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ANALYTICAL METHOD OF DESIGNING AND SELECTING TAKE-UP SYSTEMS FOR MINING BELT CONVEYORS

ANALITYCZNA METODA PROJEKTOWANIA I DOBORU UKŁADÓW NAPINANIA DLA GÓRNICZYCH PRZENOŚNIKÓW TAŚMOWYCH

This article presents a method developed to design and select tensioning systems which makes use of standard calculations. It describes procedures for selecting and analysing the operation of devices tensioning the belt, which procedures are based on the static characteristics of these devices, and a proposal for introducing a substitute belt elasticity modulus that would make the calculations of the tensioning stroke length account for the value of the initial force tensioning the belt and for its sag between sets of idlers. Static characteristics of tensioning systems have been used to describe their operation and present the advantages and disadvantages of individual design solutions.

Keywords: belt conveyors, take-up systems, designing, calculation methods

W artykule przedstawiono opracowaną metodę projektowania i doboru układów napinania taśmy wykorzystującą stosowane standardowe procedury obliczeniowe uzupełnione o zależności analityczne uwzględniające zwis taśmy między zestawami krążnikowymi i charakterystyki statyczne urządzeń napinających taśmę.

W pierwszej części publikacji opisano analityczną metodę doboru układu napinania taśmy bazującą na wynikach obliczeń sił w taśmie i szacunkowych kalkulacjach drogi napinania taśmy stosowanych obecnie w standardowych procedurach obliczeniowych (Golka i in., 2007; Gładysiewicz, 2003; Żur i Hardygóra, 1996). Następnie przedstawiono propozycję uzupełnienia stosowanej metody analitycznej o wprowadzenie diagramu drogi napinania i uwzględnienie zastępczego modułu sprężystości taśmy (Kulinowski, 2012). W obliczeniach standardowych przyjmuje się, że długość odcinka taśmy pomiędzy zestawami krążnikowymi jest równa ich rozstawowi. W rzeczywistych warunkach może się zdarzyć, że na niektórych odcinkach złożonego profilu trasy przenośnika wartość zwisu taśmy przekracza wartości dopuszczalne, wtedy długość taśmy między zestawami krążnikowymi jest znacząco większa od rozstawu zestawów. W takich przypadkach wartość modułu sprężystości taśmy uwzględnianego w obliczeniach drogi napinania jest korygowana na podstawie zależności przedstawionej w niniejszym artykule. Zastępczy moduł sprężystości taśmy powinien być uwzględniony w algorytmie obliczania długości skoku

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wózka napinającego, szczególnie dla przenośników, których złożony profil trasy może przyczynić się do lokalnego zmniejszenia sił w taśmie.

W kolejnej części artykułu przedstawiono metodę konstruowania charakterystyk statycznych urządzeń napinających taśmę (Jabłoński, 1988), które wykorzystano do opisu stosowanych układów napinania taśmy. Spośród stosowanych urządzeń napinających przedstawiono rozwiązania ze stałym położeniem bębna napinającego podczas pracy przenośnika - tzw. sztywne urządzenia napinające oraz ze zmiennym położeniem bębna napinającego w czasie pracy przenośnika - grawitacyjne, hydrauliczne, automatyczne i nadążne. Na diagramach opisujących charakterystykę napinania, poza wymaganą wartością siły w taśmie zbiegającej z bębna napędowego, przedstawiono spodziewany przebieg jej zmiany w funkcji momentu napędowego, charakterystyczny dla danego typu urządzenia napinającego taśmę (Kulinowski, 2012).

Podsumowanie artykułu stanowi przedstawienie propozycji wykorzystania charakterystyk statycznych urządzeń napinających do oceny pracy ciernego, bębnowego układu napędowego przenośnika taśmowego.

Przedstawiona w artykule metoda obliczeniowa została weryfikowana poprzez badania przemysłowe i realizacje wielu złożonych projektów przenośników taśmowych uzupełnionych analizą wyników badań symulacyjnych dynamiki pracy modeli urządzeń napinających taśmę (Kulinowski, 2012).

Słowa kluczowe: przenośniki taśmowe, urządzenia napinające, projektowanie, metody obliczeniowe

1. Introduction

The era of belt conveyors in mining transport systems dates back to the early 20^{th} century. Over the years, engineers improved the structure of conveyor sub-assemblies, the drive technology and primarily of the main element of these machines, i.e. the belt. Even before World War II, artificial materials such as artificial silk and rubber were first used for the carcass, and from the early 1940s, due to problems with natural rubber availability, attempts were made to use polyvinyl chloride for the top and bottom covers. In the 1940s, steel wire cords were used for the carcass, while the 1950s saw the introduction of polyamide and polyester for making fabric carcasses, which continues until today.

Currently, due to their transport capacity and reliable operation, belt conveyors play a dominant role in systems for hauling useful minerals, both in open pit and underground mines. Single belt conveyors transporting overburden achieve capacities of up to 50,000 tons/h. Conveyors are built that are almost 20 km long with the power of drives installed on a single machine reaching 12 MW.

Belt conveyors of the greatest capacities, speeds and installed drive power are used to transport overburden in open pit lignite mines, but designers face the most interesting engineering challenges when designing overland conveyors more than ten kilometres long, designed for operating in difficult terrain and climate. The problem there is not just about selecting and configuring drive systems, but the significant lengths of conveyors make the correct design of an effective and reliable belt tensioning system a major challenge. The correct operation of the belt tensioner reduces the failure rate of the conveyor and improves the durability of the belt, whose dimensions make its investment and operating cost clearly greater than those of all other conveyor sub-assemblies.

The selection and analysis of the operation of modern belt tensioning devices requires complex computational procedures and simulation studies which necessitate the use of specialist computer software in the design process (Kulinowski, 2012).

2. Standard method of designing a tensioning system

The standard method of calculating and selecting tensioning systems includes calculating the minimum value of the belt tensioning force and the length of the belt tensioning path, which requires estimating the forces present in the belt during steady-state operation, start-up, deceleration and stoppage.

2.1. An analytical method for selecting the belt tensioning system

2.1.1. The envelope method of calculating forces on the belt

The procedure of preparing the data of a conveyor to calculate forces present in the belt requires splitting the conveyor route geometrically into sections. Every section starts and ends in a characteristic point associated with the place where the load on the belt changes. Point 1 is customarily located where the belt comes onto the discharge pulley of the conveyor. The conveyor is split into sections according to the following criteria:

- a change of route geometry associated with a change in the inclination angle of the route, or with a horizontal or vertical curve;
- a change in the amount of material transported on the belt due to the presence of additional loading or offloading points along the conveyor route;
- the presence, in a given section, of a device causing a change of forces in the belt not due to resistance to motion, such as a drive system, a braking system, a tensioning device;
- the presence, in a given section, of devices which cause the resistance to motion to increase, e.g. deflectors, cleaning devices, changed spacing of idler sets;
- the need to determine forces in the belt at additional points of the belt.

In the presented computational algorithm, the number of characteristic points is $n = 2k + 2$, where k is the number of route sections (Kulinowski, 2012).

Below, there is an example of a belt conveyor made up of 4 horizontal sections, 2 curved sections and one sloping section, on which two loading points have been fitted (Fig. 1). There are 16 characteristic points of belt load change along the conveyor route. For each of these points, specialist software is used to determine the forces in the belt for various load states of the conveyor.

Forces at specified points of the conveyor route are determined based on a relationship which requires calculating the following forces acting on a belt section between points: belt

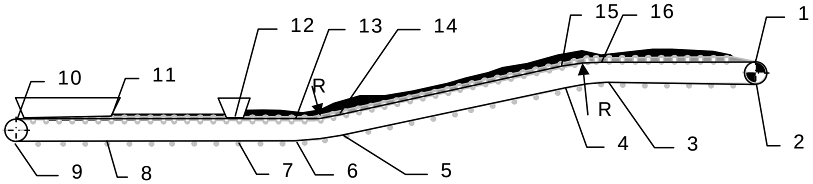


Fig. 1. A diagram of the location of characteristic points for a conveyor with a complex route profile (Kulinowski, 2012)

resistance to motion on particular sections (Kulinowski, 2012), driving or braking forces, inertia resistance etc. (Fig. 2).

$$S_{i+1} = S_i + W_{GSD(i \div i+1)} \pm W_{Hn(i \div i+1)} \pm W_{Ht(i \div i+1)} \pm m_{zr(i \div i+1)} \cdot a - P_{(i \div i+1)} \tag{1}$$

where:

S_i — force in the belt at point (i) [N],

$W_{GSD(i \div i+1)}$ — belt resistance to motion due to overcoming the friction force (main, concentrated, additional frictions) along the section between points (i) and (i + 1) [N],

P — the driving or deceleration force acting on the belt section, [N],

W_{Hn} — the resistance to lifting the handled material, [N]

W_{Ht} — the resistance to lifting the belt, [N],

m_{zr} — reduced mass of the section, [kg],

a — belt acceleration or deceleration (a = 0 for steady-state operation of the conveyor), [m/s^2].

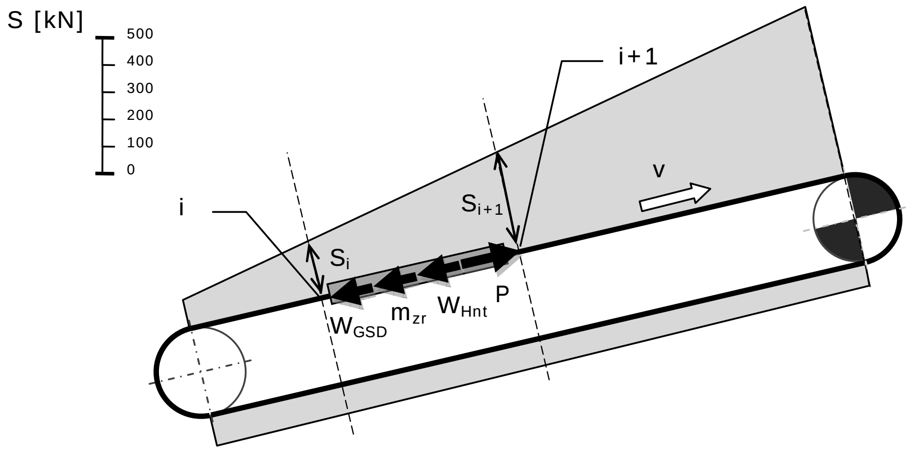


Fig. 2. An example graph of forces in the conveyor belt (Kulinowski, 2012)

The forces in the belt determined based on the above formula are corrected to ensure the correct frictional contact of the belt with drive pulleys (*11*), (*12*) and for the permissible values of belt sags between idler sets (*8*).

The values of forces in the belt calculated at individual points are used for strength calculations of the loading condition of the conveyor structure and to analyse the dislocation of the car tensioning the belt during the start-up, braking or changes in the load of handled material on the belt.

2.1.2. Method of calculating belt tensioning path length

The length of the belt tensioning path is determined based on the calculated value of the reserve for permanent deformations, the ambient temperature change, the replacement of belt splices as well as the travel of the belt-tensioning cart. The travel length depends on the belt type and the stresses occurring in it, because when the conveyor is running, the belt, which is flexible, stretches and shrinks as a result of longitudinal force action.

$$L_n = L_{ro} + L_{rt} + L_{rz} + k_d \cdot s_w \quad [\mathrm{m}] \tag{2}$$

where:

- L_n — the length of the belt tensioning path [m],
- L_{ro} — the length reserve for permanent deformations [m],
- L_{rt} — the length reserve for temperature changes [m],
- L_{ro} — the length reserve for replacing belt splices [m],
- s_w — the length of the tensioning cart travel [m],
- k_d — a coefficient accounting for the shock load of the belt amounting to: 1.5 for textile belts, 1.2-1.3 for belts with carcasses of steel cords (Żur & Hardygóra, 1996); the literature also cites the value of 2.0 (Golka et al., 2007).

If a rigid system for belt tensioning is used, the value of s_w is equal to 0, because this system serves only to compensate permanent belt elongations. During conveyor operation, the location of the belt tensioning pulley does not change, while the tensioning cart travels only during conveyor stoppages.

The increase of the length of the belt tensioning path required because of permanent deformations can be calculated based on the following relationship (Golka et al., 2007):

$$L_{ro} = k_{lo} \cdot L \quad [\mathrm{m}] \tag{3}$$

where:

- L — conveyor length [m]
- k_{lo} — correction factor for permanent deformations, equal to:
  - 0.004 for fabric belts,
  - 0.002 for SolidWoven belts,
  - 0.0005 for belts with carcasses of steel cords.

It is also assumed that the reserve of the tensioning path length for permanent deformations should amount to some 50% of the calculated length of the tensioning path for a fabric belt, and to 20%-30% of it for belts with steel cords (Gładysiewicz, 2003; Żur & Hardygóra, 1996).

The reserve length of the belt tensioning path to account for ambient temperature changes can be calculated based on the following relationship (Golka et al., 2007):

$$L_{rt} = k_{lt} \cdot L \quad [\mathrm{m}] \tag{4}$$

where:

- L — conveyor length [m],
- k_{lt} — the correction factor accounting for temperature changes, amounting to:
  - 0.005 for fabric belts;
  - 0.0003 for belts with cores of steel cords.

In addition, the reserve length of the tensioning path accounting for the possible replacement of splices is equal to one half of the product of the number of reserve belt splices and their length.

$$L_{rz} = \frac{n_{rz} \cdot L_z}{2} \quad [\text{m}] \tag{5}$$

where:

- L_z — the length of the belt splice (1.5-2) [m],
- n_{rz} — the number of reserve belt splices.

The literature (Golka et al., 2007) also mentions a correction factor accounting for an additional reserve of the tensioning path length due to belt sags, equal to one per mille of the conveyor length.

The cart travel length is determined based on the change of the belt length when transiting from one belt loading state to another. The following states of the conveyor are analysed:

- *p* — stoppage,
- *r* — start-up,
- *u* — steady-state operation,
- *h* — deceleration.

During standard calculations, the dislocation of the cart when the belt loading state changes is determined for the following periods of conveyor operation:

- *pr* — conveyor start-up, a transition from the stoppage state to the start-up state;
- *ru* — the phase of transitioning from start-up to steady-state operation;
- *uh* — process deceleration of the conveyor;
- *hp* — conveyor stopping, transitioning from the deceleration state to a stop.

In order to determine the length of travel of the tensioning cart when transitioning from the operating state *a* to *b*, marked with the index *ab* (*pr, ru, uh, hp*), one uses a relationship whose simplified form is found in the literature (Antoniak, 1990; Gładysiewicz, 2003; Żur & Hardygóra, 1996), whereas the relationship implemented in the computer application also accounts for belt sagging and takes the following form:

$$s_{ab} = \sum_{i=1}^{n} \left(\frac{\left(S_{a(i)} + S_{a(i+1)}\right) \cdot L_{(i)\div(i+1)}}{4 \cdot E_{z(i)\div(i+1)} \cdot B} \right) - \sum_{i=1}^{n} \left(\frac{\left(S_{b(i)} + S_{b(i+1)}\right) \cdot L_{(i)\div(i+1)}}{4 \cdot E_{z(i)\div(i+1)} \cdot B} \right) \quad [\text{m}] \tag{6}$$

where:

- $S_{a(i)}$ — the force in the belt at point *i* for the operating state *a* [N],
- $L_{(i)\div(i+1)}$ — the length of the route section between points (*i*) and (*i* + 1) [m],
- E_z — the substitute modulus of belt elasticity calculated based on the relationship (7) [N/m],
- B — the belt width [m].

The tensioning cart travel length s_w is determined as the distance between its extreme positions during start-up, steady-state operation, deceleration and stoppage of the conveyor (Fig. 3).

The value of the dislocation of the tensioning cart (s_{pr}, s_{ru}, s_{uh}, s_{hp}) is influenced by the change of the loading state of the belt and its flexibility characteristics. The forces in the belt depend on many factors, including on the conveyor's resistance to motion, the type of tensioning device and the method of controlling the start-up and deceleration of the drive system. However, when the conveyor is being designed, the elasticity modulus of the belt is the parameter determining the length of the tensioning cart travel. The value of this modulus is determined precisely in laboratory conditions for samples of tapes, but results of industrial research show that the belt working in a conveyor has rheological characteristics different than those of the sample studied at the laboratory (Kulinowski, 2012). The reason for this may be that the belt is significantly wider than the laboratory sample, has an arched cross-section, but mainly that it sags between sets of idlers.

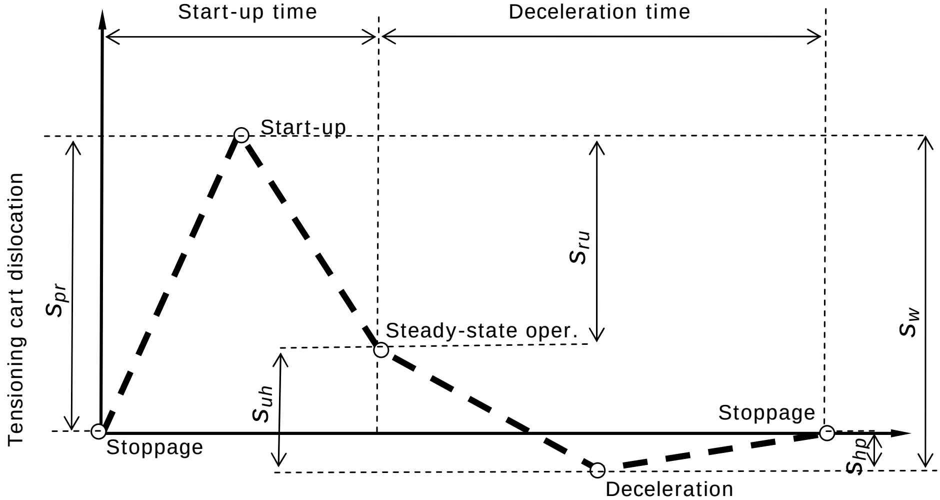


Fig. 3. Diagram of the belt tensioning path (Kulinowski, 2012)

In real conditions, the sag on some sections of a complex conveyor route profile exceeds the permissible values, and then the length of belt between sets of idlers is somewhat greater than the distance between them. In such cases the value of the elasticity modulus of the belt included in calculations of the tensioning path is corrected according to the following relationship:

$$E_z = \frac{1}{\dfrac{1}{E} + \dfrac{(q \cdot g)^2 \cdot l^2 \cdot B}{12 \cdot \left(S - q \cdot v^2\right)^3 \cdot \cos^2 \delta}} \quad \text{[N/m]} \tag{7}$$

where:

- S — force in the belt, [N],
- E — elasticity modulus of the belt sample [N/m],
- B — belt width [m],
- δ — angle of the belt section slope [m],
- l — distance between idler sets [m],
- q — the mass of 1 running metre of belt together with the handled material [kg/m],
- v — belt velocity [m/s].

The substitute elasticity modulus of the belt should be included in the algorithm for calculating the length of travel of the tensioning cart, particularly in the case of conveyors whose route profile may contribute to a local reduction of forces in the belt.

Standard calculations assume that the length of a belt section between sets of idlers is equal to the spacing of those sets. This assumption is true if the forces in the belt are significant and its sagging is negligible. In such a case, it is acceptable to replace the value E_z with the belt modulus of elasticity E.

2.1.3. Diagram of the belt tensioning path

The belt tensioning path can be illustrated with a diagram showing the location of the tensioning cart in specific states of conveyor operation. The diagram of the tensioning path (Fig. 3) can be used to read the start-up and deceleration time of the conveyor, the length of travel of the tensioning cart and the dislocation of the tensioning cart when transitioning from the stoppage to the start-up state, then from the start-up to steady-state operation and from steady-state operation to the process deceleration state of the conveyor.

2.2. Calculations of the minimum force tensioning the belt

The value of the force tensioning the belt for a given conveyor must be selected so that two conditions are fulfilled:

1. the sags of the belt between idler sets should be restricted to maintain the correct geometrical shape of the belt;
2. non-slip contact between the belt and the driven or decelerated pulley must be ensured.

The first condition defines the minimum force in the belt assuming that the sag is as permitted. The value most often assumed is $f_u/l_g = 0.015$ and during conveyor deceleration the value of $f_u/l_g = 0.04$ is allowed. This force is calculated from the following equation:

$$S_{\min} = \frac{g \cdot (m_t + m_u) \cdot l_g^2}{8 \cdot f_u} \quad [\mathrm{N}] \tag{8}$$

where:

m_t, m_u — unit masses of the belt and the output, [kg/m],
l_g — the distance between the upper idler sets, [m],
f_u — belt sagging between idler sets, [m],
$g = 9.81$ [m/s^2].

The second condition, which ensures the correct friction contact between the belt and the pulley is defined by the following relationship:

for the start-up (Fig. 4a):

$$k_p \cdot M_N \le R_b \cdot S_{2N} \left(e^{\mu\alpha} - 1\right) \tag{9}$$

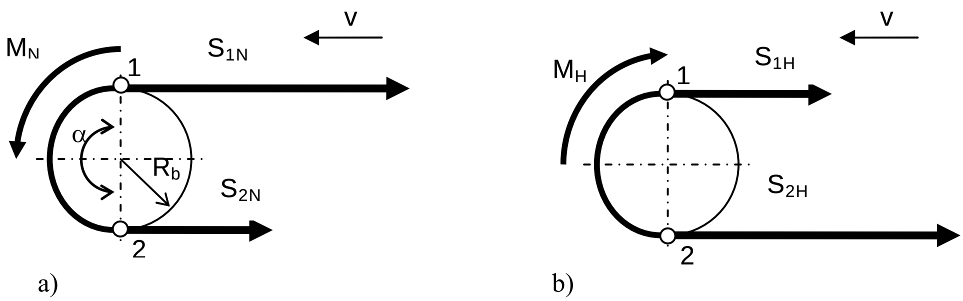


Fig. 4. Forces in the belt while the pulley is driven – a) and slowed down – b) (Kulinowski, 2012)

for slowing down (Fig. 4b):

$$k_p \cdot M_H \le R_b \cdot S_{1H} \left(e^{\mu\alpha} - 1\right) = R_b \cdot S_{2H} \frac{e^{\mu\alpha} - 1}{e^{\mu\alpha}} \tag{10}$$

where:

M_N, M_H — the driving and slowing momentum, Nm,
k_p — the coefficient for preventing slip ($k_p > 1$).

A tensioning device situated near the drive pulley, which can be driven or slowed down, should ensure the force S_2 defined by the following relationships for a specified value of torque:

– conveyor driving

$$S_{2N} \ge M_N \frac{k_p}{R_b \cdot \left(e^{\mu\alpha} - 1\right)}\ ;\ S_{2N} \ge \frac{g \cdot (m_t + m_u) \cdot l_g^2}{8 \cdot f_u} + \Delta W \tag{11}$$

– conveyor deceleration

$$S_{2H} \ge M_H \frac{k_p \cdot e^{\mu\alpha}}{R_b \cdot \left(e^{\mu\alpha} - 1\right)}\ ;\ S_{2H} \ge \frac{g \cdot (m_t + m_u) \cdot l_g^2}{8 \cdot f_u} + \Delta W \tag{12}$$

where: ΔW — the value of the correcting force for point 2, accounting for the minimum permissible values of forces in the belt along the entire length of the conveyor.

The value of the angle of wrap a is constant for a given conveyor, while the friction coefficient m is a random variable. If m is assumed in the above relationships as a constant selected from a set of random values (safe for the given conditions), linear relationships are obtained which prove the thesis that minimum forces S_2 which should be caused by the tensioner to ensure the correct operation of a belt conveyor within the interval $M \in [M_{H\max}; M_{N\max}]$ are defined by the straight lines *a*, *b* and *c* (Fig. 5).

The developed algorithm for calculating the length of the belt tensioning path has brought to light significant problems associated with the operation of conveyor belt tensioners if the route has a complex profile.

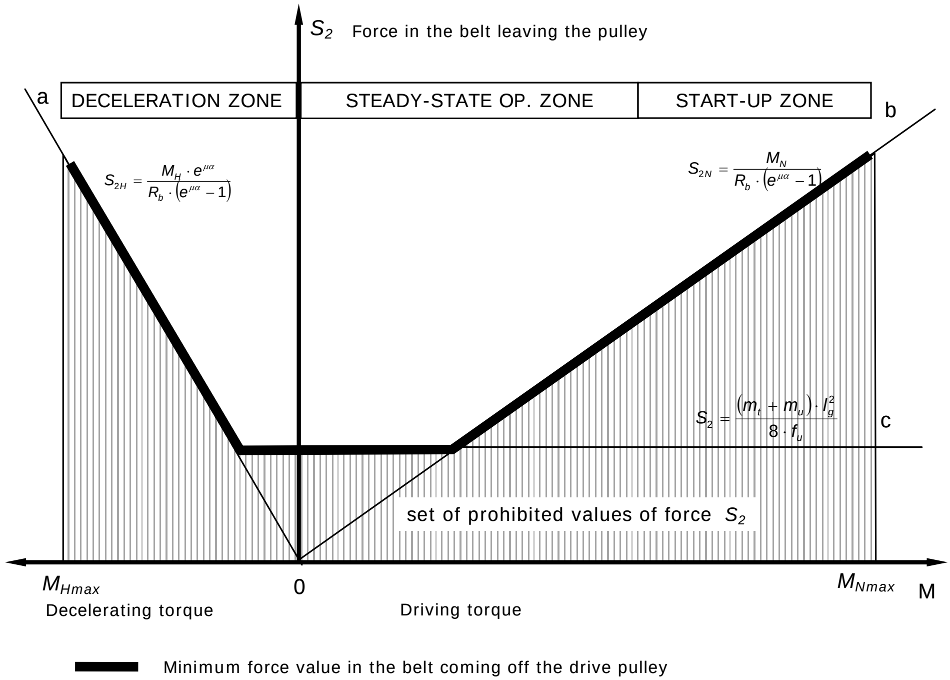


Fig. 5. Minimum values of the preliminary tension of the belt S_2 as a function of the torque M on the driven or slowed down pulley (Kulinowski, 2012)

After a series of tests were executed, the algorithm for calculating the substitute modulus of elasticity of the belt was input into the QNK™ program. The result of the multi-variant analysis of belt behaviour along the route of conveyors with a complex route profile, particularly if a variable quantity of output on the belt is accounted for, have demonstrated the utility of the developed algorithm for selecting take-up device during in the process of designing belt conveyors.

3. Types of take-up devices used in belt conveyors

The following solutions can be listed among the tensioners used:

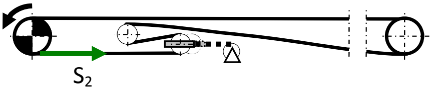
A. those with a fixed position of the tensioning pulley during conveyor operation – so called fixed take-up device;
B. those with a changing position of the tensioning pulley during conveyor operation:
   - weight (gravity);
   - constant tension – equipped with a hydraulic or pneumatic system driving the tensioning pulley;
- automatic – equipped with automation systems;
- follow-up – the required changes of the tensioning force follow changes of the driving torque on the pulley.

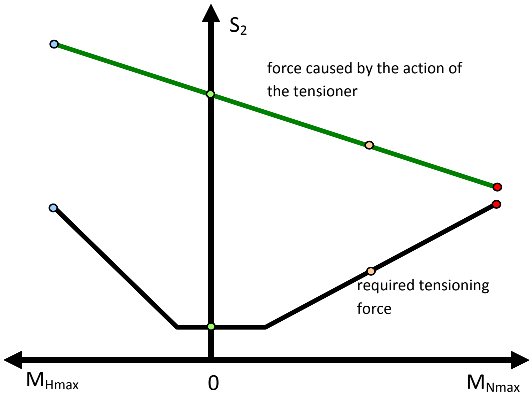
The calculated minimum values of belt tensioning force (8),(11),(12) are used to determine the required value of the force in the belt coming off the pulley as a function of the torque on the axis of the drive pulley. Then, the static characteristics of tensioners obtained from calculations, simulation studies, laboratory or industrial research are input into a simplified diagram showing those relationships.

Descriptions of the listed types of belt tensioners together with their diagrams and static characteristics are presented below. Apart from the required value of the force S_2 (Fig. 5), the graphs show the expected course of force changes in the running off belt S_2 as a function of the driving torque for the specific type of the device (Kulinowski, 2012).

TABLE 1

Static characteristics of selected types of take-up systems (Kulinowski, 2012)

<table>
<tr>
<td>a)
So-called fixed take-up devices – with a constant position of the tensioning pulley during conveyor operation


With a tensioning pulley periodically moved during conveyor stoppage.
Elastic elongations of a belt loaded with a longitudinal force cause a reduction of forces in the zone where the belt comes off the drive pulley. This necessitates causing much greater forces of preliminary belt tensioning. Permanent deformations of the belt are periodically compensated by moving the tensioning pulley during conveyor stoppages.</td>
<td>
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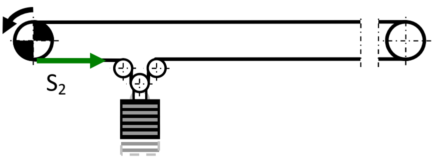
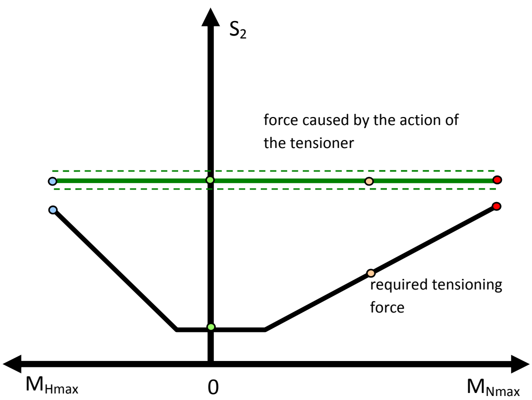
<td>b)
Gravity take-up device with unlimited tensioning path


In conditions of a variable load of output on the belt, weight (gravity) devices stabilise the belt tension forces within limits conditional on the efficiency of the tackle and rope system. In non-steady-state operation of the conveyor, tensioning force changes are observed due to inertia forces and longitudinal vibrations of the belt.</td>
<td>
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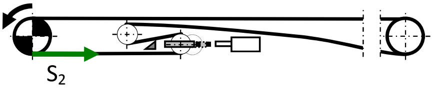
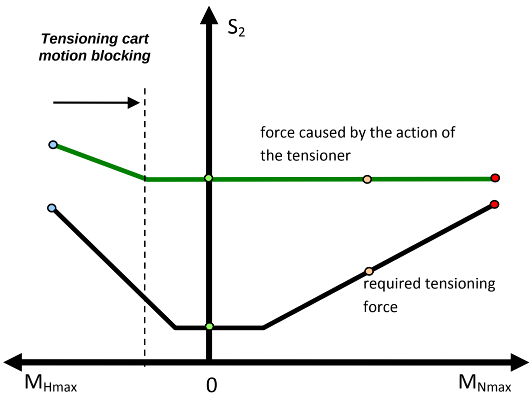
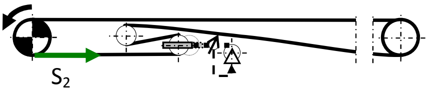
c)
Hydraulic or pneumatic take-up devices with a limited tensioning path.



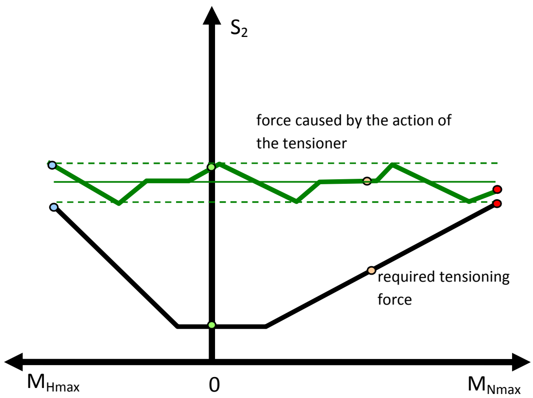
Tensioners with linear cylinders may employ various solutions: the so-called constant tension ones are connected to a trunk line with a pressurised liquid or ones with their own drive and a controlled pressure range. The characteristic feature of this type of devices is that the length of the tensioning path is limited by the length of stroke of the cylinder piston; the speed of tensioning is also limited.



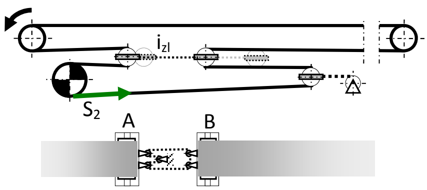
d)
Automatic take-up devices with a periodic operation.



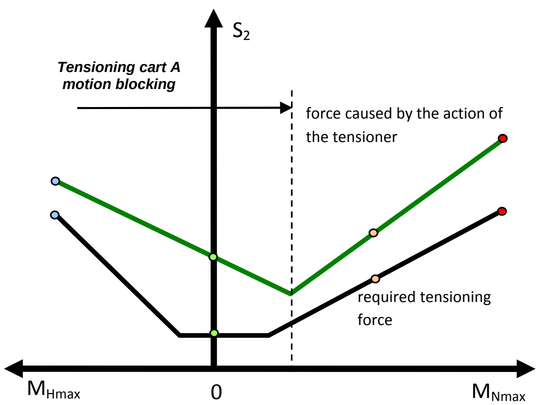
Tensioners in which the tensioning cart is pulled by the rope from a winch driven by an electric motor. They are controlled by three-position controllers, sometimes with the ability to change the force adjustment range for the duration of the conveyor start-up.



e)
Follow-up, mechanical take-up devices



Mechanical tensioners receiving their power from the main drive of the conveyor. Two tried and tested solutions are worth mentioning here.
The first has a tilting drive system suspended from the pulley shaft hanging from a system of ropes and tackle connecting it to the tensioning cart. The tensioning force of this device is the sum of a part of the weight of



the driving system and the reaction to the driving torque (Jabłoński & Kulinowski, 2006). The second type of tensioners with a similar solution are those with two tensioning carts installed before and after the drive pulley in the main drive zone and connected with a rope system of a constant ratio ensuring that the friction contact between the drive pulley and the belt is correct (Jabłoński & Kulinowski, 1993).	

4. Static characteristics as a tool for assessing the operation of a driving and tensioning system

Diagrams describing the static characteristics of tensioning systems are also successfully used to assess the operating parameters of the tensioning system when a specified method of starting up the conveyor is followed. Data necessary to construct diagrams can be obtained in the course of simulation studies conducted using a dynamic belt conveyor model (Kulinowski, 2013) or can consist in results of industrial studies of driving and tensioning systems of belt conveyors (Kulinowski, 2008, 2012).

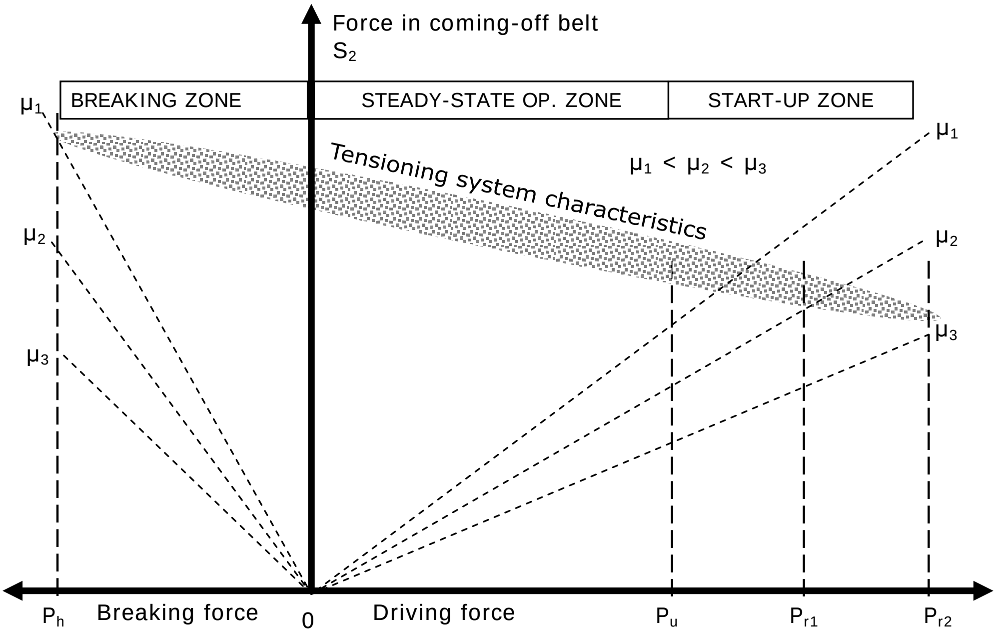


Fig. 6. The static characteristics of a fixed take-up system for a specific angle of wrap α and three values of the coefficient of friction between the belt and the pulley liner ($\mu_1 < \mu_2 < \mu_3$) (Kulinowski, 2012)

A method of analysing the operation of a frictional pulley drive of a belt conveyor making use of the static characteristics of the driving and tensioning system is presented below (Kulinowski, 2011).

At the final stage of simulation studies of a belt conveyor operation, the majority of technical/operational and design parameters have been strictly defined, with the exception of parameters of control system settings. Sometimes one of the last tasks in the conceptual design is to select the liner of the drive pulleys and/or correct the start-up (or deceleration) program of the belt conveyor to ensure slip-less work of the drive.

The procedure of using the static characteristics of the driving and tensioning system to assess its operation is as follows:

1. carry out simulation studies and archive the course of changes of forces in the belt;
2. supplement the static characteristics diagram with lines describing the minimum value of force in the coming-off belt S_2 as a function of the torque or the driving force for specified design parameters of the driving system and several values of coefficients of friction between the belt and the drive pulley liner ($\mu_1, \mu_2, \mu_3, \ldots$);
3. supplement the static characteristics diagram with the value of the force S_2 obtained from simulation studies (item 1), which force, presented as a function of the torque or the driving force, describes the characteristics of the tensioning system as a set of points;
4. analyse driving force ranges for which the value of the force S_2 of the tensioner characteristics exceeds the minimum value of the driving force.

In the diagram shown in fig. 6, the driving system works correctly within the entire range of the circumferential force (from P_h to P_{r2}) only for a liner friction coefficient greater than μ_3. Reducing the start-up force to P_{r1} will permit the slip-less operation even if the friction coefficient drops to μ_2 ($\mu_1 < \mu_2 < \mu_3$).

5. Summary

Designing a belt conveyor consists of executing a set of integrated processes to correctly select and combine its subassemblies into a unique machine meeting a defined transport requirement. An important stage in this process is to select the belt tensioning system and analyse its operation on a calculation model.

The proposed method of selecting belt tensioning systems is based on the static characteristics of these devices proposed by literature (Jabłoński, 1988) and on extending the standard method of calculating the belt tensioning path length by introducing a substitute belt elasticity modulus that would make the calculations of the tensioning path account for the value of the initial force tensioning the belt and its sag between sets of idlers.

The presented method has been verified in industrial research and also by designing (using the QNK™ computer program) many complex designs of belt conveyors supplemented with analyses of results from simulation studies of the work dynamics of belt tensioner models (Kulinowski, 2012).

Problems associated with analysing the operation of take-up systems during conveyor start-up can be solved using the model of a conveyor with distributed parameters presented in many positions of literature (e.g. Dolipski et al., 2012; Kulinowski, 2012) Simulation studies carried out using this model of the belt conveyor make it possible to assess the operation of conveyor subassemblies:

– for drives with specified characteristics of starters, mounted in selected locations along the conveyor route;
– with using a rheological model of the belt and various, parameterized types of take-up systems.

The issue of designing and selecting of take-up systems through simulation studies will be presented in the next publication

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