https://doi.org/10.7494/miag.2021.1.545.66

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Design of a machine for trenchless pipe replacement using the static cracking method

Efforts to minimize surface disturbances during earthworks are an important aspect of modern civil engineering. These expectations are met by a number of technologies that make it possible to carry out such works using trenchless technologies. The static cracking method makes it possible to extend, modernize or renovate the existing underground infrastructure. The paper presents the design of a device assigned for trenchless pipe replacement using the static cracking method. The developed device is characterized by the use of a new type of drive system with the use of articulated rods. In addition, the work proposes ways to solve the main issues in the design of this type of device.

Key words: *machine design, cracking, pipeline replacement, trenchless methods*

1. INTRODUCTION

For many years, the use of excavation methods, i.e. methods that involve digging a trench along the entire length of the conducted earthworks, has been a natural and popular method for the replacement and installation of underground infrastructures. However, the use of these methods is associated with a long duration of work and significant costs that increase with the depth and length of excavation. The effort to minimize the size of excavations has many positive ecological, economic, and legal aspects [1].

The expansion of urban agglomerations, dynamic development of transport infrastructure and growing requirements concerning the interference of earthworks with the surroundings make it necessary to search for alternative methods and technologies of conducting earthworks in relation to excavation. For this reason, trenchless methods have developed significantly in recent years as a cheaper and faster method of performing such works. A significant advantage of trenchless technologies is the relatively low impact on the environment and the direct surroundings of the worksite.

There are a number of different trenchless methods, such as microtunneling, hydraulic pipe jacking, cracking, and repair with resin agents [2]. The technologies of pipe replacement by trenchless methods can be divided into technologies with leaving the old pipe in the ground or removing it. The first of these is the subject of this paper and is known as Pipe Cracking and Pipe Bursting. The cracking method is used whenever the purpose of the work is to increase the diameter of the repaired duct and during the replacement it should be increased. Another way of renovating pipelines is repair with resin agents, which, however, results in significant reductions in the inner diameter of the replaced pipes [3].

The paper presents a proposal for the modernization of a device for trenchless pipe replacement by static cracking.

2. THE PIPE CRACKING METHOD

The cracking method involves the renovation of pipelines through the destruction of the old ones by crushing, tearing or cutting, and laying new sections of pipes in place of the old ones.

Implementation of the method begins digging a starting and ending chamber between which the section of the pipeline requiring replacement is located (Fig. 1). The next step is to insert a rod or rope into the old duct, which is used to pull the cutting head. Next, the cutting head is pulled into the old duct together with a new duct of a given diameter, creating a new channel. The head should have a diameter at least 10% larger than the diameter of the new hose. This avoids problems with the new conduit getting stuck in the soil. Additionally, the hole should not be made too large due to the possibility of soil collapse at the ground surface [4]. Debris from the old conduit after destruction remains in the soil.

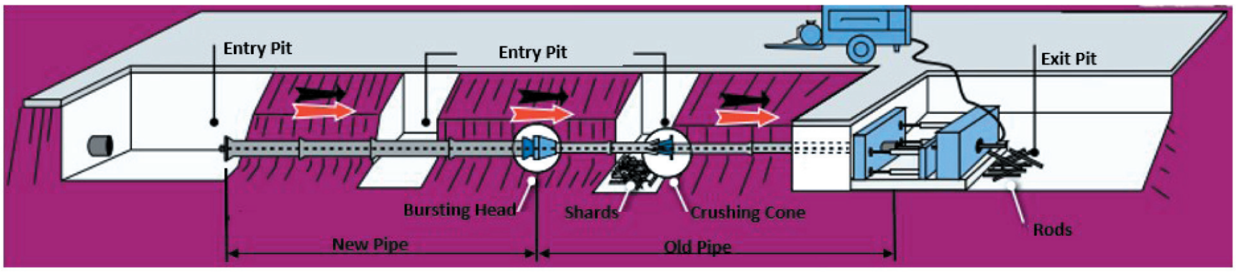

Fig. 1. Schematic of a static cracking system [5]

3. DESIGN ASSUMPTIONS

It was assumed that the maximum diameter of the new pipe is 450 mm, and the minimum diameter of the pipe to be replaced is 400 mm. The relationship between the diameters of new and replacement pipes is shown in Table 1. The green color indicates the ranges of the machine with the currently used rods, the red color indicates that the main unit meets the power reserve, while the rods are too wide. Black indicates the general relation of enlarging the diameters of already existing ducts [2].

The maximum length of the replaced pipeline for the cracking method is 120 meters [2]. Considering the classic case of a machine performing pipe replacement by the cracking method, the length of the replaced pipe was assumed to be l = 80 m.

The machine pulls the heads to preclude the occurrence of compressive stresses and exposure of the drive systems components to buckling. The rods are tensioned only.

Table 1

Possibilities of increasing diameters

New pipe diameter [mm]	Diameter of pipe to be replaced [mm]															
	50	75	100	125	150	200	225	250	300	350	375	400	450	500	525	600
63	🔴					Legend										
90	🔴	🔴				🟢		The described machine can be used								
110	🔴	🔴	🔴			🔴		Enough power, rod replacement required								
125	🔴	🔴	🔴			🟢 🔴 ⚫		Possibility of replacement by cracking method								
180		🔴	🔴	🔴	🔴											
200			🔴	🔴	🔴											
225			🔴	🔴	🔴	🔴										
250			🔴	🔴	🔴	🔴	🔴									
280				🔴	🔴	🔴	🔴	🔴								
315					🔴	🔴	🔴	🔴	🟢							
355					⚫	⚫	🔴	🔴	🟢	🟢						
400							⚫	⚫	⚫	🟢	🟢					
450							⚫	⚫	⚫	⚫	⚫	🟢				
500							⚫	⚫	⚫	⚫	⚫	⚫	⚫			
560										⚫	⚫	⚫	⚫	⚫	⚫	
630										⚫	⚫	⚫	⚫	⚫	⚫	⚫

4. DEVICE DESIGN

4.1. Required manpower

Based on the empirical equation (1) [6], the required pulling force of the pipe bursting head was determined:

$$F = \frac{\pi \cdot g \cdot \left(D^2 - d^2\right)}{4800} \ [\text{kN}] \tag{1}$$

where:

- g – acceleration of gravity (g = 9.81 m·s^{-2}),
- D – diameter of the new pipe increased by 20% (D = 540 mm),
- d – diameter of the pipe to be renovated (d = 400 mm),
- 4800 – empirical scaling factor.

Assuming the upper values of diameters, adopted in the design assumptions, the maximum working force is 845 kN. In order to ensure a surplus of the working force, the value for the designed machine of 1 MN was assumed for further calculations.

4.2. The rod design

Among the main components of the designed system, with a significant impact on the structure and operation of the device, are the rods. Their role is to ensure the transmission of the operating force, and their design determines how the force is transferred from the hydraulic system to the cutting-crushing head. In this project, two concepts of rod design were considered.

Cylindrical threaded rods

A common approach is to use rods with threaded ends (Fig. 2).

Fig. 2. Threaded rod

To drive this type of rod, an expanding and clamping jaw system is used (Fig. 3). This connection transmits force only one way, while the other way provides free return of the actuator. This solution is characterized by very good mechanical strength. The main disadvantage of this method is the necessity to twist/untwist the rods in the excavation while inserting and removing the rods from the machine. This is due to the lack of flexible connections between the rods to allow angular movement between them.

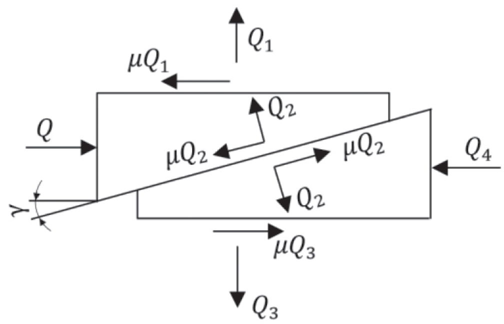

Fig. 3. Principle of operation of an expansion-clamp connection [7]

Articulated rods

To overcome the inconvenience of using the twisted rods, a new rod design is proposed in this paper (Fig. 4). The poles are connected by means of articulated joints. In this way, the flexibility of the rod set is increased and it is possible to connect and pull them out of the trench. The articulated connection also prevents bending stresses from occurring at the rod connection point. The discussed design variant is based on a pin connection.

The design of the rod necessitates a change in the method of force transmission from the friction (expansion-clamp) method to the shape method. The element that fulfils the task of power transmission is a linear pawl with an appropriate geometry. In order to implement the pawl-rod coupling, holes were made in the cylindrical part of the rod cooperating with the pawl.

Fig. 4. Design of the proposed articulated rod

For the adopted design assumptions and estimated loads of the working system, strength calculations of the rods were carried out. The calculations show that the highest stress in the rod is 312.5 MPa, while the highest surface pressure occurs in the forks and amounts to 81.2 MPa. The obtained stress values indicate that it is possible to make the rods with the use of most steels with high strength properties and good weldability. Weldability is important due to the way these elements are shaped.

4.3. The linear pawl design

The geometry of the designed linear pawl is shown in Figure 5. This element is designed to transmit the force from the hydraulic system that pulls the rods and then allows the cylinders to return to their initial position relative to the stationary rods.

Fig. 5. The linear pawl

The designed pawl was subjected to FEM strength analysis. The highest stresses occur in the area marked in red in Figure 6. The maximum reduced stresses according to the H-M-H hypothesis amount to nearly 445 MPa and occur as local concentrations. The pawl stiffness is satisfactory, with the maximum deformation less than 2 mm.

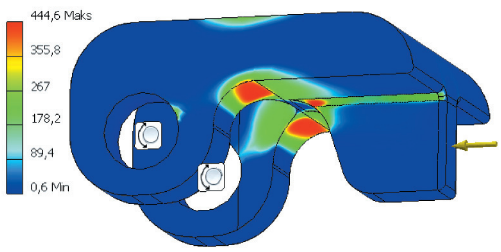

Fig. 6. Results of linear pawl FEA analysis

4.4. Calculations of pawl pins

The pin in the pawl is fixed on one side in a bushing of thickness *g*. This seating is due to the rods passing through the pawl, which makes it impossible to support the pin on both sides.

Figure 7 shows a cross-section of the pawl assembly, the pin (1) transfers the force between the linear pawl (4) and the actuator (2), the cover (5) protects the pin from slipping out, and the pawl tooth (4") allows the rods to move. The pawl cover is marked with a number (3).

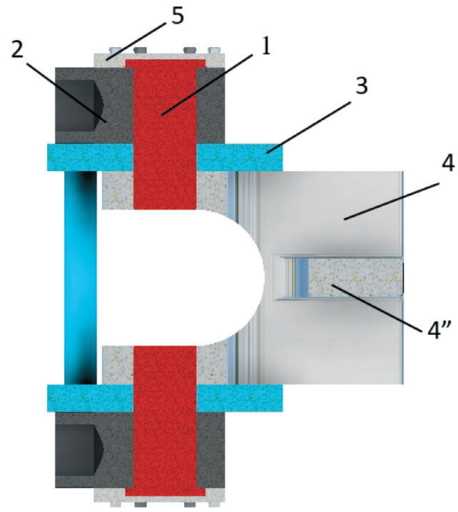

Fig. 7. The pawl assembly

The one-sided restraint of the pin (see Fig. 7, pos. 1) is unfavorable because bending of the pin occurs during subsequent operating cycles. Due to the nature of the loading, verifying fatigue calculations of the pin are required to verify the durability and reliability of the arrangement.

To check the value of the safety factor δ, the Soderberg formula [8] was used:

$$\delta = \frac{Z_{gj}}{\frac{\beta \cdot \sigma_a}{\varepsilon} + Z_{gj} \cdot \frac{\sigma_m}{R_{eg}}} \tag{2}$$

where:

- R_{eg} – bending yield strength [MPa],
- Z_{gj} – single-sided bending fatigue limit [MPa],
- β – stress concentration factor [–],
- ε – size factor [–],
- σ_a – stress amplitude [MPa],
- σ_m – average stress [MPa].

In addition, calculations were performed to verify the value of surface stresses in the pin-actuator sleeve and pin-pawl connections.

Due to the unilateral attachment of the pins in the rod, the total surface pressure in the connection is the sum of the pressure resulting from the force in the direction normal to the working surface of the pin and the pair of forces balancing the bending moment. The load model adopted corresponds to the scheme shown in Figure 8.

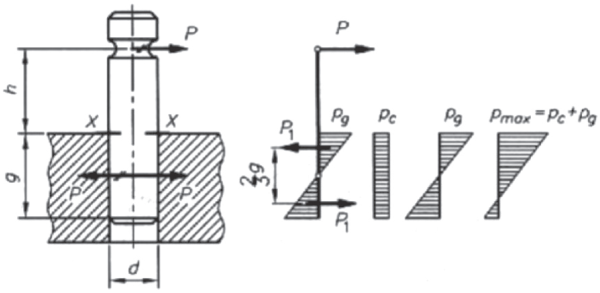

Fig. 8. Simplified model of the pin load [7]

The value of the maximum surface pressures in the pin-actuator sleeve connection is expressed as [9]:

$$p_{\max 1} = \frac{P\left(6\dfrac{h}{g_1}+4\right)}{g_1 d} \tag{3}$$

while the stresses in the pin-pawl joint:

$$p_{\max 2} = \frac{P}{g_2 d} \tag{4}$$

where:

- h – distance from where the force is applied to the surface [mm],
- g_1 – the depth of the pin insertion into the tight-fitting cylinder sleeve [mm],
- g_2 – depth of the pin insertion in a loosely fitted pawl [mm],
- P – force acting on the pin [kN],
- d – pin diameter [mm].

Based on eq. (2), the value of the safety factor in the most loaded cross-section was determined (δ = 1.29). In addition, from eqs. (3) and (4), the values of surface pressures were obtained as $p_{\max1}$ = 227.5 MPa and $p_{\max2}$ = 59.6 MPa.

The obtained value of the fatigue safety factor ensures the proper functioning of the joint. However, the surface pressure results indicate that additional bushing of the pin holes in the cylinder sleeve using high-grade, tempered steel bushings may be necessary.

4.5. The modular expansion pin

The limited space between the rods and the pawl restricts the size of the rod connection pin. Therefore, it is not possible to use a typical headed pin or a pin protected against axial displacement by means of a locking pin. The concept of pin embedding using a push-in connection must also be rejected for operational reasons, due to the long pressing pin in and the wear of elements during frequent connecting and disconnecting.

For the discussed system, a conceptual design of the expanding pin (Fig. 9) connecting the rods was developed, in which the above-mentioned problems were eliminated. The expansion pin consists of a tapered axle (1) with a hole cut out, which is fixed in the eye and fork first.

The tapered bushing (2) is inserted, followed by the tapered head bolt (3) and the tapered head pin ending in a threaded hole (4). The screw and the pin are screwed together to press against the surface and prevent the expansion pin from moving.

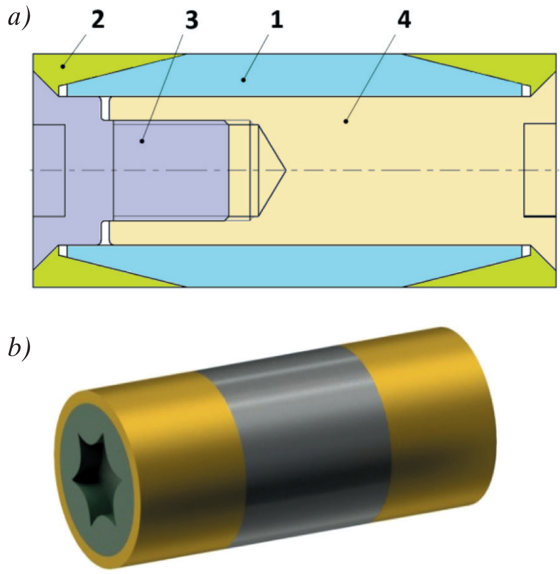

Fig. 9. Expansion pin: a) cross-section; b) view

4.6. The device frame

The frame (Figs. 10 and 11) is welded from square tubes (120×120×8). To improve the rigidity of the frame, four identical profiles are welded to connect the lower frame sections. Holes have been made in the frame and nuts have been welded to fix the bolts that fasten the machine cover. Holes were also made to place the hydraulic quick couplings.

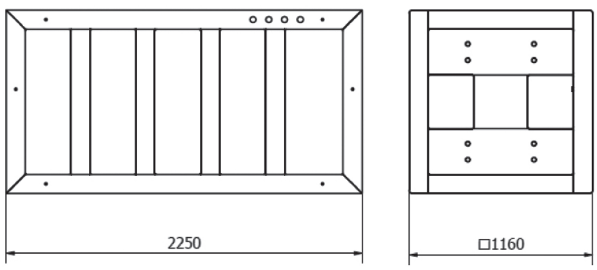

Fig. 10. Frame dimensions

A 50-mm-thick plate was welded to the front of the frame, which is responsible for transferring the force from the actuators to the side wall of the excavation (see Fig. 11).

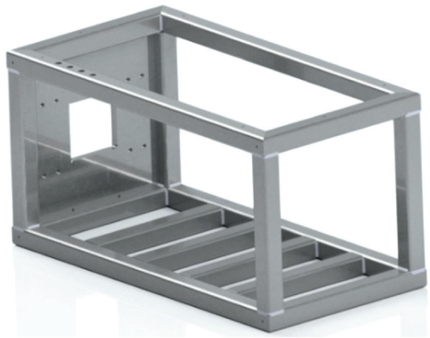

Fig. 11. Frame model 3D

4.7. Selection of actuators and hydraulic Power Pack

Two, symmetrically spaced, actuators were selected to evenly distribute the workload. They have to fulfill two tasks, the first one is being to provide the required force to pull the cutting head, and the second one is to provide enough stroke to allow the ratchet tooth to move to the next hole in the rod. The selected actuator was WHC027 – 160×90×600, which parameters are shown in Table 2 [10].

Table 2
Actuator specifications

$\varnothing D_w$	160 mm	piston diameter
$\varnothing d$	90 mm	piston rod diameter
G_w	M100×2	thread size
xL	600 mm	actuator stroke
L	842 mm	dimension without threaded part
L_c	943 mm	closing dimension
C	40 mm	extend the actuator
PD	70 mm	dimension for hydraulic connection 1
Pz	662 mm	dimension between hydraulic connections
p	25 MPa	nominal pressure

A suitable power supply should provide the appropriate pressure value and hydraulic oil volume flow. A power supply with the parameters shown in Table 3 was selected to supply the previously selected actuators.

Table 3
Hydraulic power pack parameters

Oil tank capacity	50 l
Oil flow rate	12 l/min
Working pressure	25 MPa

4.8. Hydraulic circuit diagram

Figure 12 shows a diagram of the hydraulic system of the discussed device. The operating method of the system should ensure uniform ejection of the actuators (1, 2) regardless of the load, with a flow divider (3) selected for this purpose. A controlled check valve (4) prevents uncontrolled movements of the actuators. The control uses a 4-way 3-position hydraulic solenoid valve (5) which controls the direction of hydraulic oil flow. The drive system is supplied through a spring loaded check valve (7). A safety valve (6) is used to protect the pump from pressure build-up. The source of pressure in the system is the hydraulic power pack, which consists of a pump (8), a coupling (9), a motor (10), a suction filter (11) and an oil tank (12).

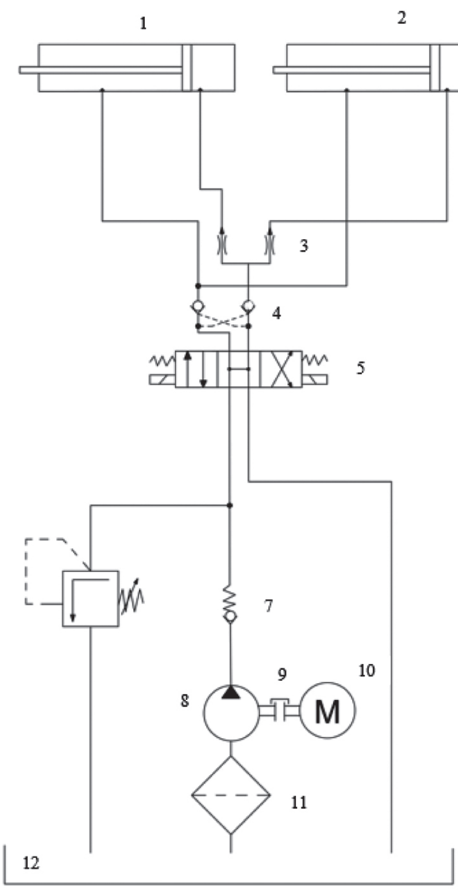

Fig. 12. Hydraulic diagram

5. DRIVE SYSTEM

The drive system (Fig. 13) consists of two hydraulic cylinders (1) mounted symmetrically on a plate that rests against the side wall of the trench during operation. The cylinders set the pawl (2) in motion. The force is transmitted from the cylinders to the pawl via two pins (4). A loose fit between the pawl and the pins allows the pawl to pivot. Longitudinal holes (3") have been made in the poles, which cooperate with the pawl tooth, allowing the force to be transferred to subsequent rods and the working unit of the device to move (rods with working head) in the direction indicated by the arrow (5).

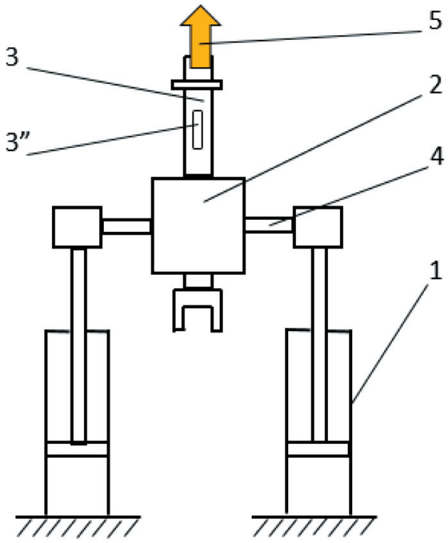

Fig. 13. Schematic of drive system

The overall dimensions of the machine are given in Figure 14, while Figures 15 and 16 show views of the developed system. The designed cracking machine consists of a frame (1) in which the machine components are located. A corrugated plate (7) is welded to the underside of the frame. The housing of the machine is made of perforated sheet metal (16) and a back solid sheet (15). An inspection door (17) is provided in the machine casing to allow access to the drive system.

During operation, the unit is supported against the excavation wall by a sheet metal plate (8) to which the hydraulic cylinders (5) are attached by means of plates (9). To protect the actuators from bending, the support brackets (10) that are welded to the frame are bolted to the actuators. The pawl (2) and pawl cover (3) are connected to the actuators with a pin. The pin is fixed in the end of the piston rod (11) and is secured against sliding out by the cap bolted with screws (12). The pawl sets the rods (4) in motion, which move along the element (6) that fixes the position of the rods vertically and ensures contact between the rods and the pawl. The machine is divided into two compartments by sheet metal (13), which has two functions: the first is to support the element (6) and the second is to protect the hydraulic lines from movement of the drive system elements. Rods are connected by means of expanding pins (14). In the upper part of the device, there are eyebolts (19) used to hang the device for transport. To facilitate hydraulic connection, the machine is equipped with hydraulic quick couplings (18).

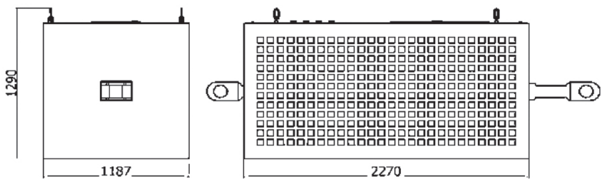

Fig. 14. Dimensions of the device

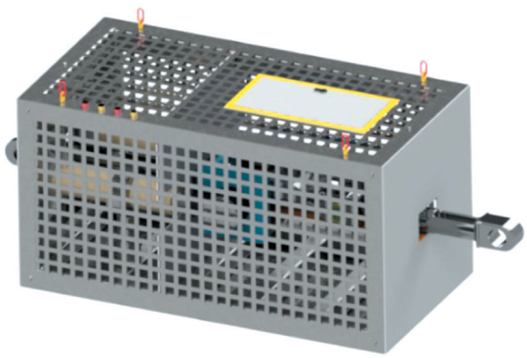

Fig. 15. Cracking machine

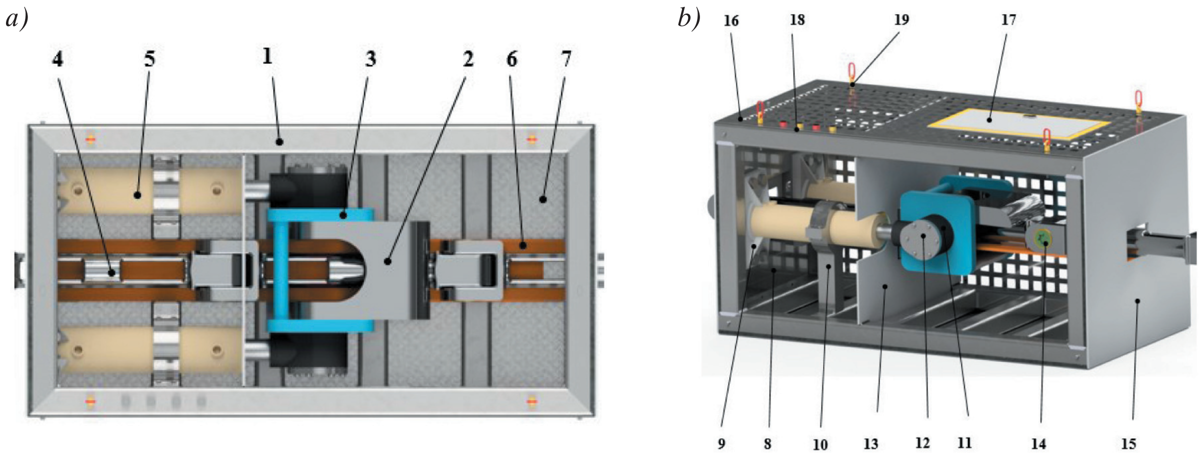

Fig. 16. Cracking machine, views: a) without top cover; b) without side cover

6. OPERATION OF THE MACHINE

To ensure the proper functioning of the device, a service trench with dimensions of 5×2 m and a depth adapted to the depth of the pipeline to be replaced must be dug. The machine (Fig. 17, pos. 1) should be placed in the trench and leveled. The axis of the drive rods should coincide with the center of the cross section of the pipe to be replaced. After placing the machine in the trench, the hydraulic hoses connect to the hydraulic quick couplings. The rods (2) are to be placed in the repaired pipeline, and the cutting/crushing head (1) attached to the last one. Expansion pins (3) should be used to connect the rods. When the hydraulic power pack is turned on, the actuators set in motion a pawl, the tooth of which falls into a hole in the rods and unilaterally blocks the movement of the pawl relative to the rods and together they pull the cutting/crushing head. The return movement of the actuator causes the pawl to move relative to the non-moving rods. The device crushes or cuts the old pipeline with the head, leaving pieces of it in the ground. The new pipe is pulled in with the head. After the renovation, the new pipe takes over the functions of the old one. After pulling, the rods are laid out in the trench (5).

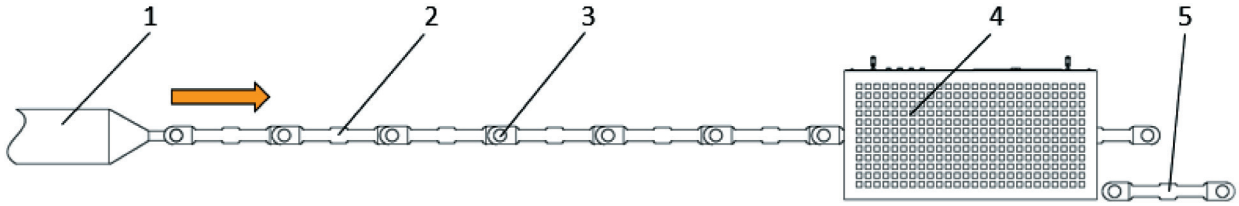

Fig. 17. Operation of the machine

7. SUMMARY

The paper presents a proposal for the modernization of the design of a device for trenchless pipe replacement by means of the static cracking method. The developed device enables the replacement of pipelines with a length of 80 m and a diameter of up to 400 mm.

In the project, special attention was paid to the most loaded elements of the machine and appropriate strength calculations were performed. The proposed solutions, especially the new concept of the rod and the method of its drive, can make the presented system competitive in relation to currently used devices for static cracking.

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