Follow-up belt tensioning in mining conveyors – selection and design solutions

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Abstract:

The article discusses the basic dependencies of the selection of the tensioning force that guarantees the operation of the belt conveyor drive without slippage between the belt and the driving drum. The static characteristics of various tensioning systems were presented, the required forces for the given systems were determined and the influence of the tensioning method on the belt durability was determined. The paper discusses the follow-up stations in long belt conveyors operating in low inclined galleries, ensuring the reduction of belt loads in all conveyor operating conditions. Examples of solutions of follow-up stations currently operating in underground mines and their principles of operation are also presented.

Keywords: belt conveyor, tensioning system, mining, follow-up station

1. Introduction

Correct operation of the drive requires ensuring appropriate belt tension in every operating condition of the conveyor - during start-up, steady operation and braking. The required force in the belt running off the drive increases in proportion to the value of the driving force, but this value cannot be lower than that determined according to other criteria, mainly the belt sag criterion. This means that the belt pretension should guarantee the correct deflection of the belt between idler sets along the entire length of the conveyor for both belt strips.

The optimal tensioning system should provide a tension value not less than the pre-tension defined above until the force equals the force required for the increasing driving force, and then increase this force proportionally to the increasing driving force.

In the event of braking, the force system reverses and the most important thing is to ensure a minimum force in the belt driving on to the drive, otherwise the belt may completely loosen and the transported material may scatter and the conveyor may not brake or even damage the belt, which may lose contact with the discharge drum and get into the discharge and then, due to its elasticity, rapidly return to the discharge drum. When determining the minimum force in the belt during braking, it is taken into account that in the case of braking, the belt sag between idler sets can be assumed to be 2.5 to 3 times greater than the sag in steady motion [1].

Most of the currently used continuous belt tensioning stations provide a constant tension force, ensuring no slip on the drive drums during start-up and steady operation, however, this value is much higher than optimal, so the belt is over-tensioned during operation below the starting load, which is a short-term load in relation to the device operation time [2].

The necessity of special tensioning devices can be partially limited by systems reducing dynamic surplus in the drive area. Depending on the length of the conveyor, installed power, inclination and capacity and the type of belt used, we can choose between "soft start" systems (working only in the start-up phase), and inverter systems and special hydrodynamic couplings, which also work in other phases of the conveyor's operation.

Each of these devices solves a number of problems appearing in the start-up phase and steady operation, but does not eliminate the phenomenon of belt length change as a function of its tension. If the drive motors show energy consumption, then this energy is transferred to the belt and regardless of how the start-up is performed and dynamic surplus is minimized - there are differences in the energy accumulated in the belt between standstill and operation of the conveyor. The use of even the best starting devices does not release the user from the necessity to use tensioning devices, the more complex the conveyor is.

Incorrectly selected tensioning and starting system causes variable loads in the range from full relaxation to sharp jerking above the permissible loads and generates the following problems increasing operating costs:

– reduction of the life of the linings of the driving drums due to the occurring belt slip,
– reduction of the life of the belt and its connections,
– knocking out rollers from the support seats,
– reduction of durability of bearings in driving, return and tensioning drums,
– damaging structural elements, especially fixing and anchoring elements,
– larger steps related to monitoring the operation of the conveyor and the condition of its elements.

Some of the costs resulting from the above considerations are not entirely attributed to the savings made in configuring the conveyor. Meanwhile, downtime failures resulting in downtime of longwall devices multiply the actual costs resulting from investment savings. Since monitoring of costs allows to determine their total level (investment plus operation), some users are already able to predict the ranges of applicability of conveyors with a specific configuration in order to optimize the ratio of expenditure to effects.

The durability of the linings of driving drums may decrease not only due to the occurrence of slip during start-up, but also in the case of improper cooperation of individual drums in multi-drum systems

resulting from uneven loading of individual motors [3]. The remaining negative phenomena are partly or entirely due to problems with belt tensioning.

2. Materials and methods

2.1. Choice of the tensioning force

The belt tension force should guarantee the drive operation without slippage between the belt and the driving drum shell. Frictional engagement depends primarily on the coefficient of friction between the friction pair. Typical values for the calculations are presented in Table 1.

Table 1. Values of the coefficient of friction μ between the belt and the driving drum according to DIN 22101 [4]

Coupling conditions - belt condition	The value of μ coefficient for linings of the driving drum			
	Steel smooth	Polyurethane grooved	Rubber grooved	Ceramic grooved
dry	0,35 to 0,40	0,35 to 0,40	0,40 to 0,45	0,40 to 0,45
wet (clean water)	0,10	0,35	0,35	0,35 to 0,40
wet (clay, loam)	0,05 to 0,10	0,20	0,25 to 0,30	0,35
For belts with PVC covers, take values lower by at least 10%				

In addition to the friction coefficient, the angle of belt around the drive drum must be taken into account. Fig. 1 shows the two basic configurations of the two-drum drive in the shape of an inverted S and Ω. This angle is usually α = 215° for the S system and α = 190° for the Ω system.
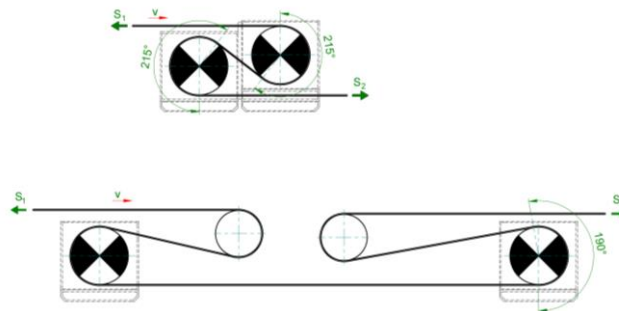
Fig. 1. Basic configurations of a two-drum drive [5]

S_1- Force in the belt running onto the drive, S_2- Force in the belt running off the drive,
v- the direction of the conveyor belt movement

The minimum belt tension behind the drive in relation to the circumferential force in the drive is determined using the Euler formula. The calculations are made **for the last drum** in the drive.

$$S_{2min} = k_z \times P_{u1} \tag{1}$$

where:

    k_z - coupling coefficient

$$k_z = \frac{k_u}{e^{\mu\alpha} - 1} \tag{2}$$

where:

    k_u - slip protection factor ($k_u = 1.2$)
    P_{u1} - circumferential force on the last driving drum (from the side of the lower belt tension)
    μ - friction coefficient between the belt and the driving drum
    α - belt wrap angle [rad]

Table 2. The values of the k_z factor for the most commonly used drive systems

Coefficient of friction μ	Coupling factor k_z	
	α=190°	α=215°
0,1	3,052	2,635
0,2	1,275	1,073
0,3	0,704	0,576
0,4	0,434	0,344
0,5	0,282	0,217

For multi-drum drives, the part of the circumferential force attributable to the last drum (from the side of the lower belt tension) is always assumed.

In practice, in the case of identical drive units on individual drums, it is enough to divide the value of the k_z coefficient by the number of drive drums "n" and multiply the k_{zn} coefficient obtained in this way by the determined total circumferential force of the drive [6].

$$k_{zn} = k_z/n \tag{3}$$

In the case of multi-drum drives, it is a mistake to insert the total angle of contact of the entire drive into the Euler formula, because the force S2 determined in this way is much lower than actually required.

$$e^{\mu(\alpha 1 + \alpha 2)} \gg e^{\mu \alpha 1} + e^{\mu \alpha 2} \tag{4}$$

For example, for a two-drum drive, with a friction coefficient of μ = 0.4 and a wrap angle of 215° on a single drum, after inserting the angle of 430 ° into the Euler formula, the obtained force value S_2 is 36% of the required one.

The tensioning force is selected using the k_z factor from Table 2, but the characteristics of the tensioning system should be taken into account [3].
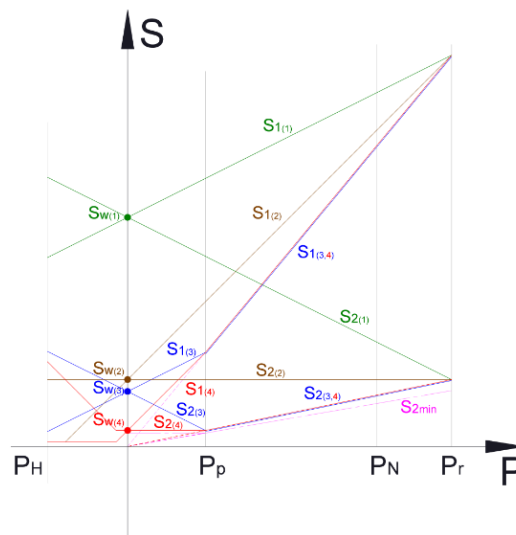
Fig. 2. Waveforms in drives with different tensioning methods [7]

(1) periodic tensioning, (2) constant tension continuous tensioning,
(3) traditional two-trolley follow-up station, (4) follow-up station with constant tension module;

P_r - circumferential force in the drive at start-up, P_N - circumferential force at nominal load,
P_p - circumferential force of the commencement of work of the follower station,
P_H - circumferential force during braking, S_1 - force in the belt running on the drive,
S_2 - force in the belt running off the drive

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The force charts shown in Fig. 2 show that the most advantageous station with the constant tension module, which gives the smallest excess of force actually occurring in the tensioning system in relation to the required force, is the most advantageous. The presented diagram applies to long conveyors in a slightly inclined galleries, where the minimum force results from the necessity to ensure the correct belt sag between idler sets, but there is no effect of the belt gravity force component.

- For active constant tension tensioning stations (gravity and hydraulic), the tensioning force must be determined for the circumferential force that occurs during starting ($S_{w(2)}$ in Fig. 2).

$$S_{w(2)} = k_{zn} \times P_r \qquad (5)$$

- For fixed tensioning systems (winch and screw), where the belt is pre-tensioned without correcting the force during conveyor operation, the determined force S_2 is the force required during start-up. The actual preload force ($S_{w(1)}$ in Fig. 2) is much higher and amounts to:

$$S_{w(1)} = S_{w(2)} + \frac{P_r}{2} = P_r \times (k_{zn} + 0,5) \qquad (6)$$

- For a traditional follow-up station with trolleys connected with a pulley block, the preload depends on the value of the circumferential force present in the system at the time of commencement of following operation. For flat conveyors it is about 30% of P_N. The value of the initial tension is determined in the same case as for the winch station by inserting into the formula for S_w the value of the circumferential force at which the follow-up station starts to work ($S_{w(3)}$ in Fig. 2).
- For the follow-up station with a constant tension module, the pretensioning force is selected on the basis of the belt sag criterion ($S_{2(4)} \geq S_{2min}$ in Fig. 2).

The maximum value of the force in the belt occurring during the steady operation in each cycle of its complete circulation has a significant impact on its durability.

The Woehler dependence referred to in [2] for determining the fatigue life of belts shows a significant influence of the longitudinal forces in the belt on its fatigue life.

$$T = \frac{N \times l}{3600 \times v \times z}\left(\frac{\sigma_z}{\sigma_{max}}\right)^5 \text{ [h]} \qquad (7)$$

where:

N - number of load cycles until fatigue failure $N = 10^7$ [cycles]
l - total length of the belt [m]
v - belt speed [m/s]
z - number of belt bends in the cycle
σ_z - unit load on the belt corresponding to the limit number of cycles [kN / m]
σ_{max} - maximum unit load in the cycle [kN / m]

In the formula for belt fatigue life, the maximum unit load in the cycle appears in the denominator in the fifth power, and the number of belt bends in the cycle in the first power. From Fig. 2, it is possible to determine the influence of the tensioning system on the belt durability by comparing the S_1 forces for individual types of tensioning and increasing the ratio of these forces to the fifth power and multiplying the result by the quotient of the number of drums.

$$\frac{T_i}{T_j} = \frac{z_j}{z_i} \times \left(\frac{S_{1j}}{S_{1i}}\right)^5 \qquad (8)$$

The results of these comparisons for a specific case presented in Fig. 2 are presented in Table 3. As the diagram in Fig. 2 was created with the assumption of a starting force at the level of 130% of the nominal load, which corresponds to very favorable conditions of soft start-up, each start-up case with a higher dynamic surplus requires more higher preload when the conveyor is operated with a winch

or constant tension station. The values of the calculation life ratio for the nominal load and with the assumption that statistically the work load is 60% of the nominal load are presented. The relation of the pretensioning forces is also presented.

Table 3. Comparison of the increase in belt life resulting from the comparison of the maximum force in the belt for identical parameters of the conveyor and belt operation [own study]

Item	Tensioning systems compared	Belt durability ratio according to formula (8) load 100% nominal		Belt durability ratio according to the formula (8) load 60% of nominal		Pretensioning forces ratio S_{wj} / S_{wi}
		Drums $z_j/z_i=1/1$	Drums $z_j/z_i=6/8$	Drums $z_j/z_i=1/1$	Drums $z_j/z_i=6/8$	
1	1(j) to 4(i)	2,24	1,68	13,40	10,05	15,17
2	2(j) to 4(i)	1,28	0,96	2,43	1,82	4,33
3	3(j) to 4(i)	1,00		1,00		3,50
4	1(j) to 2(i)	1,75		5,51		3,50
1 – conveyor with periodic tensioning 2 – continuous tensioning constant voltage conveyor 3 – conveyor with a traditional two-trolley following tensioning station 4 – conveyor with a follower station with a constant voltage module						

Table 3 also shows that **any active tensioning** significantly increases the design life of the belt compared to winch-type stations, which, due to the response time and tensioning speed, are always included in the periodic tensioning station.

2.2. Determination of the required gear ratio of the follow-up stations

In domestic underground mining, follow-up stations are used, in which a system of two tensioning trolleys is used for tensioning. The trolleys are connected by a rope system with the i_{zl} ratio, which ensures a constant ratio of forces in the belt running on the drive and running away from the drive (Fig. 3).
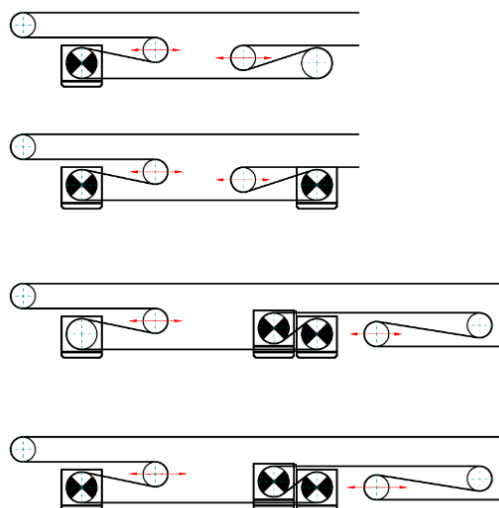
Fig. 3. Tensioning trolleys in single, double and triple drum drive systems [own sketch]

The trolley on which the S_1 force occurs can be described as controlling the tensioning, and the trolley with the S_2 force - as the executive trolley, because its displacements are greater than the displacements of the control trolley by the value resulting from the gear ratio of the rope system.

Until the force relation S_1 and S_2 is established at the level equal to $\mathbf{S_1 / S_2 = i_{zl}}$ the following station does not work. The course of forces during this time is the same as in devices with a fixed distance between the drums. The force S_1 increases and the force S_2 decreases. The average force in the conveyor belt does not change.

The station starts working when the circumferential force $P = S_1 - S_2$ appears in the drive, for which $S_1 / S_2 = i_{zl}$. The control trolley is pulled by the belt and at the same time it pulls the executive trolley, which travels the distance greater by i_{zl}. If the travel of the control trolley is marked as w_1, then the value of $\boldsymbol{\Delta l = 2 \times w_1 \times (i_{zl} - 1)}$ determines the length of the belt pulled out during tensioning, which corresponds to the belt elongation value resulting from the action of the circumferential force P.

The most important feature of the follow-up stations is the absence of their own drive. All the energy of the trolleys movement comes from the main drive of the conveyor, reducing the dynamic forces acting on the belt during the increase of the circumferential force [8].

The required ratios of the tensioning station "i" are determined considering the conditions of frictional coupling between the belt and the last drive drum from which the belt runs off the drive. It is important to determine the friction coefficient between the belt and the drum lining whether the last drum cooperates with the carrying or running belt cover. Knowing the k_z factor for the wrap angle of the last drum, presented in Table 2, and assuming that the same power is installed on each driving drum, the station ratio is determined by a simple relationship

$$i = \frac{n + k_z}{k_z} \tag{9}$$

where:

n - number of driving drums

k_z - coefficient according to the formula (2)

Table 4. The required ratios of the follow-up stations determined from the formula (9)
[own calculation]

Coefficient of friction μ	Required ratio of the follow-up station $i = S_1 / S_2$ for the drives			
	Single-drum α=190°	double-drum α=190°	double-drum α=215°	Triple-drum α=215°
0,3	2,42	3,84	4,47	**6,21**
0,35	2,83	4,65	5,53	7,80
0,4	**3,31**	**5,61**	6,81	9,72
0,45	3,87	6,75	8,35	12,03
0,5	4,54	8,08	10,21	14,82

For follow-up stations, where the trolleys are coupled via pulley blocks, the ratio value is taken as the integer value from the number shown in Table 3 (for the given coupling conditions) or lower.

## 3. Results -design solutions of follow-up tensioning stations

The first design of a self-operating station using the main drive power dates back to 1945 [9]. In this solution, the sliding steering drums located in front of and behind the drive are coupled by differential pulleys. Other solutions with differential drums, enabling longer tensioning paths, come from 1954 [10], [11]. One of the solutions with the use of cable drums coupled with a gear that ensures the appropriate ratio is presented in [12]. The problem in implementing this type of stations was the high dynamics of the tensioning trolleys, especially when the conveyor was stopped. Successful implementation of two-trolley stations took place after using the solution [13] in combination with a damping system [14], which allowed the use of this tensioning system also in the case of using brakes.

3.1. Two-stage follow-up station

The tensioning station discussed in the article [15] with the use of a variable pulley block system [16] enables the reduction of the required initial tension due to the fact that the station with a lower ratio value reaches the active state faster. However, since the low-ratio station does not guarantee the lowest

forces in the system under the conditions of nominal and starting circumferential forces in the drive, at the point specified by the designer, it automatically switches to operation with full transmission. The previously used stations of this type made it possible to obtain gear ratios i = 2/4, 2/6 or 4/6 [17].

a) work with the ratio i = 4

b) work with the ratio i = 6
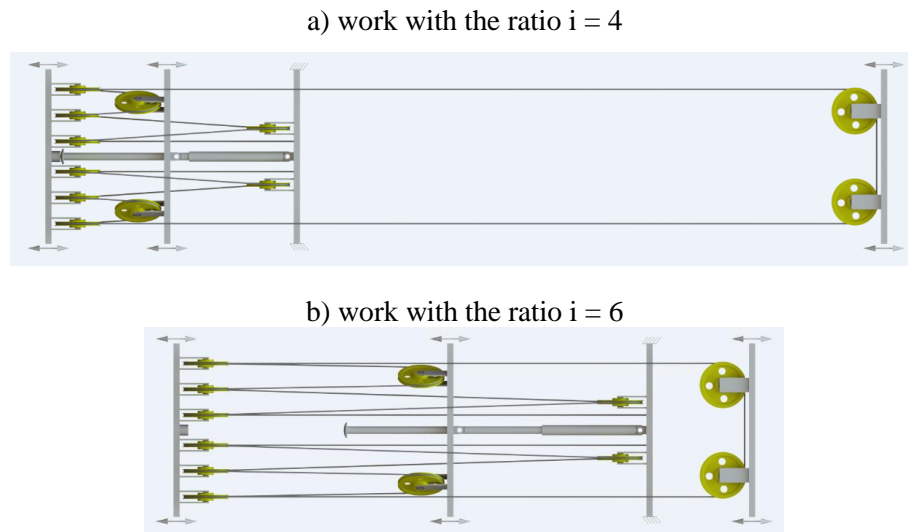
Fig. 4. Diagram of the rope circulation in the station with variable ratio i = 4/6 [own sketch]

Fig. 4 shows the implementation of the variable ratio. Two rope pulleys are mounted on the hydraulic cylinder trolley connected to the cylinder's piston rod, which acts as a hydraulic shock absorber (the trolley's bumper rests against the bumper on a high-tension trolley) and moves together with the station's high-tension trolley's pulleys as passive wheels until the cylinder stroke is used. After locking the piston on the gland of the cylinder, these pulleys stop and further movement of the high-tension trolley causes the low-tension trolley to travel with full gear ratio. The hydraulic cylinder used in the construction of the follow-up station does not have a power supply system with an external unit. While the high-tension trolley is moving, the cylinder's piston rod is pulled out by pulleys connected to it. Oil is poured into the head space by gravity from a reservoir located above the cylinder. When the high-tension trolley returns to its initial position, the cylinder acts as a damper for its movement, because the oil is pressed into the tank through the throttling-overflow system, eliminating the possibility of a sudden return of the trolley and hitting the end stops.

The effect of using a two-stage tensioning station is shown in Fig. 5.
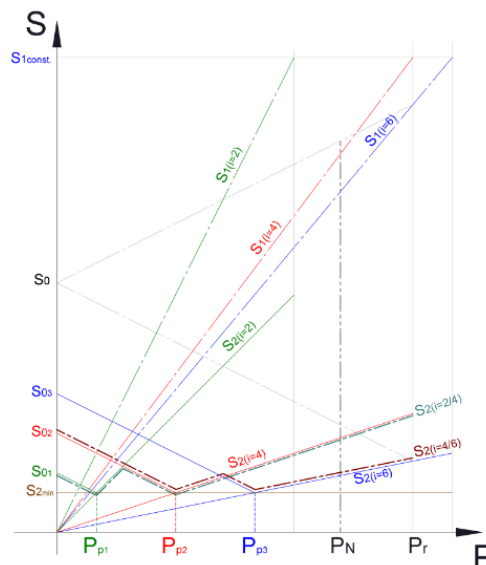
**Fig. 5.** The course of the S2 force generated by the tensioning station with variable ratio [15]

For a station with a ratio of $i = 4/6$, the conveyor starts with preload S_{02}. The station starts working after reaching the circumferential force P_{p2} and works with the gear ratio $i = 4$. After the pulleys placed on the cylinder trolley stop, the station is inactive until the drive reaches the circumferential force P_{p3}. Then the station works with the gear ratio $i = 6$.

The use of a two-stage station allows to reduce the preload compared to a station with one ratio, the ratio of which is selected according to the criterion of frictional coupling.

3.2. Follow-up station with constant tension module

The tensioning station shown schematically in Fig. 6 differs from the classic two-trolley station described in point 4.1. This station is equipped with a power unit supporting the constant-pressure tensioning cylinder and the shock-absorbing cylinder (hydraulic buffer) of the high-tension trolley [18]. For this reason, it is not necessary to use a pulley system to withdraw the piston rod of the damper. The presented diagram applies to the following station with the ratio $i = 4$ or after the elimination of the equalizing disk on the low-tension side $i = 8$. Ratios $i = 2,3$ or 4 are possible on the constant tension side.
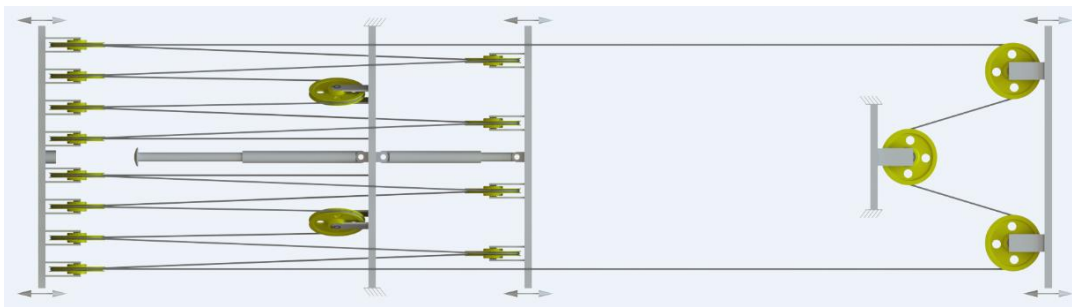
Fig. 6. Rewinding the rope of the follow-up station with a constant tension module [own sketch]

The principle of operation of the station is as follows.
– Before the commencement of the work of the follow-up station, when the ratio of forces S_1 and S_2 in the drive is below the ratio of the rope system, the high-tension trolley is on the bumper and the constant-pressure cylinder trolley moves, moving the low-tension trolley and keeping the force S_2 in the belt at the level set by the pressure in the hydraulic unit.
– When the ratio of forces S_1 and S_2 in the drive reaches the ratio of the rope system, the high-tension trolley begins to move, at the first moment causing the constant pressure cylinder trolley to return until it locks. Until the constant-pressure cylinder locks, the movement of the high-tension and low-tension trolleys does not change the belt tension. Simultaneously with the movement of the high-tension trolley, the piston rod of the damper cylinder extends. Further displacement of the high-tension trolley after blocking the constant- pressure cylinder causes the shift of the low-tension trolley proportional to the ratio, whereby the force in the belt behind the drive increases with increasing circumferential force in the drive.
– When braking, the system of forces in the drive changes due to the reversal of the driving torque. The high-tension trolley returns to its starting position, and at the same time, the constant pressure cylinder trolley is also pushed back. The force in the belt results from the given preload.
– An example of the use of this type of device in practice was presented in the paper [19].

3.3. Hydraulic follow-up station

Follow-up stations with a rope pulley guarantee the correct tension in the belt during start-up and steady operation, while sometimes selecting the belt before the drive during braking due to the return of the high-tension trolley and excessive loosening of the belt leading to the discharge may occur. As shown in Fig. 2, the follow-up stations on the braking side behave like stations with a fixed drum spacing.

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The solution that ensures the correct tension in all operating conditions is the follow-up tensioning station, in which the rope systems have been replaced by a system of hydraulic cylinders, which provides tension in the belt according to the course marked (4) in Fig. 2, both on the positive and negative circumferential side [20]. The minimum S_1 force during braking can be assumed with the sag criterion lower than that for operation with the nominal load, because in the case of braking the belt sag size between idler sets is allowed 2-3 times greater than that adopted for steady motion operation.
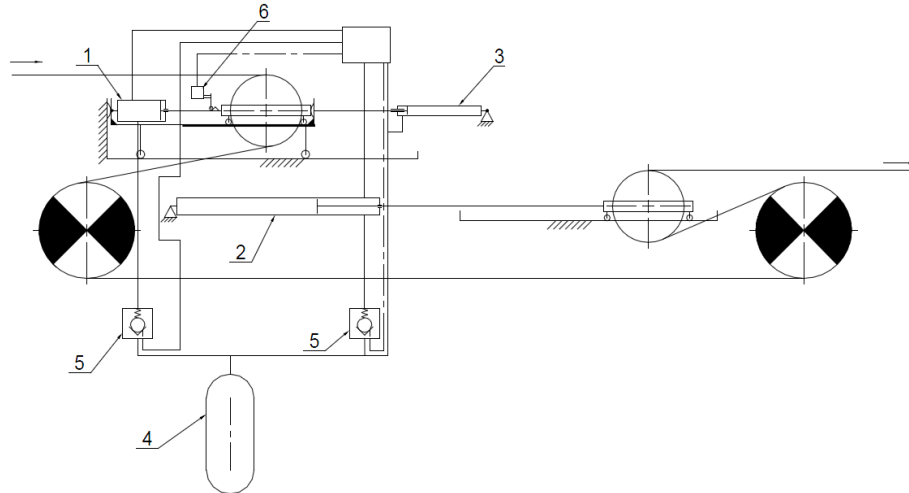

Fig. 7. Diagram of the operation of the hydraulic follow-up station [21]

The principle of operation of the station is shown in Fig. 7. The most important advantage of the device is that it does not increase the number of drums in the system. For a two-drum drive, apart from the two driving drums, there are only two non-driving drums, which simultaneously make both driving drums work with the clean cover of the belt.

There are three hydraulic cylinders in the station, connected by a hydraulic system and supplied with the same pressure from the power pack (4). Maintaining the previously adopted terms, the cylinder (1) connected to the drum from which the belt runs onto the drive is the control cylinder, and the cylinder (2) which is connected to the drum through which the belt coming off the drive is the executive cylinder. The control cylinder with the high-tension trolley is located on an additional trolley moved by the third cylinder (3), whose task is to ensure the minimum belt tension, on the side of belt running to the drive, during braking.

The diameters of the cylinders are selected in such a way that the ratio of active surfaces between the control actuator (1) and actuator (2) corresponds to the ratio from table 3.

$i_{station}$ = (under piston surface of cylinder 1)/(over piston surface of cylinder 2)

The cylinder (3) has a cross-section that guarantees the minimum force in the on-load belt, ensures the braking process is performed without the belt slipping on the drive drum and prevents the belt coming to the discharge from being fully loosened when the conveyor stops. The cross-section of the cylinder (3) must be smaller than the cross-sections of the remaining cylinders so that during their operation the cylinder (3) is always in the blocking state.

The cylinder (2) reacts to changes in the belt length and, by correcting the position of the tensioning trolley, maintains the tension in the belt resulting from the pressure in the hydraulic system. It is a segment of work with a constant force S_2, included in Fig. 2 in the range of the circumferential force from 0 to the value of P_p. The pressure in the hydraulic system keeps the other cylinders fully extended.

As the value of the circumferential force in the drive increases, the force S_1 increases with a constant value of the force S_2. An increase in the S_1 force in the belt above the value of the force on the piston of the control cylinder (1), resulting from the pressure in the hydraulic system, causes the cylinder to move. A hydraulic limit switch (6), which reacts to the movement of the cylinder, cuts off the cylinders (1) and (2) from the external power supply by means of valves (5), and the oil moves from the control

cylinder (1) to the executive cylinder (2), causing the piston rod to extend by the value resulting from the stroke of the control cylinder and the ratio of active surfaces of both cylinders. The system starts working as a follow-up station with a pulley block system. Due to the disconnection of the cylinders (1) and (2) from the hydraulic unit, the tensioning station begins to take energy from the conveyor drive, so that the required tensioning power is not limited by the power of the supply system, and in this way it is possible to obtain full follow-up of the station's operation. and the additional effect of reducing dynamic loads on the belt at start-up [8]. Switching off the conveyor or dropping the circumferential force in the drive below the minimum value causes the return to the initial state. In the event of braking, the force S1 drops even more and when it reaches a value lower than that resulting from the force on the cylinder piston (3), this cylinder moves the pre-drive drum not allowing the force S_1 to fall below the allowable one.

The stroke lengths of individual cylinders should be selected by the conveyor designer so that there is no case that there is no stroke for the correct operation of the system. The reserve of stroke of the cylinder (2) in relation to the cylinder (1) should be taken into account, because the cylinder (2) works not only in the follower mode, but also in the constant tension station mode. Use of the total stroke of one of the cylinder will result in conveyor operation as in the case of a periodic tensioning station.

3.3.1. Station assembly variants

The tensioning station shown in the diagram in Fig. 7 has hydraulic cylinders connected to non-driving drums directing the belt to driving drums in a system of two driving drums. It is not the only assembly variant that can be used as part of this solution. The station can also be built according to the tape rewinding diagram shown in Fig. 3 for the two- and three-drum version. In this case, the cylinder (2) cooperates in the loop drum, which is normally a fixed turn drum. Of course, in such a case, the second looper drum mounted on the looper trolley and connected to the winch system serves to apply the preload in the belt and compensate for the length of the belt resulting from shortening or lengthening the conveyor.
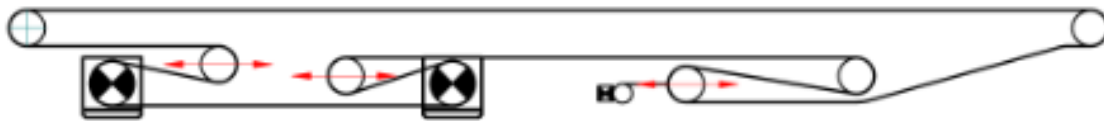

Fig. 8. The tensioning system in conveyor with the loop [21]

In the arrangement shown in Fig. 8, the loop in the conveyor with the two-trolley tensioning system is built in an inverted configuration, i.e. the belt runs to the turning drum over the top, and then returns to the loop trolley connected to the tensioning winch. Such a system is very advantageous for the operation, because all the units that require monitoring, such as: main drive units, hydraulic power unit and the loop tensioning winch, are located close to each other. An additional advantage of such a conveyor arrangement is the minimization of the risk that the losses from the upper belt may get under the drums within the loop.

3.3.2. Installation of a hydraulic station as constant tension station

In inclined conveyors, there may be situations where the S_{min} line moves up the graph shown in Fig. 2 and may align or even exceed the line describing the required force for the constant tension station. In this case, constant tension stations are most often used, installed in the place of the lowest forces in the belt, set to the minimum force that guarantees operation without belt sagging between idler sets. The frictional coupling in the drive guarantees the gravitational effect of the belt itself.

For long conveyors with moderate lift, the tensioning station is more preferably placed within the main drive, taking the tensioning force to ensure, in addition to the frictional engagement in the drive, also the correct belt sag in the return area. In conveyors equipped with brakes, it may be advisable to ensure in the drive also a minimum force in the overlapping belt during braking.

These tasks are performed by the tensile station shown in Fig. 9, which is a simplified version of the follow-up station.
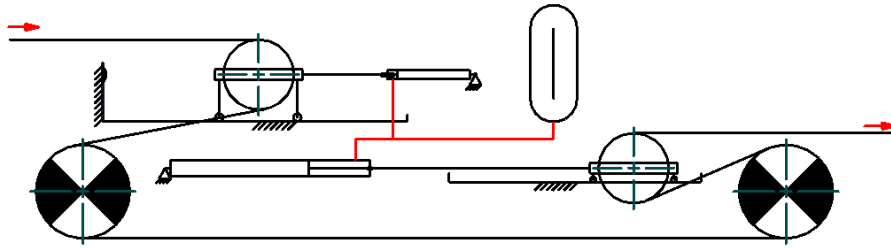
Fig. 9. Constant tension tensioning station with the function of tensioning the sliding belt during braking [21]

4. Conclusions

The follow-up tensioning stations have two advantages compared to other tensioning systems, i.e. the adjustment of the belt tension force as a function of the driving torque and the use of power from the main drive of the conveyor to move the tensioning trolleys [22]. This also applies to stations equipped with the function of working as a constant tension unit, because the external power supply in these stations is used only for tensioning regulation during standstill, operation with a minimum load and during braking. During start-up and operation with high load, these stations always use the power of the main drive, because it is the basic condition for achieving full follow-up of the tensioning force changes [8].

The feasibility studies of long belt hauls show that with the use of the following station and intermediate drives along the conveyor route, it is possible to properly operate a belt conveyor with a typical capacity of 2000 t/h, equipped with a textile belt with a strength of less than 2000 kN/m, built in a flat galleries with a length of up to 10 km.

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