DOI: 10.21440/0536-1028-2026-5-100-108

Modeling the dynamics of a belt conveyor drive with a compensating take-up unit

Alexander A. Reutov[1]
[1] Bryansk State Technical University, Bryansk, Russia
e-mail: bgtu2012@yandex.ru

Abstract

The research relevance *stems from the need to improve the efficiency of multi-drive systems of a belt conveyor.*

The research objective *is to determine the dynamic characteristics of a belt conveyor drive with a compensating take-up unit.*

The methods of research *included analyzing the scientific data; computer-aided modeling of a belt conveyor drive as a multi-mass dynamic system; modeling four operating modes of a drive; analyzing the modeling results and drawing conclusions.*

Results. *A computer-aided dynamic model of a two-drive system of a belt conveyor with a compensating take-up unit was developed using the Universal Mechanism software package. The model takes into account the viscoelastic properties of the belt, the nonlinear dependence of friction force on sliding speed, and the elasticity of the contacting surfaces. The torques of the drive pulleys under steady-state operation are represented by linear dependencies on the pulley angular velocities, while under uncontrolled start-up conditions, they are represented by nonlinear dependencies on time. Modeling examples of transient processes with step load changes and uncontrolled start-up of a drive with asynchronous electric motors allowed to determine the lowest natural oscillation frequency, percent overshoot, and damping decrement of transient oscillations under step changes in resistance force.*

Conclusions. *A drive with a compensating take-up unit is a stable servo control system with a limited range of tension and traction force variations. The compensating take-up unit changes the belt tension force as the total resistance to its motion changes, allowing for a limited increase in the drive's traction capacity as the conveyor load increases. The compensating take-up unit does not reduce the dynamic loads in the belt during conveyor start-up.*

Keywords: *belt conveyor; conveyor belt; drive pulley; friction drive; take-up unit; computer-aided model.*

Introduction. Modern belt conveyors at mining enterprises with a drive power of over 80–100 kW typically have two or three drives installed at the head section. Installing multiple standardized drives increases drive traction under limited belt tension and reduces the drives weight and dimensions as well as the cost of spare parts.

Multi-drive systems for belt conveyors have been the subject of extensive research. Paper [1] developed the most comprehensive mathematical model for the steady-state operation of a multi-drive system. The model contains equations relating the traction forces of the drive pulleys to the characteristics of the asynchronous electric drive and the conveyor belt. The model also contains inequalities reflecting the limitations of the drive pulleys traction force, the drive power of the electric motors, and the belt strength. The distribution of traction forces in two-drive and three-drive systems is obtained. Requirements for the design and operation of the belt conveyor multi-drive systems are provided in the DIN 22101 standard [2, 3], among other standards.

During operation, the overall resistance to motion of the conveyor belt changes as the conveyor load and ambient temperature change, and bearings and seals on rollers and pulleys degrade. Load-type and spring-type take-up units fall into the category of simple automatic take-up units. However, their belt tension adjustment range does not meet the requirements of heavy-duty belt conveyors. Research [4] proposed optimizing the design of an automatic take-up unit to reduce its dimensions while ensuring the required reliability.

Research [3] examined a compensating take-up unit of the 2L80 conveyor, consisting of two take-up pulleys connected by a block-and-tackle pulley system (Figure 1). The pulley system rope tension is controlled by a hydraulic sensor. The conveyor drive is structurally connected to the take-up unit, therefore is impossible to analyze the drive's operation separately from the take-up unit.

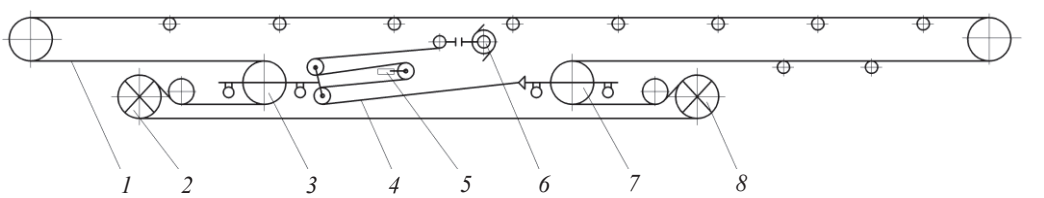

Figure 1. A conveyor drive scheme: *1* – belt; *2* – first drive pulley; *3* – left take-up pulley; *4* – the rope of the block-and-tackle pulley system; *5* – hydraulic sensor; *6* – winch of the take-up unit; *7* – right left take-up pulley; *8* – second drive pulley

Рисунок 1. Схема привода ЛК: *1* – лента конвейера; *2* – первый приводной барабан; *3* – левый натяжной барабан; *4* – канат полиспаста; *5* – гидродатчик; *6* – лебедка натяжного устройства; *7* – правый натяжной барабан; *8* – второй приводной барабан

A compensating take-up unit can also be manufactured with a gearbox and two rope pulleys instead of a block-and-tackle pulley system [5]. The compensating take-up unit changes the belt tension force as the conveyor load changes, thereby automatically adjusting the drive's traction capacity. It is reasonable to install a compensating take-up unit on a belt conveyor whose total resistance decreases with decreasing belt tension force.

A force analysis of a drive with a compensating take-up device is considered in research [6]. From the take-up pulley equilibrium conditions, it follows that ratio k_n of the take-up unit's block-and-tackle is limited by the top and bottom. Exceeding the maximum reeving ratio of the block-and-tackle causes the belt to slip on the drive pulleys. The ratio underestimation results in incomplete utilization of the drive's traction capacity and the need for increased belt tension force as well as stronger belts.

The operation of a drive with a compensating take-up unit was analyzed in works [5, 6] using a quasi-static formulation neglecting the inertial forces and dissipative properties of the compensating take-up unit. It was shown that the change in the traction force of a drive with a compensating take-up unit occurs due to changes in the belt tension force. The obtained dependences are not suitable for calculating the conveyor's start-up and other non-stationary operating modes.

Dynamic processes during conveyor start-up were studied in researches [7–11]. Theoretical dependences for the transient processes of loaded and unloaded conveyor were obtained in paper [12] using Mathcad. Transient processes for a conveyor with an electric drive are 3.5 times as long as those with a hydraulic drive. The dynamic coefficient of the hydraulic drive is 1.78 times smaller than that of the electric drive.

Researches [13–15] analyze the advantages of asynchronous conveyor drives with frequency control. Experimental measurements showed the absence of significant oscillations in the belt tension force during acceleration.

Methods for automated belt tension monitoring and control are discussed in work [16]. In paper [17], four operating modes of a belt conveyor were simulated using the MSC Adams software package: belt speed changes, sudden load increases, changes in start-up duration, and emergency braking. The model results demonstrated the effectiveness of the proposed belt tension control strategy and reduced energy consumption. Energy-efficient belt conveyor designs are also discussed in research [18].

The research objective is to determine the dynamic characteristics of a belt conveyor drive with a compensating take-up unit.

The description of the computer-aided model. Let us consider a belt conveyor drive consisting of two drives and a compensating take-up unit [5]. Each drive contains a drive pulley connected to the electric motor through a gearbox.

Modeling the conveyor drive dynamics is possible using the Universal Mechanism software package, MSC Adams, and others [19, 20]. A computer-aided model of the conveyor drive with a compensating take-up unit was created in the Universal Mechanism software package. The model includes a closed belt, looping around two drive pulleys, two take-up pulleys, a brake pulley, and a bend pulley. A fixed stop limits the leftward travel of the left take-up pulley. The diameters, distances between axes, masses, and moments of inertia of the pulleys are equal to the corresponding values of the 2L80 conveyor drive. The belt is presented in the model by 250 bodies connected by elastic-dissipative elements (hereinafter referred to as belt elements) with a stiffness of C_0, N, and a dissipation coefficient of μ_0, N·s/m. The belt tension is achieved by moving the right and left take-up pulleys under the action of the belt and compensating take-up unit forces. The initial belt tension in the computer-aided model is achieved by setting the position of the pulleys in accordance with the belt length.

The compensating take-up unit is represented by two rack mechanisms whose gear wheels share a common shaft. The gear ratio of the compensating take-up unit k_n (analogous to ratio of the take-up unit's block-and-tackle) is determined by the ratio of the gear wheels radii. The travels of the right and left take-up pulleys Δ_3 and Δ_4, m, are related by the relationship $\Delta_4 = k_n\Delta_3$ ($k_n > 1$).

The brake pulley represents the inertial properties of the conveyor linear part. The moment of inertia of the brake pulley corresponds to the kinetic energy of the conveyor linear part.

The contact interaction of the belt with the pulleys is represented by "point-circle" contact forces, described in the user manual for the Universal Mechanism software package [20].

For an asynchronous electric drive with standardized non-adjustable drives, the torque of the *i*-th drive pulley M_i, N·m, is represented by the following relationship:

$$M_i = \frac{M_{nom}}{\varepsilon_k}\left(1 - \frac{\omega_i I}{\omega_s}\right); \qquad i = 1, 2, \tag{1}$$

where M_{nom} is the nominal torque of the drive pulley, N·m; ε_k is the nominal electromechanical slip of the electric motor; ω_i is the angular velocity of the *i*-th drive pulley, rad/s; ω_s is the synchronous angular velocity of the electric motor, rad/s; I is the gear ratio of the drive.

Linear dependence (1) is well suited for slowly changing the angular velocity of the electric motor's rotor. During uncontrolled starting of an asynchronous electric motor, torque oscillations occur. The torque of the i-th drive pulley is represented by the following dependence:

$$M_i = M_{nom}\left[k_d e^{-\beta t}\sin(kt+\varphi_i)+1-e^{-\gamma t}\right]; \qquad i = 1, 2, \tag{2}$$

where t is time, s; k_d is the initial amplitude of the sinusoidal component; β, γ – coefficient of exponential decay of M_i over time; k is the angular frequency of the sinusoidal component, rad/s; φ_i is the initial phase of the sinusoidal component, rad; the parameters k_d, β, k, φ_i, and γ are determined experimentally or by modeling the electrodynamic processes of starting similar electric motors.

The torques resisting pulley rotation M_{Ci} are represented by the relationship:

$$M_{Ci} = \left(\varsigma_0 + \varsigma_1\omega_i\right) sign\left(\omega_i\right),$$

where ς_{0i} and ς_{1i} are the resistance coefficients, in N·m and N·m·s/rad, respectively.

The resistance to motion of the belt on the linear part of the conveyor F_C, N, is accounted for by the resisting torque M_C, N·m, to the rotation of the brake pulley.

Drive simulation. Four drive operating modes were simulated: under a monotonic increase in the resisting torque M_C, under a step change in M_C, under the start-up of a drive with the compensating and rigid take-up units, and under belt coasting after the drive shutdown.

The simulation was carried out with the following parameters: linear mass per meter of the belt is 20 kg/m; C_0 = 32 MN, μ_1 = 500 N·s/m; the radii of the pulleys are the same, r_i = 0.645 m; masses of the take-up pulleys with sliders $m_3 = m_4$ = 200 kg; mass of the brake pulley with a slider m_6 = 1800 kg; moments of inertia of the pulleys J_i = 120 kg·m^2 (i = 1…5), J_6 = 3000 kg·m^2; M_{nom} = 17.4 kN·m; ω_s = 3.225 rad/s; ε_k = 0.04; ς_{0i} = 1.29 N·m and ς_{1i} = 0.645 N·m·s/rad (i = 1…5); k_n = 4; ς_{06} = 150 N·m; ς_{16} = 5.0 kN·m·s/rad. The coefficients of belt friction on the pulley surfaces under zero sliding speed and under infinite sliding speed are f_0 = 0.3 and f_∞ = 0.31; the stiffness coefficient of the brake pulley slide spring is 2.0 MN/m.

Mode 1. Under a smooth change in the resisting torque M_C, the take-up and brake pulleys move in the *Y*-axis direction. Longitudinal tension-compression waves of the belt cause slight oscillations in the pulley travel speed.

The compensating take-up unit changes the belt tension force almost synchronously under changes in M_C. The misalignment of the $S(t)$ and $M_C(t)$ functions change phases is caused by the belt tension-compression waves, as well as by the nonlinearity of the model, specifically, the limitation of the left take-up pulley travel by the left stop.

The compensating take-up pulley allows for a limited increase in the drive's traction capacity as the resistance to motion of the belt increases. Simulation confirms the equality of the magnitudes of change in the resistance force F_C and ($S_{1nb} - S_{2cb}$), where S_{1nb} and S_{2cb} are the tensile forces of the belt branch running onto the first drive pulley and running off the second drive pulley, N. Changing k_n within acceptable limits changes the ratio of the left and right take-up pulleys travels, but does not affect the steady-state tension distribution along the belt length.

Mode 2. Let us determine the characteristics of the transient process under a step change in the resisting torque M_C by 20 kN·m. A step change in the resistance to motion of the belt occurs under the fast unloading of a dump truck or railcar into the conveyor's loading hopper. Under a step change in M_C, the travels of the take-up and brake pulleys are oscillatory.

Figure 2 shows graphs of the change in the force of the rack mechanism acting on the slider of the right take-up pulley F_3, kN, the total traction force of the drive pulleys F_Σ, kN, and the resistance force F_C, kN, under a step change in the resisting torque M_C at time t = 1 s.

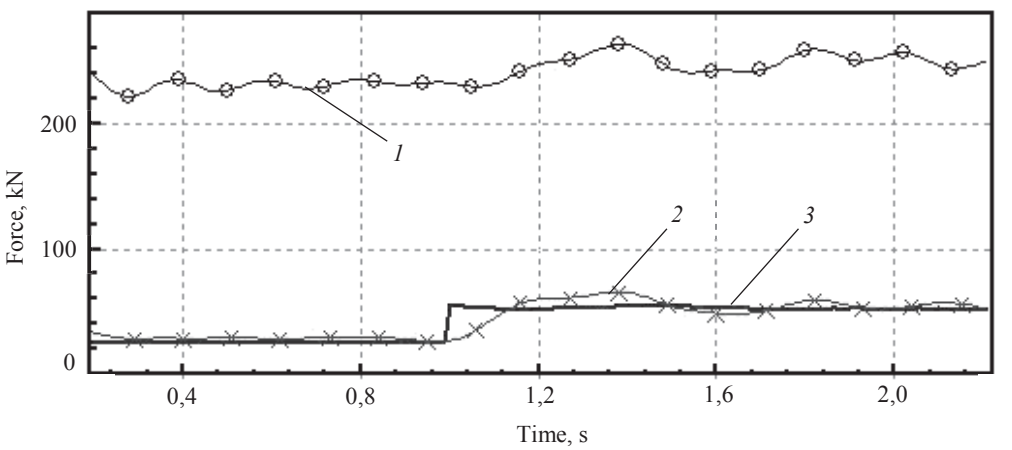

Figure 2. Graphs of the change in the force of the rack mechanism F_3 acting on the slider of the right take-up pulley – *1*, the total traction force of the drive pulleys – *2*, the resistance force F_C – *3*

Рисунок 2. Графики изменения силы действия реечного механизма F_3 на ползун правого натяжного барабана – *1*, суммарной силы тяги F_Σ приводных барабанов – *2*, силы сопротивления F_C – *3*

The graphs in Figure 2 were constructed after applying a low-pass filter of up to 5 Hz to the simulation data. For the considered example, the lowest frequency of the model's natural oscillations, corresponding to the travel of the take-up and brake pulleys in the *Y*-axis direction, is 45.7 Hz, and the damping ratio is 2.06%. The lowest frequency and damping decrement of the force F_3 oscillations after a step change in the resisting torque M_C are 1.56 and 2.58 Hz, respectively; the percent overshoot of the F_3 force is 7.54%. The values of the lowest frequency of the drive pulley angular velocity oscillations after a step change in M_C are 7.78 and 6.4 Hz, the damping decrement is 5.5 and 1.08, the percent overshoot of ω_1 and ω_2 is 2.84% and 0.9%.

Mode 3. Let us determine the belt tensile force during the start-up of a drive with a compensating and rigid take-up units under the same parameters, except for ς_{06} = 258 N·m; ς_{16} = 8.6 N·m·s/rad. The torques of the drive pulleys during the uncontrolled start-up of asynchronous electric motors are represented by dependence (2) under M_{nom} = 13.5 kN·m; k_d = 1.5; k = 12 rad/s; β = 2; γ = 3; φ_i = 0. The initial conditions for simulating the conveyor start-up were prepared by preserving the values of all coordinates when simulating the conveyor stop, i.e., the free coasting of the belt after the drive shutdown.

Tensile force of the belt element no. 50 S_{50}, located in front of the first drive pulley at the start-up, reaches the maximum value of 109.3 kN. The minimum $S_{\min}$ and maximum $S_{\max}$ values of the belt tensile force are determined as the average for four equally spaced belt elements: $S_{\min}$ = 37.6 kN; $S_{\max}$ = 83.9 kN.

The dynamic coefficient of the belt tensile force K_D is calculated through S_{50} and $S_{\max}$:

$$K_D = S_{50} / S_{\max} = 1,30.$$

To simulate the start of the conveyor with a rigid take-up unit, the sliders of the take-up and brake pulleys are fixed in positions corresponding to the belt initial tension. Fixing the pulley sliders did not change the values of K_D and $S_{\max}$ in this example. Under increased oscillation frequency of drive pulley torques k to 60 rad/s, the values K_D and S_{50} decreased to 1.23 and 103 kN, respectively.

Under no resistance to rotation of the fixed take-up pulleys, it can be assumed that the belt velocity on the arc of contact with them is equal to the velocity of the points on their surfaces. Then the belt sliding velocities at the points of running onto V_{Sni}, m/s, and running off V_{Sci}, m/s, from the drive pulleys (i = {1, 2}) are determined through the angular velocities ω_j, rad/s, (j = {1…4}) of the drive and bend pulleys. Simulation has shown that during the first second of the start-up, there is a non-monotonic (oscillatory) slippage of the belt on the drive pulleys. The sliding velocity of belt V_{Sn1} reaches 1.46 m/s, V_{Sc2} is 1.28 m/s.

Mode 4. Under belt coasting after the drive shutdown, the tension force equalizes along the belt contour. In the considered example, the time of belt coasting was 2.56 s, the maximum tension force $S_{\max}$ decreased from 121.6 to 105.7 kN, and the minimum tension force $S_{\min}$ increased from 63.8 to 83.5 kN.

Conclusions. The simulation made it possible to determine the characteristics of the transient process (lowest natural frequency, percent overshoot, and damping decrement) in a drive with a compensating take-up unit under a step change in the resisting torque. A drive with a compensating take-up unit is a stable servo control system with a limited range of tension and traction force variation.

The considered examples of starting a two-drive system with unregulated asynchronous electric motors showed that the compensating take-up unit smoothly changes the belt tension force as the resistance to its motion changes. This allows for a limited increase in the drive's traction capacity under the increased resistance to motion of the belt. The use of a compensating take-up unit is advisable in conveyors whose total energy consumption falls with a decrease in belt tension. A decrease in belt tension during conveyor's downtime and low loads slows the stretching (irreversible elongation) of the rubber-fabric belt.

A compensating take-up unit does not reduce dynamic loads in the belt under start-up. This is explained by the fixed position of the drive pulley shafts. Travel of the bend pulleys during start-up does not affect the section of the belt between the drive pulleys unless the belt completely slips on them. During the first second of an uncontrolled start-up, the belt line may slip on the drive pulleys. After the drive shutdown and the belt coasting, the tension force equalizes along the belt contour.

Summary. The developed computer-aided multi-mass dynamic model of a two-drive system of a conveyor with a compensating take-up unit allows for the analysis of steady-state and transient operating modes of the drive with standardized and non-standardized drives, determination of traction forces and the achievable power of the drives, and assessment of the belt's slippage and wear, taking into account the mechanical characteristics of the electric motors, belt, and contact interaction. The computer-aided model can also be used to analyze belt's start-up and braking modes if the torque and braking moments are defined as functions of time and pulley angular velocity.

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Received 15 April 2026

Information about the author:

Alexander A. Reutov – DSc (Engineering), Professor, professor of the Department of Production and Service in Transport Engineering, Bryansk State Technical University. E-mail: bgtu2012@yandex.ru; http://orcid.org/0000-0002-1703-6654, http://orcid.org/0000-0003-1117-8536

УДК 621.867.2:004.94 DOI: 10.21440/0536-1028-2026-5-100-108

Моделирование динамики привода ленточного конвейера с уравнительным натяжным устройством

Реутов А. А.[1]
[1] Брянский государственный технический университет, г. Брянск, Россия.

Реферат

Актуальность работы *обусловлена необходимостью повышения эффективности многоблочных приводов ленточных конвейеров.*

Цель исследования *– определение динамических характеристик привода ленточного конвейера с уравнительным натяжным устройством (НУ).*

Методы исследований. *Работа проводилась в 4 этапа: анализ научной информации; компьютерное моделирование привода ленточного конвейера как многомассовой динамической системы; моделирование четырех режимов работы привода; анализ результатов моделирования и формулировка выводов.*

Результаты. *Компьютерная динамическая модель двухблочного привода ленточного конвейера с уравнительным НУ разработана в среде программного комплекса «Универсальный механизм». Модель учитывает вязко-упругие свойства ленты, нелинейную зависимость силы трения от скорости скольжения и упругость контактирующих поверхностей. Крутящие моменты приводных барабанов при установившемся режиме работы представлены линейными зависимостями от угловых скоростей барабанов, при нерегулируемом пуске – нелинейными зависимостями от времени. Примеры моделирования переходного процесса при ступенчатом изменении нагрузки, а также нерегулируемого пуска привода с асинхронными электродвигателями позволили определить низшую частоту собственных колебаний, перерегулирование, декремент затухания колебаний переходного процесса при ступенчатом изменении силы сопротивления.*

Выводы. *Привод с уравнительным НУ является устойчивой следящей системой регулирования с ограниченным диапазоном изменения силы натяжения и силы тяги. Уравнительное НУ изменяет силу натяжения ленты при изменении суммарного сопротивления ее движению, что позволяет ограниченно увеличивать тяговую способность привода при увеличении загрузки конвейера. Уравнительное НУ не снижает динамические нагрузки в ленте при пуске конвейера.*

Ключевые слова: *ленточный конвейер; конвейерная лента; приводной барабан; фрикционный привод; натяжное устройство; компьютерная модель.*

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Поступила в редакцию 15 апреля 2025 года

Сведения об авторе:

Реутов Александр Алексеевич – доктор технических наук, профессор, профессор кафедры производства и сервиса в транспортном машиностроении Брянского государственного технического университета. E-mail: bgtu2012@yandex.ru; http://orcid.org/0000-0002-1703-6654, http://orcid.org/0000-0003-1117-8536

Для цитирования: Реутов А. А. Моделирование динамики привода ленточного конвейера с уравнительным натяжным устройством // Известия высших учебных заведений. Горный журнал. 2026. № 5. С. 100–108 (In Eng.). DOI: 10.21440/0536-1028-2026-5-100-108

For citation: Reutov A. A. Modeling the dynamics of a belt conveyor drive with a compensating take-up unit. *Izvestiya vysshikh uchebnykh zavedenii. Gornyi zhurnal = Minerals and Mining Engineering.* 2026; 5: 100–108. DOI: 10.21440/0536-1028-2026-5-100-108