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Design and Analysis of a Vertical-Shaft Organic Waste Chopper Machine with a Top-Mounted Agitator

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Abstract

Conventional organic waste chopping machines predominantly utilize horizontal shaft configurations, which are highly susceptible to asymmetric load distribution and present ergonomic challenges during maintenance. As an innovative solution, this study presents the conceptual design of a vertical-shaft chopping machine integrated with a hydraulic-actuated top-head system to enhance the efficiency of particle size reduction. The scope of this research is strictly confined to theoretical mathematical modeling and numerical simulations. The design process complies with the ASME B106.1M standard for transmission shafts and EN 13683 for operational safety parameters, excluding experimental fabrication. A comprehensive analysis was conducted on three primary materials—Mild Steel (MS), Stainless Steel 304 (SS 304), and Stainless Steel 316L (SS 316L)—under fluctuating waste loads ranging from 10 kg to 35 kg. The analytical results demonstrate several crucial findings: the minimum safe shaft diameter ranges from 16 mm (utilizing SS 316L at a 10 kg load) to 28 mm (utilizing MS and SS 304 at a 35 kg load). Furthermore, the agitator's power requirement exhibits a strictly linear increase corresponding to the applied load (from 1.34 kW to 4.68 kW), whereas the hydraulic system operates efficiently with a constant power profile of 0.25 kW. Overall, the vertical shaft configuration significantly promotes symmetrical force distribution, while the hydraulic system ensures safe and immediate access to the chopping chamber. The proposed design demonstrates substantial structural, operational, and ergonomic advantages, establishing a robust theoretical foundation for future development and experimental fabrication.

Keywords: *organic waste chopper; vertical shaft; hydraulic system; mathematical modeling; conceptual design*

Abstrak

Mesin pencacah sampah organik konvensional umumnya menggunakan poros horizontal yang rentan terhadap distribusi beban asimetris dan kurang ergonomis saat pemeliharaan. Sebagai solusi inovatif, penelitian ini menyajikan desain konseptual mesin pencacah berporos vertikal yang dipadukan dengan sistem top-head berbasis hidrolik untuk meningkatkan efisiensi reduksi ukuran partikel. Fokus penelitian ini dibatasi secara tegas pada pemodelan matematis teoritis dan simulasi numerik. Proses ini mengacu pada standar keamanan ASME B106.1M dan EN 13683, tanpa melakukan fabrikasi eksperimental. Analisis komprehensif dilakukan pada tiga jenis material utama, yaitu Mild Steel (MS), SS 304, dan SS 316L, dengan variasi beban sampah antara 10 hingga 35 kg. Hasil analisis membuktikan beberapa temuan krusial, yaitu: Dimensi diameter poros minimum yang aman berkisar antara 16 mm (SS 316L pada 10 kg) hingga 28 mm (MS dan SS 304 pada 35 kg). Kebutuhan daya agitator meningkat linier seiring penambahan beban (dari 1,34 kW hingga 4,68 kW), sementara daya sistem hidrolik tetap stabil pada 0,25 kW. Secara keseluruhan, konfigurasi poros vertikal mampu mendistribusikan gaya secara simetris, sedangkan sistem hidrolik memastikan kemudahan akses ruang pencacah yang aman. Desain ini terbukti menawarkan keunggulan struktural, operasional, serta ergonomis, memberikan landasan teoritis yang kuat untuk pengembangan serta fabrikasi di masa depan.

Kata kunci: pencacah sampah organik; poros vertikal; sistem hidrolik; pemodelan matematis; desain konseptual

1. Introduction

The escalating volume of organic solid waste across the domestic, agricultural, and industrial sectors necessitates highly efficient mechanical pre-treatment methods to facilitate subsequent biological degradation, composting, or biogas production processes. Mechanically driven organic waste processing relies fundamentally on effective particle size reduction. Optimal particle size reduction maximizes the surface-area-to-volume ratio, which consequently dictates the rate of microbial activity, moisture retention characteristics, and overall homogeneity of the resulting biomass mixture. Consequently, the efficiency of this size reduction process is intrinsically dependent on the mechanical design of the chopping equipment, particularly the kinematics of the cutting mechanism, blade geometry, and power transmission dynamics [1]. In the historical development of industrial and agricultural shredding machines, the design paradigm has predominantly favored horizontal shaft configurations. Single-shaft, dual-shaft, and even quad-shaft shredders typically orient their cutting rotors horizontally, relying on gravity coupled with hydraulic pushers (rams) to force organic materials into the cutting interface. Although these horizontal systems possess an established engineering track record and high throughput capacities for massive industrial scales, they exhibit notable mechanical and operational disadvantages when scaled down for small- to medium-scale urban or agricultural applications [2]. Mechanically, horizontal shredders frequently experience highly asymmetric load distributions. Given the highly heterogeneous nature of organic waste, the material tends to accumulate unevenly along the rotor's horizontal axis. This imbalance induces severe localized bending moments on the shaft. Coupled with constant torsional forces, this results in lateral deflection that accelerates bearing wear and exacerbates material fatigue within the shaft. Beyond these mechanical issues, horizontal machines present critical ergonomic disadvantages. Access to the internal cutting chamber for routine maintenance, blade sharpening, or the removal of solid materials causing jamming is highly restricted. These maintenance procedures often necessitate the partial disassembly of heavy transmission casings, thereby drastically increasing machine downtime and the risk of operator injury [3]. Recent advancements in waste shredder typologies have begun to investigate particle dynamics within rotary centrifugal shredders, demonstrating that shredding chamber geometry and rotor speed significantly alter particle trajectories and collision intensities. Reel-type and hammermill blade configurations have been implemented to integrate chopping and crushing functions into a single operational stage, thereby enhancing particle size uniformity and machine versatility. Despite these technical advancements in cutting mechanisms, the structural orientation of the primary drive shaft remains largely underexplored and critically unevaluated in the existing literature. The vertical shaft configuration, although utilized in specific high-impact mineral crushers or vertical hammermills, remains underexploited within the context of precision organic waste shredders designed for compact scales and optimized ergonomics. Theoretically, vertical shredders offer superior gravity-assisted feeding without the necessity of secondary hydraulic ram mechanisms, provide a 360-degree symmetrical load distribution across the radial plane, and enable a significantly more compact spatial footprint [4]. Furthermore, operator safety and maintenance ergonomics constitute a critical research gap in the contemporary design of organic waste shredders. Stringent safety standards, such as the European Standard EN 13683 regarding safety requirements for garden equipment and shredders, mandate precise safety clearances, robust protection against ejected objects, and rapid emergency access. Conventional hopper designs secured by bolts or clamps fail to provide ergonomic accessibility, frequently compelling operators to adopt unsafe postures during the clearance of blockages [5]. To address this research gap, the present study aims to provide a rigorous theoretical design and analytical evaluation of a novel organic waste shredder featuring a vertical shaft configuration, integrated with a dual hydraulic actuator-based top-head opening system and a tiltable discharge shell. The integration of this hydraulic head is designed to radically eliminate manual lifting constraints, ensuring strict compliance with high ergonomic and safety standards, while the vertical shaft serves to optimize load distribution and mitigate asymmetric wear [6]. Specifically, the primary objectives of this study are: (1) to formulate the theoretical mechanical design and determine the optimal minimum transmission shaft dimensions across various material grades (Mild Steel, SS 304, and SS 316L) under fluctuating loads, utilizing the ASME B106.1M standard; (2) to analytically quantify the torque and power requirements for the primary shredding mechanism and the auxiliary hydraulic support system; and (3) to establish a Finite Element Analysis (FEA) framework and validation strategy rooted in the geotechnical shear strength properties of municipal solid waste (MSW). The novelty of this study lies in the unprecedented integration of a vertical cutting architecture with a hydraulically automated maintenance access system, specifically optimized for small- to medium-scale organic waste processing. This research contributes a comprehensive mathematical foundation and modeling framework that bridges mechanical shredding efficiency with a human-centric ergonomic design prioritized for safety.

2. Materials and Methods

2.1. Research Procedures

The flowchart of this research is presented in Figure 1.

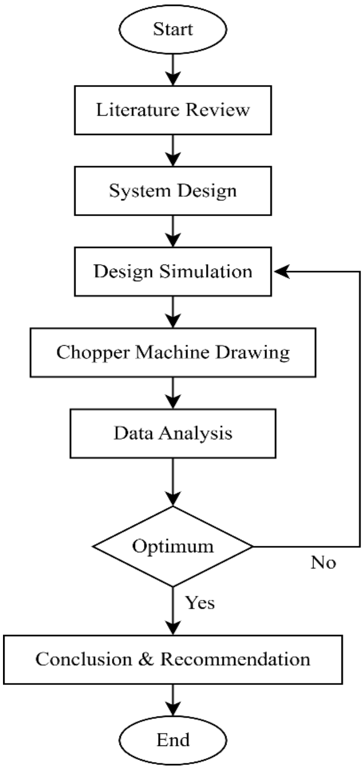

Figure 1. Research Flowchart

The research flowchart above can be explained as follows:

1. Literature review: This stage involves a comprehensive literature review of journals, books, and previous studies related to chopper machines, shredding mechanisms, blade design, and relevant analytical methods. The purpose of this stage is to establish a theoretical foundation and to identify the current state of research developments in the field.
2. System design: At this stage, the chopper machine system is conceptually designed, including the determination of the operating principle, shredding system configuration, main machine components, and functional requirements in accordance with the research objectives.
3. Design simulation: The theoretical validation strategy is formulated utilizing a FEA approach. Although the computational execution is deferred to future research, the boundary conditions and material properties are explicitly defined herein. Autodesk Inventor Nastran is recommended as the primary software for conducting the static structural analysis.

4. Chopper Machine Drawing: The results of the design and simulation are subsequently translated into technical drawings of the chopper machine. These drawings include detailed dimensions, component layouts, and technical specifications that serve as references for further development and analysis.
5. Data Analysis: The data obtained from the simulation results and design calculations are analyzed to assess the feasibility and performance of the chopper machine design, including its efficiency, reliability, and compliance with the research objectives.
6. Optimum: This stage involves decision-making to determine whether the resulting design has achieved optimal conditions. If the design is not yet optimal, the process may return to the design or simulation stage for further refinement.
7. Conclusion & Recommendation: The final stage consists of drawing conclusions based on the analysis results and formulating recommendations for further development, implementation, or future research.

2.2. Cutting Mechanics and Particle Dynamics in Vertical Choppers

The mechanism of size reduction in organic waste encompasses three fundamental mechanical forces: shearing, tearing, and impact. In conventional horizontal single-shaft configurations, the rotor operates at low to moderate speeds, relying on substantial torque to force the cutting blades to penetrate material restrained by stationary blades (stators). Conversely, the dynamics within vertical shredders operate under a distinctly different paradigm. A vertical shaft equipped with an agitator not only imparts transverse cutting forces but also leverages centrifugal effects to propel the material towards the walls of the shredding chamber. Particle dynamics within this centrifugal environment are critical; particle trajectories, collision intensities, and the probability of particle exposure to slicers or hammers are significantly governed by the rotor's rotational speed and screen geometry. Previous studies on rotary composters have demonstrated that the power absorbed to displace organic material is directly correlated with the vertical velocity component of the material's lift force, which can be approximated through mathematical regression modeling as a function of mass, chamber diameter, and angular velocity. In vertical shredders, the feed load introduced via the top-head is distributed symmetrically around the axis of rotation, thereby mitigating the unilateral bending moments that frequently induce fatigue failure in horizontal shafts. [1-5; 7; 11].

2.3. Shear Strength Properties of Organic Solid Waste

To accurately design the shaft and select the appropriate driving motor, the cutting resistance of the waste material must be precisely determined. Unlike homogeneous solid materials such as steel or wood, MSW constitutes an anisotropic and viscoelastic composite material. The geotechnical shear strength parameters of MSW govern the magnitude of the force that the shredder blades must overcome. Large-scale direct shear tests on MSW indicate that the static shear strength is optimally characterized by a cohesion (c) of approximately 15 kPa and an angle of internal friction (ϕ) of 36 degrees at a normal stress of 1 atm. These values exhibit extreme fluctuations depending on the moisture content, which ranges from 30% to 230%, as well as the proportion of organic matter. Elevated moisture levels tend to act as a lubricant, artificially diminishing the bond strength of fibrous elements, even though these organic fibers (e.g., twigs or tree bark) provide a reinforcement effect that generates tensile stress prior to ultimate failure. Consequently, this cohesion value of 15 kPa will serve as a boundary limit for modeling the cutting forces exerted on the blade in FEA simulations. [3-4; 12; 15; 20].

2.4. ASME B106.1M Standard for the Design of Transmission Shafting

Given that the shredder shaft is subjected to repetitive impact loading from organic waste, the shaft design cannot be predicated solely on static yield strength. The ANSI/ASME B106.1M-1985 standard (Design of Transmission Shafting) provides a globally recognized design protocol to address fatigue failure induced by progressive crack propagation under fluctuating loads. This standard determines the required diameter of rotating steel shafts, whether solid or hollow, by evaluating the combined loading effects of completely reversed bending moments and steady torsion. The methodology modifies the endurance limit of an ideal test specimen by applying several empirical modifying factors, including the surface condition factor (k_a), size factor (k_b), reliability factor (k_c), temperature factor (k_d), residual stress effect factor (k_g), and the local fatigue stress concentration factor (k_f) resulting from keyways or geometric transitions. Utilizing the ASME B106.1M criteria ensures that the theoretical design is not under-engineered, thereby mitigating the risks of premature shaft deflection or fracture during field operations. [2; 11; 19; 21].

2.5. EN 13683 Safety Standard

The proposed design integrates a hydraulic system for the top-head assembly not merely as a functional innovation, but as a critical response to the European Standard EN 13683 (Garden equipment - Integrally powered shredders/chippers - Safety). This standard governs the safety requirements for manually fed shredding machines. Crucial safety parameters include the elimination of potential crushing or shearing hazards at the feed opening, alongside the mandatory design of protective elements to ensure the cutting mechanism remains inaccessible to the operator's extremities. By employing a hydraulic actuator to elevate the feeding chamber during maintenance, the system eliminates the necessity for the manual lifting of heavy components, thereby mitigating musculoskeletal risks, and provides a secure mechanical-hydraulic locking mechanism during blade maintenance and clearing interventions [22-23].

2.6. Machine Architecture Design Framework

This organic waste chopper machine is conceptually designed with a primary vertical shaft configuration. In contrast to conventional shredding systems, the driving motor is optimally positioned within the top-head structure, which houses the feeding hopper. This vertical axis accommodates an agitator shaft equipped with an array of cutting blades and impact hammers, operating at a design rotational speed of 140 rpm. The nominal working capacity of the machine is engineered to process solid organic waste with a maximum load of 30 kg per operational cycle. A prominent feature of the proposed design is the integration of dual hydraulic piston actuators, symmetrically mounted on the lateral sides of the chassis. These actuators serve to vertically elevate the top-head assembly, including the driving mechanism. This architectural layout affords full visibility and 360-degree access to the internal cutting chamber. Upon reaching the requisite degree of size reduction, evidenced by the material passing through the screening matrix, the tiltable bottom shell can be tilted to discharge the final compost or biomass output. The design of the organic waste chopper machine with this configuration is illustrated in Figure 2, while the machine design dimensions are presented in Figure 3.

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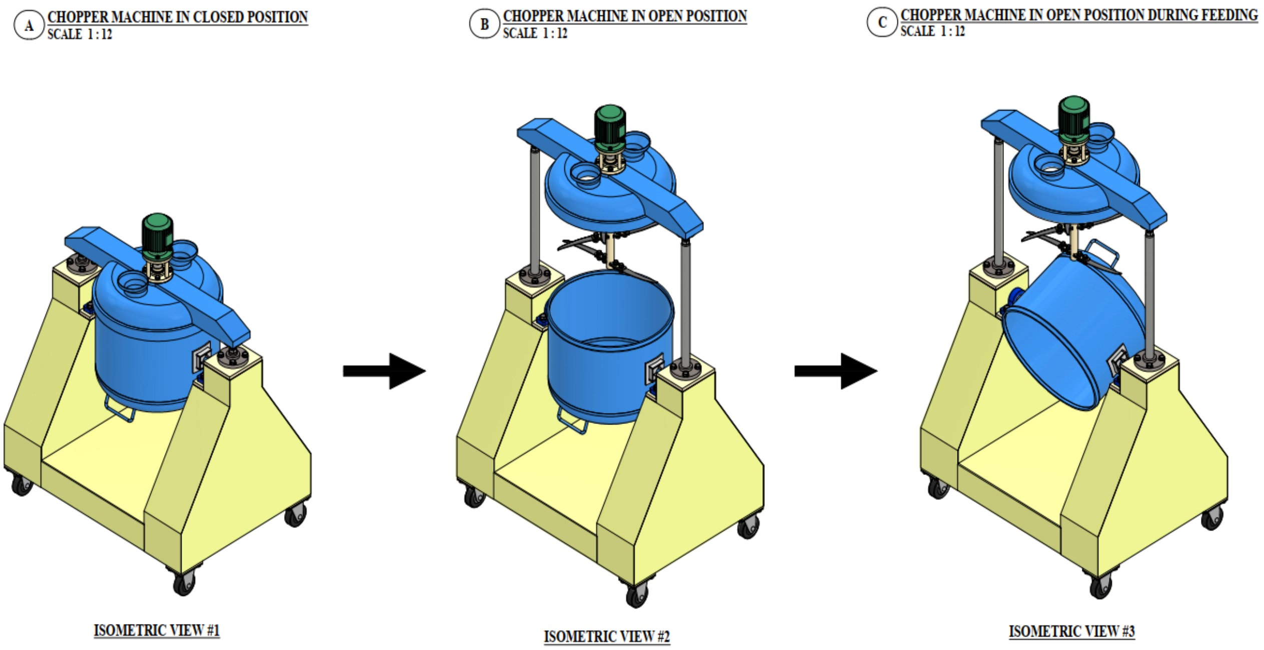
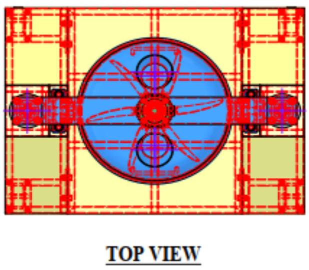


Figure 2. Design of an organic waste chopper machine



ITEM	QTY	MATERIAL	DESCRIPTION
9	4	MS	SPRING WASHER M8
8	4	MS	HEX HEAD BOLT M8
7	8	MS	FLAT WASHER M10
6	8	MS	HEX HEAD BOLT M10
5	2	MS	HYDRAULIC
4	1	MS	LEG
3	1	SS	CHOPPER AGITATOR
2	1	SS	TOP HEAD
1	1	SS	SHELL & BOTTOM HEAD

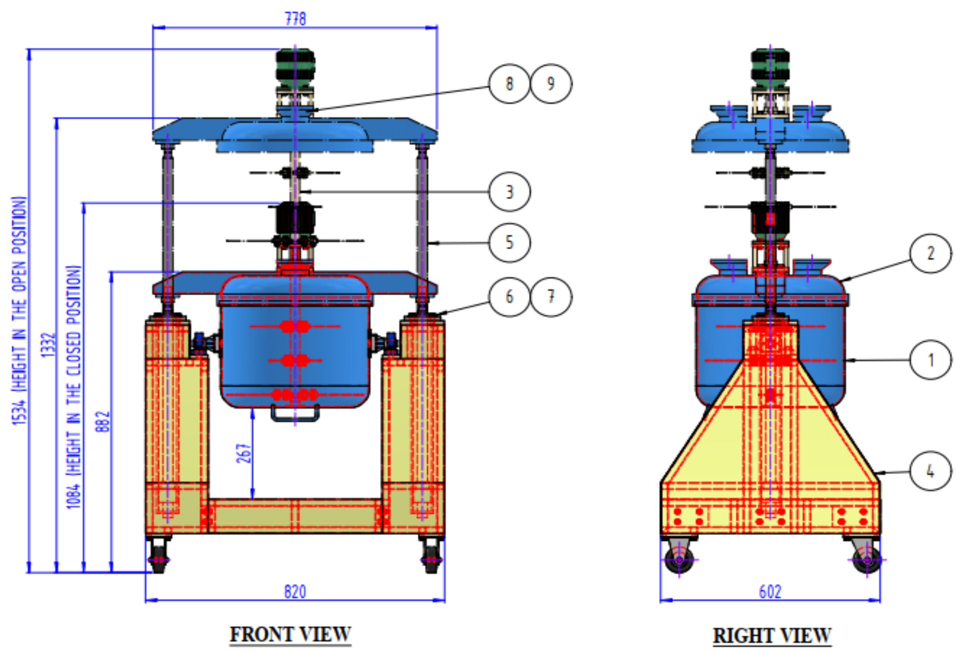


Figure 3. Design dimensions of the organic waste chopper machine (All dimensions in mm)

2.7. Numerical Simulation (FEA) Setup and Material Definition

To enhance methodological rigor, a theoretical validation strategy is formulated utilizing a FEA approach. Although the computational execution is deferred to future research, the definitions of the boundary conditions and the selection of material properties are precisely established herein [3; 7; 16-17]. The recommended software for conducting the static structural analysis is Autodesk Inventor Nastran, utilizing the following simulation parameters: (1) Load conditions: The loading is assumed to operate under steady-state conditions at the point of maximum peak torque once the shaft attains a constant rotational speed of 140 rpm. The reactive force applied to the blade tips is not designated arbitrarily; rather, it is derived from the geotechnical shear strength properties of MSW, specifically employing the Mohr-Coulomb failure criterion with a cohesion (c) of 15 kPa and an angle of internal friction (ϕ) of 36 degrees. (2) Boundary conditions (Support constraints): The vertical shaft model is secured using cylindrical bearing constraints at the two bearing support locations. This configuration releases the rotational degree of freedom about the primary Z-axis while absolutely constraining translational displacements across the X, Y, and Z axes.

2.8. Mechanical Properties for the Material Selection of Principal Components

This study evaluates three distinct steel grades for the drive shaft to demonstrate the trade-off between economic viability and mechanical performance:

1. Mild Steel (MS): A standard economical grade material, characterized by a yield strength (σ_y) of 207 MPa, a modulus of elasticity (E) of 200 GPa, and an allowable shear stress (τ_{allow}) of 62 MPa.
2. Stainless Steel 304 (SS 304): A standard corrosion-resistant grade characterized by a yield strength (σ_y) of 215 MPa and an allowable shear stress (τ_{allow}) of 65 MPa.
3. Stainless Steel 316L (SS 316L): A superior-performance alloy grade with high fatigue resistance, characterized by a yield strength (σ_y) of 317 MPa and an allowable shear stress (τ_{allow}) of 95 MPa.

2.9. Non-Fabrication Validation Strategy

Due to the exclusion of physical prototyping from the research scope, the mathematical model was cross-validated against established mechanical literature data. The plausibility of the theoretical power requirement prediction model was assessed by benchmarking it against studies reporting empirical measurements of organic compost reactor drives. In those cases, a load of 30–50 kg typically consumes an average of 1.5 to 2.5 kW at low rotational speeds (4 rpm) in the absence of significant cutting forces. Consequently, the shredder in the current study must demonstrate higher, yet physically plausible, power metrics to sufficiently account for the additional energy required for shear-cutting operations [1; 4-5; 8-10; 13-14; 16; 18].

2.10. Design Equations of the Organic Waste Chopper Machine

The following design equations are used as the basis for determining the main technical parameters of the organic waste chopper machine. These equations are primarily applied to calculate the agitator shaft diameter, agitator motor power, hydraulic piston diameter, hydraulic pump motor power, and the total electrical power requirement of the organic waste chopper machine, all of which significantly affect the performance and reliability of the shredding system. The application of these design equations aims to ensure that the machine design meets the required operational demands and is capable of operating efficiently and safely in accordance with the characteristics of the processed organic waste materials. The agitator shaft diameter is calculated using Equation (1) as follows [21]:

$$\boldsymbol{d_s = \sqrt[3]{\frac{16\, m_p\, \pi\, n^2\, r^2}{9 \times 10^2\, \tau_{allow}}}} \tag{1}$$

Where d_s is the agitator shaft diameter (m), m_p is the total mass of the organic waste product (kg), n is the rotational speed of the chopper agitator (rpm), r is the radius of the agitator blade span (m) and τ_{allow} is the allowable shear stress (Pa).

The minimum mechanical power requirement to overcome waste shear resistance and maintain rotational momentum is determined by deriving the governing equations for cutting force and torque. The agitator motor power is calculated using Equation (2) as follows [21]:

$$\boldsymbol{P_A = \frac{m_p\, \pi^3\, n^3\, r^2}{2.295 \times 10^7}} \tag{2}$$

Where P_A is the agitator motor power (kW). This formulation incorporates the inertial mass parameters (m_p), Cubic rotational speed (n^3), and the square of the radial extent (r^2) into the calculated power output, quantified in kilowatts (kW). Denominator constant (2.295×10^7) inherently accounts for standard metric conversion factors, specifically the transformation of rotational speed from revolutions per minute (rpm) to radians per second (rad/s) and mechanical power into its electrical equivalent, while also incorporating the estimated cohesive drag profile of standard wet organic waste. This calculated power is subsequently adjusted by a service factor to determine the nominal motor power rating. This margin is critical to preventing thermal failure or motor burnout in the event of abrupt shaft seizure or sudden jamming during operation.

The hydraulic subsystem is designed to elevate the static load of the top-head assembly. The structural safety analysis primarily focuses on the prevention of buckling deformation (elastic instability) in the piston rod while supporting the static dead load of the machine head. This evaluation is governed by Euler's column formula, which is integrated into the analytical calculations for determining the required hydraulic piston diameter. The hydraulic piston diameter is calculated using Equation (3) as follows [21]:

$$\boldsymbol{d_p = \sqrt[4]{\frac{96\, m_h\, g\, SF_H\, K^2\, L_H^2}{\pi^3\, E}}} \tag{3}$$

Where d_p is the hydraulic piston diameter (m), m_h is the total mass of the top head (kg), g is the gravitational acceleration (9.81 m s^{-2}), SF_H is the hydraulic safety factor, K is the effective length factor, L_H is the hydraulic stroke length (m) and E is the modulus of elasticity of the material (Pa).

The requisite hydraulic power required for actuator operation (P_H) is derived from the governing thermodynamic energy equations for fluid flow. The hydraulic pump motor power is calculated using Equation (4) as follows [21]:

$$\boldsymbol{P_H = \frac{p\, Q}{600\, \eta_m}} \tag{4}$$

Where P_H is the hydraulic pump motor power (kW), p is the hydraulic pump pressure (Pa), Q is the hydraulic pump flow rate ($m^3\ h^{-1}$), and η_m is the hydraulic pump efficiency.

The electrical generator power requirement for the organic waste chopper machine is calculated using Equation (5) as follows [21]:

$$\boldsymbol{P_G = P_A + (2 \times P_H)} \tag{5}$$

Where P_G is the electrical generator power of the organic waste chopper machine (kW).
These equations are essential for predicting device performance and evaluating the energy efficiency of the system.

3. Results and Discussion

3.1. Design Calculation Results for the Organic Waste Chopper Machine

Table 1 presents the design input parameters used as the primary basis for the engineering design and analytical evaluation of the organic waste chopper machine. These parameters include material properties, geometric dimensions, load characteristics, and hydraulic system specifications, which collectively govern the structural strength, operational performance, and safety of the proposed machine design.

Table 1. Design Input Parameters

No.	Input Parameters	Quantity	Unit
1	Material Yield Strength (σ_y) MS	207	MPa
2	Material Yield Strength (σ_y) SS 304	215	MPa
3	Material Yield Strength (σ_y) SS 316L	317	MPa
4	Permissible Shear Stress (τ_{allow}) MS	62	MPa
5	Permissible Shear Stress (τ_{allow}) SS 304	65	MPa
6	Permissible Shear Stress (τ_{allow}) SS 316L	95	MPa
7	Chopper Agitator Mass (m_a)	5	kg
8	Total Top Head Mass (m_h)	22	kg
9	Agitator Shaft Length (L_A)	0.39	m
10	Hydraulic Stroke Length (L_H)	0.45	m
11	Agitator Blade Span Radius (r)	0.19	m
12	Hydraulic Pump Pressure (p)	7	MPa
13	Hydraulic Pump Flow Rate (Q)	0.00002	$m^3\ s^{-1}$
14	Material Modulus of Elasticity (E)	200	GPa
15	Gravitational Acceleration (g)	9.81	$m\ s^{-2}$
16	Agitator Safety Factor (SF_A)	2	
17	Hydraulic Safety Factor (SF_H)	5	
18	Effective Length Factor (K)	2	
19	Hydraulic Pump Efficiency (η_m)	0.95	

Table 1 presents the fundamental parameters used as the primary reference in the design and analysis of the organic waste chopper machine with a vertical shaft configuration and a hydraulic top head system. These parameters cover material, geometric, load, and hydraulic system aspects, which collectively determine the structural strength, mechanical performance, and operational safety of the machine. The yield strength (σ_y) and allowable shear stress (τ_{allow}) of mild steel (MS), stainless steel SS 304, and stainless steel SS 316L are used as the basis for calculating the strength of the agitator shaft and other structural components. Differences in yield strength and allowable shear stress among these materials reflect variations in their load-bearing capacities, which directly influence the determination of the minimum shaft dimensions and the selection of the most suitable material in terms of strength and design efficiency. Mass parameters, including the chopper agitator mass (m_a) and the total top head mass (m_h), are used to calculate inertial forces, bending moments, and axial loads acting on the shaft and the hydraulic system during operation. These masses, combined with gravitational acceleration (g), form the basis for the analysis of static and dynamic loads experienced by the machine. Geometric parameters, such as the agitator shaft length (L_A), hydraulic stroke length (L_H), and the agitator blade span radius (r), play an important role in determining stress distribution, torsional moments, and the motion characteristics of the shredding system. In addition, the effective length factor (K) is applied in shaft stability analysis to account for support conditions and the potential for buckling. Hydraulic system parameters, including hydraulic pump pressure (p), pump flow rate (Q), and hydraulic pump efficiency (η_m), are used to calculate the piston lifting force, hydraulic power requirements, and the performance of the top head opening and closing system. These values ensure that the hydraulic system operates reliably and safely in lifting and supporting the top head during operation and maintenance. Furthermore, the material modulus of elasticity (E) is employed in the analysis of shaft deflection and stiffness, while the agitator safety factor (SF_A) and hydraulic safety factor (SF_H) are applied to ensure that the design provides adequate safety margins against structural and hydraulic system failures. Overall, the input parameters presented in this table constitute the primary foundation for design calculations, strength analysis, and performance evaluation of the developed organic waste chopper machine, enabling a systematic, measurable, and industry-standard-compliant design process.

Table 2 presents the calculated results of the shaft dimensions and power requirements of the organic waste chopper machine based on variations in material mass, material type, and specified operating conditions.

Table 2. Calculated Results of Shaft Dimensions and Power Requirements of the Chopper Machine

No.	Material	Waste (m_s) (kg)	Total (m_p) (kg)	Rotation (n) (rpm)	Agitator Shaft (d_s) (mm)	Hydraulic Piston (d_p) (mm)	Agitator Power (P_A) (kW)	Hydraulic Power (P_H) (kW)
1	MS	5	10	140	19	11	1.34	0.25
2	MS	10	15	140	21	11	2.01	0.25
3	MS	15	20	140	23	11	2.68	0.25
4	MS	20	25	140	25	11	3.35	0.25
5	MS	25	30	140	27	11	4.01	0.25
6	MS	30	35	140	28	11	4.68	0.25
7	SS 304	5	10	140	18	11	1.34	0.25
8	SS 304	10	15	140	21	11	2.01	0.25
9	SS 304	15	20	140	23	11	2.68	0.25
10	SS 304	20	25	140	25	11	3.35	0.25
11	SS 304	25	30	140	26	11	4.01	0.25
12	SS 304	30	35	140	28	11	4.68	0.25
13	SS 316L	5	10	140	16	11	1.34	0.25
14	SS 316L	10	15	140	18	11	2.01	0.25
15	SS 316L	15	20	140	20	11	2.68	0.25

16	SS 316L	20	25	140	22	11	3.35	0.25
17	SS 316L	25	30	140	23	11	4.01	0.25
18	SS 316L	30	35	140	24	11	4.68	0.25

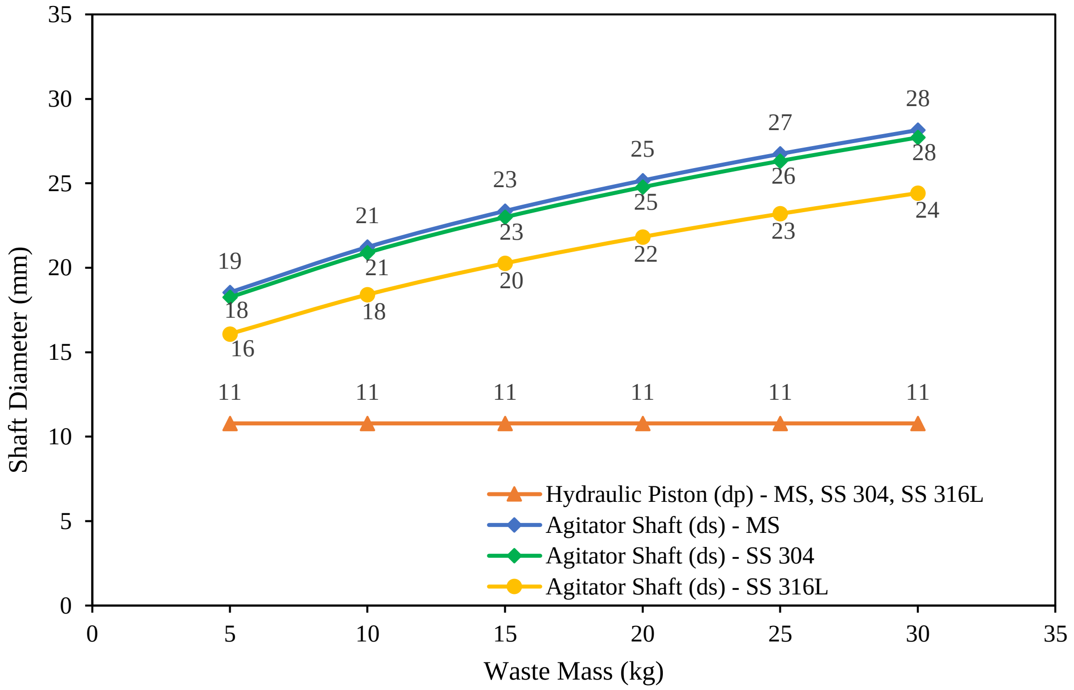


Figure 4. Effect of organic waste mass on shaft diameter

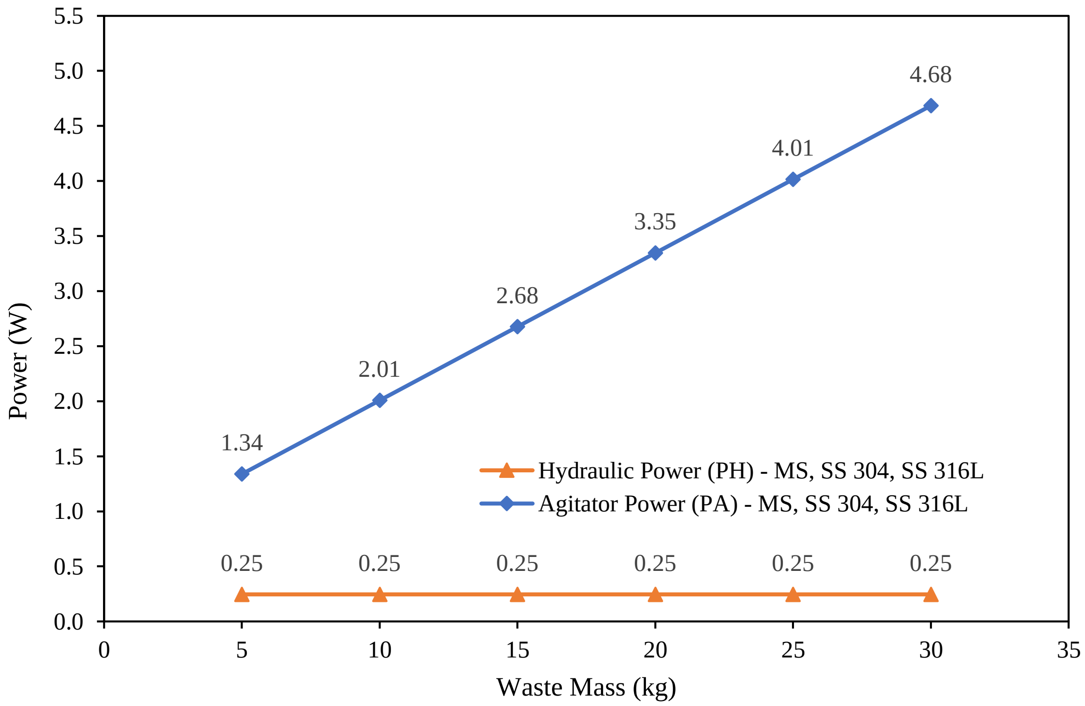


Figure 5. Effect of organic waste mass on power requirement

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Table 2, Figure 4, and Figure 5 illustrate the calculated design parameters and power requirements of the organic waste chopper machine under various structural material selections, waste load variations, and operating conditions. The materials analyzed include mild steel (MS), stainless steel SS 304, and stainless steel SS 316L, each of which has different mechanical properties that influence the shaft dimensions and the required driving power. The columns of waste mass (m_s) and total mass (m_p) represent variations in the chopping load applied to the machine, where the total mass is the sum of the waste mass and the mass of the rotating components. An increase in load mass directly affects the inertial forces and torque acting on the agitator shaft, thereby influencing the shaft dimensions and power requirements. The shaft rotational speed (n) is maintained constant at 140 rpm for all variations, allowing the analysis to focus on the effects of load and material variations on the mechanical response of the system. With a constant rotational speed, objective comparisons among different configurations can be performed. The agitator shaft diameter (d_s) column presents the calculated minimum shaft dimensions based on strength and safety criteria for each material. It is observed that, for the same load conditions, materials with higher mechanical strength, such as SS 316L, result in smaller shaft diameters compared to MS and SS 304. This finding highlights the influence of material mechanical properties on structural design efficiency. The hydraulic piston diameter (d_p) is set constant at 11 mm for all configurations, indicating that the hydraulic system is designed independently of the chopping load variations and functions solely as an auxiliary mechanism for opening and closing the machine's top head. The agitator power (P_A) column shows an increase in power demand with increasing waste mass and total system mass. This power value represents the minimum motor power required to maintain the shaft rotational speed under specific load conditions. The consistent increase in power demand indicates a linear relationship between chopping load and mechanical energy requirements. Meanwhile, the hydraulic power (P_H) is maintained constant at 0.25 kW for all configurations, as the hydraulic system operates intermittently and is not directly affected by variations in chopping load. Overall, Table 2 provides a comprehensive overview of the relationships among structural material selection, waste load variation, agitator shaft dimensions, and machine power requirements. These calculated results form an important basis for material selection, determination of key component dimensions, and specification of the motor and hydraulic system to ensure safe, efficient, and reliable operation of the chopper machine under various operating conditions.

Table 3 presents the calculated results of the power requirements of the driving system of the organic waste chopper machine, including the actual and nominal agitator power, hydraulic system power, and generator power requirements.

Table 3. Calculated Results of Power Requirement of the Chopper Machine Drive System

No.	Agitator Power (P_A) (kW)	Nominal Agitator Power (P_A) (kW)	Hydraulic Power (P_H) (kW)	Generator Power (P_G) (kW)
1	1.34	1.50	0.25	2.0
2	2.01	2.20	0.25	2.7
3	2.68	3.00	0.25	3.5
4	3.35	3.70	0.25	4.2
5	4.01	4.50	0.25	5.0
6	4.68	5.50	0.25	6.0

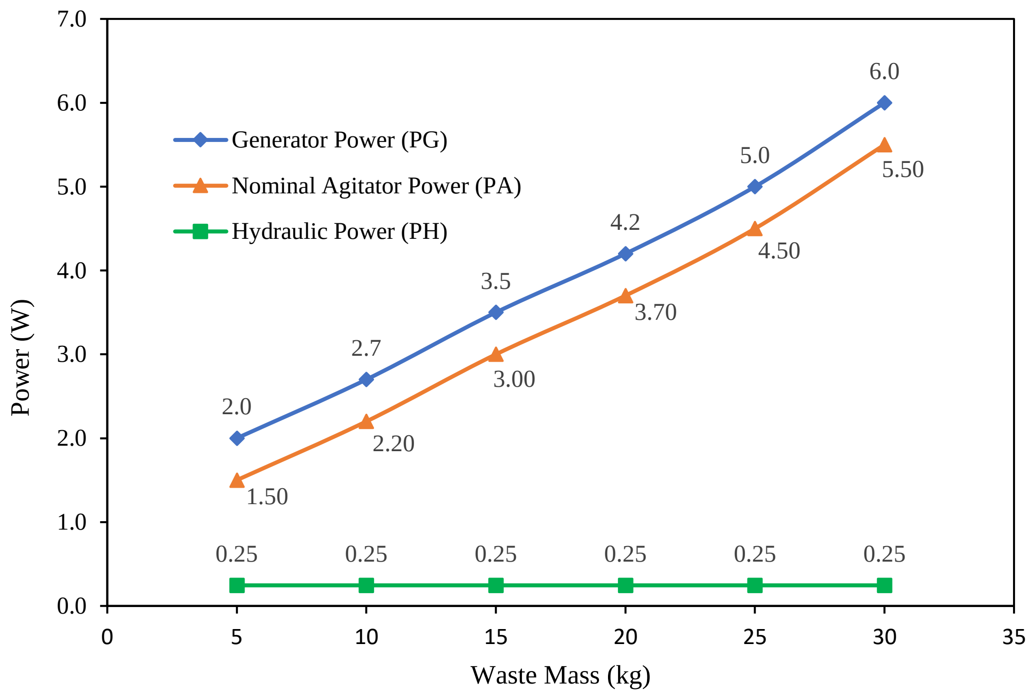

Figure 6. Effect of organic waste mass on generator power requirement

Table 3 and Figure 6 illustrate the calculated results and the selection of power requirements for the drive system of the organic waste chopper machine under various operating load conditions. The Agitator Power (P_A) column indicates the actual power requirement calculated based on the chopping load and mechanical design parameters, such as material mass, shaft dimensions, and agitator rotational speed. This power value represents the minimum power required to ensure effective chopping operation. The Nominal Agitator Power (P_A, nominal) corresponds to the motor power selected from the nearest higher commercially available electric motor rating above the calculated actual power. This selection aims to provide a safety margin against load fluctuations, mechanical power losses, and to enhance the reliability and service life of the drive motor. The Hydraulic Power (P_H) column shows the power requirement of the hydraulic system used to operate the opening and closing mechanism of the machine's top head. The hydraulic power value is maintained constant, as the hydraulic system is designed to operate intermittently and is not directly dependent on variations in chopping load. Meanwhile, the Generator Power (P_G) column indicates the recommended electrical generator capacity required to supply the total system power demand, including both the agitator motor and the hydraulic system. The generator capacity is selected based on the total power requirement while considering an appropriate safety margin, ensuring a stable and safe power supply during chopper machine operation under various working conditions. Overall, this table serves as the basis for determining suitable drive motor specifications and electrical power sources to ensure optimal, reliable, and safe operation of the organic waste chopper machine during organic waste processing.

3.2. Final Discussion

1. Introduction to Physical Analysis and Design Basis

   The design and computational results summarized in Tables 1, 2, and 3 offer comprehensive insights into the interrelationships among design input parameters, material mechanical properties, operational load variations, and power requirements for the vertical-shaft organic waste shredder equipped with a hydraulic top-head system. The input parameters outlined in Table 1 serve as the fundamental basis for all subsequent analytical phases. This analytical framework ensures that the machine's design transcends mere cutting functionality, directly addressing critical challenges pertaining to structural integrity, operational safety, and long-term system reliability.

2. Structural Efficiency and Shaft Dimensional Trends

The results presented in Table 2, Figure 4, and Figure 5 demonstrate that material variations exert a highly significant impact on the agitator shaft dimensions and overall structural efficiency. The data reveal a critical insight: at a peak load capacity of 35 kg total inertial mass, the use of conventional mild steel (MS) or SS 304 necessitates a substantially bulky shaft profile, requiring an absolute diameter of 28 mm to resist bending deformation. However, upgrading the material to Stainless Steel 316L (SS 316L) enables a reduction in the required diameter to 24 mm under identical load conditions. Even at the lower operational bound (a mass of 10 kg), SS 316L dictates a remarkably slender shaft of 16 mm, compared to the 19 mm threshold required for MS. The technical justification for this divergence is fundamentally underpinned by the ASME B106.1M fatigue failure mechanics principles, wherein the capacity to endure cyclic fatigue (endurance limit) is directly governed by the material's yield strength. Because the allowable shear stress (τ_{allow}) of SS 316L reaches 95 MPa, approximately 50% higher than the 62 MPa allowable limit of MS, this material is capable of withstanding peak fluctuations from the impulsive torsional loading of hard debris with a significantly smaller cross-section. This dimensional reduction holds substantial engineering merit; a decreased shaft diameter reduces the rotational polar moment of inertia of the component itself, which ultimately minimizes internal kinetic energy transfer losses and alleviates mechanical stress on the thrust bearing supporting the vertical load at the base of the machine. These findings are consistent with previous studies on biomass shredding machines, yet they advance the field further, as conventional designs generally overlook the implications of component downsizing on the machine's maintenance systems.

3. Mechanical Power Requirements and Operational Torque

The rotational dynamics analysis indicates that the increase in waste mass is directly proportional and exhibits absolute linearity with the surge in the agitator's mechanical power requirement (P_A). To maintain a constant cutting speed of 140 rpm amidst the geotechnical resistance from the waste matrix, the extracted mechanical energy must exceed the material's cohesive limit. Table 1 demonstrates that the power demand surges precisely from 1.34 kW under light loads (5 kg) to a peak of 4.68 kW at the maximum design capacity (30 kg). Analytically, mechanical power is a functional derivative of the product of angular velocity (ω) and shaft torque (T). Because the rotation is statically fixed at 140 rpm, the addition of material volume with an average baseline cohesiveness of 15 kPa directly expands the cross-sectional area subjected to shear forces. A thicker waste layer inherently demands a greater net torque from the blade rotor. A comprehensive quantitative data comparison substantiates the validity of this model. Literature studies on rotary composters with 30–50 kg loads record an average power consumption of 0.295 kW to 2.5 kW at rotational speeds below 4 rpm, where the material merely experiences free-fall sliding and rolling without acute cutting force penetration. Therefore, the calculated power requirement of 4.68 kW to drive a 35 kg load at 140 rpm is physically well-founded ; this supplementary power deviation is not expended merely for agitation, but rather to penetrate the viscoelastic network of plant waste and dynamically fracture its cohesive structure. The vertical rotational axis configuration also proves superior by radically mitigating the additional bending moment parameters that typically overload the bearings in classical horizontal shafts , thereby focusing the entirety of the mechanical effort on rotational shearing along the 0.19 m radial path.

4. Nominal Motor Power Intervention and Hydraulic Subsystem Analysis

   Table 3 and Figure 6 introduce a critical concept that bridges the inherent limitations of theoretical mathematical predictions with the operational realities of industrial environments. MSW does not constitute a uniform matrix; when the cutting blades impact high-fiber waste or massive, high-moisture agglomerations, the rotating components may encounter extreme resistance, inducing a momentary near-stall condition. The specification of a 5.50 kW electric motor, safely exceeding the theoretical cutting power requirement of 4.68 kW, provides an essential torque reserve margin. This design intervention is critical to mitigate the risk of thermal overload currents, which could otherwise degrade or burn out the motor's coil windings. In stark contrast to the highly variable power consumption of the cutting process, the power analysis demonstrates that the hydraulic top-head subsystem is exceptionally optimized for energy efficiency. This actuator system draws a mere, constant 0.25 kW of power to actuate two 11 mm diameter fluid piston rods. The hydraulic circuit is kinetically decoupled from the chopping load fluctuations, operating exclusively to overcome the 22 kg static dead load of the machine cover over a 0.45 m vertical elongation stroke.

5. EN 13683 Safety Compliance and Ergonomic Evaluation

   The comprehensive innovation of this research transcends the conventional focus on mere cutting efficiency by integrating fluid cylinders and a tiltable shell to resolve frequently overlooked operational constraints. Conventional maintenance approaches burden end-users with poor cutting chamber accessibility and cumbersome multi-bolt fastening configurations. The implementation of dual-lifting hydraulic actuators provides a definitive solution to comply with the European Safety Standard (EN 13683) for Garden Equipment and Integrally Powered Shredders. This standard strictly mandates the physical isolation of operators from the active cutting zone when mitigating material blockages (jamming). Governed by a 7 MPa oil pressure mechanism, the upper structure of the machine is automatically retracted, thereby eliminating the need for close-quarters manual intervention by the operator. Furthermore, the tiltable chamber design obviates the necessity of using bare hands to clear sharp residual debris. From a computational mechanics perspective, the application of Pascal's Law alongside Euler's buckling constraints demonstrates that the integration of a hydraulic Safety Factor ($SF_H = 5$) ensures the cylinder rods remain impervious to mechanical buckling. This configuration also guarantees a fail-safe mechanical locking of the top-head jaw, even in the event of an incidental loss of hydraulic fluid pressure.

6. Limitations of Theoretical Validation

   The successful conceptualization of this system is inherently constrained by its methodology, which is primarily driven by theoretical mathematical abstractions and static computational simulations, lacking empirical validation through physical fabrication and experimental prototyping. The cutting performance equations are predicated on the assumption of a steady-state, linear continuity of constant cutting force resistance. This approach risks discounting the heterogeneous, viscoelastic nature of MSW, whose shear strength fluctuates severely in response to varying moisture hydration levels (80%–130%). Dense waste agglomerations, such as high-lignin plant fronds, are anticipated to generate acute impulsive shockwaves that fall outside the predictive scope of static mathematical models. Consequently, these limitations undermine the infinite life design paradigm established by ASME B106.1M. Extreme transient resistance events, such as the blades instantaneously jamming against a destructive object, have the potential to initiate rapid fatigue crack propagation through the material's elastic modulus. This could exceed the bending tolerance of the rotary shaft before the motor's thermal protection system has the opportunity to react. Furthermore, the machine's transmission model does not precisely account for internal force damping, specifically V-belt slippage and frictional thermal dissipation, but rather simplifies the analysis by assuming an overall mechanical efficiency exceeding 90%. The anticipated 5% to 15% energy deficit attributed to belt slippage underscores the necessity of employing a Prony brake or dynamometer during the subsequent fabrication phase. Such empirical testing is strictly required to recalibrate the actual mechanical torque coefficients in response to the diverse material fractions inherent in in-situ geotechnical waste.

4. Conclusion

This study has successfully established the theoretical mechanical formulation for a vertical-shaft organic waste shredder featuring dual-hydraulic automation. The design was specifically engineered to address challenges regarding chassis reliability, asymmetric load distribution, and the ergonomic deficiencies inherent in traditional horizontal-cutting maintenance systems. The application of the ASME B106.1M design standard to mitigate fatigue crack propagation demonstrates that the minimum allowable shaft diameter is highly responsive to metallurgical improvements. Transitioning from Mild Steel (σ_y = 207 MPa) to Stainless Steel 316L (σ_y = 317 MPa) significantly reduces the required shaft diameter from 28 mm to 24 mm under a maximum system load of 35 kg. Furthermore, kinematic and rotational dynamic analyses validated a linear power progression, starting from 1.34 kW for light-duty processing up to an operational limit of 4.68 kW. This justifies the requirement for a driving motor with a nominal reserve capacity of 5.50 kW to preclude overloading. The integration of a 0.25 kW hydraulic pumping cylinder for the top-head mechanism proved to be energy-efficient, ensuring that maintenance and screen replacement procedures comply with physical isolation safety protocols without the need for intensive manual assembly. The methodology of this study is primarily constrained by its reliance on mathematical-theoretical computation and static simulation (FEA boundary) definitions, without the support of experimental prototype data. The formulas derived for estimating cutting force resistance assume linear continuity and may discount the complex elasticity and fluctuating moisture fractions within the organic waste matrix. Consequently, this may weaken the precision of estimates when the system is subjected to abrupt, millisecond-scale transient peak loads. To bridge the transition from theoretical modeling to a commercial-scale mechanical implementation, the following research stages are recommended: (1) Explicit Dynamics & DEM: Future studies should employ Explicit Dynamics FEA integrated with a Discrete Element Method (DEM) approach, utilizing geotechnical moisture properties (30%–230%) to simulate destructive shockwaves directly at the blade tips. (2) Prototyping and Standardization: Initiate a physical fabrication phase following the optimized engineering specifications (e.g., the 24 mm SS 316L shaft) while strictly adhering to EN 13683 standards for hopper geometry. (3) Empirical Validation: Conduct field performance evaluations using a Prony brake or dynamometer to extract and validate actual historical power consumption data across diverse green waste materials and to assess the final particle size reduction ratios.

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