

Article

Belt Rotation in Pipe Conveyors: Failure Mode Analysis and Overlap Stability Assessment

Leonardo S. Santos ^{1,*} , Emanuel N. Macêdo ¹, Paulo R. C. F. Ribeiro Filho ², Adilto P. A. Cunha ² 
and Noé Cheung ³ 

¹ Institute of Technology (ITEC), Federal University of Pará (UFPA), Belém 66075-110, Brazil; enegrao@ufpa.br

² Department of Mechanical Engineering, State University of Maranhão (UEMA), São Luís 65055-310, Brazil; paulofilho@professor.uema.br (P.R.C.F.R.F.); adiltocunha@professor.uema.br (A.P.A.C.)

³ Department of Manufacturing and Materials Engineering, State University of Campinas (UNICAMP), Campinas 13083-860, Brazil; cheung@fem.unicamp.br

* Correspondence: leonardo.dossantos@kaust.edu.sa

Abstract: Pipe conveyors provide sustainable solutions for environmentally sensitive or topographically complex powdered and bulk-solid handling processes; however, belt rotation is among the most critical failure modes of these equipment, influencing engineering, operational, and maintenance activities throughout the conveyors' lifecycles. Position changes in the overlap are mechanical responses to uneven contact forces between the vulcanizing rubber belt and the idler rolls, owing to the highly nonlinear process of the belt folding from a trough to a tubular shape, and no method for quantifying the belt's stability is currently available. In this study, we analyzed the failure mode of belt rotation and proposed a linearized model of an overlap stability index to evaluate the resilience of the overlap position through a case study of a short-flight curved pipe conveyor. Our proposal considers an interference model between the simulated torque of a curved flight in a pipe conveyor and the calculated torque of its equivalent straight flight by using kernel-smoothed density functions. It is adapted to incorporate adjustment factors for the filling degree based on simulations, the effect of the overlap in the forming force of the belt, the remaining useful life of the belt, and the coefficients of friction between the belt back cover and the idler rolls due to adhesion and hysteresis. An application was developed to calculate the belt's rotational holding torque and rotary moment by processing real operational data, simulated contact forces, and the relevant equipment parameters. This analysis identified the reduced transverse bending stiffness and increased belt tension forces as the root causes for position changes with a loss of contact in the upper idler rolls of curved flights 10, 13, 15–16, and 17. The contributing factors included spots of augmented contact forces during the initial stages of the belt lifespan in curved flights 15–16, which presented unstable conditions due to increased opening forces, with an OSI of 0.8657. Furthermore, we proposed corrective and preventive action plans, an optimized replacement interval for the belt, and recommendations for design changes according to the relevant standards.

Keywords: belt rotation; rotational holding torque; rotary moment; overlap stability index



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1. Introduction

The adoption of enclosed conveyor belt technologies for handling bulk solids plays an important role in ensuring sustainable consumption and production patterns, aligned with the Sustainable Development Goal 12 (SDG 12). Overall, conveyor belts generate a significant carbon footprint, demanding the utilization of synthetic and natural rubber, non-biodegradable reinforcements derived from crude oil, different chemical components, high manufacturing energy consumption, and long logistics chains, as well as generating worn-belt waste. In this sense, extending the belts' lifecycles and adopting sustainable design, engineering, operational, and maintenance criteria significantly reduces the required

resources and maximizes the value realization in bulk-solid handling processes. Quantifying operational stability in handling operations is challenging and demands a detailed analysis of critical failure modes, in addition to a whole-lifecycle management approach.

Pipe conveyors are the most prevalent among the enclosed conveyor belt technologies, which include pipe (or tubular), fold, suspended, and sandwich conveyor belts. They can negotiate tight horizontal and vertical curves, providing sustainable solutions for environmentally sensitive or topographically complex handling processes for powdered and bulk solids [1]. A special belt, sealed by the overlapping of its edges, surrounds the carried material, and the overlap needs to stay at the top and bottom positions of the carry and return flights, respectively, to ensure the proper operational stability of the pipe conveyor. This is the most challenging aspect of the process, which influences the equipment's overall performance and affects the reliability across its entire lifecycle when the belt tends to twist during curved flights [2].

The ability to form a stable tubular shape is governed by the transverse bending stiffness: the higher the stiffness, the better it counteracts the undesirable twisting moment of the belt. However, this is achieved at the cost of higher energy losses and increased rolling resistance, including the unwanted opening of the belt edges between the idler panels [3–5]. Furthermore, the cranking and overhang of the pipe belt caused by a higher transverse bending stiffness may lead to increased belt tension [6], and locally concentrated loads may trigger uncontrollable rotation. The twisting is triggered by several factors that have been identified empirically. The variation in the overlap position may be substantial, causing undesirable effects with potentially catastrophic consequences.

Changes in the overlap position are mechanical responses generated by the contact forces between the belt and idler rolls and the cross-sectional geometry of the enclosed belt, owing to the highly nonlinear process of the belt folding from a trough to a tubular shape. The intensity of the contact forces depends on various factors [5], and their mutual interactions cause undesirable belt rotation. Thus, investigating these causes and contributing factors is of fundamental significance [4]. According to [5], the rotation of the belt needs to be limited to the maximum possible extent and corrected for agreed limits; however, the phenomenon has not yet been completely explained, and practical solutions for belt tracking are not always effective [7].

Several authors have measured the contact forces in pipe belt conveyors. The authors of [2] developed a six-point testing device for pipe belt stiffness to investigate the effect of the pipe's diameter and the position of the overlap on the contact forces and compared the differences between steel cords and fabric belts. A similar six-point measurement approach was used in [8–10]. The authors of [11] placed measuring sensors on the hexagonal idler housing of a test rig to develop a multiple linear regression model that considered the tension force and material filling volume. Meanwhile, [12] used a test rig to study the effects of the pipe diameter and filling rate as key control factors on the contact force based on a sensitivity analysis. The authors of [9] evaluated the influence of the belt construction on the bending stiffness and associated forces of a pipe belt using a test rig, while [13] explored the relationship between the normal force and sag resistance using a six-point measuring device. The authors of [5,6,14–21] studied the contact force values and their dependence on the tensional force using a special test device composed of a transition region from a flat to a pipe shape. Elsewhere, [22] analyzed the contact force of a motionless pipe conveyor belt under loaded and unloaded conditions by changing its tension force. The authors of [23] used a test rig to study the form–force behavior of different pipe belt constructions, and [24] analyzed the asymmetrical effect of tension forces on the idler housing contact forces. Another study [25] evaluated the contact forces on the rollers of a large-diameter pipe conveyor, while [26] developed a measuring idler to measure the indentation rolling resistance and normal force in a pipe-belt conveyor. Elsewhere, [27] used a hexagonal wooden frame to measure the radial loads from a pipe belt.

The experimental approach for measuring the contact forces was used to validate the analytical and numerical models [7], wherein a new analytical approach was proposed

for various loading conditions by combining finite element (FE) analysis to explore the conveyor characteristics that cause contact losses between the belt and idler rolls. The current coupled models constitute a valuable tool to support the proper balance of the required belt troughability and the estimation of energy consumption; however, belt rotation remains one of the primary engineering, operation, and maintenance concerns throughout the lifecycle of a pipe conveyor belt [28,29], and no method exists to date for quantifying belt stability [30].

In this study, we analyzed the relevant engineering, operational, and maintenance issues to prevent and deal with the failure mode of belt rotation over the different stages of a pipe conveyor's lifecycle. We researched key publications in the largest databases of abstracts and citations of peer-reviewed literature on this topic and the existing standards used for the design of pipe belt conveyors. Furthermore, we proposed a linearized model to evaluate the resilience of the overlap position of a belt through a case study on a short-flight curved pipe conveyor utilized in a bituminous-coal-handling system. A software application was developed to implement the proposed method by processing real operation data, simulated contact forces using the Belt Analyst specialist software version 20.1.3.0, relevant equipment parameters, and the remaining useful life of the pipe belt. Subsequently, a field inspection was conducted on critical flights, allowing for the identification of root causes, an analysis of contributing factors, and the definition of corrective and preventive action plans.

2. Failure Mode Analysis

Pipe belt rotation is a failure mode characterized by a variation in the belt overlap position in its longitudinal axis from the target position [28]. This may affect either the carry or return flights, for which the target is expected to maintain the overlap at the top and bottom positions. The top position of the belt overlap is often referred to as the "0°" or "12 o'clock" position in field applications, and its variation usually relates to the clockwise running direction. Figure 1 shows a typical reference criterion for assessing the belt overlap position. Figure 1a illustrates the expected position of the overlap in the carry (upper position or "0°") and return (bottom position or "180°") flights of one pipe conveyor. During regular operations, position changes of the overlap may be assessed by considering the reference in the numbering of idler rolls (overlap in position 4 of the carry flight, see Figure 1(b1)), the reference in degrees (overlap in the "0°" of the carry flight, see Figure 1(b2)), or the reference in hours (overlap in the "12 o'clock" of the carry flight, see Figure 1(b3)).

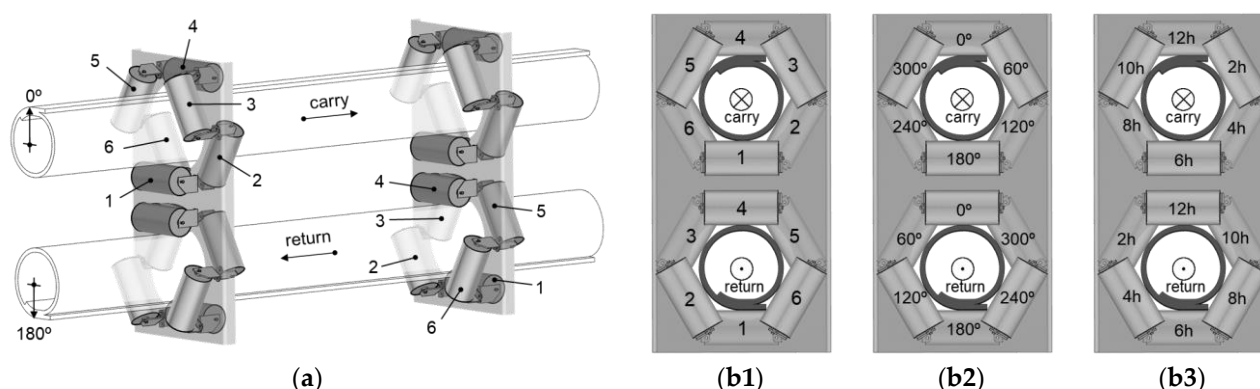


Figure 1. (a) Overlap position in pipe conveyors with the bottom position of idler panels identified as the first roll, followed by anticlockwise counting based on the belt running direction—positions 1 to 6; (b1) reference in the numbering of idler rolls considering positions 1 to 6; (b2) reference in degrees; (b3) reference in hours.

The belt rotation may be substantial, with some practical experiences of more than 360° turns being recorded. This phenomenon is one of the most critical failure modes of pipe conveyors, for which belt monitoring and training are expected to occur throughout the equipment's lifecycle. Figure 2 illustrates examples of belt rotation.

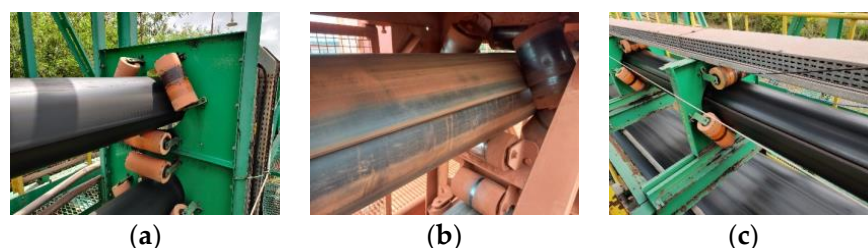


Figure 2. Examples of variation in the overlap position. (a) Fabric-reinforced belt at 90° ; (b) aramid-reinforced belt at 60° ; (c) steel-cord-reinforced belt at 60° .

Owing to the contact forces acting on the belt, particularly the lateral forces in the route curves, the pipe belt tends to rotate [5]. The undesirable rotation may not be eliminated after passing the curved flights [2], causing a misalignment of the belt edges in the transition zone, in which the belt is opened and guided to the discharge pulley. This condition may lead to persistent downstream misalignment in flat flights or the collapse of the tubular shape, with catastrophic consequences for handling operations, as shown in Figure 3.

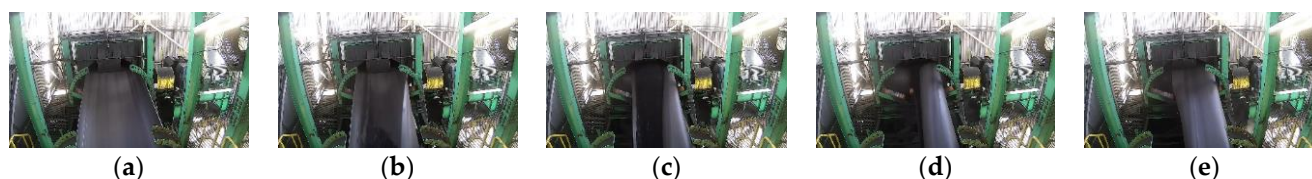


Figure 3. Misaligning then twisting of a fabric-reinforced belt in a discharge pulley after a 6 s interval. (a) Initial condition; (b) folding; (c) collapsed; (d) twisted; (e) inverted.

Figure 3 illustrates one example of the misaligning and then twisting of a fabric-reinforced belt in a discharge pulley. The belt was running with a left-rotated overlap position. Initially (see Figure 3a), it was slightly misaligned to the right side; then, it folded (see Figure 3b) and collapsed in on itself (see Figure 3c). As the operation continued, it twisted to the center (see Figure 3d) and opened in its inverted position (see Figure 3e) just six seconds after the triggering of the abnormal event.

Changes in the overlap position may lead to secondary effects, as shown in Figure 4. These include:

- Bulk spillage or powder emission: Belt rotation may affect the overlap sealing efficiency, causing spillage of the material [3] or dust emissions in the twisted regions.
- Damage to the belt edges: Belt rotation may damage the external side of the overlap edge because of the pinching of the belt between the idler rolls (in the case of the in-line panel arrangement) or the friction of the belt with the housing of the idler rolls (in the case of the offset arrangement).
- Fire on the pipe belt: Premature failures in idler rolls caused by pinching or friction between the belt and the idler housing may trigger conveyor fires that easily propagate within the facility owing to the belt's movement [31].
- Wear on the belt edges: Belt rotation may increase the stress concentration of the powder or bulk material and the friction rubber between the overlapping edges, leading to premature wear and affecting its sealing capability.
- Rainwater ingress: Carry flights are susceptible to the contamination of powder or bulk material by rainwater in the case of belt rotation, which is particularly relevant for equipment that is exposed to long shutdowns with the belt under rotated conditions.

- Propagation of cracks in the belt surface: Dynamic strain in the radial direction resulting from belt rotation favors crack propagation on the back cover surface of the belt, leading to accelerated aging owing to oxidative attack [32].

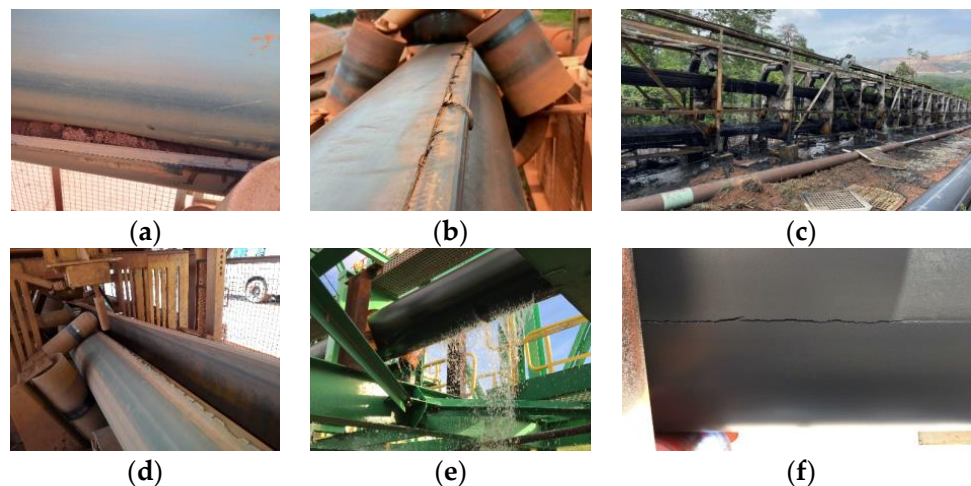


Figure 4. Secondary effects of belt rotation in pipe conveyors: (a) spillage of bulk material; (b) damage to the belt edges; (c) fire on the pipe belt; (d) wear on the inner belt edges; (e) rainwater ingress; (f) propagation of cracks.

2.1. Causes and Contributing Factors

Engineering and design considerations are crucial in preventing excessive or uncontrollable belt rotations in pipe conveyors. The proper design and specifications of the pipe conveyor belt ensure smooth commissioning and a stable operational phase. Contact forces affect the stabilizing rotary moment of the pipe belt, counteracting its twisting moment [2]. The contact forces decrease over time because of the increase in the belt's troughability. The repetitive flexural movement of the belt causes this expected behavior over its lifecycle [33], demanding lifecycle management of the equipment due to operational and maintenance-related concerns. The improper translation of requirements into specifications at early conceptual stages may generate unintentional obstacles throughout the lifecycle of the conveyor, which also include poor or omissive operational and maintenance routines after the project's startup. The following non-exhaustive list details several causes and contributing factors that may lead to uncontrollable belt rotation in pipe conveyors.

2.2. Strong Winds

Strong winds may cause belt rotation in pipe conveyors [1,34] due to sudden changes in the belt lateral forces, ultimately modifying the contact forces with the idler rolls. Designers and engineers must consider this effect such that proper training of the belt is not affected under both loaded and unloaded conditions; alternatively, weather protection should be considered [1]. In this scenario, the structure may only be side-guarded, considering the mapped areas of excessive wind speed for capital cost optimization.

2.3. Heavy Rain

Heavy rain may lead to severe belt rotation in pipe conveyors [1,34] because of sudden changes in the friction factor between the outer carcass of the belt and the idler rolls' shells. This effect can be easily identified by a decrease in the driving electrical current during intense rainfall in practical applications, followed by the pipe rotation in the route curves. For operational scenarios in which handling operations are expected to continue under adverse weather conditions, equipment guarding should be considered [1]. If heavy rain is the variable of concern, even if guarding is provided, we recommend applying test methods according to the "ISO 1817:2022 Rubber, vulcanized, or thermoplastic determination of the

effect of liquids [35] on belt samples”, considering the potential aging effects caused by liquid exposure at early conceptual stages.

2.4. Sub-Zero Temperatures

The idler rotational resistance and transverse bending stiffness of the belt increase during cold-weather operations, leading to changes in the contact forces and an increase in the indentation rolling resistance factor with a decrease in temperature [33,36]. In the case of prolonged downtime during periods of extremely cold weather, a hard start may be limited or even damage the pipe belt. Designers and engineers should consider using suitable plasticizers and low-glass-transition polymers in belt design, as well as selecting a proper cold-weather grease for the bearings used in idler rolls [1,37]. If prolonged downtime during extremely cold weather is expected, a creep driver (inspection speed) is recommended to prevent twisting and damage during a hard start. Guidance on methods of testing for the low-temperature properties of vulcanized rubbers is available in ISO 18766:2014 Rubber, vulcanized or thermoplastic—Low-temperature testing—General introduction and guide [38].

2.5. Sunlight, Ozone, Oxygen, and Heat

Under tension, rubber compounds tend to dry out, generating small cracks when exposed to intense sunlight, ozone, and heat for prolonged periods [39,40]. High ambient temperatures activate and accelerate oxidation in rubber compounds, decreasing the crack nucleation life of the belt and increasing the growth rate of fatigue cracks, independent of the natural aging of the rubber or its continuous vulcanizing processes [41]. Rubber compounds used in the back covers of pipe belts are highly dependent on the temperature and loading frequency [33,36], and an increase in the temperature reduces the indentation rolling resistance of the belt, thereby reducing its contact forces and favoring belt rotation.

Exposure to ozone increases the crack growth rate, causing scission in the polymer chain by reacting with carbon–carbon double bonds [41]. The aging process of the belt is initiated on the surface of its back cover and expands from the middle to both edges [42]. Because of belt cycling, the oxidized thin film on the back cover’s surface breaks and exposes the fresh rubber area, propagating the oxidative reaction [42]. Exposure to oxygen decreases the threshold of mechanical fatigue crack growth, and the dissolved oxygen in rubber changes its elastomer structure over time, causing embrittlement and irreversibly shortening its fatigue life [41]. The photochemical reaction caused by ultraviolet light from sunlight promotes the oxidation of the back cover surface of the pipe belt, triggering fatigue crack growth [43,44].

If this is a variable of concern, we recommend applying test methods according to ISO 703:2017 Conveyor belts—transverse flexibility (troughability) [45] to belt samples, considering peak temperatures, to estimate the contact force variations during the conceptual stages. We also recommend applying the test methods according to ISO 188:2023 Rubber, vulcanized, or thermoplastic—Accelerated aging and heat resistance tests [46]; ISO 1431-1:2022 Rubber, vulcanized or thermoplastic—Resistance to ozone cracking—Part 1: Static and dynamic strain testing [32]; ISO 6943:2017 Rubber, vulcanized—Determination of tension fatigue [47]; and ISO 27727:2008 Rubber, vulcanized—Measurement of fatigue crack growth rate [48]. For such operational scenarios, sunlight guarding or a special belt back cover should be considered in the conceptual stages. Figure 5 illustrates examples of crack propagation in a pipe conveyor.

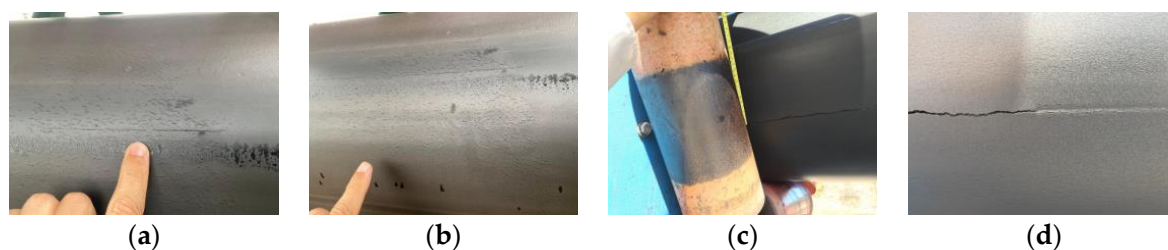


Figure 5. Propagation of cracks in pipe conveyor belts. (a,b) An oxidized thin film on the back cover's surface; (c,d) crack growth and nucleation.

2.6. Prolonged Shutdown

Different operational factors may demand the prolonged shutdown of a pipe conveyor belt during its lifecycle as part of the system requirements. This may be caused by a shutdown–turnaround–outage, a seasonal shutdown, regulations or local laws, a decrease in market demand, or reasons related to idle capacity.

Under such conditions, the belt is exposed to weathering and natural aging without experiencing opening cycles. This may reduce the opening force of the belt on the enclosed section and the troughability of the belt placed on the transition flights. In addition, the rotational resistance of idlers increases owing to weathering and long-term storage [31], for which additional time may be required to stabilize the friction loss at a constant value after a prolonged shutdown [37].

This may lead to belt rotation due to two different reasons: (i) the reduced opening force of the belt on its enclosed portion that is exposed to a prolonged shutdown, or (ii) an uneven opening force on the belt portion that remained troughed during this period. Preventive measures may include the regular empty operation of the belt for position alternation and the use of a creep driver (inspection speed) at startup.

The energy cost of periodic empty runs should be considered as part of the operation and maintenance costs early in the conceptual phases. Good practice may include marking the back cover of the belt in white ink at specific points to guide operators and maintainers in alternating the belt position to ensure that the section positioned in the carry flight is placed at the return side during the next phase, as in the case of flat and enclosed sections.

2.7. Hard-Flowing Powdered or Bulk Materials

Material flowability is one of the most important parameters to consider when designing a powdered or bulk solid handling system. The stresses induced in the material vary in all directions [49]. Its multidimensional characteristics depend on a combination of physical properties, environmental conditions, and equipment selection [50].

The moisture content is the main variable of concern, and its increase reduces the material's flowability, forming liquid bridges in narrow gaps between particles, as well as increasing the cohesive and adhesion strengths of the bulk solid [50–53]. If the first saturated region appears, the flowability decreases further but less strongly. However, if the saturation threshold is reached, the liquid surface tension disappears, and the moist bulk solid transforms into a suspension [54], as shown in Figure 6. This is particularly sensitive to processes with high variability and may affect the load profile in the pipe belt.

Figure 6a illustrates the flowability of solids in pipe conveyors as a function of the moisture content into three regions: forming liquid bridges in narrow gaps (region a, see Figure 6(b1)), the first saturated region (region b, see Figure 6(b2)), and finally the saturation threshold, where the moist bulk transforms into a suspension (region c, see Figure 6(b3)).

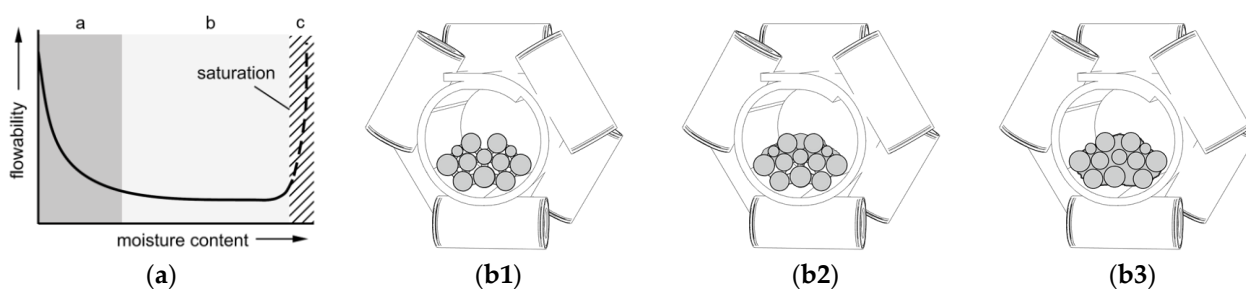


Figure 6. (a) Solids' flowability in pipe conveyors as a function of moisture content: region a—liquid bridges, region b—liquid bridges and saturated material, and region c—saturated material. (b1) Cross section details of a pipe conveyor belt with (b1) liquid bridges; (b2) liquid bridges and saturated material; (b3) saturated material.

The material load profile of pipe conveyors is influenced by several factors, including route curves, idler support spacing, belt inclination, belt sag and tension, belt speed, and equipment vibrations along the route [1]. Saturated bulk solids may modify the rotary moment of the belt at tight horizontal curves, thereby favoring belt twisting. Vertical curves or slopes may cause a spot with a high filling degree, increasing the localized pressure and generating a twisting moment owing to the uneven contact forces distributed longitudinally, as exemplified in Figure 7.

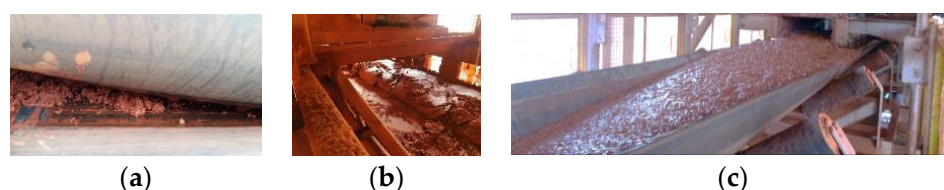


Figure 7. Uneven contact forces caused by material saturation. (a) Moist bulk on overlapping edges; (b) overfilling moist bulk; (c) saturated bulk in loading chute.

Flowability quantification should be considered for powder or bulk materials, including the evaluation of the particle size and distribution, the angle of repose, the bulk density, the angle of internal friction, cohesion, adhesion, and compressibility [50,54]. Preventive measures may include the adoption of feeding in mass flow and short residence times upstream of the pipe conveyor, minimizing impact forces and transfer speeds at the loading station, and decreasing the filling degree combined with reduced belt speed. This will necessitate the adoption of larger diameters, for which attention to troughability is required.

2.8. Dewatered Materials

Dewatered materials are special types of hard-flowing bulk solids. The remaining liquid capillary pressure causes a larger tensile strength and less favorable flow properties. When enclosed by a pipe belt, compaction may saturate the moist bulk, causing the twisting described above at the conveyor curves [54].

To reduce the risk of operating close to the saturation threshold, the moisture content may be reduced and controlled in the upstream processes at the cost of increasing dust emissions at the transfer points. Therefore, we recommend determining the dust extinction moisture for the powdered or bulk solid to determine the optimal operating point in the early design stages [55]. In addition, the use of the discrete element method (DEM) and computational fluid dynamics (CFD) when designing the transfer chutes are proven approaches to controlling dust emissions in handling systems [56,57]. Secondary measures may include the confinement of transfer regions with skirting and strip rubber, as well as the use of fog nozzles.

2.9. Deteriorating Powder or Bulk Materials

The interaction between the powder or bulk material and the belt-top cover also plays an important role in the operational stability of a pipe conveyor. Materials wetted with or containing chemicals, hot, fine, or lumpy materials, fertilizers, vegetable and mineral oils, or animal fats may cause the accelerated aging of the pipe belt, with an increase in the top cover's hardness [1,58]. The long-term effect is a decrease in the belt's opening force, which leads to belt rotation and training issues. Designers and engineers should consider the proper selection of the belt top cover and regularly measure the top cover's hardness. A Shore A Hardness of 60 ± 5 is typical for most applications [1], for which 80 Shore A is a value of concern [59,60] and may trigger a plan for the replacement of the belt owing to natural aging. Proactive measures include regular belt training, decreasing the take-up tension force based on hardness measurements, and visual inspections of the belt-top cover to detect crack propagation.

2.10. Unstable Filling Degrees

The degree of filling of the pipe conveyor plays an important role in changing the rotary moment of the belt. This condition affects the carry side of the belt, and its empty operation is often problematic because of the movement of the center of gravity of the pipe, causing training issues [1]. The unstable feeding of the pipe belt may allow empty regions over its path, leading to longitudinally uneven contact forces and favoring belt rotation, as the geometry of the cross-section defines the position of the shear center [2]. Lastly, belt overfilling may also lead to uncontrollable belt rotation, with related issues including an increased belt tension force due to the concentrated contact forces with the conveyor idler rolls. It also includes the premature damage of idler bearings, the shearing of idler bolts, the shearing of idler bolts, splicing failures, belt tearing, the removal of idler supports, or even the ripping of the belt. Figure 8 illustrates examples of overfilling issues in pipe conveyor belts.

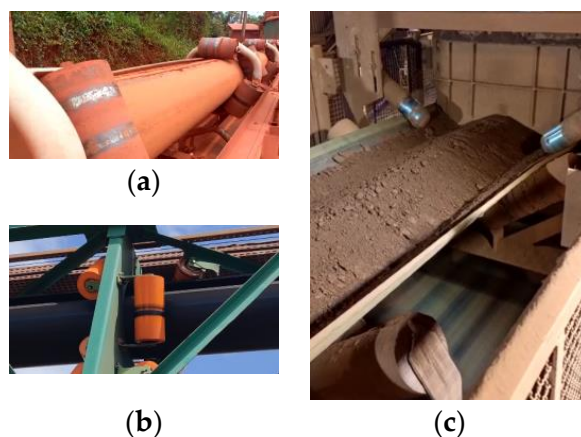


Figure 8. Overfilling in pipe conveyor belts. (a) Compressed bulk in the carry flight; (b) shearing of bolts in the upper position; (c) overload in the loading station.

In this sense, proper upstream feeder control, typically combining a dynamic weighting system and a variable-frequency drive (VFD), and the adoption of protective devices for detecting overfilling at the loading section need to be considered in the early design stages. In the case of extremely lumpy bulk solids or poor feed control, it is recommended that the maximum filling degree permitted be reduced, as shown in Table 1.

Table 1. Filling degree recommendations for pipe conveyor belts.

Recommendation	Criteria	Filling Degree
CEMA 7th [1]	No lumps and good feed control	80–85%
	Typical value	75%
	Very lumpy or feed control is poor	60% or less
JB/T 10380:2013 [61]	Lump size is less than 1/3 of nominal diameter	75%
	Lump size is less than 1/2 of nominal diameter	50–60%
	Lump size is less than 2/3 of nominal diameter	40–50%
VDI 4438:2019 [62]	Lump size is less than 1/3 of nominal diameter No lumps and good feed control	80%
	Lump size is less than 1/3 of nominal diameter Typical value	60–80%
	Lump size is less than 1/3 of nominal diameter Very lumpy or feed control is poor	30%

A failure-finding plan is recommended for testing the safety panels, tilt switches, and overfilling sensors at loading stations. The length of the region protected by the safety panels should allow for the complete stoppage of the belt after detecting overfilling, with the risk of failing to prevent the consequences mentioned above. Proactive measures also include the use of a creep driver for reconnection after an overfilling event and maintaining a constant material cross section [63].

2.11. Improper Routing

Routing is the most important structural parameter for achieving operational stability in pipe conveyor applications. An improper definition of the curvature radius may lead to recurrent twisting phenomena, causing the premature replacement of the pipe belt throughout the conveyor's whole lifecycle [1]. The route itself imposes stresses on the pipe belt, where the middle of a curve is the region of maximal elongation in the outer region of the belt, as well as the maximal compression in its inner region [23,61]. The stress distribution is noticeably increased in the overlap region through curved flights owing to the reduction in the cross-sectional geometry of the pipe belt [64]. This may result in one or more contact forces becoming equal to zero, thereby favoring material spillage and belt rotation [7,34,65]. Excessive elongation and compression in curved flights may also lead to localized overstressing and the buckling of the pipe belt, respectively, affecting its useful life.

A key factor in determining routing is the smallest possible curve radius for the horizontal and vertical curves, which depends on the structure of the conveyor, the pipe diameter, the deflection angle of the curve, the length of the curved segment, and the local tensile force [61]. The standardized CEMA recommendations for minimal curve radii consider the following designer variables: pipe belt reinforcement, the nominal pipe diameter, and the central angle (Table 2).

Proactive measures may include adopting more conservative horizontal curve radii (the larger the better), avoiding the unnecessary alternation of the horizontal direction, limiting the vertical inclination to 30°, and simulating contact forces in curved flights to ensure that no contact force is equal to zero, although this is a commonly observed phenomenon in practical applications [61]. It is also recommended that the take-up tension force be decreased based on periodic measurements of the belt diameter at predefined points along its flight during its lifecycle.

Table 2. Minimum curve radii recommendations for pipe conveyor belts [1].

Variable	Criteria	Coefficient	Value
Belt reinforcement	Nylon	p	0
	Polyester nylon		1
	Aramid		2
	Steel cord		4
Nominal pipe diameter (mm)	From 150 to 300	f	0
	From 300 to 500		1
Curve's central angle	<25 degrees	c	0
	Between 25 and 50 degrees		1
	Between 50 and 75 degrees		2
	Between 75 and 100 degrees		3
Minimum curve radii (mm)	$[300 + 100(p + f + c)]Nominal pipediameter(mm)$		

2.12. High Belt Speeds

High belt speeds may lead to belt rotation phenomena in pipe conveyors owing to the increased indentation rolling resistance and the material bunching effect. The former is caused by the viscoelastic behavior of the belt back cover on the idler roll surfaces, increasing the asymmetry of the contact zone at higher belt speeds [26,33], which may increase the rotary moment in the case of uneven contact forces; this is mainly seen in curved flights.

The latter is a resonant reaction between the transverse wave frequencies of the belt and the excitation frequencies of the idler rolls [66–68], creating zones of increased and reduced filling degrees along the belt carry side, thereby longitudinally changing the contact forces. In addition, at the critical belt speed, regimes may arise in which the amplitudes of the angular deviations become significant even without external influences. The worst case, in terms of the occurrence of significant angular vibrations, is the empty operation of the pipe conveyor under a minimum tension force [69].

The belt speed is approached differently depending on the standards or recommendations utilized. According to the seventh CEMA manual, the maximum belt speed is limited by the allowed rotational speed of the idler rolls, as well as the effect of the material dynamic behavior at transition stations [1]. The German standard VDI 4438:2019 (draft version) and the Chinese standard JB/T 10380:2013 recommend maximum belt speeds of 6.0 m/s and 7.1 m/s, respectively [61,62].

Proactive measures may include the adoption of a more conservative belt speed (below 5 m/s when practical) and evaluation of the transverse wave frequencies of the belt in the early design stages. We also recommend maintaining a safety margin for the idler roll's operating speed. Depending on the design criteria of the idlers, the maximum speed can range from 600 rpm (CEMA) [1] to 750 rpm (DIN) [70]. The safety margin may allow the end user to troubleshoot the material bunch-up by changing the belt speed (which may require increasing the driving power), thereby minimizing the unbalanced force of the idlers rolls (owing to their intrinsic eccentricity and quality flaws in the manufacturing process) or shifting the transverse wave frequencies of the belt from the resonant modes (for idler roll frequencies 1/2, 1, 2, 3 . . . times the belt frequencies).

2.13. Belt Sag

Belt sag plays an important role in the operational stability of a pipe conveyor. Using an experimental dynamic test rig, the authors of [13] identified a direct relationship between the sag and the contact forces in the straight flight of a pipe conveyor. Excessive sag may lead to material spillage due to the opening of the pipe belt edges between the panels. It may also cause overhang and cranking of the belt in the idler rolls, leading to uncontrollable belt

rotation due to the increased belt tension and concentrated contact forces in the conveyor panels [6].

Proactive measures may include the adoption of a more conservative panel spacing in straight flights, with a decreased space in curved ones; however, at increased indentation rolling resistance losses [26,33,71,72]. Therefore, the proper evaluation and balancing of the energy consumption is beneficial for operational stability. Typical spacing values range from 1.45 m to 2.36 m based on the specific weight and the nominal pipe diameter [1,73]; however, several factors may be considered, such as the pipe diameter, the conveyor routing, belt troughability, the take-up force, the number of curved flights, and their corresponding radius. Random or alternating spacing may also be used to minimize the resonant responses of the belt.

During the commissioning stage, an increase in the sag of the pipe belt is expected because of its accommodation period, which requires an initial number of cycles for stabilization. Based on this, we recommend obtaining the opening force of the belt as a function of the number of cycles from the belt manufacturer, allowing for a proper estimation of its break-in period. This may affect the initial belt training, and cautionary measures may include a temporary increase in the take-up tension force.

The commissioning of a pipe belt conveyor requires significantly more time to achieve the intended stable operation than a conventional troughed belt conveyor [61]. Additional drive power is typically required during this period, even when running an empty conveyor [64], and this condition is especially relevant for long or complex-routed pipe conveyors. Therefore, we advise that this spare capacity be anticipated, including creep-speed features. The unavailability of spare power and creep-speed capacity may delay the conveyor's commissioning plan and trigger a worst-case operational scenario soon after the project handover: failure to restart the loaded conveyor after a motor trip.

2.14. Excessive Belt Cycling

Cycling plays an essential role in the changes in the contact forces of a pipe belt over time. The belt folds from a troughed-to-tubular shape at least twice in each cycle in the molding rolls of the carry and return flights. Under normal circumstances, cyclic fatigue gradually decrease the transverse bending stiffness of the belt [74], which is triggered by the repetitive flexural movement of the belt over its lifecycle [33]. This condition mainly affects short-flight pipe conveyors and may result in the use of stiffer belts or a higher belt replacement rate, which negatively influences the total cost of ownership. In such cases, it is recommended that the take-up tension force be decreased based on periodic measurements of the belt diameter at predefined points along its flight during its lifecycle.

2.15. Improper Belt Reinforcement

The proper arrangement of belt reinforcements with a low tendency to twist is of fundamental importance to ensuring the stable operation of a pipe conveyor over its lifecycle [61]. Typically, the belt consists of different zones along its width, considering discrete variations in transverse rigidity, elongation, and line mass, for which the central part is often of an increased value [7,75]. This counteracts the rotary moment of the belt by keeping the center of gravity below its centerline, especially when operating under empty conditions.

The following techniques are commonly applied in pipe belt construction: (i) applying reinforcements or breakers to the horizontal layers of the belt width, (ii) applying an asymmetric distribution of belt carcass reinforcements, or (iii) using different material rigidities in the different zones of the belt width [7]. Belts used in pipe conveyors can be reinforced in nylon–nylon (P), polyester–nylon (EP), aramid (D), aramid–nylon (DP), and steel cord (St) [1], for which the selection is based on several factors, including the belt elongation, belt tension, pulley diameter, belt line mass, transition zone length, and conveyor routing, which are affected by the curve radius and allowable stresses in curved flights. Figure 9 shows different pipe belt reinforcements.

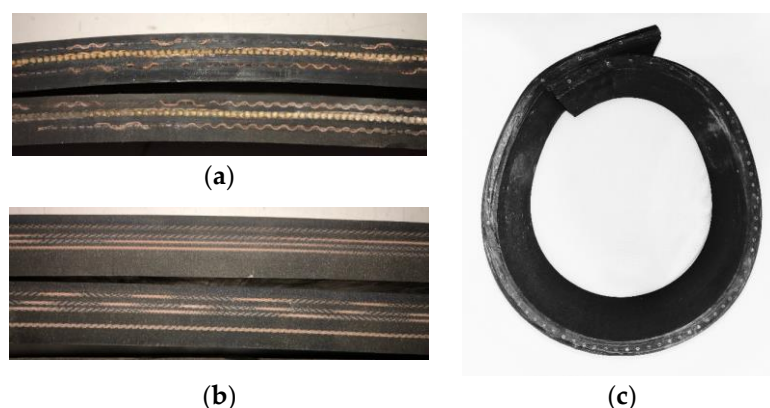


Figure 9. Examples of pipe belt reinforcements: (a) aramid; (b) ply; (c) steel cord.

Proactive measures include the coupled use of the finite element method (FEM) and DEM analyses to investigate the arrangement of the proposed reinforcements. The authors of [4] used a numerical approach and determined that the contact forces are highly sensitive in cases of empty operations, with several subtle factors affecting the force distribution, such as the friction of the belt surface, the softening of the overlap region, and the position of the overlap.

2.16. Low Belt Length/Pipe Diameter Ratio

The length of the overlap is typically expressed as the ratio between the belt length (B) and the pipe diameter (D) through the B/D ratio, which plays an essential role in ensuring the proper sealing of the pipe shape. The B/D ratio ranges from 3.5 to 4 in most applications [1,2,61,62,76]; however, it varies according to the design characteristics, belt construction, standards utilized, and previous experience of the engineering specialists. According to [77], the lateral force of a pipe belt is correlated to the lump size of the bulk solid, panel spacing, filling degree, and other parameters.

Using a static six-point pipe-belt-stiffness testing device, the authors of [2] concluded that more flexible belts require a higher B/D ratio to form a stable pipe shape. They also explored the increase in the pipe diameter using the same belt samples, which resulted in a decrease in the contact forces owing to the decrease in the opening force. Therefore, the adoption of a higher B/D ratio increases the contact forces, thereby counteracting the rotary moment of the belt. Figure 10 illustrates belt overlap at different lengths.

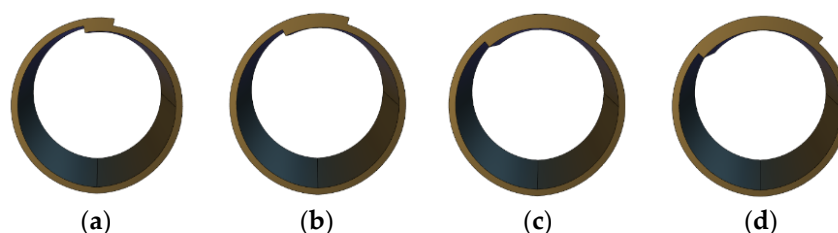


Figure 10. B/D ratio at different overlapping lengths for the nominal diameter of 375 mm. (a) B/D ratio 3.6; (b) B/D ratio 3.8; (c) B/D ratio 4.0; (d) B/D ratio 4.2.

Proactive measures may include the adoption of a more conservative B/D ratio for belt selection in the design stages; the closer it is to 4, the better the increase in the contact forces. However, the recommendations listed in the sections on high belt sag and improper belt reinforcements should be considered. The adoption of higher B/D ratios also requires caution in properly selecting the panel arrangement: (i) in-line panels may favor the pinching of the belt between the idler rolls during a change in the overlap position, even within the expected range; (ii) offset panels may favor the friction of the belt with the

housing of idler rolls in the case of too-soft belt edges, requiring the adoption of longer idler rolls.

2.17. Improper Transverse Bending Stiffness

The transverse bending stiffness of a belt is as relevant as the routing of a pipe conveyor to achieving operational stability [9]. A higher belt stiffness results in increased contact forces, resulting in better resistance to twisting and collapse in the curved flights of the conveyor [10,29,63,78]. However, increased values can lead to the opening of the belt edges between the idler panels [7], increasing the belt tension and the overall power consumption of the conveyor [10].

The transverse bending stiffness must be increased for heavier belts to form a stable pipe shape [3]; for the same modulus of elasticity, thicker and narrower belts have higher stiffness values than thinner and wider belts [79]. During operations, the belt's properties change owing to the natural aging of its rubber material, the interaction between the belt and the bulk solid, the propagation of cracks in the belt covers, and the fatigue of the belt due to repetitive opening cycles [80,81].

To determine the proper balance of this parameter, we recommend applying test methods according to ISO 703:2017 Conveyor belts—Transverse flexibility (troughability) [45] to belt samples at early conceptual stages, combined with the coupled use of FEM and DEM analysis. For energy consumption optimization and proper power specifications, designers and engineers should consider measuring the indentation rolling resistance in belt samples by applying the following methods: (i) DIN EN 16974:2016 conveyor belts for indentation rolling resistance (IRR) related to belt width requirements and testing, or (ii) the AS 1334.13:2017 methods of testing conveyor and elevator belting for the determination of the IRR of conveyor belting [33,82,83].

2.18. Poor Structural Alignment

Poor structural alignment contributes to belt rotation in a pipe conveyor, which is associated not only with the construction phase but also with improper maintenance procedures over the equipment's lifecycle. Most of the length of a pipe conveyor is enclosed through confinement on hexagonal supports [15]; therefore, the proper alignment of the supports is a key parameter for the operational stability of the conveyor [61].

The improper alignment of the panels relative to the conveyor centerline may lead to uneven contact forces, resulting in a rotary moment caused by this structural issue. In cases where one or more contact forces become equal to zero, undesirable belt rotation is likely to be triggered [7,34]. In addition, the misalignment of the transverse plane of a panel (radial displacement related to the belt centerline) will affect the belt training, as the contact with the idler rolls will be improperly angled, thereby possibly guiding the belt into a twisted position.

Good practice for ensuring proper alignment may include the following approaches: (i) numbering all hexagonal supports and mapping each center point for longitudinal alignment; (ii) applying 1/1000th of the section length as the maximum deviation for drawing the conveyor centerline [62], which typically uses the panel spacing as the minimal length; (iii) mapping the transverse plane for each panel and applying 0° of radial displacement to the conveyor centerline, for which the curve radius shall be carefully considered; and (iv) documenting the positioning during the construction and commissioning stages for future maintenance.

2.19. Heterogeneous Hardness along the Belt Length

Heterogeneous belt properties affect the rotary moment, thereby affecting the performance of the pipe conveyor. Changes in the belt transverse bending stiffness along its length often cause recurrent variations in the overlap position within an observed range, even without the action of external forces, for example, during the empty operation of the conveyor (ghost misalignment).

This heterogeneity can be caused by (i) quality issues in the manufacturing process within the same batch, for example, one low-quality belt section that is vulcanized to healthy sections (typically, sections ranging from 500 m to 700 m), or (ii) the accelerated aging of specific sections, for example, if one belt section that was stored for a long time is vulcanized to original belt sections that were not stored before installation, which is a typical maintenance scenario for handling systems.

One of the main causes of this variation in belt stiffness is the variation in hardness, which is dependent on temperature and time [84] as a result of the manufacturing process of vulcanizing. The former may be triggered by under- or overheating during the manufacturing vulcanizing process, whereas the latter may be triggered by the loss of troughability [33] because the materials are stored for a long time in a flat position (or simply because of the difference in the accommodation time between the stored and installed belt sections).

Proactive measures may include ensuring due diligence for all manufacturing processes related to the belt, strictly following the set of standards related to the designed technical specifications, and guaranteeing appropriate storage conditions for the belt according to the ISO 5285:2012 Conveyor belts—Guidelines for storage and handling [39,85]. The splicing of pipe belts with different properties is strongly discouraged, and periodic monitoring of the belt hardness for different belt sections is recommended to support maintenance decision-making processes.

2.20. Vulcanizing and Splicing Issues

Issues in belt splices can also lead to ghost misalignment on a pipe conveyor, triggered by (i) the localized hardness of the increased value as a result of vulcanizing, or (ii) squaring problems of the splice during its preparation. Owing to its high belt flexibility and tension requirements, hot vulcanization is a splicing method used for pipe belts. Steel cord belts employ standard splices, whereas fabric and aramid belts employ finger splices instead of standard ply splices [1], as shown in Figure 11.

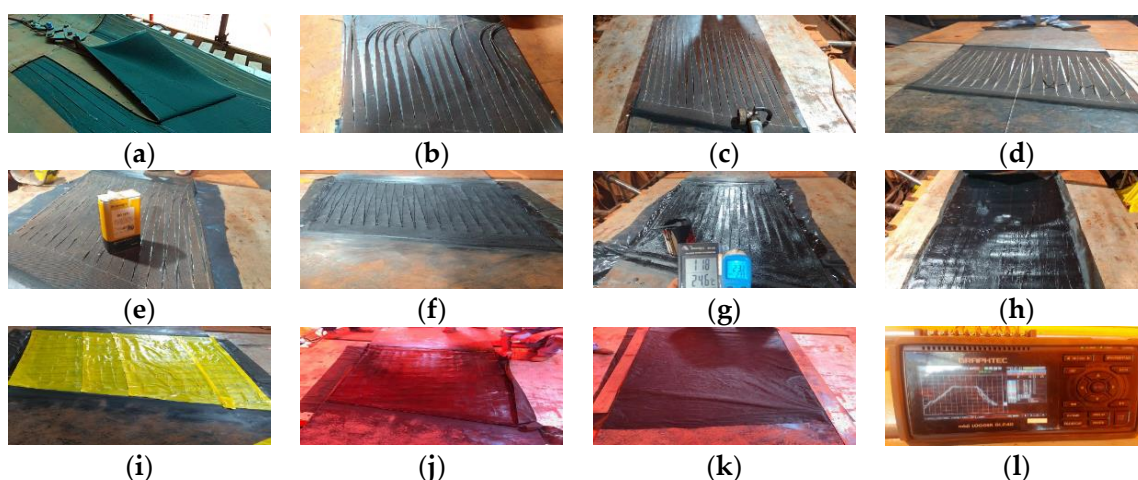


Figure 11. Finger splice on a pipe conveyor belt: (a) cover removal; (b) fingers being cut out; (c) finger fitting; (d) final squaring; (e) solvent application; (f) mounting of splice transition zone (g) rubber cement application; (h) back cover mounting; (i) bonding rubber application; (j) top cover mounting; (k) conforming of belt edges; (l) monitoring of temperature pressure.

Vulcanizing pipe belts is a sensitive manufacturing process that is affected by several factors, the most relevant of which are the vulcanizing temperature, time, pressure, moisture contamination, air trapping, expired consumables, and improper materials [86]. Poor vulcanization results in porous rubber, accelerated aging, increased hardness, and reduced troughability [87].

Proactive measures may include the close monitoring of the vulcanizing parameters that are properly specified for the selected belt and its cover properties. Documented

information on splicing should be retained, detailing the batch numbers and validity of the materials, along with the temperature, pressure, heating cycle, curing cycle, splice dimensions (square, length, width, and thickness), and belt hardness. Cautionary measures include replacing low-quality splices and applying recommendations on heterogeneous hardness along the belt length.

2.21. Excessive Tension during Installation

Belt installation is a critical activity in pipe conveyors for several reasons. This is a costly, time-consuming task with significant safety risks for both the craft involved and the belt itself. Thus, belt rotation may occur during this process because of the high initial opening force combined with an excessive tension force. Belt damage during this stage may delay or even end installation procedures, resulting in severe business consequences. Excessive belt bending, both lengthwise (owing to improper reeving during pre-splicing) and crosswise (owing to collapse or over-tension), may lead to recurrent problems during the commissioning, startup, and operational phases.

The passage of the belt for long pipe conveyors demands the simultaneous operation of the belt driving system in creep speed with an external pulling source, for which a powered winder is the recommended instrument [64,88,89]. However, technical or resource constraints may result in the use of bulldozers, wheel loaders, or tractors, without controlling the pulling force.

The steps include an analysis of the terrain and routing to determine the pre-splice location, where the belt is placed in overlapping folds after a serial sequence of splicing. The next step involves connecting the new belt to a wire rope (using a torpedo-shaped device in the case of the initial installation or after a belt breakage [64]) or to the existing belt (using a temporary splice). The wire rope (or the existing belt) is then pulled out from the conveyor structure, and the new belt is folded into the pipe shape for the first time [88,89]. Figure 12 gives examples of the connection devices for the installation or replacement of pipe belts.

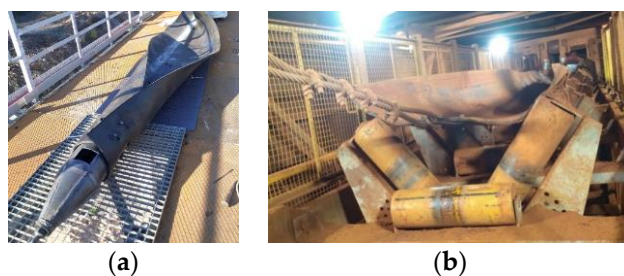


Figure 12. Pipe-belt pulling: (a) torpedo-shaped device; (b) removal of ripped belt.

Proactive measures may include the use of a digital tension dynamometer or the torque control of the external pulling device combined with the adoption of the minimum viable creep speed, which is technically and operationally influenced by the torque control of the motorization and minimal allowable passage time, respectively. The experienced craft must be positioned along the conveyor path to track the overlap position and coordinate the passage of the belt using an effective three-way communication approach. If necessary, the release of the tension force and manual positioning of the twisted section are recommended, especially if approaching 360°. Designers and engineers must consider, in the early design stages, the required pulling force for proper belt installation as a criterion for providing a creep-speed driver. Figure 13 illustrates some steps of the belt installation process.



Figure 13. Pipe conveyor belt installation: (a) belt positioning; (b) belt winder; (c) folded belt; (d) flat round reel; (e) bulldozer pulling.

2.22. Improper Tension Force

An improper tension force in a pipe conveyor belt may result in undesirable rotational phenomena. The lower belt tension in enclosed flights increases the contact forces owing to the transverse bending stiffness of the folded belt [90] and the active/passive pressure of the powder or bulk material when the belt passes through an idler assembly [71,91]. This may result in the cranking and overhanging of the pipe belt [6], as well as material spillage and dust emissions from the unsealed belt edges [3].

In practical applications, a belt under tension causes an oscillation in the driving electrical current, which is often observed after the replacement of an existing belt. In contrast, an increase in the belt tension leads to reduced contact forces on enclosed flights [90], and high belt tensions require the application of increased transverse stiffness to prevent collapses in curved flights [63]. Both effects may lead to belt rotation issues.

Contrary to the observations for enclosed flights, a different effect was perceived at transition stations, with a direct relationship between the tension and contact forces, as well as an increased contact force asymmetry [20]. This behavior was caused by the highly nonlinear belt folding process, combined with differences in the geometry of the transition flight (required to align the belt centerline from the trough to the pipe shape). This effect is further explored in relevant studies using a static test rig developed at the Technical University of Kosice [5,6,14–21,92].

The starting and stopping transients may also lead to belt rotation due to inappropriate tension forces. A reduced start-up time results in higher belt tension and the loss of contact forces. The use of variable-frequency drives (VFD) or soft starters with a more conservative acceleration time contributes to smoother transients. However, a safe stop or motor trip results in low belt tension and a rapid increase in contact forces. In this case, the use of flywheels in drive pulleys and capstan brakes for take-up recovery is recommended [93–95]. In both cases, synchronization between the upstream and downstream conveyors is highly relevant for preventing the overfilling of conveyor belts and the clogging of transfer chutes.

Good practices may include planning the take-up tension range as a design parameter, limiting the low limit according to the required friction at the driving pulleys and the installed power, and limiting the high limit according to the admissible belt tension and the minimum required contact forces of the belt. It is also recommended that the take-up tension force be adjusted based on the belt diameter measurements at predefined lifecycle stages. The commissioning and wear-out phases will probably require increasing and decreasing take-up tension, respectively, as a management approach to maintaining the required operational stability.

2.23. Improper Belt Training

Belt training is an essential task throughout a pipe conveyor's lifecycle that seeks to prevent and correct undesirable rotations. One of the simplest approaches to pipe belt training is the adoption of passive methods, in which the idler rolls are slightly angled horizontally to counteract the rotary moment of the belt (typically, the bottom roll is used to increase the repeatability of the method). Active methods have also been tested through patented solutions, for which a monitoring device is used to track the overlap position, and an actuator moves the bottom idler roll (a less common application considers moving the entire panel) [96,97].

Regardless of the chosen approach, belt training is complex, frequently necessary, and time consuming; therefore, it should not be neglected because of the risk of severe operational consequences. Over the belt's lifecycle, the contact forces decrease and wrongly trained idler rolls hide potential failure triggers that add undesirable rotary forces. This condition may even affect the return side of the belt, as shown in Figure 14, where the rotation tendency is reduced because of the overlap at the bottom [1,2]. The contact forces on the return side are also affected by an incompletely understood reduction in diameter, possibly caused by the additional line mass in the overlap region [1] and a slip on the inner belt edge. This may lead to a decrease in the upper contact forces and a higher contribution of the bottom idler roll to the belt training.

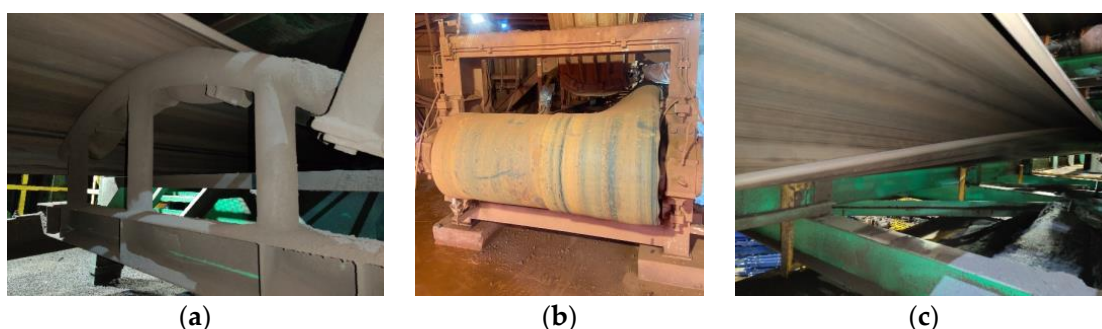


Figure 14. Belt training issues on the return side of pipe conveyors: (a) misalignment; (b) collapse at return pulley; (c) edge folding during return flight.

Good practices may include the proper mechanical completion of idler panels with documented confirmation of zero tilting of the training idlers before the hot commissioning of the conveyor. We recommend numbering all supports, including the final setup of training idlers, in the handover documentation. Regular inspections and the proper control of changes support operational stability and learning management. Furthermore, a smaller hexagonal arrangement in the return flight can be carefully evaluated during the design stage based on the specifications of the pipe belt.

2.24. Low Tension in Molding Rolls

The transition flights of a pipe conveyor impose the largest localized pressure on the belt, which affects its durability and operational stability throughout its lifecycle [18,98]. The length of the transition zone typically ranges from 25–60 times the nominal pipe diameter based on the belt reinforcement selection, and the belt is guided by molding rolls of progressive tensioning, set by their inclinations, from trough to pipe shapes [73]. These molding rolls, also known as torpedo or finger rolls, have rounded ends and are reinforced to support high reaction forces from the belt.

Research on transition flights is limited [20], although overlap formation is of fundamental importance in preventing belt rotation. In this sense, two essential processes occur in transition flights: (i) the definition of the overlapping side, and (ii) the accommodation of the belt edges governed by the friction of the overlapping region. The former process is defined by the proper regulation of the height difference between the belt edges (the

outer edge is higher), preventing the contact of the edge faces and the alternation of the overlapping sides, which may cause severe belt tensioning and uncontrollable rotation. Figure 15 shows the crossed overlap caused by low tension in the molding rolls. The latter process is defined by the belt length/pipe diameter ratio, and the molding rolls facilitate the positioning of the inner edge to its final position in an enclosed flight.

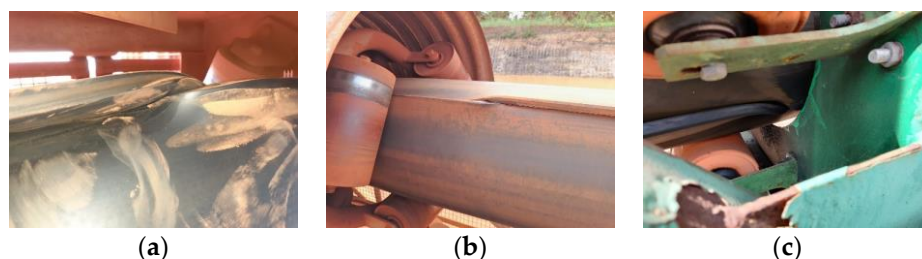


Figure 15. Alternation of the overlap sides: (a) during the replacement process; (b) after an issue on the molding rolls at the transition station; (c) on the return flight after an occurrence of twisting.

Excessive tension in the molding rolls also poses risks to the belt integrity, accelerating the wear and fatigue of the overlap region [18]. Therefore, care is required in determining the optimal fit point. In this sense, proactive measures include the proper dimensioning of supports and molding rolls, providing flexibility for setting inclinations, but preventing the looseness of bolting and changes in the adjusted positions. It is recommended that documented information be provided regarding the commissioning of molding rolls, in addition to the regular inspection of their positions, temperature, noise levels, and utilization hours. Good practices also include maintaining the same outer edge on both the carry and return sides, allowing the end user to invert the overlap position, and extending the belt usage in its wear-out phase. During the commissioning phase, a sliding additive (typically talcum [64] or graphite) may be used to reduce the friction forces in the overlap contact to compensate for the additional opening force of the belt.

2.25. Missing or Seized Idler Rolls

Failed idler rolls change the contact forces and cross-sectional geometry of the enclosed belt in a pipe conveyor. Typical failures include tube wear, bearing jamming, and shaft fixing failures, with consequent missing or seized idler rolls [31], the identification of which is difficult and may simultaneously affect different sections of the enclosed flight [15]. Ref. [99] studied the effect of a missing idler roll in a hexagonal housing and confirmed the strong dependence between the contact forces and the missing roller. It may also cause overhang and the cranking of the belt in the idler housing, leading to uncontrollable belt rotation owing to the increased belt tension and concentrated contact forces in the conveyor's panels [6]. Seized idler rolls may lead to uneven contact forces, causing idlers to withdraw from their housing [24] and triggering belt-training issues.

Appropriate design and tightness effectiveness are key elements for ensuring the reliability of idler rolls. The geometry of the hexagonal housing (60° for lateral rolls) favors the retention of contaminants and water ingress, for which primary (an outer protective shield) and secondary (a low-friction dust lip seal) sealing play an essential role in protecting the idler's bearings from premature weathering. Storage conditions (sheltered from sunlight and weathering), proper empty-run routines in the case of prolonged shutdowns, and quality-assurance procedures on manufacturing processes also maximize the useful life of idler rolls [31].

Proactive measures include the use of approved idler rolls under effective tightness tests, according to the relevant standardized methods, such as the Polish standard PN-M-46606:2010 [100], the South African standard SANS 1313-3:2012 [101], and the German standard DIN 22112-3:2022 [102]. A solid conveyor inspection plan is also strongly

recommended for mapping or replacing missing or failed idlers covering the entire conveyor structure.

Depending on belt routing, structure access, and manpower constraints, the adoption of digital twins and remote inspections may support increased productivity and operational stability. The authors of [103] proposed a digital twin approach to monitoring the contact forces in the case of missing rolls. Meanwhile, [104] listed technological advancements such as (i) the application of drones equipped with infrared and RGB cameras for automated survey flights, as shown in Figure 16; (ii) the use of acoustic disturbances to monitor microscopic changes in a backscattered laser through a fiber-optic cable along conveyor routing; and (iii) the use of self-powered smart idlers that are wirelessly connected to a cloud-connected gateway.

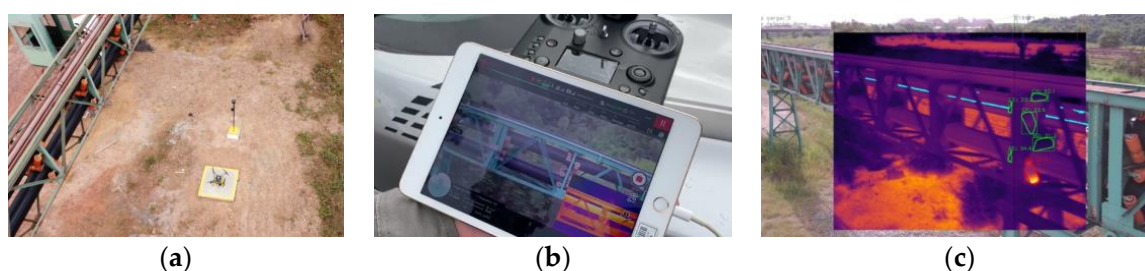


Figure 16. Automated survey flights in a pipe conveyor using Deep Track technology by Pix Force: (a) drone at launch pad; (b) survey flight in progress; (c) temperature detection using infrared and RGB cameras.

3. Methods

In this section, we propose a method to assess the tendency of a belt to maintain its stable operation in a pipe conveyor, based on the torque resulting from the interaction between the belt and the idler rolls. As detailed in the previous section, several causes and contributing factors interact with each other in highly nonlinear processes, which may lead to uncontrollable belt rotation. The method consists of obtaining the rotational holding torque of one curved-flight pipe conveyor by calculating the torque of its equivalent straight flight (theoretical) and the corresponding rotary moment of the belt by simulating or measuring the contact forces of this curved flight. We then apply an interference model using normal kernel-smoothed density functions to calculate the overlap stability index, ranging from 0 to 1.

3.1. Rotational Holding Torque

Given a theoretical straight-flight pipe conveyor running at optimal overlap positions on both the carry and return sides under all loading conditions, there is a rotational holding torque resulting from the contact forces between the pipe belt and the idler rolls, acting analogously to the radial braking forces in a block brake and maintaining the belt overlap stability. Considering that the rotary moment of the belt coincides with this holding torque and the belt does not rotate during the start-up or safe stopping of the conveyor, the overlap will remain at the 12 and 6 o'clock positions on the carry and return sides, respectively. By applying this concept to real pipe conveyors, the following characteristics must match for a theoretically equivalent straight flight:

- Material lift between the inlet and outlet;
- Longitudinal belt tension;
- Filling degree variation and bulk solids properties;
- Belt properties, environmental, and operational conditions.

Thus, this equivalent flight constitutes a reference model for assessing the effect of changes in the contact forces caused by routing or other externalities in real applications. The forces calculated from the equivalent straight flight yielded the reference rotational

holding torque; meanwhile, the forces simulated or measured from the real application yielded the rotary moment of the belt. Figure 17 illustrates the equivalent straight flight of a curved flight in a pipe conveyor.

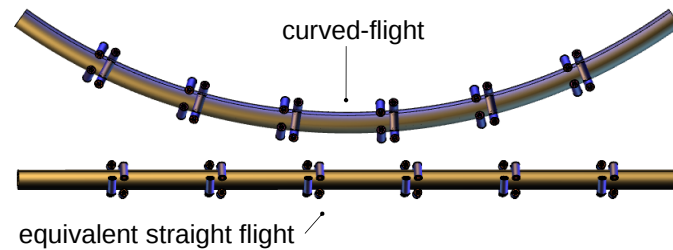


Figure 17. Equivalent straight flight of a curved flight in a pipe conveyor.

The contact of the idler roll exerts a pressure distribution on the back cover of the belt, both of which are curved in shape, causing viscoelastic deformation of the rubber. The authors of [71] assumed this curved contact to be elliptical in shape and used an analytical approach to calculate the normal forces using a generalized Maxwell model for a three-dimensional contact surface using Equations (1)–(4). The torque resulting from this asymmetric pressure distribution about the central axis of the idler roll results in IRR, in contrast to the running movement of the belt [105]. This model was adapted to describe the rotational holding torque about the central axis of the belt, which counteracts the rotary moment of the belt and governs the behavior of the overlap position during belt flight. The calculation model is shown in Figure 18.

$$w(x, y) = z_0 - \frac{x^2}{2R_1} - \frac{y^2}{2R_2}, \quad \begin{cases} -b \leq x \leq a \\ x \ll R_1 \\ y \ll R_2 \\ R_2 = R_{nom} + h \end{cases} \quad (1)$$

$$\sigma(x, y) = \frac{E_0}{2R_1 h'} (a^2 - x^2) + \sum_{i=1}^m \frac{E_i k_i}{h' R_1} \left\{ x - a + \left[(a + k_i) \left(1 - e^{\left(\frac{x-a}{k_i} \right)} \right) \right] \right\} \quad (2)$$

$$k_i = \frac{\eta_i V_b}{E_i} \quad (3)$$

$$F_{Nstr} = 2 \int_0^{c(z_0)} \int_{-b(y, z_0)}^{a(y, z_0)} \sigma(x, y) dx dy, \quad w(0, c) = 0 \therefore c = \sqrt{2R_2 z_0} \quad (4)$$

where $w(x, y)$ is the viscoelastic deformation depth of the contact surface, expressed in m; z_0 is the maximum indentation depth, expressed in m; x, y and z are a three-dimensional Cartesian plane; R_1 is the idler roll radius, expressed in m; R_2 is the external radius of the enclosed belt, expressed in m; R_{nom} is the nominal radius of the enclosed belt, expressed in m; h is the thickness of the belt, expressed in m; h' is the thickness of the back cover of the belt, expressed in m; $\sigma(x, y)$ is the pressure distribution in the viscoelastic contact surface, expressed in Pa; E_0 is the equilibrium modulus of Maxwell's generalized model, expressed in Pa; a is the point of the initial contact between the idler roll and the belt; b is the point of the final contact between the idler roll and the belt; c is the half-width of the elliptical contact surface, expressed in m; m is the number of elements of Maxwell's Generalized Model; E_i is the elastic (spring) elements of Maxwell's Generalized Model, expressed in Pa; k_i is an auxiliary variable, expressed in m; η_i is the viscous (dashpot) element of Maxwell's generalized model, expressed in Pa.s; V_b is the belt speed, expressed in m/s; and F_{Nstr} is the contact force, expressed in N, for the equivalent straight flight of the pipe conveyor.

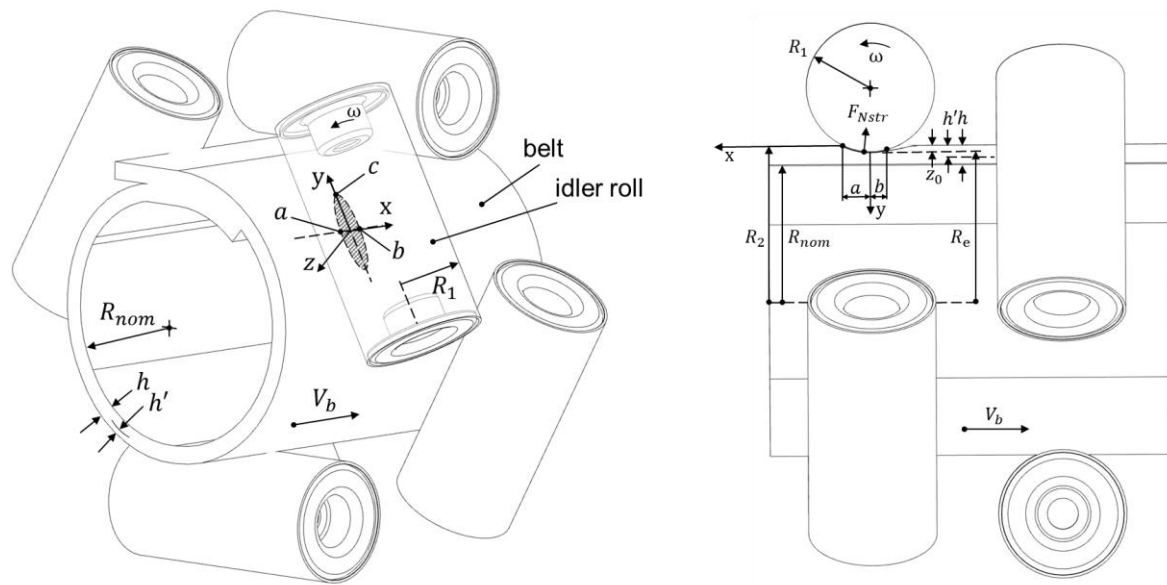


Figure 18. Variables for calculating the rotational holding torque of a pipe conveyor belt.

The torque around the center of the belt originates from the viscoelastic contact surface; therefore, the effective radius is of research interest, analogous to the calculation of the braking torque in a disk brake [106]. Thus, the effective radius is the sum of the nominal belt radius and the belt thickness less the indentation depth on the contact surface. The rotational holding torque for a given idler roll is a function of the coefficient of friction between the idler roll and the belt back cover, the contact force in the viscoelastic surface, and the effective belt radius [106].

The coefficient of friction for rubbers, adapted from [107] in the context of pipe conveyor belts, is influenced by several factors and is commonly expressed as a function of the following frictional forces:

Adhesion resulting from the intermolecular interaction between the belt's back cover and the idler rolls;

Hysteresis resulting from viscoelastic energy dissipation through the deformation of the belt's back cover;

Viscous resulting from the viscous shear of rainwater at the interface between the belt's back cover and idler rolls;

Cohesion resulting from crack initiation in the belt's back cover and the wear and microscopic surface roughness of the belt's back cover and idler rolls.

The friction forces caused by adhesion are the main contributors to the coefficient of friction; however, the hysteresis component should not be neglected [108]. Viscous forces occur mainly under poor weather conditions, and cohesion forces result from surface wear under aging mechanisms. According to [108], the contact mechanism involving rubber sliding surfaces is nonlinear, which complicates the prediction of the dependence between normal forces and the real contact area.

Full contact is formed when sufficiently large contact forces are applied, resulting in the contact area approaching the nominal value and interfacial separation approaching zero, as derived by Persson [108–110] and expressed in Equation (5). The nominal squeezing pressure p in the contact area is adapted here to consider the pressure distribution $\sigma(x, y)$ in the viscoelastic contact surface (see Equation (2)), and the adhesion coefficient of friction was derived as expressed in Equations (5)–(7):

$$A = A_0 \operatorname{erf}\left(\frac{p}{2G^{1/2}}\right) = 2 \int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} \operatorname{erf}\left(\frac{\sigma(x, y)}{2G^{1/2}}\right) dx dy \quad (5)$$

$$G = \frac{E^2 \kappa^2}{8(1-\nu^2)^2} \quad (6)$$

$$\mu_{adh} = \frac{\tau A}{F_{Nstr}} = \frac{\tau \int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} \operatorname{erf} \left\{ \frac{\sigma(x,y)}{2 \left[\frac{E^2 \kappa^2}{8(1-\nu^2)^2} \right]^{1/2}} \right\} dx dy}{\int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} \sigma(x,y) dx dy} \quad (7)$$

where μ_{adh} is the adhesion friction coefficient and is dimensionless; τ is the frictional shear stress acting on the area of contact, expressed in Pa; A is the area of real contact, expressed in m^2 ; A_0 is the area of nominal contact, expressed in m^2 ; F_{Nstr} is the contact force, expressed in N, for the equivalent straight flight of the pipe conveyor; a is the point of the initial contact between the idler roll and the belt; b is the point of the final contact between the idler roll and the belt; c is the half-width of the elliptical contact surface, expressed in m; $\sigma(x,y)$ is the pressure distribution in the viscoelastic contact surface, expressed in Pa; p is the nominal squeezing pressure in the contact area, expressed in Pa; and G is the characteristic normal pressure in the contact area, expressed in N. E is the Young's modulus of the elastic block, expressed in Pa; ν is the Poisson ratio of the elastic block and is dimensionless; κ is the RMS slope in the rough surface and is dimensionless.

We considered the hysteresis friction coefficient derived using the Klüppel and Heinrich model [111] based on the applied macroscopic pressure and velocity, as expressed in Equation (8). The mean deformation depth of rubber z_p was adapted to consider the average value of the viscoelastic deformation depth on the contact surface $w(x,y)$.

$$\mu_{hys} = \frac{1}{2(2\pi)^2} \frac{z_p}{\sigma_0 v} \int_{\omega_{min}}^{\omega_{max}} \omega E''(\omega) S(\omega) d\omega = \frac{1}{2(2\pi)^2} \frac{\left(\frac{\int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} w(x,y) dx dy}{\int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} dx dy} \right)}{\sigma(x,y) v} \int_{\omega_{min}}^{\omega_{max}} \omega E''(\omega) S(\omega) d\omega, \quad (8)$$

where μ_{hys} is the hysteresis friction coefficient and is dimensionless; $\langle z_p \rangle$ is the mean deformation depth of the rubber, expressed in m; σ_0 is the apparent normal stress from the macroscopic contact area, expressed in Pa; ω is the angular frequency of the induced excitation in the rubber, expressed in Hz; $E''(\omega)$ is the frequency-dependent loss modulus of the rubber material, expressed in Pa; $S(\omega)$ is the spectral power density owing to the rubber strain over the idler rolls, expressed in Hz; v is the sliding velocity, expressed in m/s; $w(x,y)$ is the viscoelastic deformation depth of the contact surface, expressed in m; and $\sigma(x,y)$ is the pressure distribution in the viscoelastic contact surface, expressed in Pa.

Therefore, the coefficient of friction for the contact between the back cover of the belt and the idler rolls is expressed by Equation (9) as:

$$\mu = \mu_{adh} + \mu_{hys} + \mu_{visc} + \mu_{coh}, \quad \mu_{adh} + \mu_{hys} \gg \mu_{visc} + \mu_{coh}, \quad (9)$$

where μ_{adh} is the adhesion friction coefficient and is dimensionless (see Equation (7)); μ_{hys} is the hysteresis friction coefficient and is dimensionless (see Equation (8)); μ_{visc} is the viscous friction coefficient and is dimensionless; and μ_{coh} is the cohesion friction coefficient and is dimensionless.

Finally, the total rotational holding torque of a given pipe conveyor flight comprising u hexagonal panels, each composed of six idlers, is expressed as the double summation of Equations (10)–(12):

$$R_e = R_{nom} + h - \frac{\int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} w(x,y) dx dy}{\int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} dx dy} \quad (10)$$

$$T_r = \mu F_{Nstr} R_e = \mu \left(2 \int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} \sigma(x,y) dx dy \right) \left(R_{nom} + h - \frac{\int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} w(x,y) dx dy}{\int_0^{c(z_0)} \int_{-b(y,z_0)}^{a(y,z_0)} dx dy} \right) \quad (11)$$

$$T_{total(u)} = \sum_{k=1}^u \sum_{j=1}^6 T_{r(k,j)} \quad (12)$$

where R_e is the effective pipe belt radius, expressed in m; R_{nom} is the nominal pipe belt radius, expressed in m; h is the thickness of the belt, expressed in m; a is the point of the initial contact between the idler roll and the belt; b is the point of the final contact between the idler roll and the belt; c is the half-width of the elliptical contact surface, expressed in m; $w(x,y)$ is the viscoelastic deformation depth of the contact surface, expressed in m; T_r is the rotational holding torque for a given idler roll, expressed in Nm; μ is the coefficient of friction between the idler roll and the belt back cover and is dimensionless; F_{Nstr} is the contact force, expressed in N, for the equivalent straight flight of the pipe conveyor; $\sigma(x,y)$ is the pressure distribution in the viscoelastic contact surface, expressed in Pa; u is number of idler panels in a given enclosed flight; k is the k th idler panel in a given enclosed flight; j is the j th idler roll in a given housing panel; and $T_{total(u)}$ is the rotational holding torque of an enclosed flight comprised of u panels, expressed in Nm.

As discussed in Section 2, the contact forces depend on various factors that may interact with each other in a highly nonlinear manner, causing undesirable belt rotation. Experimental approaches play a fundamental role in supporting the tuning and validation of analytical, numerical, and coupled methods. There are several variants in belt reinforcements and rubber properties according to each manufacturer's strategy, which create unique characteristics and challenges for each pipe conveyor project. Proper estimates of the rotational holding torque early in the conceptual stages drive the decision-making process, influencing the entire equipment lifecycle and the operational stability of the pipe conveyor. Therefore, conservative assumptions and simplified models save time and resources during the early initial design stages, thereby supporting the specification of small-scale tests as well as coupled FEM and DEM analyses for determining the belt contact forces, the flowability quantification of bulk materials, and measurements of the mechanical and rheological properties of the pipe belt.

Accordingly, we used the linear model proposed in [13], which calculates the contact forces as a function of the forming force of the belt, bulk gravity, and belt gravity. This coupled method uses an experimental six-point measuring device and FE analysis to obtain a simplified model of the contact forces in the straight enclosed flights of pipe conveyors, which adequately fits the research objectives of calculating a reference value for the rotational holding force. For ease of reporting, in the case study, the idler rolls were numbered as sequenced in the Belt Analyst software, with the bottom position identified as the first roll, followed by anticlockwise counting based on the belt running direction (see Figure 1). We proposed adjustment factors for the linear model based on the belt-filling degree resulting from the simulation of the applied case study. We applied steps of 10%, varying the belt filling degree from 0% to 80%, and divided the simulated forces in the conveyor's straight flight by the calculated values from the linear model proposed in [13], in order to obtain the adjustment factors proposed in Equation (14).

Herein, we assumed the forming force of the belt as calculated by Wesemeier [112] and derived from the approach of Chernenko [113], taking into consideration the concentrated radial contact force due to the belt overlap. Furthermore, the remaining useful life of the belt was considered as a percentage factor ranging from its nominal forming force (100%) to its collapsed condition (0%).

Additionally, we adopted a simplified calculation of the adhesion coefficient of friction, as proposed by Bhushan [114], which established that the friction coefficient μ_{adh} under constant shear stress is dependent on the normal load by $\propto F_{Nstr}^{-1/3}$. Furthermore, we adopted the hysteresis coefficient of friction as proposed by Ciavarella [115], who used a

simplified engineering formula derived from Persson's multiscale theory for rubber friction owing to viscoelastic losses for estimating μ_{hys} as $\kappa(E''/E)$. Next, we use the truncated value of κ fixed to 1.3 and the approximation $(E''/E) \cong 1$ as proposed by Persson [115]. The viscous and cohesion friction coefficients are substantially lower than the adhesion and hysteresis coefficients and are assumed to be negligible or to be further investigated in future experimental approaches, for which rainwater and aging variables will be incorporated. The calculation model is illustrated in Figure 19 and Equations (13)–(20).

$$\begin{aligned}
 F_{Nstr(k,1)} &= F_{1adj} \begin{cases} 0.91 F_{stif} + 0.49 G_b + (-1.18\psi^2 + 0.71\psi + 0.54) G_m, & 80\% \geq \psi \geq 50\% \\ 0.91 F_{stif} + 0.49 G_b + 0.60 G_m, & \psi < 50\% \end{cases} \\
 F_{Nstr(k,2)} &= F_{2adj} \begin{cases} 0.93 F_{stif} + 0.25 G_b + (-0.50\psi^2 + 0.41\psi + 0.25) G_m, & 80\% \geq \psi \geq 50\% \\ 0.93 F_{stif} + 0.25 G_b + 0.33 G_m, & \psi < 50\% \end{cases} \\
 F_{Nstr(k,3)} &= F_{3adj} \begin{cases} 1.15 F_{stif} - 0.05 G_b, & \text{overlap outer edge right side} \\ 0.95 F_{stif} - 0.05 G_b, & \text{overlap outer edge left side} \end{cases} \\
 F_{Nstr(k,4)} &= F_{4adj} (1.49 F_{stif} - 0.26 G_b - 0.08 G_m) \\
 F_{Nstr(k,5)} &= F_{5adj} \begin{cases} 0.95 F_{stif} - 0.05 G_b, & \text{overlap outer edge right side} \\ 1.15 F_{stif} - 0.05 G_b, & \text{overlap outer edge left side} \end{cases} \\
 F_{Nstr(k,6)} &= F_{6adj} \begin{cases} 1.06 F_{stif} + 0.26 G_b + (-1.17\psi^2 + 0.89\psi + 0.19) G_m, & 80\% \geq \psi \geq 50\% \\ 1.06 F_{stif} + 0.26 G_b + 0.34 G_m, & \psi < 50\% \end{cases}
 \end{aligned} \quad (13)$$

$$\begin{aligned}
 F_{1adj} &= 2.5894 \psi^4 - 3.2013 \psi^3 + 2.6390 \psi^2 - 1.8212 \psi + 1.3477 \\
 F_{2adj} &= F_{6adj} = 4.5918 \psi^4 - 6.6449 \psi^3 + 4.6908 \psi^2 - 1.6081 \psi + 0.9905 \\
 F_{3adj} &= F_{5adj} = 11.98 \psi^6 - 20.554 \psi^5 + 13.088 \psi^4 - 3.7948 \psi^3 + 0.486 \psi^2 - 0.0445 \psi + 0.9711 \\
 F_{4adj} &= 0.3606 \psi^4 - 0.1116 \psi^3 + 0.3268 \psi^2 + 0.4116 \psi + 0.7217
 \end{aligned} \quad (14)$$

$$G_b = Bl'hqg \quad (15)$$

$$G_m = \rho\pi(R_{nom})^2\psi l'g \quad (16)$$

$$F_{stif} = \frac{E}{1 - \nu_1\nu_2} \frac{h^3}{24R_{nom}\Delta_{ov}} l' \tau \quad (17)$$

$$\mu \cong \mu_{adh} + \mu_{hys} \cong \begin{cases} F_N^{-\frac{1}{3}}, & F_N > 1 \\ 1, & F_N \leq 1 \end{cases} + \kappa \frac{E''}{E} \quad (18)$$

$$T_{r(k)} = \mu F_N R_e = \left(\sum_{j=1}^6 \mu_{(k,j)} F_{N(k,j)} \right) (R_{nom} + h) \quad (19)$$

$$T_{total(u)} = \sum_{k=1}^u \sum_{j=1}^6 \mu_{(k,j)} F_{(k,j)} R_e, \quad R_e = R_{nom} + h \quad (20)$$

where $F_{Nstr(k,j)}$ represents the contact force of the j th idler roll belonging to the k th idler panel, expressed in N, for the equivalent straight flight of the pipe conveyor; F_N is the normal force, expressed in N; F_{stif} is the forming force of the belt, expressed in N; F_{jadj} is the adjustment factor of the j th idler roll based on the belt filling degree (dimensionless); G_b is the belt gravity, expressed in N; G_m is the bulk material gravity, expressed in N; B is the belt width, expressed in m; l' is the conveyor's panel spacing, expressed in m; τ is the remaining useful life of the belt, expressed as a percentage; h is the belt thickness, expressed in m; q is the belt density, expressed in kg/m³; ρ is the bulk material density, expressed in kg/m³; R_{nom} is the nominal pipe belt radius, expressed in m; R_e is the effective pipe belt radius, expressed in m; ψ is the filling degree rate of the enclosed belt, ranging from 0% to 80%; g is the acceleration due to gravity, expressed in m/s²; E is the elastic modulus of the belt in the lateral direction, expressed in Pa; $E''(\omega)$ is the frequency-dependent loss modulus of the rubber material, expressed in Pa; κ is the RMS slope in the rough surface (dimensionless); ν_1 and ν_2 are the Poisson's ratios of the belt in the longitudinal

and lateral directions, respectively (dimensionless); Δ_{ov} is the overlap length, expressed in m; u is the number of idler panels in a given enclosed flight; k is the k th idler panel in a given enclosed flight; j is the j th idler roll in a given housing panel; $T_{r(k)}$ is the rotational holding torque of the k th idler panel, expressed in Nm; $\mu_{(k,j)}$ is the coefficient of friction between the j th idler roll of the k th idler panel and the belt back cover (dimensionless); μ_{adh} is the adhesion friction coefficient (dimensionless) (see Equation (7)); μ_{hys} is the hysteresis friction coefficient (dimensionless) (see Equation (8)); μ is the coefficient of friction (dimensionless); and $T_{total(u)}$ is the rotational holding torque of an enclosed flight comprised of u panels, expressed in Nm. The outer edge of the overlap is defined in the molding rolls of the transition stations, based on the arbitrarily chosen side of the belt in the troughed transition flight.

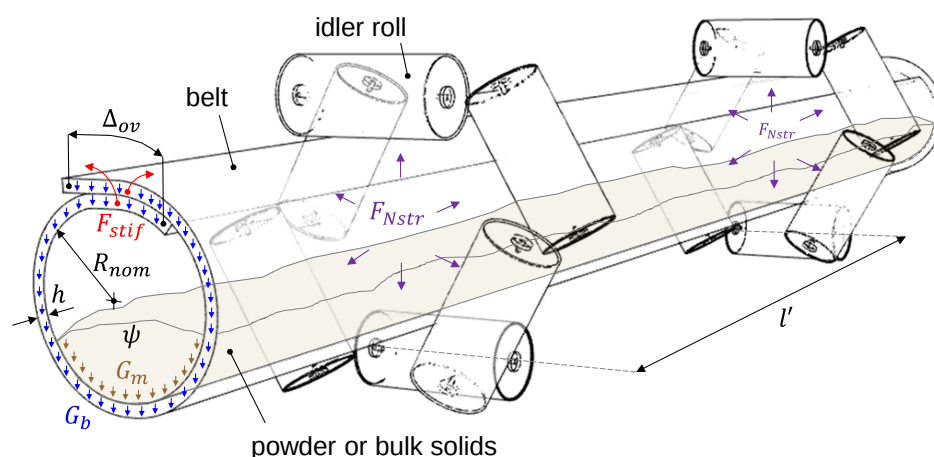


Figure 19. Variables for calculating contact forces in a pipe conveyor belt using a linear model.

3.2. Overlap Stability Index

Herein, we propose the calculation of the overlap stability index (*OSI*) of an enclosed belt flight in a pipe conveyor by assessing the interference area resulting from the normal kernel-smoothed density functions of the rotary moment of the belt flight and its rotational holding torque, ranging from 0 to 1. This method considers the minimum function between these two density functions, as well as the stress–strength interference model [116], which consists of quantifying the probability of a stress (load) parameter that is higher than a strength (capacity) parameter while assuming they are statistically independent.

The choice to apply kernel smoothing for density estimation is based either on the typical loading of a pipe conveyor or on the intuitive approach to analyzing multivariate data, which provides a notable advantage when using this method [117]. The no-load run and nominal loading are typical conditions in pipe conveyor operations, resulting in a nonparametric probability density response. Therefore, we applied a normal kernel, which is the most widely applied multivariate kernel function [118].

The kernel smoothing consists of centering a scaled function of integral one at each observation (a unimodal probability density function that is symmetric about zero). The value of the kernel estimate at one data point is the average of the n -kernel ordinates at that point. By combining the contribution from each data point, the kernel estimate will assume a relatively higher density where the true density has a large value [117]. We used the open-source Python library *scipy.stats.gaussian_kde* from the SciPy package for the calculation of the kernel-smoothed density functions.

We adopted the rotary moment of one enclosed belt flight and its rotational holding torque as the load and capacity parameters in the stress–strength model, respectively, coupled to the minimum function. The optimal stability value of *OSI* was assumed to be 0.5, which represents the full overlap of both normal kernel-smoothed density functions. As the value of *OSI* decreases (approaches 0), the bending stiffness of the enclosed belt

flight reduces and the belt tends to rotate owing to the reduced contact forces with the idler rolls.

In contrast, as the value of the *OSI* increases (approaches 1), the augmented bending stiffness of the enclosed belt flight induces a twisting phenomenon, owing to the increased tension force of the belt (see Section 2). Figure 20 illustrates the interference model used to calculate the *OSI* as expressed in Equations (21)–(23).

$$A_{interf} = \int_0^{\infty} \min\{f_{T_{total}(u)}(T), f_{M_{rot}(u)}(T)\} dT \quad (21)$$

$$Prob_{M>T} = \int_0^{\infty} F_{T_{total}(u)}(T) f_{M_{rot}(u)}(T) dT \quad (22)$$

$$OSI = \begin{cases} A_{interf} Prob_{M>T}, & Prob_{M>T} < 50\% \\ 1 - [A_{interf}(1 - Prob_{M>T})], & Prob_{M>T} \geq 50\% \end{cases} \quad (23)$$

where u is the number of idler panels in a given enclosed flight; A_{interf} is the interference area ratio between the normal kernel-smoothed density functions; $Prob_{M>T}$ is the probability of the rotary moment parameter be higher than the rotational holding torque parameter in an enclosed flight comprised of u panels; *OSI* is the Overlap Stability Index of an enclosed flight comprised of u panels; $f_{T_{total}(u)}(T)$ is the normal kernel-smoothed density function of the rotational holding torque of an enclosed flight comprised of u panels; $f_{M_{rot}(u)}(T)$ is the normal kernel-smoothed density function of the rotary moment of an enclosed flight comprised of u panels; $F_{T_{total}(u)}(T)$ is the normal kernel-smoothed cumulative distribution function of the rotational holding torque of an enclosed flight comprised of u panels; and T is the torque coordinate on the integration plan, expressed in N.m.

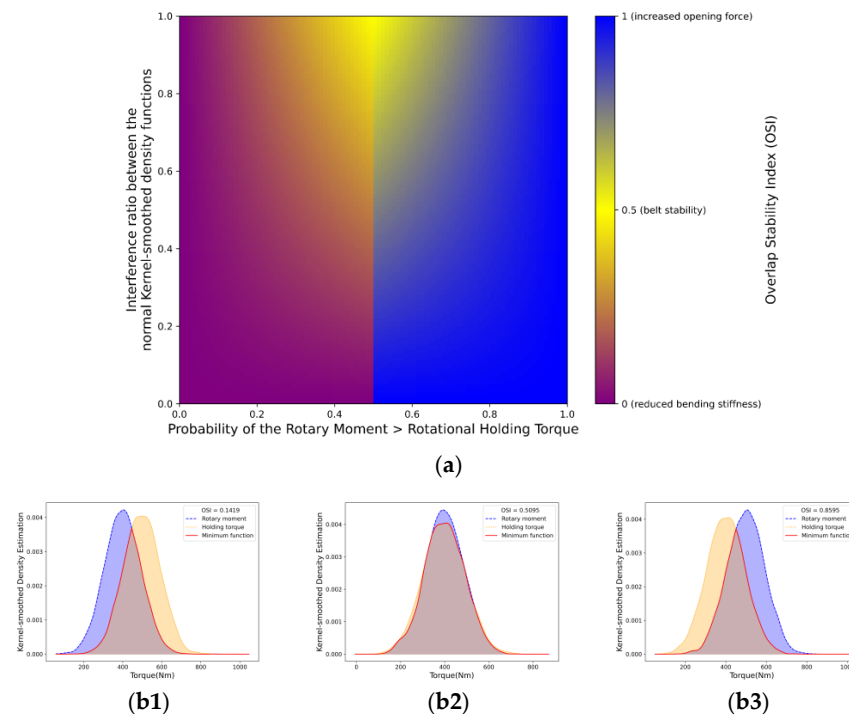


Figure 20. Interference model and kernel-smoothed density functions: (a) graphical view of *OSI*; (b1) unstable conditions due to reduced bending stiffness; (b2) stable conditions; (b3) unstable conditions due to increased opening force.

3.3. Experimental Method

We applied the proposed methodology to a case study of a short-flight pipe conveyor used in a bituminous-coal-handling system. The reasons for selecting this conveyor include certain challenging design and operational aspects, such as the utilization of the fabric belt, the high cycling rate owing to its short horizontal length, high belt speeds, the fact that it comprises right and left curves, operations under intense sunlight or moderate rain, and strategies for prolonged seasonal shutdowns. Belt rotation during the carry flight is a persistent failure mode that includes ghost misalignment and folding events during the transition flight of the discharge station. Table 3 presents the main equipment data.

Table 3. Main equipment data of the short-flight pipe conveyor.

Parameter	Unit	Value
Belt length	m	1.45
Panel spacing	m	2
Belt thickness	mm	19
Belt linear mass	kg/m	30.7
Powder or bulk density	kg/m ³	881
Nominal pipe diameter	mm	375
Overlap outer edge	Left or right	Left
Elastic modulus of the belt	MPa	10
Poisson's ratio of the belt in the longitudinal direction	-	0.45
Poisson's ratio of the belt in the transversal direction	-	0.45
Belt speed	m/s	4.9

The annual average demand for this handling system consists of ten shipments of mineral coal, for which the mean unloading time is five days each, totaling 13.7% of the annual utilization. It is predominantly used during the dry season, which ranges from July to December, as part of the unit's operational strategies for minimizing losses in the calorific value of the raw material.

The study consisted of a comparative assessment of modeled contact forces versus field observations to investigate belt rotation. The following steps were performed:

1. Acquisition of operation data. This step involved acquiring data related to the degree of belt filling during the unloading operations of a regular shipment. Mass flow data were collected, historicized, and processed using the OSIsoft PI System application for later retrieval at 5-min intervals during a 5-day unloading operation. The calibrated belt scale was located upstream of the pipe conveyor, and the data were logged in tons/h.
2. Collection of design parameters. This step comprised the gathering of technical data, including the curves and elevation of the conveyor routing, as well as the equipment specifications available in drawings, schematics, and structural and mechanical calculations.
3. Harmonization of the filling degree percentage. This step comprised the calculation of the filling degree percentage and then processing the data in 10% steps to correspond with those utilized in the software for the belt simulation. We programmed Python code and applied the conversion presented in Equation (24):

$$\psi_{harm} = \begin{cases} 0, & 0.00 \leq \psi \leq 0.01 \\ 0.1, & 0.01 < \psi \leq 0.11 \\ 0.2, & 0.11 < \psi \leq 0.21 \\ 0.3, & 0.21 < \psi \leq 0.31 \\ 0.4, & 0.31 < \psi \leq 0.41 \\ 0.5, & 0.41 < \psi \leq 0.51 \\ 0.6, & 0.51 < \psi \leq 0.61 \\ 0.7, & 0.61 < \psi \leq 0.71 \\ 0.8, & 0.71 < \psi \leq 1.00 \end{cases} \quad (24)$$

where ψ_{harm} is the harmonized filling degree rate of the belt, and ψ is the field-measured filling degree rate of the enclosed belt.

4. Simulation of contact forces in curved flights. This step comprised the calculation of the contact forces of the belt in curved flights using the Belt Analyst software, from no-load operations to 80% in increments of 10%. The data were exported in an electronic spreadsheet format for later processing.
5. Assessment of contact losses in curved flights. This step involved assessing the contact losses in the idler rolls, which may have contributed to the rotation. Warning labels were provided in the radar charts using Python coding for each idler roll position on the curved flights.
6. Calculation of contact forces in a straight flight. This step comprised the calculation of the contact forces of the belt in its theoretical equivalent straight flight using our proposed methodology, from no-load operations to 80% in increments of 10%. Python coding was used to process and store the data frames.
7. Calculation of the holding torque and rotary moment. This step involved calculating the rotational holding torque and rotary moment of the belt based on the calculated and simulated contact forces and considering the friction coefficient as proposed in our methodology. Python coding was used to generate the time series for each idler roll and the total values.
8. Estimation of normal kernel density functions. This step comprised the numerical calculation of the minimum functions and stress–strength models for the smoothed normal kernel density estimation of the rotational holding torque and rotary moment of the belt. A graphical view was provided in the probability density charts using Python coding for each curved section during the carry flight.
9. Calculation of the OSI. The index was calculated using the Python integration library, as per the proposed methodology. Python coding was used to calculate the index for each curved section during the carry flight.
10. Assessment of field installation. This step involved inspecting the critical flights detected in the assessment of contact losses, as well as those for which the OSI presented higher differences. We visually inspected the equipment during the flights of interest and discussed the findings and outlook.

4. Results and Discussions

We developed a software application using Python to process the inputs in the form of electronic text files in the “CSV” format, and we present the program variables for processing the equipment design parameters:

- Belt length
- Panel spacing
- Belt thickness
- Belt linear mass
- Powder or bulk density
- Nominal pipe diameter
- Overlap outer edge direction

- Elastic modulus of the belt
- Poisson's ratio of the belt in the longitudinal direction
- Poisson's ratio of the belt in the transversal direction
- Belt speed
- Belt's remaining life

The first csv file, related to the collection of real data, contained 1267 values of times-tamps and mass flow (in tons per hour), sampled at five-minute intervals throughout the handling operation of a single shipment (4.4 days). The second csv file, related to the simulated data in the specialist software, contained mass flow data (in tons per hour), the identifier of the curved flight, the indication of carry or return sides, and six contact forces.

Based on the conveyor profile, the specialist software generated grouped flights of various lengths to compose the equipment layout properly, as shown in Table 4. Owing to the similarities in the outputs of adjacent flights in our application, we grouped them as illustrated in Figure 21.

The outputs included comparative graphs of the mass flow and harmonized filling degree in 10% steps, as well as a summary report of the contact forces and rotational holding torque for an equivalent straight flight from no-load operations to an 80% filling degree in 10% steps. The next routines included the calculation of the rotary moment of the belt for each curved flight based on the uploaded simulation data. Figure 22 and Table 5 present the application-summarized outputs.

Table 4. Equipment profile details.

Point	Type	Vertical Radius (m)	Horizontal Radius (m)	Direction	Flight Length (m)	Lift (m)	Grouped Flights
1	Carry			Straight	171.88	16.77	1
2	Carry		270	Right	7.35	0.72	2–9
3	Carry		270	Right	5.77	0.60	
4	Carry	150	270	Right	1.40	0.15	
5	Carry	150	270	Right	1.40	0.12	
6	Carry	150	270	Right	0.50	0.04	
7	Carry	170	270	Right	7.14	0.61	
8	Carry	170	270	Right	7.17	0.01	
9	Carry	170	270	Right	16.40	0.01	
10	Carry		270	Right	46.45	0.04	10
11	Carry		270	Right	23.56	0.02	11–12
12	Carry		270	Right	23.56	0	
13	Carry		270	Right	73.44	0	13
14	Carry			Straight	42.44	0	14
15	Carry		270	Left	23.56	0	15–16
16	Carry		270	Left	23.56	0	
17	Carry		270	Left	21.44	0	17
18	Carry			Straight	45.00	0	18
19	Carry			Straight	72.00	0.15	19

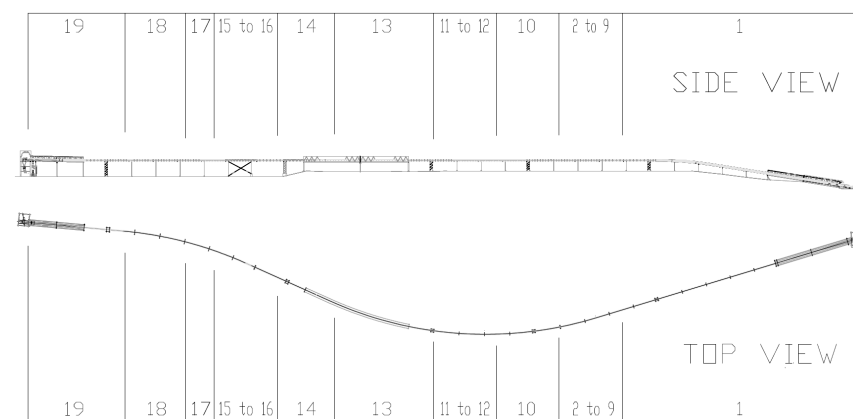


Figure 21. Equipment profile with grouped flights.

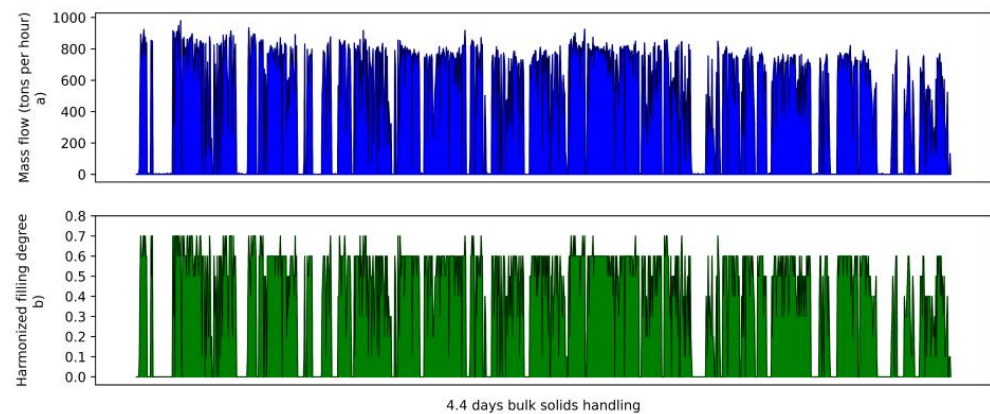


Figure 22. Mass flow and harmonized filling degree of real operations data.

Table 5. Contact forces and holding torque of the belt as a function of its harmonized filling degree.

		$\psi = 0$	$\psi = 0.1$	$\psi = 0.2$	$\psi = 0.3$	$\psi = 0.4$	$\psi = 0.5$	$\psi = 0.6$	$\psi = 0.7$	$\psi = 0.8$
Contact Forces (N)	F1	742.59	791.36	832.79	870.84	911.16	964.75	994.26	1035.66	1091.19
	F2	408.15	413.51	436.21	473.66	524.66	592.94	676.60	791.55	955.03
	F3	230.14	229.60	229.03	228.42	227.95	227.25	226.69	230.79	253.53
	F4	189.32	189.26	189.32	189.34	189.31	189.28	189.29	189.34	189.30
	F5	284.75	284.09	283.38	282.62	282.04	281.17	280.48	285.55	313.69
	F6	450.32	452.23	473.82	511.72	564.33	637.39	700.60	764.60	815.84
Rotational Holding Torque (Nm)	T1	216.28	230.11	241.84	252.61	264.01	279.15	287.48	299.16	314.82
	T2	120.93	122.47	128.98	139.70	154.28	173.75	197.55	230.16	276.40
	T3	69.54	69.38	69.21	69.04	68.90	68.69	68.53	69.72	76.33
	T4	57.63	57.61	57.63	57.64	57.63	57.62	57.62	57.64	57.63
	T5	85.38	85.19	84.98	84.76	84.59	84.34	84.14	85.61	93.74
	T6	133.02	133.57	139.75	150.58	165.60	186.40	204.37	222.52	237.04
	Total	682.78	698.32	722.39	754.33	795.00	849.96	899.70	964.82	1055.97

In this summary report for the equivalent straight flight, contact forces are numbered as sequenced in the Belt Analyst software, with the bottom position identified as F1—as well as the corresponding torque T1—followed by anticlockwise counting based on the belt's running direction (see Figure 1). Contact forces F1, F2, and F6 increased with higher filling degrees due to the contribution of the bulk material's gravity for the bottom idler rolls. As for the lateral idler rolls, the contact forces F3 and F5 decreased from a 0% to a 60% filling degree, followed by a slight increase above this percentage. This reduction is caused by the general reduction in the friction factor due to the flexural resistance of the carried material, which reduces the indentation factor [33,71]. On the other hand, the increase occurs due to the approximation of the maximum allowed filling degree for which the recommendations of the maximum filling degree apply (see Table 1). Finally, the forming force of the belt mostly influences the contact force in the upper idler roll F4.

Subsequently, the code appended both the calculated holding torque and the rotary moment in each harmonized sample for subsequent plotting in radar charts grouped by curved flight, as illustrated in Figure 23. Red tags indicate contact losses in the affected idler rolls. The final steps included the estimation of the normal kernel density functions and the calculation of the OSI for each curved flight using the results of the previous steps and a library for numerical methods, as shown in Figure 24.

Figure 23 illustrates the distribution of the rotational holding torque and the rotary moment of the belt for curved flights in the idler rolls (positions 1 to 6). The results demonstrate the asymmetry between the contact forces in the upper lateral rolls 3 and 5 for flights 10, 13, 15, 16, and 17, with an important loss of contact in position 3 and an increased tension force in position 5. Figure 24 illustrates the kernel-smoothed density plots for the same curved flights. The plots present two peaks of density corresponding to the longer periods of operation when empty or with the conveyor at a 70% filling degree. Curved flights 15 and 16 presented unstable conditions due to an increased opening force, with an OSI of 0.8657.

The simulated rotary moment of the belt and the calculated holding torque indicated close values for most grouped flights, with the OSI approaching the optimal value, except for curved flights 15 and 16, where the OSI indicates the tendency of an increased tension force of the belt resulting from the lateral force in Position 5 of the idler support. Furthermore, when assessing the contact losses, we expanded the field analysis to include flights 10, 13, and 17, considering the low contact forces in Position 3 of the idler support.

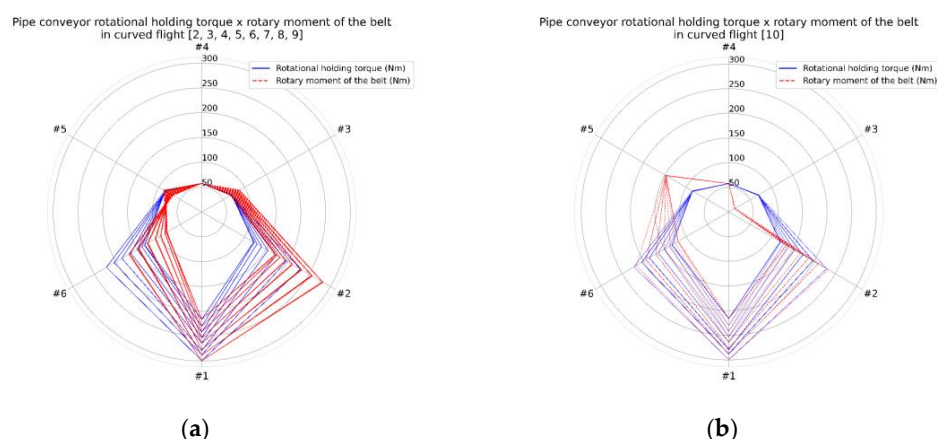


Figure 23. Cont.

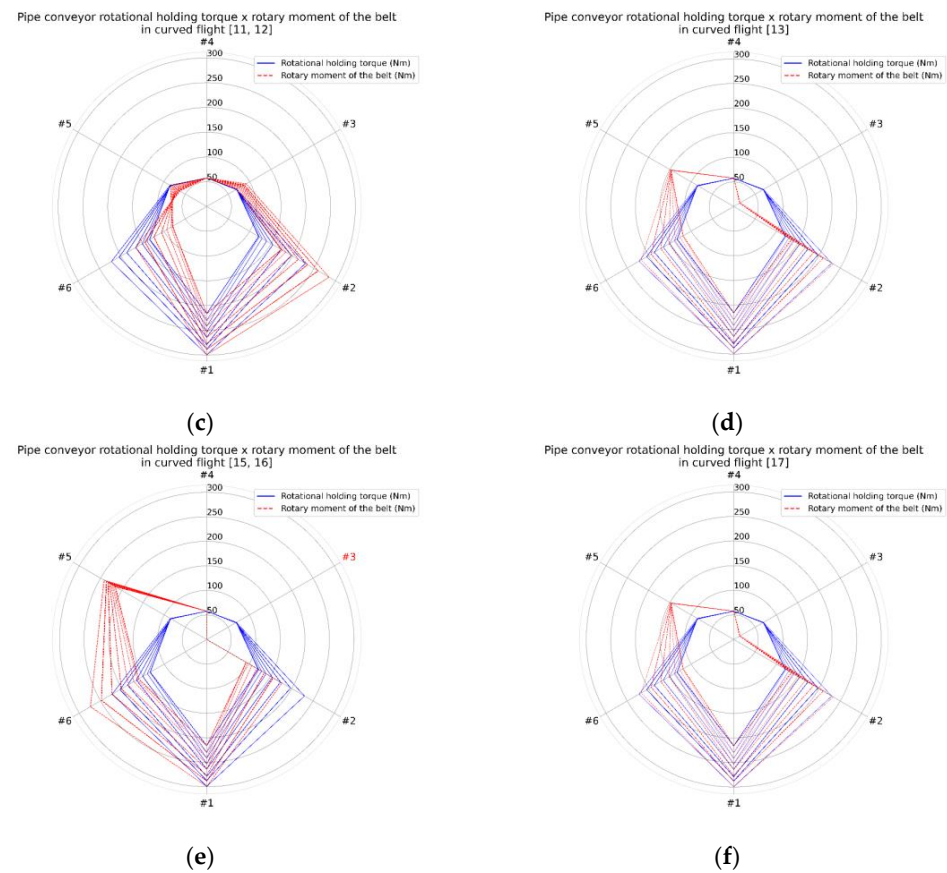


Figure 23. Radar charts of the holding torque and rotary moment of the belt for curved flights: (a) 2 to 9; (b) 10; (c) 11 and 12; (d) 13; (e) 15 and 16; (f) 17.

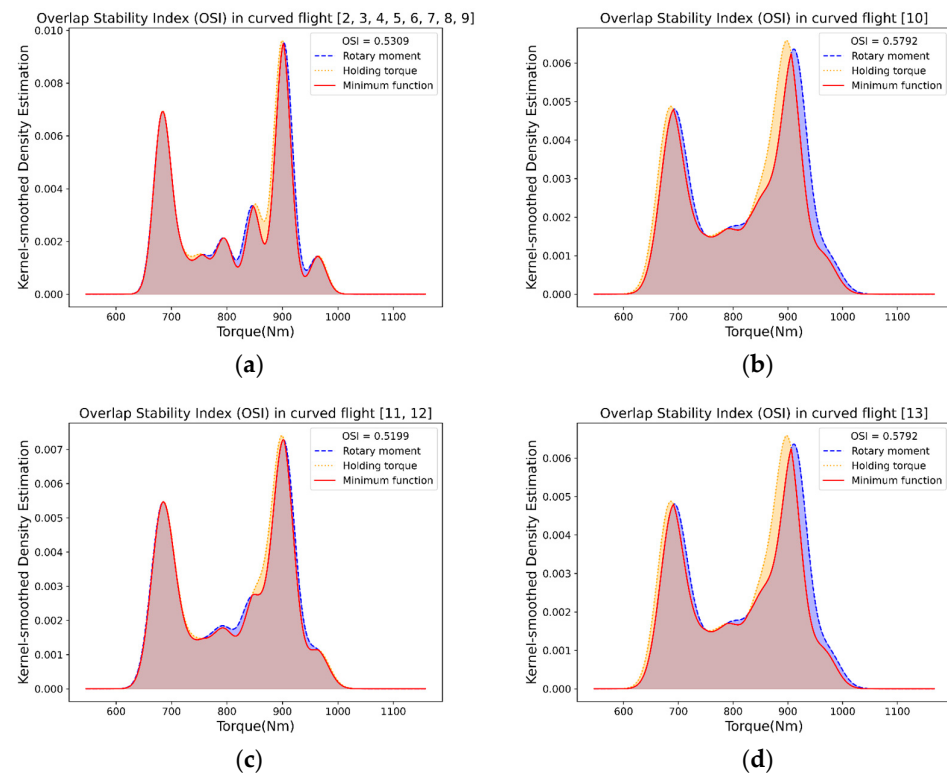


Figure 24. Cont.

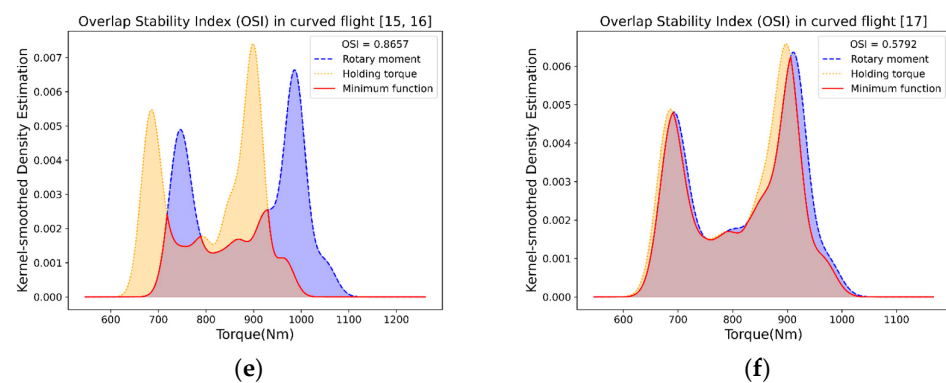


Figure 24. Kernel-smoothed density charts for curved flights: (a) 2 to 9; (b) 10; (c) 11 and 12; (d) 13; (e) 15 and 16; (f) 17.

Figure 25 shows the top view of the target points for inspection. We inspected the pipe conveyor belt in the zero-energy state and logged the position of the overlap at the time of the visual inspection, as well as the clearance between the back cover of the belt and the six idler rolls, as presented in Table 6.

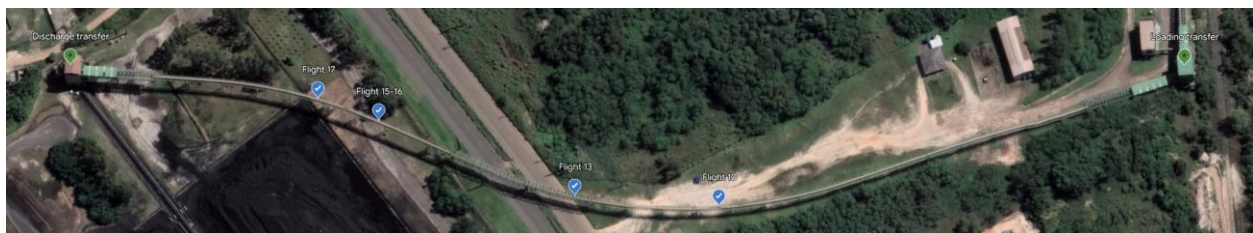


Figure 25. Top-view of selected points for field inspection in the selected pipe conveyor.

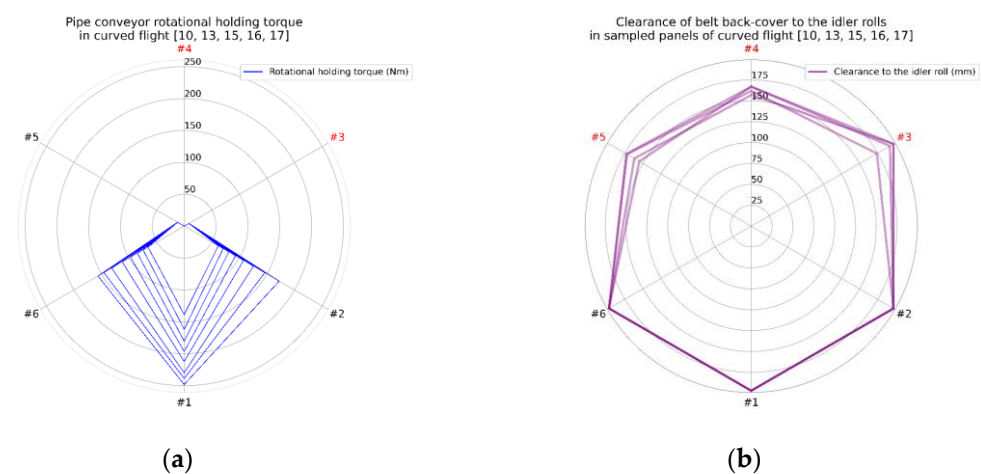
Table 6. Field survey of selected curved flights.

Curved Flight	Overlap Position (°)	Clearance to the Belt Back Cover (mm)					
		Idler 1	Idler 2	Idler 3	Idler 4	Idler 5	Idler 6
10	10	0	0	0	30	42	0
13	10	0	0	0	40	35	0
15–16	0	0	0	5	30	24	0
17	0	0	0	23	35	25	0

Field surveys have shown an important loss of contact with the upper idler rolls along the pipe conveyor, as presented in Table 7, which significantly compromises the ability to maintain the stable operation of the belt, favoring material spillage and belt rotation [7,34,65]. These findings suggest that the bending stiffness of the enclosed belt flight was reduced compared to the initially calculated value. Thus, we determine the remaining useful life of the belt in our application, using the ratio between its cumulative useful life since the last replacement (57 months) and its estimated lifespan (72 months), resulting in 21% of the belt's life remaining. Figure 26 presents the radar charts for the calculated rotational holding torque in this updated scenario.

Table 7. Proposed lifespan for the replacement of the pipe belt based on an assessment of contact forces.

Variable	Value
Average (%) per shipment	0.4
Contact force in the bottom roll F1 (N)	824.3543
Contact force in the upper roll F4 (N)	41.2177
Solved τ (%)	62.8751
Estimated lifespan of the belt (months)	72
Proposed lifespan for replacement (months)	26.7299

**Figure 26.** (a) Rotational holding torque in curved flights 10, 13, 15, 16, and 17 with loss of contact in positions 3 and 4 of idler rolls; (b) clearance between the belt and idler rolls of curved flights 10, 13, 15, 16, and 17 considering the remaining useful life of the belt, with loss of contact in positions 3, 4 and 5 of idler rolls.

The calculated results (see Figure 26a) are consistent with the observed changes in the physical properties of the belt (see Figure 26b) caused by the natural aging of its rubber material, the interaction between the belt and the bulk solid, the propagation of cracks in the belt covers, and the fatigue of the belt owing to repetitive opening cycles [80,81]. The belt tension force set during design stages and adjusted during the break-in period contributes to reducing the forming force of the belt (ν_1 in Equation (17)), justifying the observed contact losses on the enclosed flights [90]. This scenario also favors the occurrence of a ghost misalignment of the belt because of the concentrated contact forces in the bottom idler rolls, combined with the oscillation of the center of gravity of the pipe [1]. Analysis of the nominal contact forces of the belt suggests that flights 15–16 imposed stresses on the pipe belt in Position 5 of the idler support, potentially leading to localized overstressing [23,61], which is an important contributing factor in reducing the forming force of the belt in the long run.

Based on the proposed methodology and further analysis, we concluded that the combination of the causal factors, such as the reduced transverse bending stiffness of the belt and the increased belt tension force, were the root causes of the failure mode of persistent belt rotation in the carry flight of the selected pipe conveyor. The contributing factors include the augmented contact forces on flights 15–16 in the initial stages of the belt's lifespan. As a corrective action plan, we recommend reducing the take-up tension to the minimum value required to maintain appropriate friction in the drive pulley (ν_1 in Equation (17)). In addition, we recommend replacing the existing belt and adjusting the belt's lifespan for this application in the contingency mode, minimizing disruption

risks for the operational unit (τ in Equation (17)). Using the application-summarized outputs and the solver tool, we calculated the recommended useful life for the average operational filling degree. To achieve this, we calculated the remaining useful life for which the contact force in the upper idler roll was 5% of the contact force in the bottom position, as presented in Table 7. We recommend the application of a cost-and-benefit analysis for the adoption of this biannual replacement plan (26.7 months), considering the risk appetite of the operational unit and the total cost of ownership when accounting for the impact of this failure mode in the realization of value to the internal and external stakeholders.

As a preventive action plan, we recommend a design change in the belt's properties by applying test methods to belt samples, combined with coupled FEM and DEM analyses. Chemical interactions [1,58] between bituminous coal and the belt's top cover, as well as weathering during prolonged seasonal shutdowns, accelerate belt aging, notably by increasing the belt hardness. Therefore, we recommend adopting chemical-, ozone-, and ultraviolet-resistant belts and applying test methods according to the following standards:

- ISO 703:2017 Conveyor belts—Transverse flexibility (troughability) [45];
- ISO 188:2023 Rubber, vulcanized or thermoplastic—Accelerated aging and heat resistance tests [46];
- ISO 1431-1:2022 Rubber, vulcanized or thermoplastic—Resistance to ozone cracking—Part 1: Static and dynamic strain testing [32];
- ISO 1817:2022 Rubber, vulcanized or thermoplastic—Determination of the effect of liquids [35];
- ISO 6943:2017 Rubber, vulcanized—Determination of tension fatigue [47];
- ISO 27727:2008 Rubber, vulcanized—Measurement of fatigue crack growth rate [48];
- ISO/TR 7620:2005 Rubber materials—Chemical resistance [119].

5. Conclusions

We analyzed the failure mode of belt rotation and proposed a model for calculating the OSI while considering the rotary moment of the belt and its rotational holding torque. For the rotational holding torque, we considered the theoretical equivalent straight flight of a real curved-pipe conveyor to calculate its optimal distribution of contact forces, which was used as a reference parameter to assess the tendency of rotation resulting from the transverse bending stiffness of the belt or its tension force. The proposed model considers a valuable set of equipment parameters, including the remaining useful life of the belt. The linearization allows for a simplified and cost-effective implementation process. This is relevant to ensuring the sustainable benefits of this solution, as well as smooth commissioning and operational phases, supporting the decision-making process for both the operational and maintenance teams. In addition, it offers a quick assessment tool for the early design stages, thus facilitating a cost-benefit analysis while considering the effects during the later stages of the equipment's lifecycle.

We implemented the proposed methodology in a case study of a short-flight pipe conveyor utilized in a bituminous-coal-handling system, in which belt rotation in the carry flight is a persistent failure mode. Using an application in Python, we assessed the contact losses in curved flights and calculated their overlap stability indexes, considering sampled data at five-minute intervals throughout the handling operation of a single shipment. We then concluded that the combination of the causal factors, such as reduced transverse bending stiffness and the increased belt tension force of the belt, are the root causes of the failure mode of persistent belt rotation in the carry flight of the selected pipe conveyor. The contributing factors included the augmented contact forces on flights 15 and 16 in the initial stages of the belt's lifespan. Finally, we proposed corrective and preventive action plans, considering the recommendation of the replacement interval of the pipe belt in the contingency mode and the adoption of design changes according to the relevant standards.

The limitations of this study include the simulation of contact forces to obtain the rotary moment of the belt. In future studies, experimental approaches for measuring contact forces using strain gauges should be considered to provide online inputs for calculating the

rotary moment of the belt, like the studies for measuring contact forces developed at the Technical University of Kosice [5,6,14–21,92]. In this way, the overlapping stability indexes can be processed in real time, allowing for advancements in the use of digital twins, as proposed by [103], or supporting applications of artificial intelligence in edge computing, as proposed by [120].

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References

- CEMA7. *Conveyor Equipment Manufacturers Association Belt Conveyors for Bulk Materials*, 7th ed.; Conveyor Equipment Manufacturers Association: Naples, FL, USA, 2014; 978-1891171-44-4.
- Zamiralova, M.E.; Lodewijks, G. Measurement of a pipe belt conveyor contact forces and cross section deformation by means of the six-point pipe belt stiffness testing device. *Measurement* **2015**, *70*, 232–246. [\[CrossRef\]](#)
- Zamiralova, M.E.; Lodewijks, G. Shape stability of pipe belt conveyors: From throughability to pipe-ability. *FME Trans.* **2016**, *44*, 263–271.
- Zheng, Q.J.; Xu, M.H.; Chu, K.W.; Pan, R.H.; Yu, A.B. A coupled FEM/DEM model for pipe conveyor systems: Analysis of the contact forces on belt. *Powder Technol.* **2017**, *314*, 480–489. [\[CrossRef\]](#)
- Molnár, V.; Fedorko, G.; Stehlíková, B.; Michalik, P.; Kopas, M. Mathematical models for indirect measurement of contact forces in hexagonal idler housing of pipe conveyor. *Measurement* **2014**, *47*, 794–803. [\[CrossRef\]](#)
- Molnár, V.; Fedorko, G.; Andrejiová, M.; Grinčová, A.; Tomašková, M. Analysis of influence of conveyor belt overhang and cranking on pipe conveyor operational characteristics. *Measurement* **2015**, *63*, 168–175. [\[CrossRef\]](#)
- Zamiralova, M.E. Design Aspects of Pipe Belt Conveyors. Ph.D. Thesis, Delft University of Technology, Delft, The Netherlands, 2017. [\[CrossRef\]](#)
- Hötte, S. *Experimentelle Untersuchungen zu den Normalkräften bei Schlauchgurten*; TEWISS-Technik-und-Wissen-GmbH: Garbsen, Germany, 2014.
- Sun, X.; Meng, W.; Zhao, H.; Yuan, Y.; Yang, Z. Analysis on the bending stiffness and the form force of the pipe conveyor belt. *Sens. Transducers* **2013**, *161*, 655.
- Zhang, Y.; Steven, R. Pipe conveyor and belt: Belt construction, low rolling resistance and dynamic analysis. In Proceedings of the SME Annual Meeting, Seattle, WA, USA, 19–22 February 2012.
- Molnár, V.; Fedorko, G.; Honus, S.; Andrejiova, M.; Grincova, A.; Michalik, P.; Palencar, J. Research in placement of measuring sensors on hexagonal idler housing with regard to requirements of pipe conveyor failure analysis. *Eng. Fail. Anal.* **2020**, *116*, 104703. [\[CrossRef\]](#)
- Wang, S.; Li, D.; Hu, K. Analysis and experimental study on pressure characteristics of supporting roller group of pipe belt conveyor. *Shock Vib.* **2019**, *2019*, 7061847. [\[CrossRef\]](#)
- Guo, S.; Huang, W.; Li, X. Normal force and sag resistance of pipe conveyor. *Chin. J. Mech. Eng.* **2020**, *33*, 48. [\[CrossRef\]](#)
- Molnár, V.; Fedorko, G.; Andrejiová, M.; Grinčová, A.; Kopas, M. Monitoring of dependences and ratios of normal contact forces on hexagonal idler housings of the pipe conveyor. *Measurement* **2015**, *64*, 168–176. [\[CrossRef\]](#)
- Molnár, V.; Fedorko, G.; Andrejiová, M.; Grinčová, A.; Michalik, P. Online monitoring of a pipe conveyor. Part I: Measurement and analysis of selected operational parameters. *Measurement* **2016**, *94*, 364–371. [\[CrossRef\]](#)
- Molnár, V.; Fedorko, G.; Andrejiová, M.; Grinčová, A.; Michalik, P. Online monitoring of pipe conveyors part II: Evaluation of selected operational parameters for the design of expert system. *Measurement* **2017**, *104*, 1–11. [\[CrossRef\]](#)
- VMolnár; Sabovčík, M. Static testing evaluation of pipe conveyor belt for different tensioning forces. *Open Eng.* **2019**, *9*, 580–585. [\[CrossRef\]](#)

18. Molnar, V.; Fedorko, G.; Stehlikova, B.; Paulikova, A. Influence of tension force asymmetry on distribution of contact forces among the conveyor belt and idler rolls in pipe conveyor during transport of particulate solids. *Measurement* **2015**, *63*, 120–127. [\[CrossRef\]](#)
19. VMolnár; Fedorko, G.; Stehlíková, B.; Kudelás, L.; Husáková, N. Statistical approach for evaluation of pipe conveyor's belt contact forces on guide idlers. *Measurement* **2013**, *46*, 3127–3135. [\[CrossRef\]](#)
20. Stehlikova, B.; Molnar, V.; Fedorko, G.; Michalik, P.; Paulikova, A. Research about influence of the tension forces, asymmetrical tensioning and filling rate of pipe conveyor belt filled with the material on the contact forces of idler rolls in hexagonal idler housing. *Measurement* **2020**, *156*, 107598. [\[CrossRef\]](#)
21. Molnár, V.; Fedorko, G.; Honus, S.; Andrejiová, M.; Grinčová, A.; Michalik, P. Prediction of contact forces on idler rolls of a pipe conveyor idler housing for the needs of its online monitoring. *Measurement* **2019**, *139*, 177–184. [\[CrossRef\]](#)
22. Molnar, V.; Fedorko, G.; Stehlikova, B.; Michalik, P.; Kopas, M. Influence of tension and release in piped conveyor belt on change of normal contact forces in hexagonal idler housing for pipe conveyor loaded with material. *Measurement* **2016**, *84*, 21–31. [\[CrossRef\]](#)
23. Hötte, S.; Overmeyer, L.; Wennekamp, T. Form force behaviour of pipe conveyors in different curve radii. *Bulk Solids Handl.* **2011**, *31*, 164–169.
24. Molnár, V.; Fedorko, G.; Stehlíková, B.; Tomašková, M.; Hulínová, Z. Analysis of asymmetrical effect of tension forces in conveyor belt on the idler roll contact forces in the idler housing. *Measurement* **2014**, *52*, 22–32. [\[CrossRef\]](#)
25. Bombasaro, E.; Oddera, R. Field measurement of contact forces on rollers for a large diameter pipe conveyor. *Open Eng.* **2020**, *11*, 130–141. [\[CrossRef\]](#)
26. Santos, L.D.S.E.; Filho, P.R.C.F.R.; Macêdo, E.N. 2020Development of idler for measuring indentation rolling resistance in pipe conveyor belts. *Measurement* **2020**, *153*, 107413. [\[CrossRef\]](#)
27. de Graaff, R.M.T. *Troughability Testing on Troughed and Pipe Conveyor Belts-Trogbaarheid Testen Met Trog-en Pijp-Transportbanden*; Delft University of Technology: Delft, The Netherlands, 2012.
28. Maksarov, V.; Zlotnikov, E.; Olt, J. Determining the load on support rollers the pipe conveyor belt. In Proceedings of the 28th DAAAM International Symposium, Zadar, Croatia, 8–11 November 2017; DAAAM International: Maria Lankowitz, Austria, 2017.
29. Batwe, S.; Jaju, S.; Vaidya, S. Pipe Conveyor. *Int. Res. J. Eng. Technol.* **2017**, *4*, 978–983.
30. Dmitriev, V.G.; Efimov, M.S. Evaluation and method for reducing the angular deviations of the tubular conveyor belt during rotational motion, Mining Information and Analytical. *Bull. Sci. Tech. J.* **2009**, *16*, 31–53.
31. Santos, L.D.S.E.; Soares, W.K.D.S.; Filho, P.R.C.F.R. A method for investigating the influence of rainwater on the useful life of idlers in pipe belt conveyors during seasonal operations. *Eng. Fail. Anal.* **2022**, *141*, 106702. [\[CrossRef\]](#)
32. *ISO 1431-1:2022; Rubber, Vulcanized or Thermoplastic—Resistance to Ozone Cracking—Part 1: Static and Dynamic Strain Testing*. ISO: Geneva, Switzerland, 2022.
33. Santos, L.D.S.E.; Filho, P.R.C.F.R.; Macêdo, E.N. Indentation rolling resistance in pipe conveyor belts: A review. *J. Braz. Soc. Mech. Sci. Eng.* **2021**, *43*, 230. [\[CrossRef\]](#)
34. Imai, A. Pipe conveyor Development, benchmark and future trend. In Proceedings of the BulkSolids India 2011, the International Conference & Exhibition on Storing, Handling and Transporting Bulk Solids and Powders, Mumbai, India; 2011.
35. *ISO 1817:2022; Rubber, Vulcanized, or Thermoplastic—Determination of the Effect of Liquids*. ISO: Geneva, Switzerland, 2022.
36. Lodewijks, G. Determination of rolling resistance of belt conveyors using rubber data: Fact or fiction? *Bulk Solids Handl.* **2003**, *23*, 384–391.
37. Loeffler, F.J.; Gullledge, J.D., Jr. *Pipe/Tube Conveyors A Modern Method of Coal and Ash Transportation*; Pennwell Conferences & Exhibitions: Houston, TX, USA, 1995.
38. *ISO 18766:2014; Rubber, Vulcanized or Thermoplastic—Low-Temperature Testing—General Introduction and Guide*. ISO: Geneva, Switzerland, 2014.
39. Andrejiova, M.; Grincova, A.; Marasova, D. Measurement and simulation of impact wear damage to industrial conveyor belts. *Wear* **2016**, *368–369*, 400–407. [\[CrossRef\]](#)
40. Hakami, F.; Pramanik, A.; Basak, A.K. Wear of rubber material in conveyor belt system. In *Tribology of Elastomers*; Springer Nature Singapore: Singapore, 2022; pp. 23–51. [\[CrossRef\]](#)
41. Mars, W.V.; Fatemi, A. Factors that affect the fatigue life of rubber: A literature survey. *Rubber Chem. Technol.* **2004**, *77*, 391–412. [\[CrossRef\]](#)
42. Kang, W.; Liu, J.; Xiong, W.; You, T.; Wang, X.; Zeng, K.; Deng, Y.; Guo, Z.; Yuan, K. Basic mechanical and fatigue properties of rubber materials and components for railway vehicles: A literature survey. *Rev. Adv. Mater. Sci.* **2022**, *61*, 587–610. [\[CrossRef\]](#)
43. Woo, C.S.; Park, H.S. Useful lifetime prediction of rubber component. *Eng. Fail. Anal.* **2011**, *18*, 1645–1651. [\[CrossRef\]](#)
44. Brown, R.P.; Forrest, M.J.; Soulagnet, G. *Long-Term and Accelerated Ageing Tests on Rubbers*; Rapra Technology Limited: Shrewsbury, UK, 2000.
45. *ISO 703:2017; Conveyor Belts—Transverse Flexibility (Troughability)*. ISO: Geneva, Switzerland, 2017.
46. *ISO 188:2023; Rubber, Vulcanized, or Thermoplastic—Accelerated Aging and Heat Resistance Tests*. ISO: Geneva, Switzerland, 2023.
47. *ISO 6943:2017; Rubber, Vulcanized—Determination of Tension Fatigue*. ISO: Geneva, Switzerland, 2017.
48. *ISO 27727:2008; Rubber, Vulcanized—Measurement of Fatigue Crack Growth Rate*. ISO: Geneva, Switzerland, 2008.

49. Carr, M.J.; Roberts, A.W.; Wheeler, C.A. A revised methodology for the determination of bulk material cohesion and adhesion. *Adv. Powder Technol.* **2019**, *30*, 2110–2116. [\[CrossRef\]](#)
50. Ganesan, V.; Rosentrater, K.A.; Muthukumarappan, K. Flowability and handling characteristics of bulk solids and powders—A review with implications for DDGS. *Biosyst. Eng.* **2008**, *101*, 425–435. [\[CrossRef\]](#)
51. Johanson, J.R. Know Your Material—How to Predict and Use the Properties of Bulk Solids. *Chem. Eng.* **1978**, *85*, 9–17.
52. Craik, D.J.; Miller, B.F. The flow properties of powders under humid conditions. *J. Pharm. Pharmacol.* **1958**, *10* (Suppl. 1), 136T–144T. [\[CrossRef\]](#) [\[PubMed\]](#)
53. Moreyra, R.; Peleg, M. Effect of equilibrium water activity on the bulk properties of selected food powders. *J. Food Sci.* **1981**, *46*, 1918–1922. [\[CrossRef\]](#)
54. Schulze, D. *Powders and Bulk Solids*; Springer International Publishing: Cham, Switzerland, 2021. [\[CrossRef\]](#)
55. Frew, I.; Wypych, P.W.; Mar, L. *Different Modes of Dust Testing for Bulk Solids*; University of Newcastle: Callaghan, NSW, Australia, 2013.
56. Chen, X.L.; Wheeler, C.A.; Donohue, T.J.; McLean, R.; Roberts, A.W. Evaluation of dust emissions from conveyor transfer chutes using experimental and CFD simulation. *Int. J. Miner. Process.* **2012**, *110–111*, 101–108. [\[CrossRef\]](#)
57. Kesler, F.; Prener, M. DEM: Simulation of conveyor transfer chutes. *FME Trans.* **2009**, *37*, 185–192.
58. Ambrisko, L.; Marasova, D. Investigation of the ageing influence on the hardness of essential belt component. In Proceedings of the 18th International Multidisciplinary Scientific GeoConferences SGEM 2018, Albena, Bulgaria, 2–8 July 2018.
59. Harrison, A.; Roberts, A.W.; Eng, O.M. Trends in the Application of Troughed Conveyor Belts. In Proceedings of the International Materials Handling Conference, Johannesburg, South Africa; 1983.
60. Harrison, A. Propagation of longitudinal and transverse cracks on the covers of steel-cord belts. *Min. Technol.* **1982**, *64*, 738.
61. VDI 4438:2019; VDI, Entwurf Pipe Conveyor—Design Criteria, Calculation and Safety Instructions. Verein Deutscher Ingenieure: Düsseldorf, Germany, 2019; p. 4438.
62. JB/T10380; Pipe Belt Conveyor, 2nd ed. China Machinery Industry Federation (CMIF): Beijing, China, 2013.
63. Zhang, Y. Extended reach: Overland pipe conveyor with low rolling resistance belt. *Bulk Solids Handl.* **2013**, *33*, 16–21.
64. Minikin, A.; Dilffeld, M.; Fischer, W. Residuals removal at Maritza E1—Design, installation and commissioning of a tube belt conveyor. *Bulk Solids Handl.* **2014**, *34*.
65. Kulagin, D.S. Establishment of bending radius of the tubular conveyor in the horizontal plane by modelling on the computer, Mining Informational and Analytical Bulletin. *Sci. Tech. J.* **2009**, *16*, 114–129.
66. Andrianov, I.V.; Van Horssen, W.T. On the transversal vibrations of a conveyor belt: Applicability of simplified models. *J. Sound Vib.* **2008**, *313*, 822–829. [\[CrossRef\]](#)
67. Suweken, G.; Van Horssen, W.T. On the transversal vibrations of a conveyor belt with a low and time-varying velocity. Part II: The beam-like case. *J. Sound Vib.* **2003**, *267*, 1007–1027. [\[CrossRef\]](#)
68. Suweken, G.; Van Horssen, W.T. On the transversal vibrations of a conveyor belt with a low and time-varying velocity. Part I: The string-like case. *J. Sound Vib.* **2003**, *264*, 117–133. [\[CrossRef\]](#)
69. Efimov, M.S. Determination of the restoring moments from the action of the load during the rotational movement of the tubular conveyor belt. *Min. Inf. Anal. Bull. Sci. Tech. J.* **2007**, *11*, 307–309.
70. DIN 22112-1; Belt Conveyors for Underground Coal Mining—Idlers—Part 1: Dimensions. DIN: Berlin Germany, 2010.
71. Zamiralova, M.E.; Lodewijks, G. Energy consumption of pipe belt conveyors: Indentation rolling resistance. *FME Trans.* **2012**, *40*, 171–176.
72. Wheeler, C.; Munzenberger, P.; Ausling, D.; Beh, B. “Energy Efficient Belt Conveyor Design” *Tunra Bulk Solids Research Associates*; The University of Newcastle: Callaghan, NSW, Australia, 2018; p. 27.
73. Neubecker, I. An overland pipe conveyor with 22 horizontal and 45 vertical curves connecting coal mine with rail load out. *Bulk Solids Handling* **1998**, *18*, 457–462.
74. Ilanković, N.; Živanić, D.; Zuber, N. The influence of fatigue loading on the durability of the conveyor belt. *Appl. Sci.* **2023**, *13*, 3277. [\[CrossRef\]](#)
75. Barburski, M. Analysis of the pipe conveyor belt pressure on the rollers on its circuit. *J. Ind. Text.* **2016**, *45*, 1619–1634. [\[CrossRef\]](#)
76. Weigang, S.; Hongbo, J. Research Progress and Design Calculation Method of Tubular Belt Conveyor. *Chin. J. Eng. Des.* **2018**, *25*, 1–11.
77. Guo, Y.-C.; Wang, S.; Hu, K.; Li, D. Optimization and experimental study of transport section lateral pressure of pipe belt conveyor. *Adv. Powder Technol.* **2016**, *27*, 1318–1324. [\[CrossRef\]](#)
78. Haak, D. *Overview and Characteristics of Pipe Conveyor Systems and Applications—Een Overzicht En de Karakteristieken van Buisvormige Transportbanden En de Toepassing Daarvan*; Delft University of Technology: Delft, The Netherlands, 2017.
79. Zamiralova, M.E.; Lodewijks, G. Review of the troughability test ISO 703 for quantifying a uniform transverse bending stiffness for conveyor belts. *Arch. Civ. Mech. Eng.* **2017**, *17*, 249–270. [\[CrossRef\]](#)
80. Fedorko, G.; Molnár, V.; Živčák, J.; Dovica, M.; Husáková, N. Failure analysis of textile rubber conveyor belt damaged by dynamic wear. *Eng. Fail. Anal.* **2013**, *28*, 103–114. [\[CrossRef\]](#)
81. Kozhushko, G.G.; Kopnov, V.A. Fatigue strength functions in shear loading of fabric conveyor belts. *Int. J. Fatigue* **1995**, *17*, 539–544. [\[CrossRef\]](#)
82. E.N. Din 16974:2016; Conveyor Belts: IRR Related to Belt Width—Requirements, Testing. DIN: Berlin, Germany, 2016.

83. AS 1334.13:2017; Methods of Testing Conveyor and Elevator Belting Determination of Indentation Rolling Resistance of Conveyor Belting. Standards Australia: Sydney, NSW, Australia, 2017.
84. Indrajati, I.N.; Setyorini, I. Service life prediction of rubber belt conveyor using kinetics approach. *AIP Conf. Proc.* **2020**, *2197*, 090002. [\[CrossRef\]](#)
85. ISO 5285:2012; Conveyor Belts—Guidelines for Storage and Handling. ISO: Geneva, Switzerland, 2012.
86. Lemmi, T.S.; Barburski, M.; Kabzinski, A.; Frukacz, K. Effect of vulcanization process parameters on the tensile strength of carcass of textile-rubber reinforced conveyor belts. *Materials* **2021**, *14*, 7552. [\[CrossRef\]](#)
87. Dobrotă, D. Vulcanization of rubber conveyor belts with metallic insertion using ultrasounds. *Procedia Eng.* **2015**, *100*, 1160–1166. [\[CrossRef\]](#)
88. Minkin, A.; Jungk, A.; Hontscha, T. Design, replacement, and power measurements for the Long Distance pipe conveyor belt at western arch coal skyline Mine, USA. *AMS-Online* **2013**, *2*, 16–25.
89. AMinkin; Jungk, A.; Hontscha, T. Belt replacement at a long distance pipe conveyor: Belt design, installation and power measurements. *Bulk Solids Handl.* **2012**, *32*, 16–21.
90. Cherednik, P.N. Setting the critical value of the belt tension when braking a tubular conveyor. *Min. Inf. Anal. Bull. Sci. Tech. J.* **2016**, *12*, 213–219.
91. Dmitriev, V.G.; Sergeeva, N.V. Traction calculation of tubular belt conveyors. *Min. Inf. Anal. Bull. Sci. Tech. J.* **2009**, *16*, 144–169.
92. Michalik, P.; Zajac, J. Using of computer integrated system for static tests of pipe conveyor belts. In Proceedings of the 13th International Carpathian Control Conference (ICCC), High Tatras, Slovakia, 28–31 May 2012; IEEE Publications: Piscataway, NJ, USA, 2012; pp. 480–485. [\[CrossRef\]](#)
93. Jennings, A.; Perrone, P.; Cornet, J. Case study: Correcting control problems on Essroc’s multidrive station, horizontally curved conveyor. *Trans. Soc. Min. Metall. Explor.* **2013**, *334*, 472–476.
94. Lodewijks, G. Two decades dynamics of belt conveyor systems. *Bulk Solids Handl.* **2002**, *22*, 124–132.
95. Kruse, D.J. Optimizing conveyor take-up systems using dynamic analysis and the implementation of capstans. In *Bulk Material Handling by Conveyor SME Conference*; Society for Mining, Metallurgy and Exploration: Littleton, Colorado, USA, 2002.
96. Staples, P. The history of pipe conveyors. *Bulk Solids Handl.* **2002**, *22*, 210–213.
97. Drenth, K.F. The rotation of pipe conveyors Bulk Solids India Conference paper, Nehru Centre, Mumbai, 2010, pp. 1–6. Advisor: Lodewijks, G. Report no. 50303. Maritime Archive, Mechanical, Maritime and Materials Engineering, Marine and Transport Technology, Transport Engineering and Logistics. In Proceedings of the BulkSolids India 2011, the International Conference & Exhibition on Storing, Handling and Transporting Bulk Solids and Powders, Mumbai, India, 2011.
98. Fedorko, G.; Ivančo, V.; Molnár, V.; Husáková, N. Simulation of interaction of a pipe conveyor belt with moulding rolls. *Procedia Eng.* **2012**, *48*, 129–134. [\[CrossRef\]](#)
99. Molnár, V.; Fedorko, G.; Stehlíková, B.; Michalik, P. Analysis of a pipe conveyor’s idler housing failure due to a missing roller in terms of contact forces. *Eng. Fail. Anal.* **2021**, *127*, 105527. [\[CrossRef\]](#)
100. (PN-M) 46606; Przenośniki Taśmowe—Krażnikit, 2nd ed. Polski Komitet Normalizacyjny: Warsaw, Poland, 2010.
101. (SANS) 1313–3; Conveyor Belt Idlers—Part 3: Performance Specifications for Troughed Belt Conveyor Idlers—Metallic and Non-metallic—For Idler Roller Rotational Speeds of up to 750 Revolutions per Minute, 1st ed. South African National Standards: Cape Town, South Africa, 2012.
102. DIN 22112-3:2022-12; Belt Conveyors for Underground Coal Mining—Idlers—Part 3: Testing. DIN: Berlin Germany, 2022.
103. Fedorko, G.; Molnár, V.; Vasil’, M.; Salai, R. Proposal of digital twin for testing and measuring of transport belts for pipe conveyors within the concept Industry 4.0. *Measurement* **2021**, *174*, 108978. [\[CrossRef\]](#)
104. Carter, R.A. New conveyor tech: Designing for productivity. *Eng. Min. J.* **2021**, *222*, 44–47.
105. Nuttall, A.J.G. *Design Aspects of Multiple Driven Belt Conveyors*; TRAIL Research School: Delft, The Netherlands, 2007.
106. Budynas, R.G.; Nisbett, J.K. *Shigley’s Mechanical Engineering Design*, 10th ed.; McGraw-Hill Education: New York, NY, USA, 2015; p. 1082.
107. Gabriel, P. Investigation and Modelling of Rubber Friction. Ph.D. Thesis, Queen Mary University of London, London, UK, 2010.
108. Maegawa, S.; Itoigawa, F.; Nakamura, T. Effect of normal load on friction coefficient for sliding contact between rough rubber surface and rigid smooth plane. *Tribol. Int.* **2015**, *92*, 335–343. [\[CrossRef\]](#)
109. Persson, B.N.J. Theory of rubber friction and contact mechanics. *J. Chem. Phys.* **2001**, *115*, 3840–3861. [\[CrossRef\]](#)
110. Yang, C.; Persson, B.N.J. Contact mechanics: Contact area and interfacial separation from small contact to full contact. *J. Phys. Condens. Matter* **2008**, *20*, 215214. [\[CrossRef\]](#)
111. Klüppel, M.; Heinrich, G. Rubber friction on self-affine road tracks. *Rubber Chem. Technol.* **2000**, *73*, 578–606. [\[CrossRef\]](#)
112. Wesemeier, M. Theoretical investigations of the forming force on pipe belt conveyor. *Bulk Mater.* **1997**, *3*, 29.
113. Chernenko, V.D. *Calculation of Continuous Transport Systems*. Politekhnik; Politekhnik: Saint Petersburg, Russia, 2008.
114. Bhushan, B. *Introduction to Tribology*; John Wiley & Sons: Hoboken, NJ, USA, 2013. [\[CrossRef\]](#)
115. Ciavarella, M. A simplified version of Persson’s multiscale theory for rubber friction due to viscoelastic losses. *J. Tribol.* **2018**, *140*, 011403. [\[CrossRef\]](#)
116. Sundararajan, C.R. *Probabilistic Structural Mechanics Handbook: Theory and Industrial Applications*; Springer Science+Business Media: Berlin/Heidelberg, Germany, 1995.

117. Wand, M.P.; Jones, M.C. *Smoothing*, 1st ed.; Chapman & Hall: New York, NY, USA; CRC Press: Boca Raton, FL, USA, 1994. [\[CrossRef\]](#)
118. Chacón, J.E.; Duong, T. *Multivariate Kernel Smoothing and Its Applications*; CRC Press: Boca Raton, FL, USA, 2018.
119. *ISO/TR 7620:2005*; Rubber materials—Chemical resistance. ISO: Geneva, Switzerland, 2005.
120. Klippel, E.; Oliveira, R.A.R.; Maslov, D.; Bianchi, A.G.C.; Delabrida, S.E.; Garrocho, C.T.B. Embedded edge artificial intelligence for longitudinal rip detection in conveyor belt applied at the industrial mining environment. *SN Comput. Sci.* **2022**, *3*, 280. [\[CrossRef\]](#)

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