Shape Stability of Pipe Belt Conveyors: From Throughability to Pipe-Ability

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This paper presents a new approach to determine the bending stiffness of a pipe conveyor belt that is sufficient to form a stable pipe shape based on its throughability performance. The paper describes the mathematical model that determines pipe conveyor contact forces and introduces two numerical models solved using FEM in ANSYS. Results agree with the experimental data obtained using a six-point stiffness device. The mathematical model proposed can be used as a uniform validation technique for any numerical model. Appearance of one of the contact forces that equals zero is considered as a criterion for insufficient bending stiffness of belt to form a stable pipe shape. Effective modulus of elasticity quantified from the throughability parameter becomes a link to express belt pipe-ability. Impact of belt line mass and bending stiffness is investigated: for the same belt geometry, heavier belts require higher bending stiffness for the correct pipe shape formation.

Keywords: *Pipe conveyor, bending stiffness, contact forces, experiment, test rig, Finite Element Method, Throughability test, pipe-ability*

1. INTRODUCTION

The pipe belt conveyor is a popular continuous transport system utilized in the bulk handling industry. Reliable operation of a system primarily depends on the belt's ability to form a stable pipe-shape geometry. The latter is mainly governed by lateral bending stiffness of the belt. If the belt is not sufficiently rigid, the tubular shape collapses and causes spillage of bulk material. This affects provision of well-sealed transport system and leads to the futility of pipe conveyor selection over other system type. Moreover, the collapsed belt tends to exhibit a larger twist in curves along the route, which results in a problematic tracking and alignment of the belt in operation. In contrast, if the belt is excessively rigid, it can cause pipe opening between the idler stations as well as an unnecessary increase of rolling resistance, which affects the overall energy consumption of the system. Obviously, for pipe conveyors, belt bending stiffness is an important parameter that needs to be controlled and carefully identified.

The only existing standardized procedure that in some way reflects belt bending stiffness in the lateral direction is a throughability test, described in standard ISO 703 [19]. Due to the simple test procedure, belt throughability has become a common parameter widely utilized in industry for expressing belt behaviour in bending. Standards [3, 11, 20] establish recommen–dations for minimum belt throughability required specifically for conventional open-trough belt conveyors.

For pipe conveyors, to date, there is no existing criteria that can regulate bending stiffness of the belt required to form a well-sealed pipe shape. For industrial application of pipe conveyor systems, it is convenient to express that pipe-ability measure via the belt throughability parameter, as the latter is widely used in practice and can be measured from a simple test.

In addition to belt bending stiffness, a number of other physical parameters are involved (e.g., belt line mass, width, thickness, pipe diameter, etc.) that can also influence belt behaviour in bending. In this case, the impact of the physical parameters on pipe-ability of the belt must be also considered in combination.

The aim of the study is to develop an approach that can determine pipe conveyor belt bending stiffness that is sufficient to keep a stable pipe shape and express that parameter via belt throughability performance. In this case it will be sufficient to perform a simple throughability test to predict the belt's behaviour in the pipe conveyor system.

2. METHODS

To fulfil the research goal assigned, it is important to signify belt pipe-ability. In the present study, lateral belt flexibility is assumed to be sufficient for correct pipe shape formation, when the conveyor belt, folded in a tubular shape, contacts all six supporting idler rolls situated hexagonally (see Figure 1a). Consequently, pipe-ability can be identified by appearance of contact loss, when one or more of the contact forces becomes equal to zero (Figure 1b). It is important to mention that this pipe-ability expression is limited to 2D behaviour of belt and does not incorporate the impact of belt tension and length of conveyor pitch on appearance of pipe opening between the idler stations.

Obviously, load distribution between the idler rolls is a crucial parameter that needs to be correctly determined. For this purpose, a correct approach that quantifies contact forces must be selected. The

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appearance of one or more contact forces that equals zero establishes a critical value for belt bending stiffness, which in turn needs to be expressed via throughability the value

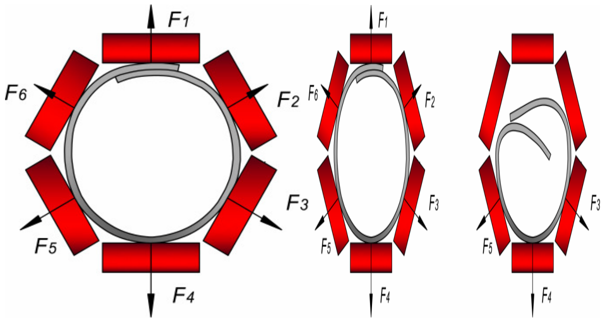

Figure 1. Pipe conveyor cross-section: a) correct pipe shape formation; b) belt collapse due to insufficient belt bending stiffness.

3. PIPE CONVEYOR CONTACT FORCES

To determine pipe conveyor contact forces three methodologies exist: an analytical approach that develops a mathematical model; a numerical solution, achieved within software; and an empirical experimentation.

The mathematical models that compute pipe conveyor contact forces are described in a number of studies (see Sergeeva [22], Dmitriev and Sergeeva [5], Kulagin [14], Dmitriev and Kulagin [4], Gładysiewicz [8], Wesemeier [23-25], Wiedenroth [26], and others). As discussed by Zamiralova *et al.* [32] the existing analytical models assume certain simplifications that need to be reconsidered for better correlation with practical experience. Particularly, further assessment requires the description of the expansion load from forming a flat belt into a pipe shape and how the resultant contact forces are determined from the external loads involved.

Contact forces and corresponding belt deformations can be determined using numerical methods implemented in various software. Most frequently, the Finite Element Method (FEM) is employed. The models of Kulagin [15], Dmitriev and Kulagin [4], Shilling *et al.*[21], Fedorko *et al.* [6], Fedorko and Molnar [7], and Wesemeier [24, 26] are examples of that kind. The models differ in their computer simulation procedures, description of belt structure and boundary conditions, convergence criteria, etc. Obviously, these factors inevitably generate different results on pipe conveyor contact forces obtained, which raises an important requirement inherent to all analytical solutions.

Both mathematical and numerical approaches require experimental validation, as validation can indicate whether assumptions and methods used are relevant and whether results obtained correlate with practical experience. Usually, a well-validated numerical model can be used to obtain results beyond the limitations of the mathematical model.

Many studies have evaluated pipe conveyor contact forces through empirical experimentation (see Zamiralova and Lodewijks [27], Hötte [9], Hötte *et al.*[10], Wiedenroth [26], Molnar *et al.* [16-18], Bahke [1], etc.) Zamiralova and Lodewijks [28-30] provide an explicit analysis of the existing studies. The researchers indicate that the results essentially depend on the selection of the test rig design, and the contact forces attained from various test rigs significantly deviate. Moreover, higher complexity of the test rig configuration increases the possibility of the presence of uncontrolled measurement errors, such as incorrect position and misalignment of the measuring equipment, uncontrolled friction forces, etc.

This indicates that experimental results need to be accompanied by an analytical solution. Without any analytical knowledge, it becomes impossible to distinguish which experimental results are relevant and which are dramatically affected by the measurement errors. In addition, a study based only on empirical experimentation, provides only an approximate trend reflecting impact of various physical parameters on pipe conveyor contact forces.

To achieve assigned research goals, it is more appropriate to develop and utilize a mathematical model, as it can directly quantify belt bending stiffness, which causes the contact loss. For the validation of that model, the contact forces are compared with the results from the experiment. The test rig selected closely replicates the analytical problem and has a simple configuration to minimize the appearance of any measurement errors. In addition, the numerical solution within FEM-based software is also developed and compared with the experimental results.

3.1. Mathematical model

An appropriate mathematical model is developed utilizing the methodology introduced by Zamiralova *et al.* [32] as a basis. The primary difference from the present study appears from the selection of the statically indeterminate system and the expression of the load from belt bending stiffness.

The problem is linearized around the reference geometry. The latter is simplified to a circular open-structure with opening on the top (see Figure 2). The reference structure is subjected to external loads that generate resultant reaction contact forces F_1, F_2, F_3, F_4, F_5, F_6.

To replicate such load distribution between the idler rolls, as suggested by Zamiralova *et al.* [32], the contact points of belt with idler rolls are substituted by the movable hinge supports with one reaction force. From the symmetry of the structure, it is possible to consider only half of the cross section with fixed bottom edge (see Figure 3). As a result, the pipe conveyor contact forces in absolute values equal: $F_1 = 2F_1'$, $F_2 = F_6$, $F_3 = F_5$, $F_4 = 2F_4'$.

Zamiralova and Lodewijks [27] carried out the experimental tests, reporting that the pipe conveyor belt formed into a pipe shape without overlap experiences additional repulsion forces at the edges of the structure. To imitate this, the additional force N_1' is incorporated within the pinned support at the edge of the structure. This support was not incorporated in the previous study [32].

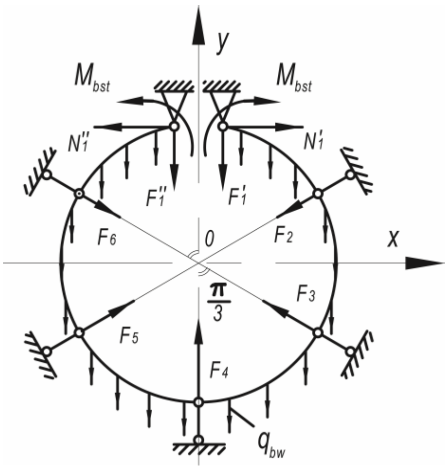

Figure 2. Statically indeterminate system that represents the symmetrical half of the cross-section. The load from belt bending stiffness is shown by a constant expansion moment M_{bst}.

The linearization implies that the complex nonlinear process of forming a flat belt into a pipe shape is substituted by an additional load from belt bending stiffness, applied onto the already pre-folded structure. Zamiralova *et al.* [32] modelled that load following recommendations of Chernenko [2]. This load is used in a number of analytical studies [4, 5, 14, 22-25] and represents an additional expansion load evenly distributed along the belt cross-section geometry, which equals:

$$q_{\text{bst}} = \frac{E_2}{1-\mu_1\mu_2}\frac{h^3 l}{12R^3}, \quad (1)$$

where E_2 is the effective modulus of elasticity of belt lateral direction; μ_1, μ_2 are Poisson ratios of the belt in the longitudinal and lateral directions, respectively; R is radius of the pipe; h - thickness of the belt; and l - longitudinal length of the belt section considered, which for a pipe conveyors equals the carry spacing.

Alternatively, to imitate load due to belt bending stiffness, the present study suggests utilization of a constant expansion bending moment $M_{\text{bst}} = EI/R$ applied at the belt edges and accepted by considering the displacement field required to form belt from a flat shape into a pipe. The expression $I = lh^3/12$ is a moment of inertia. For the analysis, the results are obtained and compared using both types of load – constant expansion moment at the edges and Chernenko distributed radial load from (1).

Due to the design limitations of the experimental test rig selected, the load from the bulk material is excluded from the analysis. In addition to the bending stiffness, the load from the belt weight is also considered. It can be determined as follows:

$$q_{\text{bw}} = \frac{m_{\text{b}}' g}{B} l, \quad (2)$$

where m_{b}' represents longitudinal weight of the belt per its unit length; g is gravitational acceleration; and $B = 2\pi R$ - belt width.

Considering that there are only three equilibrium equations and seven unknowns, the system can be classified as statically indeterminate to the fourth degree. The problem is solved using the Force Method [12, 13], following the procedure described by Zamiralova *et al*. [32].

According to the method, the given statically indeterminate structure can be released by substituting redundant forces with additional external loads. The number of redundant forces replaced equals to the degree of the system indeterminacy. For this particular case, the contact forces F_1', F_2, F_3, N_1' are replaced by the additional external forces X_1, X_2, X_3, X_4, respectively. The released structure, shown in Figure 3, can be considered equivalent to the reference system, if the displacements caused by the substituting forces equal zero: $\delta_1 = \delta_2 = \delta_3 = \delta_4 = 0$.

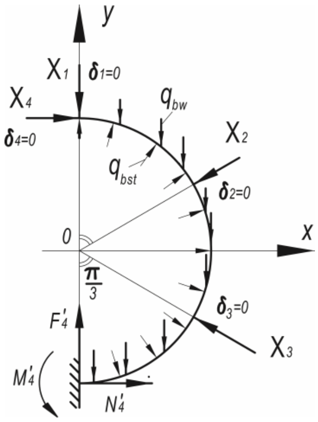

Figure 3. Statically determinate released system, where the redundant forces F_1', F_2, F_3, N_1' are replaced by unknowns X_1, X_2, X_3, X_4. The load from belt bending stiffness is shown as even radial expansion load q_{bst}.

Assuming that the deformations of the structure are linear, it is possible to articulate the system of canonical equations:

$$\begin{bmatrix}\delta_1\\ \delta_2\\ \delta_3\\ \delta_4\end{bmatrix} = \begin{bmatrix}\bar{\delta}_{11} & \bar{\delta}_{12} & \bar{\delta}_{13} & \bar{\delta}_{14}\\ \bar{\delta}_{12} & \bar{\delta}_{22} & \bar{\delta}_{23} & \bar{\delta}_{24}\\ \bar{\delta}_{13} & \bar{\delta}_{23} & \bar{\delta}_{33} & \bar{\delta}_{34}\\ \bar{\delta}_{14} & \bar{\delta}_{24} & \bar{\delta}_{34} & \bar{\delta}_{44}\end{bmatrix}\begin{bmatrix}X_1\\ X_2\\ X_3\\ X_4\end{bmatrix} + \begin{bmatrix}\delta_{1\text{P}}\\ \delta_{2\text{P}}\\ \delta_{3\text{P}}\\ \delta_{4\text{P}}\end{bmatrix} = 0. \quad (3)$$

In (3), $\bar{\delta}_{mn}$ unit displacements are caused by the unit loads, where index m characterizes each of the four displacements considered, and index n is the force that causes that displacement. $\delta_{m\text{P}}$ represents the displacements from external loads, particularly from the belt weight and bending stiffness. The displacements are found using the Maxwell-Mohr Integral. Considering [12, 13, 32], the displacements from unit loads can be determined using the moment component:

$$\bar{\delta}_{mn} = \int_L \frac{M_{1m}M_{1n}}{EI}\text{d}s + \int_L \frac{N_{1m}M_{1n}}{EA}\text{d}s + \int_L \frac{kQ_{1m}Q_{1n}}{GA}\text{d}s, \quad (4)$$

where G is the shear modulus, $A = bh$ is the cross sectional area, and $\text{d}s = R\text{d}\varphi$. The bending moments

M_{11}, M_{12}, M_{13}, M_{14}, the axial forces N_{11}, N_{12}, N_{13}, N_{14}, and the shear hoop forces Q_{11}, Q_{12}, Q_{13}, Q_{14} are obtained by independently applying unit loads $\overline{X}=1$ to the structure instead of to each of the redundant forces X_1, X_2, X_3, X_4 (see Zamiralova *et al.* [32]). In this case, the moments, for example, equal:

$$M_{11} = R\sin\varphi \text{, for } 0 \le \varphi \le \pi \text{;} \tag{5}$$

$$M_{12} = R\sin\left(\varphi - \frac{\pi}{3}\right) \text{, for } \frac{\pi}{3} \le \varphi \le \pi \text{;} \tag{6}$$

$$M_{13} = -R\sin\left(\varphi + \frac{\pi}{3}\right) \text{, for } \frac{2\pi}{3} \le \varphi \le \pi \text{;} \tag{7}$$

$$M_{14} = R(1-\cos\varphi) \text{, for } 0 \le \varphi \le \pi \text{.} \tag{8}$$

Analogically, the axial and shear hoop forces are determined.

In the present study, the moment and the forces are presumed positive, if they increase the curvature of the cross section. A similar procedure follows for the displacements from external loads. The Maxwell-Mohr Integral equals:

$$\delta_{m\mathrm{P}} = \int_L \frac{M_{1m}M_{\mathrm{P}}}{EI}\mathrm{d}s + \int_L \frac{N_{1m}N_{\mathrm{P}}}{EA}\mathrm{d}s + \int_L \frac{kQ_{1m}Q_{\mathrm{P}}}{GA}\mathrm{d}s \text{.} \tag{9}$$

According to the Principal of Superposition, the moment from external loads M_{P} is composed of the moment from the belt weight M_{bw} and also from the belt bending stiffness M_{bst}:

$$M_{\mathrm{P}} = M_{\mathrm{bw}} + M_{\mathrm{bst}} \text{,} \tag{10}$$

where for $0 \le \varphi \le \pi$ these moments equal:

$$M_{\mathrm{bw}} = q_{\mathrm{bw}}R^2(\varphi\sin\varphi + \cos\varphi - 1) \text{,} \tag{11}$$

$$M_{\mathrm{bst}} = -EI/R \tag{12}$$

or in case of distributed radial load

$$M_{\mathrm{bst}} = q_{\mathrm{bst}}R^2(\cos\varphi - 1) \text{.} \tag{13}$$

Analogically, the axial N_{P} and shear hoop forces Q_{P} from the external loads are determined.

Solving (3), the unknown forces X_1, X_2, X_3, X_4 are quantified. Finally, from equilibrium equations, the pipe conveyor contact forces are evaluated as in (14).

In the present study, the contact forces are presumed positive if they are directed as shown in Figures 2 and 3. If after evaluation, one or more of the contact forces becomes negative, it means that there is a contact loss at this point. In this case, the maximal negative redundant force is assumed to be equal to zero, and the corresponding displacement component must be removed from the system (14). After that, the calculation must be repeated.

$$\begin{cases} F_1 = 2F_1' = 2X_1; \\ F_2 = F_6 = X_2; \\ F_3 = F_5 = X_3; \\ F_4 = 2F_4' = 2\pi q_{\mathrm{bw}}R + 2X_1 + X_2 - X_3. \end{cases} \tag{14}$$

3.2. Experimental test

To validate the mathematical model, the experimental tests were performed using a six-point pipe belt stiffness device, which is owned by Phoenix Conveyor Belt Systems GmbH. The approximate sketch of the test rig is given in Figure 4. The details on the test rig design and corresponding measurement procedure are explicitly described by Zamiralova and Lodewijks in [28]. The test rig is selected over the other possible design configurations, as it is simple for the test performance, accurate in terms of the controlling friction forces, and closely correlates with the analytical model. Moreover, the test rig allows one to use the same belt samples as for the throughability test.

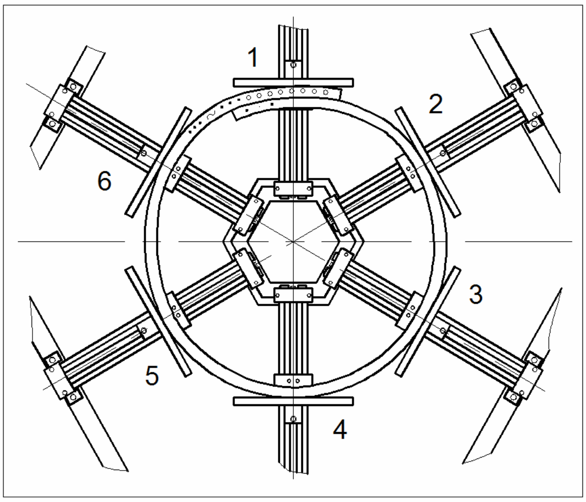

Figure 4. Approximate sketch of the static six-point pipe belt stiffness testing device of Phoenix Conveyor Belt Systems GmbH [27].

The fabric belt sample with uniform belt structure was selected for testing. The physical parameters of this sample are provided in Table 1. The results for the case "no overlap" from the experiment are compared with the contact forces, calculated for the same physical parameters using the mathematical model described in section 3.1. To study the impact of belt line mass and bending stiffness, this parameters set is used as a reference.

3.3. Numerical models

Alternatively, the contact forces are determined numerically within FEM. For this purpose two models are created and solved within ANSYS software. The models differ in their modelling procedures.

Table 1. Physical parameters of fabric belt sample used for the experiment.

Parameter	Value	Parameter	Value
Belt Type	EP	Sample mass, kg	3.672
Belt width B, m	≈1.2007	Throughability	0.368
Thickness h, m	≈0.017	Nominal diameter, m	0.400
Longitudinal length of the sample l, m	≈0.151	Length of test rig plates, m	0.200

Model 1 (Complex) closely replicates the experimental test and has a more complex modelling procedure, which can be divided into three steps using a restart function. The first step represents a nonlinear process of folding the flat belt sample into a pipe shape by applying the concentrated moments M_{bst} at the edges of the structure. In the second step, the contact pairs between the belt and the plates, and also the self-contact at the belt edges, are activated using the special element types that support "death" and "birth" functions.

The impact of the friction in the model is assumed to be consistently minimal throughout the experiment. After the belt is folded into a pipe shape, and the contacts with the plates are described, the restrained belt is released by assigning the moments at the edges to zero. This is the third step of the modelling process. At this stage, the gravitation is also activated. Figure 5 shows the results obtained within Model 1.

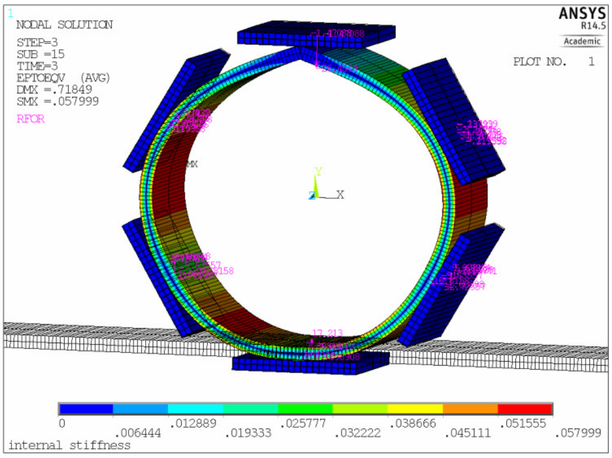

Figure 5. ANSYS solution for von Mises total mechanical strain for the Complex Model 1.

Model 2 (Simplified) is more simple model that closely correlates with the mathematical problem. It is solved within one step and represents a linearized structure of a pipe shape with an opening at the top (See Figure 6). It has six fixed nodes with one radial restraint. At the edges, the structure has an additional restraints for the pinned supports.

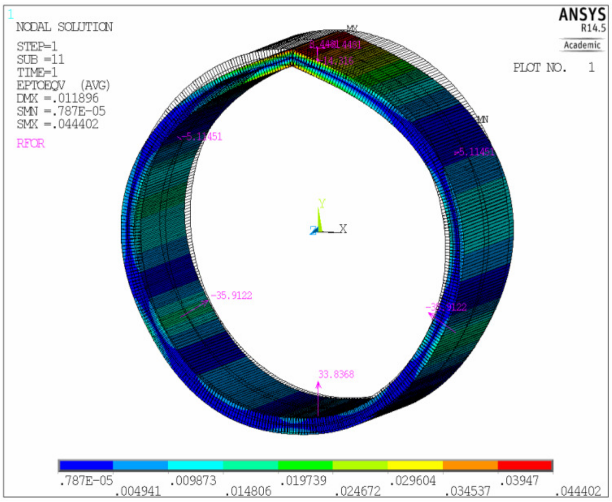

Figure 6. ANSYS solution for von Mises total mechanical strain for the Simplified Model 2.

In addition to the gravity, the load form belt bending stiffness is simulated either by applying the expansion concentrated moments M_{bst} at the edges or by applying radial distributed load q_{bst} (1). The model is solved using linear and nonlinear analysis. The correct direction of the forces is assumed as shown in Figures 2 and 3. For the opposite case, the restraint that corresponds to the maximal opposite force is removed, and the model is recalculated. The input data for both numerical models is provided in Table 1.

4. BELT THROUGHABILITY

For the analytical and numerical models, the effective modulus of elasticity of the belt in the lateral direction is required. This value can be determined from the throughability test.

The test is performed using the test apparatus, shown in Figure 7, as recommended by standard ISO 703 [19]. The belt is clamped and suspended from the horizontal bars within steel wires that can move along the horizontal bars with no impediment (see Figure 7). Maximum deflection generated after ten minutes of sample suspension is an objective of the measurement. The ratio of the maximum sag of the sample to the belt width yields a throughability value.

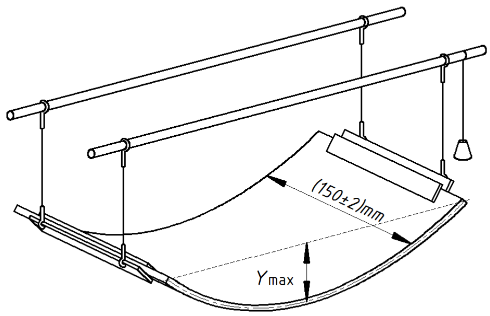

Figure 7. Throughability test ISO 703 [19]

Zamiralova *et al.* [31] made an explicit review analysis of the standard ISO 703 [19] and presented associated models that can quantify an effective modulus of elasticity of a belt based on its throughability performance. The study also reflected the effect of varied belt line mass, bending stiffness and belt geometry. The researchers utilized three methods: a numerical FEM within ANSYS software, and two mathematical models of Wang and Fertis with 3, 5 and 10 Simpson intervals [31]. The researchers indicate that FEM approach and Wang model with no more than 8 Simpson intervals are more preferable for the given strain range.

The methods described by Zamiralova *et al.* [31] for quantifying effective modulus of elasticity of belts are restricted to the small strains. Shilling *et al.* [21] reported that for pipe conveyor belts formed into a pipe shape and operating under normal operational conditions the strains do not exceed 5%. This means that the modulus of elasticity determined from the throughability test with the condition up till 5% strain can be applied for quantifying the contact forces. For the case study based on the data from Table 1, the Wang solution is applied, and the effective modulus of elasticity constituted $E = 5.47$ MPa.

5. RESULTS

Figure 8 provides a comparison of the results obtained from the mathematical model, the experiment, and nonlinear ANSYS solution from the Complex Model 1 and Simplified Model 2. As can be observed, the load from the belt bending stiffness modelled via distributed radial load q_{bst} (1) provides quite different results from the experiment in both analytical and numerical solution and generates a contact loss at the belt edges $N_1'=0$. The results obtained in ANSYS using a concentrated expansion moment M_{bst} within Model 1 and also within Model 2 generates quite close results. Compared to the experiment, these models also exhibit acceptable correlation. This means that such simplification for the restraints and loading conditions (see Figure 2) can be considered acceptable.

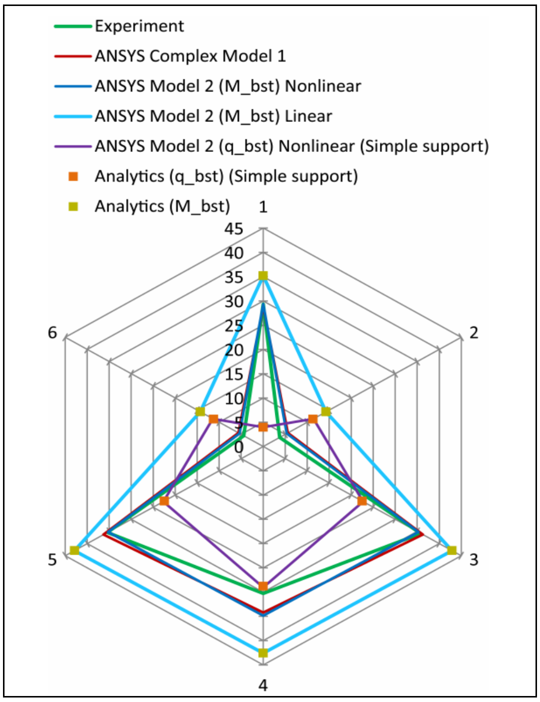

Figure 8. Comparison of the contact forces obtained from the experiment, and also using mathematical and nonlinear FEM solution. The contact forces are as presented in Fig. 1.

To study the impact of belt bending stiffness on load distribution between the idler rolls, the results are obtained for the established reference parameters and for the varied effective modulus of elasticity. Figure 9 exhibits the results, obtained within Simplified ANSYS Model 1 for linear and nonlinear analysis and also using an analytical methodology introduced in section 3.1. The contact loss appears for force F_2 prior to any other contact position. The forces have almost linear dependence and exhibit a switch when there is a contact loss. The results in Figures 8 and 9 show that the analytical method provides a strong correlation with the linear ANSYS solution of the Simplified numerical Model 2, which can be considered a satisfactory validation of the numerical model.

To Figure 10 shows the throughability values at which the contact loss appears. The graphs are obtained for the case study parameters set (Table 1) and line mass q, $q/2$, and $2q$. The graphs are achieved using Complex ANSYS Model 1 for quantifying contact forces at various moduli of elasticity E. The latter is expressed via a corresponding throughability value using techniques described in section 4.

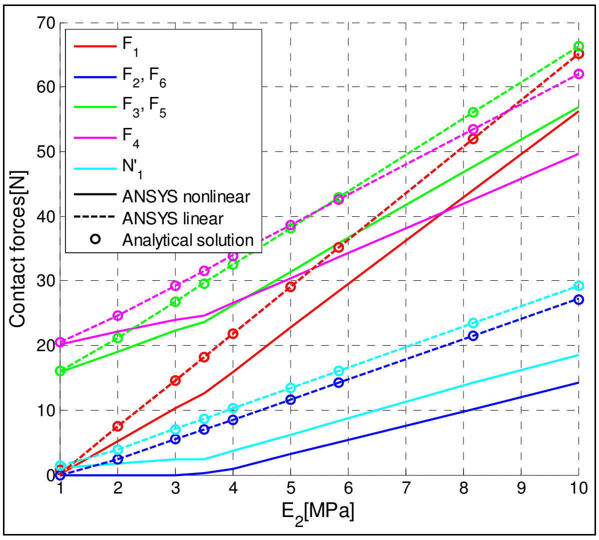

Figure 9. Contact forces obtained for varied moduli of elasticity using the mathematical approach and also the numerical Simplified ANSYS Model 2 for linear and nonlinear analysis.

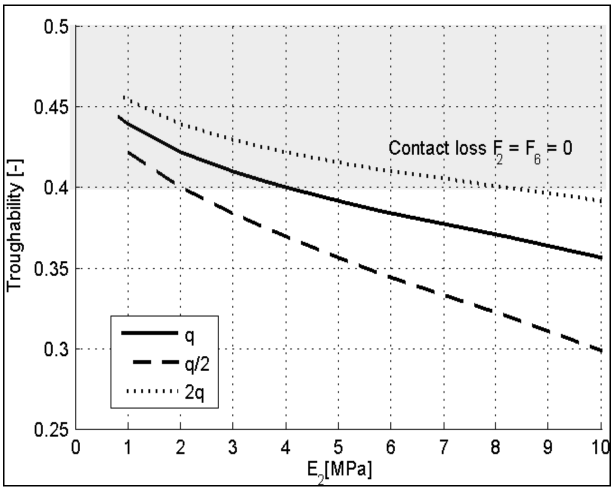

Figure 10. Throughability values versus moduli of elasticity that indicate appearance of the contact loss for the reference parameters set and line mass q, $q/2$, and $2q$.

The results show that the throughability functional dependences as well as a contact loss appears at the modulus of elasticity values that are equivalent to the change of line mass (1, 1/2, or 2). This observation agrees with the results provided in [31]. To form a stable pipe shape for the same belt geometry, heavier belts must be less flexible. For the given set of parameters (see Table 1), the belt has to exhibit a throughability less than 0.399.

6. CONCLUSIONS

This paper presents a new analytical approach that determines pipe conveyor belt bending stiffness sufficient to form a stable pipe shape, and it describes how to expresses that bending stiffness via the belt's throughability performance. Appearance of the contact loss of belt with supporting idler rolls is assumed to be a criterion for the insufficient bending rigidity of the conveyor belt.

A new mathematical approach that incorporates contact between the belt edges and the load from folding the belt from flat shape into a pipe shape is introduced. The latter is realized using a distributed radial load q_{bst} and also concentrated expansion moments M_{bst} applied at the belts' edges. Results are compared with the experimental data obtained using a six-point pipe belt stiffness device. Additionally, two FEM models that mimic the experiment and the analytical model are created and solved within ANSYS software. Results obtained via numerical and mathematical models are compared to the experiment data.

Analysis of the results shows that load from belt bending stiffness needs to be represented via expansion moment M_{bst} because it generates reasonably correct results, compared to the distributed radial load q_{bst}. Moreover, results exhibit a satisfactory correlation between the experiment and the Complex ANSYS Model 1.

At the same time, nonlinear analysis of the Simplified Model 2 generates similar results to those of the Complex Model 1, which means that the selection of the system restraints and loading conditions is performed correctly. At the same time, the mathematical approach exhibits strong correlation with the linear solution of Model 2.

These observations allow one to conclude that the mathematical model developed in this paper can be used as a uniform validation technique for any numerical model. This is especially useful, because they can vary due to a large number of modelling options, starting from the selection of the software and finishing with the choice for the convergence criteria. Complex Model 2 proposed in this paper is well validated and can be used to achieve results for parameter sets different from those used in the experiment, and to achieve results beyond limitations of the mathematical model.

In addition, this paper provides a technique to express the ability of a belt to form a stable pipe shape via its throughability performance, which becomes very useful for practicing engineers. Moreover, the impact of belt line mass and effective modulus of elasticity is also investigated: the heavier the belt, the more rigid the belt should be in bending.

Recommendations for future research are focused on further experimental validation of the analytical models proposed, and also on an impact study of the belt's geometry, line mass, and bending stiffness together with the belt's ability to form a stable pipe shape. In addition, the research has to incorporate the impact of the belt's overlap on appearance of the contact loss. Based on the techniques proposed, the uniform recommendations for pipe conveyor belt pipe-ability need to be developed and expressed via throughability values.

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СТАБИЛНОСТ ОБЛИКА ТРАКСТИХ ЦЕВНИХ ТРАНСПОРТЕРА: ОД ТРАНСВЕРЗАЛНЕ САВИТЉИВОСТИ ДО СПОСОБНОСТИ ТРАКЕ ДА ФОРМИРА СТАБИЛАН ОБЛИК

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Рад приказује нови приступ одређивању крутости на савијање траке цевастих транспортера, што је довољно за постизање стабилности облика цеви на основу њене перформансе: трансверзалне савитљивости. Описује се математички модел који одређује контактне силе код трака цевастих transportera и уводи два нумеричка модела који су решени методом коначних елемената у ANSYS софтверу. Резултати су потврђени експеримен–талним подацима добијеним коришћењем уређаја са 6 тачака за испитивање крутости. Предложени математички модел може да се користи као стандардна техника за валидацију сваког нумеричког модела. Једна од контактних сила која је

једнака нули узима се за критеријум недовољне крутости траке на савијање код формирања стабилног облика цеви. Ефикасност модула еластичности квантификована на основу параметра трансверзалне савитљивости се користи за изражавање способности траке да формира стабилан облик. Истражује се утицај масе транспортне траке и крутости на савијање: код исте геометрије траке, за исправно формирање облика траке, тежим тракама је потребна већа крутост на савијање.