

STRUCTURAL OPTIMIZATION AND WEIGHT REDUCTION OF A BELT CONVEYOR SUPPORTING FRAME USING ANALYTICAL AND FINITE ELEMENT METHODS

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Abstract:

This paper deals with the structural design of a belt conveyor equipped with a handling mechanism. The research was carried out within the framework of the KEGA 004TU Z-4/2024 and VEGA1/0302/26 projects and is focused on the design and analysis of the supporting structure in terms of its strength and operational reliability. Particular attention is paid to the methodological procedure for determining the distribution of individual loads arising at critical points of the structure during the operation of the conveyor and the handling mechanism. The paper also identifies the decisive loading conditions and their influence on the dimensioning of the supporting elements of the device. The obtained results contribute to the optimization of the structural design and to increasing the operational safety of the proposed system. The structure was found to be significantly overdesigned with a safety factor of 41.8. A subsequent optimization reduced the weight, while maintaining a safe stress level.

Key words: conveyor; construction; manipulation; transport

INTRODUCTION

Belt conveyors are among the most widely used devices in the field of material handling, automated production systems, and logistics processes. Their importance lies mainly in their ability to ensure continuous, efficient, and reliable transportation of materials at relatively low operating costs.

Before the actual construction of the conveyor, the researcher performs design and calculation steps using the Finite Element Method (FEM) and Finite Element Analysis (FEA), including belt calculation, torque calculation, motor power calculation, gearbox design, shaft design, key design, bearing selection, and chain drive design [1]. As early as the 19th century, the inventor of US Patent No. 3,633, granted on June 15, 1844 to a man named Charles Goodyear, brought new possibilities for belt conveyors as well through his invention of vulcanized rubber [2, 3].

The advantage of the modular solution is the possibility of strengthening the structure by combining individual elements, thereby increasing operational safety and cost efficiency [4]. Handling equipment is used to move objects from one place to another (e.g. between workplaces, between a loading ramp and a storage area) [5].

In current industrial applications, belt conveyors are often supplemented with various handling and sorting mechanisms that enable the automation of sorting and distribution processes for transported objects [6]. However, the integration of sorting mechanisms imposes increased requirements on the design of the supporting structure, its strength, stiffness, and overall operational reliability [7]. In industrial practice, the integration of a belt conveyor with a sorting mechanism, as described in this

study, finds critical applications across various high-demand sectors. Specifically, this system is uniquely suitable for automated logistics and distribution hubs (e.g. e-commerce fulfillment centers), where Euro containers must be continuously routed based on barcodes or RFID (*Radio Frequency Identification*) tags [8]. Another prominent application is in the automotive assembly lines, where optimized sub-assemblies or component bins are distributed to specific workstations. Furthermore, in the food and packaging industry, this rigid yet lightweight structure ensures reliable high-speed sorting of standardized crates containing perishable goods, where operational downtime due to structural fatigue or deflection would cause significant economic losses. By understanding the exact localized static and dynamic loads of the sorting units, manufacturing plants can deploy these modular conveyor sub-systems directly into existing production cells without risks of unexpected structural failure. When designing belt conveyors, it is necessary to consider not only static loads caused by the weight of the structure and the transported material, but also dynamic effects generated during the operation of the equipment. These effects may lead to increased stresses, vibrations, and local stress concentrations in critical parts of the structure, which can reduce the service life of individual components or result in structural failures. Therefore, modern approaches employ numerical methods, particularly the Finite Element Method (FEM), which enables detailed analysis of stress and deformation states under various operating conditions [9, 10].

CAD software tools also play an important role in the design process, as they enable the creation of parametric 3D structural models and their subsequent integration with FEM analyses. The use of CAD/CAE systems allows effective identification of critical structural locations, optimization of supporting element dimensions, and minimization of operational failure risks already during the design phase. Current research highlights the growing importance of digital modeling and numerical simulations in the design of conveyor systems and their structural components [11]. According to Doghri [12], it is important to support structural design with calculations based on linear elasticity, structural mechanics, and nonlinear mechanics, including computational algorithms. However, in calculations it is essential to correctly define the materials used. Supplier catalogs list many materials and dimensions; however, not all of them are readily available, and shortages frequently occur [13]. Therefore, in structural design it is necessary to use additional sources and work, for example, with mechanical engineering tables as well [14]. The authors Zhou et al. [15] indicate that the conveyor belt's warpage increases as roller spacing decreases nonlinearly with tension, and increases nonlinearly with the transverse stiffness of the belt. The presented paper focuses on the load analysis of a belt conveyor equipped with a sorting mechanism. The main objective is to determine the distribution of individual loads acting on the supporting structure of the device and to identify critical locations from the perspective of stress loading. According to Bortnowski et al. [16], the transverse vibration of conveyor belts is a key aspect of their proper and safe design, as the accurate determination of vibration frequencies is essential to prevent hazardous operating conditions and premature wear of components. Other authors, on the other hand, place emphasis on the outputs of the simulation model, and the data obtained from them serve for better maintenance planning of conveyors, reduction of belt and idler roller wear, and overall for increasing the reliability and safety of material flow in industrial enterprises [17].

The paper also presents a methodological procedure for the application of CAD modeling and numerical FEM analyses in the design and verification of the conveyor structural solution. The obtained results provide a basis for structural optimization and for improving the operational safety and reliability of the proposed system. To achieve these objectives, a comprehensive 3D CAD model of the complex conveyor system was developed. The overall assembly, including the driving units and the sorting mechanism layout, is illustrated in Fig. 1. Although conveyors are common, the integration of additional mechanisms often leads to over-dimensioning of structures due to a lack of precise combined analyses. Traditional, conservative design approaches in conveyor engineering frequently result in excessive material consumption and low optimization efficiency. Recent studies address this challenge by introducing meta-heuristic optimization algorithms – such as the Harris hawks optimization method – combined

with sensitivity analysis, achieving significant structural mass reductions of up to 19.21% while eliminating the inefficiencies of conservative design [18].

METHODOLOGY OF RESEARCH

The structural design of a belt conveyor equipped with a sorting mechanism (Fig. 1) must be subjected to load testing, which can be performed using several approaches.

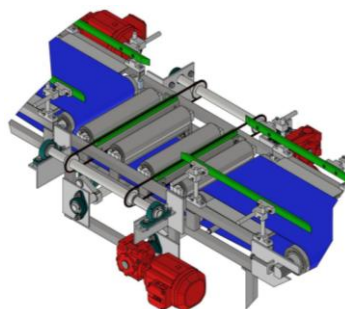


Fig. 1 Caption: The model of the belt conveyor with sorting mechanism

Alt text: Isometric CAD assembly: grey frame with parallel rollers, blue belt at both ends, green guide rails and two red gear motor units.

Long description: Rendered CAD assembly of the conveyor viewed from an elevated angle. Grey sheet-metal side plates carry a row of parallel rollers; a blue belt wraps around the drums at both ends of the unit. Several narrow green guide rails run along the top. Two red gear-motor units are attached – one on a bracket at the top left, one underneath the frame at the bottom center – with dark bearing blocks and small silver brackets on the front shafts.

From the methodological point of view, the most relevant methods for this research are analytical calculations and simulation analysis. In order to achieve the most relevant and accurate results possible, the designed structure was subjected to both types of loading analyses, and the obtained results were subsequently compared. The most reliable results could be achieved by performing real load tests on a physical prototype of the structure. However, even the comparison of theoretical and simulation analyses provides valuable results that can serve as a solid basis for further research and structural optimization.

The static loading of the conveyor belt supporting structure was determined as the load due to the self-weight of the structure, the weight of the conveyor belt and the conveyed material, as well as the weight and operational load of the sorting mechanism. The loading includes discrete concentrated forces acting at the attachment points of the sorting mechanism. The computational model was developed in accordance with the relevant standard and verified using the finite element method.

Supporting Structure Design

During the design of the belt conveyor supporting structure, it was necessary to determine the loading effects acting on the frame structure. The load generated by the transported Euro containers was idealized within the computational model as a system of concentrated forces applied at their assumed positions on the conveyor. Based on the developed static model, the support reactions, internal force distributions, and bending moment diagrams within the individual load-bearing members were determined. The obtained results subsequently served as the basis for the strength assessment and dimensioning of the conveyor supporting structure. Furthermore, the results of the analytical calculations were used as input data for a numerical analysis performed using the Finite Element Method (FEM), which enabled a more detailed evaluation of the stress-strain state of the structure and the identification of critical regions with respect to its strength and stiffness.

The load acting on the supporting structure of the belt conveyor due to the weight of the transported Euro containers was idealized in the analytical calculation as a system of six concentrated forces. The individual point loads were applied at six predefined locations situated along the axis of the struc-

tural profile, thereby creating a simplified computational model representing the actual operational loading of the structure. The design load originating from the transported Euro containers was simplified into a localized static system. The spatial distribution of the concentrated forces and the corresponding ideal support reactions are graphically represented in Fig. 2 and Fig. 3, respectively.

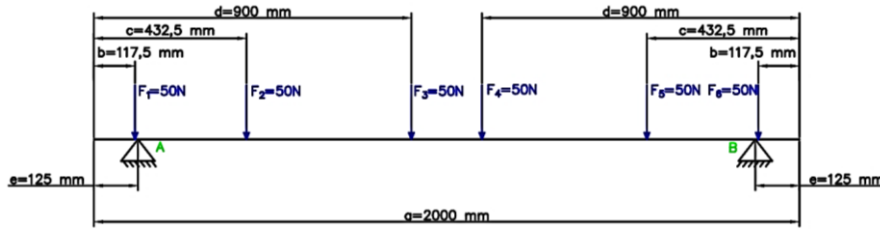


Fig. 2 Caption: Graphical Representation of Forces Acting on the Frame Structure

Alt text: Line sketch of a simply supported beam with six equal downward force arrows, F_1 to F_6 , spaced along the span; dimension lines below.

Long description: AutoCAD sketch drawing of a horizontal beam 2000 mm long with hatched pin supports labelled A and B, each inset $e = 125$ mm from the beam ends. Six blue downward arrows labelled $F_1 = 50$ N to $F_6 = 50$ N act on the beam. Dimension lines locate the forces symmetrically from both ends of the beam: $b = 117.5$ mm, $c = 432.5$ mm and $d = 900$ mm.

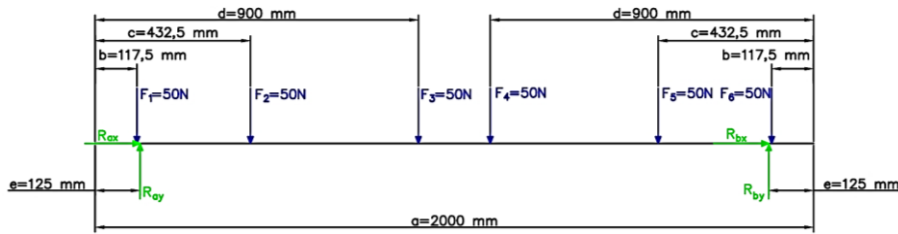


Fig. 3 Caption: Graphical Representation of Support Reactions in the Frame Structure

Alt text: The same beam sketch with two upward reaction arrows at the end supports A and B, balancing the six downward point loads.

Long description: The same 2000 mm beam drawing with the six blue 50 N force arrows. Green reaction arrows are added at both supports: vertical arrows pointing upwards labelled R_{ay} and R_{by} and horizontal arrows labelled R_{ax} and R_{bx} ; numerical reaction values are not printed. Dimension lines repeat $a = 2000$ mm, $e = 125$ mm and the force spacing b , c and d .

The load applied to the frame structure was calculated according to the following equation:

$$F_{celk} = n \cdot F = 6 \cdot 50 = 300 \text{ N} \quad (1)$$

where:

F_{celk} [N] – total load acting on the frame structure;

n [pcs] – number of applied forces;

$$n = 6$$

F [N] – magnitude of a single concentrated force;

$$F = 50 \text{ N}$$

For a symmetrically loaded structure, the support reactions were determined according to the following equation:

$$R_{ay} = R_{by} = \frac{F_{celk}}{2} = \frac{300}{2} = 150 \text{ N} \quad (2)$$

where:

R_{ay} [N] – reaction force at the left support;

R_{by} [N] – reaction force at the right support.

The maximum bending moment was calculated at the section located directly beneath the force F_3 . Due to the symmetrical distribution of both the applied loads and the support reactions, an identical bending moment value is achieved at the section located beneath the force F_4 .

$$M_{max} = R_{ay} \cdot (d - e) - F_1 \cdot (d - b) - F_2 \cdot (d - c) \text{ [Nm]} \quad (3)$$

$$M_{max} = 150 \cdot (900 - 125) - 50 \cdot (900 - 117.5) - 50 \cdot (900 - 432.5) = 53750 \text{ Nmm} = 53.75 \text{ Nm}$$

where:

$M_{\max}$ [Nm] – maximum bending moment;

F_1 [N] – concentrated force acting on the beam;

$$F_1 = 50 \text{ N}$$

F_2 [N] – concentrated force acting on the beam;

$$F_2 = 50 \text{ N}$$

b [mm] – distance of force F_1 from the edge of the beam;

$$b = 117.5 \text{ mm}$$

c [mm] – distance of force F_2 from the edge of the beam;

$$c = 432.5 \text{ mm}$$

d [mm] – distance of force F_3 from the edge of the beam;

$$d = 900 \text{ mm}$$

e [mm] – distance from support A to the edge of the beam;

$$e = 125 \text{ mm}$$

R_{ay} [N] – vertical reaction force at the left support;

$$R_{ay} = 150 \text{ N}$$

The second moment of area of the TR 50 × 35 × 3 mm profile was calculated using the following equation:

$$I_o = \frac{B \cdot H^3 - b \cdot h^3}{12} = \frac{35 \cdot 50^3 - 29 \cdot 44^3}{12} = 158\,171 \text{ mm}^4 \quad (4)$$

where:

I_o [mm⁴] – second moment of area of the profile;

B [mm] – outer width of the profile;

$$B = 35 \text{ mm}$$

H [mm] – outer height of the profile;

$$H = 50 \text{ mm}$$

b [mm] – inner width of the profile;

$$b = 29 \text{ mm}$$

h [mm] – inner height of the profile;

$$h = 44 \text{ mm}$$

Based on the analytical calculation of the continuous internal forces, the bending moment diagram was constructed. The distribution of the bending moments along the longitudinal axis of the beam, highlighting the maximum value in the midspan, is shown in Fig. 4.

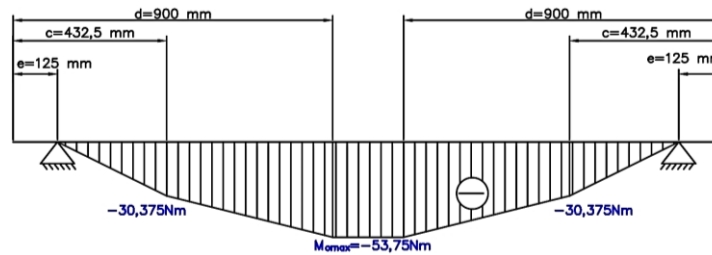


Fig. 4 Caption: Graphical representation of bending moments acting on the frame structure

Alt text: Beam sketch with the moment diagram below it: values rise piecewise-linearly from zero at the supports to a peak at midspan.

Long description: Bending-moment diagram drawn as a hatched polygon below the beam axis between the two supports. The moment is zero at each support and grows piecewise-linearly towards the center: -30.375 Nm is labelled on both branches and the extreme $M_{\max} = -53.75$ Nm lies on the flat central segment between the two middle forces. A circled minus sign marks the sign convention; dimension lines $d = 900$ mm, $c = 432.5$ mm and $e = 125$ mm are repeated above.

The section modulus of the selected profile was determined according to the following equation:

$$W_o = \frac{I_o}{y_{\max}} = \frac{158\,171}{25} = 6\,327 \text{ mm}^3 \quad (5)$$

where:

W_o [mm³] – section modulus of the profile;

$y_{\max}$ [mm] – distance from the neutral axis to the extreme fiber;

$$y_{\max} = 25 \text{ mm}$$

The bending stress was calculated using the following equation:

$$\sigma_{\text{red}} \approx \sigma_o = \frac{M_{\max}}{W_o} = \frac{53\,750}{6\,327} = 8.49 \text{ MPa} \quad (6)$$

where:

σ_{red} [MPa] – reduced stress;

σ_o [MPa] – bending stress;

$M_{\max}$ [Nmm] – maximum bending moment;

W_o [mm³] – section modulus.

For steel S355, the yield strength is $R_e = 355$ MPa

Permissible stress with a safety factor: $k = 2$:

$$\sigma_{dov} = \frac{R_e}{k} = \frac{355}{2} = 177.5 \text{ MPa} \quad (7)$$

Profile check:

$$\begin{aligned} \sigma_{red} &\leq \sigma_{dov} \\ 8.49 &\leq 177.5 \text{ MPa} \end{aligned}$$

The selected TR 50 × 35 × 3 mm profile made of S355 steel is satisfactory.

Safety factor with respect to yield strength:

$$k = \frac{R_e}{\sigma_{red}} = \frac{355}{8.49} = 41.81 \quad (8)$$

The initial analytical check of the standard TR 50 × 35 × 3 mm profile revealed a reduced stress of 8.49 MPa (Eq. 6). Compared to the permissible stress of 177.5 MPa (Eq. 7), the resulting safety factor with respect to the yield strength reaches an excessively high value of $k = 41.81$ (Eq. 8). Such a high safety factor indicates significant material overdesign, which provides an opportunity for subsequent structural optimization and weight reduction.

RESULTS

The applied methodology was based on a combination of analytical calculations of the loading of the frame structure and its subsequent verification using the finite element method (FEM). Based on the defined loading consisting of six point forces with a magnitude of 50 N, the support reactions were determined, and the distribution of internal force effects in the beam was evaluated.

The maximum bending moment was identified in the midspan region of the beam, reaching a value of $M_{\max} = 53.75$ Nm. Subsequently, a strength check of the selected TR 50 × 35 × 3 mm profile made of S355 steel was performed, and the results were compared with the numerical simulation.

The FEM analysis showed a maximum equivalent (von Mises) stress according to the HMM (Huber-Mises-Hencky) criterion of $\sigma_{red, \max} = 9.816$ MPa. The exact locations of these stress concentrations, which primarily occur in the vicinity of the beam attachments, are detailed in the numerical simulation plot in Fig. 5.

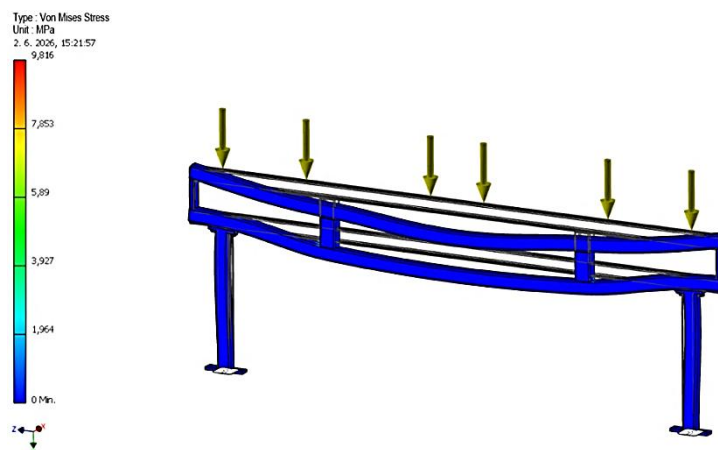


Fig. 5 Caption: Graphical representation of the stresses acting on the frame construction belt conveyor

Alt text: Color contour plot of the frame under six load arrows; the structure appears uniformly dark blue on a stress scale from 0 to 9.816 MPa.

Long description: Isometric finite-element plot titled Von Mises Stress, unit MPa. The color legend runs from 0 MPa (blue, minimum) to 9.816 MPa (red) with intermediate marks at 1.964, 3.927, 5.89 and 7.853 MPa. Six olive arrows press down on the top beam of the two-legged welded frame. At this scale the entire structure appears uniformly dark blue, i.e. close to the minimum; no yellow or red regions are visible in the plot.

At the same time, the maximum deflection of the structure was determined to be $w_{\max} = 0.047$ mm, occurring in the midspan of the conveyor, where the highest bending stresses are expected. The resulting deformation behavior and the overall global deflection fields of the analyzed framework under the full design load are visualized in Fig. 6.

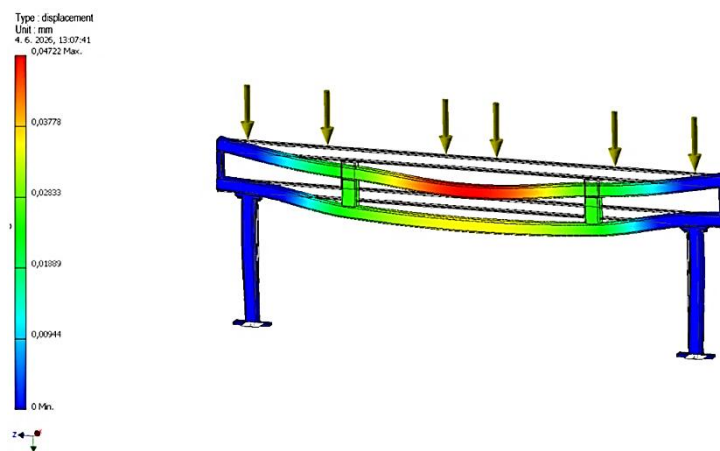


Fig. 6 Caption: Graphical representation of the deflection of the belt conveyor supporting structure

Alt text: Contour plot of the frame: legs deep blue, colors grading to red at midspan; displacement scale from 0 to 0.047 mm

Long description: Isometric finite-element plot titled displacement, unit mm, of the same frame under the six arrows. The legend runs from 0 mm (blue, minimum) to 0.04722 mm (red, maximum) with intermediate marks at 0.00944, 0.01889, 0.02833 and 0.03778 mm. The legs and beam ends are deep blue; colors grade through green and yellow to a red zone at the midspan of the horizontal beams, where the largest deflection occurs

Based on the comparison of analytical and simulation results, it can be concluded that the designed structure exhibits sufficient strength and stiffness under the considered static loading. The stress values are significantly below the yield strength of S355 steel, confirming an adequate safety margin of the design.

DISCUSSION

The designed belt conveyor structure was verified using a combination of analytical calculations and numerical validation based on the finite element method (FEM). In the analytical solution, the maximum bending moment was determined as $M_{\max} = 53.75$ Nm, with the critical region in terms of bending response identified at the midspan of the beam, where the maximum normal stresses due to bending occur.

Subsequently, a FEM analysis was performed in Autodesk Inventor Professional 2024, in which the material properties of S355 steel, boundary conditions, and loading corresponding to the operating conditions of the conveyor were defined. The setup of the model, application of constraints, and interpretation of results were carried out in accordance with recommended methodologies for numerical simulations in Autodesk Inventor [19].

The numerical analysis revealed a maximum equivalent stress according to the HMH (Huber-Mises-Hencky) criterion of $\sigma_{\text{red,max}} = 9.816$ MPa, occurring in regions of stress concentration associated with bending effects. This value is significantly lower than the yield strength of S355 steel, indicating a sufficient safety margin against plastic deformation.

At the same time, the maximum structural deflection was determined to be $w_{\max} = 0.047$ mm, representing a low level of global deformation and confirming adequate stiffness of the load-bearing system under the considered static loading.

Based on the correlation between analytical and numerical results, it can be concluded that the model shows good agreement in predicting the bending behavior of the structure, and that the design satisfies

both strength and stiffness requirements for the given operating conditions. A minor discrepancy of 15.6% was observed between the analytical bending stress (8.49 MPa) and the numerical von Mises stress (9.816 MPa). This variation is attributed to the local stress concentrations and the complex stiffness interaction of the welded frame components in the 3D FEM model, which cannot be fully captured by the simplified 1D Euler-Bernoulli beam theory used in the analytical phase. However, both methods firmly confirm that the structural integrity is fully preserved. Given the high margin of safety ($k > 40$), future steps should focus on eco-design and material optimization. Replacing the TR 50 × 35 × 3 profile with a thinner wall section (e.g., TR 50 × 35 × 1.5) or alternative aluminum alloys could lead to a weight reduction of up to 35%, significantly lower manufacturing costs, and a more efficient handling drive deployment.

CONCLUSION

This paper presented an integrated methodological procedure combining classical analytical calculations with numerical Finite Element Method (FEM) simulations to evaluate the supporting structure of a belt conveyor equipped with a sorting mechanism. The correlation between the 1D analytical beam model (8.49 MPa) [20, 21] and the 3D numerical simulation (9.816 MPa) demonstrated a minor and predictable discrepancy of 15.6%, confirming the structural integrity and reliability of the design under static operating loads. At the same time, the investigation confirmed that the selection and integration of a suitable sorting device based on defined operational requirements is a highly complex and demanding process, which strictly requires appropriate analytical procedures and optimization methods at all stages – from the initial design phase to the final implementation [22]. Since the initial layout using standard TR 50 × 35 × 3 mm steel profiles proved to be significantly over-dimensioned with an excessive safety factor of $k = 41.81$, this structural assessment shifts the focus of future research toward eco-design and weight reduction. Future work will therefore focus on geometric optimization of the framework wall thickness and the testing of alternative lightweight aluminum alloys. This systematic approach aims to achieve a potential weight reduction of up to 35%, thereby minimizing manufacturing costs and improving the overall dynamic efficiency of the integrated sorting automation. From a practical perspective, these future optimization steps aim to create a commercial prototype designed for smart factory environments operating under Industry 4.0 [23] standards. Reducing the weight by up to 35% will directly decrease the inertial forces during rapid start-stop cycles, which is a key requirement for energy-efficient sorting operations in logistics warehouses. The 35% reduction in structural weight directly contributes to enhanced energy efficiency. On a broader scale, this eco-design trend aligns with global efforts to reduce the carbon footprint of industrial enterprises. By transitioning to lightweight materials and implementing self-powered technologies, the need for battery replacement and the overall environmental impact associated with the operation of logistics centers are minimized [24]. Additionally, future research will include experimental testing of the optimized physical frame using strain gauge measurements during actual container sorting procedures. Integrating a combined numerical and experimental approach is critical for defining accurate boundary conditions and identifying potential damage zones under operational loads. By benchmarking finite element models against experimental data, the overall reliability of the framework design can be fully verified for practical industrial application [25]. This will validate the numerical safety margin in a real-world operating environment, providing logistics equipment manufacturers with a fully verified, eco-designed conveyor module ready for immediate factory-floor deployment.

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