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Mobile working platform unit as a mean for improvement of safety and convenience of emergency shaft works

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Abstract:

A level of mine safety is related directly to the condition of its shafts. Thus regular monitoring and maintenance of shaft lining and equipment is vital. However such works are always hard and dangerous, as they are conducted from conveyances. Working platforms are a significant improvement in case of emergency shaft works. However, they need some extremely precious time for assembly. An idea of pull out working platform combines safety and convenience of typical working platform with short time of its installation. Following work presents the idea and construction of the mobile working platform unit, which is a solution for improvement of shaft works conducted from the compartment of the conveyance, as the platform uses cage for transport and operation.

Keywords: shaft works, mine shaft, shaft maintenance, working platform, work safety

1. Introduction

Mine shafts, as main mine workings, are crucial for the safety of the whole mine in almost every aspect – from ventilation, through reliability of man and material transport, to providing an escape route for miners [1-6]. Condition of shaft lining, hoisting system and other equipment should be constantly monitored and all of failures or malfunctions should be immediately repaired. Lack of proper monitoring of the mine shaft and its elements can lead to serious accidents or even catastrophes, as it was in shaft no. V of the Knurów department of the Knurów-Szczygłowice colliery [1, 5, 7-9].

Advanced age of most of Polish mine shafts makes the issue of mine shaft monitoring even more important [1, 10]. However, Polish regulations on mine shaft elements monitoring are not precise. Moreover, the process of the monitoring in such old workings might be hard itself. The development of mobile monitoring devices can positively affect the quality of mine shaft monitoring, despite of the high investment cost of these solutions. Further development of mobile monitoring solution is to be expected [5].

Currently the mine shaft monitoring comes down to periodical destructive and non-destructive testing conducted by experts. However, the process of shaft lining testing might be complicated, as it is conducted from the conveyance, which sometimes makes it difficult to reach the lining. It is also required to stop regular operation of the shaft hoisting system for the time of conducting the test. Periodical character of the tests can also result in late information of potential damages or failures [1, 5, 11-16].

The situation of emergency and maintenance works in mine shafts, such as repairs of the lining, pipelines or cables, is similar. These elements ae accessible only from the conveyance or working platform that has to be built in the shaft or on the conveyance construction. The most convenient way of conducting the shaft works is to build a suspended working platform. However such solutions are extremely expensive and complex constructions and require a long period of time for their design and assembly. For these purposes they are used only in case of shaft sinking, equipping or very complex shaft modernization. In case of emergency or maintenance works, application of such constructions is not economically justified [3-4, 17-21]. Working platforms assembled to conveyances construction are also used. They were an inspiration for the pull out platform and they are presented in following section.

Another modern solution combining safety and convenience of suspended working platform with mobility of the conveyance is the special-purposes conveyance that was used by Shaft Sinking Company (PBSz SA) in the 1-Bzie shaft of the Jastrzębie-Bzie colliery [22-24].

The solution of the pull out working platform described in this work was used in the shaft no. II of Borynia department of the Borynia-Zofiówka coal mine to improve repair works. This shaft is equipped with four-compartment conveyances, guided using stiff steel guides [25].

2. Existing constructions of shaft working platforms

Working platforms used in shafts of underground mines are designed to provide access to the shaft lining and equipment, such as pipelines or cables. They are assembled to the top transom of the conveyance, to the floor of cage's compartments or to the conveyance's sidewalls. Shaft working platforms have to provide proper level of safety for people using them and possibly the shortest time of their assembly and disassembly. Every element of the working platform which exceeds the conveyance's outline has to be transported in the cage and manually assembled. A working platform is usually assembled at the level, where shaft works are to be conducted, in extreme conditions of the cage suspended at great height.

In case of the shaft stoppage ongoing for at least few days (which is a very rare situation), permanent platforms are installed. In such case, constructions used in civil engineering are used, built or suspended on the shaft buntons. It can be spotted in shafts where hoisting system operation is suspended for a long period of time, e.g. in case of repair of the lining in large range.

For purpose of testing and carrying out works in the operating shaft, stiff platforms assembled to the top transom of the conveyance, constructed inside the cage or swing platforms installed to other elements of the cage's construction are used.

Stiff platform installed on the top transom of the cage usually consists of the rigid frame assembled with screw to elements of the construction of the cage's top transom and the actual platforms made of steel sheet and assembled to beams, which are installed on the frame using pins. To provide safety for people working on the platform, it is equipped with barriers and an overhead protection. Platforms with beams and protections are disassembled after every working shift, while the frame is usually disassembled after the end of the works.

Platform constructed inside the cage is usually made of steel or wooden beams, installed parallelly to the conveyance's axis. The most popular construction of such platforms comprises thick wooden boards blocked with transverse steel beams, assembled to the cage's construction with screws.

Swing platform is usually a steel construction installed from the inside of the cage, with the possibility of rotation around the point of its assembly. The construction of swing platform is based on steel cables lowered along the cage. Platforms are assembled to the cables. The construction of the swing platform allows to work on even three levels simultaneously, as well as quick assembly and disassembly of the platform. However, the construction is a subject of big torque, which makes stabilizing the platform and thus providing safety for people working on it a real issue.

Time for tests, repairs and modernizations of the shaft elements is limited, especially in the shaft with operating hoisting system. Shafts are usually available for contractors only on single shifts on weekends or holiday. The time for which the hoist is blocked with the cage at the level of where the shaft works are to be conducted is spent on: loading of the platform's elements and its assembly together with all protections, actual shaft works (such as lining repair), disassembly of the platform and unloading of its elements. The issue is the long time of the preparatory actions (assembly and disassembly of the platform). The aim is to shorten their time, to make the shaft works more effective with providing proper level of safety.

The solution for reducing the time of assembly and disassembly of the working platform is application of constructions which allows to pull out or unfold platform's elements installed in a unit, which is placed inside the cage.

German patent DE29908954U1 [26] presents working platform based on a lift. Elements of the working platform are located in the corpus of the device and they can be pull out for purpose of increasing the area of the platform surface. These elements are telescopically connected. Proper stiffness of the construction is provided by side beams, which are pulled out together with the main element of the platform. They also play a role of side barrier. Movable elements are connected with the device corpus with telescopic rails. However, the method of attachment does not provide sufficient durability for high payloads and allows for small enlargement of the working platform area.

French patent FR2830557 [27] presents a solution of working platform, which allows to increase the width of the working platform area. Platform extensions are connected with the main platform with vertical barrier element. To increase the platform width protective barrier is disconnected and the extension is rotated around its axis, moving along the arch from its idle position to outside of the platform. However, this construction of the working platform cannot be used in case of restricted space, which makes it impossible to move the extension.

Patent WO2006052131 [28] presents a sliding working platform, which is an element of a lift. Two guides made of C-profile are attached to the permanent platform along its longer sides. Movable platforms are assembled to the guides and they can be moved to extend the permanent platform on its both sides. The method of attachment does not provide sufficient support for high payloads and it is a source of high forces acting on the moveable platforms.

Krause company offers so-called telescopic working platform, construction of which allows to telescopically extend its length from 1.75 up to 3.50 m. It consists of closed profile containing moveable element, which allows for the platform extension and together with dedicated accessories creates a complete working platform. It is designed for maximum working height of 3.0 m with payload of 200 kg. It is used to access objects vertically and it requires support construction on both sides of the platform. Its construction results in relatively small payload of the platform.

Solutions for extending the working platform presented above cannot be used in a mine shaft, because they do not provide sufficient payload, range, safety as well as convenience in transport or assembly in difficult conditions. However, they became an impulse for design of the device that would help to improve safety and convenience of conducting shaft works.

3. Mobile working platform unit

The idea behind the invention of the mobile working platform unit was to design a light and easy to move construction of a working platform. The aim was to develop a device equipped with moveable elements that creates working platforms oh high durability and providing high level of safety. The mobile working platform unit can be transported in a mining cage and operate in the shaft from the conveyance. Its elements are easy to assemble, disassemble and transport.

Mobile working platform unit is designed to play a role of a working platform for emergency and maintenance shaft works. It consists of pull out platform element assembled to a corpus, which is equipped with undercarriage. A view of the platform is presented in Fig. 1.

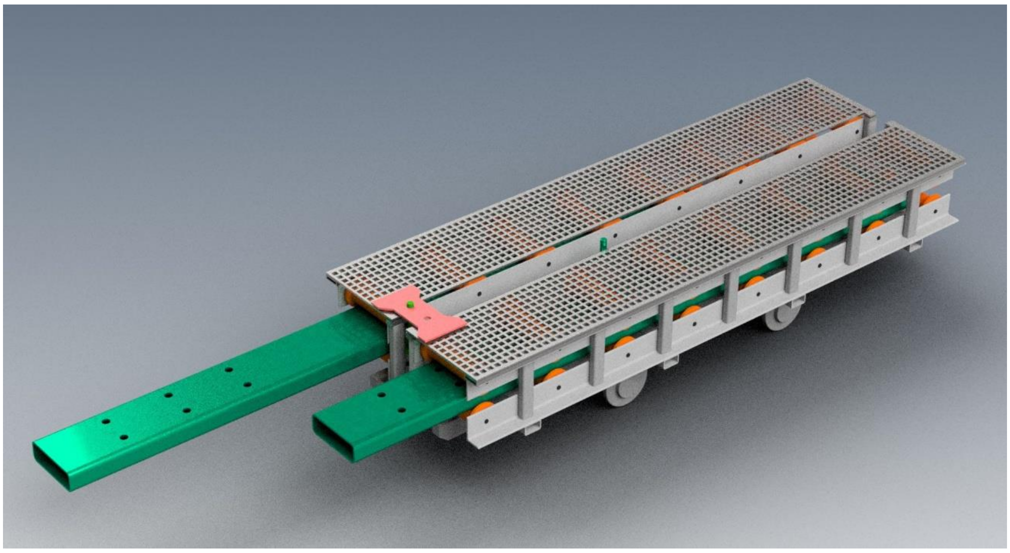

Fig. 1. A view of the mobile working platform unit for shaft works

The mobile working platform unit, according to patent [25] comprises one or two corpuses, each of them constructed of four beams: two top and two bottom, connected permanently and two roller guides: top and bottom, permanently fixed to the corpus. A moveable bridge is assembled between two roller guides. Permanent working platform is installed on the top of the device's corpus. The corpus is permanently fixed to the undercarriage.

The mobile working platform is connected to the construction of a conveyance using technological holes in the beams. Roller guides are groups of spool-like rollers, assembled rotary in the corpus' beams, both top and bottom.

The moveable bridge, which is a closed-profile beam, is assembled between bottom and top roller guides, which allows to easy move the bridge and to stabilize it in safe and reliable manner. There are symmetrical pairs of mounting holes evenly spaced on the bridge. Mounting holes allow to lock longitudinal motion of the bridge. The corpus of the mobile working platform unit is covered with permanent working platform, made of steel grate with square eyelets.

The key feature of the device is the m,oveable working platform which is installed on thebridge exceeding the outline of the device's corpus and thus the outline of the cage. It is made of steel grate, the same as the permanent platform. Elements of the moveable platform are assembled using pins, permanently fixed to the platform's element. A distance between the surfaces of the platform and the bridge is fixed using sleeve-like spacers used with the pins. Pins' spacing is equal to the spacing of the mounting holes on the moveable bridge. Pins placed in the mounting holes lock the motion of the moveable platform. Every platform used with the mobile working platform unit has to be equipped with an even number of pins and not less than four. Moveable platforms are also equipped with protective barriers segments and optionally with an overhead protection. A view of the platform unit equipped with different types of moveable platforms is shown in Fig. 2.

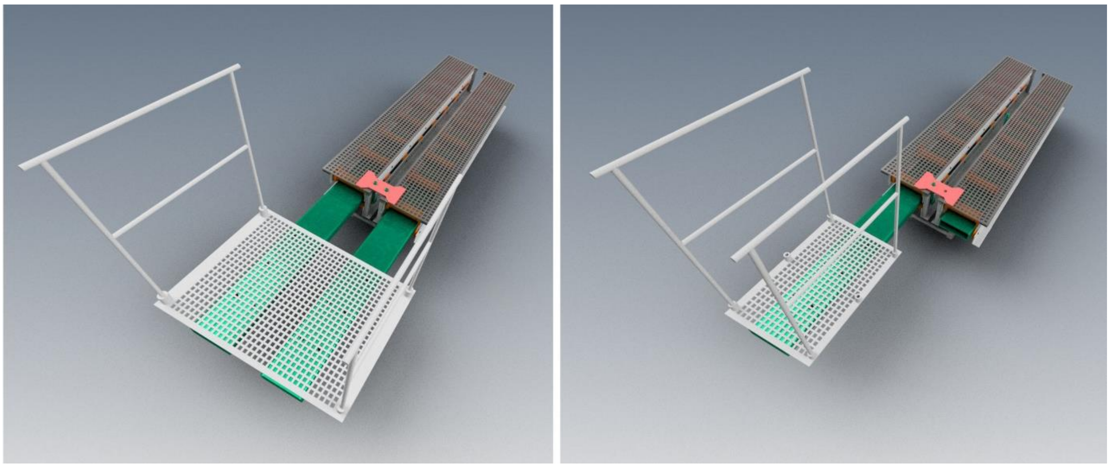

Fig. 2. A view of the mobile working platform unit equipped with different types of moveable platforms

The permanent platform is connected with the moveable bridge using minimum two profile overlays, made of a flat bar, equipped with pins fixed at right angles to its surface. The overlays, connecting platform's grate and bridge's mounting holes are a mechanical lock, preventing bridge's longitudinal motion.

The corpus of the mobile working platform unit is fixed to a undercarriage frame. The undercarriage comprises at least two single axis or one two-axle cart. The parameters of the carts (axle's width and wheel size) have to be suited to the parameters of the mine's rail system.

Dimensions of the mobile working platform unit are suited to the dimensions of the conveyance in which the platform is to be used. Fig. 3 presents a view of the mobile working platform unit placed in the conveyance. Fig. 4 pictures an example of application of the device in a mine shaft equipped with four cages. Design of the mobile working platform unit is shown in Fig. 5 and 6.

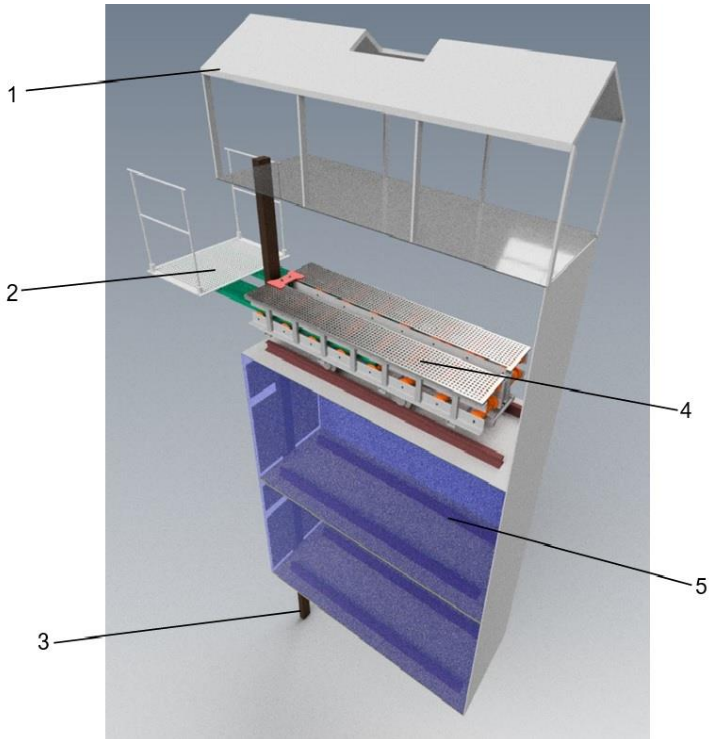

Fig. 3. The mobile working platform unit in a conveyance:
1 – cage's overhead protection, 2 – moveable platform, 3 – guide, 4 – corpus, 5 -cage

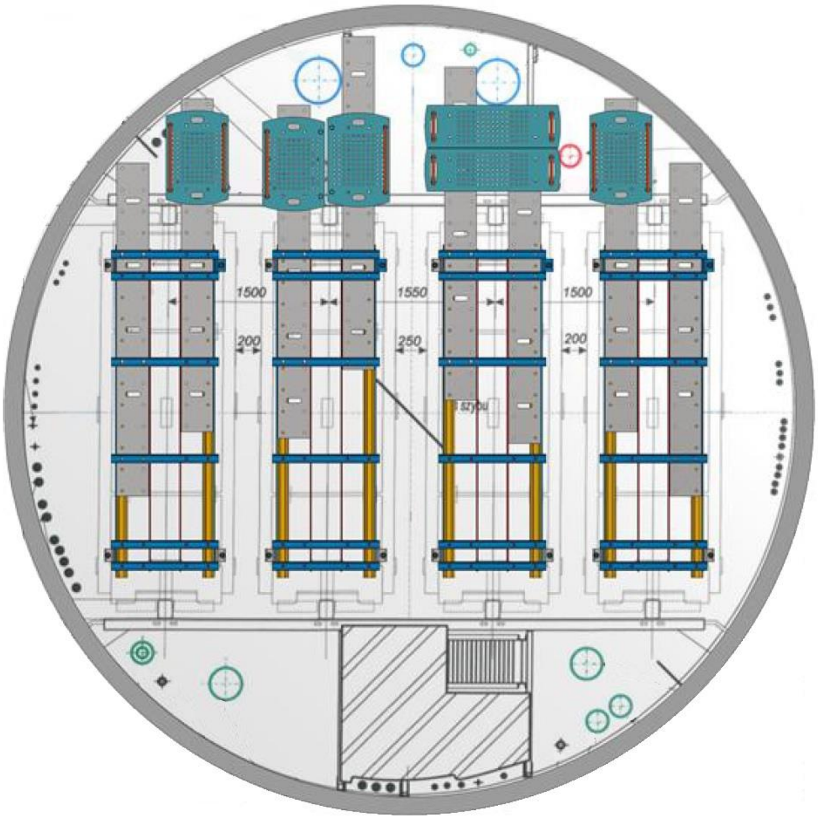

Fig. 4. An example of possible application of mobile working platform units with different types of moveable platforms in a mine shaft equipped with four cages

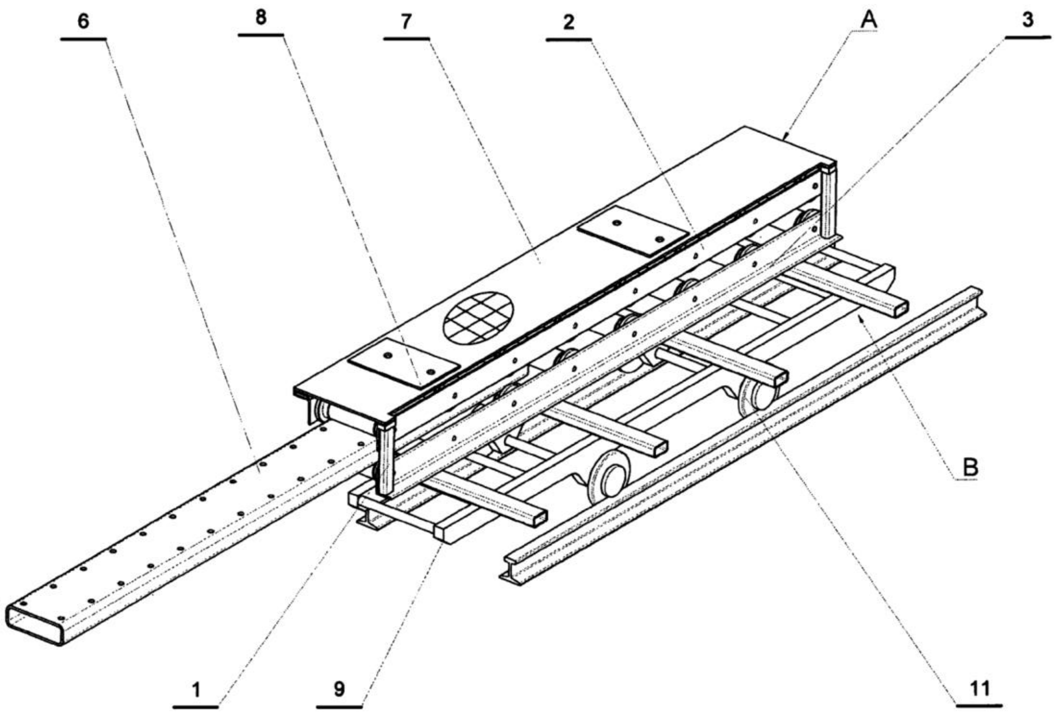

Fig. 5. Design of the mobile working platform unit:
A – corpus, B – undercarriage, 1 – corpus connectors, 2 – corpus' top beam, 3 – corpus' bottom beam, 6 – moveable bridge, 7 – permanent platform, 8 – profile overlay, 9 – undercarriage frame, 11 – cart

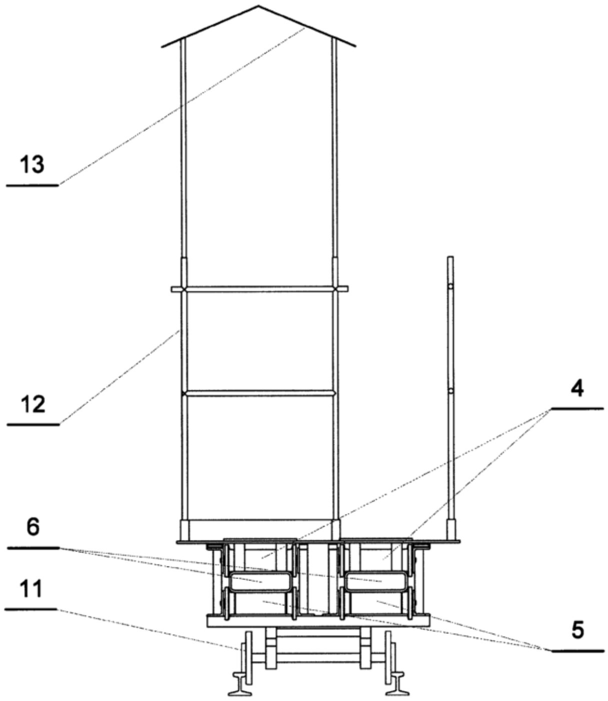

Fig. 6. Design of the mobile working platform unit:
4 – top roller guide, 5 – bottom roller guide, 6 – moveable bridge, 11 – cart, 12 – protective barriers, 13 – overhead protection

As it can be seen in Fig. 4, the mobile working platform unit provides access to the area of the shaft which depends on the configuration of the conveyances installed in the shaft. Moveable platform can be "pulled out" in only one direction, thus the accessible area is limited. However, it is not an issue in terms of planned applicability of the device, as it was designed to improve level of safety and convenience of works, which are conducted from the inside of the cage's compartment. Works on the other sides of the cage are usually conducted using top transom of the cage and major repairs and modernizations of shafts and their equipment requires application of standard suspended working platforms.

Working platforms for mining engineering and for shaft works in particular, has to be provide proper level of safety for working people and have relatively high payload. This is the reason why some of the constructions of working platform used in civil engineering cannot be applied in mine shafts. To prove that the mobile working platform unit meets the requirements of safety and payload, numerical simulations were conducted to analyse its behaviour. Values of bending stress and displacement obtained in the simulations confirmed that the platform provides proper level of safety. Results of the simulations are shown in Fig. 7 and 8.

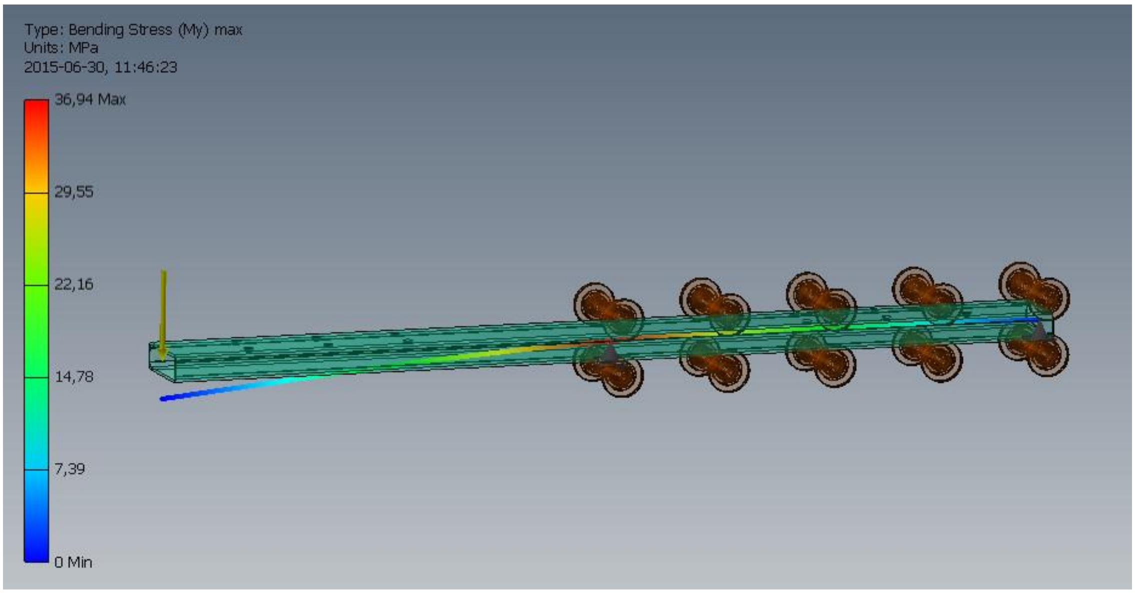

Fig. 7. Bending stress map

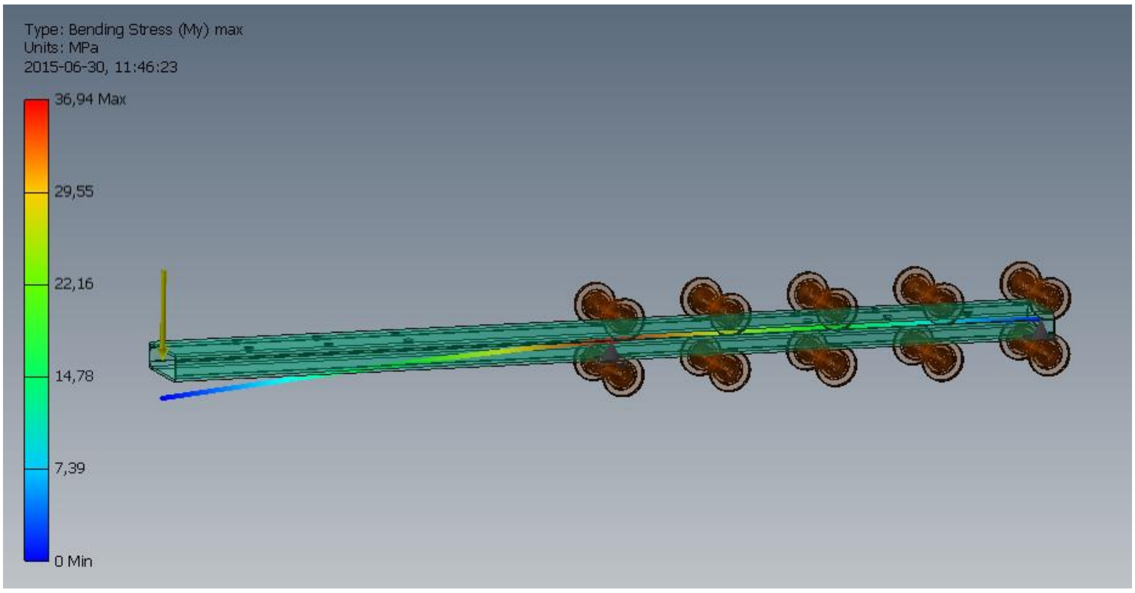

Fig. 8. Displacement map

4. Summary

Proper maintenance of the lining and equipment of mine shafts is crucial for safety of the whole mine. The problem is that all of the shaft works, including monitoring of shaft elements, maintenance and emergency works, require stoppage of the shaft operation and application of working platform of special construction. Moreover, shaft works are difficult and dangerous. Thus, the key features of working platforms are high level of safety and short time of their assembly and disassembly. Analysis of this factors was an impact for the design of the mobile working platform unit for shaft works.

The advantage of the mobile working platform unit is minimization of time required for assembly and disassembly of the construction of a platform exceeding the outline of the conveyance. The invention allows to conduct maintenance work of shaft lining, equipment or installations in a safe manner. The moveable bridge and moveable platform equipped with barriers and optionally the overhead protection significantly improves safety of people working in the shaft. The unit ensures full stability of the pull-out bridge construction by locking the motion of both the bridge and the corpus, as the unit is fixed to the construction of the conveyance. The mobile working platform unit also plays a role of transport platform for all of its elements for the time of their transport in the cage.

The platform cannot be used for all types of shaft works, as accessible of the shaft is limited, so it cannot be a replacement for all types of shaft working platforms, like typical suspended working platforms. Also it can be used only for minor repairs of shaft lining or equipment. However, the platform was designed to improve shaft works conducted using working platforms installed in the cage's compartments and it meets designed requirements.

Presented construction of the working platform might positively affect the safety and effectiveness of conducting shaft works, including its monitoring, maintenance and emergency works. Thus it can improve economic score of such ventures or even help to increase level safety of the whole mine.

5. Patents

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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.

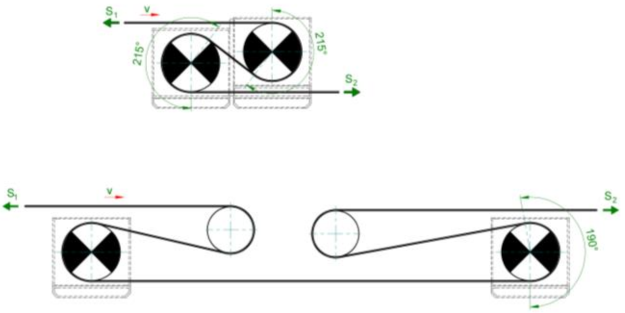

Fig. 1. Basic configurations of a two-drum drive [5]

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.

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

where:

k_z - coupling coefficient

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

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]

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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].

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

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.

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

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].

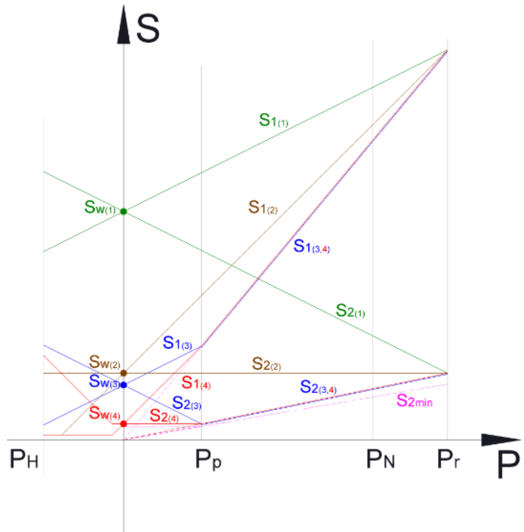


Fig. 2. Waveforms in drives with different tensioning methods [7]

(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

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).

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

– 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:

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

– 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.

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

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.

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

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 $\mathbf{i_{zl}}$ ratio, which ensures a constant ratio of forces in the belt running on the drive and running away from the drive (Fig. 3).

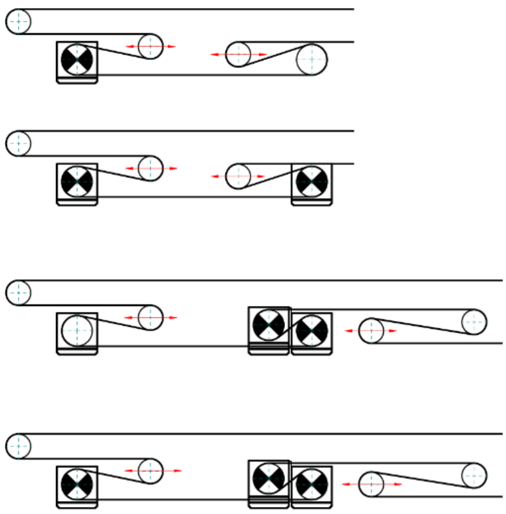


Fig. 3. Tensioning trolleys in single, double and triple drum drive systems [own sketch]

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

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

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 = S₁ / S₂ 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.

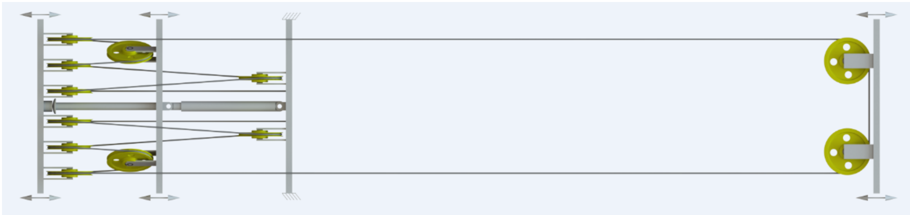
3. Results -design solutions of follow-up tensioning stations

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

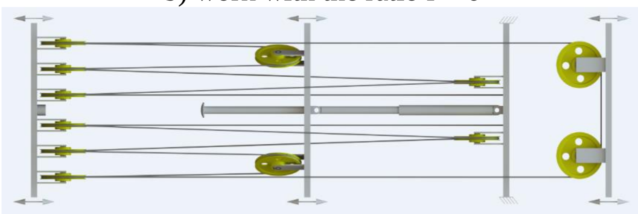


Fig. 4. Diagram of the rope circulation in the station with variable ratio i = 4/6 [own sketch]

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.

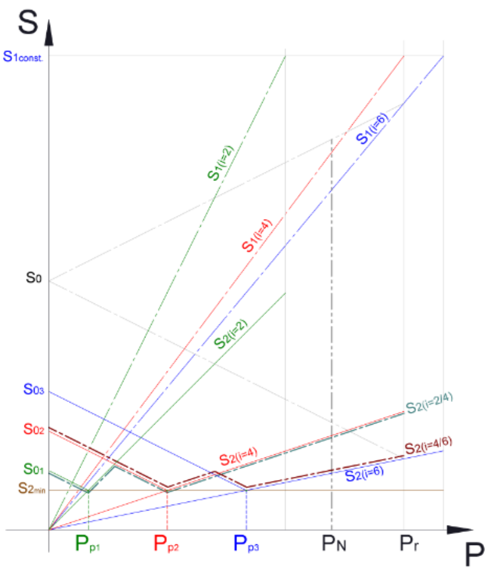


Fig. 5. The course of the S2 force generated by the tensioning station with variable ratio [15]

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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.

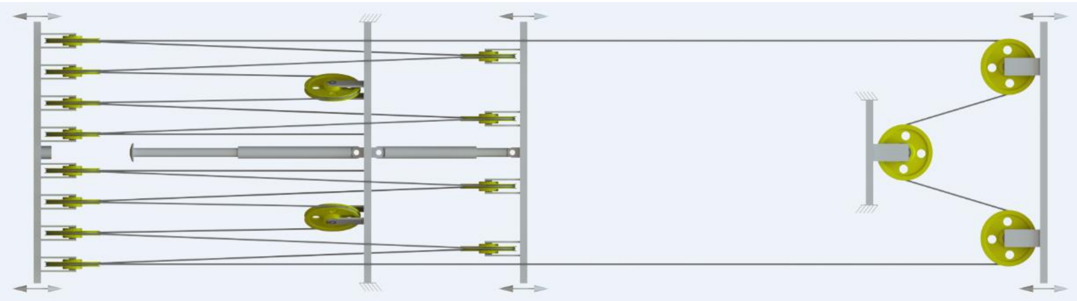

Fig. 6. Rewinding the rope of the follow-up station with a constant tension module [own sketch]

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.

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.

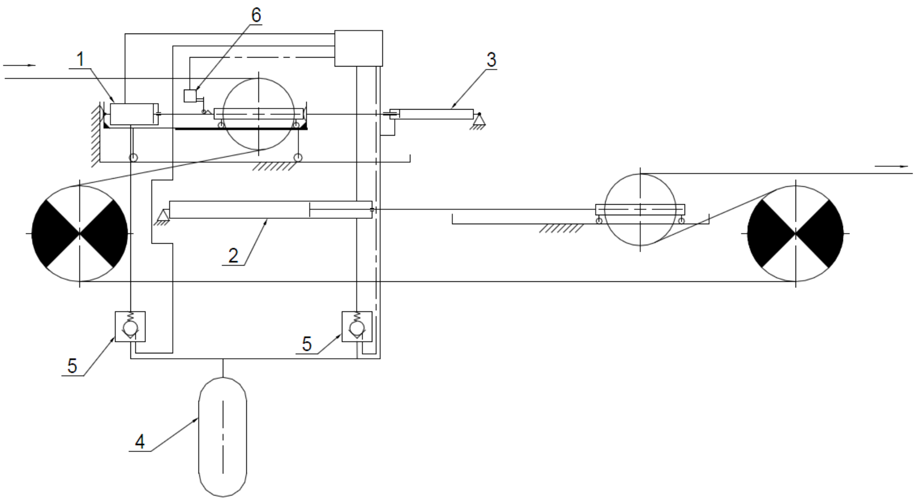

Fig. 7. Diagram of the operation of the hydraulic follow-up station [21]

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.

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

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.

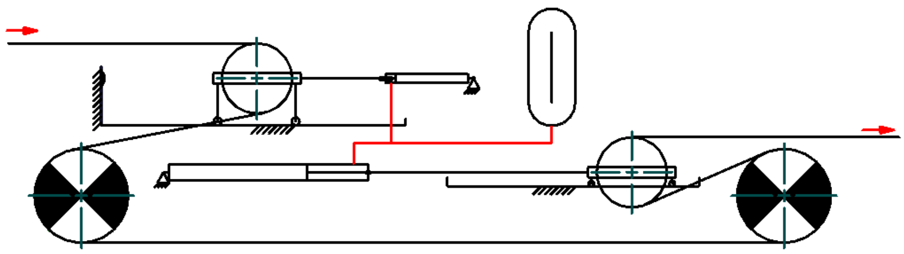

Fig. 9. Constant tension tensioning station with the function of tensioning the sliding belt during braking [21]

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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Strength analysis for cycloidal gears with the new concept of power transmissions

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Abstract:

The article presents a new design solution of hydraulic gerotor machines. The main attention was paid to the analysis of strain behaviour of a cycloidal gear set. The gears are the main assemblies of hydraulic gerotor machines, which are powered by driving the gear set. In the state of the art, the rotational motion of the gear set was effected by driving the inner gear. This article proposes a modified method of power transmission in which the outer gear is now a driving gear. Such a solution also results in modifications to the hydraulic machine design. This have led to a concept of a new hydraulic gerotor machine, which differs from the traditional designs. The device according to the new design is capable of carrying substantially higher loads than devices of the previous designs.

Keywords: strength analysis, plastics, Finite Elements Method (FEM), conceptual designing

1. Introduction

Hydraulic machines and hydraulic equipment have been used from the very beginning of human industrial activity. The first to appear were piston pumps used in maritime transport and railways [1]. Gear machines of more advanced design appeared later. Hydraulic gear machines are divided into the following three types:

- gear machines with external meshing,
- gear machines with internal meshing,
- gerotor machines.

The gerotor machines appeared at the latest. Their mass production was started by the Henry Nichols company in the USA in the 1930s in the USA.

Gerotor machines are of an innovative design solution and have many utility advantages. They have a simple structure, small dimensions and weight, high efficiency and low pulsation of this efficiency. A set of gears with internal cycloidal meshing is the key group of gerotor pumps.

Typical hydraulic georotor machine is presented in Fig. 1.

The main working unit of this machine consists of two cycloidal wheels, which are the idler wheel (1) and the driving wheel (2). They are wheels with internal meshing. The idler wheel, in this assembly, has a greater number of teeth than the driving wheel. The difference in the number of teeth is always z2-z1 = 1. The figures (Fig. 1, Fig. 2) show an example of a set of wheels, where the driving pulley (1) has the number of teeth z1 = 6, and the idler wheel z2 = 7

Two cycloidal wheels, the active wheel (1) and the passive wheel (2) are the main working subassemblies of the machine. These wheels have an internal gearing. The passive wheel in this assembly has a greater number of teeth than the active wheel. The difference in the number of teeth is always z_2-z_1 = 1. Figures (Fig. 1, Fig. 2) show an example of a wheel assembly, where the active wheel (1) has the number of teeth z_1 = 6 and the passive wheel z_2 = 7.

The wheel assembly is placed in the central body (3) and on the sides there are the front console (4) and the rear cover (5).

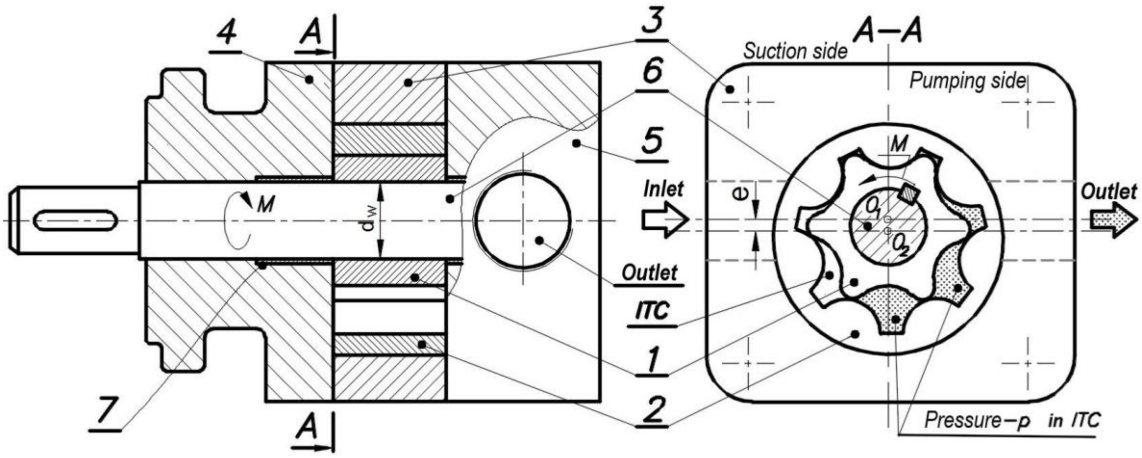

Fig. 1. Design and principle of operation of a hydraulic gerotor machine

Drive and energy are transmitted from the shaft (6) to the active wheel (1). The drive shaft (6) is placed in bearings (7), fixed in the front console (4) and the rear cover (5). Intertooth chambers (ITCs) appear between the wheels after the wheel assembly is installed on the shaft (6) and in the body (3).

The driving torque M sets the cycloidal wheel assemblies (1) and (2) into rotation. As a result of this motion, the working fluid is transported in the inter-tooth chambers from the inlet hole to the outlet hole. As it rotates, pressure in the pumped fluid increases until the working pressure p is reached. The vertical symmetry axis of the O_1O_2 wheel assembly divides the hydraulic gerotor machine into two sides, the suction side and the discharge side. The working pressure p on the discharge side (right, A-A section) acts on the wheel assembly and on components of the body.

Both wheels rotate in the same direction and there is an eccentricity e between the wheel centres. The active wheel (1) rotates around the centre of the axis O_1, while the passive wheel (2) rotates around the centre of the axis O_2.

During operation of a hydraulic machine, the wheel assembly is subjected to mechanical and hydraulic loads. These loads result in a complex state of stress and deformation. Finite element analysis (FEM) was used to determine the stresses.

FEM analysis was used by Gamez-Montero, Castilla, Khamashta and Codina [2]. They proved that maximum stress was found in a pair of teeth moving around the central point of the toothing.

FEM analyses on the strength of epitrochoids working in rotational motion were also carried out by Maiti [3]. Maiti demonstrated that deformations of each tooth have different parameters and allow the flow of the fluid pumped through the machine's inter-tooth canals [3].

The analyzes of flat two-dimensional models were described in [2, 3]. In papers [4, 5, 6], 3D models of cycloidal wheel assemblies were presented. Wheel assemblies made of plastics were analysed and the conclusions from earlier publications were also confirmed [2, 3].

Other work concerned estimation of the working parameters of gerotor machines [7], as well as their geometry [8]. Maiti, Nag and Nagao found that the initial torque depends on the position of the drive shaft [7]. The work [8] shows the unification of geometrical solutions for designing.

Numerical analysis with use of the finite element method enabled determining the behavior of the cycloidal wheel assembly. The steel wheel assembly, loaded with a working pressure p = 4 MPa and a torque M = 7.16 Nm was analysed. These loads were selected for the analysis as in the previous tests the parameters p = 4 MPa and M = 7.16 Nm were used [4, 5, 6].

The method of fixation and loading with pressure p and torque M can be seen in Fig. 1 and Fig. 4. The mechanical load with the torque M was applied to the wall of the keyway in the inner wheel, as shown in Fig. 1 (cross-section A-A).

The hydraulic load of the working pressure p was applied on the discharge side in the inter-tooth chambers ITC (Fig. 1, cross-section A-A). The pressure p in the inter-tooth chambers was applied in the same way as in the computational model in Fig. 4.

The method of fixation results also from the operation mode of the machine as in Fig. 1. The active wheel is fixed in the inner hole to simulate fixation of the inner wheel (1) on the shaft (6). The outer wheel (2) is also fixed in the hole of the central body (3), allowing the wheel assembly to rotate around the axis O2. This was done in the same way as for the model shown in Fig. 4.

The wheel set was also fixed along the Z axis to reflect the pressing the wheel assembly against the surface of rear cover (5). This fixation and pressure are shown in Fig. 4 as so-called frontal fixation along the Z axis and pressure p on the wheel face.

Result of this analysis is shown in Fig. 2, where the distribution of stresses and displacements for a loaded set of cycloidal wheels can be observed.

Based on the numerical analysis, the complexity of the stress state in the set of wheels under load was confirmed. This state includes both compressive static loads, as well as fatigue and contact loads.

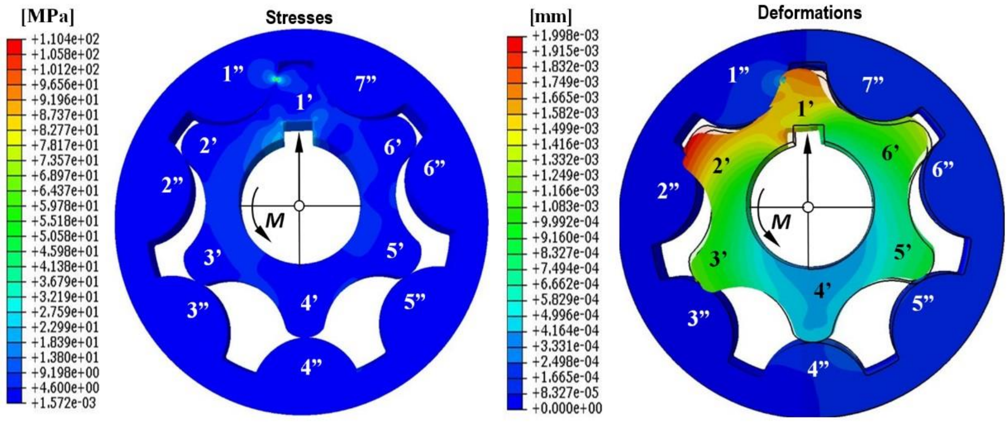

Fig. 2. Distribution of reduced stresses and displacements in a cycloidal wheel assembly

The highest reduced stresses appear in the active wheel at the point of contact of the 1'-1 "tooth pair and in the area of the keyway. The maximum value of these stresses was about 110 MPa.

Working loads also cause displacements in the cycloidal wheel assembly. It can be observed, in Fig. 2, that the greatest displacements were found in the inner wheel for the tips of the 1' and 2' teeth and on the wall of the keyway (red contours). However, these displacements do not have a significant effect on the operation of the hydraulic machine.

The most dangerous displacement takes place in the 4'-4 ” tooth pair. The working pressure of fluid in the inter-tooth displacement chambers (ITC) acts on both wheels, causing the 4 ' tooth to deflect to the left. The 4 ' tooth in the inner wheel then bends towards the suction side. As a result of this deflection, there is an inter-tooth radial clearance (hr). This causes the machine to leak and its efficiency to drop as a result of leaks.

However, the analysis shows that the displacement of the nodes for tooth 4 'is very small - less than 0.0005 mm.

Issues related to designing and operation of hydraulic gerotor machines are realized by the Fluid Power Research Group (FPRG) from the Faculty of Mechanical Engineering at the Wrocław University of Science and Technology (www.fprg.pwr.wroc.pl).

Projects covering technological, designing, operational and visualization aspects are the results [4, 5, 6, 10]. Visualization allowed to define the flow directions and show the cavitation issue [10]. The conclusions from numerical analyses, indicating that higher stresses prevailed in the inner wheel [2, 3, 4, 5, 7], were also confirmed.

It is therefore necessary to introduce changes aimed at increasing the load capacity of cycloidal wheels. To achieve this goal, however, it is necessary to introduce changes to the design of the wheel assembly, but also in a principle of operation of the hydraulic gerotor machine.

In each of the solutions, known to the author, drive in the wheel assembly is transmitted from the inner wheel to the outer wheel.

The solution used so far has several disadvantages. These disadvantages were proved both by numerical analyzes [2, 3, 4, 5] and tests [6]. These disadvantages can be summarized in several points and they are as follows:

- significant stress in the inner wheel,
- high concentration of stress in the contact places of wheels teeth, especially in the 1'-1 " tooth pair,
- high stress in the area of the hole in the active (inner) wheel, caused by transmission of the drive from the drive shaft to the active wheel,
- small diameter of the driving shaft d, limited by the diameter of the tooth base in the active wheel,
- asymmetric stress distribution in the wheel assembly.

All these disadvantages limit the range of the operational parameters with which hydraulic gerotor machines.

It should also be added that the smaller size of the inner active wheel causes that stress relaxation in this wheel is difficult, and thus stress concentrations appear in it.

It can be observed that there is a need to change the design concept for hydraulic gerotor machines. New design of a hydraulic gerotor machine should be work objective. This solution should ensure higher operational parameters, without increasing dimensions and weight of the machine.

2. Materials and Methods

2.1. New solution

New solution was developed by changing the role of both cycloidal wheels, so that the outer wheel is now the driving wheel, and the inner wheel is now the driven wheel.

The solution was described in the patent application (Gerotor hydraulic machine. Application No. P. 416532. Case mark 451-9/16).

The hydraulic gerotor machine according to the new concept, in which the role of the wheels was changed is shown in Fig. 3. In the system shown in Fig. 3, the drive is first transmitted to the outer wheel (2), while the inner wheel (1) now acts as a passive (driven) wheel.

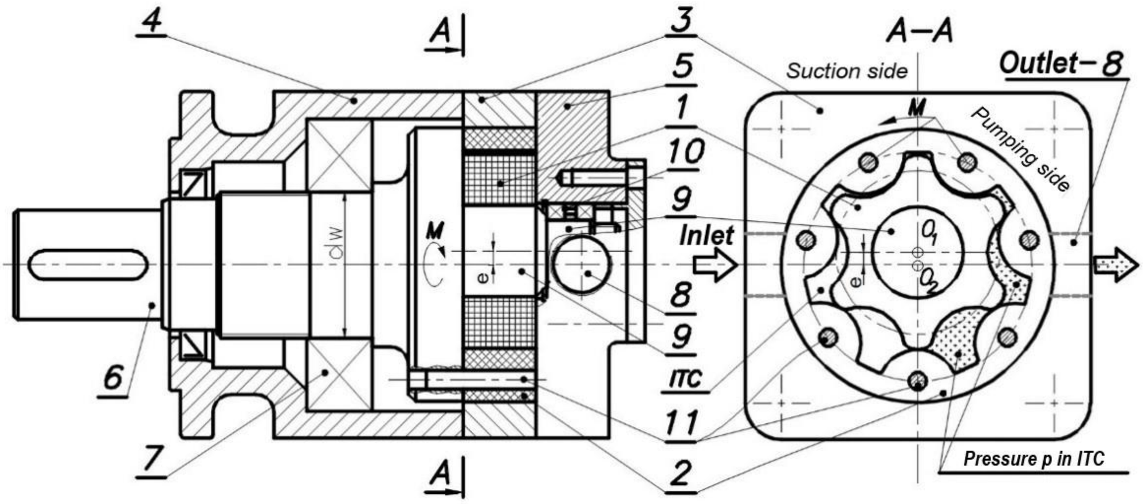

Fig. 3. Scheme of a hydraulic gerotor machine according to the new concept

The inner (1) and outer (2) wheels are still the main working components of the hydraulic gerotor machine (Fig. 3).

A set of cycloidal wheels (1) and (2) is located in the central body of the hydraulic machine (3). The hydraulic machine body assembly consists of a central body (3), a front console (4) and a rear cover (5). Pins (11) connect outer wheel (2) to driving shaft (6). Front console (4) contains bearing assembly (7) for the driving shaft (6). The outlet hole for pumped liquid (8) is located in the rear cover (5).

Fig. 3 shows that the inner wheel (1) is mounted on the second shaft (9). The shaft (9) is supported by a bearing unit (10), placed in the hole of the rear cover (5).

The connection between the inner wheel (1) and the shaft (9) is close fit.

In the solution shown in Fig. 3, the cycloidal wheels (1) and (2) are still in contact with each other by their inter-tooth contacts, as in the previous solution (Fig. 1). It can therefore be said that:

- the outer wheel (2) rotates with the drive shaft (6) in bearings (7) around its axis O_1,
- the inner wheel (1) rotates with the shaft (9) in bearings (10) around its axis O_2.

The eccentricity e is both between the wheel axles (1) and (2) and between the shaft axles (6) and (9) on which these wheels are embedded. The outer wheel (2) is connected to the drive shaft (6) using the special pins (11).

These pins (11) are used to transmit power and energy from the shaft (6) to the outer wheel (2). The A-A section (Fig. 3) shows that the number of pins corresponds to the number of teeth in the outer wheel (2). The number of pins equal to the number of teeth in the outer wheel, ensures an even distribution of the load in this wheel. In the example shown in this description, there are seven pins (Fig. 3).

Energy of torque M is transmitted from the driving shaft (6) to the outer wheel (2) via pins (11). The outer wheel (2), which now operates as an active wheel, sets the inner wheel (1), mounted on the shaft (9), into rotation.

The liquid flows from the left side (see section A-A in Fig. 3) to the inter-tooth chambers and is forced by the rotation of the wheels to the outlet (8) on the right side (A-A). The mechanical energy is then converted into hydraulic pressure in the inter-tooth chambers.

2.2 Computational model

A geometric model of the cycloidal wheel assembly was developed using the construction diagram of the gerotor pump shown in Fig. 1 and 3. The model is shown in Fig. 4. The toothing profiles of both wheels were determined on the basis of the literature [9].

During operation, the wheel assembly is subjected to hydraulic and mechanical loads. The hydraulic load of the model results from the pressure p of the working medium, while the mechanical load results from the application of the driving torque M.

The working pressure p acts in the inter-tooth chambers and on the front surfaces of both wheels. The wheel assembly remains in equilibrium because the torque M applied to the outer wheel is equal to the torque induced by the pressure p. Mechanical load by torque M is transferred to the larger outer wheel, what can be observed in the model.

The torque M was applied to seven holes made in the teeth of the outer wheel. In this way the torque M was distributed evenly between the seven teeth of the outer wheel. The model should be restraint, and the method of restraining also results from the principle of operation of gerotor pump.

Fig. 3 shows that the inner wheel is mounted on the shaft and can only be rotated around the axis O_1Z. In the model shown in Fig. 4, this wheel is radially fixed on a shaft with a diameter $d_1 = 25$ mm and rotates around the axis O_1Z.

The outer wheel in Fig. 3 can rotate inside the body around the O_2Z axis.

In the model shown in Fig. 4, this wheel is restraint radially on a diameter $d_2 = 75$ mm, but it can rotate around the axis O_2Z. The system is also loaded with the pressure p of the working medium in the gap between the wheel assembly and the body components. The wheel assembly is thereby pressed against the body by a pressure p exerted on the wheel face surfaces.

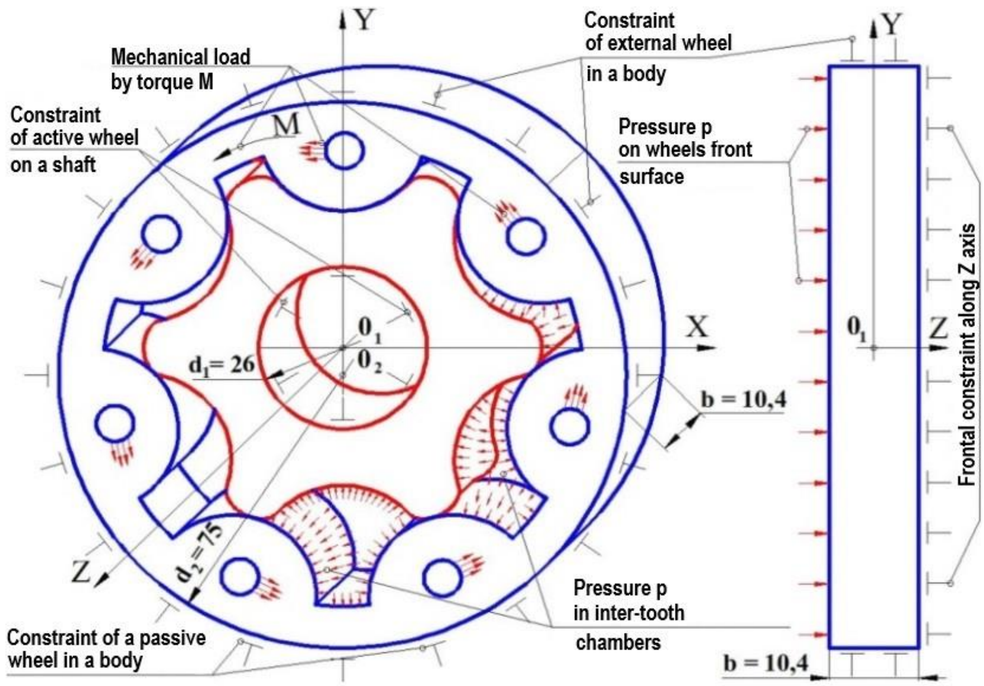

Fig. 4. Computational model of wheel assembly for a new method of drive transmission – a diagram of loads and restraints

Therefore, in the model shown in Fig. 4, the front surface of the wheel assembly is loaded with pressure p, while the opposite face is fixed frontally along the Z axis.

The numerical model of the cycloidal wheel assembly was created using the ABAQUS version 6.14-2 system. The license for this program, number 05UWROCLAW, was made available by the Wrocławskie Centrum Sieciowo-Superkomputerowe at the Wrocław University of Technology. HEXA-type cubic elements shown in Fig. 5a were used to develop the finite element mesh shown in Fig. 5b. These are typical elements for creating solid models [11]. The use of HEXA-type elements made it possible to create 3D model of the wheel assembly. The HEXA element is cube-shaped with 8 nodes. The number of nodes indicates that it is a first order element. Each node of the HEXA element can be moved in relation to X axis, Y axis and Z axis [11].

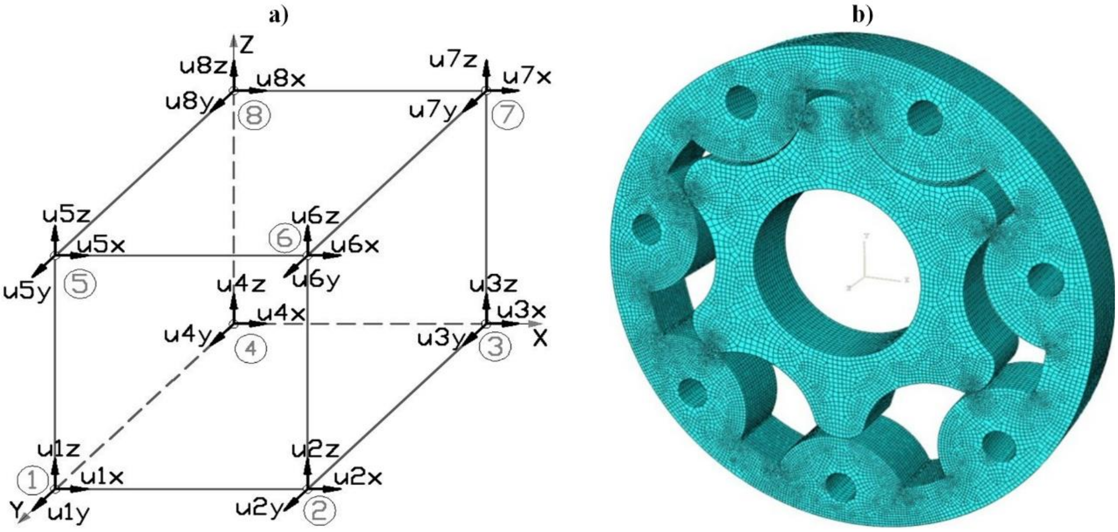

Fig. 5. HEXA type element and mesh of cycloidal wheel assembly created of HEXA type finite elements

Triangular and quadrilateral elements were used to build two-dimensional meshes of 2D cycloidal wheels [2, 3]. They were both first order and second order elements.

2D models gave the results close to the real ones. Creating 3D model allows to increase calculations accuracy, due to the better correlation of the 3D model with real conditions, as the wheel assembly was a 3D structure. Total load and restraints cannot be included in 2D models. Only in the 3D model it is possible to include pressure to the front wheels surface and the frontal restraint along the Z axis (see Fig. 4). 3D model takes into account total load to the wheel assembly. Such a model allows to estimate the axial clearance (in direction of the Z axis). This was not possible in the case of 2D models. Even if axial clearance will have a slight share in the displacement, this value should be taken into account in the analysis of the hydraulic operation of the gerotor machine. For these reasons, 3D model of the cycloid wheel assembly was created.

The finite element mesh for the wheel assembly was created of approximately 500,000 HEXA elements. The elements formed the outline of the wheel assembly in the XY plane. Fifteen layers of HEXA elements were created to model the thickness of the wheel assembly. These layers were spaced approximately every 0.7 mm along the OZ axis until obtaining the proper thickness of the wheel assembly b = 10.4 mm (Fig. 4).

Mesh densification was entered at the location of predicted stress concentration, see Fig. 5b. Points of densification are the teeth contacts of the cooperating wheels at the input side (left side of the model in Fig. 5b). Lower densification was in the corners at the base of the teeth of the outer wheel.

Previous tests show that the stresses in the tooth are of a contact nature [2, 3, 12, 13]. The HEXA element is also very good for modelling just such contacts [12, 13].

The ABAQUS system also enabled to model the contact of cooperating surfaces using a special algorithm for this system. The contact of the teeth of the wheels was assumed at the low pressure zone (left side of Fig. 4 and Fig. 5b).

The coefficient of friction μ for the contact was selected for the steel-steel contact and the most unfavorable case was adopted, i.e. for dry friction, which is $\mu = 0.1$.

The numerical model enabled determining the stresses and strains for the wheels assembly with high accuracy. Discretization errors did not exceed 0.02% for deformations and about 1.3% for stresses. These errors were estimated using the relationships provided in the literature [11].

Test program and selection of parameters for numerical analysis

At the beginning, the wheel assembly model operating under working pressure p = 4 MPa and torque M = 7.16 Nm was analyzed. The initial parameters pressure p = 4 MPa and torque M = 7.16 Nm were taken basing on previous publications [4, 5].

Aim of this analysis was to find design solutions that would allow to exceed this limit. For this reason, the parameters p = 4 MPa and M = 7.16 Nm were taken as the starting values, as the lowest.

In the next stages, working pressure p and torque M were increased. According to the test program, the model of wheel assembly was analyzed for pressures p = 4, 8, 12, 16, 20 and 24 MPa and with a proportionally higher torque M.

The load parameters are presented in Table 1.

Table 1. Computational parameters

Working pressure p [MPa]	Torque M [Nm]
4	7.16
8	14.32
12	21.48
16	28.66
20	35.82
24	42.96

Cycloidal wheels are usually made of high-grade steel. The steel was 18HCrMo4 steel (former symbol 18HGM) intended for carburizing. Strength parameters of 18HCrMo4 steel are given in Table 2.

Table 2. Technical parameters of 18CrMo4 steel 18CrMo4 (18HGM)

Item	Parameter	Symbol	Value
1.	Yield point	*Re*	830 MPa
2.	Young’s modulus	*E*	210000 MPa
3.	Steel/steel friction coefficient	μ	0.1
4.	Safety factor [14]	x	1.4 ÷ 1.6

To analyze the solution, the permissible stress and displacements for cycloidal wheels assembly should be determined.

The permissible displacements depends on the radial clearance hr. The hr_{DOP} can be determined on the basis of the literature [9]. It was assumed that the permissible hr_{DOP} = 0.1 mm and the hr clearance cannot exceed the hr_{DOP}.

The permissible stress should be determined using the Re for 18CrMo4 steel and dividing it by the safety factor x for gears [14]. When the safety factor is x = 1.5, the allowable stress is σ_{DOP} = 550 MPa.

The numerical analysis was continued until one of the above permissible values (hr_{DOP} = 0.1 mm, σ_{DOP} = 550 MPa) was exceeded.

3. Results of numerical analysis

Fig. 6 shows distributions of stresses and displacements for wheel assembly, when direction of drive transmission is changed. These distributions differ from those seen in the previous solutions [2, 4, 5] and in Fig. 2.

The maximum stress appears for tooth 1’, the same as in the traditional solution shown in Fig. 2. However, there are also the following differences:

– the maximum stress is located on the pressing side in the tooth pair 1’-7”, not on the suction side, as in the traditional solution (in the 1’-1” tooth pair),
– the maximum stress (in the contact of the 1’-7” pair) is 72 MPa and is about 35% lower than in the traditional solution (for the 1’-1” pair in Fig. 2).

The change in position of maximum stress is the result of a change in the direction of energy transmission.

In the new solution, the drive is transmitted to the outer wheel via pins (No. 11 in Fig. 3). In this case, the groove is omitted and therefore there is no groove in the hole in the inner wheel. The connection between the shaft and the inner wheel was made fit (Ø25H7/p6). For this reason, the hole in the inner wheel was strained, but the stress is lower than the stress in the contact of tooth pair 1'-7".

The greatest displacement in the described solution is now in the outer wheel. In the traditional solutions described in the literature, the inner wheel was significantly more strained [2, 4, 5] and more deformed (Fig. 2). In the new solution, the opposite is true. However, it can be seen from Fig. 6 that the maximum displacement (red) should not have a significant impact on operation of the wheel assembly.

Deformation most disadvantageous for the machine operation was found for the 4'-4" tooth pair, due to bending of this pair of teeth towards the suction zone, the same as for the previous solutions.

However, it should be noted that the displacement as a radial clearance hr is much lower than the permissible value hr_{DOP} = 0.1 mm. This displacement is low for both wheel assembly designs.

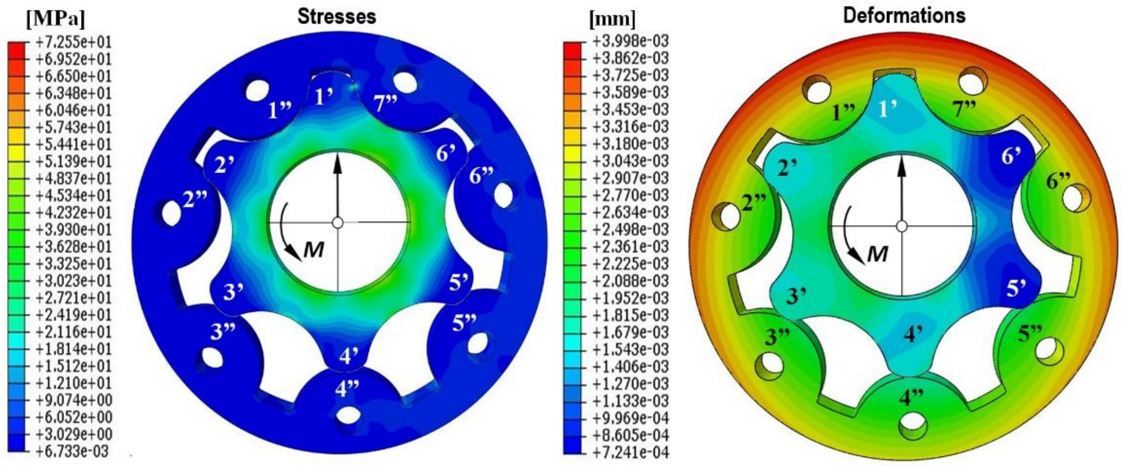

Fig. 6. Distribution of reduced stresses and deformations in a cycloidal wheel assembly according to the new solution

Fig. 7 and Table 3 present the result of the analysis in a comprehensive manner for both methods of drive transmission, which are shown in Fig. 1, 2, 3 and 6.

By analyzing Fig. 7 and data from Table 3, it can be observed that the wheel assembly according to the new solution (Fig. 3 and Fig. 6) shows lower stress in the 1'-1" tooth pair than the traditional solution (Fig. 1, 2). However, the traditional solution shows a lower radial clearance hr. Therefore, the new solution is better in terms of stresses, while the traditional one is better in terms of the radial clearance hr.

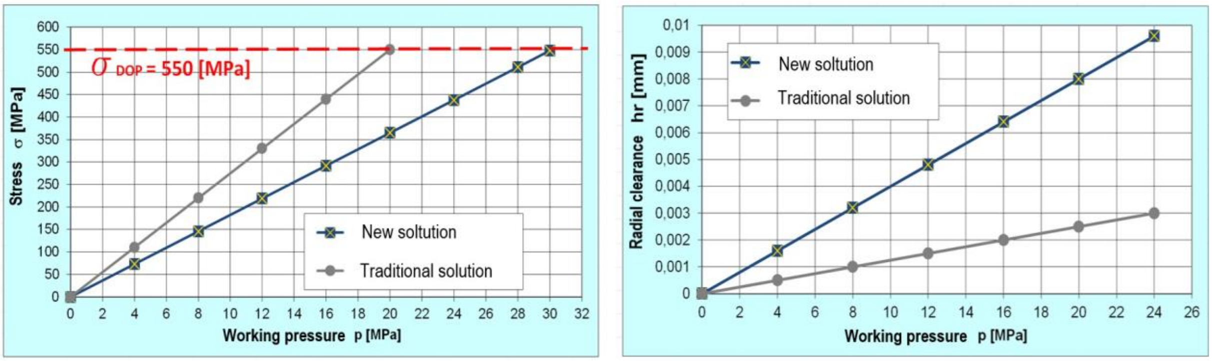

Fig. 7. Change of reduced stresses and deformations depending on the working pressure p for two types of cycloidal wheels

As the load (pressure p) increases, strain in the wheel assembly increases and the increase is linear (Fig. 7).

In the new solution, the stress is lower. According to the author, this may be due to a more even distribution of the force producing torque M. The force producing torque M is evenly distributed over the seven pins. This factor reduces the stresses in wheel assembly made according to the new solution, as shown in Fig. 3.

Table 3. Strength analysis with the use of FEM for the cycloidal wheel assembly

Working pressure p [MPa]	Reduced stress σ [MPa]		Radial clearance h_r [mm]	
	Traditional solution [Fig. 2]	New solution [Fig. 6]	Traditional solution [Fig. 2]	New solution [Fig. 6]
4	110	73	0.0005	0.0016
8	220	146	0.001	0.0032
12	330	219	0.0015	0.0048
16	440	292	0.002	0.0064
20	550	365	0.0025	0.008
24		438		0.0096
28		511		0.0112
30		548		0.0192

When analyzing Fig. 7 and Table 3, it can be concluded that gerotor machine with the new type of drive, enables obtaining the higher working pressure p than with a traditional drive transmission system.

Attention should also be paid to strength criterion. The analysis shows that for both designs (Fig. 2 and 6), a possible failure can occur as a result of destruction of the wheel assembly by increased stresses. This destruction can happen much faster before the increase in radial clearance hr causes loss of tightness and thus a decrease in volumetric efficiency for hydraulic gerotor machine.

4. Conclusions

Presented solution shows the method for increase the load limit for a cycloidal wheel assembly and a hydraulic gerotor machine.

In literature [9] describing the behaviour of wheel assembly under load, it was assumed that all pairs of teeth on the passive side are in contact [9]. An additional assumption was that the load is transferred evenly. According to this assumption, the forces acting in pairs of teeth in contact with each other are equal [9]. The numerical analysis does not confirm these assumptions.

However, the analysis confirms the need for searching a new design of hydraulic gerotor machines.

The result of analysis indicates that the hydraulic gerotor machine, designed according to the new concept, can operate with a higher working pressure p. Working loads for the new unit can be about 35% higher than for a wheel assembly using a traditional transmission method (Fig. 1 and 2).

The possibility of applying new concepts in the design of gerotor machines should be considered both from the deformation and allowable stresses point of view.

Stresses in the wheels should not exceed the stress limits (σ_{DOP} = 550 MPa), causing possible damage to the wheel assembly. At the same time, deformation of teeth and the resulting inter-tooth clearance hr should not exceed the adopted limit. Exceeding this value will result in loss of tightness, leakage and reduced machine efficiency. Based on literature data, it was assumed that the maximum clearance hr should not exceed the permissible clearance of hr_{DOP} = 0.1 mm [9].

Regarding the stress, the new solution of the wheel assembly can operate up to the p = 30 MPa of working pressure, while the traditional one can operate up to the pressure of p = 20 MPa. Therefore, the new solution is more advantageous in this aspect.

However, regarding the inter-tooth clearance hr, the traditional solution has better parameters. For both solutions the clearance hr is much lower than the permissible clearance, established as hr_{DOP} = 0.1 mm. When analysing the diagram in Fig. 7 for the clearance hr, it is clear that it does not exceed hr = 0.012 mm.

Therefore, the deformations are kept within the limits and there is no threat of unsealing the gerotor machine for both types of drive transmission. It can be observed that the stress level is more important here. For both solutions, an increase in stress will destroy the wheel assembly sooner than a decrease in efficiency as a result of leakage.

When choosing a solution, the stress criterion should be taken into account rather than the criterion of deformations. Therefore, it becomes justified to introduce a new type of drive transmission for hydraulic gerotor machines. The new type of drive lowers the stress in the wheel assembly.

Reducing the stress demonstrates the importance of introducing a new method of transmitting the drive to the cycloidal wheel assembly. The analysis shows that the new solution can operate up to the working pressure of p = 30 MPa.

However, there will also be other factors that can reduce the assembly strength. These factors include manufacturing and assembly inaccuracies, fatigue strength, temperature-dependent dimensional changes and other operating conditions.

All these factors are difficult to predict and therefore also difficult to include in numerical analysis. However, an attempt should be made to reduce the final load capacity range for the analyzed wheel assembly.

Based on the analysis the following can be observed:

– A wheel assembly with a changed direction of drive transmission can theoretically work up to a pressure of about $p \leq 30$ MPa. However, after taking into account the factors reducing the strength, the load capacity range should also be reduced to the working pressure $p \leq 25$ MPa.

According to the author, the presented solution can be used for hydraulic systems with the increased working pressure p.

The above analysis does not exhaust the subject and further studies on design modifications can be continued.

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Research on the use of hydro-pneumatic accumulators in order to reduce the flow rate and pressure pulsations of oscillating hydraulic intensifiers

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Abstract:

Oscillating hydraulic pressure intensifiers, of the minibooster type, are supplied at the inlet, in the primary, by low-pressure pumps and provide, at the outlet, in the secondary, high pressure to the hydraulic consumers (linear or rotary hydraulic motors under load). The pressure increase in the secondary, proportional to the amplification factor of the intensifier occurs at a much lower flow rate than the supply flow rate, and thus the two hydraulic parameters (pressure and flow rate) at the outlet of the intensifier are affected by oscillations. Because of this, the miniboosters are designed for static applications, which require low displacements of hydraulic motors. The authors aimed to expand the field of use of miniboosters by reducing the flow rate and pressure pulses with the help of hydro-pneumatic accumulators mounted on the primary and secondary of the intensifiers. If these pulsations can be mitigated, then low-pressure pump units of small dimensions, equipped with miniboosters, can be used in dynamic mining-specific applications, in complete safety, such as, for example, those involving relatively uniform displacement, under load, of some hydraulic jacks. A numerical simulation model developed in Simcenter Amesim highlights the effect of using hydro-pneumatic accumulators on the mitigation of flow rate and pressure pulsations of the oscillating hydraulic intensifiers. Numerical simulations performed with and without hydro-pneumatic accumulators mounted on the primary and secondary of the intensifier highlight the following aspects:

- hydro-pneumatic accumulators can be used successfully for the partial damping of flow rate and pressure pulses, but they must be dimensioned for each specific application and work optimally for a relatively narrow pressure range;
- using hydro-pneumatic accumulators can sufficiently improve the uniformity of displacement and the velocity of displacement of a hydraulic cylinder, so that it can be used in less demanding dynamic applications, as well.

Keywords: minibooster, numerical simulation, high-pressure, flow rate and pressure pulsations, hydro-pneumatic accumulators

1. Introduction

Hydraulic pressure amplifiers with oscillating pistons [1-9] are known in the literature under several names: oscillating hydraulic pressure amplifiers, oscillating pumping units, pressure intensifiers, boosters, miniboosters (miniBOOSTER Hydraulics).

The oscillating hydraulic pressure intensifier (OHPI) is used to generate higher pressure using a low-pressure hydraulic power source. Considering the high-pressure flow pulses, OHPIs can be single acting, SAOHPI (higher pulsations; they pump on a single direction of piston movement) or double acting, DAOHPI (lower pulsations; they pump on both directions of piston movement).

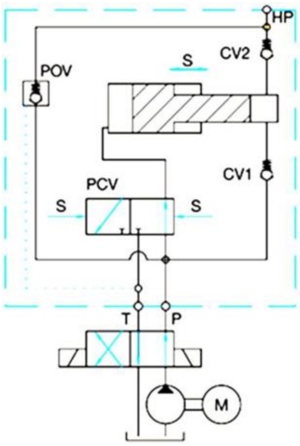

Fig. 1. Structure and operation of the SAOHPI: operating principle [11]

The basic structure of an SAOHPI:

- **assembly of two pistons** of different diameters, connected by a rod;
- **PCV**= piston control valve (bistable piston distribution valve);
- **CV1**, **CV2**= check valves;
- **POV**= pilot operated check valve;
- **P, T** = ports for consumers of a 4/2 directional control valve;
- **M** = drive motor of a hydraulic pump with fixed flow rate and low pressure.

The position of the pistons will determine, at the end of each stroke, a signal **S** to the **PCV**, which will cause a change in the direction of piston travel. This "pulsating" cycle of piston movement, with a maximum frequency of 20 Hz [10], lasts until the end pressure is reached, after which the piston stops. Onwards, they will only move to maintain the end pressure at **HP** port.

The operating principle of an SAOHPI (Fig. 1) is as follows: a large fluid volume and low pressure pushes a large diameter piston, which is in contact with another piston, of small diameter; as an effect of this action, the small diameter piston will push a small volume of fluid, with high pressure, **HP**, equal to the low pressure, **LP**, amplified by the ratio of piston surfaces. The high pressure, **HP**, will always be proportional to the supply pressure of the large piston [12, 13].

The authors aimed to expand the field of use of miniboosters, reducing the flow rate and pressure pulses with the help of hydro-pneumatic accumulators mounted on the primary and secondary of the intensifiers; in this regard, they set the following objectives:

– identification of flow rate and / or pressure pulses caused by the way an oscillating pressure intensifier works;
– quantitative and qualitative measurement of flow rate and pressure pulses; their influence on the operation of a hydraulic cylinder connected in the secondary of the oscillating pressure intensifier;
– quantitative and qualitative measurement of the pressure pulses from the primary of the intensifier, where the low-pressure hydraulic pump is connected;
– taking constructive measures to reduce flow rate pulses and / or pressure pulses and presenting the effect of these measures.

The four mentioned objectives will be achieved in two work stages of an applied research project, namely:

– **numerical simulation** of the dynamic behavior for a hydraulic supply system containing a low pressure pump group, which feeds the primary of a minibooster and a hydraulic cylinder, which is fed from the secondary of the minibooster [14];
– **experimental identification** of the numerical simulation model, on an equivalent test stand, which will be developed under the project [15-20].

In order to fulfill the previously presented objectives and because, at this point, the development of the stand has not been completed yet, so it is not possible to perform physical experiments in the laboratory, *this article only deals with issues related to the first stage* (numerical simulation).

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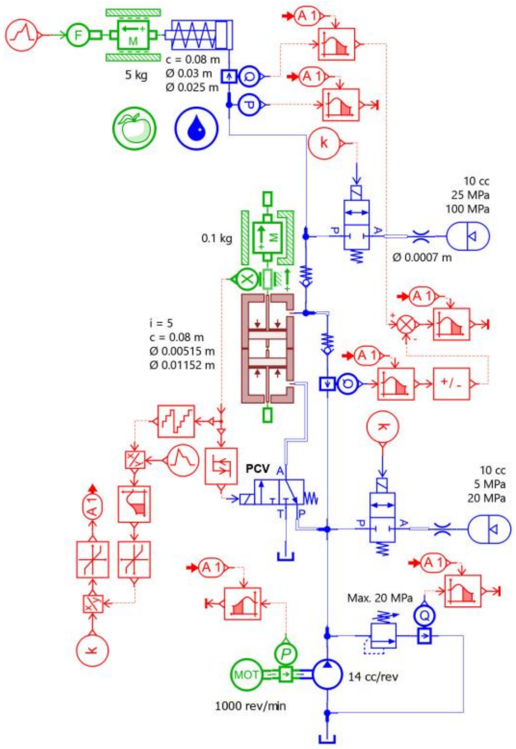
2. Materials and Methods

Both the numerical simulation model and the equivalent test stand on which it will be identified experimentally have been / will be developed for a dynamic application using an HC7 minibooster, respectively a hydraulic supply system composed of: a low-pressure pumping group; a Hydraulics A / S minibooster: HC7-5.0-B-12, whose primary is fed by the pumping group; a hydraulic cylinder, which is fed from the secondary of the minibooster and moves a load over the entire stroke.

This article partially presents the results of numerical simulations; on request, all the results obtained from running the numerical simulation model in Fig. 2, and the graphs corresponding to the texts written in “italics”, from chapter 3 can be delivered.

2.1. Presentation of the numerical simulation model

On the numerical simulation model in Fig. 2, developed using the Simcenter AMESim software, both the constructive measures taken to reduce the pulsations and their effect will be virtually tested. In this sense, the simulation model comprises two hydro-pneumatic accumulators and two 2/2 directional control valves, with electric control, which connect / disconnect the accumulators to / from the hydraulic pumping system and the hydraulic cylinder, to present on the same graph the differences between the two situations studied.



The structure and initial data of the numerical simulation model

The oscillating hydraulic intensifier:

- amplification ratio = 5/1; the two diameters of the pistons: 5.15x10^{-3} m and 11.52x10^{-3} m; total mass of the pistons = 100x10^{-3} Kg; the maximum pressure: in the primary = max. 20 MPa; in the secondary max. 100 MPa.

PCV is also part of the intensifier.

The electric motor: *rotation speed = 1000 rev/min.*

The hydraulic pump with gears:

- capacity = 14x10^{-6}m^{3}/rev.

The pressure valve: *acting as a safety valve; set at 20 MPa; natural frequency of 68 Hz.*

The 2 hydraulic accumulators: *- volume = 10x10^{-6}m^{3};*

- the initial pressure: in the primary = 5...20 MPa; in the secondary = 25...100 MPa; natural frequency of 14 Hz.

The hydraulic cylinder:

- stroke = 80x10^{-3} m;

- diameter of the piston = 30x10^{-3} m;

- diameter of the rod = 25x10^{-3} m.

The 2/2 directional control valves.

Fig. 2. Numerical simulation network

The numerical simulation model in Fig. 2 also embeds other initial data, namely: other blocks with various functions, various measurements, signal processing (moving average) and calculation of pumping frequency; hydraulic oil HEP46; - sampling rate 1000 Hz, 3001 points, simulation time 3 s, a tolerance of 1x10^{-7}. The numerical simulation aims to highlight the influence of the intensifier on the hydraulic system. In order not to influence the results presented, the pump, the directional control valves and the hydraulic cylinder have no volume losses and have an ideal behavior, as well as in the case of the two chambers of the intensifier, which if they do not have flow losses, represent the most unfavorable situation in terms of pressure and flow rate pulses. The components with real behavior are: the hydraulic pipes take into account the compressibility of the fluid, flow resistance created by surface roughness and the inertia of the fluid column; the gas cushion of the hydro-pneumatic accumulators has a real gas behavior and the one-stage safety valve has a dynamic behavior

(2nd order system), as well as the two directional control valves. The oil used is HEP46 with extreme pressure additive.

2.2. Mode of operation for simulation model; brief description

The electric motor drives the hydraulic pump with a constant rotational speed (Fig. 2). This one supplies, alternately, constant flow in the primary and the secondary of the intensifier, with the help of the PCV directional valve; due to the ratio of the areas of the two chambers of the intensifier, the pressure is **amplified 5 times**.

The two check valves are connected in the secondary of the intensifier; their role is to prevent high pressure from reaching the pump.

In the secondary of the intensifier the single acting hydraulic cylinder is also connected. On this numerical simulation model, the 2/2 directional control valves have the role of disconnecting the two hydraulic accumulators, in order to show **the differences in the operation of the hydraulic system, with the hydraulic accumulators connected** or **disconnected**.

Because on the experimental test stand the intensifier is connected directly to the hydraulic cylinder port, only the check valve and a direct hydraulic connection are interposed between the volume of the secondary of the intensifier and the dead volume of the hydraulic cylinder, without having an additional volume of fluid to influence the results of the simulation.

3. Results

For the identification and quantitative and qualitative measurement of flow rate and / or pressure pulses, as well as their influence (effect) on the operation of the hydraulic cylinder, initially four simulations were performed: a) the first of them did not have any accumulator connected; b) the second had only one accumulator connected to the secondary of the intensifier; c) the third had an accumulator connected to the primary of the intensifier; d) the fourth had both accumulators connected.

After performing the four simulations mentioned above, it was found that:

- connecting an accumulator to the primary of the intensifier does not bring any benefit in the primary or secondary;
- *connecting an accumulator in the secondary of the intensifier visibly improves the uniformity of the flow and pressure only at the hydraulic cylinder port*;
- ***by connecting one accumulator in the primary and one in the secondary***, the amplitude of the pressure pulsations in the primary decreases, and the uniformity of the flow and pressure in the secondary improves considerably compared to the previously mentioned case, in which only one accumulator was connected in the secondary.

Taking into account the above observations and in order to make the results presented as clear and easy to compare as possible, the following graphs show ***in red the results of the simulation without accumulators*** and ***in blue the results of the simulation with two accumulators***, one in the primary, and the other one in the secondary.

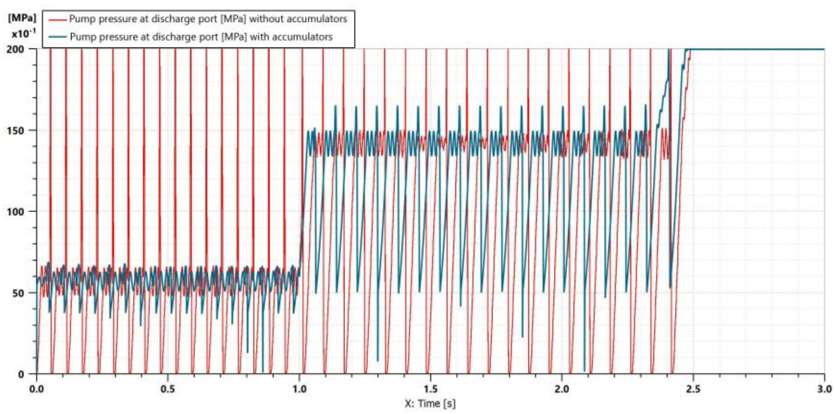


Fig. 3. Variation of the pressure at the discharge port of the hydraulic pump over time

Fig. 3 shows the pressure at the discharge port of the pump. It can be seen that the accumulator helps reduce the amplitude of the pressure pulses and tends to even out the resistant torque. On the inactive stroke of the intensifier, without accumulators the pressure reached 20 MPa, and with accumulators, the maximum pressure in the primary reaches the value of 16 MPa.

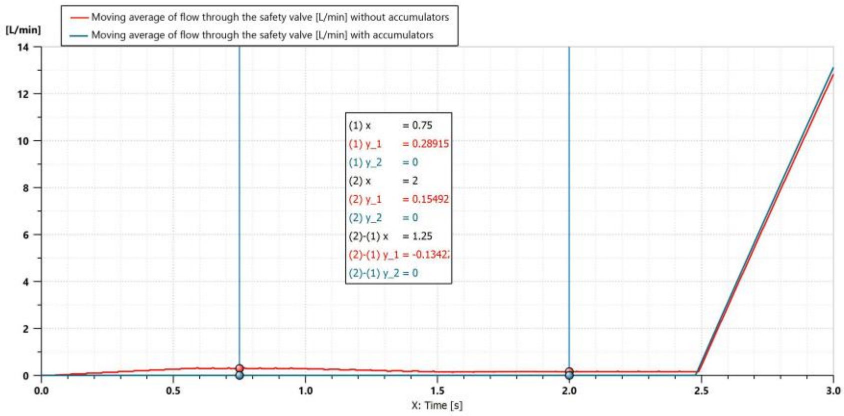


Fig. 4. Moving average of flow rate through the safety valve

The advantage of maintaining a pressure with a value of less than **20 MPa** in the primary is that in the case of using accumulators, the safety valve no longer discharges (on average) the 0.2 l/min (Fig. 4) at a pressure of 20 MPa with a frequency of pulsations of approximately 25 Hz. Eliminating these frequent discharges prolongs the service life of the safety valve. The safety valve is not the only component in the primary of the intensifier that is affected by pressure variations, pulsations caused by the way the intensifier works; another such component is the pump. It is considerably shortened in service life if it is repeatedly subjected to large pressure variations. When pressure variations have large amplitudes and their value reaches zero, they tend to destroy the pump housing relatively quickly, due to the phenomenon of fatigue of the housing material and noise.

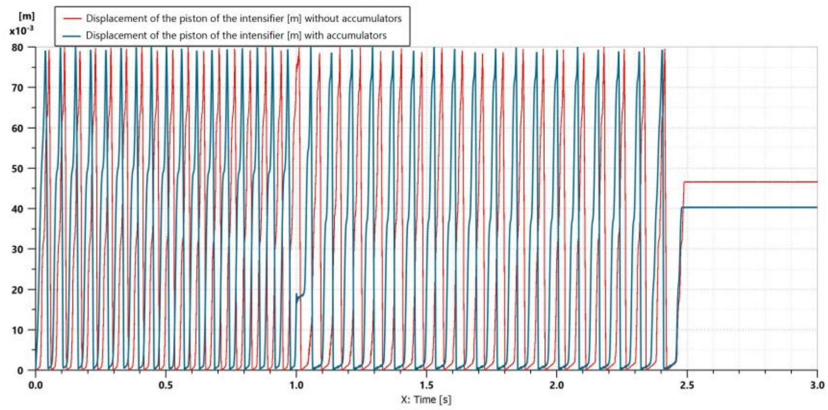


Fig. 5. Variation of the stroke of the hydraulic oscillator (intensifier's piston) over time

Fig. 5 shows the variation of the hydraulic oscillator stroke over time, in the two studied cases: it can be seen that the system with accumulators reacts late to the sudden increase of the load; in the second case, around second 1 of simulation, the appearance of a linear distribution around 18 mm value (blue curve) with a duration of 0.03 s can be noticed. The appearance of this linear distribution is caused by the sudden increase of the load in the secondary of the intensifier; this downtime occurs due to the elasticity of the primary.

Another aspect worth mentioning is the fact that, usually, hydraulic systems containing accumulators have a delayed response, which on the graph in question, for relatively small loads, does not seem to be true; in fact, only in this case, as it can be seen in Fig. 6, the pumping frequency for the simulation with accumulators increases slightly, because the pressure in the primary is less than

20 MPa, and part of the flow rate of the gear pump is no longer lost through the safety valve, as in the case of accumulator-free simulation.

In Fig. 6, it can be seen how the pumping frequency of the intensifier is affected by the load in the secondary and the flow rate lost through the safety valve in the primary.

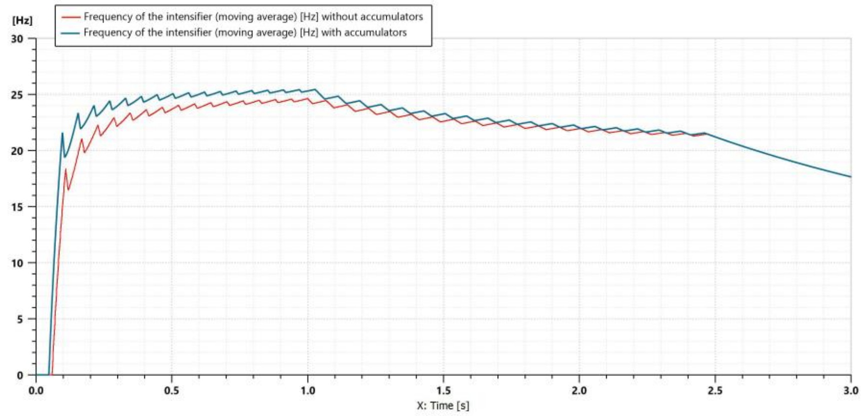


Fig. 6. Variation of the moving average frequency of the hydraulic oscillator (intensifier piston) over time

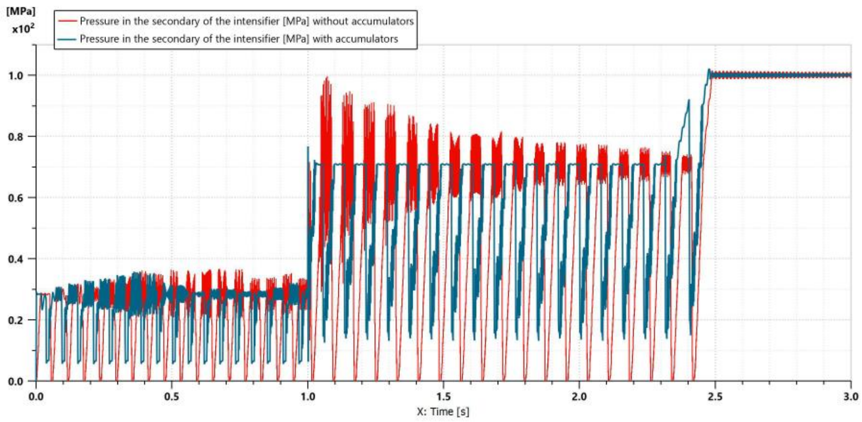


Fig. 7. Variation of the pressure in the secondary of the intensifier over time

The variation of the pressure in the secondary of the intensifier over time is shown in Fig. 7. It can be seen how the hydraulic accumulator "cuts" the pressure peaks, which can reach 100 MPa, when a relatively large load suddenly appears on the secondary of the intensifier. Also on this graph, one can see that, when using accumulators, the pressure does not reach very low values.

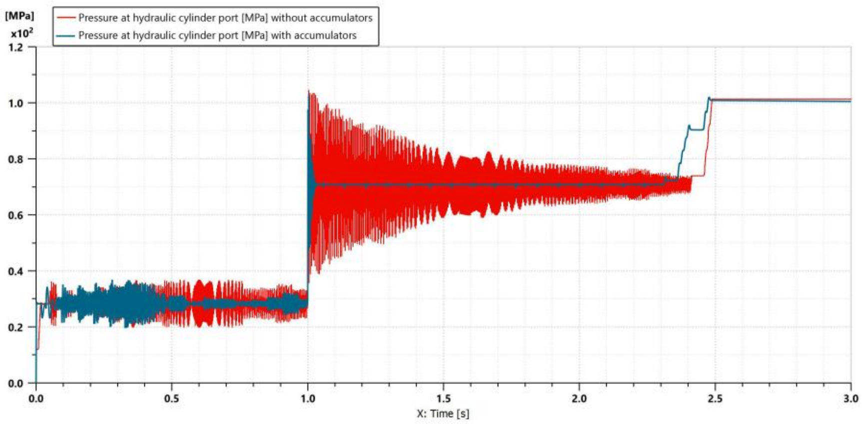


Fig. 8. Variation of the pressure at the port of hydraulic cylinder over time

Fig. 8 shows the variation of the pressure at the hydraulic cylinder port over time. The hydraulic accumulators reduce the amplitude of the pressure pulsations in the secondary, with average values between 20 MPa and 30 MPa, at a maximum of 1.6 MPa, around 70 MPa for which the two hydraulic accumulators were dimensioned.

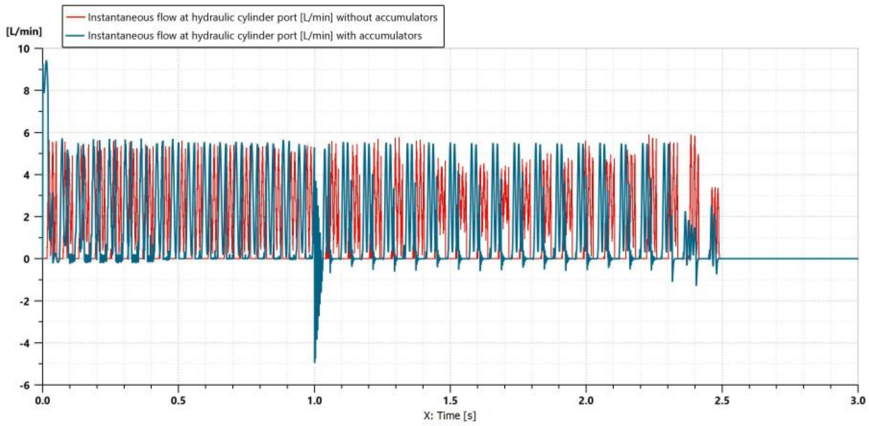


Fig. 9. Variation of the instantaneous flow rate at the hydraulic cylinder port over time

Fig. 9 shows the time variation of the instantaneous flow rate at the port of the hydraulic cylinder. At the beginning of the simulation, for a very short period of 0.02 s, almost all the pump flow bypasses the intensifier and pressurizes the hydraulic cylinder, until the pressure in the secondary increases sufficiently and the intensifier begins to amplify the pressure, sending to the cylinder an instantaneous flow of maximum 6 l/min.

From the results of the numerical simulation, it can also be observed:

The influence that the sudden application of a large load has on the flow rate, the accumulator compensating for this shock;

How the hydraulic accumulator helps even out flow; it does not perfectly equalize the flow because the volume of the accumulator is not large enough. If the volume of the accumulator is increased from $10 \times 10^{-6} m^3$ to $12 \times 10^{-6} m^3$ or a higher value, the hydraulic cylinder reaches a considerably decreased flow due to the elasticity of the accumulator;

On the moving average of the pressure in the hydraulic cylinder, three linear distributions can be noticed; two of them are created by the load on the rod of the hydraulic cylinder, and the last one occurs because the hydraulic cylinder reaches the end of the stroke.

Fig. 10 shows the variation of the moving flow rate average at the hydraulic cylinder port over time. On this graph, one can see that the intensifier discharges a flow rate between 1.5 and 2 l/min, depending on the pumping frequency, performed at relatively low pressures, and when the pressure increases, the pumping frequency decreases simultaneously with the flow, having an average value of 1.28 l/min for high pressures.

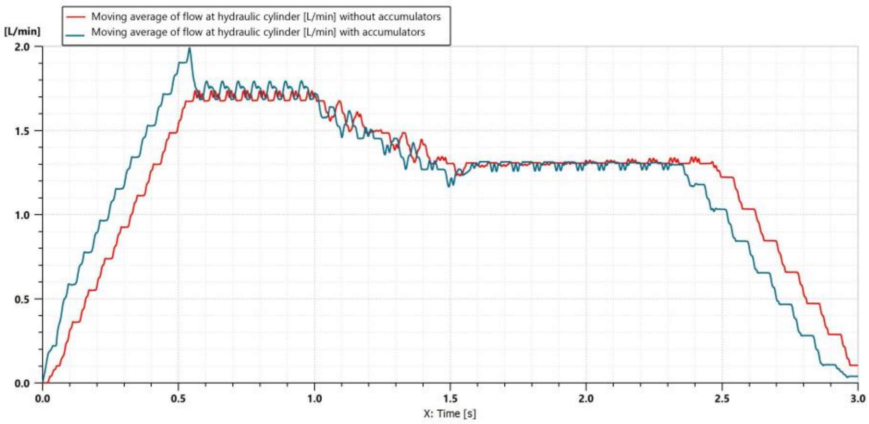


Fig. 10. Variation of the moving flow rate average at the hydraulic cylinder port over time

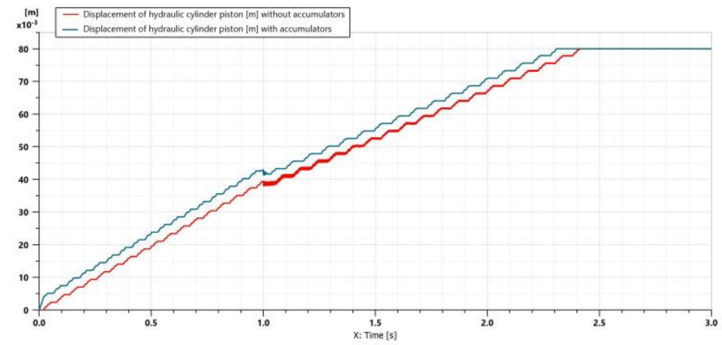


Fig. 11. Variation of the displacement of the hydraulic cylinder piston over time

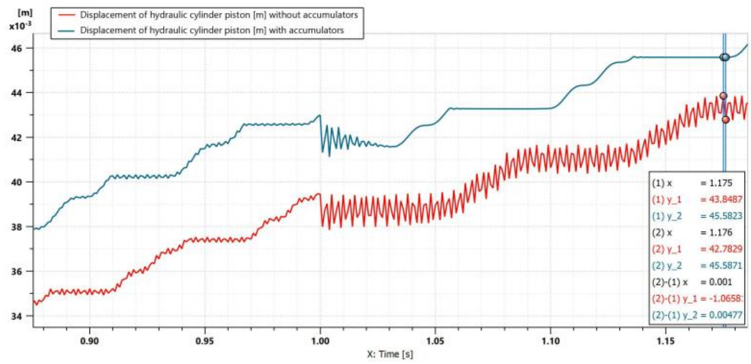


Fig. 12. Detail: displacement of the piston of the hydraulic cylinder

Fig. 11 and 12 show the time variation of the displacement of the hydraulic cylinder piston. On them, one can notice that the accumulator helps even out the displacement of the piston of the hydraulic cylinder, in the case of loads that produce values of pressures of approx. 70 MPa, in which case the accumulator in the secondary of the intensifier was chosen and charged with nitrogen.

On the detail in Fig. 12, one can see that the vibration with an average amplitude of 1.07 mm, which corresponds to a pressure of 70 MPa, is considerably reduced, in the case of using hydraulic accumulators, to a value with an amplitude of 0.0048 mm. A visible reduction of the vibration amplitude can also be observed for the pressure of 30 MPa, this effect being a secondary one; so the accumulator helps with damping for this pressure as well, but the damping is not optimal.

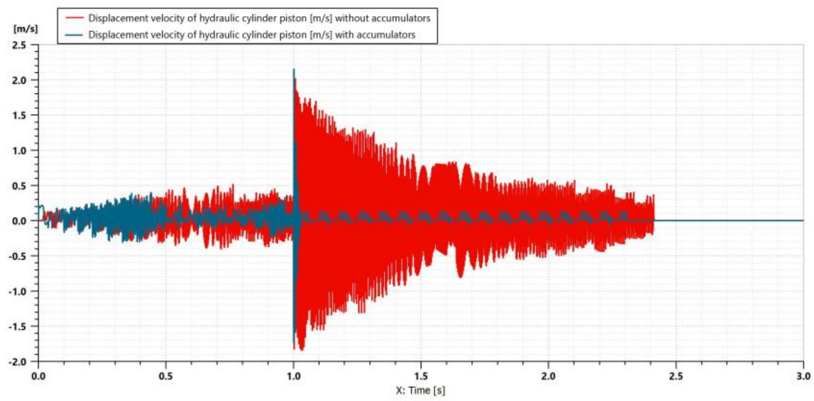


Fig. 13. Variation in time of the speed of movement of the hydraulic cylinder piston

Fig. 13 shows the variation of the speed of the hydraulic cylinder piston over time. One can easily notice the influence of the hydraulic oscillator on the speed of movement of the hydraulic cylinder rod, and how these speed variations are considerably reduced, if the load has values of 70 MPa and the two accumulators are connected.

The numerical simulation also shows:
- the amplitude of the speed variation was measured, in case of using hydraulic accumulators. This variation has a maximum value of 0.2 m/s, and without accumulators, the average value is 2 m/s, for pressure values of 70 MPa. Visible damping also occurs for the pressure of 20 MPa;
- the acceleration of the piston of the hydraulic cylinder. The accumulators restrict by a few orders of magnitude the variation of the amplitude of the acceleration;
- the force produced by the hydraulic cylinder piston and the force that opposes the displacement of the hydraulic cylinder piston;
- the force produced by the piston of the hydraulic cylinder. For the working pressure of 70 MPa; the average amplitude of the force variation is 20000 N, this variation being reduced to a maximum value of 1277 N.

4. Discussion

Lack of public information about several constructive - functional characteristics of the miniboosters (piston diameters, oscillation strokes / frequencies of the piston assembly and the PCV, the size of the clearance between the moving parts and the bores in which they move, the flow to the primary output port of intensifier) requires obtaining them, by direct or indirect measurements, in the experimental identification on the stand, followed by adjustment of the numerical simulation model (experimental validation of the simulation model). This is the direction for the continuation and completion of this research.

5. Conclusions

Hydraulic accumulators can be used successfully for the partial damping of flow and pressure pulses, but they must be dimensioned for each specific application and work optimally for a relatively narrow pressure range. In order to ensure the effective operation of the accumulator, as a pressure pulsation filter for a frequency of 15 to 25 Hz (intensifier frequency average range), the natural frequency of the accumulator must be close to the lower limit of the above range, and the frequency of accumulators used is 14 Hz.

Using suitable hydraulic accumulators can sufficiently improve the uniformity of movement and travel speed of a hydraulic cylinder, so that it can be used in less demanding dynamic applications. The pressure pulsations in the intensifier's primary, those that reach the hydraulic gear pump, were also reduced due to the accumulator in the intensifier's primary; the shocks produced by the load in the secondary of the intensifier no longer reach the hydraulic pump, being compensated by the accumulator.

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KOMAG activities in the domestic and international research areas

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Abstract:

The paper presents some information about the scientific, research and technical potential of the KOMAG Institute of Mining Technology. Some examples of achievements in the domain of machines and equipment for underground mining and preparation of minerals are given. Testing activities in the KOMAG accredited laboratories are highlighted, among others tests of powered roof supports, hydraulic executive elements, safety of products. Research project results, concerning widely understood environmental protection, are described.

For more than seventy years KOMAG has been an important scientific base for the mining industry, offering innovative solutions and competing successfully on the Polish and foreign markets with other research organizations of similar type. Its scientific and technical output includes over 6000 technical documentations of machines and equipment implemented in Poland and abroad as well as over 4400 patents and utility patents which confirm an innovative character of technical solutions developed at KOMAG.

Keywords: mining machines, testing potential, research projects, certification

1. Introduction

Last year the KOMAG Institute of Mining Technology celebrated the Jubilee of its seventy year activity. An establishment of the Institute was preceded by organizational-and-formal activities oriented onto changing not only the name of the organization, but also the scope of its functioning. The history of the Institute dates back to 1950, when the Central Office of Projects, on the base of the Minister's of Mining Regulation, dated 5th May, was transformed into the Design Office of Mining Machines. It was the first step towards a creation of organizational and technical forms of design and testing base for the Polish mining industry, tending towards a revival after the World War II, for a mechanization of exploitational processes of hard coal and a creation of mining machinery industry.

During the seventy-year period several changes of the name and subordination of the Institute took place. Since January 1, 2009 KOMAG has been operating as the Institute of Mining Technology, at present supervised by the Minister of the State Assets [1].

Scientific research and technical projects, realized at KOMAG, include mechanical and mechatronic systems, among others, machines and equipment for an underground exploitation of deposits, mechanical preparation of minerals as well as the systems of supply, control, diagnostics and monitoring of machines and equipment. Concepts, projects and technical documentations of machines and equipment, having a broad scope of applications as well as expert opinions, concerning a selection of machines and equipment for complex functions and tasks but also specific conditions of their location and operation, are developed. Model, laboratory and industrial tests are also conducted. The research projects and tests are oriented onto increasing life and reliability of machines and equipment as well as work safety and ergonomics.

Specialistic research and testing activities, realized in the accredited laboratories, among others in the scope of testing powered roof supports, hydraulic executive elements, tests of products' safety and material engineering are also carried out.

Some research projects concern a widely understood environmental protection in mining, post-mining and in other industrial areas.

The projects, realized at the Institute, also incorporate technology transfers, standardization, a protection of intellectual and industrial property rights and a certification.

The scope and subject-matter of research-and-development projects have been subject to modifications and up-dating since the beginning of the KOMAG activity, which resulted from the current demand of developing industry subject to restructuring processes.

KOMAG has been and still is an important scientific base for the mining industry, including the mining machinery industry. As one of few organizations, despite dynamic economic changes, it operates on the base of the Polish capital and competes successfully on the Polish and foreign markets with other research organizations of similar scope of activity.

An unquestioned output of the 70-year activity of the KOMAG Institute encompasses over 6000 technical documentations of machines and equipment implemented in mines of minerals in Poland and abroad as well as over 4400 patents and utility patents which confirm an innovative character of technical solutions developed at KOMAG.

An achievement of such a high position of the Institute in the domestic and international research areas was possible due to a close collaboration with producers and users of machines and equipment for the mining industry and also with technical universities, research-and-development institutes, mining supervisory organizations and local administration. The synergy effect of the scientific and industrial sectors includes products and services which are safe for a user and environment friendly.

In the process of creating state-of-the-art machines and equipment the IT and telecommunication technologies are used. Mechatronic systems, in which diagnostic and monitoring systems as well as robotics play an essential role, are generated. Computer-aided designing and other methods, called engineering of knowledge, are commonly used. Due to these activities the KOMAG Institute takes an active part in the process of building the economy based on knowledge and innovations.

The KOMAG strategy takes into consideration future needs of users and producers of machines and equipment for the mining industry, offering interdisciplinary research and testing services as well as advisory activity. An important trump is a young, ambitious staff of scientific, research and engineering-and-technical employees of broad competences. Industrial partners of the Institute from the branch of mining machines and equipment come from big capital groups and on the other hand they represent a group of medium, small and micro enterprises. A collaboration with industrial partners is realized at the support of the State Agency of Entrepreneurship Development. The results of research projects, realized by KOMAG, are used in particular by: the Polska Grupa Górnicza S.A. (Polish Mining Group, J.S.C), the Jastrzębska Spółka Węglowa S.A. (Jastrzębska Coal Company J.S.C.), the Tauron Wydobycie S.A. (J.S.C.), the Węglokoks S.A. (J.S.C.) and Lubelski Węgiel Bogdanka S.A. (J.S.C.)

2. Potential of the KOMAG Institute in the aspect of challenges of market-oriented economy

The mission of KOMAG includes a creation of innovative solutions for the economy. In the vision KOMAG is defined as a research institute of organizational-and-proprietary structure adapted for the market activity in the European Research Area. The Institute's organizational culture creates a friendly climate for generating new ideas and for realizing innovative activities i.e. transforming new ideas into new products.

2.1. Subject and scope of the KOMAG research activity

The subject of KOMAG activity encompasses a realization of scientific, research and development projects in the field of mechanization of extracting and beneficiation processes as well as in the field of air protection, surface protection and waste management connected with mining and processing of minerals, adapting the research projects results to an industrial application.

The projects are developed, among others, in the following fields:

- smart mechatronic systems,
- innovative solutions controlling hazards and increasing work safety,
- innovative transportation systems for conveying people in minerals' production plants,
- technologies and technical means for a beneficiation and a classification of minerals,
- environment management in industrial areas,
- technologies and methods for environmental protection,
- smart solutions in supply, control, diagnostic and monitoring systems of machines and equipment,
- innovative hydraulic and pneumatic systems of machines and equipment,
- innovative drive systems.

The KOMAG Institute also realizes research-and-development projects, based on contracts concluded with producers of machines and equipment, their users and authorities of towns, communes and other customers in the scope given below:

- winding machines and equipment,
- transportation equipment,
- powered roof supports,
- power hydraulics and hydraulic systems,
- cutting and auxiliary machines,
- explosion-proof equipment,
- equipment for ventilation and dust control in roadway workings,
- equipment for "small mechanization",
- electric equipment of explosion-proof machines and equipment,
- mechatronic systems for machines and equipment,
- Internet of Things.

2.2. Research and testing potential

The research and testing potential of the KOMAG Institute of Mining Technology includes personnel and material resources grouped in divisions, laboratories and departments. Scientific, research and testing projects are conducted there. These activities are aided by organizational departments, taking an active part in the projects planning and management processes as well as in the processes of quality and knowledge management. A crucial role in these activities is played by the testing potential gathered in the Institute laboratories.

Three accredited testing laboratories are active in the organizational structure of the KOMAG Institute of Mining Technology. They which conduct tests according to the scope of their accreditation.

- the Laboratory of Tests,
- the Laboratory of Applied Tests,
- the Laboratory of Material Engineering and Environment.

2.2.1. The Laboratory of Tests

A construction of a new testing hall with unique, as regards the world scale, test rigs for testing powered roof supports (Fig. 1) in the eighties of the last century and with the rigs for testing legs and high-pressure hydraulic elements of mining machines (Fig. 2) in the nineties, and also of hydraulics valves (Fig. 3) formed the grounds for a creation of bases, enabling to conduct research projects on extremely important issues of safety in the mining industry [2, 3].

Fig. 1. A view of the rig for testing powered roof support units [3]

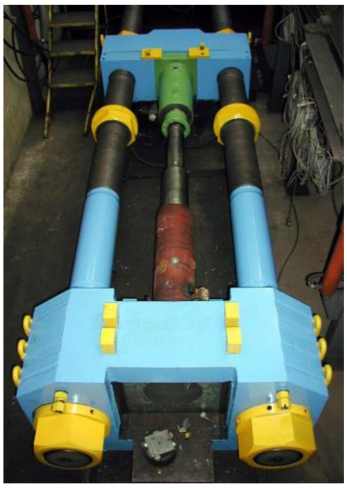

Fig. 2. A view of the rig for testing legs and elements of high pressure hydraulics for mining machines and equipment [3]

The Research Laboratory has a certificate of the Polish Center for Accreditation (PCA), obtained in 1995 in the scope of complex tests of powered roof supports units (Certificate No. AB 039), according to EN 1804-1;2;3 European Standard. At present the Laboratory of Tests realizes as follows:

- rig tests of powered roof supports for certification purposes (new or modernized support units) in the framework of the technical condition assessment (support units in operation),
- tests of power and control hydraulics elements,
- strength tests of products and construction materials.

During over 25 years of accreditation the Laboratory of Tests has performed approximately 3.500 accredited tests of all types of construction products, contributing to an improvement of work safety both in the underground mines as well as in other industries.

At the test rigs of the Laboratory of Tests over 400 types of powered roof support units were tested based on the orders of industrial partners from Poland and abroad. Research-and-development projects, oriented onto methods and means, minimizing the effects of the rock bursts and bumps and

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thus improving safety in a significant way as regards longwall mining systems, were conducted as well. The results of tests, carried out in the Laboratory, are used by domestic and foreign producers and users of powered roof supports as well as of the elements of power and control hydraulics. Besides, in the Laboratory several hundred tests of hydraulic pipelines, friction props, clamps, steel sprags, mesh linings and chock supports were realized.

In the Laboratory of Tests new test rigs are under construction. Testing methods, control systems of testing processes, checking and keeping archives processes are subject to modernization. The testing infrastructure enables to watch the tests on-line. The undertaken modernization activities up-grade significantly the method of conducting tests and thus they increase customers' satisfaction.

2.2.2. The Laboratory of Applied Tests

The Laboratory of Applied Tests was established in 2001 in the result of separating a certain part of tests and measurements, supporting a realization of scientific, applied and industrial projects. Initially its scope of activity covered measurements of mechanical, electrical, hydraulic and pneumatic quantities. Rig tests of dust collecting machines and equipment were conducted on a specialistic rig, located at KOMAG. In course of time the scope of conducted tests encompassed vibracoustic measurements, measurements of dust contents, and tests of power consumption of drive systems of machines used for mining minerals. Obtaining the accreditation of the Polish Center for Accreditation (PCA) by the Laboratory (Certificate No. AB 665), enabled to realize tests connected with certification processes. In this scope the Laboratory closely collaborates with the KOMAG Division of Attestation Tests, Certifying Body. At present the Laboratory of Applied tests offers as follows [4,5]:

- tests of equipment for conformity with the ATEX Directive – Fig. 3,
- measurements of acoustic and vibration parameters,
- measurements of electrical quantities,
- testing the intrinsical safety of circuits,
- measurements of electrostatic, mechanical and geometrical quantities,
- environmental tests and climatic tests,
- tests of IP protection code,
- tests of geometrical structure of surface,
- aging tests of elastomers,
- tests of electric drives,
- tests of hydraulic, pneumatic and mechanical systems,
- measurements of quick-changing processes and response time,
- measurements of light intensity,
- tests of machines and equipment as regards certification,
- tests of electronic devices as regards thermal shock.

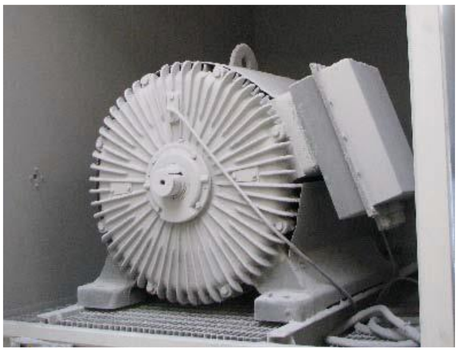

Fig. 3. Tests in the dust chamber for conformity with the ATEX Directive – IP code

The Laboratory extends the scope of offered tests continuously and takes an active part in inter-laboratory comparisons, improving the quality of offered testing services. It has also been taking an active part in a realization of research, development, research-and development and targeted projects.

2.2.3. The Laboratory of Material Engineering and Environment

The Laboratory of Material Engineering and Environment was established in 2006. Using the technical and financial means of the Institute, the documentation was elaborated and specialistic, mechatronic test rigs were constructed. Taking an advantage of the financial means, granted by the Ministry of Science and Higher Education, the Laboratory was equipped with testing-and-measuring apparatus. In February 2008 the Laboratory obtained the accreditation of the PCA - the Polish Center for Accreditation (Certificate No. AB 910) and its scope was extended. At present the Laboratory of Material Engineering and Environment conducts the following tests [6,7]:

- material tests of metal elements of machines and equipment and elements of furniture in the scope of chemical composition and corrosion resistance,
- tests of dangerous chemical substances in materials used in consumers' products (Fig. 4),
- tests of resistance of plastics to light radiation and atmospheric impacts,
- climatic tests of textiles,
- tests of consumers' products,
- tests of global and specific migration of articles having contact with food for conformity with the requirements of the EU Regulation No. 10/2011,
- tests of electric and electronic equipment for conformity with the requirements of the RoHS 3 Directive,
- tests of packaging's for conformity with the requirements of the 94/62/EU Directive,
- tests of surfaces of sports fields and playgrounds for conformity with the requirements of the REACH Regulation and of the standards,
- tests of dangerous chemical substances in materials used in consumers' products,
- safe use of products for children (Fig. 5).

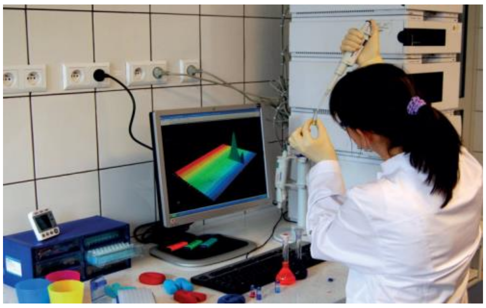

Fig. 4. Test rig for testing chemical compositions of products

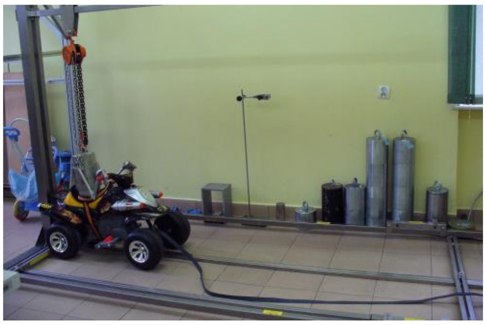

Fig. 5. Test rig for testing a safe use of products for children

Now three accredited testing laboratories, functioning in the structure of the KOMAG Institute, act on the basis of flexible scopes; which allows testing laboratory to make changes in methodology and other parameters which fall within the competence of the laboratory as confirmed by the accreditation body. The experience, gained from the ten-year period of applying such scopes, is a proof that such a system is efficient and useful [8].

3. Scope of the KOMAG research activity

According to the statute of the KOMAG Institute of Mining Technology the scope of the Institute's basic activity includes as follows:

- a realization of research and development projects,
- an adaptation of research and development projects' results to industrial needs,
- an implementation of research and development projects' results to industrial practice.

Development of machines and equipment

Technical achievements in the field of machines include, among others, shearers and monorails (Fig 6). As far as shearers are concerned the KSW-750E. The KSW-880 E and the KSW-800NE (Fig 7) should be mentioned.

Fig. 6. Haulage unit of the GAD-1 suspended monorail [9]

Fig. 7. KSW-800 NE longwall shearer [10]

The research and design projects, concerning innovative solutions of mining machines and equipment, also include:

- tunnelling machines and roadway equipment,
- hydraulic and pneumatic systems for machines,
- pneumatic machines and devices for so called "small mechanization" such as: drill rigs, drill jumbos, bolting devices, loaders, drills, hydraulic hammers,
- devices for transport of materials and personnel,
- hoisting as well as cutting-and-transport machines,
- conveyor, wheeled, stranded, and rail-mounted transport systems with use of different drive technologies, including, among others, belt and scraper conveyors,
- advanced drive and control systems, including diesel drives,
- bunkers and devices for a storage of loose materials, gravitational chutes and spouts,
- pump systems for hydrotransport.

An important part of activity covers research-and-development projects concerning spraying devices used in the processes of cutting and haulage. The spraying devices reduce methane explosion and coal dust explosion hazards, being efficient at a simultaneous reduction of consumption of water indispensable in these processes. Some examples of developed and implemented solutions for a longwall shearer, for a longwall system, for a heading working and for a transfer area in the conveyor system of the run-of-mine haulage are shown in Fig. 8, 9, 10, 11 and 12.

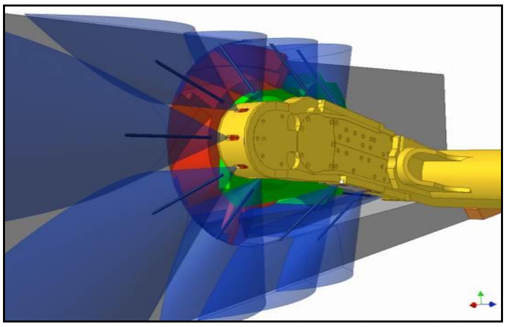

Fig. 8. Computer simulation of operation of the air-and-water spraying installation built in the RW-200N ranging arm of the KSW-460NE shearer [11]

Fig. 9 Air-and-water spraying installation built in the RW-200N ranging arm of the KSW-460NE shearer [11]

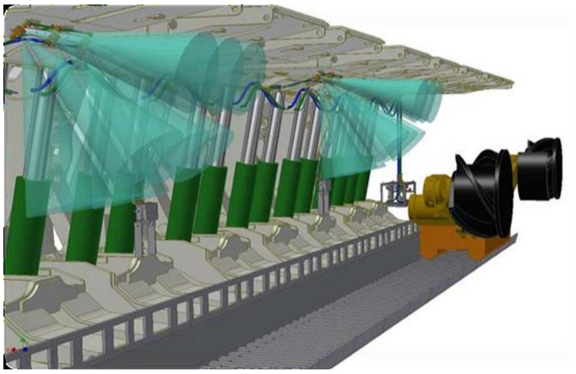

Fig. 10. Visualization of the longwall air-and-water spraying system of KOMAG type [12]

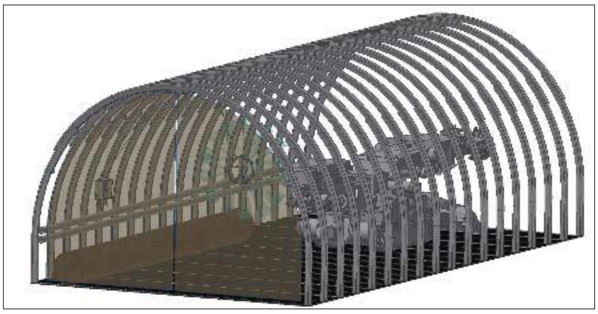

Fig. 11. Visualization of the EMZ-BRYZA roadway dust barrier [12]

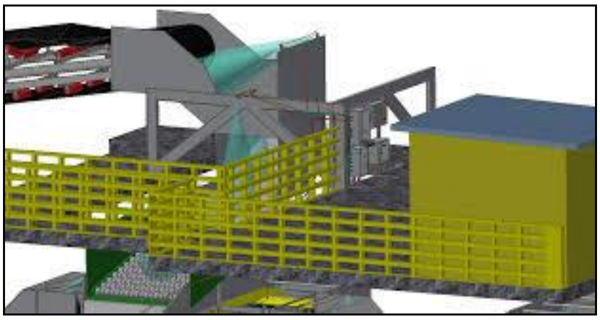

Fig. 12. Visualization of a part of the NEPTUN spraying system – a transfer point in the area of raw coal bunkers [13]

Besides, in this field KOMAG renders services in the scope of technical consultancies, analytical and computational services with use of computer technologies and it elaborates analyses, technical assessments and expert opinions in the scope of selecting machines and equipment for an application in specific mining and geological conditions.

Apart from the innovative design solutions presented above, some information on powered roof supports should be given as well.

KOMAG conducts research-and-development as well as implementation projects in the field of powered roof supports, including an elaboration of new and a modernization of the existing powered roof support units, units of supports for a special application (e.g. supports of crossings: longwall and roadway), individual supports of special designation (e.g. rescue supports), systems of hydraulic supply and control and an installation of pipelines.

Three types of the KOMAG shield supports were implemented for a mechanization of longwall faces of the height from 1.0 to 5.0 meters.

In 1978 the FAZOS Factory started a production of the chock-and-shield supports: FAZOS-12/28/Oz and FAZOS-15/31-Oz elaborated at KOMAG. The GLINIK Factory started a manufacture of the GLINIK-08/22-Oz and GLINIK-05/15-Oz supports. A development of the design and production of powered roof supports caused that in the seventies of the last century over 22,000 of the FAZOS-12/28-Oz powered roof support units were produced.

Two examples of powered roof supports, used at present, are shown in Fig 13 and Fig. 14.

Fig. 13. KHW 12/28-POz/Pp powered roof support unit [14]

Fig. 14. HYDROMEL 16/34-POz powered roof support unit [14]

At present 3D methods of designing and numerical calculations with use of the FEM method are applied in the process of developing designs of powered roof supports.

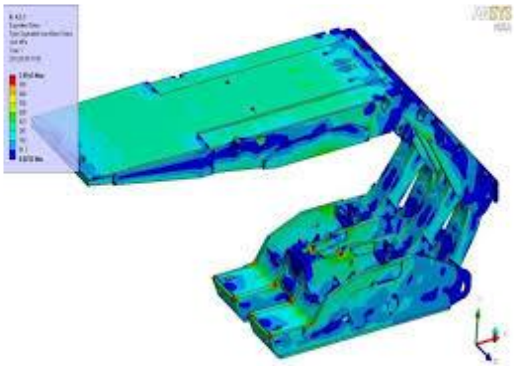

Fig. 15. An example of using the Finite Elements Method (FEM) in the designing process of a powered roof support unit [14]

KOMAG also offers services concerning technical consultations, analytical and computational services, using computer technologies. It also elaborates analyses, technical assessments and expert opinions from the scope of selection, design, manufacture and exploitation of powered roof supports.

Presenting development processes of machines and equipment of KOMAG preparation systems of minerals which plays a significant role in the KOMAG scope of activity, should be mentioned.

KOMAG conducts research-and-development, design and implementation projects in the field of processing minerals (Fig. 16), in particular a development of concepts, projects and documentations of devices such as: pulsating jigs for coal beneficiation and a recovery of coal from coal waste, bucket conveyors, classifiers (Fig. 17), crushers and grinding mills, screens, sizers, automatic samplers and dewatering centrifuges. Over the period from 1955 to 2020 technical documentations of over 320 beneficiation devices (new and modernized ones), including over 200 separators in the pulsating water medium, were elaborated. The jigs operate in Polish and foreign hard coal mines in Brazil, China, India, Romania and Vietnam, and their modified constructions are used in production plants of aggregates. A modification of jigs construction enabled to apply them for processing mine waste deposited on dumps. Due to an improved construction of jigs it is possible to recover simultaneously the coal which is in the waste and full-value aggregates used in building or road-construction technologies.

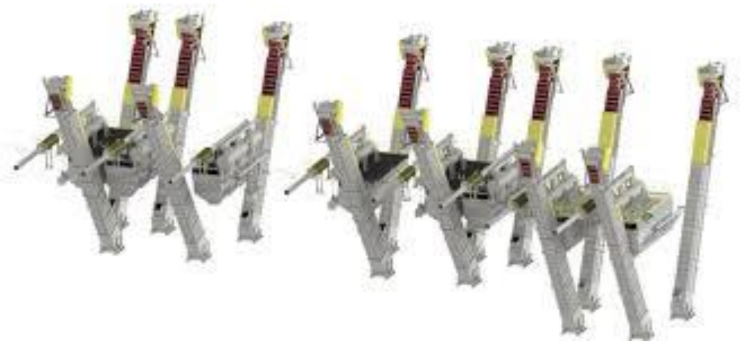

Fig. 16. Beneficiation node at the Budryk Mine after modernization [15]

Fig. 17. Pulsatory classifier of dividing and separating organic and mineral impurities from natural aggregates [16]

KOMAG also offers services in the scope of technical as well as analytical-and-computational consultancies, using the latest computer technologies. It elaborates analyses, technical assessments and expert opinions in the field of processing minerals. It conducts author's supervision of manufacture and start-up of the designed machines and equipment and it also offers testing of preparation processes of minerals in the laboratory, half-technical and industrial scale.

Outstanding technical achievements concern environmental technologies and vibroacoustics. Some examples are shown in Fig 18 and 19.

Fig. 18. Silencer of the air draw and ejector installed at the Jankowice Mine [17]

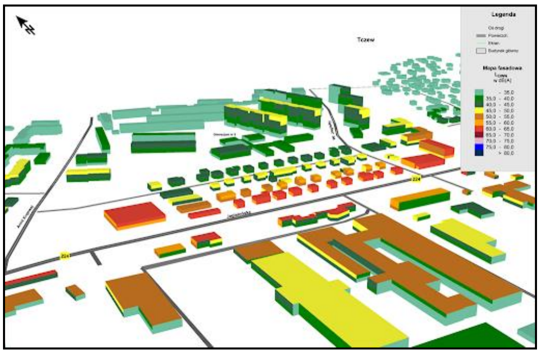

Fig. 19. Acoustic map of one of the roads in the Pomorskie Voivodeship [18]

Models of acoustic field distribution in closed spaces, in the areas of urban agglomerations, transport systems and of industrial plants in the aspect of environmental impacts were widely used. The same concerns acoustic maps and programmes of environmental protection against noise, means of reducing noise and vibrations connected with a modernization of cubature objects, projects of land development calculations of costs and take-off activities.

As it has already been mentioned projects on mechatronic systems are oriented onto automation systems for machines, equipment, technologies and computer applications of testing, communication, internet character and also of sensor system solutions. One of the outstanding achievements concerns a development and an implementation of the GATHER data software, incorporated in the system of electronic identification of elements of powered roof supports (Fig 20). The researchers are able to realize complicated mechatronic projects, taking advantage of their experience in mechanical engineering, automation, IT and electronics. At their disposal they have specialistic test rigs, enabling to construct the designed systems and their complex testing before an industrial implementation (Fig. 21).

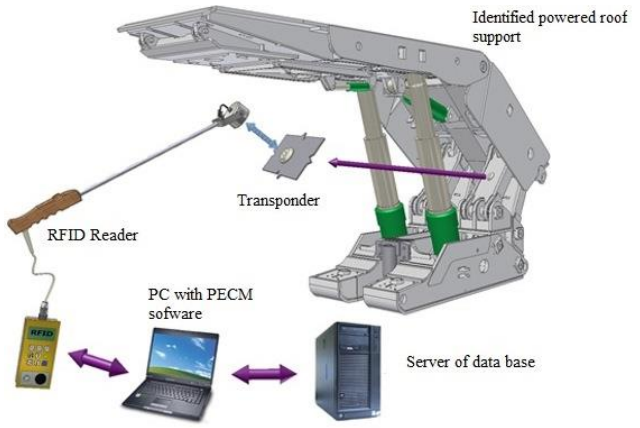

Fig. 20. Electronic identification of elements of powered roof supports [19]

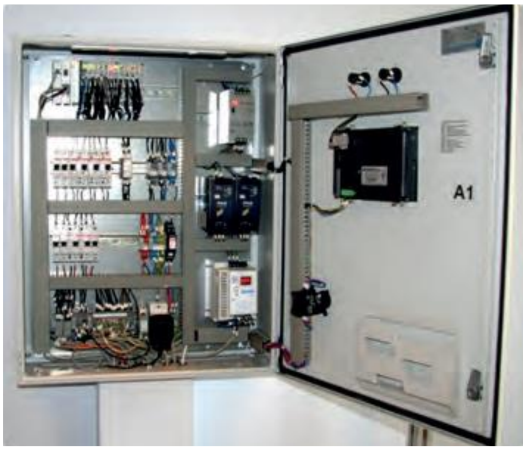

Fig. 21. Control system of the laboratory test rig

4. Accredited Body Certifying Products

The KOMAG scope of activity also covers assessments and certification of products, conducted at the Division of Attestation Tests, Certifying Body which is [20]:

- the accredited body certifying products No. AC 023 which conducts a certification of machines and equipment mainly designed for an application in mining plants in accordance with the scope of accreditation,
- the notified body No. 1456 which conducts conformity assessments with three Directives:
  - 2006/42/WE (testing of WE type),
  - 2014/34/UE (module B: testing of UE type, module D: conformity with the type, based on ensuring the quality of a production process, module F: conformity with the type based on the product verification, module C1: conformity with the type based on internal control of production and testing of products under supervision, module E: conformity with the type based on ensuring the product quality, module G: conformity based on the unit verification),
  - 2009/488/WE (testing of type – module B).
- the body conducting tests and assessments of products subject to approvals for an application in mining plants on the base of Article 113, Item 3 of the Act from 9th June 2011 – Geological and Mining Law – realizes tests and assessments in the scope of:
  - elements of mine shaft hoists, among others, hoisting machines and conveyances, pulleys, suspension gears of hosting ropes, load-carrying suspension gears of hoisting conveyances, shaft signaling and communication devices,

- products used in the workings of underground mining plants, among others, rope transport devices, suspended monorails, floor-mounted railways and their subassemblies; mine cars as well as vehicles with a diesel drive for transportation of personnel; electric machines and equipment and switchgear for the voltage above 1 kV of AC or above 1.5 kV of DC; communication, safety and alarm systems as well as integrated control systems of mining and face systems; conveyor belts.

5. International collaboration

For many years KOMAG has been collaborating, on a large scale, with over fifty foreign scientific-and-research organizations, universities, enterprises and companies from Spain, the Netherlands, Germany, Portugal, Romania, Slovenia, Hungary, United Kingdom, Finland, Latvia, Ukraine, the Czech Republic, Greece, Bulgaria, the Slovak Republic and France. This collaboration has concerned, in particular, the research areas of priority character, supported by the European Commission in the framework programmes such as: health, energy, new materials and IT as well as ecology.

The KOMAG representatives take part in the activities of international research organizations such as the European Association for Coal and Lignite – EURACOAL, the Research Fund for Coal and Steel, the Coal Advisory Group of the European Commission and in the European Standardization activities within the CEN/TC 196/WG-3: Machines for underground mines – Roof support. KOMAG also participates in the activities of the following working groups: Safety of Toys and Equipment for Explosive Atmospheres. The objectives of the undertaken research projects, oriented onto a realization of the basic strategic tasks, aim at an integration with the European Research Area in the scope of designing, testing and manufacturing of machines and equipment as well as an increase of competitiveness of Polish technical solutions on the European market. A high scientific position of the Institute in the European Research Area was officially confirmed, when KOMAG was acknowledged to be the Centre of Excellence in the scope of state-of-the-art and reliable mechanical systems which are operator and environment friendly within the V Framework Programme of the European Commission. International scientific and research projects of innovative character are directly co-financed by the financial sources of the Research Fund for Coal and Steel, the HORIZON 2020, the Erasmus and the Programme HORIZON EUROPE, started this year, which opens new possibilities for a development of research projects, in particular on reducing environmental foot print, improving circularity in extractive and processing value chains, raw material preparation for clean steel production, raw materials for EU strategic autonomy and successful transition to a climate – neutral and circular economy, sustainable and innovative mine of the future, innovative solutions for efficient use and enhanced recovery of mineral by – products from processing of raw materials [21].

Presenting the achievements of the KOMAG Institute in the field of international collaboration, an organization of cyclic scientific-and-technical conferences and workshops, which are one of the tools contributing to a dissemination of specialistic knowledge and a promotion of innovations, should be mentioned. An organization of such undertakings, their subject-matter, coherent with the priority development directions of knowledge in Europe, a participation of best specialists from Poland and from foreign countries have a positive impact on the image of the KOMAG Institute and its perception as a modern scientific-and-research Institute, having an important position in the international research area and playing an important role in the development process of the economy based on knowledge and innovations.

6. Awards and distinctions granted to KOMAG

One of the appreciation forms, confirming the significance and innovativeness of KOMAG results of scientific, research and technical projects, includes awards and distinctions obtained at different exhibitions and competitions in Poland and abroad. A few examples of such outstanding achievements in recent years are as follows:

- Silencer of air draw and ejector.
- BH 300B-HYDKOM 75 Loader.
- KOGASTER intrinsically safe control system.

- Air-and-water pressure instalation.
- PECM system with the GATHER module.
- Auxiliary mine transport arrangement.
- Ventilation system of objects of low acoustic emission.

The above mentioned innovative solutions were awarded at the International Warsaw Exhibition of Innovations -IWIS, the Poznan Fairs, the World Exhibition of Innovations, Brussels Innova, iENA, Concours Lepine, EUROINVENT, INTARG®, the International Fair of Mining, Power Industry and Metallurgy Katowice. They were also awarded by the Minister of Science and Higher Education.

7. Final remarks and conclusions

A scientific development of employees is an important priority objective of the KOMAG Institute. A well-educated staff, having a creative approach to challenges, expressed by the state-of-the-art industry and the requirements of legal regulations concerning the activity of scientific organizations, has a decisive impact on the position and rank of the Institute. The programme of the KOMAG employees' scientific development, which has been active for 20 years, plays an important role in the process of shaping the staff potential.

A system of doctoral, preliminary and "Young Scientist" grants enables young employees to enter the path of a scientific development. Under a supervision and care of the Director's Advisors' Team in which there are professors, representing different branches of science, young employees identify scientific objectives, scope and methodology of their future doctoral theses. At present six KOMAG employees work on doctoral theses of practical application oriented onto solving particular problems experienced by the Institute.

Dynamically changing economic and social conditions force KOMAG to meet more and more sophisticated requirements of interdisciplinary character. The main requirement, which has a big significance for the process of the Institute further development, includes a creation and an industrial implementation of technologies, mechanical and mechatronic systems, automation and robotics oriented on to safer and more efficient gaining of minerals.

Economic conditions determine changes in innovative activities which require an introduction of regular up-dating and extending corrections of the KOMAG strategic objectives. The last document, concerning the Institute development strategy, encompasses the period till the year 2025 [16]. It incorporates the requirements of the Industry 4.0 which are related to a different business model of companies and branches of industry. The KOMAG strategy is oriented onto an application of the state-of-the-art, smart information technologies and automation at each stage of the product manufacture, starting from the phase of design, through tests, production and maintenance as well as recycling. Sensor technologies, Internet of Things and Big Data Sets, Computational Clouds and telecommunication technologies, enabling a rapid data transfer, will be implemented onto practice. An introduction of digital transformation will encompass all the areas of the KOMAG activity.

The strategy of KOMAG activities till the year 2025 is concentrated on the research fields, where the Institute reached the world level, participating in the process of constructing the ecosystem of innovations and of the e-economy. It includes directions of smart specializations in the scope of safety, energy, natural environment and waste management. Nine objectives are determined: Innovativeness, Commercialization, Occupational Safety, Safe Use of Products, Environmental Protection, International Collaboration, Staff Development, Promotion of Innovations - Knowledge Share, Finance, Social Impact of System.

In the scope of creating innovative technologies and technical means, the KOMAG Institute of Mining Technology will continue a close collaboration with Polish and foreign scientific and industrial partners. An essential role will be played by industrial partners. A collaboration with them will be carried out within contracts, scientific-and-industrial networks, technological platforms, centres of advanced technologies.

Over seventy years of the KOMAG history obliges for a realization of the strategic objectives which guarantee a high position of the Institute in the Domestic and European Research Areas and an extension of collaboration with producers and users of machines and equipment in the scope of technical and technological solutions which are safe as well as operator and environment friendly.

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Universal disinfecting installation UMID for decontamination of viruses, bacteria and fungi

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Abstract:

The article discusses the disinfection problem in terms of combating the SARS-CV-2 virus that causes an infectious disease called COVID-19. In the epidemic era the focus was on reviewing the methods and means of disinfecting everyday items that are currently used. The concept and design of a solution for disinfection of outerwear of employees moving along the main communication routes in mining plants, developed by KOMAG in cooperation with Elektron s.c and Budryk mine belonging to JSW S.A., is presented. The structure and principle of operation of the device are presented together with the selected disinfectant based on silver colloid. The first stages of creating a prototype solution are discussed, as well as the effects of implementing the developed universal UMID disinfecting installation for decontamination of viruses, bacteria and fungi. Additionally, a demonstration disinfecting installation built for the needs of KOMAG employees is presented.

Keywords: coronavirus, mist, disinfection

1. Introduction

In 2020, the world faced the pandemic caused by the SARS-CV-2 virus (Fig. 1). To this day, the spreading virus causes an infectious disease called COVID-19, which, although in most cases is mild, can lead to pneumonia or multi-organ failure resulting in death in some groups [1]. A major threat at the beginning of the development of the pandemic was the lack of the so-called "Herd immunity" in society, with the result that there is a high risk of collapse in the functioning of the health service [2].

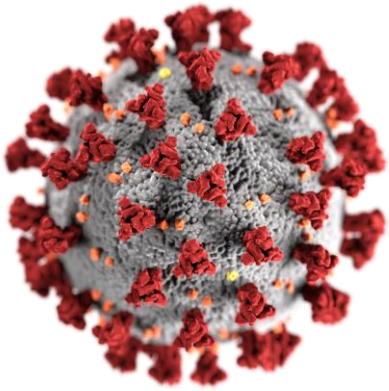

Fig. 1. Visualization of SARS-CoV-2 virus [3]

The COVID-19 epidemic that is present in everyday life has completely reorganized people's lives, both from a medical and social point of view, both at home, in shops and at work, significantly reducing the sense of security. Everyone has got used to the need to maintain social distance, to disinfect hands, and in particular cover the mouth and nose. Unfortunately, a virus that is spread by airborne droplets from coughing or sneezing can settle on a variety of everyday items, including clothes. The virus can survive on such surfaces for up to 9 days, and in extreme cases even 28 days [4]. That is why disinfection of places and things exposed to contact with the virus turns out to be so important [5]. Workplaces, especially hard coal mines, are an excellent source of infections. In the case of hard coal mines, the high incidence of disease is influenced by people due to the nature of the work (going into cages into the underground of the mines and close cooperation of employees). At the turn of spring and summer 2020, some mines had to suspend or limit mining for several weeks due to the scale of coronavirus infections among miners. In total, in five coal companies, more than 10,000 have become infected since the epidemic began. employees [6]. The activities undertaken by companies and mines, consisting in ad hoc disinfection, reducing the number of employees per shift, did not solve the problem. Therefore, KOMAG specialists, having high competences and extensive experience in water atomization, gained within the ROCD European project [7,8,9,10] designed a disinfection and decontamination solution based on spraying a silver colloid using the compressed air.

2. Review of disinfection methods

The most popular disinfection methods include biocides based on alcohol, which are used to wipe potentially contaminated surfaces [11,12]. Disinfecting agents are available in various variants: concentrates, ready-made liquids or wet wipes. Their advantage is low costs, but the disadvantage is the time needed to wipe the surface and incomplete wiping. Ozonation (O_3), which shows disinfecting properties [13] is another method of disinfection as ozone is rapidly decomposed into diatomic oxygen and atomic oxygen (Fig. 2). It is one of the strongest oxidants as it destroys both viruses, fungi, bacteria and mites. The disadvantage is that it is harmful to living organisms and cannot be used in the presence of humans.

Fig. 2. Visualization of ozone particles [14]

The method of irradiation with UV-C rays is also known and used to control viruses. UV-C radiation is one of the three types of waves emitted by the Sun [15]. This type of waves does not reach living organisms on the Earth, thanks to the ozone layer, but we know that they are suitable for rooms disinfection. They are produced by special lamps that can be used in the absence of people because they are harmful to the skin and eyes, while destroying bacteria and viruses and their spores. The advantage is fast action and the disadvantage of lack of penetration, making them work, where the microorganisms are on external surfaces (Fig. 3).

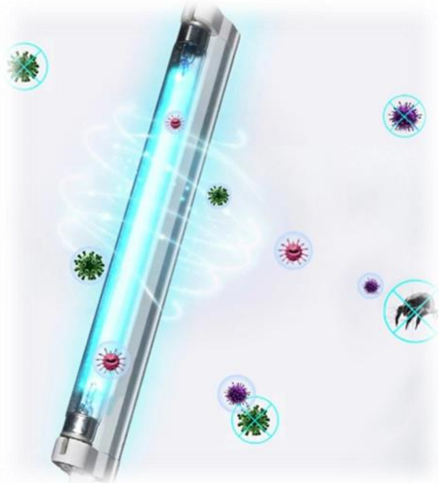

Fig. 3. Visualization of a germicidal lamp action using UV-C light [16]

The presented methods of disinfection have many advantages, but also have some disadvantages [17]. Unfortunately, none of the solutions presented relates to the problem of disinfection of clothing of the people entering workplaces, in particular coal mines. In coal mines, where several to several dozen people go underground together, the virus can infect a significant part of the employees, disrupting the work of the entire mine. That is why KOMAG decided to develop such a disinfection method that enables efficient and safe disinfection of clothes of employees moving in communication routes of workplaces.

3. Concept and design of the solution

KOMAG, just a dozen or so days after the announcement of the epidemic in Poland, developed and built a prototype of an innovative solution for disinfecting viruses, intended to be used in hard coal mines. Several hours passed from the moment of taking the decision to develop such a solution, to the creation of the concept and the test model based on disinfectant mist sprayed by compressed air. The test model was built from the available components used to control dust in the hard coal mining industry, i.e. two-media nozzles, a disinfectant tank and a valve reducing the compressed air pressure (Fig. 4).

Fig. 4. Operational tests of the developed prototype [18]

Spraying system, which delivers and atomises a disinfectant was the main component of the developed solution. The structure of the device is shown in Fig. 5.

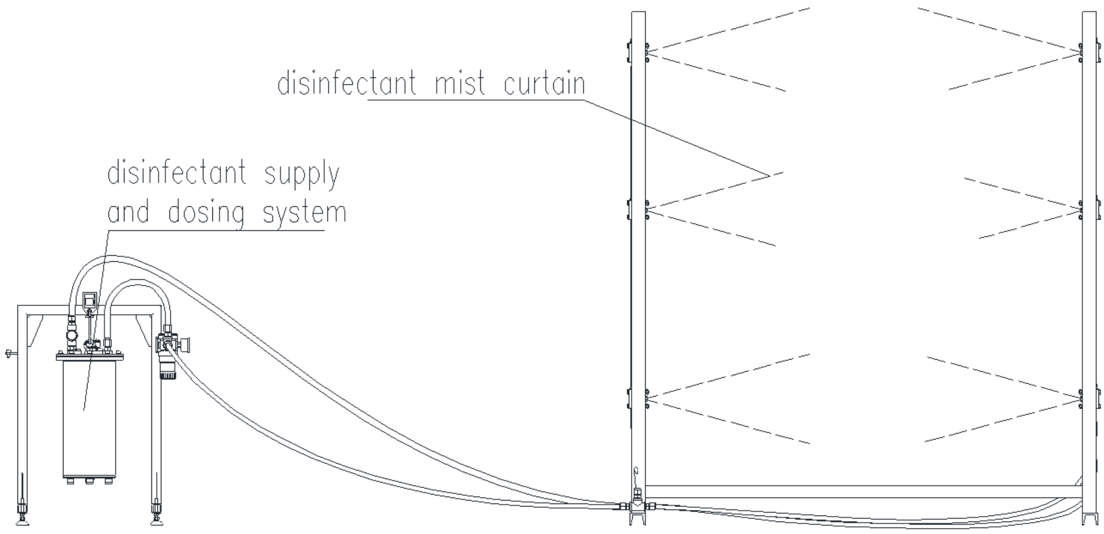

Fig. 5. General view of the installation [18]

In the solution, compressed air is used to pressurize the tank containing the disinfectant and transport it to the spraying nozzles. It is also a carrier of a disinfectant after mixing inside the spray nozzle. The disinfectant comes out of the tank through a nozzle equipped with a tube reaching the bottom. The basic components of the supply and dosing unit are shown in Fig. 6.

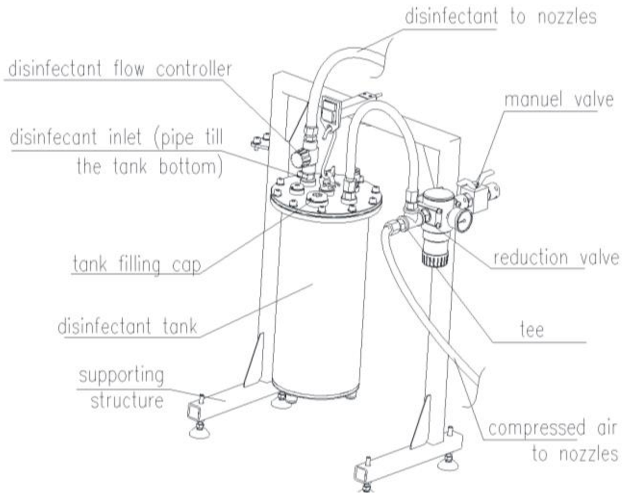

Fig. 6. The system for supplying and dosing the disinfectant [18]

When people pass through the mist curtain, a disinfectant is sprayed on them. Walking takes place in an upright position, arms extended along the body to disinfect as much of the outerwear surface as possible (as shown in Fig. 7)

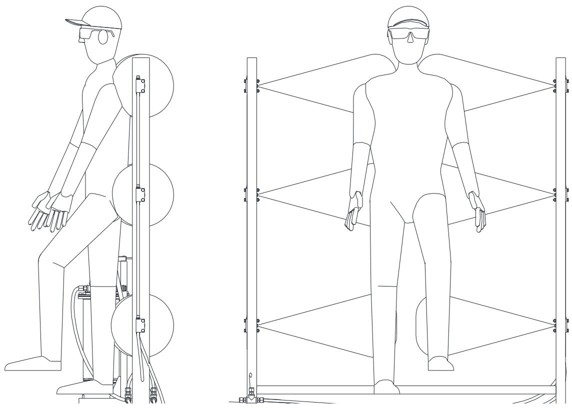

Fig. 7. Method for passing through the disinfectant mist curtain [18]

The model constructed in this way was used to implement a prototype of the UMID solution in Budryk mine of Jastrzębska Spółka Węglowa (Fig. 8). As a result of the implementation, the installation was automated by using the motion sensors (detecting the entry of a person into the decontamination zone) and the nozzles with a flat spray jet to increase the area of sprayed drops.

Fig. 8. A prototype of the UMID universal disinfecting installation in Budryk mine, made by Elektron s.c. [18]

The applied improvements and positive opinions about the prototype allowed to start work on the documentation of the serial installation (Fig. 9). As a result, the installation was developed with a spraying system that atomises a disinfectant with air-water mist nozzles as the main component. The disinfecting installation requires the supply of a working medium in the form of compressed air with a pressure in the range of 0.3-0.6 MPa. Compressed air is used to pressurize the disinfectant and transport it to the spray nozzles. Compressed air is also responsible for atomization of the disinfectant in the spray nozzles. The standard solution is also equipped with a disinfectant flow controller. The installation is equipped with an automatic control activated by a photocell. The electric control unit i.e. distribution box on which the following components are installed: power switch, signal lamps (green - power, orange - solenoid valve on).

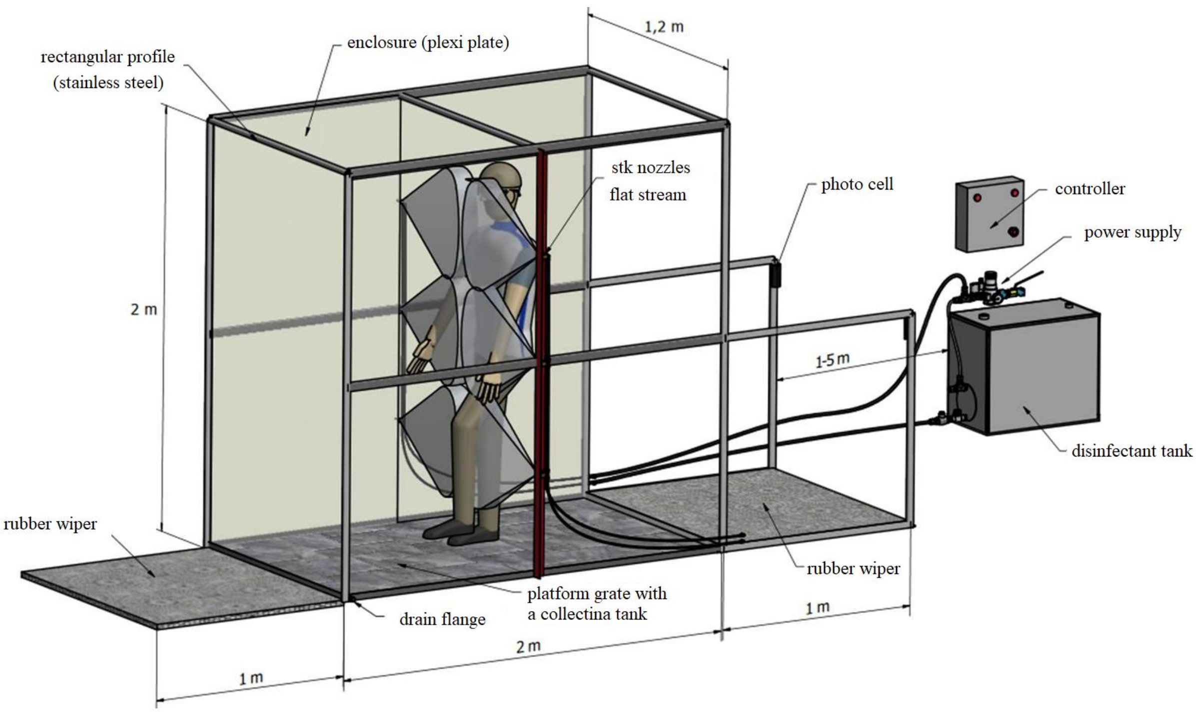

Fig. 9. Universal UMID mist disinfection installation - the serial solution [19]

The solution was used in several hard coal mines in Poland (Budryk mine, Borynia mine, Zofiówka mine) and in the Specialist Services Center of the Central Mining Rescue Station Cen-Rat Sp. z o. o. (Fig. 10). The installed devices use a disinfectant based on metallic silver particles stabilized with polyol, the numerous scientific studies confirmed its antiviral effectiveness. Nano-silver has an affinity for proteins found on the surface of viral capsids and causes their inactivation or physical encapsulation. As a result, silver prevents the ingress of viruses, which inactivates them permanently [20].

Fig. 10. Universal UMID mist disinfection installation **-** the serial solution used in coal mines [19]

Technical characteristics of UMID disinfecting installation:

A) **Operational environment:**

Ambient temperature:	5÷40°C,
Relative humidity:	up to 80% in temp. +20°C,
Supplying hoses:	elastic,

B) **Supply parameters of nozzles**

Compressed air pressure:	p_{min} = 0.4 MPa,
Compressed air working pressure:	p = 0.4÷0.6 MPa,
Air flowrate (in the entire system):	Q_p =600 dm^3/min,
Disinfectant flowrate (in the entire installation)	Q_s =0.6÷1.0 dm^3/min,
Type of nozzles:	STK-ZZ-P
Number of nozzles in the chamber:	6

C) **Technical data of the control unit**

Supply voltage:	230 V; 50 Hz
Rated current:	2 A

4. Development of UMID installation

The experience gained and the effects of implementing the UMID installation in hard coal mines prompted the KOMAG management to build a demo version of the installation for own applications to neutralize and eliminate viruses and fungi on the outerwear of employees entering the institute. The developed demonstration UMID mist disinfecting installation was equipped with an installation for spraying a nano-silver-based disinfectant using the air-water nozzles [21]. The nozzles spraying the disinfectant are placed in the side walls of the disinfecting chamber, made of profiles for high storage, where it has an open entrance and exit. The disinfecting installation requires supply of a working medium in the form of compressed air with a pressure in the range of 0.4-0.6 MPa. Compressed air is used to pressurize the tank with the disinfectant and push it into the lines feeding the two-way nozzles. In addition, compressed air atomises the disinfectant supplied, already in the spraying nozzles. The amount of disinfectant is limited to the volume of the tank, amounting to a dozen or so litters, and the flow controller used at the outlet of the tank allows adjusting the consumption depending on the needs of the installation. The demonstration device is turned on and off by a twilight and movement sensor which, after detecting movement in its field of operation, activates a solenoid valve located on the compressed air supply pipe, which in turn activates a controlled check valve, allowing the disinfectant to flow to the spraying nozzles. The optical sensor is installed in the side of the chamber, on its outer wall, in a special cover. This made it possible to limit the uncontrolled activation of the sensor, activating the sensor only when a person or a hand was present, just in front of the cover opening (Fig. 11).

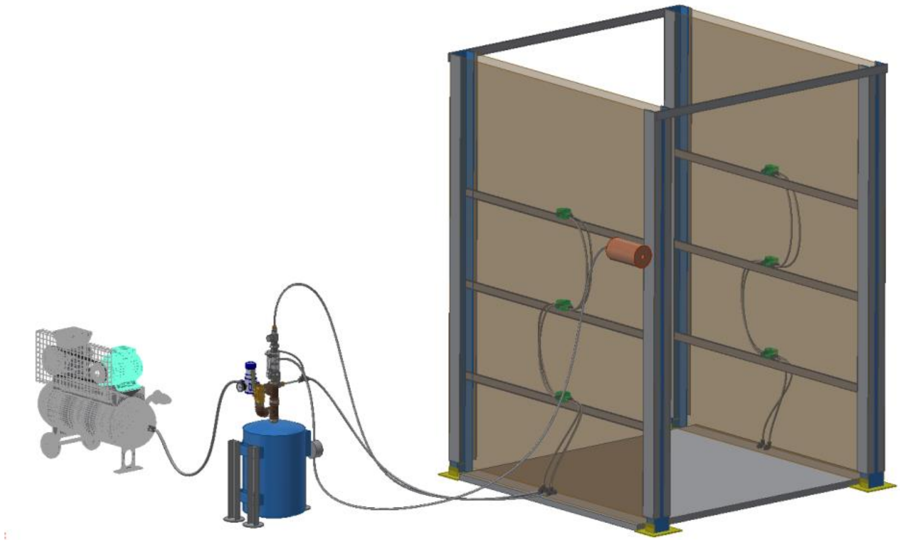

Fig. 11. 3D model UMID demonstrative mist disinfecting installation [21]

The demonstration UMID installation was placed in front of the main entrance to KOMAG, where everyone entering the main building can be disinfected (Fig. 12). The disinfectant used in the installation enabled people passing through without the need to wear protective gloves and glasses. The solution was used in positive temperatures and did not significantly affect the discomfort of passing people.

Fig. 12. Demonstration mist disinfection installation,
when a person enters the main KOMAG building [21]

5. Conclusions

The growing threat of the spreading SARS-CV-2 virus in Poland and in the world, causing an infectious disease called COVID-19, has forced the search for new solutions and ways to reduce it. Such a solution was developed by ITG KOMAG just a few days after the epidemic was announced in the country. The proposed disinfection by fogging, consisting in the production of drops of a disinfectant with a diameter of several to several dozen micrometres, allows reaching almost every fragment of the disinfected surface, inaccessible to other solutions. The universal mist disinfecting installation developed by ITG KOMAG is designed to neutralize bacteria, viruses and fungi on the outer clothing of workers moving on communication routes in coal mines and other buildings. The device works by spraying drops of a disinfectant by compressed air, without need for additional spray pumps. The solution works in automatic mode, when a person is passing near the motion sensor, which starts the disinfectant discharge from the spraying nozzles in the disinfecting chamber. The disinfectant used in the device, based on metallic silver particles stabilized with polyol, allows it to be used in the presence of people, and its effectiveness against viruses has been confirmed in numerous scientific studies. Additionally, coating the surface with a disinfectant based on nano-silver creates a microscopic nano-layer on the surface, protecting against deposition and penetration of microorganisms, reducing the accumulation of pathogens. The installation developed at KOMAG was patented in the Patent Office of the Republic of Poland and noticed by the jury during the Invention and Innovation Fair INTARG 2020, where it won the gold medal and the World Invention Intellectual Property Associations WIIPA award [22]. In addition, the solution was awarded in the Innovator of Silesia competition organized by the Upper Silesian Accelerator of Market Entrepreneurship [23].

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Concept of the facility for testing the wear of chain links in the aspect of synergism of environmental factors

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Abstract:

The article presents the concept of a test stand for laboratory comparative tests of the multifactorial wear of chain links used in scraper conveyors, the aim of which will be to improve the durability of chains by using more favourable materials from which the chain links are made. The tests will focus on the zones of links cooperation under load. The concept was preceded by a theoretical introduction illustrating the nature of the wear of chain links and the synergy of wear processes. 3D model of the test stand and proposals for technical solutions of each component were discussed. The test stand will enable testing the environmental factors increasing the intensity of the wear process in several arrangements. These include, among others mineral abrasive, mine water and dynamic forces acting on the chain. The method of verifying the concept of the test stand using the simplified prototype made in the 3D printing technology was also presented.

Keywords: increase of durability, link chain, wear

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Introduction

Cooperating components of machines are subjected to wear and tear. There are many external factors causing and/or increasing the wear degree of the machine nodes. There are the following most common forms of damage:

- separation fracture (decohesion),
- corrosion,
- plastic deformations,
- tribological wear [1,2].

All the above-mentioned forms of damage can be observed in the subassemblies of scraper conveyors operating in hard coal mines - this also applies to link chains. However, analysis of the state of the art shown that tribological wear is to a significant extent responsible for the total damage to the components of scraper conveyors. This wear often intensifies or accelerates all the above-mentioned forms of damage [1, 3-8].

Material and Methods

2.1. Basic forms of tribological wear

In the literature, tribological wear is defined as a type of wear caused by frictional processes. There is a change in the structure and physical properties of the outer contact layers, and thus a loss in the volume and weight during the processes related to tribological wear. The wear intensity can be defined as the total result of different types of destructive interactions. Tribological wear can be defined by several quantities. The most popular are the parameters defining the loss in volume, weight or some dimensions characteristic for a given wear (e.g. in the case of link chains it may be a pitch) [1, 2].

The main process causing destruction of cooperating components is a criterion for classification of tribological wear (Fig. 1). Here we can distinguish the following [1, 2]:

- adhesive wear caused by degradation of adhesive joints,
- abrasive wear caused by micro-cutting and micro-grooving processes through the interaction of the surface with the micro-roughness present,
- fatigue chipping of surface layers of cooperating components, caused by cyclic variable load or abrasive movement,
- corrosive wear.

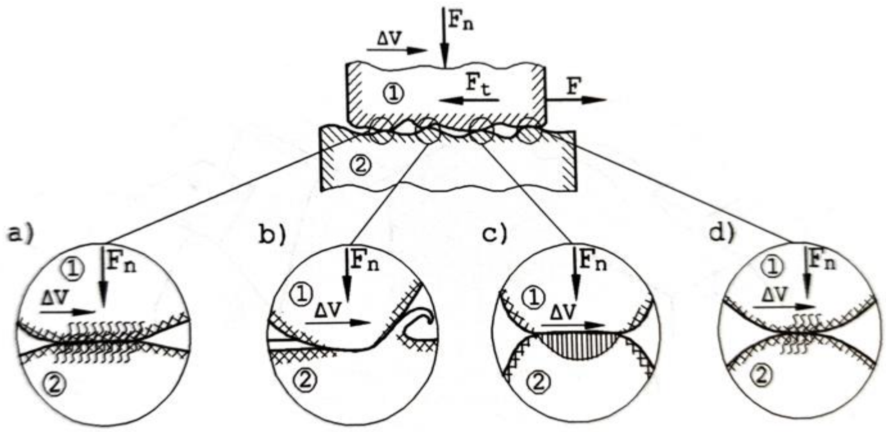

Fig. 1. Tribological wear mechanisms (a-adhesive, b-abrasive, c-fatigue, d-corrosion) [1]

Abrasive wear is the basic mechanism of damage in the tribological system of two cooperating chain links. In general, it consists in the separation of material particles of the abrasive surfaces as a result of micro-abrasion and micro-grooving processes. In the absence of additional abrasive material between cooperating surfaces, damage is caused by micro-roughness of the cooperating surfaces themselves. When there is additional abrasive between the cooperating surfaces, the degradation process is significantly intensified [1].

Dynamic effect is an another factor that intensifies the abrasive wear. In the literature, abrasive-dynamic wear is sometimes classified as a sub-category of impact wear, defined as a collision of solid bodies. Abrasive-dynamic wear is related to the presence of two following mechanisms:

- hitting the abrasive particles on the worn surface and further sliding movement of these particles on the worn surface,
- sliding against each other of two worn surfaces in the presence of abrasive particles with the interaction of external forces generating impact [1, 9].

Frequent starts, uneven load, hits of the transported material on the conveyor, constantly generate dynamic forces that can cause impact on the abrasive between the surfaces of the chain link.

Tribo-chemical wear is another wear mechanism mentioned in the literature, defined as the formation of surface reaction layers in a result of tribo-chemical reactions. These surfaces, in a result of the abrasive mechanisms, are removed, opening the newly-reacting surface with environmental components (e.g. mine water). This reactivity is explained by the plastic deformation of the friction surface, which generates local changes in density, causing the migration of electrons from the places with shear stresses to places with tensile stresses (where the potential is negative). Temperature differences resulting from warming up of friction surface is another catalyst for tribo-chemical reactions. Then, electrons flow from regions with higher temperature to the places with lower temperature. Within the tribo-chemical wear, a distinction is made between oxidative and corrosive wear. The first one consists in the cyclic removal and recreation of the oxide layer on the surfaces of the cooperating components, in the atmosphere of dry air. It is classified as a normal type of wear, and its intensity is insignificant under stabilized conditions. In turn, corrosive wear is the removal of the brittle oxide layer in a cyclical manner during the abrasion of the cooperating surfaces. This process is significantly intensified by the presence of wear products, which take the role of hard abrasive grains. In the literature, the concept of tribo-corrosive wear is used to describe the forms of combined wear intensified by the synergistic effect of abrasive and corrosive factors [1, 9-15]. The mechanism of tribo-corrosive wear is schematically presented in Fig. 2.

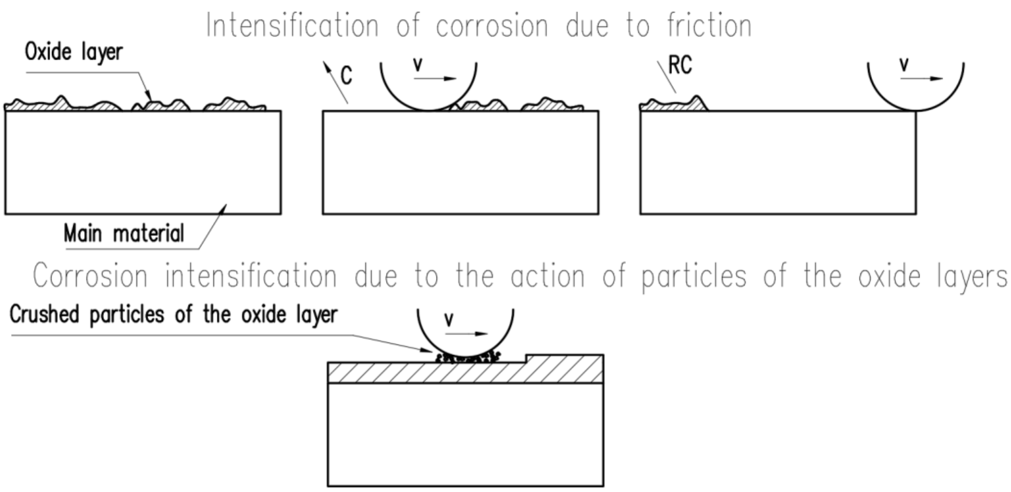

Fig. 2. Mechanism of tribo-corrosive wear (velocity vector, K-corrosion; R-repassivation) [11]

More broadly, tribo-corrosion is sometimes treated as the interaction of corrosion with various types of mechanical wear. On this basis, the literature defines the so-called tribo-corrosion system understood as a kind of tribological system, taking into account, along with mechanical inputs, material and environmental properties, also electrochemical inputs. This type of system is best described by the relationships influencing the wear of the chains in the area of contact of mine scraper conveyors components [1, 16-18].

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2.2. Wear and degradation of chain links

Cooperation of chain links with the drum and the conveyor route, cooperation between the links as well as difficult and complex environmental conditions in hard coal mining plants determine complexity of the wear processes of link chains in terms of friction, corrosion and fatigue. The most important factors increasing the degradation of links and components of scraper conveyors are the following:

– presence of rock and coal dust in the zone of cooperation of chain links and during their movement in the segments of the conveyor route
– corrosive effects of mine water coming from spraying systems and flowing out of the goaf (causing the so-called Rebinder effect),
– corrosive effect of aerosols of mine air and mine water,
– dynamic loads from start-ups of conveyor drives, their uneven loading, frequent reloading and blocking.

The above-mentioned degradation factors of chain links, most often lead to their breaking after a given operating period. Despite the production of chains from high-quality steel according to specialized technologies ensuring high durability and strength requirements, chain breaks are the most common failures of scraper conveyors, which, first of all, pose a great threat to working miners and generate significant economic costs related to breakdowns. This is especially disadvantageous when considering the serial mining system in coal mining, where the stopping of the haulage machine on the face or longwall heading usually stops the entire mining process. What's more, the chains break usually without any previous symptoms, that is why taking the preventive actions is not impossible posing a risk.

Results and discussion

3.1. Technical assumptions

Analysis of the possibility for designing a test stand, intended for laboratory comparative tests of chain wear (within the link contact zone), with particular attention to impact of environment in underground mining plants was the basic assumption of the conceptual work. Development of the stand design to a degree that would enable its construction, and then testing it was the authors objective. The following assumptions were made for the test stand:

– the stand should reflect the cooperation of two chain links, one fixed and the other moving in the range of the angle of turning one against the other by about 30˚,
– drive component of the stand based on an electric motor with a power of approx. 1.1-1.5 kW and a rotational speed of approx. 1420 rpm,
– load to the fixed link with an axial force of about 1000 N,
– 14x50 link chain will be tested.

3.2. Concept of the test stand

The first step in the conceptual work was to determine the kinematics of the test stand based on the adopted technical assumptions. Accordingly, the diagram as in Fig. 3 was adopted.

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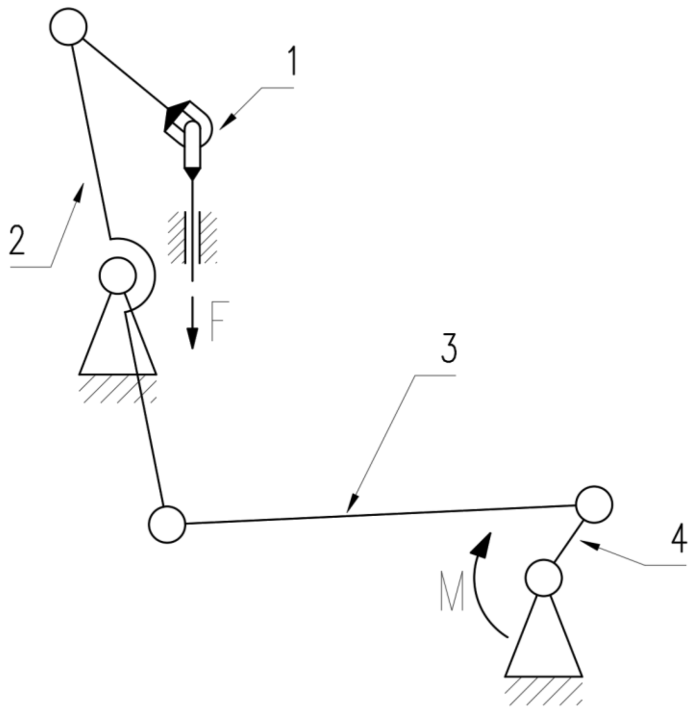

Fig. 3. Kinematic diagram of the conceptual test stand
(1-cooperating links, 2-rocker arm, 3-linkage, 4-crank, F-loading force, M-torque)

According to the above diagram (Fig. 3), the torque will be transmitted directly to the crank system. The crank will be connected by a link to the articulated rocker arm. Movement of the crank will allow the rocker to rotate around the link to a limited extent, corresponding to the required range of rotation in the chain link. The rocker arm on the other side will be connected via articulated arm, on which the half of the link that will move in the link will be fixed. The second half of the link will be suspended on the movable "half-link", loaded with the force F in the direction coinciding with the longitudinal axis of the fixed link. The load will be fixed on a guide which will be free to move in the direction of the force.

After the analysis of the kinematic scheme, the components of the drive assembly were selected. Three-phase electric motor of a body size IEC 90, foot-mounted (B3) and with the possibility of flange-mounting (B14) type 90LP/4 TF 180E by Nord was selected. As standard, the motor is equipped with an inverter (type SK 180E), which enables setting the rotational speed of its shaft. The speed control does not require an additional control system and will be controlled by a knob already mounted on the inverter [19]. In addition to the motor, a Kacperek HM-202/90B14 worm gear was selected [20]. With the assumed gear ratio i=11.42, the test stand will be able to make maximum 123 friction cycles (back and forth) on a one sample (177,120 cycles per 24 hours).

The conceptual model of the test stand is shown in Fig. 4.

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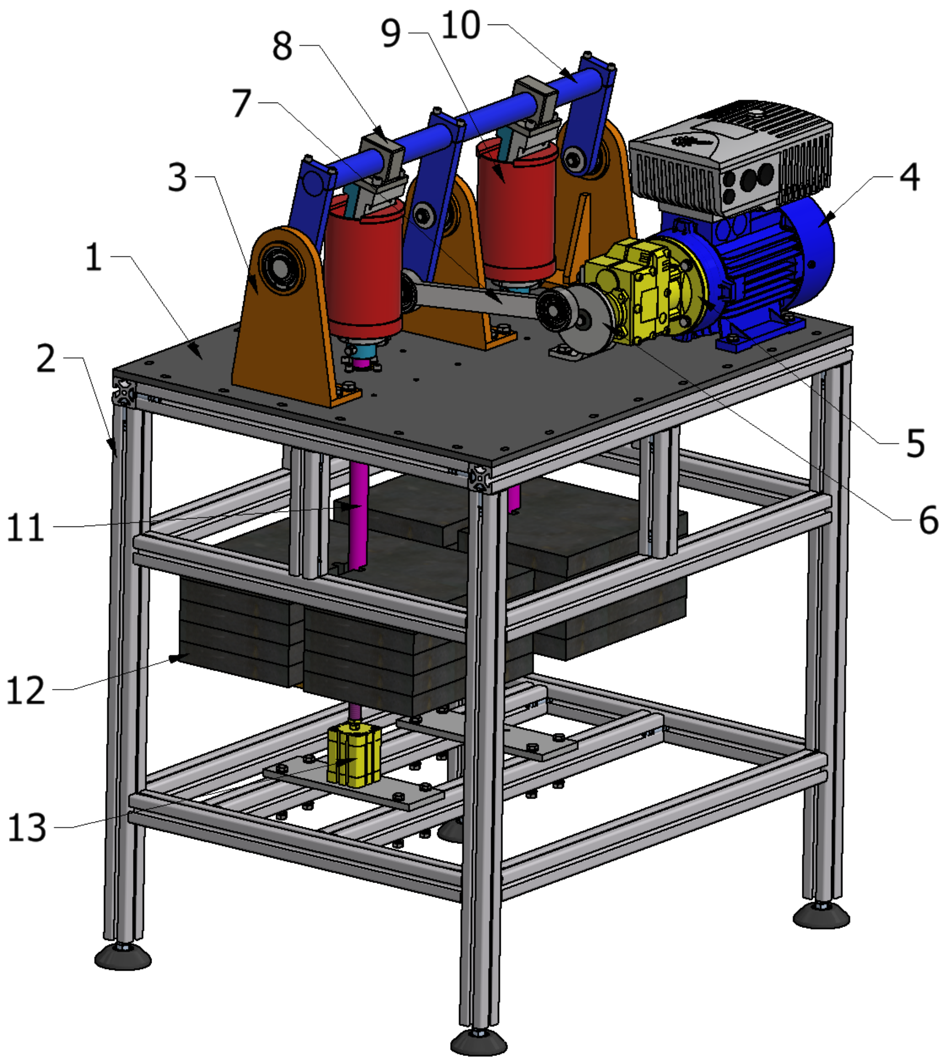

Fig. 4. Model of the test stand - general view (1- table top, 2-table frame, 3- stands, 4- electric motor with inverter, 5- reducer, 6-eccentricity (crank), 7-connector, 8-upper link holder, 9 - container for abrasive, 10-rocker arm, 11-load bar, 12-weights, 13-pneumatic cylinder)

The assembly is driven by a three-phase electric motor. The selected motor has a body that contains both the legs and the connection flange. This allows the motor to be screwed to the table through the legs and to fix on it the reduction gear flange. On the input side, the gearbox is equipped with a hub with a key (for connection with the motor), and on the output side with a shaft, also equipped with a key. An eccentricity (crank), made as a pipe (with a groove) with a disc containing an eccentric opening, will be mounted directly on the shaft. The eccentric sleeve acts as a crank mechanism, converting the rotary motion into a reciprocating motion.

A pin assembled with the connector will be installed in the eccentric opening. The connector is the component that transmits the reciprocating movement to the rocker arm, which in turn converts it back into a rotary movement around the link, but limited to about 30°. A steel rod, ending on both sides with bearing hubs will be the connector– on one side for connection with the drive, and on the other side with the middle arm of the three-armed rocker arm, installed on three legs to make swinging possible. So-called movable links will be attached to the rocker arm, cooperating with fixed links, which are loaded by a weight. Both movable and fixed links are located in containers to which abrasive is dosed.

The containers cover precisely the abrasive and possibly the water added with it, due to the need to limit the amount of abrasive needed for testing, and also the need to reduce dust, which is particularly dangerous for the operators working in the presence of carbon abrasive. A cross-section through the abrasive tank is shown in Fig. 5.

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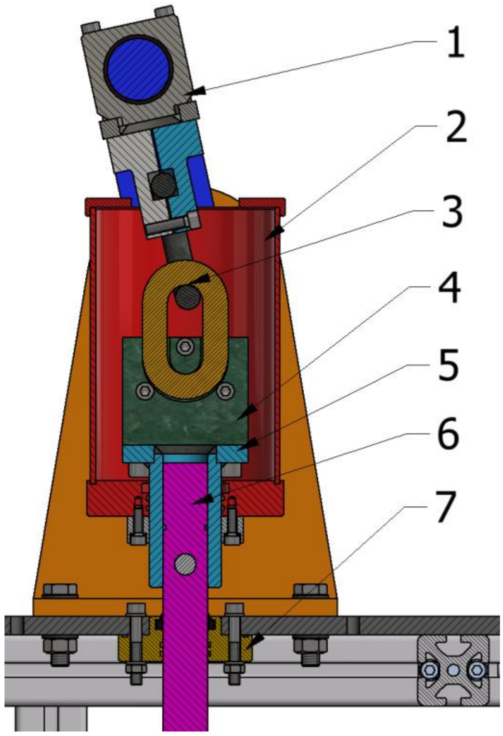

Fig. 5. Cross-section of the tank for abrasive
(1- grip of upper link, 2 – tank for abrasive, 3 – cooperation point between links,
4- grip of bottom link, 5- sealing sleeve, 6 – loading rod, 7 – guiding sleeve)

The load that simulates the longitudinal forces in the chain will be installed on a bar attached to the bottom link. The bar will require displacements restriction in directions perpendicular to its axis. For this purpose, a guide screwed from the bottom of the table will be introduced (Fig. 5), allowing only the axial displacement of the bar. At its end, a set of weights (Fig. 4, item 12) will be installed, simulating the tensile force occurring nominally in the chain. Weights will be made of rectangular pieces of metal sheets, and their number will depend on the assumed load. At the bottom of the table, it is possible to install auxiliary devices, e.g. a relief jack, as well as a unit responsible for generating dynamic loads. It was initially assumed that this role would be played by a pneumatic actuator axially connected to a loading bar.

Initial nominal load resulting from the suspended weights was determined on the basis of FEM analyses, which showed the stresses ~ 40 MPa in the contact area at a load of 1000 N. It seems to be enough to emphasize the processes of micro-cutting and micro-grooving, however, this value will be finally verified empirically on the constructed test stand. In the case of selecting and determining the operating parameters of the pneumatic actuator, dynamic tests of the scraper conveyor will be carried out to determine the simplified characteristics of dynamic excitations. This characteristic will be implemented into the control system of the test stand.

The test stand is relatively small (Fig. 6). The height of the table top, approximately 860 mm from the floor, creates favourable ergonomic conditions for testing.

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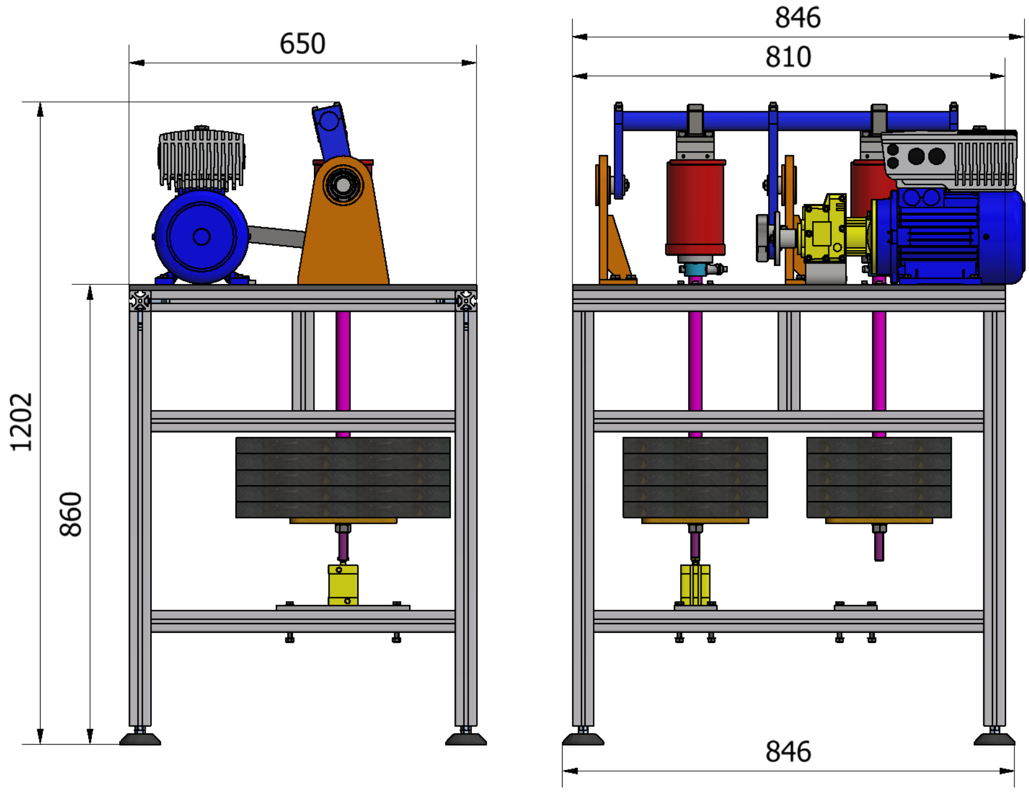

Fig. 6. Test stand dimensions

The test stand will be used for testing the wear of chain joints, taking into account the synergy of combination of environmental factors. The selected technical solutions will enable testing in the following configurations:

- chain wear tests without the presence of factors increasing corrosion and abrasive wear,
- chain wear tests in the presence of abrasive,
- chain wear tests in the presence of abrasive and demineralized water,
- chain wear tests in the presence of abrasive and saline water,
- chain wear tests with dynamic forces without the presence of abrasive,
- chain wear tests in the presence of abrasive with dynamic forces,
- chain wear tests in the presence of abrasive and demineralized water with dynamic forces,
- chain wear tests in the presence of abrasive and saline water with dynamic forces.

Mine links of size 14x50 will be tested due to the expected faster wear of the abrasive nodes (compared to larger sizes). Moreover, it is possible to test the samples made of potentially more wear-resistant materials, the introduction of which into the chains production could improve their durability. They are intended to be unconventional (e.g. nanocrystalline) steels or conventional steels subjected to novel processing methods. Such materials would be used to make test specimens in the form of rollers with an internal rounding corresponding to the radius of cooperation of the two joints. The shafts will be mounted in a special holder enabling, on the one hand, their replacement, and, on the other hand, assembly in the cell holders without making any structural adjustments. The structure and mounting of the sample in the holder are shown in Fig. 7.

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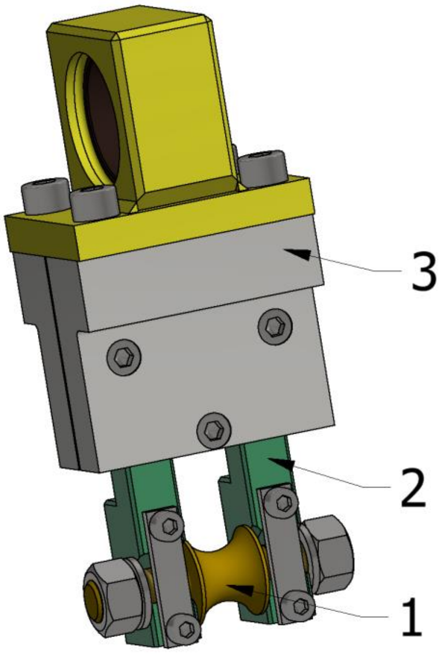

Fig. 7. Concept of assembly of a special sample
(1-universal sample, 2-sample holder, 3-upper link holder

3.3. Verification of design assumptions

Verification of design assumptions in form of a mechanism model and production of its components using the incremental 3D printing method was the final stage of the conceptual work. Components of the stand were fixed on the OSB board with the standard connecting components (also used to adjust the clearances between the components). A small 12VDC geared motor was the driving component of the stand mechanism. Rotational speed control was based on the supply voltage change using a simple converter with a potentiometer. The model and the first prototype are shown in Fig. 8 and 9.

Tests of the model made in 3D printing technology proved correctness of technical and construction assumptions.

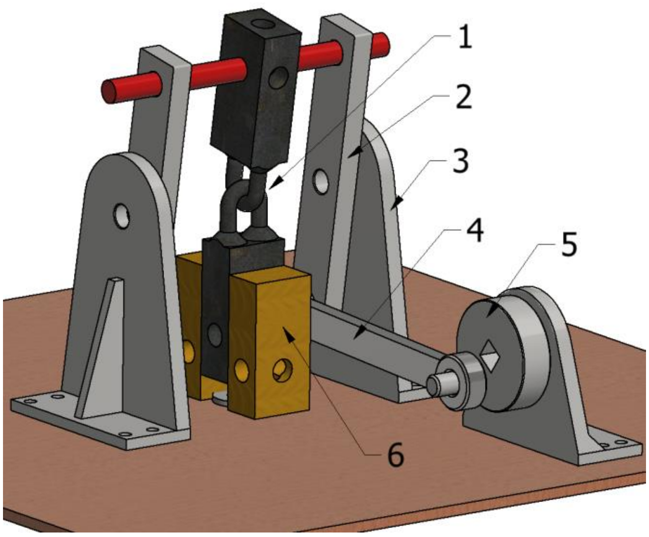

Fig. 8. Model of the stand for 3D printing
(1-cooperating links, 2-rocker arm, 3-stand, 4-connector, 5-eccentricity (crank), 6-guides)

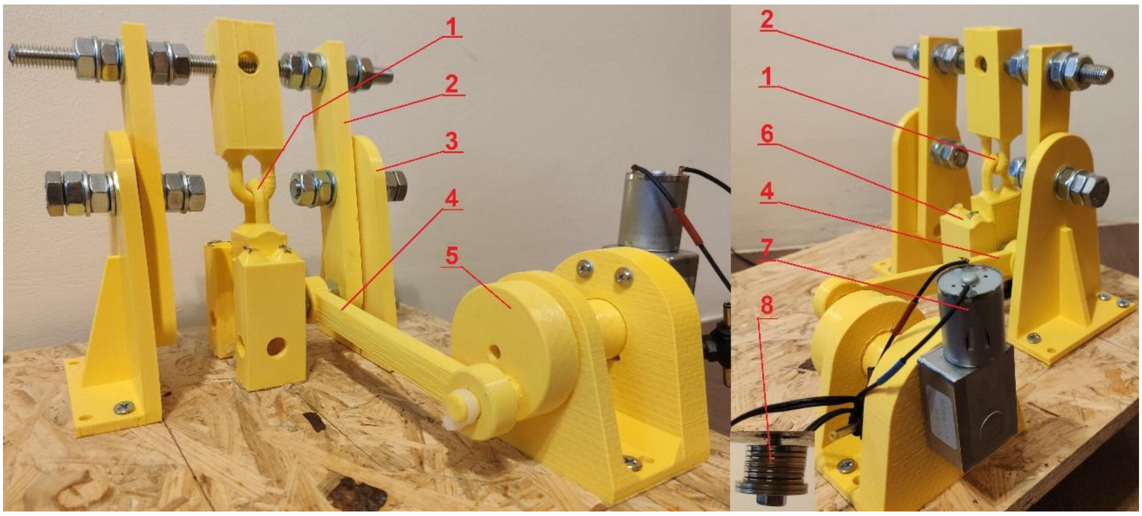

Fig. 9. Mechanism of the stand (3D printout)
(1-cooperating links, 2-rocker, 3-stand, 4-connector, 5-eccentricity (crank), 6-guides, 7-DC gear motor, 8-load suspended from the bottom)

4. Conclusions

The described test stand allow to tests and identify multi-factor wear providing significant knowledge on wear of materials used in link chains, mainly in scraper conveyors. This knowledge is necessary for the design and engineering staff in the development of innovative and state-of-the-art solutions for machines equipped with chains of increased service life. This, in turn, will allow for development of the machine design solutions optimized for the actual environmental conditions and dedicated to various industries.

The analysis and the designing work showed that the construction of the test stand for testing the wear of chain links, considering the synergy of impact of many destructive factors, is justified and possible to be implemented with the use of a simple mechanical system. The proposed stand is marked by small overall dimensions.

It is expected that the results of future research work will be useful in engineering works in the design of new solutions for cable transport machines. Moreover, the authors hope to be able to relate the research work results to other mechanical nodes with tribo-corrosive processes, exposed to many aggressive agents.

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