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SIMULATION METHOD OF DESIGNING AND SELECTING TENSIONING SYSTEMS FOR MINING BELT CONVEYORS

SYMULACYJNA METODA PROJEKTOWANIA I DOBORU URZĄDZEŃ NAPINAJĄCYCH DLA GÓRNICZYCH PRZENOŚNIKÓW TAŚMOWYCH

This article deals with a methodology developed to design and select tensioning systems with using simulation studies on a belt conveyor model. It describes the structure of a dynamic model of a belt conveyor with distributed parameters and details of physical models of the applied belt tensioning systems. For proper display and comparison of work described tensioning systems are shown examples of the results of simulations carried out on a discrete model of the mining belt conveyor.

Presented methodology has been verified through industrial research and realization of a number of complex projects requiring simulation tests of take-up systems of belt conveyors.

Keywords: belt conveyors, take-up systems, designing, calculation methods, dynamic analysis, simulation tests

W artykule przedstawiono symulacyjną metodę projektowania i doboru układów napinania taśmy, wykorzystującą wyniki badań modelowych, będącą często wymaganym rozszerzeniem stosowanych analitycznych metod obliczeniowych (Kulinowski, 2013b). Z uwagi na zjawiska falowe występujące w taśmie podczas nieustalonych stanów pracy przenośników taśmowych, rozruchu, hamowania i pracy przy zmiennej ilości nadawy, wyniki obliczeń uzyskane z wykorzystaniem standardowych metod obliczeniowych można uznać jedynie za szacunkowe lub wstępne do przeprowadzenia badań symulacyjnych. Charakter rozruchu przenośnika taśmowego zależy od typu zastosowanego urządzenia rozruchowego i napinającego, mas wprowadzanych w ruch oraz od własności sprężystych taśmy. Problemy związane z analizą pracy układów napędowych i urządzeń napinających podczas rozruchu przenośników cięgnowych rozwiązano z wykorzystaniem modelu dyskretnego o parametrach rozłożonych (Dolipski et al., 2012; Kulinowski, 2012) opisanego w pierwszej części publikacji. Badania symulacyjne przeprowadzone na takim modelu umożliwiają ocenę pracy przenośnika taśmowego wykorzystującego taśmę z rdzeniem tkaninowym lub stalowym, dlatego też kolejny rozdział artykułu poświęcono modelom reologicznym taśm przenośnikowych. Klasyczne metody projektowania nie uwzględniają złożonych charakterystyk różnych typów układów napinania, dlatego też w kolejnej części artykułu zamieszczono sparametryzowane dyskretne modele fizyczne wybranych typów urządzeń napinających, stosowanych powszechnie

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w górniczych przenośnikach taśmowych: sztywnych, grawitacyjnych, hydraulicznych i nadążnych. Dla właściwego zobrazowania i porównania pracy opisanych układów napinania przedstawiono przykładowe wyniki badań symulacyjnych przeprowadzonych na dyskretnym modelu przenośnika taśmowego.

Przedstawiona w artykule metodyka została zweryfikowana poprzez badania przemysłowe oraz realizacje wielu złożonych projektów przenośników taśmowych wymagających przeprowadzenia badań symulacyjnych dynamiki pracy modeli urządzeń napinających taśmę (Kulinowski, 2012).

1. Introduction

The design process of belt conveyors can become a complicated task especially if you need to analyze the work of belt tension systems in difficult mining conditions. The complexity of the route profile of the mining conveyor, in combination with the various location and capacity of loading points, causes that the design calculations with using standard computational methods become less accurate. In the case of application of the analytical calculation method, presented in the previous article (Kulinowski, 2013b), for the selection of work parameters of a fixed tensioning system or gravity take-up device, calculation error will be insignificant. On the other hand, the use of hydraulic tensioning device, with limited tension force, speed and tensioning cart displacement or mechanical, proportional follow-up tensioning device, requires the use of more complex numerical calculation procedures, including simulation studies carried out using the dynamic model of the belt conveyor. This allows to analyze the operation of the conveyors with different characteristics of start-up systems, take into account the rheological properties of the belt, the variability of the loading conveyor and most importantly, allows for modeling the dynamics and to analyze the behavior of the belt take-up systems.

2. A numerical method of analysing the operating dynamics of a belt conveyor

The basic, widely used method for analysing the dynamics of transient states of conveyor belt operation is based on the assumption that all moving elements of the conveyor, i.e. those in translational motion like the transported material and the belt, and the rotating ones, i.e. the idlers and driving system subassemblies, are concentrated in one point with a the mass of and move with a constant acceleration or deceleration. However, due to the elastic characteristics of the belt, not all moving elements of the conveyor accelerate at the same time, so during the start-up, wave phenomena occur in the belt due to the progress of stress waves caused by the operation of driving systems and belt tensioners. It does happen, particularly on very long conveyors, that the time delay between putting individual sections of the belt in motion ranges from several to tens of seconds. Thus calculation results produced using standard methods for assessing the dynamics of the start-up can be considered only as approximate or preliminary for conducting more complex analytical procedures. The character of the conveyor start-up depends on the characteristics of the starter and tensioner, the masses put in motion and the elastic properties of the belt.

Problems associated with analysing the operation of take-up systems during conveyor start-up can be solved using the model of a belt conveyor with distributed parameters presented

in fig. 1 multi-mass model of a belt conveyor with distributed parameters (Dolipski et al., 2012; Kulinowski, 2012) Simulation studies carried out using this model make it possible to assess the operation of conveyor subassemblies:

– for drives with specified characteristics of starters, situated in selected locations along the conveyor route;
– using various, parameterized types of take-up systems.

2.1. A dynamic belt conveyor model

In the physical model designed for analysing dynamic phenomena (Fig. 1), the reduced masses of drives, i.e. electric motors, clutches, gears and driving pulleys including the appropriate sections of the belt, the transported material and the reduced mass of the appropriate number of idlers were concentrated in the points where drives are installed. The reduced masses associated with the upper and lower belt were concentrated, respectively, in centres of mass distributed along the conveyor route.
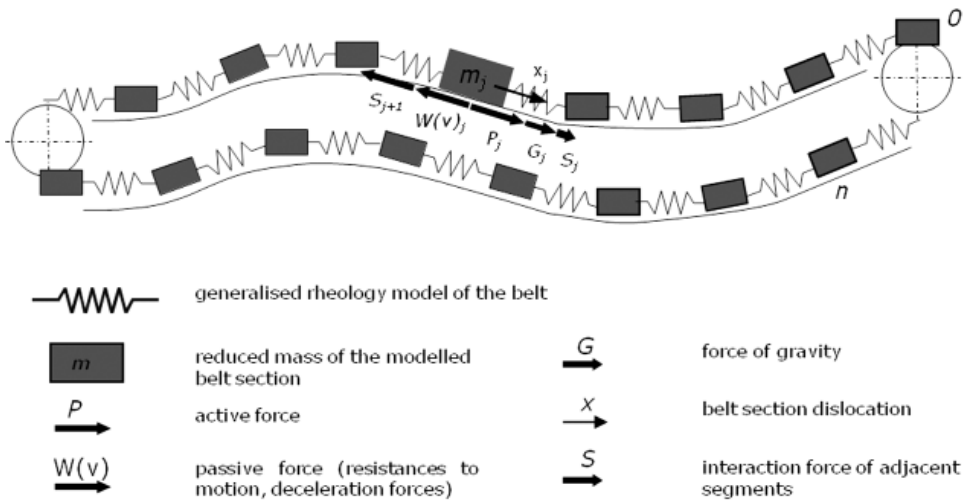

Fig. 1. Multi-mass model of a belt conveyor with distributed parameters

Variable x_i represents the dislocations of particular points on the belt ($i = 1, 2, ..., n$). The resistance to belt motion along a given section of the conveyor is represented by W_i and it is assumed that the value and the sense of the resistance force depends on the belt speed v_i. Depending on the stated purpose of the model study, the value of resistances to motion is determined using standard methods (DIN 22101, PN-M-46552) or the single resistance method (Gładysiewicz, 2003). The components of the gravity force of the belt and the transported material placed on it, tangential to the direction of belt movement, are represented by G_i, while δ_i is the angle of slope relative to the modelled level of the ith section of the belt. Active forces affecting the belt and coming from the drive are symbolised by P_i (Kulinowski, 2012).

Due to the rheological phenomena occurring in the belt during the operation of the conveyor, the standard rheological model of the belt, which is a serial combination of the Hooke and Kelvin-Voigt models, is most often adopted for analysing the single-axis state of stresses (Table 1).

The purpose of the planned simulation studies determines the correct frame of reference to be adopted for the chosen set of differential equations. If it is important to observe dynamic phenomena occurring in specific sections of the belt or splices, the values of forces, resistances to motion and masses must be made dependent on the x coordinate of the location of the belt section. However, what is most often important in research and design practice is the impact of the belt on structural elements of the conveyor, so the conveyor model is described by equation (1).

The mathematical model of a belt conveyor is described by a system of second-order ordinary differential equations. Its matrix form is as follows (Kulinowski, 2012):

$$\mathbf{M} \cdot \ddot{x}(t) = \mathbf{N} \cdot \dot{x}(t) + \mathbf{K} \cdot x(t) + \mathbf{P} - \mathbf{W} + \mathbf{G} \tag{1}$$

where:

    $\mathbf{M}$ — reduced mass matrix;
    $\mathbf{N}$ — damping coefficient matrix;
    $\mathbf{K}$ — elasticity coefficient matrix;
    $\mathbf{W}$ — resistance to motion matrix;
    $\mathbf{P}$ — active force matrix;
    $x(t)$ — dislocation matrix;
    $\mathbf{G}$ — matrix of component gravity forces.

Fig. 2 below presents a fragment of a block model of a conveyor with a two-parameter rheological model of the belt. The motion equation for the jth segment of this model is described by relationship (2).
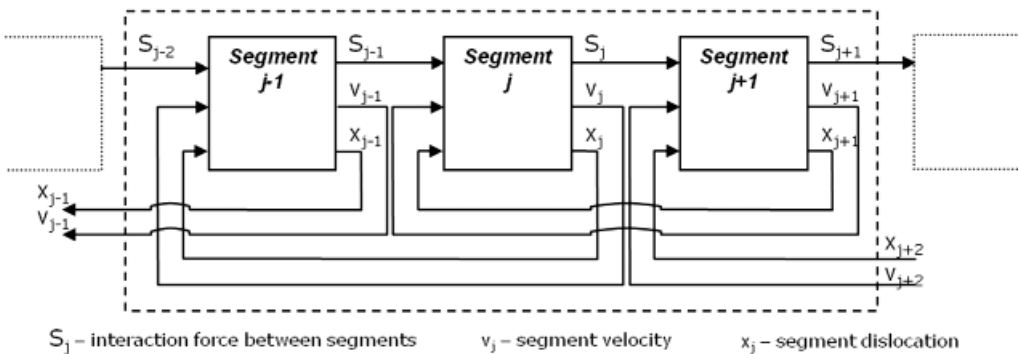

Fig. 2. A fragment of the block diagram of the model of a conveyor (Kulinowski, 2012)

If there are intermediate drives, tensioners, discharge pulleys etc. at other points of the conveyor route than in the vicinity of the head pulley or the tail pulley, reduced masses of mobile elements of these devices were taken into account and associated with the appropriate point of

the belt. If these devices are a source of active forces, the description of their operation is also included in the simulation model.

The method of modelling the driving force P_i depends on the type of start-up system employed. For the purpose of the simulation studies conducted, models of drives were developed which allow the conveyor to be started up using electromechanical drive systems widely applied in mining, which allow the following start-ups: direct, with the use of frequency converters and thyristor starters, using hydrodynamic couplings or contactor/resistance starters for wound-rotor motors.

The equation of motion for the jth segment of a conveyor with a two-parameter rheological model of the belt has the following form:

$$m_{zrj}\ddot{x}_j = S_j - W_j - S_{j-1} + G_j + P_j \tag{2}$$

where:

S_j — sum total of elasticity and damping forces in the rheological model of the belt, [N];
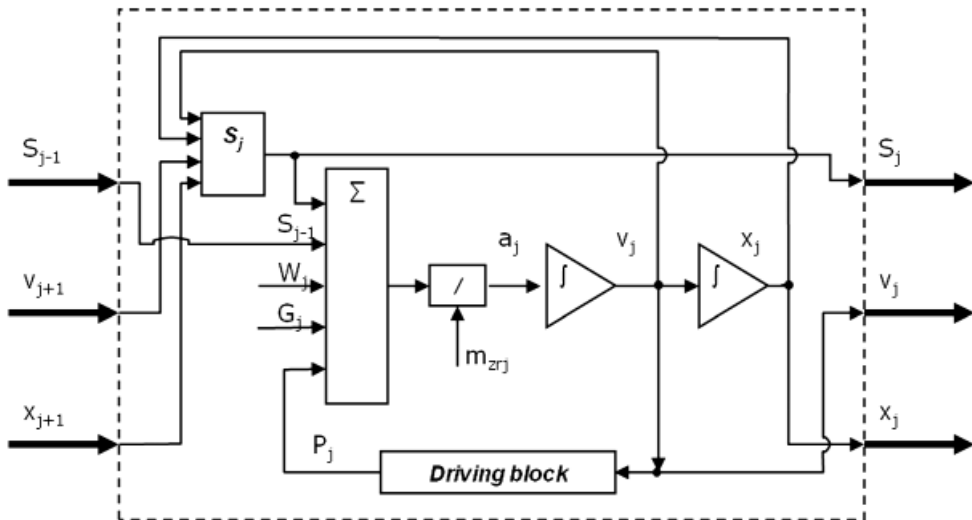
$$S_j = k_j(x_{j+1} - x_j) + \eta_j(\dot{x}_{j+1} - \dot{x}_j) \tag{3}$$

where:

k_j — elasticity coefficient of the belt rheological model, [N/m];
η_j — damping coefficient of the belt rheological model, [Ns/m];
x — dislocation of the segment, [m].

W_j – resistance to motion due to overcoming friction forces
G_j – component of the force of gravity
S_{j-1} – force of the adjacent segment's effect

Fig. 3. A block diagram of the model of a driving segment in the dynamic conveyor model
(Kulinowski, 2012)

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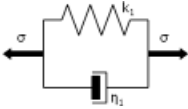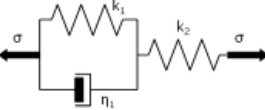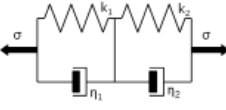## 2.2. Basic rheological models of conveyor belts

The belt model adopted for simulation studies is of major significance in designing, selecting and analysing the operation of belt tensioning systems. The components used to produce conveyor belts with fabric carcasses have the characteristics of large-molecule polymers with a 3D structure, and for belts with steel cords – of composites. Thus a conveyor belt is an anisotropic body whose mechanical characteristics are rheological. The assumption that the rheological model of the belt selected for studies is linear suggests adopting the following simplifications (Markusik, 1998):

    – deformations and stresses in the belt are small (maximum 15% of the rupturing force);
    – a single-axis stress state occurs in the belt;
    – the mechanical properties of the model are subject to the Bolzmann superposition principle.

Simulation studies of the operation of belt conveyors include belt properties described by the most frequently used models compiled in Table 1 (Furmanik & Zarzycki, 2011; Jabłoński, 1988; Markusik, 1998; Nordell & Ciozda, 1984; Pypno, 1994).

TABLE 1

The rheological models of conveyor belts used (Kulinowski, 2012)

Two-parameter Kelvin-Voigt model	The Kelvin-Voigt model represents energy dissipation and stress relaxation phenomena. The model is used to analyse short-term, transient states of belt conveyor operation.

Standard three-parameter model	The standard model is a serial combination of the Hooke and Kelvin-Voigt models. It represents such characteristics of conveyor belts as:
	  – immediate flexibility;
	  – stress relaxation;
	  – energy dissipation;
	  – creep.

Four-parameter model	The four-parameter model, which is a serial combination of two Kelvin-Voigt models, is used for analysing both short-term, transient dynamic states and longer periods of steady-state operation of belt conveyors.
	It represents such characteristics of conveyor belts as:
	  – immediate flexibility;
	  – stress relaxation;
	  – energy dissipation;
	  – creep.

The selection of the belt rheology model depends on the purpose of the simulation studies. If the analysis covers short-lasting, transient states of conveyor operation occurring during its start-up and breaking, the two parameter Kelvin-Voigt model with a short stress relaxation time calibrated during simulation tests is sufficient.

If the analysis concerns changes occurring during the steady-state operation of the conveyor with a variable load of handled material, it is necessary to use a standard model or a four-parameter one, which is a serial combination of two Kelvin-Voigt models (Kulinowski, 2012).

Rheological models which account for permanent deformations of the belt, i.e. the four-parameter Burgers model and the five-parameter model are not used at present (Kulinowski, 2012).

2.3. Physical models of take-up systems

2.3.1. A fixed take-up system

A rigid belt tensioner is a mechanical system designed for tensioning the belt when the conveyor is stopped. When the conveyor is running, the tensioning pulley remains stationary, so the main function of this mechanism is to offset permanent belt elongations (Kulinowski, 2013b).

The model describing the operation of a rigid tensioner assumes that before conveyor start-up, the tensioning cart with the pulley is moved to create the force of the so-called preliminary
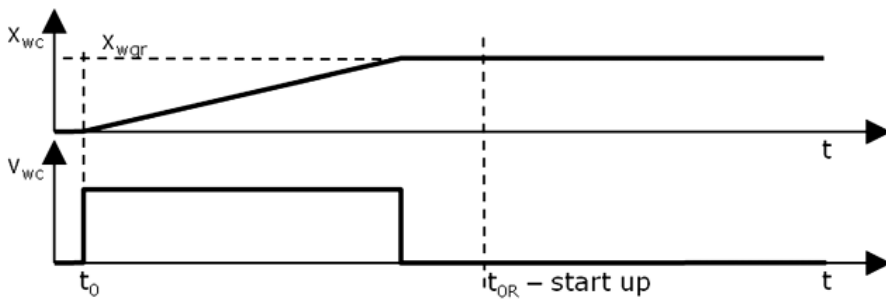

Fig. 4. The course of the change of the location and velocity of the tensioning pulley (Kulinowski, 2012)

tension on the belt. The location of the belt will change at the constant velocity of v_{wc} from the starting moment t_0 until the moment the tensioning cart reaches the position x_{wgr} (Fig. 4). While the conveyor operation is simulated, the tensioning pulley remains stationary ($v_{wc} = 0$).

In the physical model of a rigid tensioning device presented in Fig. 5 the velocity of the additional point v_d is given by the relationship:

$$v_d = v_{i+1} + 2v_{wc} \ [m/s] \tag{4}$$

where:

v_d —— additional point speed, [m/s];
v_{wc} —— winch velocity, [m/s];
v_{i+1} —— speed of the centre of gravity $i+1$, [m/s].

The calculated value of the speed and dislocation of an additional point x_d is substituted in the differential equation system describing the conveyor model.
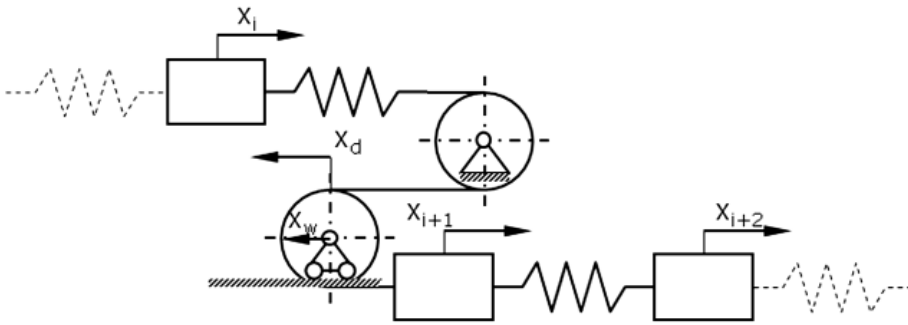
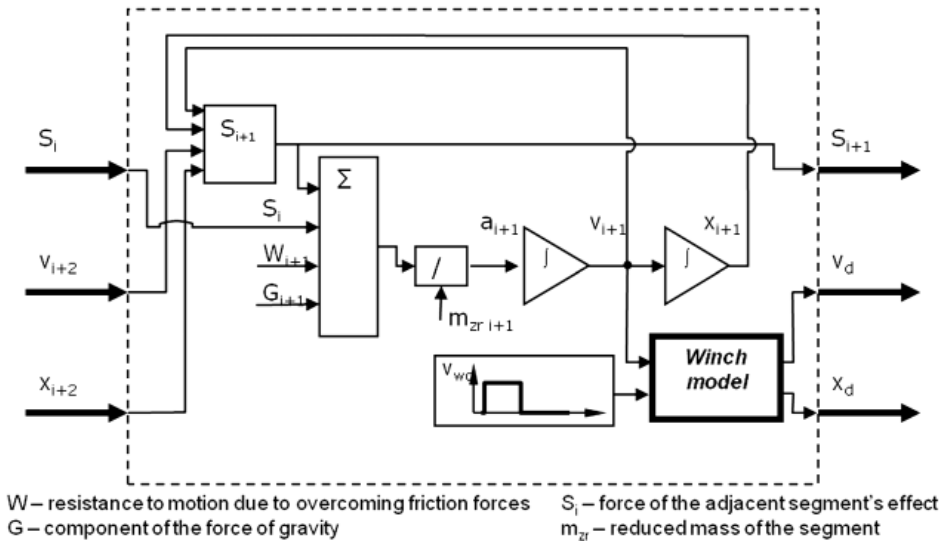Fig. 5. The physical model of a rigid tensioner (Kulinowski, 2012)

W – resistance to motion due to overcoming friction forces S_i – force of the adjacent segment's effect
G – component of the force of gravity m_{zr} – reduced mass of the segment

Fig. 6. A block diagram of the model of a rigid tensioning segment in the dynamic conveyor model
(Kulinowski, 2012)

2.3.2. A gravity take-up system

A physical model of a gravity tensioning system is illustrated by the diagram in Fig. 7 and the model of the segment in a dynamic conveyor model is presented in Fig. 8 (Kulinowski, 2013b).

The acceleration of the weight is determined based on the following relationship:

$$a_o = \frac{2 \cdot S \cdot i_{zl} \cdot \eta_{zl}}{m_o \cdot n_o} - g \quad [\text{m/s}^2] \tag{5}$$

where:
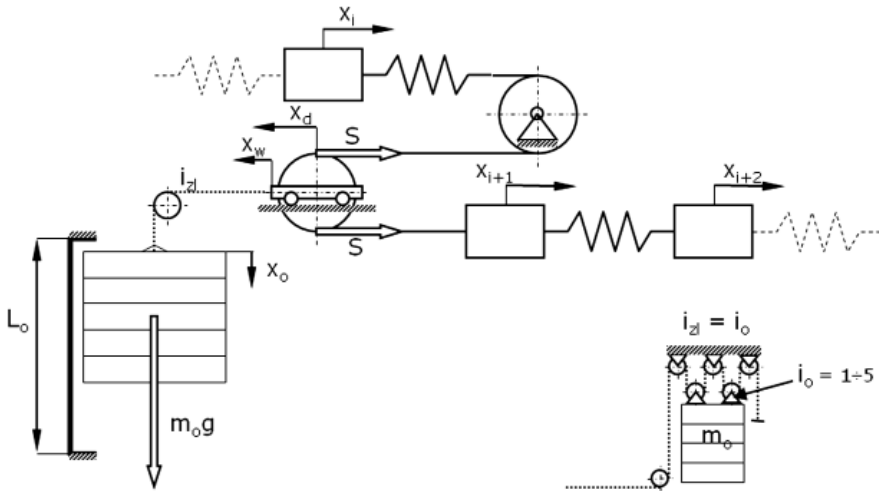
m_o — weight mass, [kg];
n_o — number of weights, [-];

Fig. 7. The physical model of a gravity tensioner (Kulinowski, 2012)

W – resistance to motion due to overcoming friction forces S_i – force of the adjacent segment's effect
G – component of the force of gravity m_{zr} – reduced mass of the segment
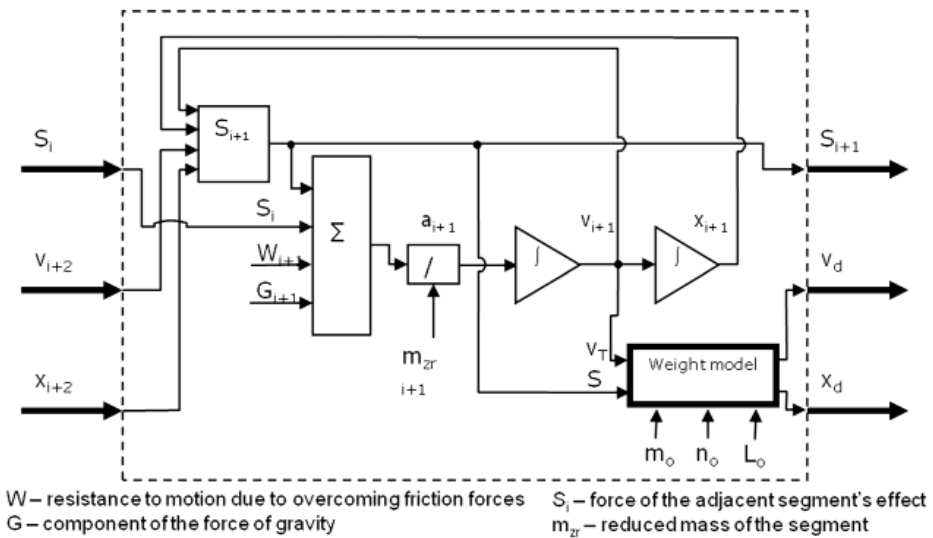
Fig. 8. A block diagram of the model of a gravity tensioning segment in the dynamic conveyor model
(Kulinowski, 2012)

S — force in the belt, [N];
i_{zl} — ratio of the tackle system, [-];
η_{zl} — tackle system efficiency $\eta_{zl} = f(v_o)$, [-].

After performing an integration operation to determine the acceleration of the weight $- a_o$, its dislocation $- x_o$, and after accounting for the structural limitations of the length of the tensioning

path L_o, the value of the dislocation of the tensioning cart is calculated. The calculation of this value accounts for the ratio of the tackle system and its efficiency, which varies depending on the speed of the weight v_o. The value of the velocity of the additional point v_d was determined from relationship (4).

Operating parameters of a gravity-based tensioning system are adjusted using the number and the mass of weights as well as the tensioning path length.

2.3.3. A hydraulic belt tensioning system

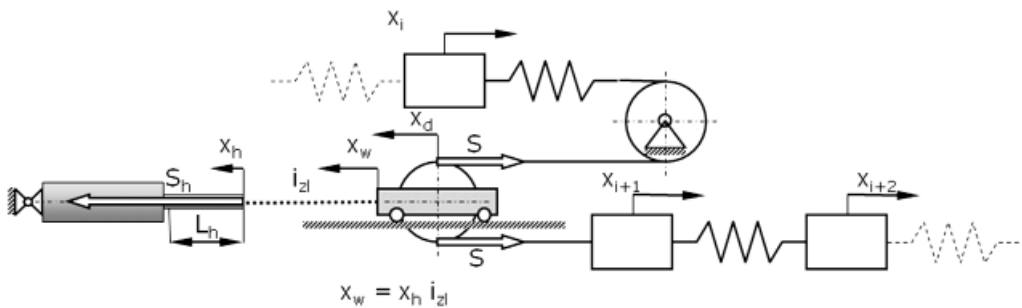Fig. 9 presents the physical model of a hydraulic take-up system (Kulinowski, 2013b).

Fig. 9. The physical model of a hydraulic tensioner (Kulinowski, 2012)

The acceleration of the piston of the cylinder is determined based on the following relationship:

$$a_h = \frac{S_h - 2 \cdot S \cdot \dfrac{i_{zl}}{\eta_{zl}}}{m_w \cdot i_{zl}^{\,2} + m_h} \quad [\text{m/s}^2] \tag{6}$$

where:
m_w — the mass of the cart with the tensioning pulley, [kg];
m_h — the mass of the moving part of the tensioning system connected with the piston, [kg];
S_h — the piston force, [N];
S — the force in the belt, [N];
i_{zl} — the ratio of the tackle system, [-];
η_{zl} — the tackle system efficiency $\eta_{zl} = f(v_o)$, [-].

During iterative calculations, after the operation of integrating the acceleration of the cylinder piston – a_h, its velocity is determined. During the real operation of a hydraulic tensioning system, the velocity of piston extension – v_h is restricted by the capacity of the hydraulic system, so the maximum permissible speed of piston extension was included in the theoretical model. The length of the piston stroke – L_h can also restrict the operating ability of the tensioning system,

so the location of the piston end x_h is controlled and when the extreme position is reached, the cart's ability to travel further is blocked.

The velocity of the additional point v_d was determined, just as for the remaining tensioning systems, based on relationship (4).

The work of the hydraulic tensioner model is controlled by the value of the tensioning force S_h and by restricting the velocity and length of the tensioning path – $v_{h\,max}$, L_h (Fig. 10).

W – resistance to motion due to overcoming friction forces S_i – force of the adjacent segment's effect
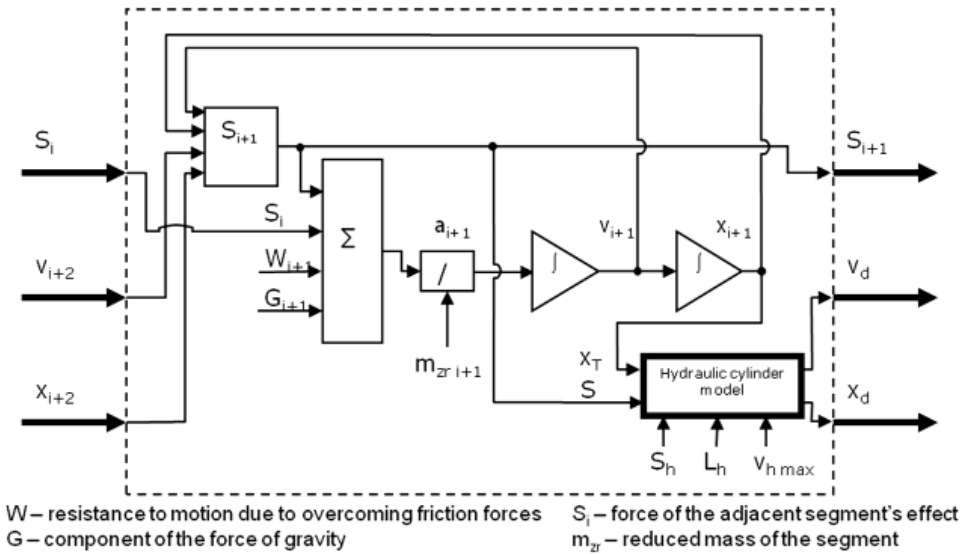G – component of the force of gravity m_{zr} – reduced mass of the segment

Fig. 10. A block diagram of the model of a hydraulic tensioning segment in the dynamic conveyor model (Kulinowski, 2012)

2.3.4. Follow-up mechanical belt tensioning system

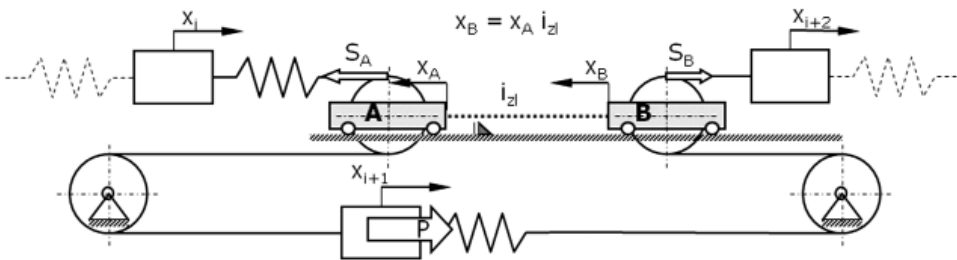The physical model of a follow-up, mechanical belt tensioner (Kulinowski, 2013b) is shown in Fig. 11.

Fig. 11. A physical model of a mechanical proportional tensioning system (Kulinowski, 2012)

The acceleration of cart A, located in the zone of large forces on the belt, is determined based on the following relationship:

$$a_A = \frac{2(S_A - S_B \cdot i_{zl} \cdot \eta_{zl} - F_{tl}}{m_A + m_B \cdot i_{zl}^2} \quad [\text{m/s}^2]$$

(7)

where:

$m_{A,B}$ — masses of carts with tensioning pulleys, [kg];
S_A — force in the belt in the high force zone (coming onto the pulley), [N];
S_A — force in the belt in the low force zone (coming off the pulley), [N];
F_{tl} — damping force, [N];
i_{zl} — the ratio of the tackle system, [-];
η_{zl} — tackle system efficiency $\eta_{zl} = f(v_A)$, [-].

Diagrams in Fig. 12 and Fig. 13 show a block model of a follow-up, proportional belt tensioning system.

S_i – interaction force between segments v_i – segment velocity x_i – segment dislocation
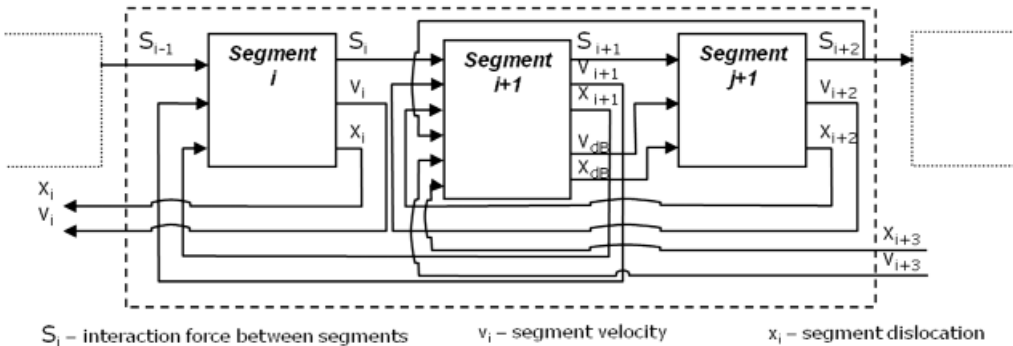
Fig. 12. A fragment of the block diagram of the model of a conveyor with the module of a proportional mechanical tensioning system (Kulinowski, 2012)

3. Simulation studies of tensioning systems

Below are presented the simulation results of work of tensioning devices used in mining belt conveyors. The simulation tests were carried out on a discrete model of the horizontal, fully loaded belt conveyor with a length of 1,200 m. The drive system (2 × 250 kW) was mounted on the head of the conveyor, and the tensioning device was located in the area of the belt running-off the driving pulleys. Both motors were run at the same time by starting contactor. To assess the effectiveness of the tensioning system work, based on the simulation results, the static characteristics of tensioners were determined (Fig. 14b, Fig. 15b, Fig. 16b, Fig. 17b).

In the case of a rigid tensioning system (Fig. 14) can be observed the work of drive without slip, resulting from the high value of a initial belt tension (> 200 kN). During stationary work of
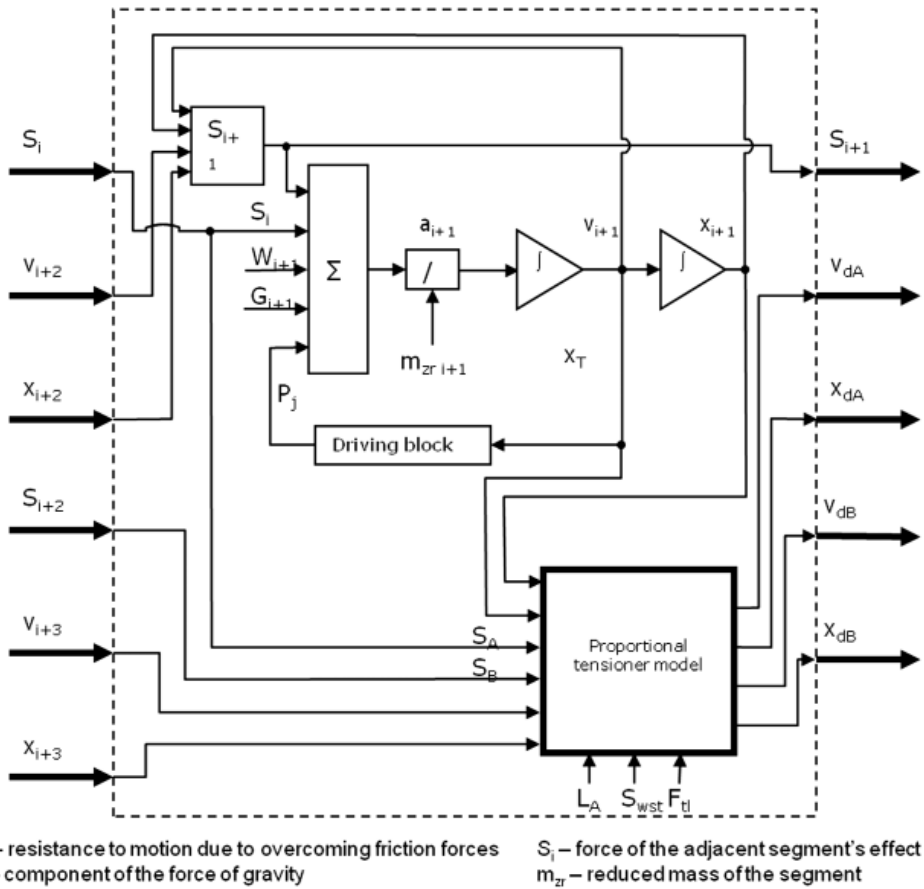
W – resistance to motion due to overcoming friction forces S_i – force of the adjacent segment's effect
G – component of the force of gravity m_{zr} – reduced mass of the segment

Fig. 13. A block diagram of the model of a follow-up proportional tensioning segment
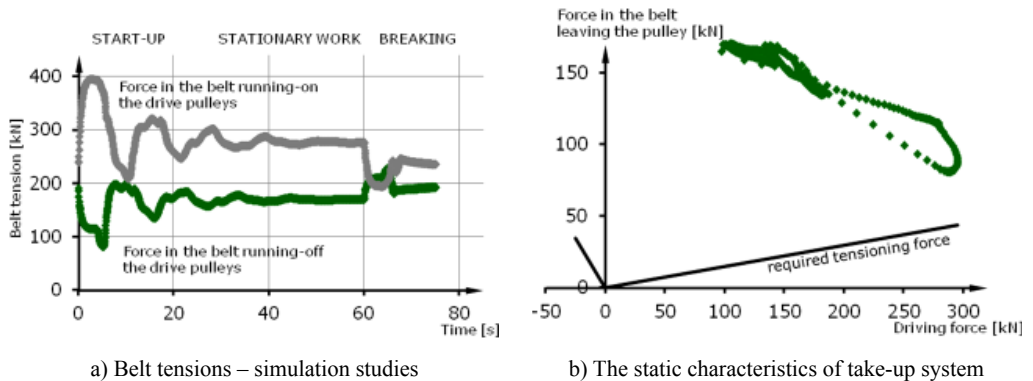in the dynamic conveyor model (Kulinowski, 2012)

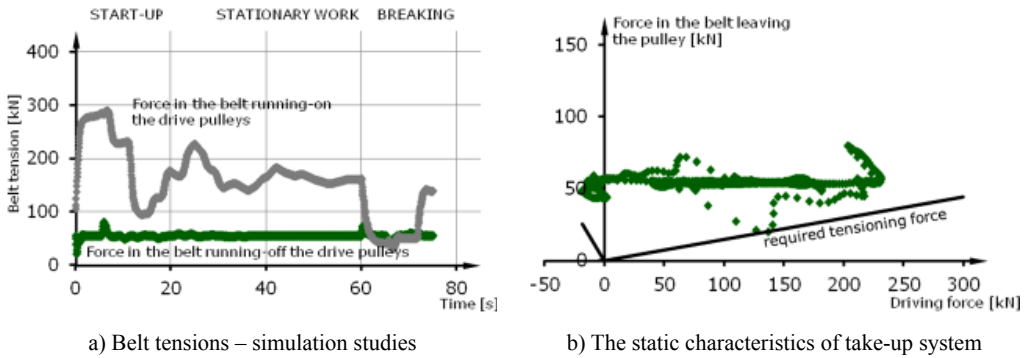a) Belt tensions – simulation studies   b) The static characteristics of take-up system

Fig. 14. The fixed take-up system

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a) Belt tensions – simulation studies   b) The static characteristics of take-up system

Fig. 15. The gravity take-up system

a) Belt tensions – simulation studies   b) The static characteristics of take-up system

Fig. 16. The hydraulic belt tensioning system

a) Belt tensions – simulation studies   b) The static characteristics of take-up system
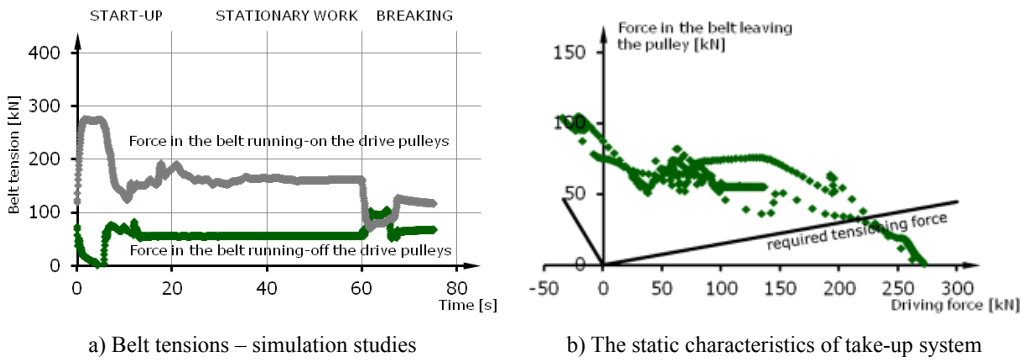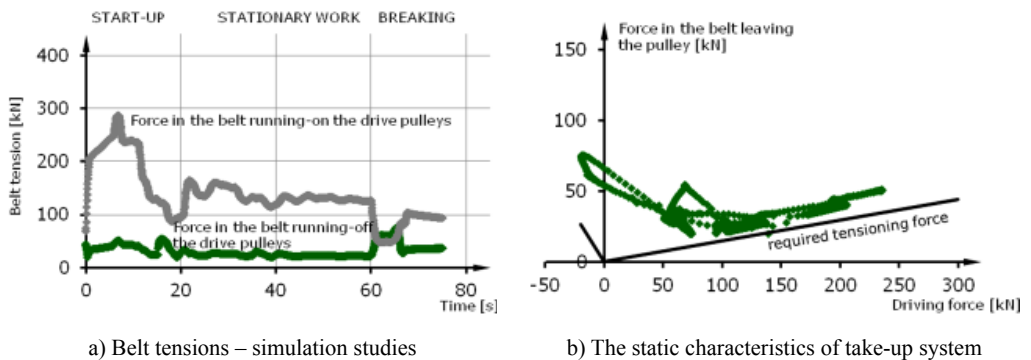
Fig. 17. The follow-up mechanical belt tensioning system

the conveyor there is a large surplus of the belt tension relative to the value required. Maximum force in the belt during start-up reaches a value of about 400 kN.

For subsequent types of tensioning devices was reduced significantly the tensioning force values, which reduced the maximum force in the belt to the level of 300 kN. In the case of the gravity tensioning system, stabilizing a force in the belt at the level of 50 kN, the conveyor worked properly during starting, stationary work and braking (Fig. 15). On the other hand, the hydraulic tensioning system, due to reduced flow of the hydraulic system caused a temporary drop of the belt tension below the required value, which led to a temporary slip of the belt on the drive pulley (Fig. 16).

The follow-up mechanical belt tensioning system assured correct, minimal belt tension throughout the period of the conveyor work, by adjusting the level of tension force proportional to the changing driving force (Fig. 17).

4. Summary

An analysis of the tensioner requires conducting a series of simulation studies, so a dynamic, multi-mass model of a belt conveyor with distributed parameters which accounts for the rheological properties of conveyor belts forms an important part of the proposed method. This article contains a detailed description of parameterized dynamic models of belt tensioning systems applied in mining.

An interesting example of using a dynamic model of a belt conveyor to design a new type of belt tensioners consisted in a series of simulation tests carried out on a model of a follow-up belt tensioning system (Kulinowski & Wójcicki, 2013). The most interesting observation was the possibility of conducting the start-up with the initial tension three times lower than in the case of working with a winch only. In steady-state operation, the force in the belt of a conveyor with a follow-up tensioning station is equal to between 40% and 80% of the value of the force in the belt of a conveyor with a winch tensioner, for a comparable load of transported material. This means that the use of follow-up belt tensioners instead of devices with a stationary tensioning pulley during the operation of a conveyor will allow the durability of the belt, its splices and other structural elements of the conveyor to be significantly extended (Kulinowski & Wójcicki, 2013).

References

Antoniak J., 1990. *Urządzenia i systemy transportu podziemnego w kopalniach*. Wyd. „Śląsk”, Katowice.

Dolipski M., Remiorz E., Sobota P., 2012 *Determination of dynamic loads of sprocket pulley teeth and seats by means of a mathematical model of the longwall conveyor.* Arch. Min. Sci., Vol. 57, No 4, p. 1101-1119.

Furmanik K., Zarzycki J., 2011. *O modelowaniu własności reologicznych taśm przenośnikowych.* Transport Przemysłowy i Maszyny Robocze, 1(11): 23-28.

Gładysiewicz L., 2003. *Przenośniki taśmowe. Teoria i obliczenia*. Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław.

Jabłoński R., 1988. *Analiza i synteza urządzeń napinających taśmę w przenośnikach taśmowych dużej mocy*.

Kulinowski P., 2008. *Identyfikacja parametrów techniczno-ruchowych przenośników taśmowych z wykorzystaniem mobilnego systemu pomiarowego.* Maszyny Górnicze, 3: 35-43.

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Kulinowski P., 2012. *Metodyka zintegrowanego projektowania górniczych przenośników taśmowych*. Wydawnictwa AGH.

Kulinowski P., 2013. *Badania symulacyjne jako element zintegrowanego procesu projektowania w aspekcie eksploatacji przenośników taśmowych*. Eksploatacja i Niezawodnosc – Maintenance and Reliability, 1.

Kulinowski P., 2013. *Analytical Method of Designing and Selecting Take-up Systems for Mining Belt Conveyors.* Arch. Min. Sci., Vol. 58, No 4, p. 1301-1315.

Kulinowski P., Wójcicki W., 2005. *Pętlicowe nadążne urządzenie napinające taśmę dla podścianowych przenośników taśmowych kopalń węgla kamiennego*. Transport Przemysłowy, 1(19): 7-9.

Markusik S., 1998. *Reologia taśm przenośnikowych*. Gliwice.

Nordell L., Ciozda Z., 1984. *Transient belt stresses during starting and stopping: Elastic response simulated by finite element methods.* Bulk Solids Handling, 4(1): 99-104.

Pypno C., 1994. *Wpływ temperatury na własności mechaniczne taśm przenośnikowych*. Praca doktorska, Politechnika Śląska, Katowice.

Żur T., Hardygóra M., 1996. *Przenośniki taśmowe w górnictwie*. Wyd. „Śląsk”, Katowice.