

# KINEMATIC CORRECTION OF THE PRESSURE PLATE MOTION IN A DIE-CUTTING PRESS USING RACK-DRIVEN ECCENTRIC MECHANISMS

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## ABSTRACT

This study develops a kinematic correction for the pressure plate of a die-cutting press driven by a main wedging mechanism and auxiliary rack-driven eccentrics. The input stroke is adjusted so that every corrected curve reaches the same normalized maximum,  $S_i = 1$ , while retaining the 80 mm working stroke. Calculations cover three relative eccentricities ( $\lambda_e = 0.05, 0.075$  and  $0.10$ ) and four eccentric start positions (upper, lower, left and right). At  $\lambda_e = 0.10$ , the left position increases the calculated dwell angle within  $S_i \geq 0.99$  from  $14.4^\circ$  to  $17.4^\circ$  and reduces the magnitude of the dominant negative acceleration invariant from 1.5299 to 1.0564. Tests on a related sectional-plate eccentric stand showed peak-torque reductions of 18–31% along the fibers and 23–42% across them. These measurements show temporal load redistribution during sequential die engagement on the sectional-plate stand.

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## 1. INTRODUCTION

Flat-bed die-cutting converts sheets of paperboard into carton blanks by cutting, creasing, and, where required, embossing them against a stationary die (Rehei, 2011). These operations establish the blank geometry and the fold or relief features before the carton is erected. The final approach of the moving pressure plate is therefore critical: small changes in its displacement and velocity alter how the tools enter and load the board.

The die-cutting section is a key unit of the equipment used to manufacture carton blanks. Cutting, creasing, and embossing occur during a brief interval of contact between the moving pressure plate and the die. During

this interval, the whole tool engages the blank simultaneously over its entire area, so the load on the drive rises steeply, and the parallelism and motion law of the pressure plate govern the uniformity of the pressure distribution. Experimental pressure measurement using microcapsules and image sensing showed that die-cutting machines require reliable contact-force diagnostics and that the pressure distribution directly affects the quality of the technological operation (Jiao et al., 2018). Consequently, the way the pressure plate approaches the die during the finishing part of the stroke is at least as important as the peak force it develops.

A large body of work is therefore devoted to shaping and controlling the pressure-plate motion. Modern die-

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cutting equipment is increasingly supported by servo control, synchronized axes, and dedicated motion-control methods that improve trajectory accuracy; a passive velocity-field control method for die-cutting machines demonstrated that the control quality depends on the coordination between the cutting contour and the drive response (Zhang et al., 2018). Mechanical optimization of the drive remains equally active: a lightweight drive-rod study for precision die-cutting machines reduced the moving mass while preserving stiffness, thereby lowering deformation and improving the dynamic stability of the press (Wang et al., 2023). Cam-based drives offer another route to a prescribed motion law, and optimizing a cam drive system using a B-spline method reduced vibration and improved accuracy, although it introduces its own manufacturing problem that a wedging mechanism avoids (Sun & Fang, 2024). The optimization of a dual-elbow-bar mechanism further showed that die-cutting kinematics can be improved by treating the press drive as a coupled kinematic and structural problem rather than a purely geometric chain (Lin et al., 2015). General spline-based motion synthesis with non-uniform rational B-splines (Nguyen et al., 2019) and smooth trajectory generation with cubic B-spline curves subject to velocity and acceleration limits (Li et al., 2022) provide the methodological background for shaping a continuous, low-jerk technological motion law for cyclic equipment.

Creasing and relief embossing are especially sensitive to the finishing phase of the stroke because their outcomes depend on tool displacement and the board's short-term response. Ultrasound-assisted embossing studies have linked the degree of forming, detail reproduction, and shape accuracy to the interaction among the tool, pressure, and board structure (Hunniger et al., 2022), including the material's thermal and mechanical responses under brief compression (Kappeler et al., 2020). Compression tests also show a non-linear rise in force with compression distance (Kappeler et al., 2022). Studies of creasing and folding identify crease depth, delamination, and material response as additional controlling factors (Nagasawa et al., 2003; Beex & Peerlings, 2009; Nygård et al., 2009; Huang & Nygård, 2010). Together, these findings justify examining the displacement history near the top of the pressure-plate stroke.

Among the mechanical drives of the pressure plate, wedging (self-locking lever) mechanisms are widely used in printing and packaging machinery (Poliudov, 2005) because they generate a large technological force from a comparatively small load on the driving links. Their synthesis, kinematics and force behaviour have been studied for single and combined wedging contours (Vlakh & Mykhailiv, 2023), for a combined double-crank wedging drive that increases the contact duration between the tools and the blank (Rehei et al., 2023), for a combined double-wedging mechanism validated against SolidWorks simulation (Vlakh et al., 2025b) and,

more recently, through a CAD/CAE-based kinematic and force analysis of a double-wedging drive (Rehei et al., 2026). Alternative drives based on a screw–nut transmission have also been investigated experimentally for paperboard cutting and provide a stable motion, but they differ from wedging systems in force transmission, dimensions, and integration into existing presses (Ternytskyi et al., 2021). A promising way to retain the advantages of the wedging drive while improving the finishing phase is to superimpose a small auxiliary correction motion onto the main wedging displacement, for example, via an additional driven crank-and-gear transmission (Vlakh et al., 2025a).

The three technological operations carried out in the die-cutting section differ in how the tool interacts with the cardboard, and each imposes its own requirement on the finishing phase of the pressure-plate stroke (Figure 1). Cutting (Figure 1a) is performed by a sharp steel rule that is pressed through the full thickness of the blank against a hardened counter-surface, while the rubber ejectors beside the rule strip the blank after the cut. Creasing (Figure 1b) uses a blunt, rounded rule that does not sever the board but forces it into a matching channel of the creasing counter-die, forming a residual bead along which the package is later folded. Relief embossing (Figure 1c) is produced by a male–female die set that locally compresses the board between the two forming surfaces, so that a three-dimensional relief is retained in the material after the load is removed.

Cutting requires the board to fracture, whereas creasing and embossing rely mainly on controlled deformation. More time near the top of the stroke may allow the compressed fibers to relax, but a kinematic dwell interval alone does not establish the actual contact time or the resulting product quality. Earlier studies by the authors examined drive arrangements intended to prolong tool–blank interaction: a combined double-crank wedging drive (Rehei et al., 2023), a combined double-wedging mechanism (Vlakh et al., 2025b), and a wedging drive with an additional crank-and-gear transmission (Vlakh et al., 2025a). The present work builds on that design objective while treating any benefit to cutting, creasing, or embossing as a hypothesis that still requires direct testing.

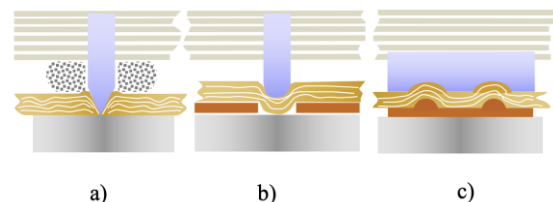


**Figure 1.** Schematic of processing a cardboard blank in a press using a cutting rule (a), a creasing rule (b), and an embossing tool set

This study develops and evaluates a kinematic correction for the pressure plate of a flat-bed die-cutting press. Its main contribution is a procedure for selecting the input

stroke  $S\lambda$  before the displacement curve is calculated, so that four rack-driven eccentric start positions can be compared at the same normalized maximum displacement. The analysis covers relative and absolute displacement, velocity, and acceleration for  $\lambda_e = 0.05$ , 0.075, and 0.10. A separate laboratory experiment is used only to examine the broader principle of temporal load redistribution. The stand has a sectional plate with individual eccentric drives, not the proposed rack-driven single plate, and its torque measurements are therefore

indirect supporting evidence rather than validation of the new mechanism.

## 2. METHODOLOGY

The proposed press combines the main wedging drive of the pressure plate with eccentric correction mechanisms driven by fixed toothed racks. Figure 2 shows the layout used for the kinematic analysis.

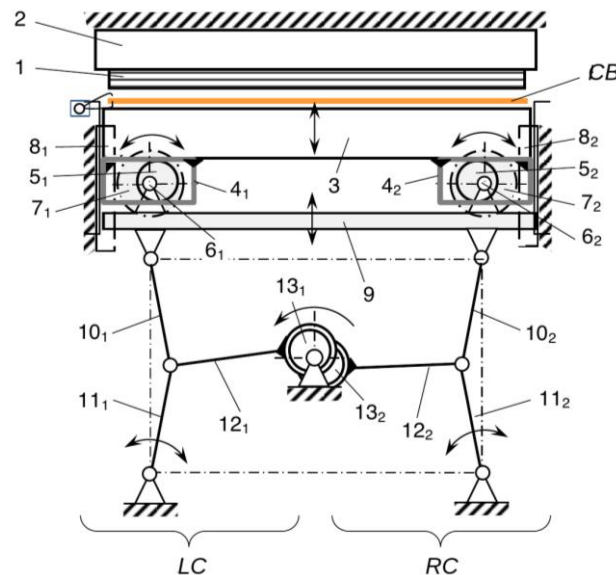


**Figure 2.** Scheme of the proposed die-cutting press with rack-driven eccentric correction mechanisms

The press consists of a flat die-cutting form 1 fixed to a stationary horizontal plate 2 and a movable pressure plate 3 that is guided in vertical guideways and carries a left slot 4<sub>1</sub> and a right slot 4<sub>2</sub>. A left eccentric 5<sub>1</sub> and a right eccentric 5<sub>2</sub> are mounted in these slots and fixed on a left shaft 6<sub>1</sub> and a right shaft 6<sub>2</sub>, together with a left gear wheel 7<sub>1</sub> and a right gear wheel 7<sub>2</sub>. The gear wheels mesh with fixed vertical left and right toothed racks 8<sub>1</sub> and 8<sub>2</sub>. The shafts 6<sub>1</sub> and 6<sub>2</sub> are hinged to the upper side of a horizontal frame 9, whose lower side carries the upper rockers 10<sub>1</sub> and 10<sub>2</sub> of the left and right wedging contours. The upper rockers are connected through the lower rockers 11<sub>1</sub> and 11<sub>2</sub> and the connecting rods 12<sub>1</sub> and 12<sub>2</sub> to the left and right eccentric mechanisms 13<sub>1</sub> and 13<sub>2</sub>, which are mounted on the main drive shaft. In this way, the left contour (LC) and the right contour (RC) form the main wedging drive, while the rack-driven eccentrics 5<sub>1</sub> and 5<sub>2</sub> form the additional correction drive that acts directly on the pressure plate.

The press operates as follows. After a cardboard blank (CB) is fed into the die-cutting zone, the pressure plate 3 is in its lowest position, the eccentric mechanisms 13<sub>1</sub> and 13<sub>2</sub> are displaced to their extreme right and left positions, and the additional eccentrics 5<sub>1</sub> and 5<sub>2</sub> are shifted downward in the slots 4<sub>1</sub> and 4<sub>2</sub>. Rotation of the main drive shaft counter-clockwise moves the connecting rod 12<sub>1</sub> to the left and the connecting rod 12<sub>2</sub> to the right; this symmetric motion wedges the rockers 10<sub>1</sub>, 11<sub>1</sub> of the left

contour and 10<sub>2</sub>, 11<sub>2</sub> of the right contour and lifts the horizontal frame 9. As the frame rises, the gear wheel 7<sub>1</sub> rolls along the fixed rack 8<sub>1</sub> and rotates counter-clockwise, turning the left eccentric 5<sub>1</sub> through the shaft 6<sub>1</sub>, while the gear wheel 7<sub>2</sub> rotates the right eccentric 5<sub>2</sub> clockwise. Because the eccentrics sit in the horizontal slots 4<sub>1</sub> and 4<sub>2</sub>, their rotation produces an additional upward displacement of the pressure plate precisely during the finishing operation, when the die rules penetrate the blank. The cycle is completed when the eccentric mechanisms 13<sub>1</sub> and 13<sub>2</sub> reach their extreme positions and the additional eccentrics 5<sub>1</sub> and 5<sub>2</sub> reach their upper positions; further rotation of the drive shaft returns the pressure plate downward during an idle stroke.

The base motion is taken from the single-wedging contour analyzed by Vlach and Mykhailiv (2023). Related double-crank and double-wedging drives (Rehei et al., 2023; Vlach et al., 2025b), together with the modernized pressure-plate drive of Kuznetsov et al. (2017), provide the structural context. The auxiliary-motion concept follows the driven crank-and-gear arrangement reported by Vlach et al. (2025a), while the subsequent CAD/CAE study by Rehei et al. (2026) supports using a wedging contour as the base system. In the present design, the racks rotate the eccentrics as the plate rises. The correction is governed by the relative

eccentricity  $\lambda_e$  and the eccentric start position, while the prescribed maximum plate stroke is retained.

The kinematic correction is built on the displacement law of the base single-wedging contour. For a crank angle  $\varphi$ , the relative (dimensionless) displacement of the pressure plate produced by one wedging contour is obtained from the geometric synthesis of the four-bar wedging loop as

$$S_{base,i} = H - 2 l_3 \cos \varphi_{1,i} \quad (1)$$

where  $H$  is the relative height of the wedging contour (the distance between the extreme positions of the guide point expressed in units of the driving-crank radius  $R_1$ ),  $l_3$  is the relative length of the wedging link and  $\varphi_{1,i}$  is the current inclination of the wedging link, which is related to the transmission angle  $\gamma_i$  of the four-bar loop by

$$\varphi_{1,i} = \beta_0 - (\pi - \gamma_i) \quad (2)$$

Here  $\beta_0 = \arcsin(W / (2 \cdot OO_1))$  is the initial angle of the loop,  $W$  is the relative width of the contour,  $OO_1$  is the relative distance between the fixed axes, and  $\gamma_i$  is the transmission angle computed from the four-bar closure conditions. The complete geometric synthesis of the base wedging mechanism, together with the auxiliary angles that define  $\gamma_i$ , was derived and verified earlier (Kuznetsov et al., 2012) and is not repeated here; only the quantities that enter the corrected calculation are reproduced. The index  $i$  denotes the discrete crank position  $\varphi_i = i \cdot \Delta\varphi$ .

The base contour has relative width  $W = 5$  and relative height  $H = 3.4$ . With the initial wedge angle  $\gamma_B = 5^\circ$ , the relative link length is  $l_3 = H \cdot \sin \gamma_B / \sin(\pi - 2\gamma_B)$ ; the remaining geometric parameters are  $\lambda_1 = OO_1/R_1$ ,  $\lambda_2 = l_2/R_1$  and  $\lambda_3 = l_3/R_1$ . The correction uses  $\lambda_{rD} = 0.318$  and three relative eccentricities,  $\lambda_e = 0.05, 0.075$  and  $0.10$ . The absolute working stroke is  $S_{max} = 80$  mm, and the crank speed is  $n = 60$  rpm ( $\omega_r = \pi \cdot n/30$ ). Curves are evaluated at an angular step  $\Delta\varphi = 1^\circ$ ; the scaling relations below convert the dimensionless displacement, velocity, and acceleration to absolute values.

For each eccentricity  $\lambda_e$  and start position  $P$  (upper, lower, left or right), the input stroke  $S_\lambda$  is found before the kinematic curve is evaluated. Its value is chosen so that the corrected displacement spans the common normalized range  $0 \leq S_i \leq 1$ :

$$\max_i S_i(S_\lambda, \lambda_e, P) = 1. \quad (3)$$

The base relative displacement curve is first scaled to the calculated input stroke:

$$S_{in,i} = S_\lambda \frac{S_{base,i} - S_{base,min}}{S_{base,max} - S_{base,min}} \quad (4)$$

The current angular coordinate includes the eccentric start phase  $\alpha_{D,0}$ . It is zero for the upper and lower eccentric positions and  $\pi/2$  for the left and right eccentric positions:

$$\alpha_{D,i} = \alpha_{D,0}(P) + \frac{S_{in,i}}{\lambda_{rD}}. \quad (5)$$

The eccentric contribution is then added to the corrected input curve according to the selected start position; the four explicit laws are given below Figure 3:

$$S_i(\lambda_e, P) = S_{in,i} + \Delta S_e(P, \alpha_{D,i}). \quad (6)$$

For the absolute displacement scale, the corrected relative value is multiplied by the required press stroke. In the calculations used for the graphs,  $S_{max} = 80$  mm and  $n = 60$  rpm:

$$s_i(\lambda_e, P) = S_{max} S_i(\lambda_e, P). \quad (7)$$

The sign of the eccentric term, its constant offset and the initial phase define the four start positions relative to guide point  $D$ : upper, lower, left and right. Figure 3 shows these orientations and their normalised displacement laws.

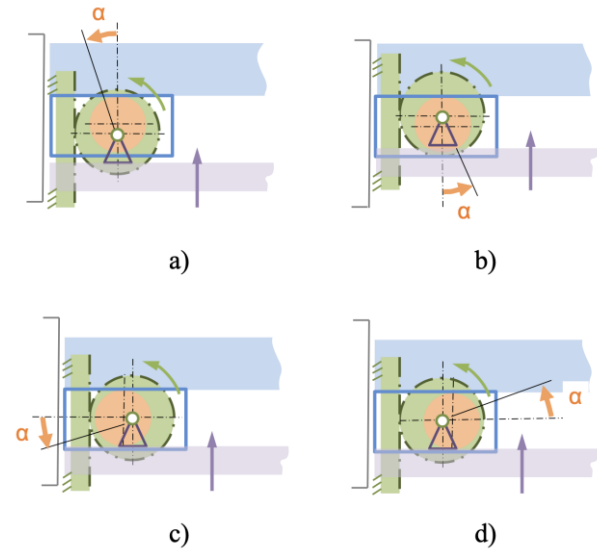


**Figure 3.** Start positions of the rack-driven eccentric correction mechanism: (a) upper, (b) lower, (c) left, and (d) right eccentric positions

Upper eccentric position (Figure 3a):

$$S_i = S_{in,i} + \lambda_e \cos(\alpha_D) - \lambda_e. \quad (8)$$

Lower eccentric position (Figure 3b):

$$S_i = S_{in,i} - \lambda_e \cos(\alpha_D) + \lambda_e. \quad (9)$$

Left eccentric position (Figure 3c):

$$S_i = S_{in,i} - \lambda_e \cos(\alpha_D). \quad (10)$$

Right eccentric position (Figure 3d):

$$S_i = S_{in,i} + \lambda_e \cos(\alpha_D). \quad (11)$$

**Table 1.** Corrected input stroke  $S_i$  for the upper, lower, left, and right eccentric positions ( $\lambda_{rD} = 0.318$ )

| $\lambda_e$ | Position | $S_i$    |
|-------------|----------|----------|
| 0.05        | Upper    | 1.097616 |
|             | Lower    | 0.902295 |
|             | Left     | 1.000182 |
|             | Right    | 0.999868 |
| 0.075       | Upper    | 1.142496 |
|             | Lower    | 0.857324 |
|             | Left     | 1.000300 |
|             | Right    | 0.999814 |
| 0.10        | Upper    | 1.183620 |
|             | Lower    | 0.816095 |
|             | Left     | 1.000447 |
|             | Right    | 0.999767 |

The calculations were performed in Python at 361 crank positions,  $\varphi_i = i \cdot \Delta\varphi$  ( $i = 0 \dots 360$ ,  $\Delta\varphi = 1^\circ$ ). At each position, the program computes the geometric closure of the wedging loop and evaluates  $S_{base,i}$  from Eqs. (1)–(2), and scales it according to Eq. (4). For every  $\lambda_e$  and P, a bisection search finds  $S_i$  from Eq. (3); Eqs. (4)–(6) and (8)–(11) then give the corrected displacement. Central finite differences with respect to crank angle give the velocity and acceleration invariants. Absolute values follow from  $S_i = S_{max} \cdot S_i$  (Eq. 7),  $v = S_{max} \cdot \omega_r \cdot V_i$ , and  $a = S_{max} \cdot \omega_r^2 \cdot a_i$ . Threshold-crossing angles reported in Table 2 were obtained by linear interpolation between adjacent  $1^\circ$  samples. No angular-grid convergence study was performed, so the acceleration extrema should be read at the stated resolution. Figures 5–8 and Tables 1–4 were generated from the same dataset for the base curve and all twelve corrected variants.

**Table 2.** Dwell angle  $\Delta\varphi$  ( $^\circ$ ) within the cardboard-thickness bands  $\delta = 0.005$ , 0.010 and 0.015 for the base curve and the four eccentric positions.

| Variant                  | $\lambda_e$ | $\delta = 0.005$ ( $S_i = 0.995$ ) | $\delta = 0.010$ ( $S_i = 0.990$ ) | $\delta = 0.015$ ( $S_i = 0.985$ ) |
|--------------------------|-------------|------------------------------------|------------------------------------|------------------------------------|
| Base                     | -           | 10.13                              | 14.37                              | 17.66                              |
| Upper eccentric position | 0.05        | 9.42                               | 13.40                              | 16.48                              |
|                          | 0.075       | 8.99                               | 12.79                              | 15.73                              |
|                          | 0.10        | 8.57                               | 12.20                              | 14.98                              |
| Lower eccentric position | 0.05        | 10.41                              | 14.77                              | 18.14                              |
|                          | 0.075       | 10.41                              | 14.77                              | 18.14                              |
|                          | 0.10        | 10.35                              | 14.68                              | 18.03                              |
| Left eccentric position  | 0.05        | 11.02                              | 15.67                              | 19.28                              |
|                          | 0.075       | 11.59                              | 16.47                              | 20.26                              |
|                          | 0.10        | 12.25                              | 17.41                              | 21.42                              |
| Right eccentric position | 0.05        | 9.39                               | 13.34                              | 16.40                              |
|                          | 0.075       | 9.07                               | 12.90                              | 15.85                              |
|                          | 0.10        | 8.79                               | 12.51                              | 15.37                              |

### 3. EXPERIMENTAL SETUP

The experiment focused on a specific question: whether eccentric pressure-plate movement can shift die engagement over time and lower the maximum drive torque. A full press for the proposed rack-driven correction was not available, so a related laboratory stand was used instead. Its monoblock pressure plate can be replaced with a sectional plate, with two sections, each with an individual eccentric drive. This arrangement reproduces temporal load redistribution, but not the motion law nor the force path of the proposed rack-driven single-plate mechanism.

The experimental stand (Figure 4) is driven by a stepper motor 1 through a worm gearbox 2, which rotates the main shaft 3, supported in bearings 4 on the frame 12. Eccentrics 5 and 5' fixed on the shaft drive the pressure-plate sections 6 and 6', which move vertically on cylindrical guides 7 and linear bearings 8; the die-cutting form 11 is fixed to a stationary support plate 9 that is set on the guides by nuts 10. The angular offset between the two eccentrics can be adjusted so that the sections engage the blank sequentially while remaining parallel

throughout the working cycle. The shaft speed is set by the regulator 14 and the periodic motion law by the stepper-motor control system 15, thereby allowing the shaft speeds of 30, 60, and 90 rpm used below to be compared.

Drive-shaft torque was measured using BF350-3AA strain gauges set at 45 degrees in a bridge circuit. The signal was digitized with a 24-bit HX711 converter, processed by an ATmega328P microcontroller, and sent via an HC-05 Bluetooth module. Calibration involved applying known forces at a set lever arm, which allowed us to convert ADC codes to torque M. The specimens were commercial folding box board (FBB) with thicknesses of 0.3, 0.4, 0.5, 0.6, and 0.7 mm, cut in both the machine direction (MD) and cross direction (CD). The dataset includes peak-torque estimates for each condition, but does not have the individual replicate readings, calibration residuals, or environmental-drift records needed for a full repeatability or measurement-uncertainty analysis. Because of this, standard deviations, confidence intervals, and significance tests are not reported, and the following comparisons are descriptive only.

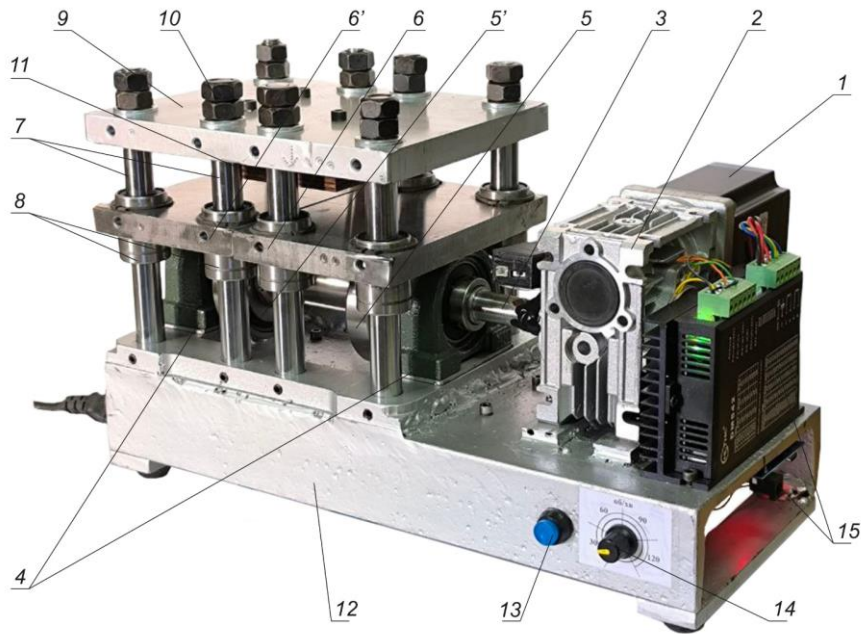


**Figure 4.** Experimental stand with a sectional pressure plate and individual eccentric drives (1 – stepper motor, 2 – worm gearbox, 3 – main shaft, 4 – bearings, 5, 5' – eccentrics, 6, 6' – pressure-plate sections, 7 – cylindrical guides, 8 – linear bearings, 9 – support plate, 10 – nuts, 11 – die-cutting form, 12 – frame, 13 – switch, 14 – speed regulator, 15 – stepper-motor control system)

#### 4. RESULTS

To compare motion near the top of the stroke, the relative cardboard thickness  $\delta$  is defined as

$$\delta = \frac{\Delta}{S_{max}} \quad (12)$$

For a nominal blank thickness  $\Delta \approx 0.8$  mm and  $S_{max} = 80$  mm,  $\delta = 0.010$ . The kinematic dwell angle  $\Delta\varphi$  is the crank-angle interval for which  $S_i \geq 1 - \delta$ . A larger  $\Delta\varphi$  therefore means a wider geometric interval near the die in the rigid-body model. It does not account for contact compliance or force and must not be read as a measured contact time. The additional values  $\delta = 0.005$  and  $0.015$  correspond to nominal thickness bands of approximately 0.4 mm and 1.2 mm, respectively. The 0.8 and 1.2 mm bands are parametric cases, not specimens from the 0.3–0.7 mm experimental set.

Figure 5 shows the reference displacement curve for the base single-wedging mechanism. The displacement rises smoothly from  $S_i = 0$ , reaches  $S_i = 1$  near  $\varphi = 159^\circ$ , and returns during the idle stroke. The base dwell angle within  $S_i \geq 0.99$  is  $14.37^\circ$ . This curve is the reference for all corrected variants.

Figure 6 compares all four start positions at  $\lambda_e = 0.05$ ,  $0.075$  and  $0.10$  near the top of the stroke. Every corrected curve reaches  $S_i = 1$ , but its width changes. For  $\delta = 0.010$ , the base dwell angle is  $14.37^\circ$ . The left position increases it to  $15.67^\circ$ ,  $16.47^\circ$ , and  $17.41^\circ$  as  $\lambda_e$  rises from  $0.05$  to  $0.10$ ; the largest increase is  $21.1\%$ . At  $\lambda_e = 0.10$ , the upper

and right positions shorten the interval to  $12.20^\circ$  and  $12.51^\circ$ , while the lower position remains close to the base at  $14.68^\circ$ . The same ordering holds for  $\delta = 0.005$  and  $0.015$ . These values describe the geometry of the motion and do not, by themselves, demonstrate an improvement in cutting, creasing, or embossing.

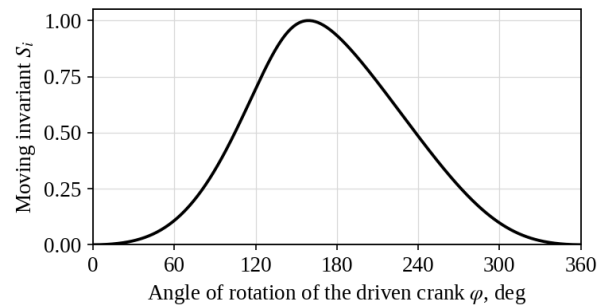


**Figure 5.** Relative displacement (moving invariant  $S_i$ ) of the base wedging mechanism over the cycle

Figure 7 and Table 3 compare the velocity invariant  $V_i$ . The base range is  $-0.4622$  to  $+0.7430$ . At  $\lambda_e = 0.10$ , the left position lowers the positive peak to  $+0.7072$  (4.8% below the base), whereas the right position raises it to  $+0.8787$  (18.3% above the base) and deepens the negative peak to  $-0.5404$ . Among all twelve corrected variants, the smallest positive peak is  $+0.7028$  for the left position at  $\lambda_e = 0.075$ . The left position therefore combines a wider kinematic dwell with a lower approach-velocity peak, although the technological effect still requires direct testing.

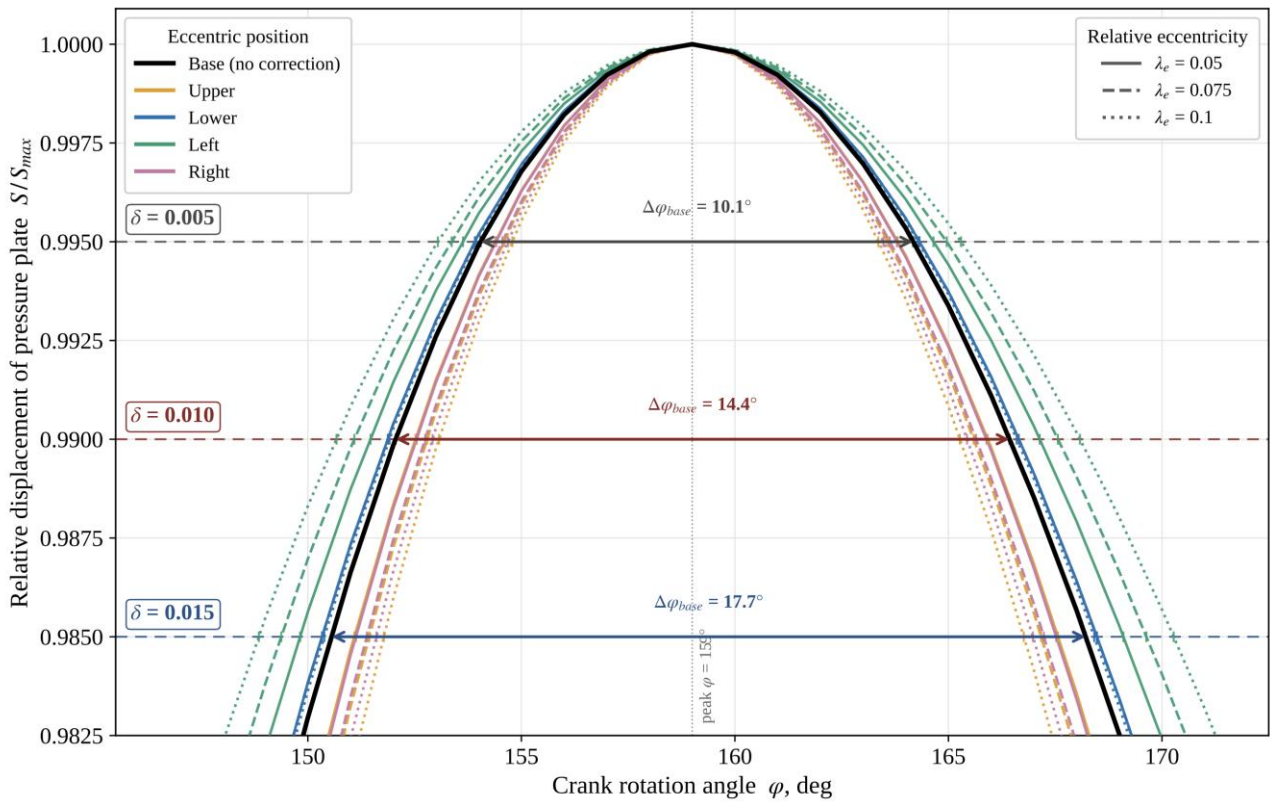


**Figure 6.** Corrected relative displacement  $S_i$  of the pressure plate near the top of the stroke for the four eccentric start positions (1 – upper, 2 – lower, 3 – left, 4 – right) and  $\lambda_e = 0.05, 0.075, 0.10$ , with the base curve; the dwell angle  $\Delta\varphi$  is read at three relative cardboard thicknesses  $\delta = 0.005, 0.010$  and  $0.015$  (the base-curve value is marked separately)

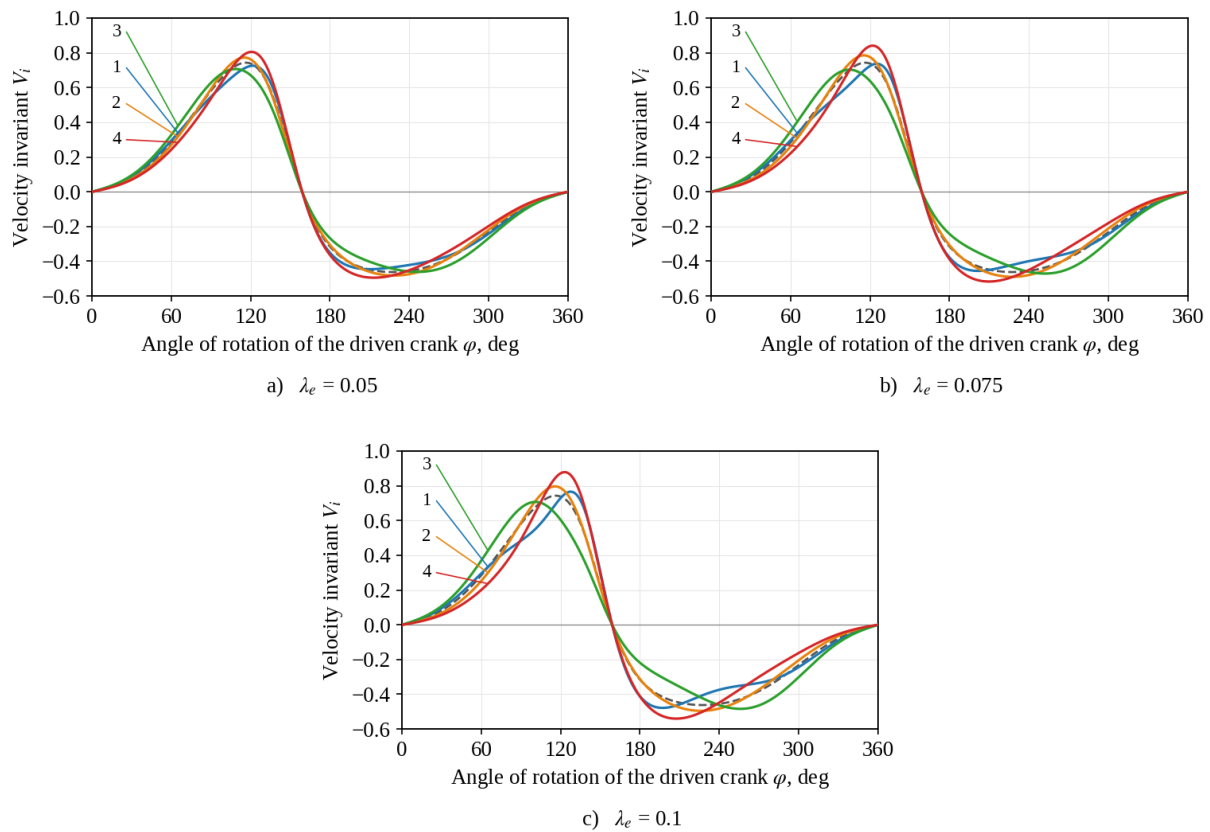


**Figure 7.** Relative velocity  $V_i$  for (a)  $\lambda_e = 0.05$ , (b)  $\lambda_e = 0.075$  and (c)  $\lambda_e = 0.10$  (1 – upper, 2 – lower, 3 – left, 4 – right; dashed – base)

**Table 3.** Relative velocity extrema  $V_i$  for the eccentric positions.

| Position  | $\lambda_e$ | $V_i$ min | $V_i$ max |
|-----------|-------------|-----------|-----------|
| Base      | —           | -0.4622   | 0.7430    |
| Upper (1) | 0.05        | -0.4458   | 0.7255    |
|           | 0.075       | -0.4557   | 0.7370    |
|           | 0.10        | -0.4787   | 0.7660    |
| Lower (2) | 0.05        | -0.4824   | 0.7727    |
|           | 0.075       | -0.4897   | 0.7853    |
|           | 0.10        | -0.4956   | 0.7964    |
| Left (3)  | 0.05        | -0.4614   | 0.7066    |
|           | 0.075       | -0.4710   | 0.7028    |
|           | 0.10        | -0.4841   | 0.7072    |
| Right (4) | 0.05        | -0.4949   | 0.8052    |
|           | 0.075       | -0.5168   | 0.8411    |
|           | 0.10        | -0.5404   | 0.8787    |

Figure 8 and Table 4 show that acceleration is more sensitive to the start position. The base range is  $-1.5299$  to  $+0.5869$ . At  $\lambda_e = 0.10$ , the left position reduces the magnitude of the dominant negative peak to  $1.0564$ , a 31.0% reduction. The upper and right positions increase that magnitude to  $2.0302$  and  $2.0053$ , while the lower position remains essentially unchanged at  $1.5299$ . For a

fixed equivalent moving mass, these values scale only the inertial component; the total drive load also depends on contact forces, compliance, and friction. Across the four  $\lambda_e = 0.10$  variants at  $S_{max} = 80$  mm and  $n = 60$  rpm, velocity spans  $-0.2717$  to  $+0.4417$  m/s and acceleration spans  $-6.4119$  to  $+2.7067$  m/s<sup>2</sup>.

**Table 4.** Relative acceleration extrema  $a_i$  for the eccentric positions.

| Position  | $\lambda_e$ | $a_i$ min | $a_i$ max |
|-----------|-------------|-----------|-----------|
| Base      | —           | -1.5299   | 0.5869    |
| Upper (1) | 0.05        | -1.7097   | 0.4991    |
|           | 0.075       | -1.8548   | 0.4621    |
|           | 0.10        | -2.0302   | 0.5888    |
| Lower (2) | 0.05        | -1.4912   | 0.6594    |
|           | 0.075       | -1.5047   | 0.6870    |
|           | 0.10        | -1.5299   | 0.7100    |
| Left (3)  | 0.05        | -1.2929   | 0.6103    |
|           | 0.075       | -1.1746   | 0.6325    |
|           | 0.10        | -1.0564   | 0.6574    |
| Right (4) | 0.05        | -1.7676   | 0.6640    |
|           | 0.075       | -1.8865   | 0.7528    |
|           | 0.10        | -2.0053   | 0.8570    |

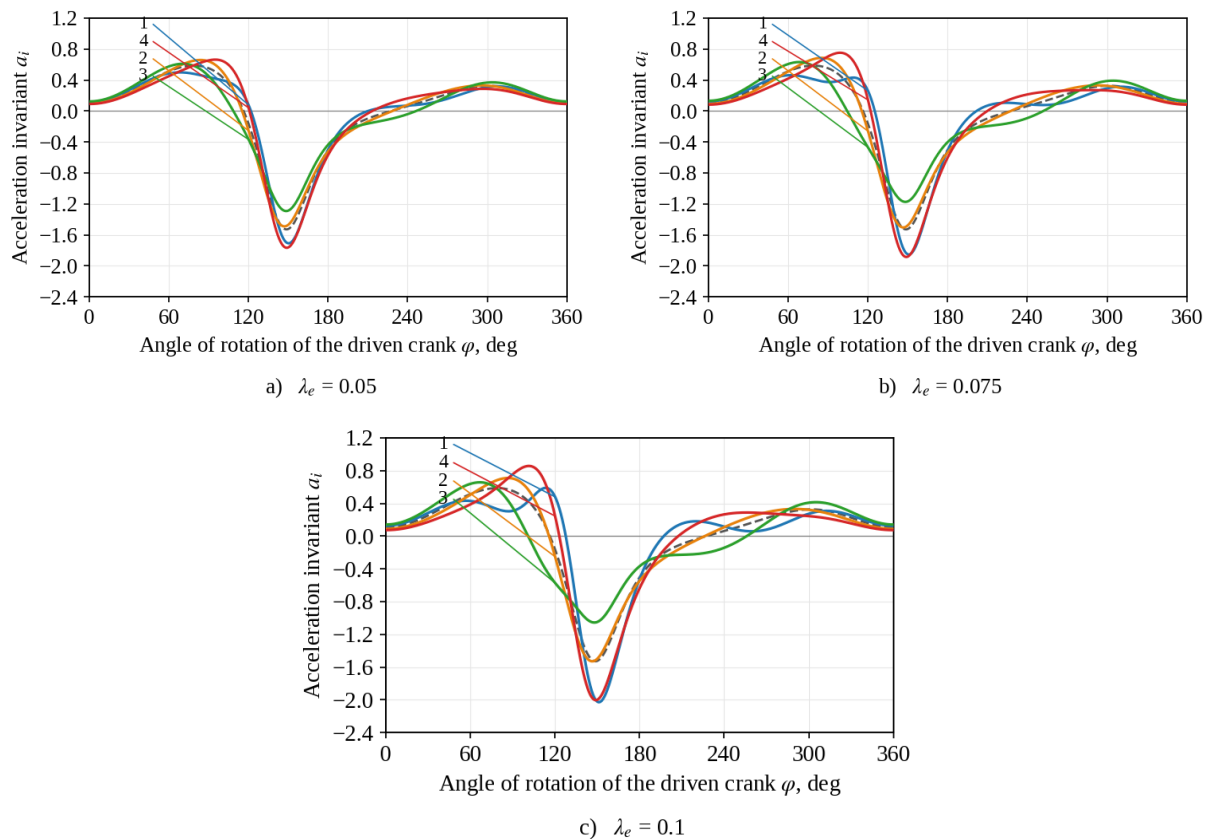
**Figure 8.** Relative acceleration  $a_i$  for (a)  $\lambda_e = 0.05$ , (b)  $\lambda_e = 0.075$  and (c)  $\lambda_e = 0.10$  (1 – upper, 2 – lower, 3 – left, 4 – right; dashed – base)

Figure 9 shows a recorded torque trace for a 0.5 mm blank cut along the fibers at 30 rpm. The monoblock plate produces one peak of about  $1.63$  N·m. With the sectional plate, sequential engagement produces two lower peaks of about  $1.10$  and  $1.31$  N·m. Using the higher sectional-

plate peak, the point-estimate reduction is 19.6%. This is an experimental analog of temporal load redistribution, not a direct test of the rack-driven correction.

Figure 10 summarises the measured peak torque at 30, 60 and 90 rpm for cutting in the MD and CD directions. Peak torque increases nonlinearly with thickness for both plate types, and CD values are consistently higher than MD values. This pattern is consistent with paperboard

anisotropy. At every measured condition, the sectional plate has a lower point estimate than the monoblock plate; without repeatability and uncertainty data, no inferential claim is made.

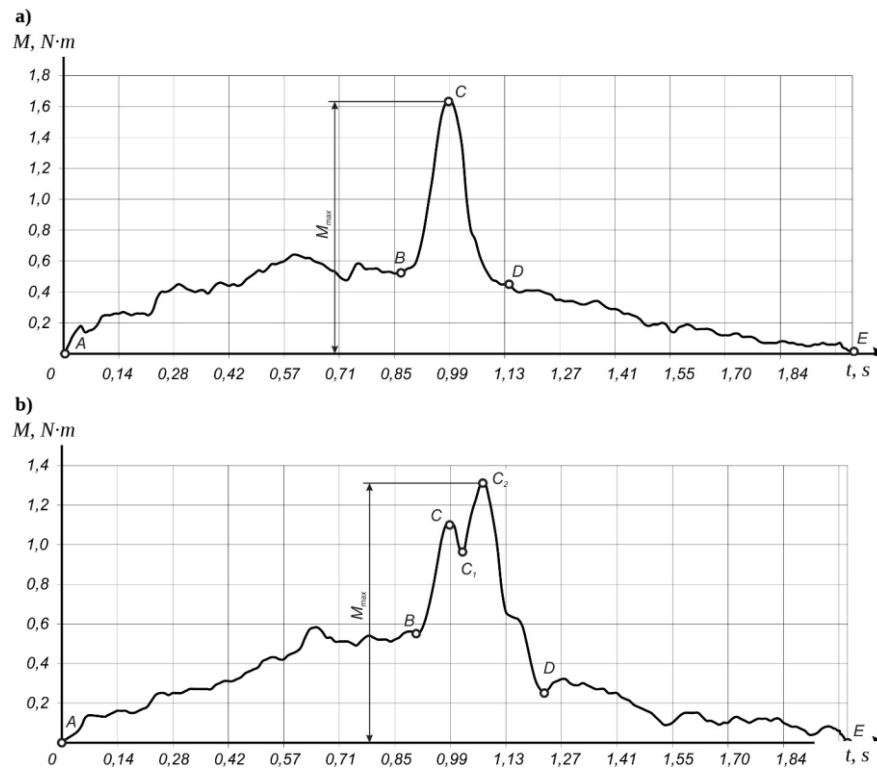


**Figure 9.** Drive-shaft torque  $M$  versus time  $t$  during die-cutting of a 0.5 mm blank along the fibers at 30 rpm: (a) monoblock pressure plate – a single peak C; (b) sectional eccentric pressure plate – two successive peaks C and C<sub>2</sub> (A–E are the characteristic points of the working cycle)

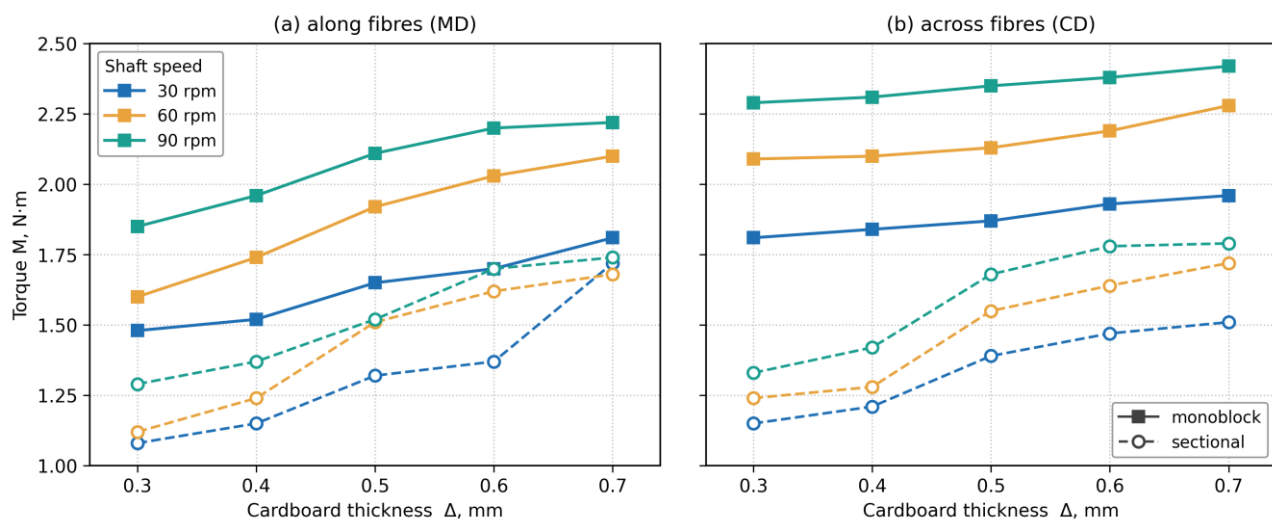


**Figure 10.** Peak drive-shaft torque  $M$  versus cardboard thickness  $\Delta$  for the monoblock and the sectional eccentric pressure plate at 30, 60 and 90 rpm: (a) cutting along the fibers (MD); (b) cutting across the fibers (CD)

Table 5 expresses the sectional-plate reduction relative to the monoblock plate under the same measured condition. The point estimates range from 18.2% to 30.8% in MD and from 22.9% to 41.9% in CD, with the largest differences for the thinner blanks. These data support the

general load-redistribution principle but do not quantify the performance of the proposed rack-driven single-plate mechanism.

**Table 5.** Relative reduction of the peak drive-shaft torque provided by the sectional eccentric plate (%), compared with the monoblock plate.

| $\Delta$ , mm             | 30 rpm | 60 rpm | 90 rpm |
|---------------------------|--------|--------|--------|
| <b>Along fibres (MD)</b>  |        |        |        |
| 0.3                       | 27.0   | 30.0   | 30.8   |
| 0.4                       | 24.3   | 29.3   | 30.6   |
| 0.5                       | 20.0   | 23.4   | 28.4   |
| 0.6                       | 18.9   | 20.7   | 23.2   |
| 0.7                       | 18.2   | 20.0   | 22.5   |
| <b>Across fibres (CD)</b> |        |        |        |
| 0.3                       | 36.5   | 40.7   | 41.9   |
| 0.4                       | 33.9   | 39.3   | 38.5   |
| 0.5                       | 26.2   | 27.6   | 28.5   |
| 0.6                       | 23.0   | 24.7   | 25.6   |
| 0.7                       | 22.9   | 24.6   | 24.8   |

## 5. VALIDATION SCOPE AND LIMITATIONS

The numerical and experimental parts of this study address different questions. The rigid-body model examines the displacement, velocity, and acceleration of the proposed rack-driven single-plate mechanism at a  $1^\circ$  angular resolution. This model does not account for contact compliance, friction, manufacturing tolerances, or a grid-convergence study. As a result, its findings are kinematic predictions rather than actual measurements of machine performance.

The sectional-plate stand provides experimental evidence for the general idea that engaging dies one after another can distribute the load over time. However, its sectional plate, separate eccentric drives, and force path differ from those of the proposed rack-driven single plate. Also, the available dataset does not allow for a repeatability or uncertainty analysis after the fact. Because of this, the

reported torque reductions should be seen as descriptive estimates, not as direct proof or statistically confirmed improvements for the new mechanism.

## 6. CONCLUSION

A kinematic correction was formulated for a single-wedging pressure-plate drive with rack-driven eccentrics in four start positions. Selecting the input stroke  $S_i$  before each calculation keeps  $0 \leq S_i \leq 1$  and retains the 80 mm working stroke.

Eccentric orientation materially changes the finishing motion. At  $\lambda_e = 0.10$ , the left position increases the dwell angle within  $S_i \geq 0.99$  from  $14.37^\circ$  to  $17.41^\circ$  and gives a positive velocity peak of  $+0.7072$ , compared with  $+0.7430$  for the base mechanism and  $+0.8787$  for the right position. The lowest peak overall is  $+0.7028$  at the left position, with  $\lambda_e = 0.075$ .

At  $\lambda_e = 0.10$ , the left position reduces the magnitude of the dominant negative acceleration invariant from 1.5299 to 1.0564, whereas the upper and right positions increase it to 2.0302 and 2.0053. These values describe only the inertial component for a fixed equivalent moving mass.

The left position is the most promising tested orientation for a slower, wider finishing motion. This is a kinematic conclusion; product quality was not measured.

On the related sectional-plate stand, the peak-torque point estimates were 18.2–30.8% lower in MD and 22.9–41.9% lower in CD. They support temporal load redistribution, but remain indirect and lack repeatability and uncertainty estimates.

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