

Design Verification, Uncertainty Analysis and Criticality-Based Maintenance of a Long Belt Conveyor in a Cement Plant: The Case of Conveyor CV-301 at the Grande Cimenterie du Katanga, Democratic Republic of the Congo

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Abstract: Unplanned stoppages of long belt conveyors interrupt entire production lines in mines and cement plants, yet field verifications rarely report the uncertainty of the parameters they rely on. This study verifies the design of conveyor CV-301 at the Grande Cimenterie du Katanga (Democratic Republic of the Congo), which carries 600 t/h of a gypsum–rock mixture over a centre distance of about 497 m and whose downtime is dominated by belt tearing. Measured installation data were combined with an analytical model based on ISO 5048 and DIN 22101 covering load cross-section, motion resistances, drive capability, tension design (slip, sag, and start-up), belt strength, and idler bearing life. Unmeasured parameters were bounded from standards and the literature and propagated with a 10,000-run Monte Carlo simulation, and failure modes were ranked with a failure modes, effects, and criticality analysis (FMECA). At 1.3 m/s, the capacity ranges from 710 to 855 t/h depending on the surcharge angle, a 70%–85% fill degree. The nominal effective tension is 22.6 kN, and the minimum slack-side tension of 18.8 kN is governed by belt sag rather than drive slip. The EP 1000 belt retains a start-up safety factor of at least 14.1 in all simulated cases (5th percentile 16.7), and the 80 kW drive can sustain a net lift of at least 12.7 m (5th percentile 15.3 m). Calculated idler bearing lives exceed 2×10^5 h. The installation is therefore not undersized; reliability gains depend on loading-zone protection, splice quality, contamination-resistant idlers, and condition monitoring.

Keywords: Belt Conveyor, DIN 22101, ISO 5048, Belt Tension, Monte Carlo Simulation, FMECA, Cement Industry.

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I. INTRODUCTION

Belt conveyors are the dominant means of continuous bulk handling in mining, quarrying, and cement production because they move large tonnages over long distances at low specific energy and labour cost (Spivakovsky & Dyachkov, 1983; Conveyor Equipment Manufacturers Association [CEMA], 2014). Their economic advantage depends on high availability: a single conveyor in series with crushers, stockpiles, and kilns becomes a critical path, and each stoppage starves downstream equipment. Energy studies have also shown that conveyors are frequently operated far

from their design point, which raises both energy use and wear (Zhang & Xia, 2011).

Design and verification methods are well standardized. ISO 5048 (International Organization for Standardization [ISO], 1989) and DIN 22101 (Deutsches Institut für Normung [DIN], 2011) express the effective tension as a sum of main, secondary, special, and slope resistances, and define the tension conditions needed to prevent slip and excessive belt sag. However, these methods require parameters that are rarely measured on site, notably the artificial friction coefficient f , the masses of rotating idler parts, and the belt–pulley friction coefficient μ . Field measurements report f

values between about 0.012 and 0.035 depending on load, idler type, and maintenance condition (Gładysiewicz et al., 2017), a range wide enough to change design conclusions. Verifications that use single nominal values therefore give an incomplete picture of the actual safety margins.

Failure studies of textile-carcass belts identify impact of falling material at loading points as a primary cause of punctures and longitudinal tears (Fedorko et al., 2014; Andrejiová & Grinčová, 2016), while idler bearing failures and splice separations dominate the remaining downtime. Failure modes, effects, and criticality analysis (FMECA; International Electrotechnical Commission [IEC], 2018) is the standard tool for ranking such failure modes, but its scores are often reported without any assessment of their robustness.

The Grande Cimenterie du Katanga (GCK), at Likasi in the Haut-Katanga province of the Democratic Republic of the Congo, operates conveyor CV-301 between its crushing plant and storage hall. Plant records identify belt tearing as the first cause of stoppage on this conveyor, followed by splice repairs, idler replacement, and tracking adjustments (Nkulu Kabembo, 2025). A preliminary verification based on catalogue capacity tables concluded that the conveyor was adequately sized but did not quantify tensions, drive reserve, or the effect of parameter uncertainty.

This article addresses that gap. Its objectives are (a) to verify the conveying capacity of CV-301 with the ISO 5048 cross-section model and to compare it with catalogue values; (b) to compute the motion resistances, drive reserve, and tension distribution, identifying which condition governs the take-up design; (c) to assess the belt and idler bearings against their strength and life criteria; (d) to propagate the uncertainty of unmeasured parameters with a Monte Carlo simulation; and (e) to rank failure modes with an FMECA whose robustness is tested, so as to derive maintenance, safety, and monitoring measures.

II. METHOD

➤ Study Site and Measured Data

Data were collected at the GCK plant through site inspections, the plant's technical documentation, and interviews with maintenance staff during the 2024–2025 academic year (Nkulu Kabembo, 2025). The general arrangement of CV-301 is shown in Figure 1, and the measured or documented characteristics are listed in Table 1. The documented length of 997.21 m refers to the endless belt; neglecting take-up loops, it corresponds to a centre distance A of about 497 m (Equation 5), which is the length used in the resistance calculations.

Table 1 Measured and Documented Characteristics of Conveyor CV-301

Parameter	Symbol	Value
Conveyed material	–	Crushed gypsum–rock mixture, abrasive
Bulk density	ρ	1.32 t/m ³
Required mass flow	Q_m	600 t/h, 24 h/day
Belt length (endless)	L_b	997.21 m
Belt width	B	1,200 mm
Belt type	–	EP 1000/4 textile carcass, 12 mm (covers 6 + 2 mm)
Carrying idlers	l_3, λ	3 equal rolls, 465 mm; trough angle 20°
Return idlers	–	Flat, 1,400 mm
Drive / tail / snub pulley diameter	D, d, D_1	800 / 630 / 500 mm (face 1,400 mm)
Motor	P_M, n_e	86 kW, 1,480 rpm
Gear unit output	P_s, n_s	80 kW, 36.57 rpm
Take-up	–	Gravity (counterweight tower with winch)
Operating speed adopted by the plant	v	1.3 m/s

Note. Data from Plant Documentation and Site Inspection (Nkulu Kabembo, 2025).

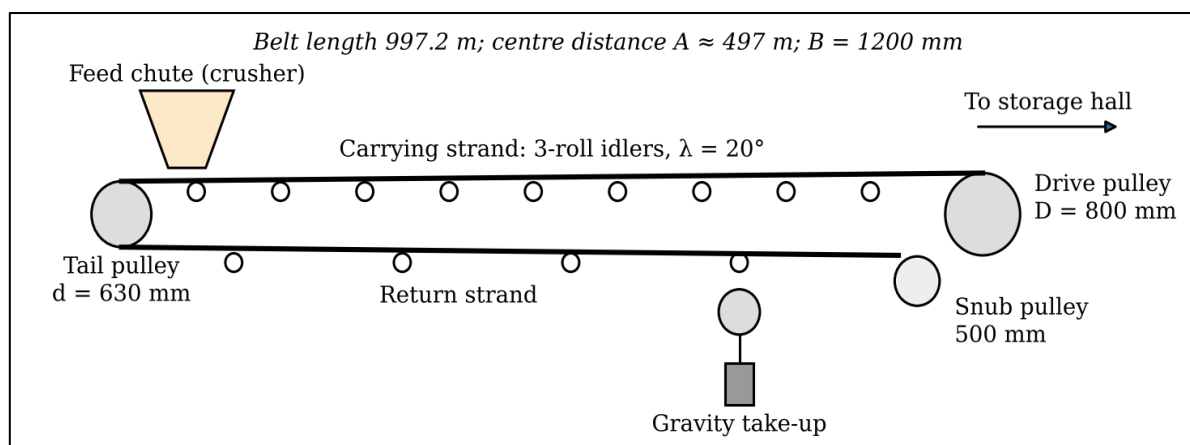


Fig 1 General Arrangement of Conveyor CV-301

Note. Not to Scale. Idler Positions and the Location of the Gravity Take-Up are Schematic.

➤ Analytical Model

All calculations were implemented in Python 3 with NumPy. Unless stated otherwise, the conveyor is treated as horizontal (net lift $H = 0$), because its lift has not been surveyed; the effect of lift is analysed separately.

• Conveying Capacity

The volumetric and mass flows follow from the load cross-section S and the belt speed v :

$$Q_v = 3600 S v \quad (1)$$

$$Q_m = \rho Q_v \quad (2)$$

For a three-roll trough of equal rolls, ISO 5048 (ISO, 1989) splits the cross-section into a surcharge segment S_1 above the trough edges and a trapezoid S_2 inside the trough (Figure 2a):

$$S_1 = \frac{[l_3 + (b - l_3) \cos \lambda]^2 \tan \theta}{6} \quad (3)$$

$$S_2 = [l_3 + \frac{b - l_3}{2} \cos \lambda] \frac{b - l_3}{2} \sin \lambda, \quad b = 0.9B - 0.05 \quad (4)$$

Where l_3 is the roll length, λ the trough angle, θ the dynamic surcharge angle, and b the usable belt width (m). Catalogue capacities (Rulmeca Group, 2014) were compared with Equations 3 and 4 for θ between 15° and 25° .

• Geometry and Drive Kinematics

The centre distance was recovered from the endless belt length, and the belt speed imposed by the drive from the gear-unit output speed:

$$L_b \approx 2A + \frac{\pi(D+d)}{2} \quad (5)$$

$$k = \frac{n_e}{n_s}, \quad v_t = \frac{\pi D n_s}{60} \quad (6)$$

The output torque and the maximum effective tension the drive can transmit at the drive pulley of radius $R = D/2$ are:

$$C_s = \frac{60 P_s}{2\pi n_s}, \quad F_{U,max} = \frac{C_s}{R} \quad (7)$$

• Motion Resistances

Following ISO 5048 and DIN 22101 (DIN, 2011), the effective tension at the drive pulley is the main resistance F_H increased by the length-dependent coefficient C for secondary resistances, plus special resistances F_S (scrapers, skirtboards) and the slope resistance F_{St} :

$$F_U = C F_H + F_S + F_{St} \quad (8)$$

$$F_H = f A g [q_{RO} + q_{RU} + (2q_B + q_G) \cos \delta] \quad (9)$$

$$q_G = \frac{Q_m}{3.6 v}, \quad F_{St} = q_G H g \quad (10)$$

Where f is the artificial friction coefficient, $g = 9.81$ m/s², q_{RO} and q_{RU} are the rotating masses of carrying and return idlers per metre, q_B and q_G the masses of belt and material per metre (kg/m), and δ the inclination. The resistance of the return strand alone, needed for the tension distribution, is:

$$F_{ret} = C f A g (q_{RU} + q_B) \quad (11)$$

The power required at the motor, with an overall drive efficiency η , is $P_M = F_U \cdot v / \eta$. The admissible net lift is the lift at which F_U reaches the drive limit:

$$H_{adm} = \frac{F_{U,max} - F_U(H=0)}{q_G g} \quad (12)$$

• Tension Design

Traction is transmitted by friction. For a belt element of length $dx = R \cdot d\alpha$ on the pulley, radial equilibrium gives $N \cdot dx = T \cdot d\alpha$ and tangential equilibrium $dT = \mu N \cdot dx$, hence $dT/T = \mu \cdot d\alpha$. Integrating over the wrap angle α yields the Euler–Eytelwein condition for no slip:

$$\frac{T_1}{T_2} \leq e^{\mu\alpha} \quad (13)$$

Where T_1 and T_2 are the tight-side and slack-side tensions. Three conditions bound T_2 . First, slip must be prevented during start-up, when the effective tension is increased by a start-up factor ξ . Second, the sag between carrying idlers spaced a_0 must not exceed the admissible ratio $(h/a)_{adm}$, which imposes a minimum tension at the tail end of the carrying strand. Third, the same criterion applies to the return strand with spacing a_U :

$$T_2 \geq \frac{\xi F_U}{e^{\mu\alpha} - 1} \quad (14)$$

$$T_{min} = \frac{a_0(q_B + q_G)g}{8(h/a)_{adm}} \quad (15)$$

$$T_{tail} = T_2 + F_{ret} \geq T_{min}, \quad T_1 = T_2 + F_U \quad (16)$$

For a head drive, the minimum tension occurs at the drive exit. T_2 was set to the largest of the three requirements, and the corresponding counterweight of a take-up located near the drive is $G_{TU} \approx 2T_2/g$.

• Belt Strength and Idler Bearing Life

The safety factor of a textile belt of nominal strength k_N (N/mm) is evaluated in steady state (T_1) and at start-up ($T_1 + (\xi - 1)F_U$):

$$s = \frac{k_N B}{T_1} \quad (17)$$

A minimum of about 10 is commonly required for textile belts in steady operation to allow for splice efficiency and dynamic loads (DIN, 2011; Rulmeca Group, 2014). The load on the central carrying roll and the basic rating life of its bearings follow Rulmeca Group (2014) and ISO 281 (ISO, 2007):

$$P_c = a_0(q_B + F_p q_G) g F_s F_m \quad (18)$$

$$L_{10h} = \left(\frac{C_r}{P}\right)^3 \frac{10^6}{60n} \quad (19)$$

Where $F_p = 0.65$ is the participation factor of the central roll for a 20° trough, F_s and F_m are service and environment factors, $P = P_c/2$ is the load per bearing, C_r the dynamic load rating, and n the roll speed.

➤ Parameter Classification and Uncertainty Analysis

Parameters that could not be measured were assigned a nominal value and a plausible range from standards,

manufacturer data, and published measurements (Table 2). Uncertainty was propagated with a Monte Carlo simulation of $N = 10,000$ runs (pseudo-random seed 2025), sampling each bounded parameter independently from a uniform distribution, which is the least informative choice when only bounds are known. Outputs are reported as 5th, 50th, and 95th percentiles (P5, P50, P95) and minima. A one-at-a-time sensitivity analysis was also performed by setting each parameter to its bounds while holding the others at nominal values.

Table 2 Assumed Parameters: Nominal Values, Ranges, and Sources

Parameter	Nominal	Range	Basis
Artificial friction coefficient f	0.020	0.016–0.030	DIN (2011); measured 0.012–0.035 (Gładysiewicz et al., 2017)
Secondary-resistance coefficient C ($A \approx 500$ m)	1.20	1.15–1.25	ISO (1989); DIN (2011)
Belt mass q_B (kg/m)	17.4	15–19	12 mm EP belt, ≈ 14.5 kg/m ² (ISO, 2013)
Carrying idler rotating mass q_{RO} (kg/m)	16.7	13–20	≈ 20 kg per set at $a_0 = 1.2$ m (Rulmeca Group, 2014)
Return idler rotating mass q_{RU} (kg/m)	5.0	4–6	≈ 15 kg per roll at $a_U = 3.0$ m
Special resistances F_s (kN)	1.0	0.5–2.0	Two scrapers and skirtboards (DIN, 2011)
Belt–pulley friction μ (lagged pulley)	0.35	0.25–0.40	Dry to wet lagging (DIN, 2011)
Wrap angle α	200°	190° – 210°	Snub pulley arrangement
Start-up factor ξ	1.4	–	DIN (2011): 1.3–1.7
Idler spacing a_0 / a_U (m)	1.2 / 3.0	–	Typical for $B = 1,200$ mm (CEMA, 2014)
Admissible sag $(h/a)_{adm}$ carrying / return	1% / 2%	–	Conservative value (CEMA, 2014; DIN, 2011)
Drive efficiency η	0.90	–	Gear unit and couplings
Idler roll and bearing	$\varnothing 133$ mm, 6205	–	$C_r = 14.8$ kN (SKF, 2018); $F_s = F_m = 1.1$

Note. Parameters with a Range were Sampled in the Monte Carlo Simulation; the Others were Held Constant. None of These Values was Measured on CV-301.

➤ Failure Modes, Effects, and Criticality Analysis

Following IEC 60812 (IEC, 2018), the conveyor was decomposed into functional sub-assemblies (belt, splices, idlers, pulleys, drive, take-up, cleaning devices). For each failure mode, causes, effects, and detection means were identified from plant records and interviews, and severity G , occurrence O , and non-detection D were each scored on a four-level scale. The criticality index is:

$$I_C = G \cdot O \cdot D \quad (20)$$

Modes with $I_C \geq 18$ were classed as priorities; this threshold corresponds to an important failure ($G = 3$) that

recurs ($O = 3$) with limited warning ($D = 2$). Because the scores rely on expert judgement, robustness was assessed by recomputing I_C when any single index is shifted by ± 1 level.

III. RESULTS

➤ Geometric Compliance

The installed pulleys, rolls, and belt width match manufacturer recommendations for a 1,200 mm EP 1000 belt with four plies (Table 3). The minimum transition distance for a 20° trough with a textile belt is $1.0 B = 1.2$ m; it could not be measured and remains to be checked on site.

Table 3 Installed Versus Recommended Dimensions

Component	Installed	Recommended	Assessment
Belt width	1,200 mm	Standard width	Compliant
Drive / tail / snub pulley diameter	800 / 630 / 500 mm	800 / 630 / 500 mm	Compliant
Pulley face width	1,400 mm	1,400 mm	Compliant
Carrying / return roll length	465 / 1,400 mm	465 / 1,400 mm	Compliant
Transition distance ($\lambda = 20^\circ$)	Not measured	≥ 1.2 m	To verify

Note. Recommended Values for Textile-Carcass Belts from Rulmeca Group (2014) and DIN (2011).

➤ Conveying Capacity

With $b = 1.03$ m, Equations 3 and 4 give $S = 0.1149$, 0.1307 , and 0.1477 m² for $\theta = 15^\circ$, 20° , and 25° . The catalogue value $I_{VT} = 498$ m³/h at 1 m/s ($S = 0.1383$ m²) corresponds to an equivalent surcharge angle of 22.3° . At the adopted speed of 1.3 m/s, the mass capacity is therefore 710 t/h ($\theta = 15^\circ$), 807 t/h ($\theta = 20^\circ$), and 855 t/h (catalogue), giving fill degrees of 85%, 74%, and 70% for the required 600 t/h

(Figure 2b). The minimum speed delivering 600 t/h ranges from 0.91 m/s (catalogue) to 1.10 m/s ($\theta = 15^\circ$).

The gear ratio is $k = 40.47$, and the peripheral speed of the drive pulley is $v_t = \pi \times 0.80 \times 36.57/60 = 1.53$ m/s, 18% higher than the speed adopted by the plant. At v_t , the capacity would be 1,007 t/h (catalogue) and the fill degree 60%. The actual belt speed therefore needs to be measured; the results below are given for 1.3 m/s, with 1.53 m/s as a check.

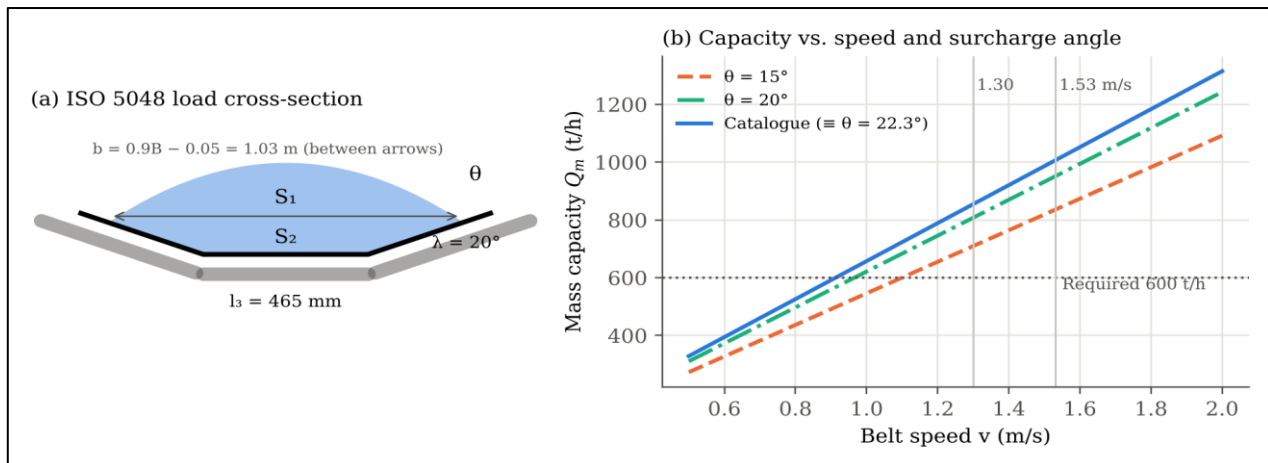


Fig 2 Load Cross-Section and Conveying Capacity of CV-301

Note. Panel b uses $\rho = 1.32$ t/m³. Vertical Lines Mark the Adopted Speed (1.30 m/s) and the Speed Imposed by the Drive (1.53 m/s).

➤ Resistances, Drive Reserve, and Power

Table 4 summarizes the nominal results. For a horizontal conveyor at 1.3 m/s, $q_G = 128.2$ kg/m, $F_H = 18.0$ kN, and the effective tension is $F_U = 22.6$ kN, which requires 29.4 kW at the drive pulley and 32.7 kW at the motor, i.e.,

38% of the installed 86 kW. The gear-unit output torque is $C_s = 20.89$ kN·m, so the drive can deliver up to $F_{U,max} = 52.2$ kN. The resulting reserve corresponds to an admissible net lift of 23.5 m at nominal friction, decreasing to 14.9 m if $f = 0.030$ (Figure 3).

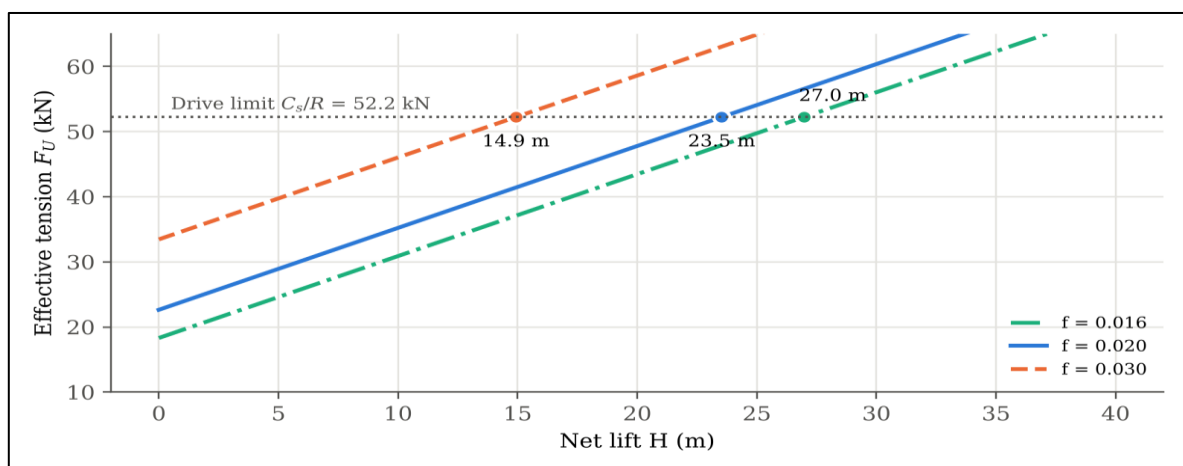


Fig 3 Effective Tension as a Function of Net Lift for Three Artificial Friction Coefficients

Note. Belt Speed 1.3 m/s, $Q_m = 600$ t/h. Markers Indicate the Admissible Lift at Which F_U Reaches the Drive Limit.

➤ Tension Distribution and Governing Condition

With $\mu = 0.35$ and $\alpha = 200^\circ$, $e^{\mu\alpha} = 3.39$. Preventing slip at start-up ($\xi = 1.4$) requires $T_2 \geq 13.2$ kN, whereas limiting the carrying-side sag to 1% requires a tail tension of 21.4 kN; after deducting the return-strand resistance $F_{ret} = 2.6$ kN, this gives $T_2 \geq 18.8$ kN. Sag therefore governs the take-up design for μ above about 0.28 (Figure 4b), and drive slip governs

only for wet or worn lagging. The resulting tension distribution is shown in Figure 5: $T_1 = 41.4$ kN in steady state and 50.5 kN at start-up. A take-up located near the drive would need a counterweight of about 3.8 t; the installed counterweight could not be weighed and should be compared with this value.

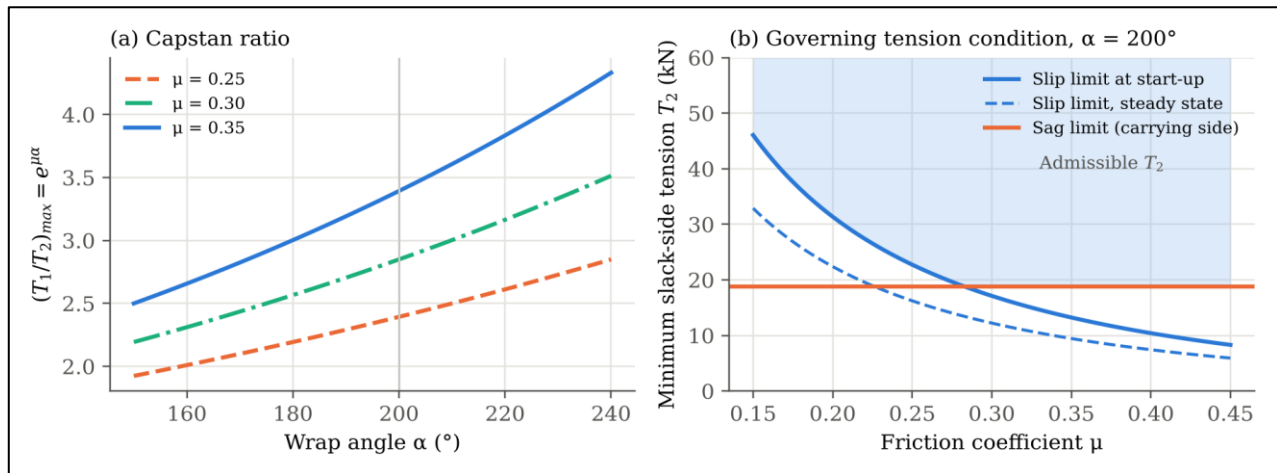


Fig 4 Traction Conditions at the Drive Pulley

Note. Panel b: Nominal $F_U = 22.6$ kN, $\xi = 1.4$. The Shaded Area is the Admissible Region for T_2 .

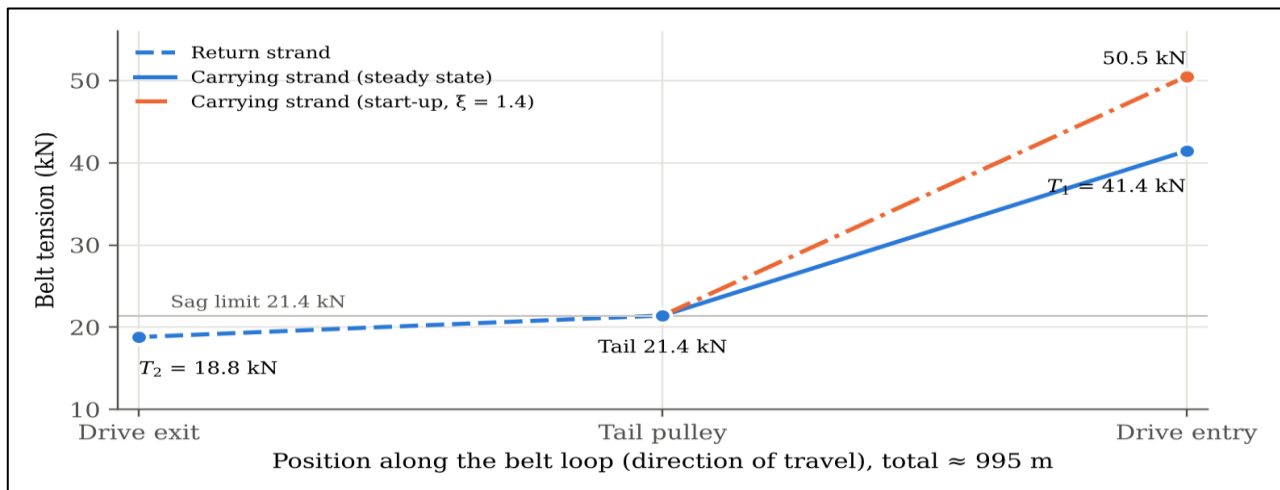


Fig 5 Belt Tension Distribution Along the Loop (Nominal Values, $v = 1.3$ m/s)

Note. Tensions Vary Linearly Along Each Strand Because the Conveyor is Assumed Horizontal with Uniform Resistance.

Table 4 Nominal Calculation Results for a Horizontal Conveyor

Quantity	$v = 1.30$ m/s	$v = 1.53$ m/s
Material load q_G (kg/m)	128.2	108.8
Main resistance F_H (kN)	18.0	16.1
Effective tension F_U (kN)	22.6	20.4
Motor power required P_M (kW)	32.7	34.7
Slack-side tension T_2 (kN), governing condition	18.8 (sag)	15.9 (sag)
Tight-side tension T_1 , steady / start-up (kN)	41.4 / 50.5	36.3 / 44.5
Belt safety factor, steady / start-up	29.0 / 23.8	33.1 / 27.0
Admissible net lift H_{adm} (m)	23.5	29.9
Counterweight near drive G_{TU} (t)	3.8	3.3

Note. Nominal Parameter Values of Table 2; $F_{U,max} = 52.2$ kN in Both Cases.

➤ Belt Strength

The belt strength is $k_N \cdot B = 1,200$ kN. Under nominal conditions the safety factor is 29.0 in steady state and 23.8 at start-up. As an upper-bound check, if the drive delivered its full 52.2 kN with $\mu = 0.35$, T_1 would reach 74.0 kN and $s = 16.2$; under this extreme loading, s would fall below 10 only if μ were lower than 0.16, a value outside the range expected for lagged pulleys.

➤ Idler Bearing Life

For the central carrying roll, Equation 18 gives $P_c = 1.43$ kN, i.e., 0.72 kN per bearing. At 1.3 m/s a $\varnothing 133$ mm roll turns at 187 rpm, and Equation 19 yields $L_{10h} \approx 7.8 \times 10^5$ h. Applying an additional impact factor of 1.5 under the loading zone still gives about 2.3×10^5 h. Under the assumed roll and bearing type, fatigue is therefore not a plausible cause of the frequent idler failures reported at GCK.

➤ Uncertainty Analysis

Table 5 and Figure 6 report the Monte Carlo results. Over the full parameter ranges, the effective tension spans 19.2–32.9 kN (P5–P95) and the required motor power 27.7–47.6 kW, always below the installed 86 kW. The start-up safety factor never falls below 14.1 (P5 = 16.7), and the

admissible lift never falls below 12.7 m (P5 = 15.3 m). The one-at-a-time analysis shows that f alone explains most of the variation in F_U (from 18.3 to 33.5 kN across its range), far more than C , the special resistances, or the masses (Figure 6a).

Table 5 Monte Carlo Results (N = 10,000, Horizontal Conveyor, $v = 1.3$ m/s)

Output	Min	P5	P50	P95
Effective tension F_U (kN)	17.0	19.2	25.9	32.9
Motor power required P_M (kW)	24.6	27.7	37.5	47.6
Slack-side tension T_2 (kN)	17.2	17.8	19.0	27.7
Safety factor, steady state	16.9	20.4	26.5	31.3
Safety factor, start-up	14.1	16.7	21.5	26.1
Admissible net lift H_{adm} (m)	12.7	15.3	20.9	26.3
Counterweight near drive G_{TU} (t)	3.5	3.6	3.9	5.6

Note. Uniform Distributions Over the Ranges of Table 2.

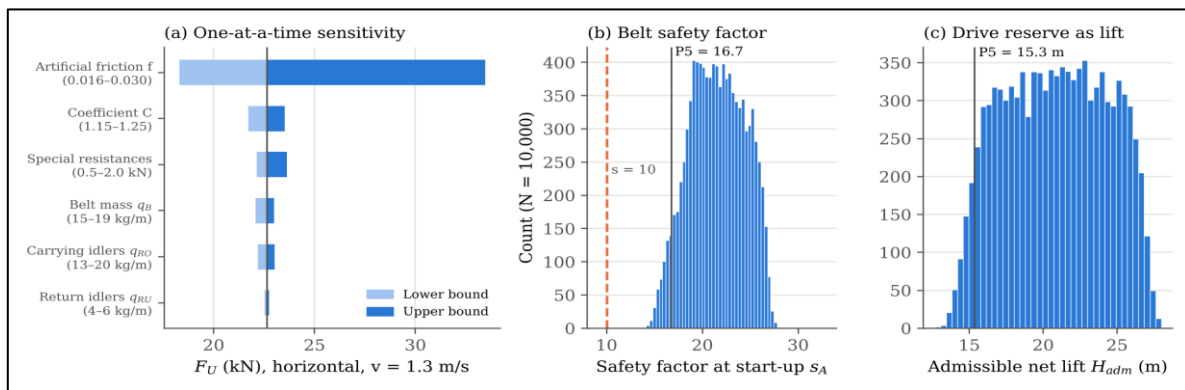


Fig 6 Sensitivity and Uncertainty Analysis

Note. Panel a: One-at-a-Time Variation of Each Parameter between its Bounds; Vertical Line = Nominal F_U . Panels b and c: Monte Carlo Distributions; the Dashed Line in Panel b is the Recommended Minimum $s = 10$.

➤ Criticality Analysis

Table 6 summarizes the FMECA. Three failure modes reach the highest criticality ($I_C = 24$): longitudinal belt tearing, splice separation, and idler seizure; belt cover wear reaches the threshold ($I_C = 18$). The robustness test (Figure 7) shows that the ordering is stable, because no lower-ranked mode can exceed 27 under a one-level change, but that threshold membership is sensitive: a one-level overestimate of any index would bring a top-ranked mode down to 12. The scores therefore rank priorities reliably but should be confirmed with plant downtime data before being used for budget allocation.

Table 6 Failure Modes, Effects, and Criticality Analysis of CV-301

Component	Failure mode	Main cause	Effect	Detection	G	O	D	I_C
Belt	Longitudinal tear	Lumps, tramp metal at chute	Long stoppage	Visual	4	3	2	24
Splice	Separation	Faulty vulcanizing, fatigue	Stoppage	Inspection	4	2	3	24
Idlers	Seizure	Dust ingress, seal failure	Belt wear, fire risk	Noise, heat	3	4	2	24
Belt	Cover wear	Abrasion, carry-back	Shorter life	Thickness	3	3	2	18
Gear unit	Wear, oil leak	Lubrication	Stoppage	Oil, vibration	4	2	2	16
Drive pulley	Lagging wear	Slip, abrasion	Loss of traction	Speed switch	3	2	2	12
Belt	Mistracking	Idler misalignment, off-centre loading	Spillage, edge wear	Switches	2	4	1	8
Scrapers	Blade wear	Abrasion	Carry-back	Visual	2	4	1	8
Motor	Overheating	Overload, fouled cooling	Trip	Thermal relay	3	2	1	6
Take-up	Jamming	Cable or sheave defect	Slip	Visual	3	1	2	6

Note. G = Severity; O = Occurrence; D = Non-Detection; $I_C = G \times O \times D$ (scale 1–64). Scores from Plant Records and Interviews.

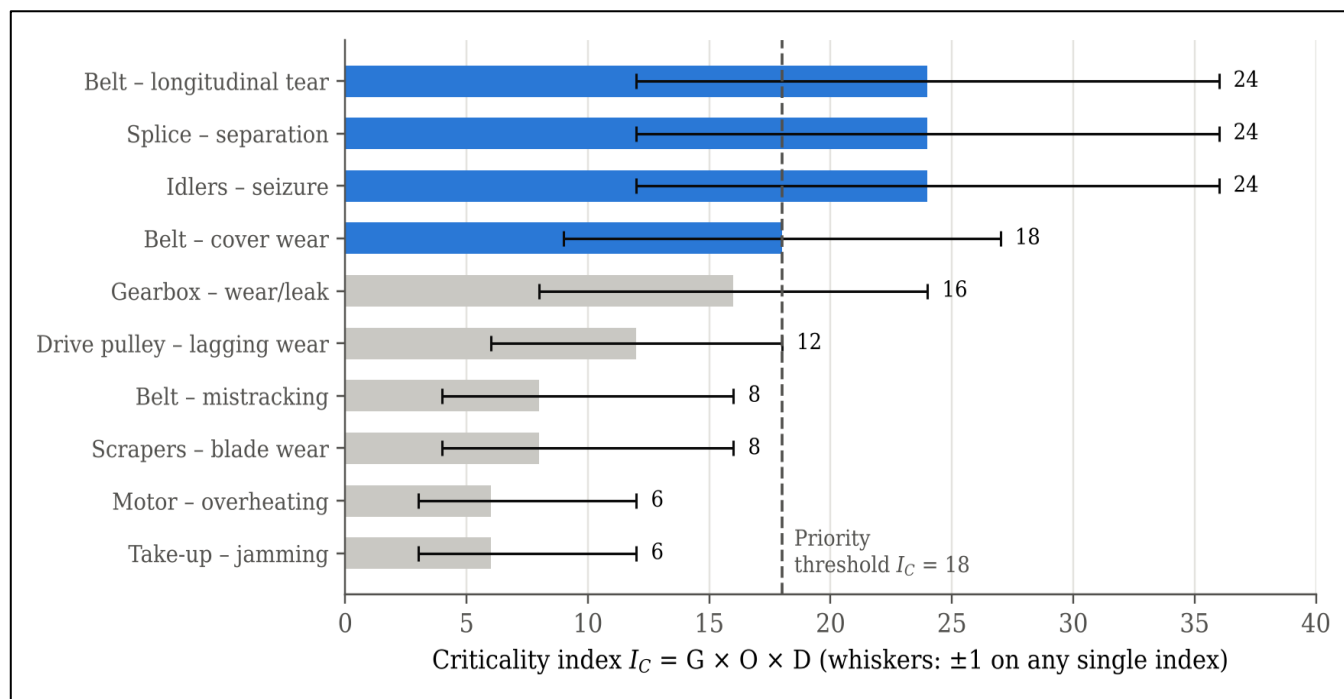


Fig 7 Criticality Ranking and Robustness of the FMECA Scores

Note. Whiskers Show the Range of I_C When Any Single Index is Shifted by ± 1 Level (Bounded to 1–4).

IV. DISCUSSION

➤ Adequacy of the Installation

The results converge on one conclusion: CV-301 is not undersized. Its capacity covers the required throughput even under the most conservative surcharge angle, the drive works at about 40% of its rating, and the belt retains a large strength margin under all simulated conditions. The preliminary verification based on catalogue tables (Nkulu Kabembo, 2025) is confirmed in direction but refined in magnitude: the catalogue capacity implies a surcharge angle of 22.3° , which may be optimistic for a coarse, abrasive crushed material, so that the actual capacity margin may be closer to 18% than 42% at 1.3 m/s.

Two quantities nevertheless call for field verification. First, the 18% gap between the adopted speed and the speed imposed by the drive can only be resolved by measurement; a higher actual speed would reduce the fill degree and spillage but increase impact energy at the loading point. Second, the admissible lift of 15–24 m is the main constraint on the drive; if the surveyed lift approached this value, the drive reserve would become small, especially with high resistance ($f = 0.030$) from poorly maintained idlers. The dominance of f in the sensitivity analysis agrees with field measurements showing that idler condition and belt loading strongly influence this coefficient (Gładysiewicz et al., 2017), and it means that idler maintenance directly protects the drive reserve. Conversely, at about 38% of its rating under nominal conditions, the motor operates in a region of reduced efficiency, which supports the speed-control strategies proposed for under-loaded conveyors (Zhang & Xia, 2011).

➤ Mechanisms Behind the Observed Failures

Because tensile overload, drive slip, and bearing fatigue can all be excluded as dominant mechanisms, the frequent belt tears observed at GCK are most consistent with mechanical damage at the loading point. Experiments on textile-carcass belts have shown that the energy of falling material, governed by drop height and lump mass, controls the onset of carcass damage (Fedorko et al., 2014; Andrejiová & Grinčová, 2016). This agrees with the first rank of longitudinal tearing in the FMECA and points to the transfer chute rather than the belt grade as the place for intervention. Likewise, the very long calculated bearing lives indicate that idler seizures stem from contamination and seal failure in a dusty environment rather than from load, which favours idlers with labyrinth seals and condition monitoring over heavier bearings.

These findings translate into the measures summarized in Table 7. They target the three top-ranked failure modes and the two parameters that most affect the safety margins: the resistance level, through idler condition, and the slack-side tension, through take-up integrity.

Table 7 Proposed Maintenance, Safety, and Monitoring Measures

Target	Measure	Justification
Longitudinal tear	Impact idlers and impact bed under the chute; grizzly for oversized lumps; tramp-metal magnet; rip detector interlocked with the drive	FMECA rank 1; impact damage mechanism (Fedorko et al., 2014)
Splice separation	Qualified hot vulcanizing procedure; splice inspection at each planned stop	FMECA rank 1
Idler seizure	Sealed idlers; monthly thermographic survey; replacement of seized rolls	Long fatigue life implies contamination-driven failure
Resistance level	Belt scale and motor current logging to track F_U	f dominates the uncertainty of F_U
Slip and take-up	Speed switch on the tail pulley compared with drive speed; counterweight travel limit switches	$T_2 \geq 18.8$ kN must be maintained
Personnel safety	Pull-cord stops on both sides (sections ≤ 50 m); fixed guards at nip points; lockout before intervention	European Committee for Standardization [CEN] (2010); ISO (2019)

Note. Intervals and Devices should be Adapted to Manufacturer Instructions and Plant Failure History.

V. LIMITATIONS

This study has four main limitations. First, the idler masses, friction coefficients, idler spacing, and roll type were not measured; their effect was bounded by the uncertainty analysis, but uniform distributions and independent sampling are simplifying assumptions. Second, the lift and actual belt speed of CV-301 are unknown, so results are given for a horizontal conveyor with an admissible-lift envelope. Third, the model is quasi-static and does not capture belt dynamics during start-up and stopping, for which DIN 22101 recommends dynamic analysis on long conveyors. Fourth, the FMECA scores are based on expert judgement because quantitative downtime records were not available in a form suitable for statistical analysis.

VI. CONCLUSIONS

An analytical verification of conveyor CV-301 according to ISO 5048 and DIN 22101, combined with a Monte Carlo uncertainty analysis and a robustness-tested FMECA, leads to the following conclusions:

- The conveyor is adequately sized: at 1.3 m/s, its capacity is 710–855 t/h for a required 600 t/h, depending on the surcharge angle.
- The nominal effective tension is 22.6 kN, about 43% of the drive limit of 52.2 kN; the drive can sustain a net lift of at least 12.7 m in all simulated cases.
- The take-up design is governed by belt sag rather than slip; a slack-side tension of about 18.8 kN, or a counterweight of about 3.8 t near the drive, is required.
- The EP 1000 belt keeps a start-up safety factor of at least 14.1; belt tearing is therefore attributable to impact damage at loading rather than tensile overload.
- Longitudinal tearing, splice separation, and idler seizure are the priority failure modes; their ranking is robust, but their absolute scores require validation with downtime data.

Future work should measure the belt speed, lift, idler masses, and motor current on site to replace the assumed parameters, record downtime by failure mode to calibrate the

FMECA, and evaluate the transient behaviour of the conveyor during start-up and stopping.

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➤ Data Availability

The installation data are reported in Table 1 and in Nkulu Kabembo (2025). The calculation script and simulation outputs are available from the corresponding author on reasonable request.

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[To be completed by the authors: conceptualization, methodology, investigation, formal analysis, writing – original draft, writing – review and editing, supervision.]

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APPENDIX

Appendix: Nomenclature

Symbol	Definition (unit)	Symbol	Definition (unit)
A	Centre distance (m)	k_N	Nominal belt strength (N/mm)
a_0, a_U	Carrying and return idler spacing (m)	L_b	Endless belt length (m)
B, b	Belt width and usable width (m)	Q_m, Q_v	Mass (t/h) and volumetric (m ³ /h) flow
C	Secondary-resistance coefficient (–)	q_B, q_G, q_{RO}, q_{RU}	Masses per metre of belt, material, carrying and return idlers (kg/m)
C_r	Bearing dynamic load rating (N)	S, S_1, S_2	Load cross-section and its parts (m ²)
I_C	FMECA criticality index (–)	s	Belt safety factor (–)
C_s	Gear-unit output torque (N·m)	T_1, T_2	Tight-side and slack-side tensions (N)
D, d	Drive and tail pulley diameter (m)	v, v_t	Adopted and drive-imposed belt speed (m/s)
f	Artificial friction coefficient (–)	α	Wrap angle (rad)
F_H, F_S, F_{St}	Main, special, and slope resistances (N)	θ, λ	Surcharge and trough angles (°)
$F_U, F_{U,max}$	Effective tension and drive limit (N)	μ	Belt–pulley friction coefficient (–)
G, O, D	Severity, occurrence, and non-detection indices (–)	ξ	Start-up factor (–)
H, H_{adm}	Net lift and admissible lift (m)	ρ	Bulk density (t/m ³)
Symbol	Definition (unit)	Symbol	Definition (unit)
A	Centre distance (m)	k_N	Nominal belt strength (N/mm)