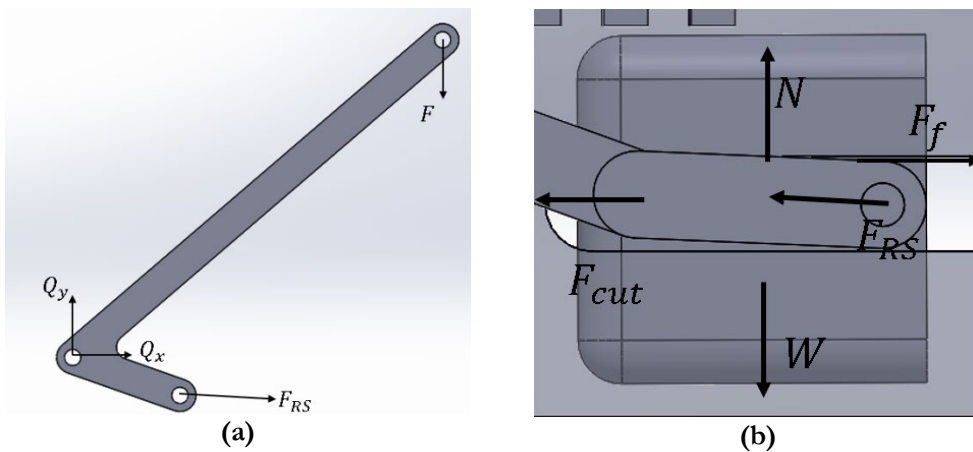


Figure 5. FBD of (a) First Link and (b) Cutting Force in Bottom Pusher



Figure 6. FBD of (a) Pipe AC and (b) Pipe BD in Top Pusher.

Utilizing the FBD shown in Figures 5 and 6, the dimension from Figures 3 and 4, and applying the static equilibrium equation, the value of each reaction force is obtained. Moreover, the values of cutting forces are also obtained. For bottom pusher, the value of cutting force is shown by F_{cut} while for top pusher it is the reaction force D_y . From the static equilibrium equation for top and bottom pusher, it is found that the value of F_{cut} and D_y for 25 N are 128.56 N and 111.67 N, while for 50 N are 255.9 N and 223.34 N, respectively.

In determining the most critical part, only the value of those reaction forces that use 50N is considered since the higher the value of force, the more stress each component has. Those reaction forces are shown in Table 4.

Table 4. Reaction Force Value and Resultant For 50 N Force.

Reaction Force	Value (N)	Resultant (N)
Q_x	-273.71	290.56
Q_y	97.52	
A_x	19.31	196.96
A_y	-196.01	
D_x	93.52	242.13
D_y	223.34	

From the value that is provided in Table 4, the stress analysis can finally be done for some of the parts in the product that is considered to be in critical situation. The stress analysis will be done on pin Q, pipe BD, and pipe AC. The reason is because pin Q experiences a high resultant force, pipe BD experiences compression, and pipe AC that experiences bending stress. Furthermore, pipe BD and pipe AC will also experience stress concentration because of welding that is used to connect both pipes into a tubular T – joint (N'Diaye *et al.*, 2009).

In Pin Q, there is a single shear force, so the shear stress can be calculated by utilizing Equation (1). Therefore, the value of τ is **3.7 MPa**.

Pipe BD experiences compression stress and stress concentration. For welded tubular T-joint in axial loading, the value of stress concentration factor is $k_t = 5$. The normal stress of Pipe BD can be found using this value of stress concentration factor and multiply it with Equation (2). The value of normal stress σ of Pipe BD is:

$$\sigma_B = 39.52 \text{ MPa}$$

Lastly, for Pipe AC, to determine the stresses, the shear force diagram (SFD) and bending moment diagram (BMD) are generated as shown in Figure 7. From the SFD and BMD, the value of maximum shear force for Pipe AC is **196.01 N** and the value of maximum bending moment is **7.84 N.m**. To get the stresses of Pipe AC, the stress concentration factor and the moment of inertia for hollow cylinder are needed. From stress concentration factor for welded tubular T-joint in in-plane-bending loading, the critical values are $k_t = 1.9$ and the value of moment of inertia, I is found by using Equation (3). The normal stress of Pipe AC is found by multiplying $k_t = 1.9$ with the Equation (2) and the shear stress is found by using Equation (1). Therefore, the value of normal stress and shear stress of Pipe AC are:

$$\sigma_x = 55.15 \text{ MPa}$$

$$\tau_{xy} = 4.38 \text{ MPa}$$

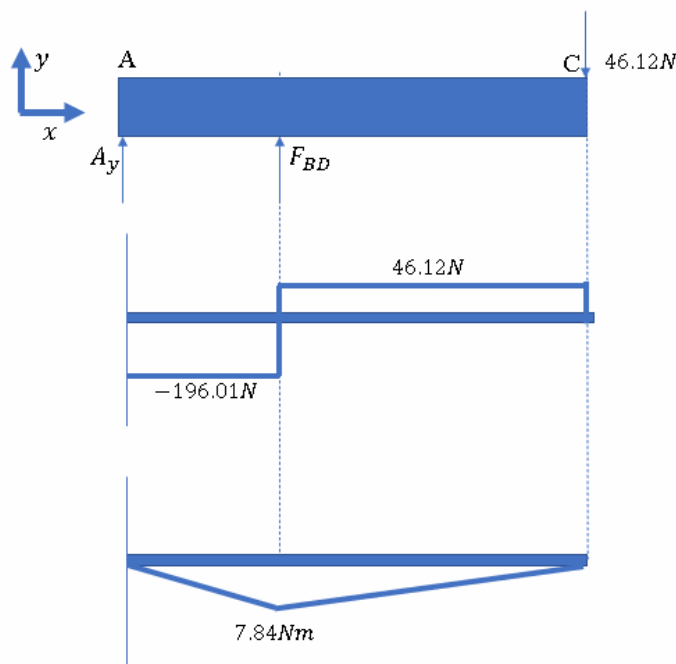


Figure 7. SFD And BMD

After stress analyses on pin Q, pipe BD, and pipe AC, the values of σ and τ are obtained. From the values, the Pipe AC has the highest stress which shows that Pipe AC will fail first. In this case, only Pipe AC that is analyzed because this pipe can be considered as one of the most critical parts of the product which can also be used to determine the overall safety factor of the product. Lastly, the principal stress at the bottom part of Pipe AC is calculated using Equation (4). Therefore, the principal stresses of Pipe AC are:

$$\sigma_A = 0 \text{ MPa and } \sigma_B = -55.15 \text{ MPa}$$

Since the material that is used in this product has the same compressive and tension yield strength, Maximum Shear Stress Theory is used to find the factor of safety. Because $0 \geq \sigma_A \geq \sigma_B$, MSS Theory case 3 is implemented and Equation (5) is used to find the static factor of safety. The static factor of safety of the product is:

$$n_s = 3.72$$

3.1.2. Strength Under Fatigue Stress

In determining the fatigue strength, the applied fluctuating load will be 0 to 50N which is very similar to the real world situation when the product is being used to cut meat. Hence, the calculation of the fatigue analysis can be seen below.

- Surface modification factor, k_a

For cold-drawn surface finish, the factor $a = 4.51$ and exponent $b = 0.862$, then the value of k_a can be found:

$$k_a = 0.862$$

- Size modification factor, k_b

Using Equation (8) to determine d_e , the value of d_e is found which is 7.4 mm , then k_b can be found:

$$k_b = 1.00095$$

- Load modification factor, k_c

The loading factor that is used for this product is the k_c for bending, which are:

$$k_c = 1$$

- Reliability factor, k_e

Since the reliability that is used is 95%, then the value of k_e is:

$$k_e = 0.868$$

- Temperature factor, k_d , and Miscellaneous-effect factor, k_f , are assumed to be 1.

Combining the known value of Marin's factors and S_e' and using Equation (7), the value of S_e can be found.

$$S_e = 192.85 \text{ MPa}$$

After that, the fatigue stress concentration factor, K_f was determined. With fillet radius of 2 mm for the welding, then the value of \sqrt{r} can be found and using Equation (9) the value of Neuber constant (\sqrt{a}) is found, which are $\sqrt{r} = 0.28$ and $\sqrt{a} = 0.089054$. Therefore, using Equation (10) and (11), q and K_f are determined.

$$q = 0.758$$

$$K_f = 1.6822$$

Since it is known that the load is fluctuated from 0 to 50N, the value of σ_{min} and σ_{max} can be found by using Equation (2), which are:

$$\sigma_{min} = 0 \text{ MPa}$$

$$\sigma_{max} = -55.15 \text{ MPa (Compression)}$$

Then, the value of midrange component, σ_m , and amplitude component, σ_a , can be found by using Equation (12) and (13), respectively, and then multiply it with the K_f . Therefore:

$$\sigma_m = 46.38 \text{ MPa}$$

$$\sigma_a = 46.38 \text{ MPa}$$

After the value of σ_a and σ_m are found, the fatigue factor of safety can be determined by utilizing Equation (14), and the value of n_f is:

$$n_f = 2.14$$

Based on the value of n_f which is more than 1, it shows that the service life of the product can be predicted to be infinite.

3.2. Discussion

It was found that the lowest cutting force from applying 25 N and 50 N is equal to 111.67 N and 223.34 N respectively. Based on research, it was found that the needed force to cut fruit is 97.7 N and 218.4 N to cut

meat (Brown *et al.*, 2005). Although the value of the cutting force is still higher than the needed cutting force, the value of the needed applied force will still be 25 N and 50 N because there are deviations in the force needed to cut the fruit and meat which will cause the needed force to be higher than 97.7 N and 218.4 N.

This designed product has one part that is going to be welded which is the tubular T pipe that is installed on the top pusher. In order to weld the T pipe, the welding technique that is going to be used is the shielded metal arc welding. Shielded metal arc welding (SMAW) is a welding process where coalescence of metals is produced by heat from an electric arc maintained between the tip of a consumable electrode and the surface of the base material in the joint being welded (American Welding Society Education, 2021)(Liu & Hu, 2022). The reason this technique is chosen is because this technique does not need complex equipment to do the welding, and it is also very cheap compared with gas metal arc welding (GMAW). Besides that, the type of welding that is going to be used is notch free welding because it has the highest strength.

The analysis resulted with applied force of 25 N and 50 N can be amplified up to 4.5 times using the installed mechanism. This shows that the user will be able to save up the energy up to 4.5 times. Other than that, the users are also ensured to not be tired after using the product for an extended period of time. Lastly, the three steps of cutting process will also save up time.

4. CONCLUSION

This study can be concluded into four important points:

1. The cube cutter was designed and analyzed in this study. The cube cutter was made to help the worker in the Small and Medium Enterprise (UMKM) food industry to be able to cut food ingredients directly into cube shape without any other tool needed. Other than that, the analysis has also proven that the applied force 25 N and 50 N will be able to be amplified up to 4.5 times. The applied force of 25 N or similar to 2.5 kg of force will be able to cut fruit and vegetables, while the applied force of 50 N will be able to cut meat.
2. The static analysis that was done on the design with the applied force of 50 N shows that the overall static safety factor of the product is 3.72. This value of safety factor shows that the designed product will not yield if a force of 50 N is applied because it has a safety factor that is higher than 1.
3. The fatigue analysis was done using dynamic loading of 0 to 50 N. The fatigue safety factor was obtained to be 2.14. Since the value of the fatigue safety factor is bigger than 1, it means that the product will have infinite life.
4. The recommendation that can be given for further research of the paper is regarding the design of the knife. In this paper, the knife is assumed to be always very sharp, which leads to no analysis regarding the life span of the knife.

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