METHODOLOGY FOR THE INTEGRATED DESIGN AND STRUCTURAL VERIFICATION OF A TWIN-SHAFT INDUSTRIAL SHREDDER FOR MIXED RECYCLABLE MATERIALS

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Abstract - Twin-shaft industrial shredders are widely used for processing heterogeneous recyclable materials; however, their mechanical design is often based on empirical practice rather than integrated engineering procedures. This study presents a methodology for the integrated design and structural verification of a compact twin-shaft industrial shredder intended for processing wood, polymeric materials, aluminium, and occasional light-steel components. The proposed approach combines functional requirement analysis, analytical mechanical design, component selection, computer-aided modelling, and finite element analysis within a unified engineering framework. The methodology was applied to the design and verification of the drivetrain, cutter assembly, shafts, bearings, and supporting frame. Analytical calculations and numerical simulations confirmed adequate strength, stiffness, and structural integrity under representative operating and overload conditions. The proposed methodology provides a systematic engineering framework for developing compact, reliable, and maintainable twin-shaft shredders prior to prototype manufacturing and can be adapted to similar industrial recycling machines.

Keywords: Mechanical design, Recycling systems, Industrial shredders, Twin-shaft, Design Analysis, Practical framework, FEA.

1. Introduction

The increasing generation of municipal, commercial, and industrial waste has intensified the need for efficient technologies for material recovery and reuse.

The transition from a linear to a circular economy requires recycling systems capable of processing heterogeneous waste streams while reducing landfill disposal, environmental impact, and the consumption of primary raw materials [1–4]. Mechanical size reduction is a critical stage in these systems because it directly affects material handling, separation, transportation, and subsequent recycling operations.

Industrial shredders are widely used for the primary size reduction of plastics, wood residues, electronic waste, metallic scrap, construction debris, and mixed industrial materials. Among the available configurations, twin-shaft shredders are particularly suitable for heterogeneous feedstocks because they operate at relatively low rotational speeds and generate high torque. Their two counter-rotating shafts are equipped with intermeshing cutters that progressively grip, shear, tear, and fracture the processed material. Compared with high-speed granulators and impact-based machines, this operating principle provides improved control of material feeding, lower dust generation, and reduced impact loading [5–9].

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Despite their widespread industrial application, twin-shaft shredders are frequently designed through empirical procedures, manufacturer-specific experience, and iterative prototype modification. The selection of the electric motor, gearbox, shafts, bearings, synchronization gears, cutter geometry, and supporting structure is commonly performed using conservative assumptions. Although such an approach may provide operational reliability, it can also result in oversized components, excessive structural mass, increased manufacturing costs, and inefficient material utilization. These limitations become particularly important in compact machines intended to process materials with substantially different mechanical properties.

Previous studies have investigated individual aspects of shredding equipment, including cutter geometry, fragmentation mechanisms, energy consumption, and numerical simulation of separate machine components [10–14]. Finite element analysis has been used to identify stress concentrations, deformation zones, and potential failure locations in shafts, cutters, pins, and supporting elements. However, the available studies commonly address these elements independently and do not establish a unified relationship between the functional requirements, cutting assembly, power-transmission system, synchronization mechanism, shaft-support arrangement, and structural frame [15],[16]. Consequently, a systematic methodology integrating analytical mechanical design with numerical structural verification remains necessary.

The processing of mixed recyclable materials introduces additional engineering challenges because wood, polymeric materials, aluminium, and occasional light-steel components exhibit substantial differences in hardness, ductility, shear strength, fracture behaviour, and cutting resistance. These variations directly affect the required cutting force, transmitted torque, shaft stresses, bearing reactions, gearbox loading, and structural deformation [17–20]. The shredder must therefore be considered as an integrated mechanical system in which the cutting assembly, drivetrain, bearing arrangement, synchronization mechanism, and supporting structure interact under nominal and overload conditions. Analytical calculations complemented by computer-aided design and finite element analysis make it possible to evaluate these interactions before prototype manufacturing and to identify critical structural regions requiring geometric modification or localized reinforcement [21–23].

The objective of this study is to develop and apply an integrated engineering methodology for the design and structural verification of a compact twin-shaft industrial shredder intended for processing mixed recyclable materials. The proposed methodology combines the definition of functional and operational requirements, cutter-system design, analytical dimensioning and selection of the drivetrain and principal mechanical components, computer-aided modelling, and finite element verification of the supporting structure. The analytical stage includes the determination of transmitted power and torque, cutting forces, gear loading, shaft stresses, and bearing reactions. The resulting loads are subsequently applied in the numerical simulations, thereby ensuring consistency between mechanical dimensioning and structural assessment. The methodology is validated by verifying the strength and stiffness of the developed machine configuration under static and representative operational loading conditions. The principal contribution of the study is a unified engineering procedure in which the drivetrain, cutting assembly, shaft supports, synchronization components, and structural frame are evaluated as interacting subsystems rather than as isolated machine elements. The proposed framework provides a practical basis for developing compact, maintainable, and structurally reliable twin-shaft shredders and can be adapted to machines with different processing capacities and recyclable material compositions.

2. Integrated Mechanical Design

2.1 Main Points

The proposed twin-shaft industrial shredder was developed following an integrated engineering methodology combining analytical mechanical design, component selection, computer-aided design (CAD), and finite element analysis (FEA). The adopted approach considers the shredder as an integrated mechanical system in which the drivetrain, cutting assembly, supporting structure, and safety components are designed and verified as a unified engineering solution.

The methodology comprises the sequential stages presented in the following sections, including the definition of the design requirements, development of the general layout, design of the principal mechanical components, analytical calculations, structural verification, and final validation of the proposed machine.

2.2 Design Requirements

The first stage of the proposed design methodology establishes the operational and functional requirements of the industrial shredder. These requirements provide the foundation for the subsequent engineering calculations, component selection, structural analysis, and optimization procedures. Their definition ensures that the final machine configuration satisfies the intended operational objectives while maintaining reliability, operational safety, and cost-effectiveness.

The primary requirement concerns the identification of the materials intended for processing. The proposed shredder is designed to process heterogeneous recyclable materials, including wood, polymeric materials, aluminium, and occasional light steel components (Figure 1). The selected materials exhibit substantial variations in mechanical properties, including hardness, fracture toughness, ductility, density, and shear strength. These differences result in significant variations in cutting forces and dynamic loading during the shredding process. Consequently, the proposed shredder must be capable of withstanding the most severe operating conditions while ensuring stable operation and efficient processing of all target materials.

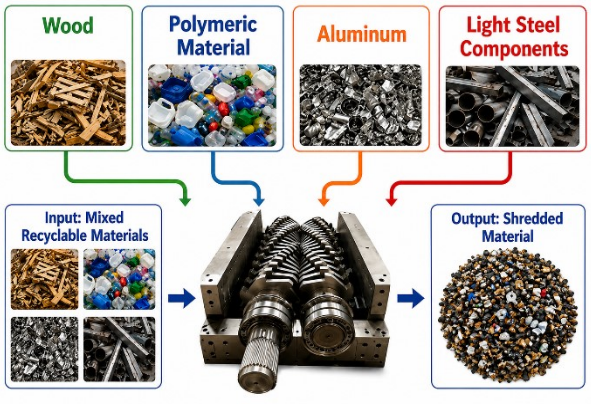

Figure 1: Heterogeneous recyclable materials [24]

The second design requirement is the specification of the required processing capacity. Based on the intended industrial application, the nominal production capacity was established within the range of approximately 300÷500 $kg \cdot h^{-1}$. This requirement governs the selection of the drive system, reduction gearbox, shaft rotational speed, cutter dimensions, hopper capacity, and structural stiffness of the supporting frame. The selected capacity represents an appropriate compromise between productivity, energy consumption, machine dimensions, and manufacturing cost.

Another important design consideration is the required particle size of the shredded material (Figure 2). The target particle size depends on the subsequent recycling process, including material separation, transportation, washing, pelletizing, or remelting. The final particle dimensions are primarily governed by the cutter geometry, cutter spacing, shaft rotational speed, and the interaction between adjacent cutting elements. These parameters must therefore be optimized to ensure uniform material fragmentation while minimizing energy consumption.

Operational safety represents another essential design requirement. Owing to the high torque generated by the twin-shaft shredding system, appropriate safety measures must be incorporated into both the mechanical structure and the control system.

These measures include a fully enclosed cutting chamber and a reinforced hopper that prevents operator access to the rotating cutters.

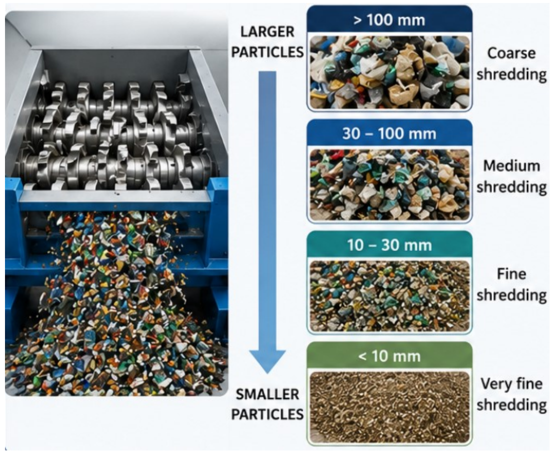

Figure 2: Particle size

Additional safety provisions include an emergency stop function, variable-frequency-drive overload protection, automatic shaft reversal in the event of cutter blockage, and adequate guarding of all rotating transmission components. The implementation of these measures enhances operator safety while reducing the risk of mechanical failure under overload conditions.

Ease of maintenance is another important design consideration. Since industrial shredders operate under severe loading conditions and are continuously exposed to abrasive contaminants, the principal mechanical components should allow convenient inspection, lubrication, adjustment, and replacement. Particular attention should therefore be given to the accessibility of bearings, cutter assemblies, synchronization gears, couplings, and transmission components [25], [26].

Based on these functional requirements, the principal design specifications adopted in the present study are summarized as follows:

- Processing of mixed recyclable materials, including wood, plastics, aluminium, and occasional light steel components;
- Nominal processing capacity of approximately 300–500 $kg \cdot h^{-1}$;
- Low-speed, high-torque twin-shaft shredding principle;
- Continuous and stable operation under variable loading conditions;
- Compact structural configuration with optimized power transmission;

- Safe operation through mechanical guarding and overload protection;
- Modular construction to facilitate maintenance and future component replacement.

The defined design requirements establish the design constraints for the proposed machine and provide the basis for the subsequent analytical design, optimization, and structural verification of the twin-shaft industrial shredder.

2.3 General Design and Dimensions

After establishing the functional and operational requirements presented in the previous section, the conceptual mechanical layout of the industrial shredder was developed. The overall configuration was designed to provide a compact, rigid, and maintainable structure capable of processing heterogeneous recyclable materials under the specified operating conditions. Particular attention was given to the arrangement of the drivetrain, shredding chamber, hopper, supporting frame, and operator access in order to ensure reliable operation and convenient maintenance.

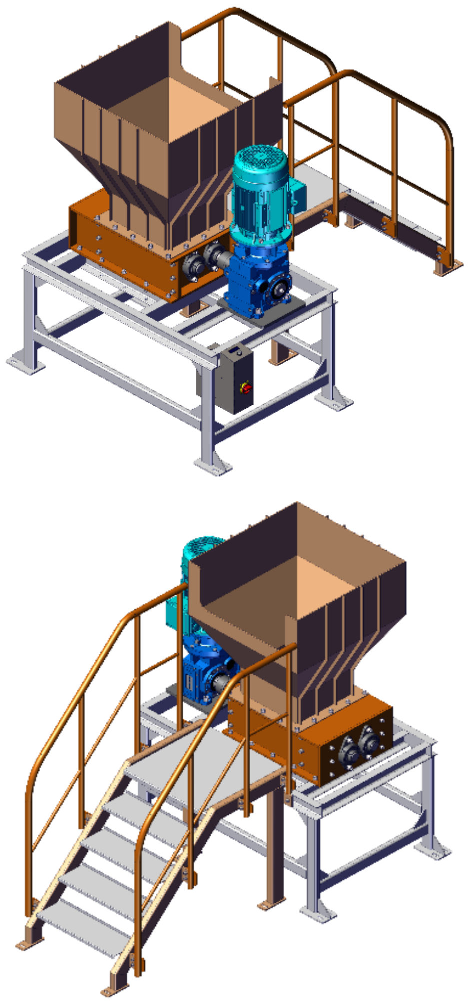

Figure 3: General Design of the Industrial Shredder

In Figure 3 is shown the general design of the shredder machine. In Figure 4 are the main dimensions of the whole construction, including the stairs and the frame. The size and the setup of the whole assembly is considered for easy use and maintenance.

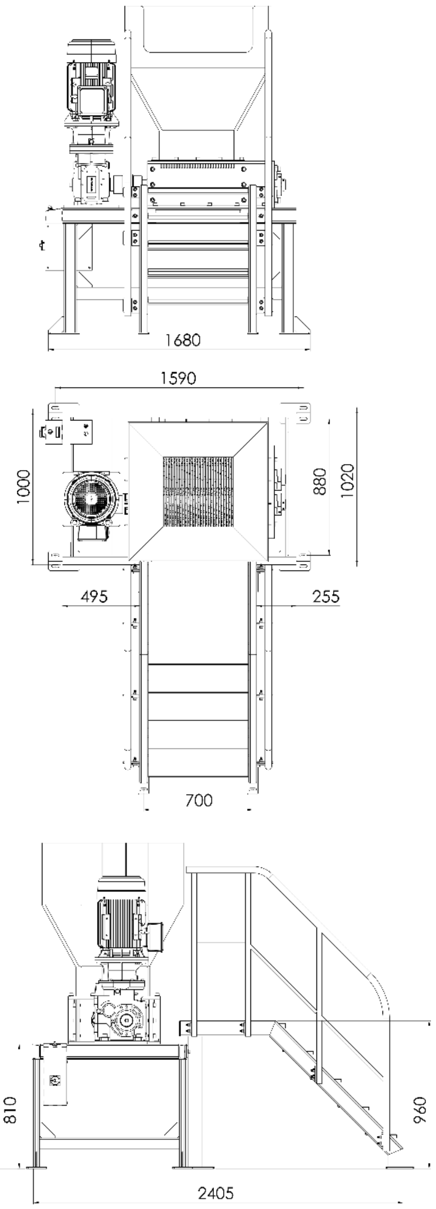

Figure 4: General Dimensions of the Industrial Shredder

2.4 Main Components of the Machine

Table 1. Shredder main components

Position	Semi-Assembly
1	Shredder Box
2	Hopper 900x900
3	Motor-reducer group
4	Frame support
5	Stairs

In Figure 5 are shown the main components (main sub-assemblies) of the shredder machine. They are described in Table 1.

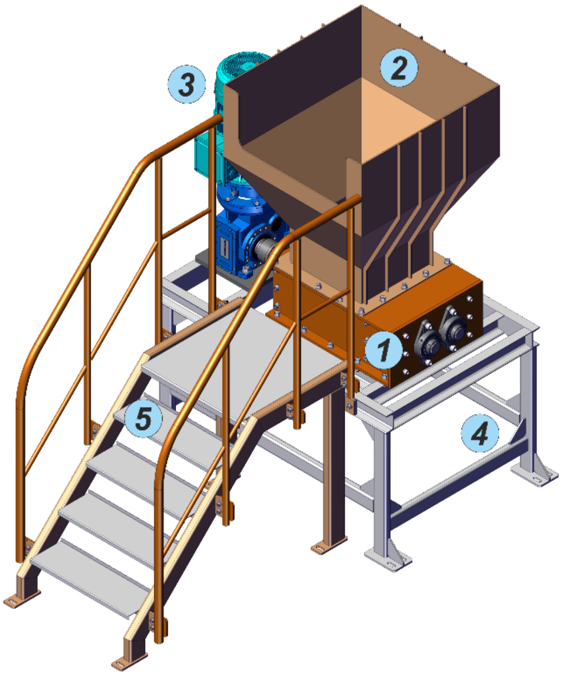

Figure 5: General Design of the Industrial Shredder

Figure 6 shows the structure of the shredder box and all the working components of the shredder machine. They are described in Table 2.

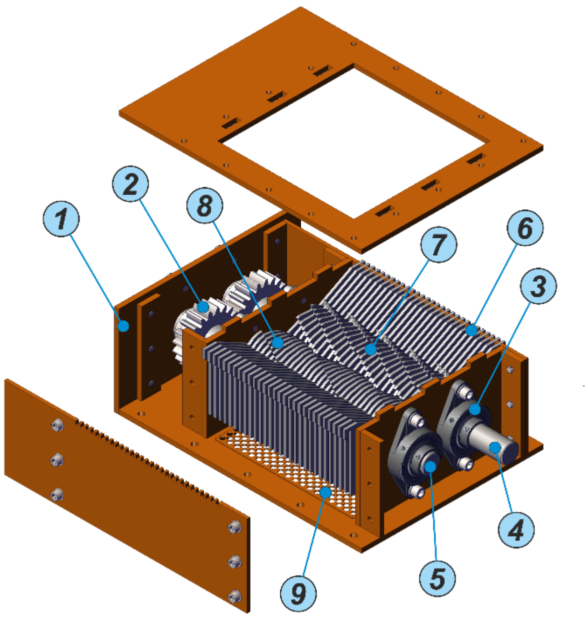

Figure 6: Shredder box elements

1 – Welded box – The base box which is housing all the elements, and also the cover plates.
2 – Gear z=20 – The gears that transmit the torque to the second shaft.

Table 2. Shredder box components

Position	Components
1	Welded box
2	Gear z = 20
3	Bearing SKF UCFL212
4	Input shaft
5	Second shaft
6	Cutter wall plate
7	Cutter 1
8	Cutter 2
9	Sorting net

3 – SKF UCFL212 bearing – Set of six bearings working on both shafts.
4 – Input shaft – The main shaft that receives the power and torque from the motor-reducer.
5 – Second shaft – Second shaft which concludes the twin-shaft assembly.
6 – Cutter wall plate – Cutter plates mounted to the wall of the welded box for extra work.
7 – Cutter 1 – Working Cutter 1, input shaft, different geometry.
8 – Cutter 2 – Working Cutter 2, second shaft, different geometry.
9 – Sorting net – A net that sorts the size of the particles.

2.5 Cutter Geometry

The cutter assembly represents the primary working subsystem of a twin-shaft industrial shredder and plays a decisive role in determining the machine's overall performance, efficiency, and operational reliability.

Unlike conventional cutting tools designed for continuous machining processes, shredder cutters are subjected to complex loading conditions involving repeated impact, compression, shearing, tearing, and localized plastic deformation. Consequently, the geometrical configuration of the cutter directly influences the magnitude of the cutting forces, power consumption, particle size distribution, wear resistance, and the service life of the entire shredding system.

As a result of the various used materials, the cutter geometry must provide an appropriate balance between efficient material engagement, stable material feeding, and sufficient structural strength to withstand cyclic loading and shock impacts. Parameters such as cutter diameter, tooth profile, hook angle, rake angle, clearance angle, cutter thickness, spacer arrangement, and intermeshing depth collectively determine the effectiveness of the shredding process and the resulting mechanical loading of the drivetrain components.

In addition to influencing the cutting mechanism, the cutter geometry directly affects the transmitted torque, shaft bending moments, bearing reactions, and synchronization gear loading. Therefore, the cutter design cannot be considered independently from the remaining mechanical components of the shredder. Instead, it must be integrated into the overall design methodology.

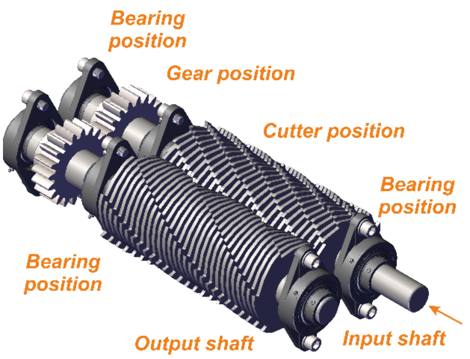

Figure 7: Cutter elements position in the assembly

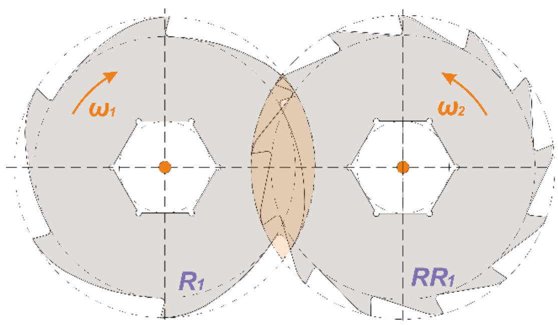

Figure 8: Cutter R1 and cutter RR1 contact overlap

In Figure 7 is shown the assembly of the working cutter components. There are two different types of cutters with different number of cutting teeth for a better working condition.

Figure 8 shows the geometry of the first cutter – R1 and the second – RR1 and most importantly, the contact spot, formed by the radial distance between the two shafts. This guarantees the proper cutting of different type of components, with different properties, geometry, and size.

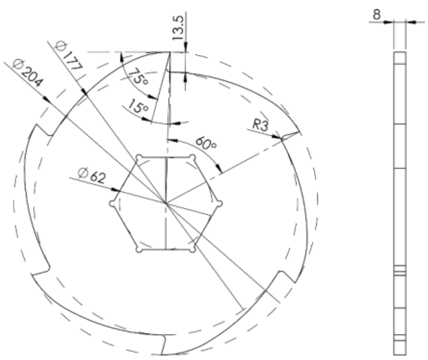

Figure 9: Cutter R1 technical drawing

Table 3. Cutter geometrical parameters

Parameter	**Symbol**	**Description**	**Value [mm] $[^0]$**
Outer diameter	D	Cutter diameter	204
Inner bore (circle)	d	Shaft bore diameter	62
Cutter thickness	t	Individual cutter thickness	8
Number of teeth	z	Number of cutting hooks	6 (R1) 13 (RR1)
Tooth height	h	Cutting depth	13.5
Hook angle	γ	Cutting angle	15^0
Tooth shape angle	α	Geometry of tooth	75^0
Cutter overlap	e	Intermeshing maximum depth	44

Figure 9 presents a technical drawing of the main parameters of the R1 cutter, with Table 3 describing the parameters. Table 3 is also relevant for cutter RR1 – Figure 10, because of the similar geometry. The only difference is the number of teeth – Z_2 = 13.

The specific geometry of the cutter is using a hexagonal form for the bore because of the easy and practical solution for transferring the torque between the shaft and the cutters.

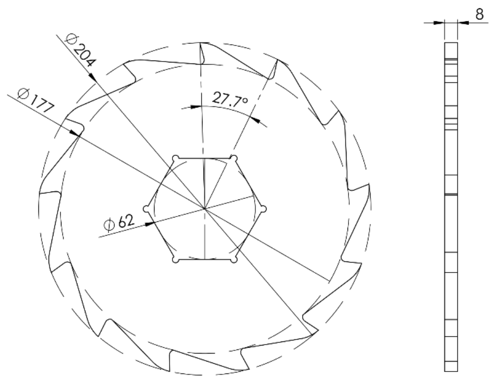

Figure 10: Cutter RR1 technical drawing

For the proper cutting conditions and to ensure the cutters work and efficiency, several different geometries for the hexagon shape are considered, with a 5-degree change on each step – Figure 11.

Position 1 is the first cutter, with Position 7 the middle (yellow colour), and Position 12 is the last, which made a proper 60-degree turn to match the starting position.

It is visible in the hexagon shape, marked with different colours, how the step of 5 degrees is performed.

This same logic is applied for cutters RR1–RR2. Two sets by 12 cutters are applied for both shafts. This guarantees the working capacity of the machine and also ensures the small size of recycled materials.

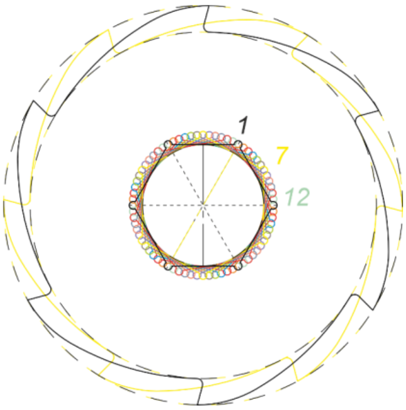

Figure 11: Cutter R1 – R12

The final shape of the Cutter assembly is show in Figure 12.

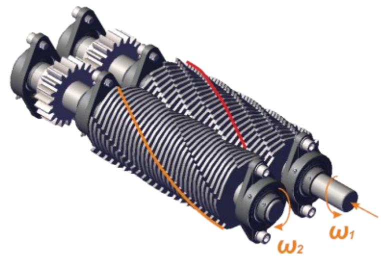

Figure 12: Cutting curve for both Cutters

For the application (wood + plastic + aluminium + steel pieces), the cutter material is one of the most important design decisions in the entire shredder.

The material should have the following properties:
- High wear resistance
- Good toughness (to avoid tooth chipping)
- Heat-treatability
- Good laser cutting capability
- Reasonable cost

In Table 4 are suggested several types of material, suitable for usage of the cutters, each with their own advantages / disadvantages.

Table 4. Cutter geometrical parameters

Material	Laser cutting	Wear resistance	Toughness
Hardox 450	Excellent	Very good	Excellent
Hardox 500	Excellent	Excellent	Very good
Hardox 550	Good	Excellent	Good
42CrMo4	Excellent	Moderate	Excellent
X153CrMoV12	Possible	Excellent	Poor–moderate
Tool steel S7	Possible	Very good	Excellent

Hardox 500 – Chosen Material

- Specifically designed for abrasive and impact-loaded applications;
- Can be purchased as plate and laser cut directly;
- No complicated heat treatment required after cutting;
- Good combination of hardness (~500 HB) and toughness;
- Widely used in shredders, crushers, and recycling equipment;
- Less likely to chip than other tool steel.

3. Calculations and Structural Verification

3.1 Power Calculation

In Table 5 are presented the already chosen parameters of the shredder, which defines its working principle.

Table 5. Defined parameters

Parameter	Value
Motor power / P	15 kW
Motor type	4 pole
Motor speed / n	1470 rpm
Gear ration / i	49.8 : 1
Gearbox type	Helical bevel
Gearbox efficiency / η	98 %

Motor torque – T [Nm]

$$T = \frac{9550.P}{n} \quad (1)$$

Output rotational speed - n_2 [rpm]

$$n_2 = \frac{n_m}{i} \quad (2)$$

Output torque - T_2 [Nm]

$$T_2 = T.i.\eta \quad (3)$$

Angular velocity - ω_2 [s^{-1}]

$$\omega_2 = \frac{\pi.n_2}{30} \quad (4)$$

$T = 97.45$ Nm; $n_2 = 29.5$ rpm; $T_2 = 4756$ Nm; $\omega_2 = 3.1$ s^{-1}

3.2 Gear and Shaft Calculation

In Table 6 are presented the already chosen values of the shredder gears and the shaft which will affect the calculation of the power transmission.

Table 6. Defined values

Parameter	Symbol	Value
Spur gear number of teeth	$z_1 = z_2$	20
Spur gear module	m	8
Gear thickness	b	80 mm
Pitch diameter	$d_1 = d_2$	160 mm
Shaft material	-	42CrMo4
Shaft diameter (inner circle)	d	60 mm
Output torque	T_2	4756 Nm

Tangential force - Ft

$$Ft = \frac{2.T_2}{d_1} \quad (5)$$

Radial force - Fr

$$Fr = Ft.tan20^0 \quad (6)$$

Shaft Torsional Stress - τ

$$\tau = \frac{16.T_2}{\pi.d^3} \quad (7)$$

Ft = 59,45 kN; Fr = 21,64 kN; τ *= 112,2 MPa;*

$$[\tau] > \tau \quad (8)$$

where: [τ] - Allowable Shear Stress

σ- Yield Strength (for 42CrMo4) = 900 MPa

$$[\tau] = 0.3.\sigma \quad (9)$$

[τ] = 270 MPa

$$270\ MPa > 112,2\ MPa \quad (10)$$

$$k = \frac{[\tau]}{\tau} \quad (11)$$

where: k – safety factor; *k = 2,41*

Gears:

$$\sigma_F = \frac{Ft}{b.m.Y} \quad (12)$$

where: σ_F – bending stress [MPa]
Y - Lewis Form Factor; Y = 0.322 (for gear 20 teeth)

σ_F *= 288,5 MPa (acceptable for 42CrMo4)*

Pitch line velocity - ϑ *[m/s]*

$$\vartheta = \frac{\pi.d.n}{60} \quad (13)$$

ϑ *= 0,25 m/s*

A very low pitch line velocity means dynamic gear effects are small.

3.3 Cutting force

For the materials used in the shredder, the most useful property is the shear strength, which affects the cutting conditions and the cutting forces. Typical values are shown in Table 7.

Table 7. Materials properties

Material	Shear strength
HDPE	20–30 MPa
PP	25–35 MPa
PET	35–50 MPa
Softwood	5–20 MPa
Hardwood	15–30 MPa
Aluminium 6061	120–170 MPa

For a twin-shaft shredder, the cutter does not shear the entire 8 mm thickness simultaneously. The material fails progressively as the two cutter teeth rotate. Therefore, the engaged material thickness is usually much smaller than the cutter thickness.

For industrial shredders processing aluminium and occasional steel:

- Aluminium 6061-T6 – 5 mm plate – 150 MPa (τ_S)
- Steel (S235) – 5 mm plate – 220 MPa

For both (aluminium and steel) the effective shear area is calculated:

$$A = t.s \quad (14)$$

where: A – shear area (mm²);
t – cutter thickness (mm);
s – material thickness (mm);
τ_S – shear strength (MPa)
A = 40 mm²

$$F_C = \tau_S .A \quad (15)$$

where: F_C – cutting force (N)

For Steel (S235) - F_C *= 8800 N*

If we use a safety factor – $K_S = 2$;

F_C ***= 17 600N***

$$T_C = F_C.R \quad (16)$$

where:
T_C – required torque [Nm];
R – cutter radius [m];
T_C *= 1800 N*

T_2 / T_C *= 4756 / 1800 = 2.6* (extra safety margin)

$$R_A = \frac{F_t.x_1}{x_1 + x_2} \quad (17)$$

$$R_{B1} = \frac{F_t.x_2}{x_1 + x_2} \quad (18)$$

$$R_{B2} = \frac{F_c.(x_3 + x_4)}{x_3 + x_4 + x_5 + x_6} \quad (19)$$

$$R_{B2} = R_C \quad (20)$$

R_A = 33,38 kN; $R_B = R_{B1} + R_{B2}$ = 26,07 + 8,8 = 34,87 kN

R_C = 8,8 kN

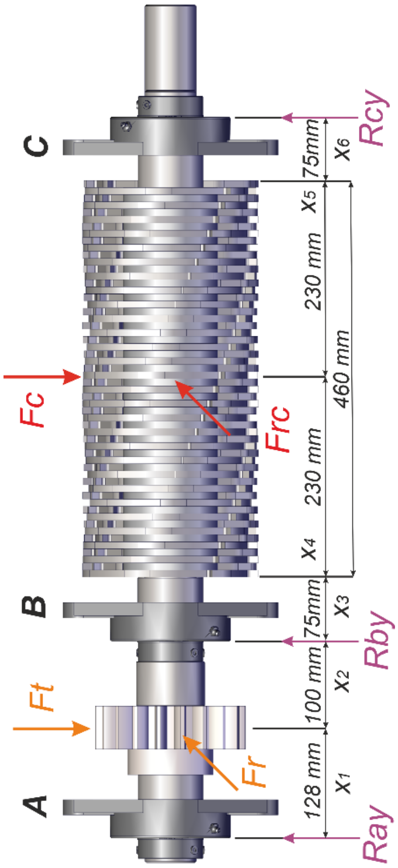

Figure 13: Position and measures of the main components

In Figure 13 is shown the main forces that are applied on the shaft and the cutters. All the reactions in the bearings are in the limits of the chosen bearings.

Maximum bending moment – M [kNm]

$$T_C = (x_3 + x_4).R_{max} \quad (21)$$

where: $R_{max} = R_B$

M = 10,64 kNm

Bending stress – σb [MPa]

$$\sigma b = \frac{32.M}{\pi.d^3} \quad (22)$$

σb = 502 MPa

$$\sigma = \sqrt{\sigma b + \tau} \quad (23)$$

σ = 514,34 MPa < [σ]

The design construction will withstand the calculated forces and this result is the most unfavourable with the biggest applied forces, using steel material. With the aluminium, wood and plastic, the forces and the stress will be much smaller.

The analytical calculations confirmed that the proposed twin-shaft industrial shredder satisfies the mechanical strength and operational requirements for processing heterogeneous recyclable materials.

The analytical evaluation of the cutting process was performed considering the least favourable operating condition, in which the shredder is required to process occasional mild steel components in addition to wood, polymeric materials, and aluminium. Since mild steel exhibits the highest shear strength among the considered recyclable materials, it generates the maximum cutting resistance and therefore represents the critical design case. By adopting this conservative approach, all structural components of the machine were designed to withstand the highest expected operational loads.

3.4 Hopper Capacity

1. Upper section
- Top opening: 888 × 888 mm
- Bottom opening: 448 × 448 mm
- Height: 400 mm

2. Lower transition section
- Constant cross-section: 448 × 448 mm
- Height: 200 mm

Total H = 600 mm;
V_1 = 0,182 m^3 (from model verification)
V_2 = 0,04 m^3 (from model verification)
V = 0, 222 m^3

Usually, the industrial hoppers do not get filled in 100 % and there is a filling coefficient, but here we do not apply it, because there is a extra section that we do not use.

Effective cutting area – Ac [m^2]

$$A_C = l_1.l_2 \quad (24)$$

where: l_1, l_2 – dimension of the cutter box

A_C = 0,2025 m²

A twin-shaft shredder does not convey the full chamber volume every revolution. Only part of the cutting chamber is effectively occupied by material because of voids, intermeshing cutters, and intermittent feeding. We choose 12 % capacity filled. In Table 8 is shown the capacity with different type of materials.

Table 8. Hopper Capacity for Different Materials

Material	Bulk Density (kg/m³)	Hopper Capacity (kg)
Plastic bottles (loose)	50	11.1
Mixed plastic	250	55.5
Plastic flakes	350	77.7
Wood chips	250	55.5
Hardwood pieces	400	88.8
Mixed municipal recyclables	300	66.6
Aluminium scrap	550	122.1
Mild steel scrap	850	188.7

$$Q_V = A_C.p.n_2.60.\eta_V \quad (25)$$

Where: Q_V - theoretical volumetric flow [m³/h]

p – cutter pitch [m] = 0.014 m

η_V – capacity – 12 % - 0.012

Q_V = 0,602 m³/h

In Table 9 is the working capacity after the calculation and the material used.

Table 9. Working Capacity

Material	Density (kg/m³)	Capacity (kg)
Plastic bottles (loose)	50	30.1
Mixed plastic	250	150.5
Plastic flakes	350	210.7
Wood chips	250	150.5
Hardwood pieces	400	240.8
Mixed municipal recyclables	300	180.6
Aluminium scrap	550	331.1
Mild steel scrap	850	511.7

4. Finite Element Analysis

4.1 Introduction

Finite Element Analysis (FEA) represents an essential stage in the structural verification of the proposed twin-shaft industrial shredder. While the preceding analytical calculations provide an initial assessment of the strength and loading conditions of the principal machine components, the simulation creates an evaluation of the stress distribution, deformation behaviour, and overall structural stiffness under realistic operating conditions.

The finite element method is particularly suitable for complex welded structures and machine assemblies.

The primary objective of the simulation is to verify that the supporting frame possesses sufficient stiffness and mechanical strength to withstand both the static and operational loads generated during the shredding process. The loading conditions were derived directly from the analytical calculations presented in the previous section, including the weight of the rotating cutter assembly, the calculated cutting resistance corresponding to the worst-case loading condition, and the operational forces transmitted through the shaft and bearing supports. By using analytically determined loading parameters, consistency between the theoretical design methodology and the numerical verification is ensured.

The numerical simulations were performed under linear elastic material assumptions, as all calculated stresses were expected to remain below the yield strength of the structural material. The analysis focused primarily on the determination of the total deformation, equivalent (von Mises) stress distribution, and the overall structural response of the supporting frame under static and operational loading conditions. Particular attention was paid to regions susceptible to stress concentration.

The obtained numerical results were subsequently compared with the analytical calculations to verify the consistency of the proposed design methodology. This combined analytical–numerical approach increases the reliability of the design process and provides a comprehensive structural assessment of the proposed industrial shredder prior to prototype manufacturing.

4.2 FEA under Weight

Figure 14 shows the initial setup for the weight load analysis. We applied the weight of the shredder box + hopper (550 kg – 5500 N) and the weight of the motor-reducer group (150 kg – 1500 N (rounded)).

The Mesh image is shown in Figure 15.

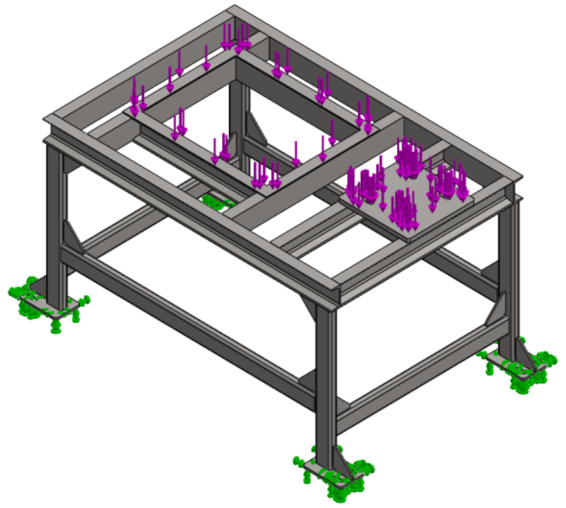

Figure 14: FEA setup of frame support – simplified model

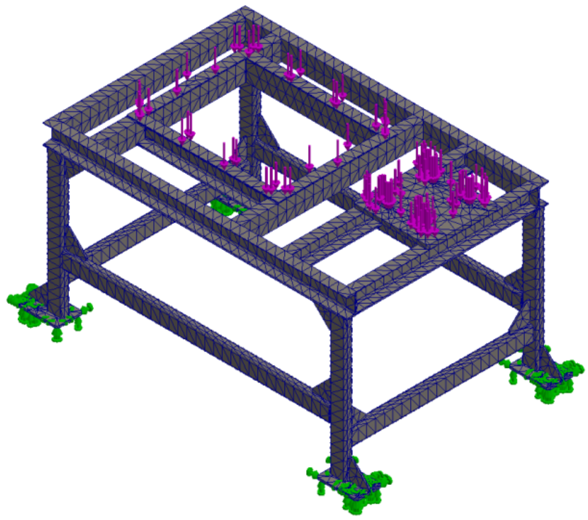

Figure 15: FEA mesh

After running the study, the stress analysis and the displacement are shown in Figure 16 and Figure 17. It is pointed out that the stress correlates with the previous calculations.

The displacement at the position of the motor-reducer group is very small, but also the weight load on the position of the shredder box is negligible.

The numerical results are in good agreement with the preceding analytical calculations and confirm that the supporting frame exhibits adequate stiffness and load-carrying capability under gravitational loading. The maximum displacements remain very small and are not expected to affect the alignment of the mechanical components. Moreover, the stress distribution indicates that the applied loads are safely accommodated by the frame without compromising its structural integrity.

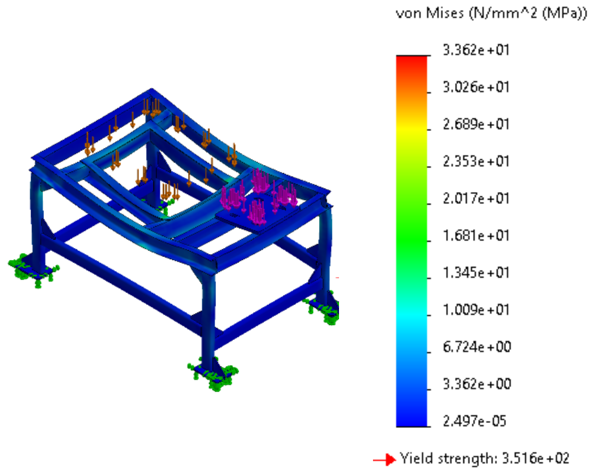

Figure 16: FEA – stress results

Maximum Bending stress – 33,62 MPa;

Maximum Displacement – 0,23 mm

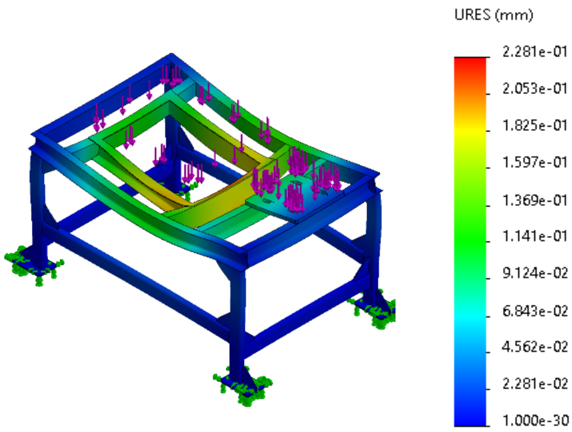

Figure 17: FEA – displacement results

4.2 FEA under Load (Cutting force + Weight)

Figure 18 shows the initial setup for the analysis, including the cutting force + weight of the shredder box. This is the area that is the most loaded under different conditions and under usage of different materials. The simulation is performed with the maximum cutting force calculated in Chapter 3 - F_c = ***17 600N.***

With the load from the weight of the shredder box + hopper we calculate the maximum loading force – ***F = 23 100 N***.

After running the study, the stress analysis and the displacement are shown in Figure 19 and Figure 20. It is pointed that the stress correlates with the previous calculations.

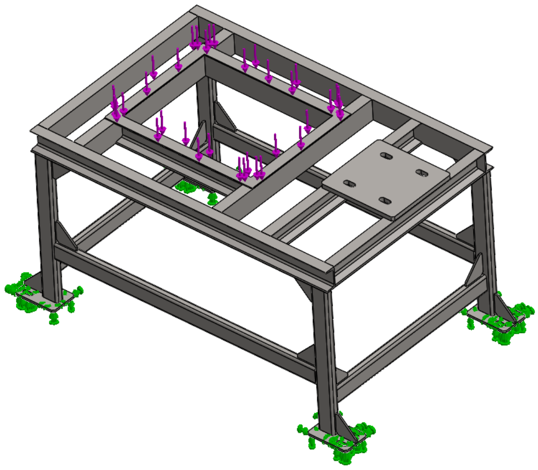

Figure 18: FEA setup of frame support with force applied – simplified model

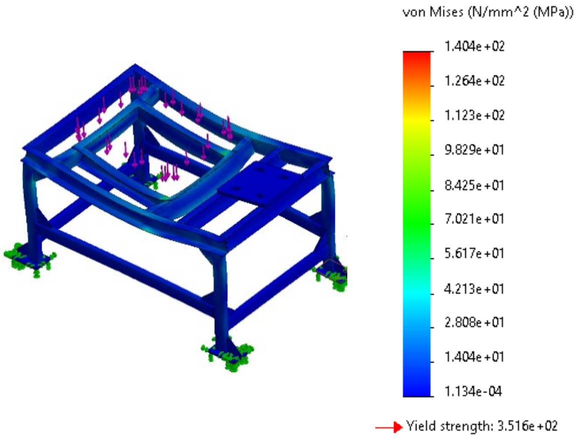

Figure 19: FEA – stress results with maximum load

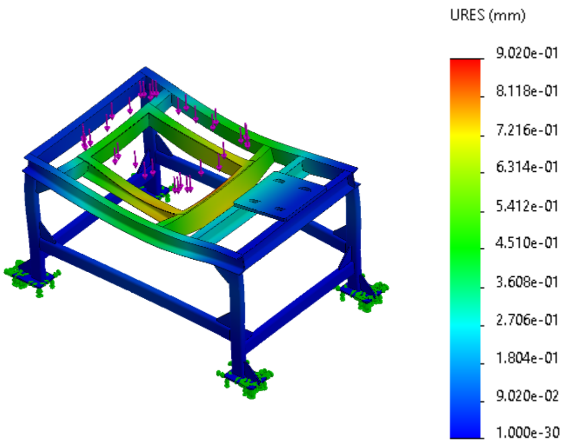

Figure 20: FEA – displacement results with maximum load

Maximum Bending stress – 140,4 MPa and Maximum Displacement – 0,9 mm.

5. Design Validation

The proposed design methodology was validated through analytical calculations and finite element simulations to verify the functional and structural performance of the developed shredder.

The analytical results demonstrated that the selected drive system, shafts, synchronizing gears, bearings, and coupling safely transmit the required power and torque, while the calculated cutting forces, shaft stresses, and safety factors confirmed adequate mechanical strength. Finite element analysis verified sufficient stiffness and structural integrity of the supporting frame, with equivalent stresses and deformations remaining within acceptable engineering limits.

Overall, the proposed approach integrates functional requirements, analytical design, and numerical verification into a coherent engineering procedure for twin-shaft industrial shredders.

6. Conclusions

This study presented an integrated methodology for the design and structural verification of a twin-shaft industrial shredder intended for processing heterogeneous recyclable materials. The proposed approach combines analytical design with finite element analysis to provide a systematic procedure for developing reliable and structurally efficient shredding equipment.

The design procedure includes the definition of functional requirements, cutter geometry, transmission design, analytical sizing of the principal mechanical components, and numerical verification of the supporting frame. Conservative design conditions based on mild steel shredding ensured safe operation for materials with lower mechanical resistance. Finite element analysis confirmed adequate frame stiffness and strength, with stresses below the allowable limits and deformations within acceptable engineering tolerances.

The proposed methodology enables accurate dimensioning of the principal mechanical components, identification of critical structural regions, and verification of machine performance before prototype manufacturing. It can also be applied to twin-shaft shredders with different processing capacities and material compositions.

Future work will focus on experimental validation of the prototype, optimization of cutter geometry based on wear behaviour, dynamic loading analysis under transient operating conditions, and multi-objective optimization to further improve structural efficiency and energy consumption.

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