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# Theoretical Analysis of a Cotton-Stack Breaker with Curved Pegs: Particle Motion, Peg Deflection, and Performance Estimation

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**Abstract.** Background: Cotton-stack breakers loosen stored seed cotton and transfer it to subsequent processing operations. Their performance depends on the interaction between the rotating drum, its pegs, and the fibrous cotton mass.
Objective: To develop a theoretical framework for comparing a conventional straight-peg drum with a proposed drum equipped with curved pegs, and to identify the data needed for experimental verification.
Methods: The analysis combines drum kinematics, a linear particle-motion model of a cotton fragment under driving, frictional, and resistance forces, a rotor torque balance, Euler–Bernoulli beam and beam-column models of peg deflection, and a throughput relationship.
Results: A corrected closed-form solution of the linear particle-motion model is obtained, together with a steady-state torque condition, a bending-stress design criterion, and a dimensionally consistent throughput expression. Preliminary design estimates indicate higher throughput (35–50 vs 25–35 t h−1), lower specific energy consumption (1.2–1.8 vs 1.8–2.5 kWh t−1), and lower cotton damage (1.5–3 vs 4–7 %) for the curved-peg design. These figures are not supported by a documented test protocol or raw measurements and are indicative only.
Conclusions: The framework supports further design calculations and controlled testing of cotton-stack breaker working bodies; the reported advantages of curved pegs remain hypotheses until validated experimentally.

**Keywords:** cotton processing; cotton-stack breaker; curved pegs; rotating drum; particle dynamics; peg deflection; throughput; mechanical design

## Introduction

Cotton-processing plants require reliable equipment for removing seed cotton from stored stacks and delivering it to downstream cleaning and processing units. A stack breaker performs this task by engaging the cotton mass with a rotating working body and transferring the loosened material to a conveying system [1]. The design of the working body affects material flow, energy demand, mechanical loading, and the likelihood of damaging the cotton [2].

A conventional working body consists of a cylindrical drum with straight pegs fixed to its surface. As the drum rotates, the pegs penetrate the cotton layer and detach portions of the material [3]. The proposed design uses curved pegs, intended to alter the engagement path and encourage additional loosening and separation. The potential advantage of a curved geometry must be weighed against the risks of increased bending stress, excessive contact forces, and unstable material transport [4].

The geometry alone does not establish improved performance: the effect depends on drum speed, peg angle and curvature, penetration depth, cotton moisture and density, and the arrangement of the receiving conveyor [5]. A consistent analytical description that links these parameters is therefore needed before experimental work is designed.

The aim of this paper is to formulate such a theoretical framework. The specific objectives are to: (i) formulate a simplified model for the motion of a cotton fragment; (ii) describe the rotor torque balance; (iii) establish a beam-based model for peg deflection; (iv) present a basic throughput estimate; and (v) identify the evidence required to compare conventional and curved-peg designs [6].

## Materials and Methods

### Equipment and working-body concept

The RBA-type stack breaker considered here comprises a traction unit, a milling cutter or rotary working body, a boom, a working-body drive, a conveyor drive, a boom-lifting mechanism, a receiving chute, a slewing platform and drive, a loading conveyor, a frame, running gear, a take-away conveyor, a steering system, and a boom support. These assemblies cooperate to position the working body, loosen the stack, and transfer the detached cotton.

The conventional working body is a rotating cylindrical drum carrying straight pegs. In the proposed concept the pegs have a curved, arc-like profile. The design hypothesis is that this profile changes the direction and duration of contact with the cotton fragment, potentially improving loosening and transfer.

[Figure 1 – insert the general view of the RBA-type stack breaker. Figure 2 – insert the drawings of the conventional and curved-peg working bodies. Use high-resolution, numbered figures with dimensions, and confirm permission status before submission.]

### Drum kinematics

For a drum rotating with angular velocity ω and effective outer radius $R$, the tangential velocity of the peg tip is

$$v = \omega R \tag{1}$$

where $v$ is the linear velocity of the peg tip (m $s^{-1}$), ω is the angular velocity (rad $s^{-1}$), and $R$ is the effective drum radius (m). If the rotational speed is expressed as $N$ in revolutions per minute, $\omega = 2\pi N/60$. The tangential velocity is a key input to the contact-force and material-transfer analysis.

### Motion of a cotton fragment

A detached cotton fragment is treated as a body of mass $m$ moving along the transfer axis $x$. The peg applies a driving force $F_q$, while friction $F_r$ and material resistance $F_i$ oppose the motion. Newton's second law gives

$$m\ddot{x} = F_q - F_r - F_i \tag{2}$$

For a linear resistance model, $F_i = c\dot{x}$, where $c$ is an effective damping/resistance coefficient. The friction force is approximated as $F_r = \mu N$, and under the simplifying assumption $N = mg\cos\theta$,

$$m\ddot{x} + c\dot{x} = F_q - \mu mg\cos\theta \tag{3}$$

Here μ is the effective friction coefficient, $g$ is gravitational acceleration, and θ is the peg inclination angle defined for the chosen contact geometry. A working range of μ = 0.35–0.55 was adopted from the preliminary design documentation. For constant $F_q$ and θ, define $k = c/m$ and $A = (F_q - \mu mg\cos\theta)/m$, so that

$$\ddot{x} + k\dot{x} = A \tag{4}$$

with initial conditions $x(0) = 0$ and $\dot{x}(0) = 0$. In real cotton handling the driving force, contact normal force, and resistance vary with time, so this model is a first-order approximation rather than a complete description of fibre dynamics.

### Rotor torque balance

The angular acceleration of the drum is determined by the difference between the driving and resisting torques:

$$I\,(d\omega/dt) = M_d - M_r \tag{5}$$

where $I$ is the rotor's mass moment of inertia (kg m$^2$), $M_d$ is the applied drive torque (N m), and $M_r$ is the resisting torque (N m). For a uniform solid cylindrical drum of mass $m_b$ and radius $R$,

$$I = \tfrac{1}{2} m_b R^2 \tag{6}$$

The resisting torque is approximated as

$$M_r = nF_qR \tag{7}$$

where $n$ is the number of pegs simultaneously engaged with the cotton. An additional velocity-dependent expression for the force, involving cotton density, a coefficient, and the peg frontal area, was considered in the design documentation; it requires calibration and dimensional checking for the actual contact mechanism and is not used quantitatively here.

### Elastic deflection of a peg

A peg subjected to a distributed transverse load is approximated using Euler–Bernoulli beam theory. For a straight slender peg under small deflection,

$$EI\, d^4w/ds^4 = q(s) \tag{8}$$

where $E$ is Young's modulus (Pa), $I$ is the second moment of area of the peg cross-section (m$^4$), $w(s)$ is the transverse deflection (m), $s$ is the coordinate along the peg (m), and $q(s)$ is the distributed load (N m$^{-1}$). For a peg carrying an axial compressive force $P$, a simplified beam-column model is

$$EI\, d^4w/ds^4 + P\, d^2w/ds^2 = q(s) \tag{9}$$

A constant distributed load $q(s) = q_0$ is considered. The boundary conditions must represent the actual mounting of the peg, for example a fixed base and a free tip rather than simply supported ends, and the sign convention for $P$ must be stated. The maximum bending stress is estimated from the bending moment:

$$\sigma_{max} = M_{max}/W_z \le [\sigma] \tag{10}$$

where $M_{max}$ is the maximum bending moment, $W_z$ is the section modulus (m$^3$), and $[\sigma]$ is the allowable stress for the selected material and design conditions.

### Theoretical throughput estimate

The mass throughput is estimated as

$$Q_s = 3600\, n_z V_z \rho_p \eta \tag{11}$$

where $Q_s$ is the mass throughput (kg h$^{-1}$), $n_z$ is the number of effective peg engagements per second (s$^{-1}$), $V_z$ is the volume detached per engagement (m$^3$), $\rho_p$ is the bulk density of the cotton (kg m$^{-3}$), and $\eta$ is a dimensionless process coefficient. If $n_z$ is defined per minute, the conversion factor must be 60 instead of 3600. The volume detached per engagement is approximated by a cylinder:

$$V_z = \pi d_z^2 h_z/4 \tag{12}$$

where $d_z$ is the characteristic diameter of the effective engagement area (m) and $h_z$ is the penetration depth (m). A process coefficient of $\eta = 0.70$–$0.85$ was adopted as an assumed range.

### Data basis and computational tools

This is an analytical study; no new controlled experiment is reported. The parameter ranges ($\mu$, $\eta$) and the comparative performance indicators for the two designs were taken from the preliminary design documentation available to the authors. Fragment displacement and peg deflection for different peg angles and cutter speeds were examined in Maple.

## Result

### Analytical solution of the particle-motion model

Integrating Eq. (4) with $x(0) = 0$ and $\dot{x}(0) = 0$ gives, for $k \neq 0$,

$$\dot{x}(t) = (A/k)[1 - \exp(-kt)] \tag{13}$$

$$x(t) = (A/k)t - (A/k^2)[1 - \exp(-kt)] \quad (14)$$

Equations (13)–(14) are the mathematically consistent solution of the stated linear model; the displacement expression in the original draft contained a sign inconsistency, and the form above satisfies both initial conditions [7]. The velocity approaches the limiting value $A/k$ with a time constant $1/k$, and a forward-moving fragment requires $A > 0$, i.e. $F_q > \mu mg \cos\theta$. For $k \to 0$ the solution reduces to $x(t) = At^2/2$.

[Figure 3 – insert the original Maple plots of fragment displacement and peg deflection for different peg angles and cutter speeds, together with the parameter values, initial and boundary conditions used [8].

## Rotor torque condition

At steady rotational speed $d\omega/dt = 0$, and Eq. (5) requires $M_d = M_r$. With Eq. (7), the drive torque needed to sustain rotation scales with the number of simultaneously engaged pegs, the peg force, and the drum radius [9].

## Peg deflection and stress criterion

Equations (8)–(10) provide the governing relations for peg deflection and the strength criterion $\sigma_{max} \leq [\sigma]$. For the beam-column model, the integration constants follow from the support and tip conditions of the actual attachment. Because peg dimensions, material properties, load distribution, and allowable stress are not specified in the available material, Eq. (10) is a design criterion and not a numerical verification [10].

## Throughput relationship

With $n_z$ expressed per second, Eq. (11) is dimensionally consistent and yields kg $h^{-1}$: $s^{-1} \cdot m^3 \cdot kg\ m^{-3} \cdot 3600\ s\ h^{-1}$. The per-minute definition used in the original draft, combined with the factor 3600, was inconsistent and has been corrected. Substituting Eq. (12) shows that throughput increases linearly with the engagement frequency, the penetration depth, the bulk density, and $\eta$, and quadratically with the engagement diameter $d_z$ [11].

## Preliminary comparative performance estimates

Table 1 lists the comparative ranges for the conventional straight-peg design and the proposed curved-peg design. No test method, sample size, operating conditions, or raw data accompany these values; they are preliminary estimates and are not independently verified experimental findings.

**Table 1.** Preliminary comparative performance estimates (no documented test protocol).

| Indicator | Conventional straight-peg design | Proposed curved-peg design |
|---|---|---|
| Throughput (t $h^{-1}$) | 25–35 | 35–50 |
| Specific energy consumption (kWh $t^{-1}$) | 1.8–2.5 | 1.2–1.8 |
| Cotton damage (%) | 4–7 | 1.5–3 |
| Reliability (%) | 85–90 | 92–97 |
| Service life (years) | 5–8 | 8–12 |

If confirmed, the ranges would indicate higher throughput and reduced specific energy consumption and cotton damage for the curved-peg design. The ranges for throughput, specific energy consumption, and service life meet at their boundary values, and no uncertainty estimates are provided [12].

## Discussion

### Interpretation of the framework

The framework identifies three coupled design questions. First, drum speed and peg geometry determine the velocity and contact path imposed on the cotton. Second, the resulting contact forces govern both material transfer and the mechanical load transmitted to the peg and rotor. Third, the elastic response of the peg affects durability and may alter the contact trajectory during engagement. Curved pegs could change the engagement path and encourage loosening, but any advantage depends on their curvature, inclination, length, cross-section, and operating speed [13].

### Model limitations

The linear particle model is useful for examining trends but simplifies a heterogeneous, deformable fibre mass into a single lumped body. Cotton is compressible and entangled; friction and resistance are likely nonlinear and dependent on moisture, contamination, bulk density, and loading history [14]. A more complete model could use experimentally calibrated force–displacement relations, a multibody or discrete-element representation, or a coupled experimental–numerical approach.

The beam model also needs a realistic support condition. A peg welded or bolted to a rotating drum is usually closer to a cantilever than to a simply supported beam, so the final design calculation should use a cantilever or other boundary model consistent with the real attachment. The axial-force term in Eq. (9) should use a clearly stated sign convention, because compression can reduce stability and increase deflectionx [15]. Fatigue, impact loading, stress concentration at the peg root, corrosion or wear, and possible buckling should also be considered.

This paper is a theoretical synthesis rather than a report of a completed controlled experiment [16]. Several numerical inputs and the original graph values are not included, and the comparative performance ranges have not been independently verified. The results therefore establish a modelling framework and design hypotheses, not proof of a quantified performance improvement.

### Recommended experimental validation

A validation programme should compare the conventional and curved-peg working bodies under controlled conditions. At minimum, the following should be documented:

1. **Working-body geometry:** drum diameter, peg count and spacing, peg length, curvature radius, inclination angle, cross-section, and attachment method.
2. **Operating conditions:** rotational speed, feed rate, working width, stack height, cotton moisture content, bulk density, and ambient conditions.
3. **Performance measurements:** mass throughput, electrical power consumption, specific energy per tonne, percentage of visibly damaged cotton, and transfer losses.
4. **Mechanical measurements:** peg-root strain or deflection, vibration, drive torque, bearing temperature, and wear after a defined operating period.
5. **Experimental design:** repeated trials, instrument calibration, uncertainty estimates, a clearly defined damage-assessment method, and statistical comparison between designs.

Results should be reported with sample sizes, mean values, standard deviations or confidence intervals, and a transparent description of the test protocol. Numerical modelling should be validated against at least one measured response, such as fragment displacement, rotor torque, or peg deflection [17].

## Conclusion

A theoretical framework was assembled for a cotton-stack breaker with a rotating drum and curved pegs, covering drum kinematics, cotton-fragment motion, rotor torque balance, peg deflection, and throughput estimation.

For the simplified linear motion model, the displacement satisfying zero initial displacement and velocity is $x(t) = (A/k)t - (A/k^2)[1 - \exp(-kt)]$, where $k = c/m$ and $A$ is the net driving force per unit mass.

The curved-peg concept may modify the cotton-engagement path and promote loosening, but its performance depends on geometry, operating speed, cotton properties, and the structural response of the pegs.

The reported ranges for throughput, energy consumption, damage, reliability, and service life are potentially favourable but preliminary, and should be validated through controlled, repeatable testing.

Before design approval or journal submission, the model requires a dimensionally consistent throughput definition, realistic peg boundary conditions, complete input parameters, reproducible plots, and experimental validation.

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**Conflict of interest**
[The authors should declare any competing interests, or state that there are none, if accurate.]

**Data availability**
The data used in this theoretical study are limited to the preliminary design information available to the authors. Any additional calculation files, experimental data, and original figures should be deposited or made available according to the target journal's policy.

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