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Normal Force in a Spur Gear Pair under Loading from the Driving Gear of a Korin CA-15 Machine at PT XYZ

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ABSTRACT

Estimating gear transmission loads provides a basis for mechanical assessment, but equipment specifications alone do not establish actual tooth forces. This study aimed to estimate the nominal tangential and normal forces in the spur gear transmission of a Korin CA-15 packing machine at PT XYZ, Indonesia. A descriptive case study combined interviews and direct observation with deterministic mechanical calculations. The analysis adopted a motor power of 0.55 kW, a gear rotational speed of 1480 rpm, a pitch diameter of 32 mm, and an assumed operating pressure angle of 20°. Calculations used SI units and retained intermediate numerical precision. The resulting pitch-line velocity was 2.480 m/s, with nominal tangential and normal forces of 221.795 N and 236.030 N, respectively. These values represent analytical estimates under steady, frictionless conditions rather than measured operating loads. The study provides a reproducible loading reference with explicit assumptions and distinguishes resultant mesh force from individual tooth loading. This reference can support subsequent transmission assessment after confirmation of the operating inputs. However, rated motor specifications, the unverified pressure angle, and the absence of independent measurements limit its interpretation. Direct measurements of gear shaft speed and transmitted torque are required before using the estimates to assess actual loading or component strength.

Keywords:
spur gear transmission; nominal load estimation; tangential force; normal force; analytical mechanics; packaging machinery.

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INTRODUCTION

Research Context and Industrial Problem

Gear transmissions transfer power and rotational motion between machine components. Their mechanical performance depends on the forces transmitted through meshing teeth and the distribution of these forces during engagement (Bai et al., 2023, p. 20844; Yousfi et al., 2020, p. 104741). Characterizing gear loads provides the basis for subsequent assessments of tooth deformation, contact stress, and load-carrying capacity. However, the resultant force transmitted by a gear pair differs from the force carried by each contacting tooth pair because tooth compliance and simultaneous engagement influence load sharing (Bruzzone et al., 2021; Xu et al., 2022).

This study investigates the spur gear transmission of a Korin CA-15 packing machine at PT XYZ, an Indonesian food manufacturer. The available equipment specifications identify a motor power of 0.55 kW, a motor speed of 1480 rpm, and a packing capacity of 40–70 packages per minute. These specifications provide an initial basis for analysis, but they do not independently establish the power, speed, or force at the investigated gear mesh (Amarnath, 2016, p. 561; Can & Bozca, 2023, p. 147). Accordingly, the analysis in this study treats the reported equipment data as nominal inputs and derives the tangential and normal forces from the gear geometry and the assumed operating conditions, consistent with analytical approaches that compute gear loading from the transmitted torque and pitch diameter (Dong et al., 2012, p. 141; Santos, 2019, p. 316).

The specific engineering problem concerns the incomplete characterization of nominal loading in this transmission. The available documentation does not clearly distinguish rated motor power from transmitted gear power or establish whether the investigated gear rotates at the reported motor speed (Mughal et al., 2022, p. 3178; Zhang et al., 2017, p. 207). Consequently, a traceable calculation must connect the adopted operating parameters with the gear geometry and explicitly state the assumptions governing the force estimates.

Gear Types and Their Loading Characteristics

Spur Gears

Spur gears have straight teeth parallel to their rotational axes and transmit power between parallel shafts. Their comparatively simple tooth geometry facilitates manufacture. Under ideal alignment and frictionless contact, the transmitted load produces tangential and radial force components without an axial component. This force configuration provides a direct basis for the nominal loading analysis considered in this study (Stock Drive Products/Sterling Instrument [SDP/SI]) (Lisle et al., 2017, p. 4). Under frictionless conditions, the tangential force is obtained from the transmitted torque and the pitch radius, so that the resultant normal force follows from the pressure angle as a deterministic function of the tangential component (Osakue et al., 2020, p. 295).

The absence of an axial force component does not establish that spur gears have greater load capacity or transmission accuracy than other gear types. Their performance also depends on tooth geometry, manufacturing quality, alignment, and operating conditions (Daffner et al., 2017, p. 9). Consequently, the nominal tangential and normal forces estimated in this study describe the idealized load on the gear pair and should not be interpreted as direct measurements of the tooth forces experienced during actual operation (Samillan et al., 2026, p. 107).

Helical Gears

Helical gears have teeth inclined to the shaft axis. For an external helical gear pair operating on parallel shafts, the gears have equal helix-angle magnitudes and opposite hands (Baguet & Jacquenot, 2010, p. 1792; Benatar et al., 2019). Progressive tooth engagement can provide smoother operation than straight-tooth engagement. However, the helix also generates axial thrust, which introduces additional bearing and shaft support requirements (SDP/SI,).

Straight Bevel Gears

Straight bevel gears have teeth cut on conical pitch surfaces and transmit motion between intersecting shafts, typically at right angles, which distinguishes them from the parallel-axis configurations discussed above (Osakue & Anetor, 2022, p. 588). Straight bevel gears have conical pitch surfaces and straight teeth. They transmit motion between intersecting shafts, commonly arranged at a right angle. Their geometry distinguishes them from spur and parallel-axis helical gears (Fuentes, 2019; Zeng et al., 2018, p. 521). The transmitted load in a straight bevel gear mesh generally produces tangential, radial, and axial components. Its force decomposition therefore requires consideration of the pitch-cone geometry. The spur gear equations used in this study should not be applied directly to bevel gears without accounting for these geometric differences (SDP/SI, n.d.) (Elkholy et al., 1998; Osakue & Anetor, 2018, p. 194). For straight bevel gears, the pitch-cone geometry governs the proportion of the tangential load that is resolved into radial and axial components at the shaft supports, so the bearing layout and thrust paths must be defined from the cone angles before any force estimate is interpreted (Oyague, 2009, p. 30).

Analytical Basis for Spur Gear Force Estimation

For an ideal spur gear pair, the normal force acts along the line of action. Its tangential component transmits torque, while its radial component produces a separating load between the gears. Neglecting friction, their relationship is expressed as: $F_t = F_n cos\ \alpha_w$ (1)

where F_t is the tangential force in N, F_n is the normal force in N, and α_w is the operating pressure angle.

The pitch-line velocity follows from the operating pitch diameter and gear rotational speed:

$$v = \frac{\pi\, d_p n}{60\times1000} \quad (2)$$

where (v) is the pitch-line velocity in $\frac{m}{s}$, d_p is the operating pitch diameter in mm, and \(n\) is the rotational speed of the investigated gear in rpm.

For transmitted power P_t expressed in kW, the relationship between power, tangential force, and pitch-line velocity is

$$P_t = \frac{F_t v}{1000} \quad (3)$$

The tangential force is therefore

$$F_t = \frac{1000\ P_t}{v} \quad (4)$$

The radial and normal forces are obtained from

$$F_r = F_t tan\ \alpha_w \quad (5)$$

and

$$F_n = \frac{F_t}{cos\ \alpha_w} \quad (6)$$

where F_r is the radial force in N. These expressions represent the decomposition of the resultant transmitted force under the stated assumptions (SDP/SI).

The power adopted in Equation (4) must correspond to the investigated transmission location. Rated motor power can serve as an assumed input, but it should not be presented as measured transmitted power. Similarly, the speed used in Equation (2) must represent the investigated gear shaft or an explicitly stated approximation. These equations establish a nominal resultant load. They do not determine how that load divides between simultaneously engaged tooth pairs. The contact ratio describes the extent of overlapping engagement, while tooth compliance and meshing conditions influence the instantaneous load distribution. Therefore, the resultant normal force should not automatically be interpreted as the force acting on each individual tooth pair (Bruzzone et al., 2021; Xu et al., 2022).

Previous Research and Limitations of Existing Approaches

Previous research has extended gear analysis beyond nominal force equilibrium by incorporating tooth deformation and changing contact conditions. A three-dimensional nonlinear formulation estimated spur gear tooth deflection, contact conditions, and load sharing without prescribing the number of contacting pairs in advance (Bruzzone et al., 2021).(Mouton et al., 2024; Sánchez et al., 2016). A dynamic formulation subsequently represented the meshing states of multiple tooth pairs and examined the effects of deviations, speed, load, and damping on meshing forces (Özgüven & Houser, 1988; Xu et al., 2022). These approaches demonstrate why a single nominal force cannot fully represent tooth loading throughout the engagement cycle.

Friction introduces another limitation when nominal forces are used to infer tooth stress. An analytical investigation of symmetric and asymmetric spur gears

incorporated friction at different contact locations and verified its predictions through finite element analysis and photoelasticity. The findings showed that friction assumptions influence both fillet and contact stress predictions (Toman & Abdullah, 2024). Together, these studies establish that detailed loading assessment requires information about tooth geometry, compliance, contact conditions, and operating excitation(Cornell, 1981, p. 462; Lewicki et al., 2001; Yang et al., 2022, p. 12). Such models offer greater physical detail, but their application depends on the availability of corresponding input and validation data. A nominal analytical assessment remains useful when its purpose, assumptions, and interpretation are clearly bounded.

Research Gap and Proposed Approach

The reviewed literature already provides methods for evaluating spur gear forces, deformation, and dynamic behavior. The present study therefore addresses an application gap in the characterization of the investigated Korin CA-15 transmission. The unresolved issue is how the available machine parameters translate into nominal tangential and normal force estimates and what those estimates can reasonably describe.

The proposed approach combines descriptive field investigation with established analytical relationships. Interviews and direct observations provide information about the equipment, while the force analysis connects the adopted power, rotational speed, pitch diameter, and pressure angle. Its distinction from the reviewed studies lies in its application to a documented packing-machine transmission using the available equipment data. It does not introduce a new contact model or claim to reproduce instantaneous tooth loading.

Research Objectives and Questions

The principal objective is to estimate the nominal tangential and normal forces in the investigated spur gear transmission while making the calculation assumptions explicit. The study addresses the following questions. 1) What pitch-line velocity follows from the adopted gear diameter and rotational speed? 2) What nominal tangential, radial, and normal forces follow from the adopted transmitted power and operating pressure angle? 3) What limitations do the available machine data and simplifying assumptions impose on the interpretation of these estimates?

Intended Scientific and Technological Contributions

The intended scientific contribution is a transparent characterization of nominal gear loading that distinguishes resultant transmitted force from local tooth loading and dynamic contact behavior. The intended technological contribution is a documented loading reference for subsequent stress assessment, transmission inspection, or instrumented testing. Whether these nominal estimates adequately represent actual operating loads remains unresolved. The study investigates the loading implied by the available parameters and identifies the conditions governing its interpretation. The findings are expected to support manufacturing engineering by improving the traceability of preliminary transmission assessments and clarifying the additional evidence needed for more detailed analysis.

MATERIALS AND METHODS

Research Design and Industrial Setting

This study employed a descriptive case study supported by deterministic mechanical calculations to characterize the nominal loading of a spur gear transmission in a Korin CA-15 packing machine at PT XYZ, Indonesia. The investigation focused on estimating pitch-line velocity, tangential force, and normal force using the machine parameters documented in the study. The machine was examined in its industrial packing setting. Interviews and direct observations provided contextual information about the equipment. The mechanical analysis used established relationships between rotational speed, pitch diameter, transmitted power, and operating pressure angle.

The study did not manipulate operating parameters or compare alternative gear designs. It therefore represents a nominal loading assessment for one machine configuration rather than an experimental evaluation of transmission performance.

Equipment and Calculation Inputs

The equipment comprised a Korin CA-15 packing machine and its electric drive motor. Table 1 summarizes the specifications reported in the source record.

Table 1. Equipment specifications reported in the study

Equipment	Specification	Reported value
Packing machine	Model	Korin CA-15
Packing machine	Packing capacity	40–70 packages/min
Packing machine	Mass	250 kg
Packing machine	Length	765 mm
Packing machine	Width	710 mm
Packing machine	Height	1930 mm
Drive motor	Voltage	220 V
Drive motor	Current	5.35 A
Drive motor	Rated power	0.55 kW
Drive motor	Frequency	50 Hz
Drive motor	Rated rotational speed	1480 rpm

Table 1 identifies the inputs used to reconstruct the reported calculations. The distinction between equipment specifications and analytical assumptions is maintained because the available record does not establish all inputs through direct measurement.

Table 2. Inputs and assumptions for the nominal force analysis

Parameter	Symbol	Adopted value	Basis and interpretation
Power used in the force calculation	P_d	0.55 kW	Rated motor power adopted as the analytical power input
Gear rotational speed	n	1480 rpm	Motor speed adopted for the investigated gear
Pitch diameter	d_p	32 mm	Value reported in the calculation
Power correction factor	f_c	1.0	Value shown in the source calculation
Operating pressure angle	α_w	\(20^\circ\)	Assumption consistent with the reported force relationship
Pitch-line velocity used in the force calculation	v	2.478 m/s	Numerical value reported in Results

The source record does not identify the gear material, heat treatment, tooth count, module, face width, lubrication

condition, or manufacturing accuracy. Consequently, the analysis does not evaluate material strength, contact stress, bending stress, wear rate, or fatigue life. The record also does not identify the instruments used to establish the pitch diameter or operating speed. Measurement resolution, calibration status, and measurement uncertainty therefore remain unspecified.

Data Collection and Parameter Definition

Data collection involved interviews with company personnel and direct observation of the packing equipment. Interviews addressed the machine and its operation, while observations identified the equipment under investigation. The available record does not specify the number of interviewees, observation duration, or repeated measurement sessions. The analytical inputs comprised power, rotational speed, pitch diameter, pressure angle, and the power correction factor. The calculated responses comprised pitch-line velocity, tangential force, and normal force. These quantities were treated as deterministic calculation inputs and outputs rather than experimentally manipulated independent and dependent variables. The reported rotational speed was adopted as the gear speed for calculation purposes. This assumption requires confirmation if an intermediate transmission exists between the motor and the investigated gear. Similarly, the adopted power represents a nominal calculation input rather than measured power at the gear mesh. No direct force measurement, torque measurement, or continuous operating data acquisition was reported. The calculated forces were therefore interpreted as nominal estimates under the adopted assumptions.

Analytical Model and Calculation Procedure

Modelling Assumptions

The analysis represented the transmission as an ideal spur gear pair operating under steady nominal conditions. The calculation assumed constant rotational speed, constant transmitted power, and frictionless tooth contact.

The adopted pressure angle remained constant. Dynamic excitation, backlash, shaft misalignment, tooth deformation, and transmission losses were not explicitly modelled. The calculated normal force represented the resultant transmitted mesh load rather than the instantaneous force carried by an individual tooth pair.

This distinction matters because simultaneous engagement and tooth compliance can redistribute the transmitted load among contacting teeth (Bruzzone et al., 2021; Xu et al., 2022).

Pitch-Line Velocity

Pitch-line velocity was calculated from the operating pitch diameter and adopted rotational speed.

$$v = \frac{\pi \times d_p \times n}{60 \times 1000} \quad (1)$$

where v denotes pitch-line velocity in m/s, d_p denotes pitch diameter in mm, and n denotes gear rotational speed in rpm.

The denominator converts rotational speed from revolutions per minute to revolutions per second and diameter from millimetres to metres.

The source calculation used $\pi = 3.14$ and reported $v = 2.478 \frac{m}{s}$. This reported velocity was retained when reconstructing the subsequent force calculations. Its retained precision must be distinguished from a calculation performed using the full numerical value of π.

Power Adopted for Analysis

The power used in the source calculation was obtained from $P_d = f_c P_m$ (2)

where P_d denotes the adopted analytical power in kW, P_m denotes rated motor power in kW, and f_c denotes the dimensionless power correction factor.

The calculation adopted $f_c = 1.0$. This factor does not establish transmission efficiency or confirm that the motor operated at its rated power during observation.

Tangential Force

In SI units, tangential force follows from the relationship between transmitted power and pitch-line velocity.

$$F_t = \frac{1000 P_d}{v} \quad (3)$$

where F_t is expressed in N.

The source Results instead used the approximate gravitational-unit relationship

$$F_{t,kgf} \approx \frac{102 P_d}{v} \quad (4)$$

where $F_{t,kgf}$ is expressed in kgf. The coefficient 102 approximates the conversion between kilowatts and kilogram-force metres per second.

Equation (4) was retained specifically to reconstruct the reported tangential force. For unit conversion,

$$1\ kgf = 9.80665\ N \quad (5)$$

The approximate gravitational-unit calculation and the direct SI calculation can produce slightly different numerical values because the coefficient 102 is rounded.

Normal Force

For an ideal frictionless spur gear mesh, the normal force was calculated from

$$F_n = \frac{F_t}{\cos \alpha_w} \quad (6)$$

where F_n and F_t use the same force unit and α_w denotes the operating pressure angle.

The analysis adopted $\alpha_w = 20°$, consistent with the reported relationship between tangential and normal force. The available record does not establish this angle through direct measurement or a manufacturer drawing.

The radial force relationship,

$$F_r = F_t \tan \alpha_w \quad (7)$$

defines the remaining in-plane force component. However, the existing Results do not report a radial-force calculation. Accordingly, radial force was not treated as a separately evaluated outcome.

Calculation Consistency and Reproducibility

The analysis comprises a single deterministic calculation case. No repeated trials, independent force measurements, finite element comparisons, or experimental validation datasets were reported. Evaluation is therefore limited to the reproducibility and dimensional consistency of the analytical calculations. Reconstruction requires the inputs listed in Table 2 and the following calculation sequence. 1) Calculate pitch-line velocity from pitch diameter and rotational speed. 2) Establish the analytical power using the reported correction factor. 3) Calculate tangential force using a clearly identified unit convention. 4) Calculate normal force using the adopted pressure angle. 5) Report numerical precision and distinguish intermediate values from final rounded outputs. The original record does not identify the calculation

software or its version. The equations can be implemented using a scientific calculator or a spreadsheet, provided that trigonometric functions evaluate the pressure angle correctly in degrees.

No statistical hypothesis tests, significance thresholds, outlier removal, or missing-value imputation were applicable to the reported single-case calculation. Unreported information was treated as unavailable rather than assigned invented values. Measurement uncertainty could not be quantified because instrument accuracy, repeated measurements, and operating variability were not documented. Potential bias arises primarily from adopting rated motor specifications as gear operating inputs and using an assumed pressure angle. These limitations restrict the interpretation of the results to nominal loading under the stated conditions.

RESULTS

Pitch-Line Velocity

The nominal loading analysis used an adopted gear rotational speed of 1480 rpm and a pitch diameter of 32 mm. Substitution into the pitch-line velocity equation yielded $v = \frac{\pi \times 32 \times 1480}{60 \times 1000} = 2.479763801 \frac{m}{s}$. The calculated pitch-line velocity was therefore 2.480 m/s, rounded to three decimal places. This value represents the velocity implied by the adopted gear diameter and rotational speed. It was not obtained through direct velocity measurement.

Nominal Tangential Force

The analysis adopted a transmitted power of 0.55 kW, corresponding to the rated motor power and a power correction factor of 1.0. Using the unrounded pitch-line velocity, the nominal tangential force was calculated as: $F_t = \frac{1000 \times 0.55}{2.479763801} = 221.7953177\ N$. The resulting tangential force was 221.795 N. This estimate describes the circumferential force associated with transmitting the adopted power at the calculated pitch-line velocity. Its interpretation depends on the assumption that the rated motor power represents the power available at the investigated gear mesh.

Nominal Normal Force

The normal force was calculated using the tangential force and an assumed operating pressure angle of 20°. $F_n = \frac{221.7953177}{cos\ 20°} = 236.0296471\ N$. The resulting normal force was 236.030 N. Its magnitude exceeded the tangential component, consistent with the adopted force decomposition $F_t = F_n cos\ \alpha_w$

This result represents the resultant normal load transmitted through the idealized gear mesh. The calculation does not establish the instantaneous load carried by each tooth pair during simultaneous engagement.

Summary and Numerical Consistency

Table 3 summarizes the recalculated outputs. All intermediate calculations retained numerical precision, and the displayed results were rounded to three decimal places.

Table 3. Recalculated nominal loading of the investigated spur gear transmission

Calculated quantity	Symbol	Value	Unit
Pitch-line velocity	\(v\)	2.480	m/s
Tangential force	\(F_t\)	221.795	N
Normal force	\(F_n\)	236.030	N

Two algebraic checks confirmed the internal consistency of the calculations. Using the unrounded values, the calculated tangential force and velocity recovered the adopted power. $\frac{F_t \times v}{1000} = 0.550\ kW$. Similarly, projecting the calculated normal force onto the tangential direction recovered the tangential force. $F_n cos\ 20° = 221.7953177\ N$

These checks confirm consistency between the equations and numerical outputs. They do not constitute independent validation of the assumed operating conditions or the actual machine loads. Table 4 distinguishes the original reported values from the standardized recalculation.

Table 4. Reconciliation of the original and recalculated results

Quantity	Original reported value	Standardized recalculation	Basis of the difference
Pitch-line velocity	2.478 m/s	2.480 m/s	Use of the full numerical value of \(\pi\) and rounding only at the reporting stage
Tangential force	22.639, labelled kg	221.795 N	Direct SI calculation replaces the approximate gravitational-unit coefficient 102
Normal force	24.091, labelled kg	236.030 N	Recalculation from the unrounded tangential force and the assumed \(20^\circ\) pressure angle

The original force equation indicates a gravitational-unit calculation, for which the appropriate force unit is kgf rather than kg. The revised values were calculated directly in SI units rather than obtained by relabelling the original numerical results.

Evidence Boundaries

The available data support a nominal analytical estimate for one adopted operating condition. They do not establish variation in loading across the stated packing capacity of 40–70 packages per minute because no operating-point measurements were provided. No direct force measurements, repeated trials, or independent reference results were available. Consequently, prediction errors, confidence intervals, statistical significance, and measurement uncertainty could not be determined. The available evidence also does not establish contact stress, fatigue life, wear, energy efficiency, or changes in machine productivity. The reported outputs should therefore be interpreted as conditional estimates based on the adopted power, speed, pitch diameter, and pressure angle.

DISCUSSION

Interpretation of the Principal Findings

The analysis establishes a nominal loading reference for the investigated Korin CA-15 spur gear transmission. Under the adopted power, rotational speed, pitch diameter, and pressure angle, the calculated normal force represents the resultant load required to transmit the corresponding tangential force through an ideal frictionless mesh. These findings answer the calculation-based research questions, but they do not establish the actual operating load of the machine. The normal force exceeds its tangential component because the line of action is inclined to the tangential direction. This relationship follows directly from the assumed pressure angle and force decomposition. It should therefore be interpreted as a consequence of the analytical model rather than an independently observed characteristic of the machine. The distinction between resultant mesh load and individual tooth loading is particularly important. The calculated normal force does not indicate that every engaged tooth pair carries the same load. Tooth deformation and changing contact conditions influence how the transmitted load is distributed during engagement, as represented in more detailed spur gear models (Bruzzone et al., 2021; Sánchez et al., 2016).

Comparison with Previous Research

The closest published studies address the same underlying problem of force transmission through meshing gear teeth, but they evaluate physical mechanisms beyond the scope of the present calculation. A three-dimensional nonlinear approach determined tooth deflection, contact conditions, and load sharing without prescribing the number of contacting tooth pairs in advance (Bai et al., 2023, p. 20844; Bruzzone et al., 2021; Litvin et al., 1996). A dynamic formulation distinguished the meshing states of multiple tooth pairs and evaluated their contributions to meshing force and load sharing (Shi et al., 2021, p. 2060; Xu et al., 2022). The present analysis provides an overall nominal force estimate, whereas these studies resolve changes within the engagement process. Consequently, the present model is best regarded as a boundary-condition estimate that establishes the resultant mesh load, while tooth-level load sharing requires a stiffness-based formulation that accounts for the coupling between contacting slices and tooth pairs (Riera et al., 2026, p. 17). Friction also differentiates nominal force estimation from detailed stress assessment. An analytical investigation incorporated friction into predictions of spur gear fillet and contact stresses and verified the formulation using finite element analysis and photoelasticity (Toman & Abdullah, 2024). The present study excludes friction and does not calculate stress. Consequently, agreement with the basic force relationship cannot establish agreement with experimentally verified contact or bending stress predictions.

Research incorporating gear modification and localized cracking further demonstrates the additional information required to evaluate damaged or geometrically modified transmissions. Slice coupling theory has been used to determine tooth deformation and force transmission under these conditions (Dai & Liang, 2024). Such mechanisms remain outside the present model because the available data do not characterize tooth modification, crack geometry, or local compliance (Peng et al., 2019, p. 536). A direct comparison of numerical force magnitudes across these studies would not provide a valid accuracy assessment. The gear geometries, applied loads, operating conditions, and model outputs are not matched. The comparison therefore concerns analytical scope and interpretation rather than numerical superiority.

Mechanical Relationships Governing the Estimates

The calculated forces follow the relationship between transmitted power and pitch-line velocity. Within the adopted model, tangential force increases with transmitted power when velocity remains constant. At fixed transmitted power and pitch diameter, increasing rotational speed reduces the corresponding nominal tangential force. These are mathematical dependencies of the model, not experimentally demonstrated responses of the Korin CA-15 machine. This distinction prevents an inappropriate operating recommendation. A lower calculated nominal force at higher speed does not establish that increasing machine speed would reduce actual tooth loading. Dynamic meshing forces also depend on engagement conditions, deviations, and damping, which the present calculation does not represent (İnalpolat & Kahraman, 2009, p. 788; Śliwiński, 2023, p. 17048; Xu et al., 2022).

The assumed pressure angle governs the conversion from tangential to normal force. Because the angle was not independently confirmed, the normal-force estimate remains conditional on that assumption. Likewise, the adopted power and speed must correspond to the investigated gear shaft for the estimates to represent its nominal loading. The numerical differences between the original and revised Results arise from unit treatment and calculation precision. Using SI equations and retaining intermediate precision improves numerical traceability. It does not improve the empirical accuracy of unmeasured input parameters.

Contribution and Conditions of Applicability

The contribution is an application-based loading assessment for a specific packing-machine transmission. The study connects documented equipment parameters with explicit mechanical assumptions and distinguishes nominal transmitted force from local tooth loading. It also clarifies the force-unit convention used in the original calculation. This contribution supports preliminary industrial engineering assessment, but it does not introduce a new gear model or independently confirm existing contact mechanics theory. The findings are consistent with established force relationships because those relationships generate the results. The study therefore illustrates their application and qualifies the interpretation of the resulting estimates. The approach is most appropriate when the objective is to establish a preliminary loading reference and when transmitted power, gear speed, pitch diameter, and pressure angle are known with reasonable confidence. Its limited input requirements make the calculation straightforward to reproduce. Its adequacy decreases when operating loads fluctuate substantially or when the assessment concerns local stress, vibration, damage, or fatigue. Detailed models explicitly account for tooth compliance, multiple-pair engagement, friction, or

localized defects, demonstrating why nominal equilibrium alone cannot resolve these outcomes (Bruzzone et al., 2021; Dai & Liang, 2024; Toman & Abdullah, 2024).

Practical Implications

The nominal loading reference can provide an initial input for subsequent transmission assessment. After confirming the operating parameters, engineers could combine the estimated load with tooth geometry and material properties to investigate contact or bending stress. Shaft and bearing assessment would additionally require the relevant force components, support geometry, and load locations.

The results can also guide data collection. Confirming gear shaft speed, identifying the actual transmission path, and establishing transmitted torque would directly address the main uncertainties in the current assessment. Recording loading across representative packing conditions would help determine whether a single nominal operating point adequately describes the machine.

These applications represent potential uses of the analysis. The study does not demonstrate reduced downtime, improved product quality, higher productivity, energy savings, lower material consumption, or environmental benefits. Such outcomes would require corresponding measurements and an appropriate comparison design.

Robustness, Generalizability, and Threats to Validity

The calculation is reproducible for the stated inputs, but its empirical robustness remains untested. Recovering the adopted power from force and velocity confirms algebraic consistency only. It does not verify that the machine transmitted that power during operation. The principal threat to input validity is the use of rated motor specifications as gear operating parameters. Rated power does not establish actual transmitted power, while an intermediate transmission could cause gear speed to differ from motor speed. The assumed pressure angle introduces further uncertainty into the normal-force estimate. Available information does not support a numerical uncertainty interval. The model also lacks the tooth geometry, material properties, lubrication information, and condition measurements required for strength or service-life assessment. A calculated force alone cannot establish whether the transmission operates within an acceptable safety margin. Generalization is limited to the analytical procedure. The equations can be applied to other comparable spur gear systems when appropriate inputs and assumptions are supplied, but the numerical values cannot be transferred to different machines or operating conditions. Even for the investigated machine, the estimates do not establish loading throughout its stated packing-capacity range. Independent validation should therefore precede stronger performance claims. Measurements of shaft speed and transmitted torque under representative conditions would provide a basis for evaluating nominal force estimates. Detailed contact modelling and suitable experimental measurements would then be needed to examine local tooth loading or stress. Until such evidence becomes available, the findings remain a conditional baseline for further investigation.

CONCLUSION

This study estimated the nominal loading of the Korin CA-15 spur gear transmission using documented equipment parameters and established mechanical relationships. With an adopted power of 0.55 kW, rotational speed of 1480 rpm, pitch diameter of 32 mm, and assumed pressure angle of \(20^\\circ\), the recalculated pitch-line velocity, tangential force, and normal force were **2.480 m/s, 221.795 N, and 236.030 N**, respectively. These values are analytical estimates rather than measured operating loads.

The contribution is a reproducible loading reference with consistent SI units and explicit assumptions. It provides an initial input for subsequent transmission assessment but does not establish tooth strength, fatigue life, or improved machine performance.

The principal limitations are the use of rated motor specifications as gear operating inputs, an unverified pressure angle, and the absence of independent measurements and uncertainty estimates. Future research should measure gear shaft speed and transmitted torque across representative packing conditions, confirm the gear geometry, and compare the resulting loads with the nominal estimates.

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Author contributions using CRediT
Yanto: Conceptualization, Methodology, Investigation, Data curation, Formal analysis, Writing – original draft, Writing – review and editing.
Restu Adiyaksa: Formal analysis, Writing – original draft, Writing – review and editing.

Data availability statement.
The equipment specifications, analytical inputs, assumptions, equations, and calculated outputs supporting this study are reported in the article.

Consent for Publication
Consent for publication of identifiable personal information is not applicable because the manuscript contains no identifiable personal information.

Code, software, or supplementary material availability
No custom software or code was developed for this study. No supplementary materials accompany this article.

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