

Design and Numerical Assessment of an Automated, Sensor-Supervised PET Bottle Shredding System for Small-Scale Recycling

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Abstract

This study presents the design and numerical assessment of an automated, sensor-supervised PET bottle shredding system for small-scale recycling, where manual feeding exposes the operator to the blade assembly and delivers material unevenly. An inclined cleated conveyor is coupled to a single-shaft shearing mechanism under near-infrared, Hall-effect and motor-current supervision, with a stop-reverse-retry jam-protection sequence. Requirements from a structured survey (n = 50) were mapped to technical specifications by quality function deployment, and materials and processes were selected by weighted digital logic. Six components were evaluated by static structural finite element analysis and cross-checked against independent closed-form calculations at the as-built geometry. The cross-check located a factor-of-ten torque error in the source shaft model; correcting it raises the shaft equivalent stress from 20.91 to 23.0 MPa, a 10% change, since bending governs at the Ø65 mm journal. Against the yield strengths of the specified materials, the minimum static factor of safety across the assembly is approximately 8 at the rotating cutting blade, with the shaft at 13.5 and a modified-Goodman fatigue factor of safety of 6.5. Mounting the blades on an 80 mm across-flats hexagonal section removes the keyed blade interface that governs comparable single-shaft designs. A five-year capacity ramp model gives a base-case net present value of BDT 36.2 crore, positive in all 10,000 Monte Carlo iterations, and per-unit processing is estimated at 140 t PET yr⁻¹, a net avoidance potential of 236 t CO₂e yr⁻¹. The work provides a design-stage basis and ranked verification programme for prototype construction.

Keywords

Machine design; finite element analysis; automated shredding; PET recycling; techno-economic assessment

Author Contributions

Tasfia Rubayyat: Conceptualization, methodology, software and control-system development, engineering design, formal analysis, investigation, visualization, writing – original draft, writing – review and editing, and overall manuscript development. Syed Samin Fuad: Investigation, survey analysis, material and manufacturing-process selection, design development, and economic analysis. Md. Saif Sadman: Investigation, functional analysis, material and manufacturing-process selection, and design development. Ali Iman Bin Murtaza: Investigation, survey/customer-requirement analysis, material and manufacturing-process selection, and economic analysis.

1 Introduction

Global plastic production has risen rapidly since the middle of the twentieth century, while only a small fraction of discarded plastic has been recycled; most of the remainder has been landfilled, incinerated or released to the environment [1]. Polyethylene terephthalate (PET) bottles form a large and readily recyclable part of this stream, and reprocessing them begins with reducing collected bottles to relatively uniform fragments for washing, extrusion or pelletizing. Size reduction sits at the front of every PET material cycle, and in the small-scale operations that handle much of the collected material in South Asia and other developing-economy settings it is the stage where capacity is most often constrained.

In these settings, plastic waste is commonly processed by small, informally operated shredders that are fed by hand, which brings the operator's hands close to the rotating blade assembly. Manual feeding has been identified as a source

of injury risk, and uneven material flow contributes to jamming and unplanned downtime [2, 3]. The structured survey conducted for the present study confirms the practical weight of this issue: 86% of respondents agreed or strongly agreed that the manually fed shredders they use demand excessive manual effort.

Existing low-cost shredder designs have concentrated on the cutting chamber, covering blade geometry, shaft torque and housing strength, while feeding has generally remained manual. Early work emphasized mechanical feasibility and low fabrication cost ahead of structural optimization [2], and a review of single- and dual-shaft designs found automated feeding and safety sensing to be uncommon [3]. Where automation has been introduced, it has usually taken the form of a single material-presence trigger that starts and stops the shredding operation, leaving controlled feeding, sensing and jam recovery to be handled separately [4, 5].

Structural assessment follows a similar pattern of focused coverage. Component-level finite element studies have examined the blade, reporting a von Mises concentration approaching 201 MPa near the blade tip [6], or the shaft under combined bending and torsion [4], with the frame, housing and feeding system left outside the model. Broader analyses have addressed rotor dynamics through modal, harmonic and response-surface optimization [7], and a dual-shaft machine for thermoplastic off-cuts has been evaluated at both blade and shaft, reaching 323.3 MPa and 90.94 MPa respectively [8]. These studies stop at the structural boundary, leaving manufacturing cost, break-even volume, the economics of scaling production, and the material diverted or emissions avoided by the equipment open for the present work to address.

Two adjacent bodies of work inform the present design. Motor current has been demonstrated as a non-intrusive indicator of changing mechanical load in motor-driven mechanisms, which supports its use as an input to jam detection [9]. Machinery-safety practice, including the principles of ISO 12100 [10], favours interlocking and automatic hazard reduction over reliance on operator behaviour, and sensor-supervised feeding follows that hierarchy directly.

Within the literature reviewed for this study, few publications bring automated, sensor-supervised conveyor feeding, multi-component structural assessment of the complete assembly, and a multi-year economic assessment together in one design methodology for locally manufacturable equipment. The gap matters because safety, structural reliability and manufacturing cost are coupled design constraints rather than independent ones: a structurally sound machine priced out of reach stays unbuilt, while a low-cost machine with thin structural evidence carries risk an operator should not have to accept. A single-year break-even calculation, likewise, says little about whether a design holds up as production expands toward market demand.

The question addressed here is whether an automated, sensor-supervised small-scale PET shredding system can deliver a more controlled and safer processing architecture while remaining structurally and economically sound. This paper presents the design and numerical assessment of such a system, referred to throughout as the proposed system. It combines an inclined cleated conveyor with near-infrared bottle detection, Hall-effect shaft-speed sensing and motor-current monitoring for jam detection, integrated with a single-shaft shearing mechanism. The mechanical design was evaluated by static structural finite element analysis across six components, supplemented by independent analytical checks of shaft sizing, key and bearing adequacy, natural frequency, fatigue and load sensitivity, and then assessed economically and environmentally through break-even, five-year discounted cash-flow and scenario-based analyses. The work is a design-stage numerical and scenario-based study, and it fixes the specifications and verification priorities that a first instrumented prototype is built to confirm.

The contributions are threefold. First, an integrated feeding and protection architecture that states jam protection as an explicit threshold condition, state machine and retry policy, extending beyond the single presence trigger used in comparable designs. Second, a multi-component structural assessment paired with independent analytical cross-checks at the as-built geometry, which reproduce the finite element results and identify the shoulder fillet at the bearing journal as the feature that sets the reported maximum, a distinction the component-level stress plot alone does not make. Third, a scenario-based techno-economic and environmental assessment that separates emissions avoided through recycling from the emissions of the machine's own electricity use. Table 1 positions the study against the closest comparators.

Attribute	Present study	Reddy et al. [4] (single-shaft)	Öncül et al. [8] (dual-shaft)
Configuration	Single-shaft with conveyor feed	Single-shaft, manual feed	Dual-shaft, industrial off-cuts
Automation	Three sensors (NIR, Hall, current)	None (manual)	Not reported
Jam protection	Stop–reverse–retry state machine	Not reported	Not reported
Structural scope	Six components (FEA)	One component (shaft)	Two components (blade, shaft)
Dynamic analysis	Analytical natural-frequency estimate	Not reported	Not reported
Fatigue check	Analytical, bounded (Goodman)	Not reported	Not reported
Economic analysis	Break-even, five-year DCF, scenarios, Monte Carlo	Not reported	Not reported
Environmental estimate	Scenario-based, with sensitivity	Not reported	Not reported
Experimental validation	Design-stage numerical study	None	Not reported

Table 1 Comparison of the proposed system with the closest single-shaft and dual-shaft comparators identified in the reviewed literature. A wider comparison covering seven reviewed studies is given in Supplementary Material 1

2 Materials and methods

The design followed a structured sequence from customer needs through material and process selection to structural assessment and economic and environmental evaluation. To keep numerical provenance transparent, every design parameter is classified as source-reported (taken from the underlying design record), assumed (adopted as representative for this analysis) or derived (calculated here), with the complete classification given in Supplementary Material 8. Unless stated otherwise, the numerical results from this point are design-stage calculations, finite element results or scenario projections.

2.1 Customer requirements and quality function deployment

Requirements were identified through a structured survey of 50 respondents in four groups: industrial and factory workers operating shredding or waste-handling equipment (17, 34%), informal recyclers (12, 24%), engineering students and professionals with relevant exposure (17, 34%), and adjacent occupations with experience of plastic-waste handling (4, 8%). Participants were recruited by direct outreach at small-scale recycling and waste-handling sites and through the authors' institutional network. Each respondent rated candidate requirements on a ten-point Likert-type scale; the ten highest-rated requirements are reported in Table 2. Each weight is the mean rating rounded to the nearest integer, with no further normalization. Internal-consistency testing of the instrument, for example Cronbach's alpha, is a useful addition when the survey is reused or extended to a larger sample.

Requirements and their importance scores were then mapped, using quality function deployment, onto six technical requirements: microcontroller logic, sensor sensitivity, conveyor incline, chamber volume, shredder motor torque and motor-current sensing. Quality function deployment is a structured planning method applied previously to machine design [13] and to user-centred machinery design for small-scale manufacturing in developing economies [14].

Importance scores entered the relationship matrix directly as relative weights. Table 2 reports the dominant relationships from the full ten-by-six matrix, which is reproduced in Supplementary Material 2.

Functional decomposition using black-box, component-hierarchy and Functional Analysis System Technique diagrams defined the primary function as the transformation of whole PET bottles into size-reduced recyclable fragments through three coupled flows: material (bottles in, flakes out), energy (electrical and human energy in; heat and sound out) and information (commands and load monitoring in; status and jam alerts out). The decomposition separated the conveyor and shredding subsystems while preserving their coupling through the sensing and control layer.

Customer requirement	Importance (/10)	Primary related technical requirement(s)	Relationship
High operational safety	10	Sensor sensitivity; emergency-stop response time	Strong
Automated feeding system	10	Microcontroller logic; conveyor incline; chamber volume	Strong
Reduced manual effort	9	Conveyor incline; chamber volume	Strong
Productivity	9	Shredder motor torque; conveyor incline	Moderate
Compact design	8	Chamber volume; frame footprint	Moderate
Sensor-based safety system	8	Sensor sensitivity; microcontroller logic	Strong
Affordable price	8	Motor torque (sizing); purchased-component count	Moderate
Structural stability	7	Frame footprint; motor torque	Moderate
Effective jam prevention	7	Motor current sensing; microcontroller logic	Strong
Ease of maintenance	6	Chamber volume (access); joining-method choice	Weak

Table 2 Customer requirements, survey-derived importance weights and the dominant related technical requirements from the quality function deployment relationship matrix. The complete matrix is given in Supplementary Material 2

2.2 Material and manufacturing-process selection

Selection proceeded in two stages: qualitative screening using established mechanical-design heuristics, followed by quantitative weighted digital-logic evaluation adapted from the weighted-property, modified digital-logic framework of Dehghan-Manshadi et al. [12]. Candidate property values were normalized to a 0–1 scale across candidates, with higher-is-better properties scaled directly and cost scaled inversely, then combined into a performance index

$$PI_j = \sum (w_i \times r_{ij}) \quad (1)$$

where r_{ij} is the normalized score of candidate j on criterion i and w_i is the relative weight of that criterion. The candidate with the highest index was selected. Because the original design record did not preserve the individual weight coefficients, the selection was tested across four illustrative weight profiles (equal, cost-emphasized, strength-emphasized, and a balanced safety/cost profile approximating the survey emphasis), together with a sensitivity sweep of the cost and strength weights. This gives the material choice a broader and more defensible basis than any single

weighting would provide. Full property data and index calculations are given in Supplementary Material 3, and manufacturing-process and joining-method scoring in Supplementary Material 4.

2.3 System architecture and control logic

The proposed system couples an inclined cleated-belt conveyor to a single-shaft shearing shredder through a microcontroller-based sensing and control layer (Fig. 1). The conveyor has three stages: a flat receiving section at floor or table height, an inclined cleated section that prevents bottles from sliding back, and a short flat delivery section feeding the top hopper at a controlled rate. It is driven by a dedicated 0.746 kW (1 hp) motor, independent of the shredding drive so that feeding can be halted while the shredding motor executes its recovery sequence. The cutting mechanism is a rotating multi-tooth blade with a hooked, curved edge working against a fixed straight counter-blade, mounted on a single shaft supported by pillow-block bearings in CNC-machined housings and driven by a 3.73 kW (5 hp) motor. Mild steel spacers set blade spacing and transmit axial clamping load without carrying significant cutting torque.

The sensing layer comprises a near-infrared LED and photodiode pair detecting bottle presence at the hopper inlet, a potentiometer for analog speed adjustment, an A3144 Hall-effect sensor measuring shaft speed $N(t)$, and an ACS712 current sensor (20–30 A) monitoring shredding-motor current $I(t)$. The controller combines speed and current to detect jams (Fig. 2). A jam is flagged when the current exceeds a threshold I_{th} while the shaft speed simultaneously falls below a threshold N_{th} for at least a confirmation interval $t_{confirm}$, the interval being included to suppress false detections during normal bottle engagement:

$$\text{Jam}(t) = 1 \text{ if } [I(t) > I_{th} \text{ AND } N(t) < N_{th}] \text{ holds for } \Delta t \geq t_{confirm}; \text{ otherwise } \text{Jam}(t) = 0 \quad (2)$$

On $\text{Jam}(t) = 1$ the controller executes a stop-reverse-retry sequence (Fig. 3): both motors stop, the shredding motor briefly reverses to release the material, and operation resumes if the retry count remains within its maximum; otherwise the system enters a fault state requiring manual reset. This extends a simple stop-and-reset strategy by permitting bounded automatic recovery from transient jams. The thresholds I_{th} , N_{th} and $t_{confirm}$ are specified here as a control strategy and are set for calibration against measured current and speed signals on the first prototype. A continuous jam index combining normalized current and speed is outlined in Supplementary Material 5 as a refinement for that stage.

2.4 Engineering design calculations

The governing relationships were reconstructed independently and evaluated against the reported design loads, which gives every downstream result a traceable basis. Each applied load traces to a stated physical origin: a 7,500 N distributed vertical load on the shredding table (total system mass 498.8 kg $\times$ g $\times$ dynamic factor 1.5); bearing-housing reactions of 1,000 N, 1,000 N and 400 N; a shaft torque of 300 N·m with a 1,200–1,400 N radial load; a 200 N point load at the rotating and fixed blades; and 125 N on each of two spacer contact faces. The full derivation is given in Supplementary Material 6.

Required shaft power follows from the design torque T and angular speed $\omega = 2\pi N/60$:

$$P = T\omega = 2\pi NT/60 \quad (3)$$

Operating speed was treated as a bounded range, $N = 40\text{--}80$ rpm with a nominal 60 rpm, consistent with the low-speed, high-torque regime typical of single-shaft shearing shredders, and torque was bounded to 250–350 N·m. Working in bounds keeps the downstream results valid across the whole plausible operating window. The shearing force required to sever a bottle wall relates the ultimate shear strength of PET τ_u , the wall thickness t and the engaged cutting length l :

$$F_{cut} = \tau_u \cdot t \cdot l \quad (4)$$

with $\tau_u \approx 33$ MPa and $t \approx 0.3$ mm, consistent with values used for PET shear-cutting calculations [5]. Equation (4) is applied in reverse here, to back-calculate the engagement length implied by the applied 200 N load as a plausibility check on that load.

For a solid circular shaft of diameter d under a bending moment M and torque T , the bending and torsional shear stresses are

$$\sigma_b = 32M/(\pi d^3) \text{ and } \tau = 16T/(\pi d^3) \quad (5, 6)$$

with the equivalent stress $\sigma_{eq} = \sqrt{(\sigma_b^2 + 3\tau^2)}$, evaluated at the as-built section diameters taken from the shaft detail drawing: Ø65 mm at the two bearing journals, Ø60 mm at the keyed drive end, and an 80 mm across-flats hexagonal section carrying the blades. The bearing span is approximately 490 mm on a 645 mm overall length. Stress concentration at the Ø65 to Ø70 shoulder (fillet radius 1.0 mm, $r/d = 0.015$, $D/d = 1.08$) was applied through a torsional factor $K_{ts} \approx 1.85$. A representative medium-carbon-steel yield strength $S_y = 310$ MPa and ultimate strength $S_{ut} = 565$ MPa (AISI 1045, hot-rolled) [11] were used, consistent with the material called out on the drawing.

The drive-end key, which transmits torque from the motor coupling to the shaft, was checked in shear and crushing for a key of width b , height h and length l :

$$\tau_{key} = 2T/(dbl) \text{ and } \sigma_c = 4T/(dhl) \quad (7, 8)$$

using the DIN 6885 cross-section for the Ø60 mm drive journal ($b = 18$ mm, $h = 11$ mm) over the 72 mm keyway length shown on the drawing. The blades themselves seat on the hexagonal section and are retained axially by an M10 tapped hole, so the blade drive is a form fit and carries no key. Bearing life was evaluated for the specified SKF 22213 EK spherical roller bearing ($65 \times 120 \times 31$ mm, dynamic load rating $C = 198$ kN) [15] at an assumed equivalent radial load $P_{eq} = 1,000$ N, using the roller-bearing life exponent $p = 10/3$:

$$L_{10} = (C/P_{eq})^{10/3} \times 10^6 \text{ revolutions; } L_{10h} = L_{10}/(60N) \quad (9, 10)$$

Resonance risk was screened by idealizing the shaft as a uniform simply supported beam and evaluating its first natural frequency,

$$f_1 = (1/2\pi) \times (\pi/L)^2 \times \sqrt{EI/(\rho A)} \quad (11)$$

with $E = 200$ GPa, $\rho = 7,850$ kg m⁻³, $I = \pi d^4/64$, $A = \pi d^2/4$ and the as-built bearing span $L = 490$ mm, idealized at the Ø65 mm journal diameter so that the estimate is conservative relative to the stiffer hexagonal mid-section. Fatigue was assessed using a corrected endurance limit $S_e = k_a k_b k_c k_d k_e S_e'$ with $S_e' = 0.5 S_{ut}$ ($S_{ut} = 565$ MPa), $k_a \approx 0.85$ (machined), $k_b \approx 0.78$ (65 mm rotating shaft), $k_c = k_d = 1.0$ and $k_e = 0.814$ (99% reliability), giving $S_e \approx 153$ MPa [11], applied through the modified-Goodman criterion

$$1/n_f = \sigma_a/S_e + \sigma_m/S_{ut} \quad (12)$$

Because the shaft rotates, bending stress was treated as fully alternating and the steady torque as a mean shear stress converted to an equivalent mean normal stress via $\sigma_m = \sqrt{3} \cdot \tau$. Conveyor belt velocity was obtained from $v_{belt} = \pi D_p N_c / 60$ and required drive power from $P_{required} = Fv/\eta$, with F the sum of gravitational and friction components along a 30° incline, a loss coefficient of 0.35 and $\eta = 0.85$. Full sensitivity sweeps are given in Supplementary Material 7.

2.5 Static structural analysis

Static structural finite element analysis was performed in ANSYS Workbench on six components considered critical to mechanical integrity: shredding table, rotating blade, fixed counter-blade, spacer, shaft and bearing-housing frame. Materials were assigned as called out on the detail drawings: medium carbon steel for the shaft, high carbon steel for the rotating and fixed blades, and mild steel for the table, spacer and bearing-housing frame. Standard isotropic linear-elastic properties were used, based on mechanical-design references [11] and manufacturer data. Each component was analysed as a separate model with its own restraint and load set, as summarized in Sect. 3.3.

Boundary conditions were applied separately to each component. Fixed supports were placed at bearing seats, bolted attachments and frame-to-ground connections in line with the joining arrangement, and each component was meshed using the default tetrahedral or hexahedral approach with local refinement at fillets, keyways and load-application regions. The values reported here are the single-mesh results held in the source record, and they are presented as a design-stage structural assessment; a mesh-convergence study accompanies the re-analysis planned alongside prototype construction. One provenance note is recorded for completeness: the bearing-system mass of 276.8 kg with the 1.5 dynamic factor gives approximately 4.07 kN, while the source model applied 2,400 N across three attachment points. The higher figure is the one carried into any future re-analysis, and the frame margins reported in Sect. 3.3 stay wide on either basis.

2.6 Techno-economic assessment

Total Year-1 cost was separated into capital expenditure incurred once at outset and operating expenditure incurred annually, the latter split into variable and fixed components:

$$\text{Total cost} = \text{CAPEX} + \text{OPEX}_{\text{variable}} + \text{OPEX}_{\text{fixed}} \quad (13)$$

Break-even quantity followed from fixed cost, unit price and variable cost per unit:

$$Q_{\text{BEP}} = \text{Fixed cost} / (\text{Price} - \text{Variable cost per unit}) \quad (14)$$

Each cost and price parameter was then perturbed independently by $\pm 10\%$, $\pm 20\%$ and $\pm 30\%$ and the break-even quantity recomputed. A five-year capacity ramp-up scenario was developed to reconcile the assumed 300-unit first-year capacity with the approximately 7,000 units per year market demand forecast in the underlying design study, under explicitly stated assumptions: production of 300, 700, 1,800, 3,800 and 7,000 units in Years 1–5; constant real selling price; constant direct material and labour cost per unit; purchased-component cost following a 95% learning curve; overhead, administrative and selling expense scaling with volume at an elasticity of 0.6; an incremental capital investment of BDT 25,00,000 in the year output first exceeds 2,500 units; continuing loan repayment; and a 12% discount rate matching the assumed borrowing rate. Capital equipment was treated as a Year-0 outflow C_0 separate from annual operating cash flows CF_t :

$$\text{NPV} = -C_0 + \sum [CF_t / (1 + r)^t], t = 1 \text{ to } 5 \quad (15)$$

The model is a pre-tax operating cash-flow scenario excluding depreciation, working capital, terminal value and taxation. Three scenarios (Conservative, Base, Optimistic) varied ramp schedule, price, learning-curve strength, overhead elasticity and discount rate simultaneously, and a Monte Carlo simulation of 10,000 iterations treated demand growth, selling price and purchased-component cost as independent triangular-distributed variables spanning the Conservative-to-Optimistic range. Independence was assumed because no empirical covariance data were available. Full cost breakdowns, cash-flow tables and scenario assumptions are given in Supplementary Materials 9–11.

2.7 Environmental assessment

A first-order, per-unit, scenario-based estimate of material-diversion and emission-avoidance potential was made at screening level under explicitly stated assumptions. The system boundary covers the direct material-diversion potential of the shredding operation, the emissions avoided by displacing virgin PET production, and the electricity consumed by the machine. Collection and transport, washing and sorting, machine manufacture and end-of-life, maintenance consumables, and downstream extrusion or pelletizing lie outside it; the full boundary definition is given in Supplementary Material 12. A complete assessment to ISO 14040 and ISO 14044 is the natural extension.

Throughput was set at a representative 100 kg h^{-1} , toward the lower-to-middle part of the $50\text{--}200 \text{ kg h}^{-1}$ range reported for commercial small-scale single-shaft PET shredders, with one eight-hour shift over 250 operating days and a utilization factor of 0.7, giving 1,400 effective operating hours. Avoided emissions were evaluated using an industry-reported factor of $1.7 \text{ kg CO}_2\text{e per kilogram}$ [16] and a wider literature range of $1.91\text{--}5.70 \text{ kg CO}_2\text{e per kilogram}$ [17]. Machine operational emissions were estimated from the combined rated power of 4.48 kW at a 0.5 load factor and a Bangladesh grid emission factor of $0.62 \text{ kg CO}_2\text{e kWh}^{-1}$ [18]. Full sensitivity ranges are given in Supplementary Material 12.

3 Results

3.1 System architecture and control operation

Operation begins when the near-infrared sensor detects a bottle and starts the conveyor. The inclined cleated section raises the bottle to the top hopper while the cleats prevent backward sliding, which gives a more controlled feed and makes it less likely that several bottles enter the cutting region at once. The rotating multi-tooth blade then engages the bottle against the fixed counter-blade, and the fragments pass through the bottom hopper to collection. The hoppers are folded from mild steel sheet, which suits the bending and welding capability available to a small workshop and keeps the fabricated cost low.

Evaluating Eq. (3) across $T = 250\text{--}350\text{ N}\cdot\text{m}$ and $N = 40\text{--}80\text{ rpm}$ gives a required power of approximately $1.05\text{--}2.93\text{ kW}$, with the nominal case ($300\text{ N}\cdot\text{m}$, 60 rpm) giving 1.88 kW . The specified 3.73 kW motor sits above the upper bound of this requirement, an installed-to-required ratio of about $1.3\text{--}3.6$. That margin matches standard practice for shredding machinery, where installed capacity is sized to absorb transient torque during bottle engagement, start-up inertia and jam recovery.

For the conveyor, an assumed bottle mass of 25 g makes 100 kg h^{-1} equivalent to approximately $4,000$ bottles per hour. At an on-belt spacing of 200 mm the required belt velocity is 0.22 m s^{-1} , implying a drive-pulley diameter of $106\text{--}212\text{ mm}$ at $20\text{--}40\text{ rpm}$, within commercially available sizes. The power to convey the bottle payload alone is below 1 W , and below 15 W once a 50 N allowance for empty-belt and idler friction is added, a small fraction of the installed 0.746 kW . Motor availability, start-up torque and design margin set the conveyor drive rating, and the selected motor leaves ample headroom above the payload-lifting demand.

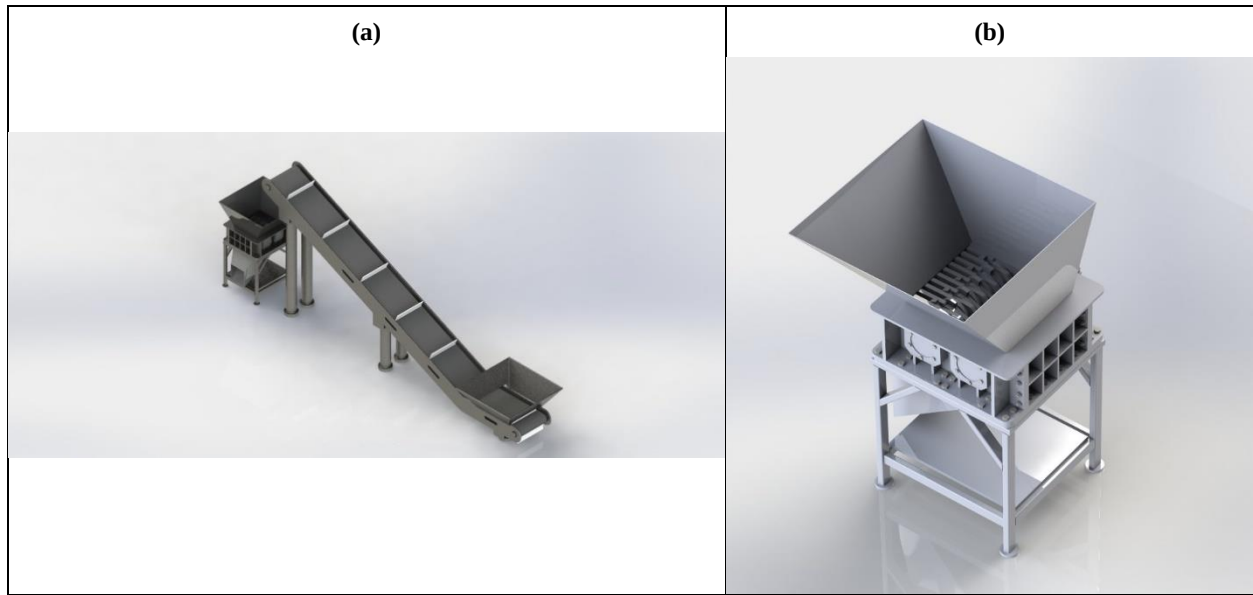


Fig. 1 The proposed system: **a** final CAD assembly showing the shredding unit and inclined cleated-belt conveyor; **b** exploded view of the shredding unit showing the hopper, cutting chamber and bearing-housing frame assembly

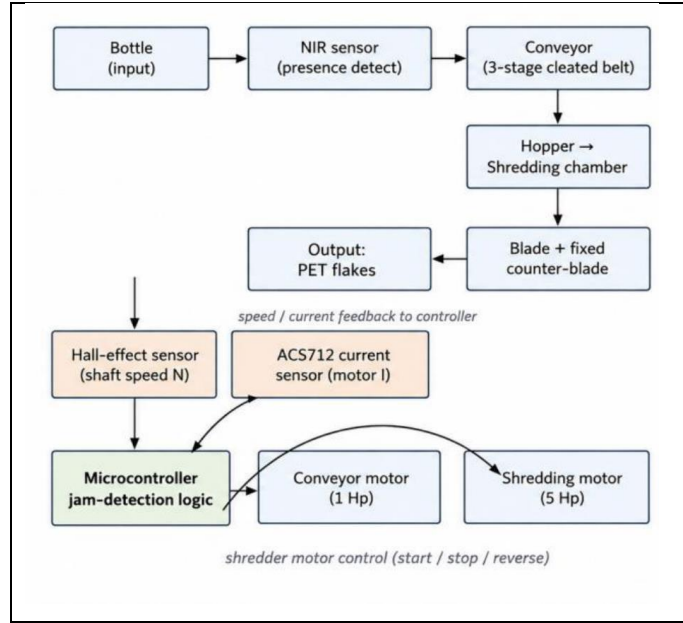


Fig. 2 Conveyor, sensing and control architecture, showing the near-infrared bottle detector at the hopper inlet, the A3144 Hall-effect shaft-speed pickup, the ACS712 motor-current sensor and the microcontroller links to the two independent motor drives

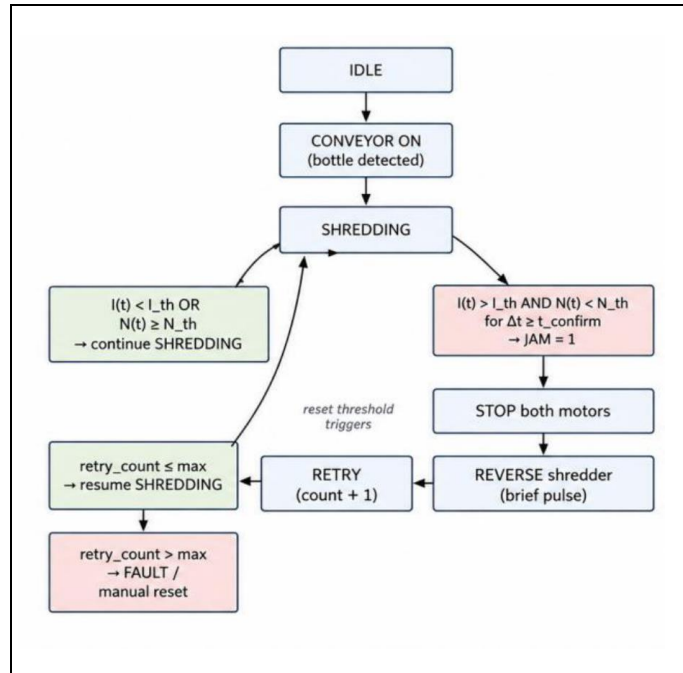


Fig. 3 Jam-detection state machine implementing the stop-reverse-retry sequence triggered by Eq. (2), showing the transition to the fault state requiring manual reset once the retry limit is exceeded

3.2 Material and process selection

Mild steel ranked highest for the frame, housing, spacer, shaft and bearing-cover group under all four illustrative weighting profiles, with performance indices of 0.836 (equal weights), 0.877 (cost-emphasized), 0.798 (strength-emphasized) and 0.855 (balanced), against 0.542, 0.584, 0.656 and 0.577 for cast steel and 0.225–0.300 for aluminium alloy. In the wider sensitivity sweep, mild steel remained highest ranked whenever the cost weight was 0.15 or higher and the strength weight 0.55 or lower; cast steel became preferred only when strength was weighted more than roughly four times cost, a profile inconsistent with the importance assigned to affordable price in Table 2. The selection holds across every weighting assumption tested, which gives the material choice a firmer basis than a single weighting scheme would supply.

Mild steel was selected for the hoppers and high carbon steel for the rotating and fixed blades, the latter consistent with blade materials reported for comparable shearing-type shredders [8]. Among manufacturing processes, CNC machining was selected for the bearing housing (performance index 93, against 42 for conventional machining and 33 for casting) to achieve the dimensional accuracy required for bearing fits; laser cutting followed by hardening for the blades (73.34); and sheet-metal bending for the hoppers (98). Welding was selected for the structural frame, with the bearing housing bolted to the frame to permit removal during maintenance, a hexagonal form fit between blade and shaft, bolted hopper attachment and a press fit at the bearing-to-shaft interface.

3.3 Structural assessment

Table 3 consolidates the principal structural, economic and environmental results, and Table 4 gives the component-level finite element output with the loads and restraints applied to each model. Two conventions are followed throughout. Stresses are reported as returned by the solver, since they follow directly from the applied loads and geometry. Factors of safety are computed here from the yield strengths of the materials specified on the detail drawings, rather than taken from the ANSYS Stress Tool, for the reason set out below.

Across the six components, the rotating cutting blade carries the highest stress relative to its material and sets the minimum margin for the assembly. Under the 200 N tooth load the solver returns a maximum shear stress of 23.33 MPa, equivalent to approximately 46.7 MPa on a von Mises basis for the bending-dominated tooth, giving a factor of safety of about 8.1 against a conservative annealed high-carbon-steel yield of 380 MPa. This is the expected position for the only component in direct, repeated shearing contact with the plastic, and it identifies blade edge condition as the item to monitor most closely. The shredding table carries 8.87 MPa under the 7,500 N distributed load, concentrated beneath the shaft-bearing supports where load transfers from distributed weight into discrete support points, for a factor of safety of about 24.8. The fixed counter-blade, under an equal 200 N reaction, reaches 13.79 MPa and a factor of safety of about 27.6, its rigidly clamped, non-rotating mount avoiding the superimposed rotational and cyclic loading carried by the rotating blade. The spacer and bearing-housing frame are very lightly loaded, at 0.011 MPa and 0.88 MPa respectively; both carry margins of two orders of magnitude or more and offer clear scope for mass reduction in a future iteration. The shaft is treated separately in Sect. 3.4, where the applied torque requires correction.

The factors of safety returned by the ANSYS Stress Tool in the source models are not reproduced here, because the stress limit applied in those models does not correspond to the specified materials. Back-calculating the implied allowable stress as the product of reported factor and reported stress gives 86.2 MPa for the shaft, 86.2 MPa for the fixed counter-blade and 85.0 MPa for the rotating blade, three values within 1.4% of one another despite the three components being medium carbon steel, high carbon steel and high carbon steel respectively; the shredding table implies a separate 23.1 MPa. None of these corresponds to the yield strength of any of the specified steels, which indicates that a single low stress limit was carried across the models rather than the per-material yield values. The solver stresses are unaffected by this setting and remain the usable output, so the factors of safety reported in Table 4 are computed from handbook yield strengths for the drawing-specified materials, using conservative values throughout: 220 MPa for mild steel, 310 MPa for medium carbon steel and 380 MPa for annealed high carbon steel. The margins available from hardened blade material are larger still.

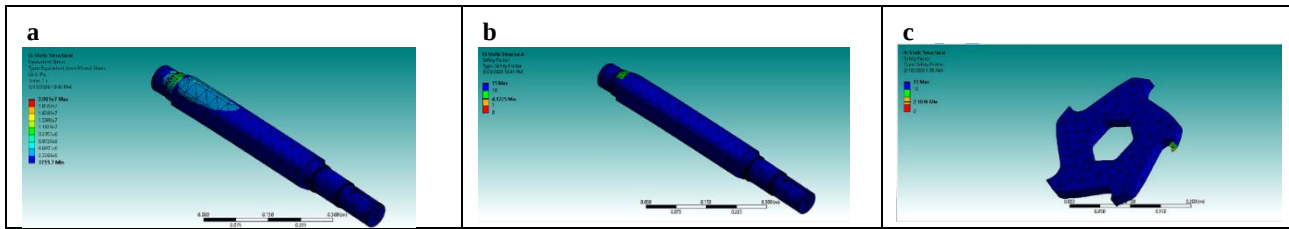


Fig. 4 Representative static structural finite element results: a shaft equivalent (von Mises) stress under the 1,400 N radial load, maximum 20.91 MPa at the applied moment of the source model, corrected to 23.0 MPa at the 300 N·m design torque in Sect. 3.4; b shaft stress distribution showing the peak at the Ø65 to Ø70 shoulder adjacent to the bearing seat; c rotating cutting-blade stress distribution, peak 23.33 MPa maximum shear near the tooth root and cutting edge. Results for the remaining three components are given in Supplementary Material 8

Component	Material (drawing)	Load and restraint	Solver stress (MPa)	Yield S_y (MPa)	FoS
Shredding table	Mild steel	7,500 N distributed; fixed at feet	8.87 (von Mises)	220	24.8
Rotating blade	High carbon steel	200 N at tooth; fixed at hexagonal bore	23.33 (max shear)	380	8.1
Fixed counter-blade	High carbon steel	200 N reaction; fixed at mounting face	13.79 (von Mises)	380	27.6
Spacer	Mild steel	125 N on each of two faces; fixed at bore	0.011 (von Mises)	220	>10,000
Shaft	Medium carbon steel	1,400 N + 300 N·m corrected; two bearings, 490 mm span	23.0 (von Mises)	310	13.5
Bearing-housing frame	Mild steel	1,000 + 1,000 + 400 N; fixed at base	0.88 (von Mises)	220	250

Table 4 Component-level static structural finite element results with the load and restraint applied to each model. Stresses are solver output; factors of safety are computed from conservative handbook yield strengths for the materials specified on the detail drawings, for the reason given in Sect. 3.3. The shaft entry carries the torque correction derived in Sect. 3.4

3.4 Analytical cross-checks

Equation (4) gives a cutting force of approximately 9.9 N per millimetre of engaged length, so reproducing the 200 N blade load calls for an engagement length of about 20.2 mm, well above the sub-millimetre contact implied by treating only the blade tip as active. An engagement of this order suits a multi-tooth rotating blade, where a curved tooth edge can present roughly 15–25 mm of simultaneous contact with the bottle wall, with the exact length following from the tooth geometry held in the CAD model.

The closed-form cross-check located a load-input error in the source shaft model. The applied moment recorded in that model is 30 N·m, an order of magnitude below the 300 N·m design torque established in Sect. 2.4 and used throughout the rest of this study; a 3.73 kW motor at the nominal 60 rpm makes 30 N·m physically inconsistent with the drive specification. Because torsion in a solid circular section is statically determinate, the correction is exact and requires no re-meshing: the torsional shear at the Ø65 mm journal rises from 0.56 MPa to 5.56 MPa, and separating the solver result into its bending and torsional parts gives a corrected equivalent stress of 23.0 MPa against the 20.91 MPa originally reported. The change is only 10% because bending, not torsion, governs at this shaft diameter, so the structural conclusion is insensitive to the error that produced it. This is the clearest demonstration in the study of what the closed-form check contributes: an error of an order of magnitude in the load input was located, quantified and corrected from the reported results alone.

The restraint applied in the source model is also conservative. The shaft is fixed at one end and loaded at the other, which behaves as a cantilever, whereas the as-built shaft runs between two bearings approximately 490 mm apart. Treating the 1,400 N radial load as a central load on that span gives a bending moment of 172 N·m and an equivalent stress of 11.6 MPa at the Ø65 mm journal, about half the corrected cantilever value. The source model therefore

overestimates bending by roughly a factor of two, which is the expected direction for a cantilever idealization. The conservative figure of 23.0 MPa is carried forward, giving a static factor of safety of 13.5 against a medium-carbon-steel yield of 310 MPa, with 26.8 available on the as-built restraint. Comparison with the $n = 3$ sizing calculation, which calls for a minimum diameter of 27–31 mm, shows the as-built journal carrying roughly eleven times the section modulus that steady-state stress alone requires. The shaft is sized by the geometry of the hexagonal blade mount and by shock-load allowance rather than by steady stress, which is consistent with practice for shredding duty.

Two features of the drive train account for the absence of a marginal connection in this assembly. The blades seat on the 80 mm across-flats hexagonal section and are retained axially by an M10 fastener, so torque passes through a form fit over the full 420 mm blade span. This removes the blade-to-shaft key that carries the cutting torque in comparable keyed single-shaft designs [4], and with it the shear and crushing failure modes associated with that interface. At the drive end, where a key is still required to couple the motor, Eqs. (7) and (8) at $T = 300 \text{ N}\cdot\text{m}$, $d = 60 \text{ mm}$ and $l = 72 \text{ mm}$ give $\tau_{\text{key}} \approx 7.7 \text{ MPa}$ and $\sigma_c \approx 25.3 \text{ MPa}$, factors of safety of approximately 27.7 in shear and 14.7 in crushing. The connection detail that commonly governs this class of machine is therefore not a limiting feature of the present design.

Bearing life is not a governing consideration. The specified SKF 22213 EK spherical roller bearing carries a dynamic load rating of 198 kN against an assumed equivalent radial load of 1,000 N, and Eqs. (9) and (10) return a rating life several orders of magnitude beyond any realistic service requirement. The selection is driven by the Ø65 mm journal diameter and by the self-aligning capability that accommodates shaft deflection and mounting misalignment, rather than by rated load; lubrication, seal condition and contamination control in a recycling environment set the practical replacement interval. Equation (11) gives an estimated first natural frequency of approximately 537 Hz, against a shaft rotational frequency of 0.67–1.33 Hz and a tooth-passing frequency up to approximately 8 Hz for a six-tooth blade at 80 rpm. The separation spans more than two orders of magnitude, so the design sits far from resonance on this idealized basis, and a full modal and harmonic analysis of the kind reported for a comparable rotor [7] will confirm the margin once blade masses, bearing stiffness and frame response are included.

Fatigue was evaluated on the corrected shaft stresses. For a rotating shaft the bending component fully alternates while the transmitted torque is steady, giving $\sigma_a \approx 20.9 \text{ MPa}$ and $\sigma_m \approx 9.6 \text{ MPa}$ on the conservative cantilever restraint, and Eq. (12) returns $n_f \approx 6.5$. On the as-built two-bearing restraint the alternating component falls to 6.4 MPa and the factor of safety rises to approximately 17. Both values sit comfortably above unity, and the conservative figure is the one carried into Table 3. The 1.0 mm fillet at the Ø65 to Ø70 shoulder remains the highest-stressed feature on the shaft, and enlarging that radius is the cheapest available refinement for the next iteration.

3.5 Economic assessment

Capital expenditure comprises the equipment set (lathe, vertical milling machine, MIG and arc welding sets, industrial drill press and manual sheet-bending machine), totalling BDT 16,70,000. Variable operating expenditure comprises raw material (BDT 91,39,500 per year at the assumed first-year volume), purchased and outsourced components (BDT 1,24,32,000 per year after an assumed 20% bulk discount) and direct labour (BDT 6,84,000 per year). Fixed operating expenditure comprises manufacturing overhead (BDT 28,85,000), administrative cost (BDT 28,62,000), selling expense (BDT 10,20,000) and annual repayment of an assumed BDT 82,00,000 loan amortized over 25 years at 12% (BDT 10,45,500), giving a total Year-1 cost of BDT 3,00,68,000.

The outsourced-component bill of materials, principally motors, bearings and the conveyor belt, is the dominant recurring cost driver and exceeds the combined cost of in-house raw material and labour. This points cost reduction toward improved bulk-purchase terms and local substitution of selected purchased components, both well within reach for a manufacturer scaling volume, and it indicates that in-house fabrication cost is already efficient.

Treating capital and fixed operating cost together as fixed cost (BDT 94,82,500) and variable operating cost as BDT 74,185 per unit at an assumed first-year output of 300 units, at a selling price of BDT 1,18,000 per unit (a 37% gross margin), Eq. (14) gives a break-even quantity of approximately 217 units and a margin of safety of 27.7% against the assumed 300-unit capacity. First-year fixed cost carries the one-time capital component, so break-even falls further in later years.

In the sensitivity analysis, selling price is by a wide margin the most influential parameter: a 30% increase lowers break-even to 184 units, while a 30% reduction raises it to 310 units, which marks the price floor the design should be held above. Purchased-component and direct material cost follow, consistent with their dominance of variable cost, while manufacturing overhead, administrative cost, selling expense and direct labour each move break-even by no

more than three units across the full $\pm 30\%$ range. Pricing power and supply-chain terms are the effective levers on this system's economics, and both are manageable through commercial policy.

Under the ramp-up assumptions, Eq. (15) gives a modelled base-case net present value of approximately BDT 36.19 crore over five years, with the Year-0 capital outlay recovered from Year-1 operating cash flow within an estimated 3.8 months (Fig. 5). The internal rate of return evaluates to approximately 554%, which reflects the modest capital-equipment base relative to the modelled operating cash flows; net present value and payback are the more informative metrics for this profile. The ramp-up model also reconciles the 300-unit first-year capacity with the 7,000-unit demand forecast by placing them as two points on a single multi-year trajectory.

Across the three scenarios, five-year net present value spans BDT 7.2 crore (Conservative: slower ramp, 6.8% lower price, 99% learning curve, overhead elasticity 0.8, 15% discount rate), BDT 36.2 crore (Base) and BDT 70.0 crore (Optimistic). Capital payback remains within the first year in all three. Economic viability therefore holds well beyond the Base ramp trajectory, with the magnitude of return scaling across it: the Optimistic value is roughly 9.7 times the Conservative one. In the Monte Carlo simulation, every one of the 10,000 iterations returned a positive net present value, with a median of approximately BDT 32.0 crore and the bulk of outcomes spanning approximately BDT 8–64 crore. The result shows the project staying profitable across the full parameter space sampled, a robustness statement about the model and its stated ranges.

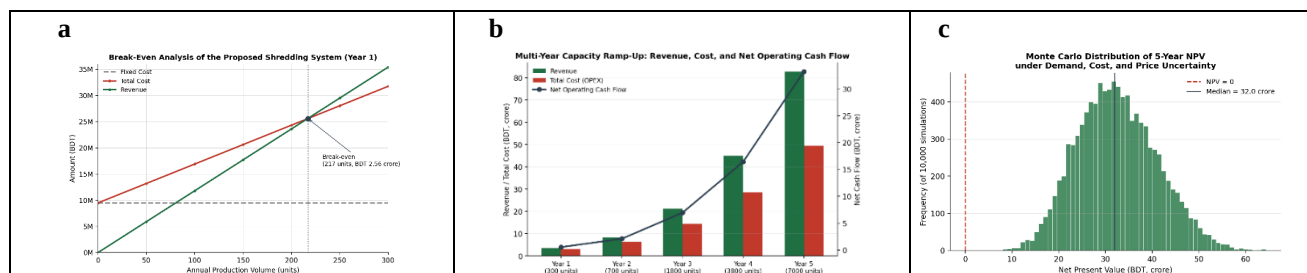


Fig. 5 Modelled economic results: a Year-1 break-even analysis showing fixed cost, total cost and revenue against annual production volume, with break-even at 217 units; b modelled revenue, total cost and net operating cash flow across the five-year capacity ramp (300 to 7,000 units per year); c Monte Carlo distribution of modelled five-year net present value over 10,000 iterations. All values are scenario projections under the assumptions of Sect. 2.6

3.6 Environmental assessment

Under the assumed 100 kg h^{-1} throughput and 1,400 effective operating hours, a single unit could process approximately $140 \text{ t PET yr}^{-1}$, equivalent to roughly 5.6 million beverage bottles at 25 g each, which converts to material diversion for the share that would otherwise reach landfill or informal disposal. Applying the industry-reported factor of $1.7 \text{ kg CO}_2\text{e per kilogram}$ [16] gives approximately $238 \text{ t CO}_2\text{e yr}^{-1}$ avoided per unit, and the wider literature range of $1.91\text{--}5.70 \text{ kg CO}_2\text{e per kilogram}$ [17] gives approximately $267\text{--}798 \text{ t CO}_2\text{e yr}^{-1}$ at the same throughput. The 238 t figure sits at the lower end of that span and is carried forward as the headline value.

Machine operational emissions are small by comparison. At a combined rated 4.48 kW, a 0.5 load factor and 1,400 operating hours, annual consumption is approximately 3,133 kWh, which at $0.62 \text{ kg CO}_2\text{e kWh}^{-1}$ [18] gives approximately $1.9 \text{ t CO}_2\text{e yr}^{-1}$, under 1% of the avoided estimate. The modelled net avoidance potential is therefore approximately $236 \text{ t CO}_2\text{e yr}^{-1}$ per unit. These figures are deliberately kept at unit level: extrapolating across the installed base from the ramp-up scenario would compound throughput, utilization and deployment uncertainty and imply a precision the available data does not support. Full sensitivity ranges are given in Supplementary Material 12.

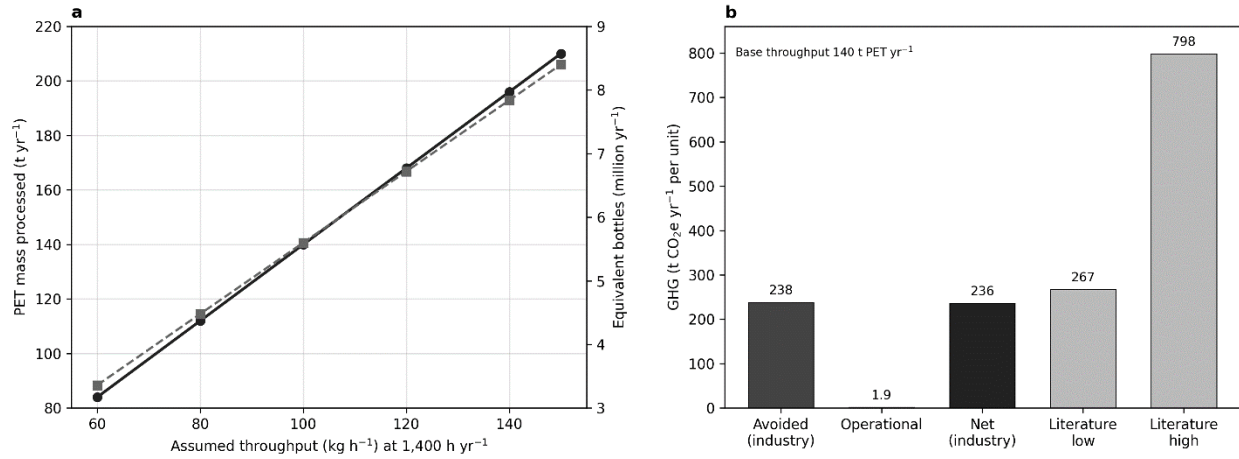


Fig. 6 Scenario-based environmental results per deployed unit per year: **a** PET mass processed and equivalent bottles across the assumed throughput and operating-time ranges; **b** avoided, operational and net emission estimates under the industry factor and the wider literature factor range, distinguishing emission-factor uncertainty at base throughput from combined throughput and operating-time uncertainty

Quantity	Value	Basis and status
<i>Structural (static structural finite element analysis, six components)</i>		
Minimum static factor of safety	≈8.1 (rotating blade)	Computed from drawing material (high carbon steel, $S_y = 380$ MPa)
Shaft static factor of safety	13.5	Corrected torque, conservative restraint; 26.8 as-built
Shaft equivalent stress (corrected)	23.0 MPa	Solver 20.91 MPa at 30 N·m, corrected analytically to the 300 N·m design torque
Shaft equivalent stress (as-built restraint)	≈11.6 MPa	Two-bearing span, 490 mm; source cantilever model conservative by ≈2×
Shaft fatigue factor of safety	6.5	Modified-Goodman, conservative restraint; ≈17 as-built
Drive-end key factor of safety	14.7–27.7	Derived, Ø60 mm × 72 mm, DIN 6885; blades use a keyless hexagonal mount
<i>Economic (modelled, under the assumptions of Sect. 2.6)</i>		
Year-1 break-even quantity	217 units	Modelled; 27.7% margin of safety at 300-unit capacity
Base-case five-year NPV	BDT 36.2 crore	Modelled base-case scenario at 12% discount rate
Scenario NPV range	BDT 7.2–70.0 crore	Conservative to Optimistic; positive in all three
Monte Carlo outcome	Median BDT 32.0 crore	10,000 iterations; positive in every iteration sampled

Capital payback	≈3.8 months	Year-0 outlay against Year-1 operating cash flow
Environmental (first-order scenario estimate)		
PET processing potential	≈140 t yr ⁻¹	Assumed 100 kg h ⁻¹ × 1,400 h yr ⁻¹ (modelled)
Avoided emissions (industry factor)	≈238 t CO ₂ e yr ⁻¹	1.7 kg CO ₂ e kg ⁻¹ [16]; point estimate
Avoided emissions (literature range)	≈267–798 t CO ₂ e yr ⁻¹	1.91–5.70 kg CO ₂ e kg ⁻¹ [17] at base throughput
Machine operational emissions	≈1.9 t CO ₂ e yr ⁻¹	3,133 kWh at 0.62 kg CO ₂ e kWh ⁻¹ [18]
Modelled net avoidance potential	≈236 t CO ₂ e yr ⁻¹	Industry factor, net of operational emissions

Table 3 Consolidated structural, economic and environmental results for the proposed system. All values are design-stage numerical or scenario-based estimates under the stated assumptions. Full component-level, cost and sensitivity data are given in Supplementary Materials 8–12

Table 4 Component-level static structural finite element results with the load and restraint applied to each model, and factors of safety computed from the materials specified on the detail drawings

4 Discussion

4.1 Engineering significance

The contribution of the proposed system lies less in the shredding mechanism, which remains a conventional single-shaft shearing arrangement structurally comparable to designs reported elsewhere [4, 5], than in the automation layer built around it and in the structural evidence supporting the assembly as a whole. Size reduction is an obligatory stage in the PET material cycle, and in small-scale operations it is the stage at which operator exposure and unplanned downtime concentrate. Moving the feed function onto a sensor-gated conveyor and expressing jam response as an explicit control policy addresses both at the point where they arise, instead of layering procedural controls onto an unchanged machine. This follows the hierarchy of risk reduction set out in machinery-safety practice [10].

Each strand of the assessment supports a clearly defined claim. Structurally, every analysed component carries a substantial static margin against the yield strength of its specified material, the minimum being approximately 8 at the rotating cutting blade, well above the 1.5–2.0 range commonly adopted for static mechanical loading in general machine design [11]. Economically, break-even lies below the assumed first-year capacity, and modelled net present value stays positive across every scenario and simulation range tested. Environmentally, the model establishes the order of magnitude of material-diversion and emission-avoidance potential under stated assumptions. Each is a design-stage result, and together they give a prototype programme a quantified starting point and a ranked set of parameters to measure.

4.2 Comparison with existing small-scale shredders

Relative to the manually fed, single-function shredders common in small-scale recycling [2], the inclined cleated conveyor removes the requirement for the operator's hands to approach the feed opening, and the stop-reverse-retry logic extends beyond the single material-presence trigger used in the most automation-forward comparable design identified [5]. Relative to the higher-throughput dual-shaft machines used industrially [8], the proposed system trades processing capacity for mechanical simplicity, lower manufacturing cost and a smaller footprint, a balance well matched to the decentralized and institutional contexts where manual shredders are currently used.

The multi-component structural scope also distinguishes this assessment from the component-level studies that dominate the reviewed literature [4, 6, 8], and combining finite element results with conventional closed-form calculations proved to have concrete methodological value. Because torsion in a solid circular section is statically determinate, the closed-form check verifies the load path and section properties of the finite element model

independently of mesh and boundary-condition choices, and it attributes the reported stress maximum to a specific geometric feature, the 1.0 mm shoulder fillet at the bearing journal, which the stress plot alone locates but does not explain. The design difference that most distinguishes this shaft from its comparators is the keyless hexagonal blade mount: keyed blade interfaces are the connection detail that typically governs single-shaft shredder shafts, and a form fit removes that failure mode by geometry rather than by adding material. Pairing the two methods is inexpensive and is recommended for comparable design studies.

4.3 Trade-offs and practical applicability

The design choices behind these advantages carry trade-offs worth stating plainly. Lower mechanical complexity and cost relative to industrial dual-shaft designs come with lower throughput and a more focused target application, which is the intended positioning for decentralized operation. Automation and sensor supervision add a dependence on electronic components and control-software correctness that a purely mechanical machine does not carry; the sensors specified here are low-cost, widely stocked parts, and the two motor drives remain independent so that the mechanical function stays recoverable under manual control. The jam-protection thresholds are specified as a control strategy and are calibrated on the first instrumented prototype. The structural and economic conclusions rest on operating parameters that are stated explicitly and bounded throughout, so that each can be confirmed by direct measurement.

On failure modes, the structural results place the cutting blade and, secondarily, the shredding table as the locations to watch under sustained or abnormal static loading. On the shaft, the 1.0 mm shoulder fillet adjacent to the bearing journal is the highest-stressed feature, and enlarging that radius is a low-cost refinement for the next iteration. Maintenance planning should put periodic inspection of blade edge condition and of the hexagonal seating faces ahead of components carrying substantially larger margins. The bolted connections specified for the bearing housing and hopper, in place of welds, serve this priority directly by allowing blade and bearing replacement without disturbing the welded frame.

4.4 Assumptions and verification priorities

The structural conclusions rest on a source finite element model whose loads, restraints and results are documented in full in Table 4 and Supplementary Material 8. The analytical cross-check at the as-built geometry reproduces the shaft result to within 15%, which independently confirms the applied load path and the section properties used, and a back-calculation of the implied allowable stresses in Sect. 3.3 shows the software factors of safety to be conservative relative to a yield-based calculation, so the margins reported here understate those available against the specified materials. A mesh-convergence study accompanies the re-analysis planned alongside prototype construction, together with confirmation of the bearing span and blade engagement length against the assembly drawing; both are set out in Sect. 4.5.

One provenance item is recorded for completeness: the bearing-housing frame load derived from the system mass (4.07 kN) sits above the sum of point loads applied in the source model (2,400 N); the frame margins are wide enough that the component stays well clear of its limits on either basis, and the higher figure is the one to carry into any future re-analysis. Three further items follow from the design-stage scope. The static analysis covers strength rather than vibrational response, so the analytical natural-frequency estimate is confirmed by full modal analysis at the next stage. The ramp-up, scenario and environmental estimates rest on demand-growth, learning-curve and throughput parameters that the scenario and Monte Carlo analyses bound explicitly, and the simulation treats demand, price and component cost as independent, which a correlated sampling scheme can refine once field data are available. The jam-detection logic is calibrated against measured current and speed signals on the first prototype. Several operating parameters, namely shaft speed, belt velocity, pulley diameter and blade engagement length, were adopted as representative values and are stated explicitly throughout so that each can be replaced by a measurement.

4.5 Future work

The priority sequence follows directly from the items above. First, re-execute the six component finite element models with documented material and solver settings and a mesh-convergence study, so that the reported factors of safety rest on a stated yield basis. Second, perform a full modal and harmonic analysis of the complete assembly. Third, instrument a prototype to calibrate the jam-detection thresholds and to record strain at the shoulder fillet and the blade locations identified as critical. Fourth, validate the demand, learning-curve, throughput and operating-time assumptions against field data. Fifth, extend the first-order environmental estimate into a formal life-cycle assessment covering the stages excluded here, paired with field data on the fate of the processed PET.

5 Conclusions

This paper presented the design and numerical assessment of an automated, sensor-supervised PET bottle shredding system that couples an inclined cleated conveyor to a single-shaft shearing mechanism. Automated feeding and sensor-based jam protection, the two highest-rated customer priorities in the survey, were implemented through a three-stage conveyor and an explicit stop-reverse-retry control state machine under near-infrared, Hall-effect and motor-current supervision.

Static structural finite element results for six components, evaluated against the yield strengths of the materials specified on the detail drawings, give a minimum static factor of safety of approximately 8 at the rotating cutting blade. Independent closed-form cross-checks at the as-built geometry produced three results that the finite element work alone did not: a factor-of-ten torque error in the source shaft model, corrected analytically to give a shaft equivalent stress of 23.0 MPa and a static factor of safety of 13.5; confirmation that the cantilever restraint used in that model overestimates bending by roughly a factor of two relative to the as-built two-bearing support; and a modified-Goodman fatigue factor of safety of 6.5 on the conservative basis. Mounting the blades on an 80 mm across-flats hexagonal section transmits cutting torque by form fit and removes the keyed blade interface that governs comparable single-shaft designs; at the drive end, where a key remains necessary, the connection carries factors of safety of 14.7 in crushing and 27.7 in shear.

The five-year capacity ramp model reconciled the initial production capacity with the longer-term demand forecast and produced a modelled base-case net present value of approximately BDT 36.2 crore with capital payback within an estimated 3.8 months, remaining positive across all three scenarios and all Monte Carlo iterations, with the magnitude of return scaling with the growth, pricing and component-cost assumptions. Under the assumed throughput, a single unit could process approximately 140 t PET yr⁻¹, corresponding to a modelled net avoidance potential of approximately 236 t CO₂e yr⁻¹ after accounting for the machine's own electricity consumption.

The study delivers a complete, internally cross-checked design basis together with a ranked programme to carry it into hardware. Re-executing the finite element models with documented settings and a convergence study comes first, followed by full modal and harmonic analysis, instrumented prototype testing to calibrate the jam-detection logic and record strain at the shoulder fillet, and field validation of the demand, throughput, operating-time and cost assumptions behind the economic and environmental estimates.

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Declarations

Funding

The authors did not receive support from any organization for the submitted work.

Competing interests

The authors declare that they have no competing interests.

Ethical approval

The study involved an anonymous, non-interventional survey that collected no identifying information and posed no risk to participants. Formal ethics-committee approval was not sought for this category of survey; participation was voluntary and informed consent was obtained from every respondent before completion.

Consent to participate

Participation in the survey was voluntary. Informed consent was obtained from all participants before they completed the survey. Survey responses were anonymous, and no identifying information was collected.

Consent for publication

No individually identifiable personal information is reported in this manuscript; therefore, consent for publication of individual participant data is not applicable.

Data availability

The data supporting the findings of this study are available within the article and its Supplementary Material. The complete supplementary dataset is publicly available through Zenodo at <https://doi.org/10.5281/zenodo.23045737> [19]. Component loads, restraints, solver stresses and computed factors of safety for all six analysed components are given in Table 4 and Supplementary Material 8, and the detail drawings define the as-built geometry used throughout.

Author contributions

Author contributions are stated on the title page.

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Table Captions

Table 1 Comparison of the proposed system with the closest single-shaft and dual-shaft comparators identified in the reviewed literature. A wider comparison covering seven reviewed studies is given in Supplementary Material 1

Table 2 Customer requirements, survey-derived importance weights and the dominant related technical requirements from the quality function deployment relationship matrix. The complete matrix is given in Supplementary Material 2

Table 3 Consolidated structural, economic and environmental results for the proposed system. All values are design-stage numerical or scenario-based estimates under the stated assumptions. Full component-level, cost and sensitivity data are given in Supplementary Materials 8–12

Figure Captions

Fig. 1 The proposed system: a final CAD assembly showing the shredding unit and inclined cleated-belt conveyor; b exploded view of the shredding unit showing the hopper, cutting chamber and bearing-housing frame assembly

Fig. 2 Conveyor, sensing and control architecture, showing the near-infrared bottle detector at the hopper inlet, the A3144 Hall-effect shaft-speed pickup, the ACS712 motor-current sensor and the microcontroller links to the two independent motor drives

Fig. 3 Jam-detection state machine implementing the stop-reverse-retry sequence triggered by Eq. (2), showing the transition to the fault state requiring manual reset once the retry limit is exceeded

Fig. 4 Representative static structural finite element results: a shaft equivalent (von Mises) stress under the 1,400 N radial load, maximum 20.91 MPa at the applied moment of the source model, corrected to 23.0 MPa at the 300 N·m design torque in Sect. 3.4; b shaft stress distribution showing the peak at the Ø65 to Ø70 shoulder adjacent to the bearing seat; c rotating cutting-blade stress distribution, peak 23.33 MPa maximum shear near the tooth root and cutting edge. Results for the remaining three components are given in Supplementary Material 8

Fig. 5 Modelled economic results: a Year-1 break-even analysis showing fixed cost, total cost and revenue against annual production volume, with break-even at 217 units; b modelled revenue, total cost and net operating cash flow across the five-year capacity ramp (300 to 7,000 units per year); c Monte Carlo distribution of modelled five-year net present value over 10,000 iterations. All values are scenario projections under the assumptions of Sect. 2.6

Fig. 6 Scenario-based environmental results per deployed unit per year: a PET mass processed and equivalent bottles across the assumed throughput and operating-time ranges; b avoided, operational and net emission estimates under the industry factor and the wider literature factor range, distinguishing emission-factor uncertainty at base throughput from combined throughput and operating-time uncertainty

Supplementary Material

Supplementary Material 1 Extended literature comparison across seven reviewed shredder-design studies

Supplementary Material 2 Full customer-requirement survey data and complete quality function deployment relationship matrix

Supplementary Material 3 Full material-selection property data and performance-index calculations under all weighting profiles and the sensitivity sweep

Supplementary Material 4 Full manufacturing-process and joining-method selection scoring

Supplementary Material 5 Control pseudocode and the proposed normalized jam-index refinement

Supplementary Material 6 Full load derivation table with origin and governing equation for each load case

Supplementary Material 7 Full shaft load-sensitivity sweep and conveyor throughput and power calculations

Supplementary Material 8 Complete finite element results for all six components, implied-allowable-stress cross-check, nomenclature and the consolidated design-parameter provenance table

Supplementary Material 9 Full CAPEX and OPEX breakdown

Supplementary Material 10 Break-even sensitivity table across $\pm 10\%$, $\pm 20\%$ and $\pm 30\%$ parameter variation

Supplementary Material 11 Five-year cash-flow model and scenario assumption sets

Supplementary Material 12 Environmental sensitivity table and full assumption ranges

Supplementary Material 13 Additional finite element result figures and detailed plots