



A new approach to controlling a hydraulic indirect elevator with a variable-speed pump

Łukasz Stawiński¹ · Andrzej Kosucki¹ · Adrian Morawiec¹ · Justyna Skowrońska¹

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Abstract

The paper describes a new approach to the issue of controlling an indirect elevator with a bidirectional variable-speed pump and a simple controller based on the position sensor. The aim of this paper is to present a method of controlling the speed of the elevator to ensure smooth movement and proper positioning of the car on the floor, regardless of its load and ropes rigidity. The main feature of the proposed solution is the use of a frequency inverter in vector mode to control the speed of the car in both directions. The control function is based on virtual cams comparing actual measurements from the car position sensor. The proposed control strategy has been experimentally verified on the existing indirect elevator drive, and the obtained results indicate a very high accuracy in maintaining and shaping the speed and positioning of the car. The conducted research confirms the possibility of using a new method of controlling hydraulic and indirect elevators. The benefits of this method include a less complex hydraulic system, the control of overloads in the car and vibrations in the rope system, and the possibility of energy recovery.

Keywords Electro-hydrostatic drives · Elevators · Velocity control · Position control · Frequency inverters · Variable-speed pump

1 Introduction

Control is one of the most important aspects of designing hydraulic machines and systems. Choosing the appropriate control method enables the optimum operation of the entire system and its components and can positively influence, e.g., productivity and energy efficiency. Proper control plays a key role in machines that are used to transport people, such as elevators. It is important to ensure the safety and comfort of traveling. There must be no excessive acceleration in the car during the start and stop of the elevator, and the floor of the car must be flush with the floor of the floor after reaching the selected stop. The issue of control of hydraulic systems is widely discussed in [1], including hydraulic elevator systems [2]. One can observe a tendency to move away from the control of working fluid flow through throttle valves, as such an approach is disadvantageous in terms of energetic efficiency.

Energy is converted into heat during the flow through the throttle valve.

The issues of energy management and energy recovery in elevators are increasingly taken into account when developing new drive systems. Dalala et al. [3] discussed an energy management system for an elevator equipped with energy storage. The dc-link of the regenerative motor drive is connected to the energy storage device through a dc/dc power converter. In this case, there is a reverse flow of power so that it can be used later to lift the load. The proposed control strategy has an additional degree of freedom due to the fact that the dc reverse voltage has two limits. In turn, Jabbour et al. [4] and [5] discussed control strategies for an energy recovery system in an elevator. The supercapacitor-equipped energy recovery system (ERS) is connected to the dc link of the lift motor drive via a bidirectional dc–dc converter. In addition, two fuzzy logic controllers are part of the system, whose task is to regulate the dc-link voltage via the dc–dc converter.

Energy recovery and storage systems can be found in many devices using hydraulic drive systems. For example, Chen et al. [6] presented the control of an energy recovery system in a hybrid hydraulic excavator equipped with

✉ Łukasz Stawiński
lukasz.stawinski@p.lodz.pl

¹ Faculty of Mechanical Engineering, Institute of Machine Tools and Production Engineering, Lodz University of Technology, Stefanowskiego 1/15, 90-924 Lodz, Poland

an accumulator and a valve–motor–generator system is described. The motor speed is controlled by a variable-speed generator. On the other hand, the flow rate control of the proportional flow valve and the throttle is responsible for the cylinder speed. Lin et al. [7] also proposed a control system in a hybrid hydraulic excavator with ERS. In this case, the AMGERS (accumulator–motor/generator energy regeneration system) was used, which provided better control of the response and speed of the excavator boom, thus allowing for better control performance. Wang et al. [8] presented the design of an excavator ER (energy recovery) controller in a system using a composite control strategy. In the system discussed, an electric generator is driven by a hydraulic motor. The speed of the generator is controlled by a PID controller with load estimation and leakage flow compensation.

To obtain a good representation of the assumed speed of the elevator car, research is being carried out on new control strategies and elements of the drive systems [9]. The main parts of the system were a bidirectional variable-speed pump and an electric motor powered by a converter operating in torque regulation mode. The flow rate of the working fluid was controlled by the speed setting of the pump and not by the use of valves. Simulation tests showed that the control system used was robust to leakage at the pump and parameter mismatch. Based on a similar idea, a simulation model and a real drive system to control the speed of the scissor lift were created [10]. It used a fixed displacement pump, the flow rate of which was dependent on the speed of the asynchronous motor driven by a frequency inverter (FI). The effect was to maintain a constant speed of the work platform despite the variable ratio of the kinematic system of the scissor lift.

Li et al. [11] presented an electrohydraulic elevator system with two cylinders equipped with three proportional flow control valves. In addition, the system is equipped with pseudo-derivative feedback (PDF) controller, whose task is to acquire the closest approximation to the set car velocity pattern. This makes it possible to control the speed of the lifting car. A PD controller with step limits ensures that the non-synchronous error between two cylinders on the two sides of the car is minimized. A speed control system for a variable voltage variable frequency (VVVF) hydraulic elevator equipped with a pressure accumulator to improve energy efficiency is presented in Ref. [12]. The system studied consists of the main circuit with a pump and a motor and an accumulator circuit with a motor and a hydraulic pump. The speed of the lift in the discussed system can be controlled using a PID control algorithm. Yang et al. [13] discussed the speed control of a VVVF hydraulic elevator. In this case, however, the system consisted of two parts: the first with a VVVF motor and a fixed displacement pump, while the second was an electric motor, a hydraulic motor, and energy feedback.

Kim et al. [14] described the speed control of a hydraulic elevator through a specially designed two-stage nonlinear controller. The operating principle of this controller was based on the Lyapunov redesign method. Zhao et al. [15] discussed an electrohydraulic hybrid elevator drive system with a hydraulic accumulator. Here, a field-oriented control (FOC) strategy is used to control the elevator speed. In this case, a multi-loop control strategy is adopted. Both the inner loop (current) and the outer loop (speed) use a PI controller for operation.

Pena et al. [16] describe the architecture of a hydraulic counterbalance elevator system. The authors proposed a physical model of the architecture, where the ratio of the pump/motor control disc angles is the control input for the cab speed output control. However, a PI controller was used to control the displacement.

Xu et al. [17] discussed the speed control of a hydraulic elevator through a conventional PID controller and a self-adjusting fuzzy PID controller. The experimental results confirmed the superiority of the latter in terms of control performance. Hu et al. [18] discussed the velocity tracking control strategy of a hydraulic elevator system. The paper proposes an algorithm for PD and feedforward–feedback control.

Kumar et al. [19] presented an elevator control system with a hydraulic motor and the PLC. The controller in this system was used to adjust the directional valve to control the hydraulic motor. This approach resulted in benefits, such as a reliable and constant lifting speed.

Mohammed et al. [20] and [21] discussed a hydraulic elevator speed control system using a proportional valve and a PI controller.

A control strategy for an offshore platform lifting system is presented in Ref. [22]. In this case, the system is equipped with proportional speed control valves, which are used to control the lifting speed of the platform. Wang et al. [23] describe the control of a hydraulic actuator for elevator systems through the appropriate control of proportional valves—a directional valve, a throttle valve, and a pressure valve. The proposed hybrid observer, based on the output feedback controller, realizes the multi-objective task of position tracking and energy saving. Ranjan et al. [24] described a control system in a hydraulic excavator equipped with a hydropneumatic accumulator. The implementation of the actuator position control is carried out by a PID controller cooperating with a proportional flow control valve (PFCV). An additional component of the control system is the model predictive controller (MPC), which in turn affects the performance. A modern control system for a hydraulic excavator powered by a variable-speed and displacement pump is proposed in Ref. [25]. The pump in turn is driven by a variable-speed motor. A matching method based on segmented speed and continuous pump

output control was used to ensure optimal dynamic and energy characteristics of the system.

Some drive systems combine speed control and energy management. There are systems also controlling the position of actuators. A control scheme for a hydraulic excavator using a pressure compensator and an energy storage device is discussed in Ref. [26]. This approach was able to provide adequate control performance. The load position control of a forklift with the ERS energy recovery system is discussed in Ref. [27]. By using a permanent magnet synchronous motor coupled to a reversible hydraulic machine, no servo valves are required. The application of the principle of variable-displacement control enables the speed and position of the forks of the forklift to be controlled. In Fu et al. [28], the topic of forklifts with an energy recovery system is again addressed: in this case, a strategy with two control modes. The first one is a mode using a conventional valve to control throttling and thus control boom speed. In the second control mode (the so-called HMG mode)—the implementation of boom speed control is done through two sets of HMGs. Yu et al. [29] discussed the control strategy of a hybrid forklift equipped with an energy-saving and storage system. In this case, the speed of the cylinder is given as a feedback signal from the main valve to the PID controller; in turn, the speed of the electric motor is controlled by the PID controller precisely based on the speed of the cylinder.

Most of the presented drives are based on closed-loop control systems. Usually, the satisfactory effects of speed mapping and positioning are obtained. These types of systems are usually quite complex and require measuring devices operating during the whole duty cycle. The presented drive system joins new possibilities of (FI) usage in hydraulic drives with the advantages of classical elevator control system simplicity based on the limit switches (cams). It is also possible to implement closed-loop speed or position control systems.

The main purpose of this paper is to present a new drive system with a variable-speed pump operating in the open loop control system for an indirect elevator. The system takes advantage of FI capabilities, e.g., setting acceleration and deceleration times, communication, and safety capabilities (vector mode).

This paper is organized as follows. Section 2 describes the currently used systems for hydraulic elevator drives, pointing to their advantages and disadvantages. Section 3 contains a description of the system including design, operational parameters, and basic calculations. Section 4 presents the results of experimental tests of the new system loaded by different masses of payload and the determination of precision of the hydraulic cylinder and car positioning and speed mapping factors. An analysis of the test results, conclusions, and implementation possibilities are presented in Sect. 5.

2 Problem statement and proposed solution

The currently used hydraulic elevators are equipped with the two most common drive systems. One is based on a series of pressure and throttle valves and the other uses a proportional directional valve or, less rarely, a directional servo valve. Both systems must be properly controlled to ensure smooth elevator movement and proper positioning of the car on the floor. According to the standards (EN 81-20 and EN 81-50), the requirement is to maintain a positioning accuracy not greater than ± 10 mm avoiding accelerations above 1 m/s^2 . In the case of indirect lifts, an additional problem to comply with these restrictions is the susceptibility of steel ropes. If the starts or stops are too abrupt, they can vibrate and behave like a spring. In addition, the greater the car payload, the more the ropes stretch, which makes positioning difficult. Simplified hydraulic schemes of the two most popular elevator control systems and the new one proposed in this paper are shown in Fig. 1. Next to the hydraulic schemes, diagrams of the car velocity are provided, based on the supply scheme for the relevant control elements. A typical diagram of the car velocity movement consists of five stages: accelerating the car to the transport speed, driving the car with the transport speed, braking the car to the creep speed, driving the car with the creep speed, and braking the car until it stops. The arrows in the diagrams show the direction of movement of the cabin, i.e., lifting and lowering, respectively. The working principle of each system is described below.

The multi-valve electrohydraulic control solution is used (Fig. 1a). Some valves have appropriate channels in which the oil, under a given pressure, closes or opens the oil flow, respectively (3, 4, 7, 8, 9, 10). Other valves have solenoid coils and thus allow the oil to flow through them (5, 6, 9). Implementation of the car speed diagram takes place thanks to the appropriately selected stages of supplying the solenoid valves, which affect the pressures in the lines supplying the pressure valves. Creep speeds are realized through appropriately set throttling valves. The actuator (12) is lifted by switching on the motor (1) driving the pump (2) and powering the valves coils (5 and 6). Lowering does not require the work of the motor, because the actuator lowers under the payload after supplying the valve coils (6 and 9). The rupture valve (11) prevents from falling as a result of the actuator failure or a hose break and the pressure relief valve (3) limits the maximum pressure in the system. In this solution, the potential energy, as the actuator is lowered is converted to heat as the oil is forced through the narrow channels of all the valves. It is a complex system, which is additionally susceptible to load changes, which affects the accuracy of the positioning of the car on the floors.

The proportional directional valve solution does not require as many valves as the previous one (Fig. 1b).

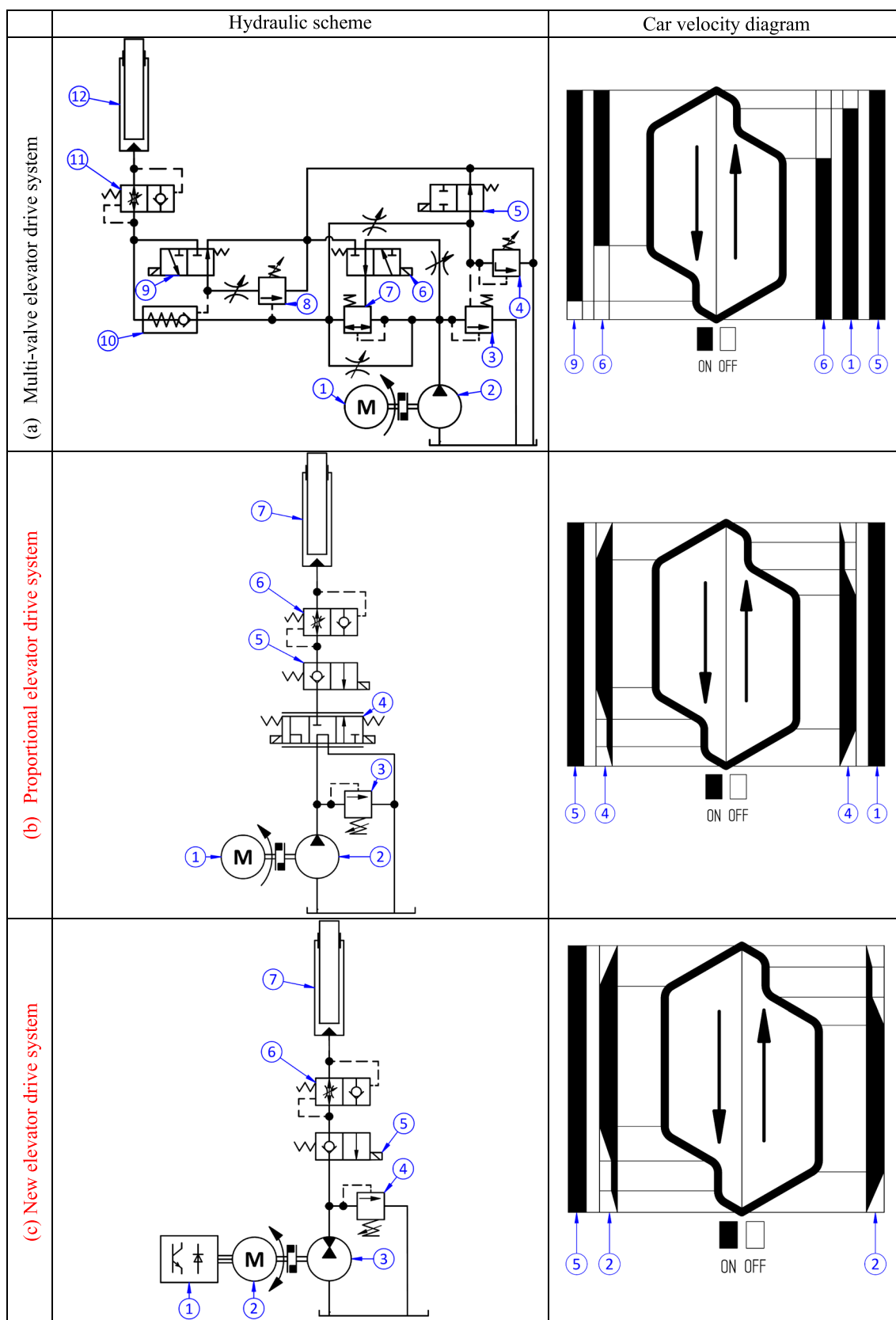


Fig. 1 Comparison of elevator drive system concepts

Appropriate actuator movements are performed by supplying the directional valve with an appropriate current value, which is proportional to the stage of opening of the respective channels inside the valve. When the car is lifting, motor (1) is powered, which drives pump (2) and oil flows to the directional valve (4). The controller supplies one of the valve coils according to the set function to gradually open the flow from the pump to the actuator. Creep speeds are realized by closing the oil flow to the actuator. During this time, some of the oil is drained back into the reservoir. This is not an economical solution, because the pump operates at an increased level of power, caused by an increase in pressure due to the reduction of the passage in the proportional valve. Additionally, the heating up of the oil is a side effect. When lowering, as in the multi-valve solution, the motor does not require any power. The oil flow from the actuator (7) is opened by applying power to valve coil (5), and then appropriate power to the second coil of the directional valve (4). The oil also rubs through the narrow channels of the directional valve and heats it. The rupture valve (6) prevents from falling as a result of the actuator failure or a hose break. Valve (3) limits the maximum pressure in the system. The positioning of the car in such systems is also difficult and dependent on the load—especially during lowering, where there is a dependence on the flow through the orifice in the directional valve. In addition, the controller cooperating with a proportional valve requires an appropriate amplifier, which generates additional energy consumption.

The proposed new drive system is a combination of the advantages of both systems (Fig. 1c). As before, the pressure relief valve (4) and the rupture valve (6) are responsible for safety. The lifting and lowering cycles take place by changing the direction of the motor's rotation and shaping its speed accordingly. When lowering, the valve coil (5) must be supplied. The pump (3) begins to receive the oil from the actuator (7) and drives the electric motor (2) which works as a generator. The braking torque is generated by a FI (1) with the vector mode, which ensures stable operation.

The main purposes of this paper are:

- the design of a new elevator hydraulic system with a simple quasi-open loop control system,
- manufacture and examination of the elevator.

The most important thing is to properly position the car so that it reaches the same assumed position with the greatest possible accuracy (less than 10 mm) regardless of the weight of the transported load. The assumption of the system is smooth control of the car velocity with the fixed displacement pump, asynchronous motor, and FI based on the simple quasi-open control system. The additional advantage of

the proposed solution is the possibility of energy recovery (while lowering) and elimination of oil heating.

3 New drive description

3.1 Structure

The indirect rope elevator supplied by the hydraulic actuator was equipped with the new drive allowing free shaping of lifting and lowering the car speed using the volumetric method of control. The scheme and photos of the test stand

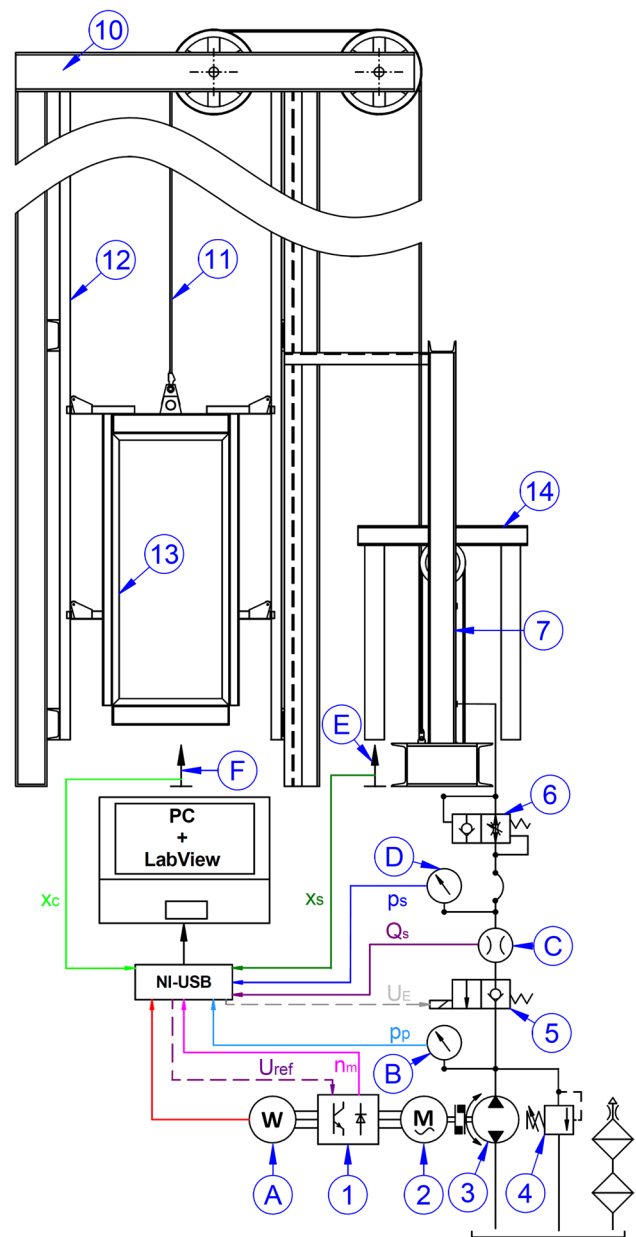
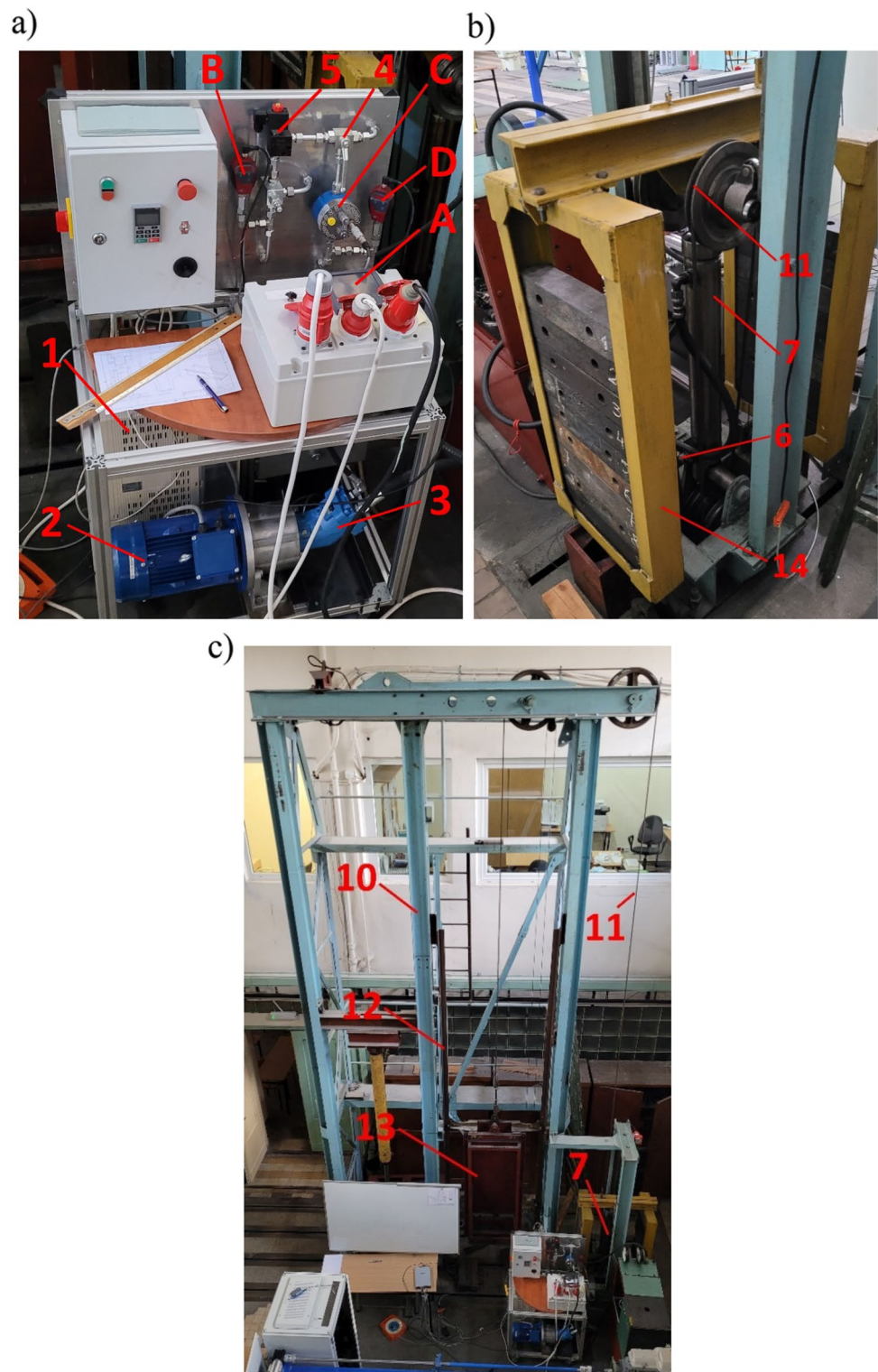


Fig. 2 Scheme of the test stand with prototype hydrostatic system

Fig. 3 The test stand with the new drive system: **a** drive and control system; **b** actuator load structure; **c** test stand structure



with new the hydrostatic system are presented, respectively, in Figs. 2 and 3. It is not a typical indirect rope elevator, due to the transmission ratio of the rope system, which is 4 (usually up to 2) and is not recommended by the elevator manufacturers. In the test rig, only a single rope system is used.

The elevator structure consists of the steel structure of the stand (10), supporting rope system (11), and guides (12) for the car (13). A rope system is connected to the car and driven using a hydraulic actuator (7). Regardless of the car payload, the actuator can be loaded with a fixed mass (14)

attached directly to the piston rod. This allows the system to be tested with the additional mass on the side of the hydraulic actuator. The stand was equipped with two wire encoders (E) and (F) to measure the positions of the piston rod and the car, respectively.

The proposed drive system consists of a reversible fixed displacement piston pump (3) driven by the asynchronous motor (2) fed by the FI (1). The used FI apart from allowing free shaping of the pump speed (including change of rotation direction) is equipped with energy return to the grid module. The solenoid-operated check valve 2/2 (5) was used to secure the system after stopping and to enable the change of the direction of the car's movement. The rupture valve (6) protects the system in the case of a hydraulic cylinder or hose failure. The system is also equipped with a pressure relief valve (4) to limit the maximum system operating pressure. The system is equipped with a set of sensors for analyzing the operation of the drive: pressure sensors (B) and (D), and the flowmeter (C). To manage the measuring and control, the LabView software on PC (8) connected to interface NI USB 6434 (9) was used. The test stand was also equipped with a grid parameters analyzer (A) which allows measuring and registering, among others, the current power consumption of the drive. It will be used in further studies of the energy consumption of the new elevator drive system.

3.2 Operation

The operation of the new system (Fig. 2) is based on controlling the pump (3) speed using FI (1) to force the demanded oil flow to or from the hydraulic actuator (7). The determined oil flow is proportional to the current speed of the asynchronous motor (2) speed. To obtain safe and stable control of the motor speed, the FI must operate in vector control mode.

The car (13) lifting operation is realized by clockwise rotation of the asynchronous motor (2) with assumed speed. The oil flows from the pump (3) outlet by a solenoid-operated check valve (5) to the hydraulic actuator inlet (7). The velocity of the piston rod v_s is proportional to the speed of the pump and can be calculated for the lifting v_{su} and lowering v_{sd} , respectively, as follows:

$$v_{su} = \frac{4 \cdot q_p \cdot \omega_p \cdot \eta_{vp}}{\pi \cdot D^2}, v_{sd} = \frac{4 \cdot q_p \cdot \omega_p}{\pi \cdot D^2 \cdot \eta_{vp}},$$

where q_p , ω_p , and η_{vp} are displacement, angular velocity, and volumetric efficiency of the pump, respectively, and D is the diameter of the hydraulic actuator piston.

The car (13) lowering operation is realized by counterclockwise rotation of the asynchronous motor (2) with assumed speed. The oil flows from the hydraulic actuator inlet (7) through the supplied solenoid-operated check valve

(5) to the pump outlet (3). The pump loads the electric motor with torque proportional to the pressure and pumps displacement and it is the active load (contrary to lifting where the load is passive). This kind of load means that the asynchronous motor operates as a generator and returns the electric power to the electric grid. The velocity of the piston rod is also proportional to the oil flow and can be calculated as follows.

In both cases of cabin movement (lifting and lowering), the assumed motor speed is possible thanks to the use of FI. The start-up functions can be shaped either using the capabilities of the FI, allowing for arbitrary speed acceleration and deceleration times, or using an external speed reference system, in this case, based on a measurement and control card with LabView software [25]. As can be seen, it is possible to set both the speed level and the duration of the individual start-up stages. Other FI features such as e.g., "acceleration/deceleration time shock reduction", which allows limiting drive jerk, can also be used. This is a useful feature for reducing the spring back of ropes. This paper shows an external elevator control signal. The acceleration and deceleration times as well as the motor speed are set in the controller. Switching points of the elevator movement stages (according to a typical car speed diagram) are based on the car position sensor. The technical parameters of the main elements of the test stand are presented in Table 1. The operational parameters of the elevator are presented in Table 2.

4 Experimental tests

Experimental studies were carried out based on similar operating parameters of a typical elevator, i.e., the assumed car speed diagram, different car payloads, and positioning. Shaping the speed of the car is done by appropriate control of the motor speed fed by FI, according to the given input function. The points responsible for the start of each motion stage can be determined by software or by external triggers such as limit switches located along the shaft or virtual switches. In the tested system, the acceleration and deceleration times (t_1 – t_7) as well as the points of change of the elevator operation stages (H1–H4) were included in the external control system. This use of limit switches should provide repeatable positioning of the car at cycle end points regardless of load. Such a solution also dispenses with the need for speed or position feedback systems, but does not preclude the use of such control systems. In this paper, the operation of the system using limit switches is presented. The change in the motor speed is based on the comparison of the values entered by the controller with the current position of the car. The tests were carried out based on the input function shown in Fig. 4a. The virtual switches that

Table 1 Components of the test stand

No	Component	Parameters
1	Frequency inverter: Yaskawa CIMR-UC4E0014AAA	400 V, 5 kW, 14 A
2	Electric motor: MS100L3-4 B5	400 V 4 kW, 1430 rpm
3	Piston pump: Hydro Leduc XPi 12 0,523,820	12 cm ³ /rev, 38 MPa, 0–3150 rpm
4	Pressure relief valve: Ponar A-VMP-PIB-12-SP	35 MPa
5	Solenoid-operated check valve: Winner Hydraulics EP-08W-05-M-05	35 MPa, 30 lpm
6	Rupture valve: VUBA-03-FF-S	35 MPa; 67 lpm
7	Hydraulic actuator: Hydro-Com UCJ2F-16–80/45/500Dz	80/45/500, 16 MPa
8	PC + LabVIEW Software	Dell Inspiron 17 G3 3779
9	Data acquisition device: NI USB 6434	16 AI, 2 AO, 24 DIO USB
A	Power network analyzer: LUMEL ND45	400 V AC; 20/5 A; acc. $\pm 0.1\%$
B, D	Pressure sensors: Hydrotechnik MultiEpc 100	0–40 MPa; acc. $\pm 0.2\%$
C	Flowmeter: QG 110 3185-03-S-35.030	0.2–30 l/min; acc. $\pm 0.7\%$
E, F	Position sensors: Hohner Automation 90.1404.FX + 58–11,112-2000	4 m; acc. $\pm 0.03\%$

Table 2 Operational data

Components	Parameters
Hydraulic actuator stroke	0.5 m
Car displacement	2 m
Piston rod transport speed	0.06 m/s
Car transport speed	0.24 m/s
Nominal pressure	16 MPa
Mass of the car	158 kg
Fixed mass mounted to the piston rod	366 kg
Nominal car payload	732 kg
Rope system transmission ratio	4

were used to change the stages of the car's travel are shown schematically in Fig. 4b and c for the lifting and lowering cycle, respectively. The values of parameters entered in the control system are given in Table 3. The tests were carried out with different car payloads, which resulted in different loads on the hydraulic actuator (Table 4).

An example chart of selected variables during the entire duty cycle with the average payload (419 kg), consisting of lifting the car and after a short stop lowering (according to the function shown in Fig. 4a) are presented in Fig. 5. Figure 5a shows the reference velocity of the motor n based on the given input function. The experimentally measured motor velocity is drawn in green. Comparing these two curves, only slight differences can be seen during the acceleration and braking stages. The rest of the driving stages are very similar and there are no significant differences. Figure 5b shows a comparison of the assumed velocity of the actuator piston rod, based on the calculated function of the motor velocity, with the velocity of the actuator piston rod measured experimentally v_s on the stand. There is a slight difference between the assumed and measured velocities,

especially in the steps of the cabin transport speed. During the lifting phase, the measured value is lower than the assumed value, while in the lowering phase, it is slightly higher. This is due to the efficiency of the systems (hydraulic and rope), which is slightly lower during lifting and reduces the speed of movement. During lowering, the cabin and the rope system with the actuator fall under the influence of gravity, which slightly increases the speed of the piston rod. The same character has the measured value of the car velocity v_c shown in Fig. 5c. In addition, greater velocity oscillations are visible when changing the stages of the cabin's movement. This is due to the flexibility of the rope, which stretches and contracts under the influence of inertial forces. Figure 5d additionally shows changes in the position of the hydraulic cylinder piston rod x_s and the car H . Switching points of the control function and positions after the lift has stopped are visible. The cabin accurately reproduces the set trajectory of movement and reaches the set positions. Referring to the requirements of the maximum accelerations that can prevail in the cabin, the measured accelerations of the car a_c and the piston rod of the hydraulic cylinder a_s are presented in Fig. 5e. These values are below 1 m/s^2 , which shows the high smoothness of driving and travel comfort.

The presented method of controlling the drive system of the elevator confirms a very good representation of the assumed velocities of both sections (piston rod and car).

The compilations of charts of the velocity–position diagram developed for piston rod and car carrying different payloads are shown in Figs. 6a and 7a, respectively. Tests were carried out with car payloads ranging from 0 to 732 kg, shown in different colors as per the legend. Despite different loads of the car, the assumed velocity of the piston rod is mapping with satisfactory precision, and it is maintained at a similar level. The greater the load, the lower (during lifting) and higher (during lowering)

Fig. 4 Limit switches: **a** input function; **b** mounting points—lifting; **c** mounting points—lowering

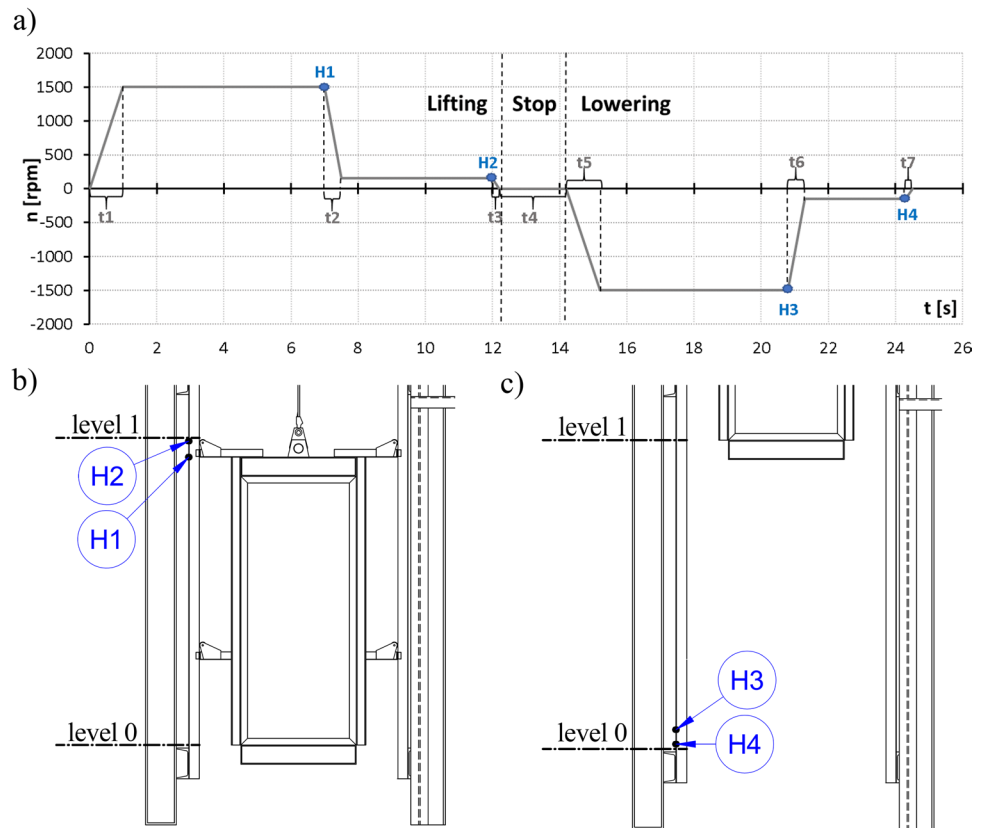


Table 3 Control function parameters

Description of the parameter	Lifting	Lowering
acceleration time to the car transport speed	$t_1 = 1$ s	$t_5 = 1$ s
deceleration time to the car creep speed	$t_2 = 0.5$ s	$t_6 = 0.5$ s
deceleration time to stop the car	$t_3 = 0.2$ s	$t_7 = 0.2$ s
car stop time after lifting cycle	$t_4 = 2$ s	
starting point of braking to the car's creep speed	$H_1 = 1.5$ m	$H_3 = 0.1$ m
starting point of braking to stop the car	$H_2 = 1.59$ m	$H_4 = 0.01$ m
motor transport speed	$n_1 = 1500$ rpm	
motor creep speed	$n_2 = 150$ rpm	

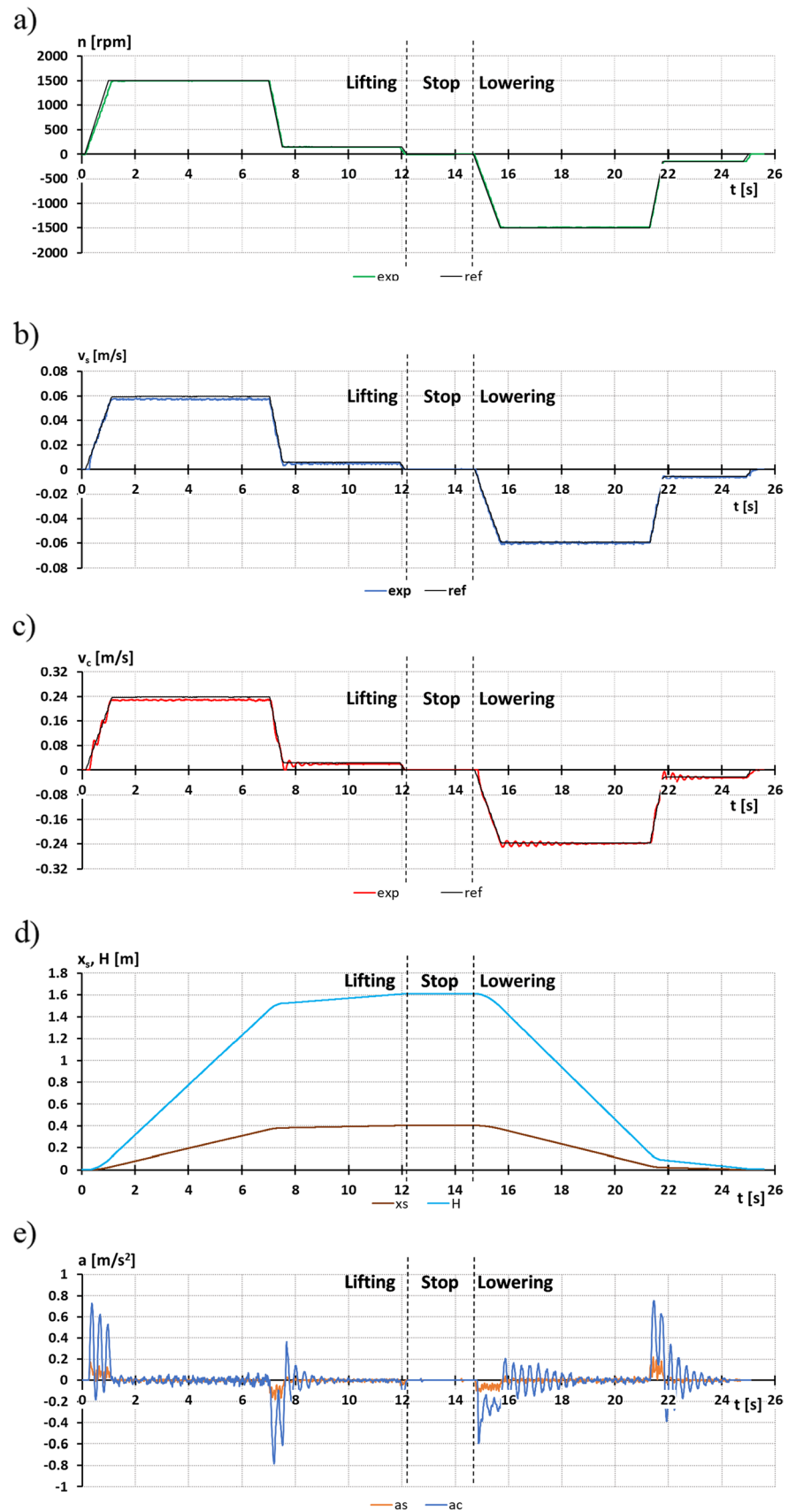
Table 4 Operational loads

Payload mass in the car [kg]							
0	105	210	314	419	524	627	732
Piston rod load [N]							
9790.38	13,910.58	18,030.78	22,111.74	26,231.94	30,352.14	34,393.86	38,514.06

the speed of the piston rod what is slightly visible. This is due to the flow losses in the pump, which increase with increasing pressure in the pump discharge line. Therefore, when lifting, the pump pumps slightly less oil into the system. When lowering, the pump acts like a hydraulic motor

and allows excess oil to flow into the tank, allowing the cylinder to drain oil more quickly. As the load increases, there is also an increase in oscillation caused by the vibration of the ropes and the flexibility of the hydraulic hoses. This is much more evident in car speed graphs. This is

Fig. 5 Exemplary duty cycle with a payload of 419 kg: **a** motor/pump velocity; **b** piston rod velocity; **c** car velocity; **d** piston rod and car positions, **e** piston rod and car accelerations



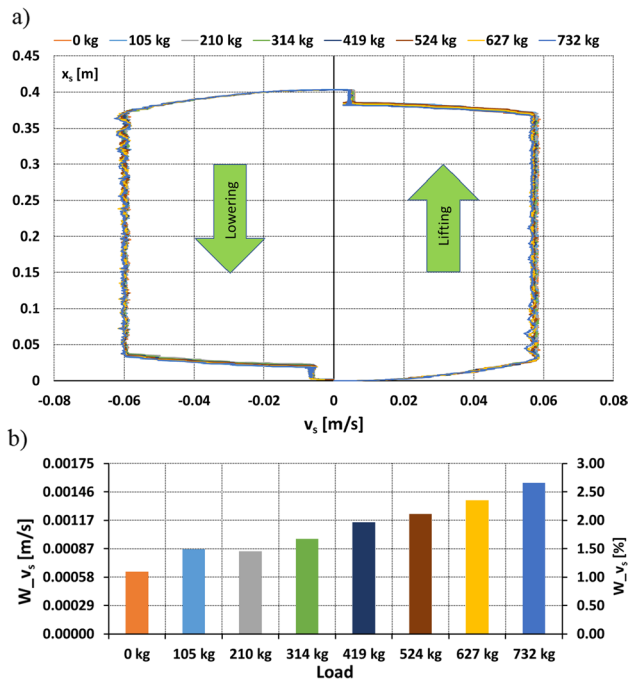


Fig. 6 The piston rod velocities for different payloads: **a** velocity diagram; **b** mapping factor

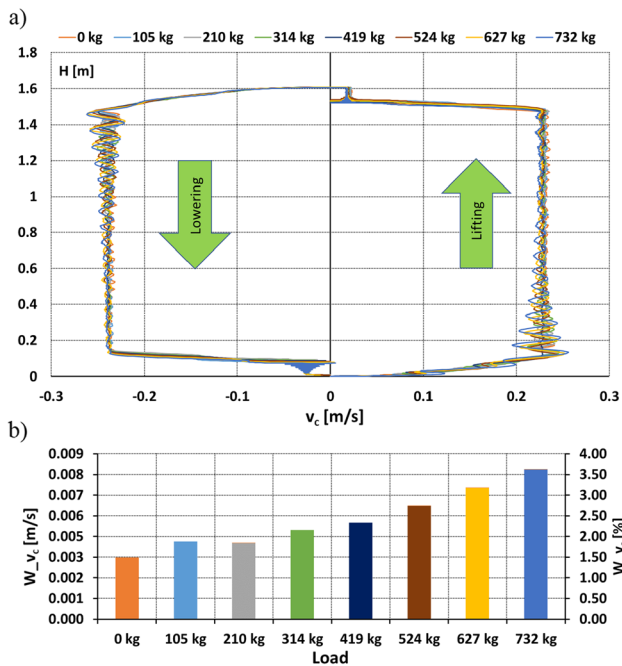


Fig. 7 The car velocities for different car payloads: **a** velocity diagram; **b** mapping factor

due to the direct attachment of the car to the rope, which behaves like a spring during unstable movements. In the system used, it is only one rope, but in a real lift, several

strands are used (at least 4). In addition, vibration dampers are used. This should significantly eliminate vibrations in the car.

The most important aim in this study was the fact that the car was positioned at a given height. As can be seen in Fig. 7a, the car reaches the set position of 1.6 m regardless of the load. This action is repeatable, which proves the correct operation of the proposed drive and control system.

To look closely at the velocity deviations of the elevator's drive elements, appropriate factors were derived. The changes in this factor as a function of payloads are presented in Fig. 6b and Figure 7b, respectively. The velocity mapping factors W_{v_s} and W_{v_c} have been introduced and calculated according to the following equations:

$$W_{v_{s(c)}} = \frac{\sum_{i=1}^n |v_{sa(ca)} - v_{s(c)}|}{v_{ss(cs)}} \cdot 100[\%] W_{v_{s(c)}} = \frac{\sum_{i=1}^n |v_{sa(ca)} - v_{s(c)}|}{k} [m/s],$$

where $v_{sa(ca)}$, $v_{s(c)}$, $v_{ss(cs)}$ are the piston rod (car) assumed, current, and transportation velocities, respectively, and k is the number of samples. As can be seen, the accuracy of speed mapping depends on load, but it is at the high level of about 1–2.5% of the inaccuracy of the piston rod speed concerning the assumed one.

In the case of the car speed mapping (Fig. 7b), high reproducibility of the speed can be seen; however, additional oscillations are visible related to the rope system with a single rope, which has a certain stiffness. The speed mapping factors depending on payload mass are at the level of about 1.27–3.5% of the inaccuracy of the car speed concerning the assumed one.

An interesting user-side issue of car and piston rod accelerations was also investigated. The results are presented in Fig. 8. The maximum acceleration of the car during lifting and lowering is ac_{max} and ac_{min} , respectively. None of them exceeds 1 m/s^2 , regardless of the car's load and the direction of the elevator movement. As can be seen, a constant level of acceleration is maintained regardless of the load. In the case of cylinder piston rod accelerations when the cabin is raised and lowered, these are max and min, respectively. In this case, the values are much lower due to the lower speeds achieved by the actuator. This is thanks to the use of a 4:1 rope system transmission.

The high quality of the piston rod and car positioning independent of the movement direction and the load is confirmed by the data shown in Table 5 and Fig. 9. The presented Δx_s and ΔH factors are calculated as follows:

$$\Delta x_s = x_{si} - \frac{\sum_{i=1}^k x_{si}}{k} [m] \Delta H_i = x_{si} - \frac{\sum_{i=1}^k x_{si} H_i}{k} [m],$$

where k is the number of samples, and x_{si} and H_i are the end positions of the piston rod and the car in a sample,

Fig. 8 Piston rod and car maximum accelerations for different payloads

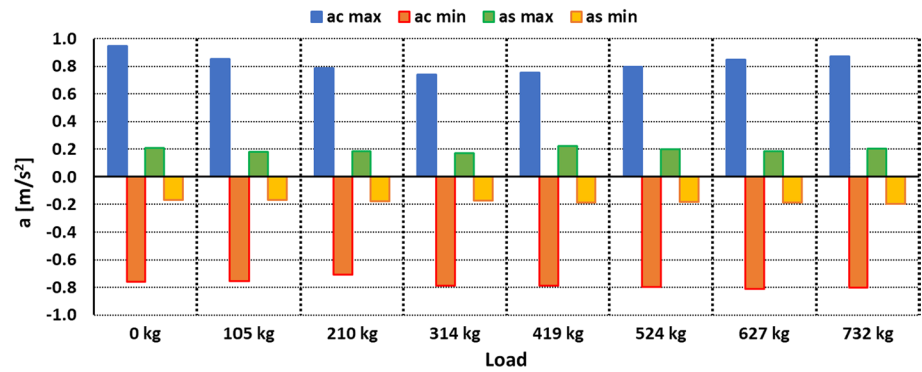
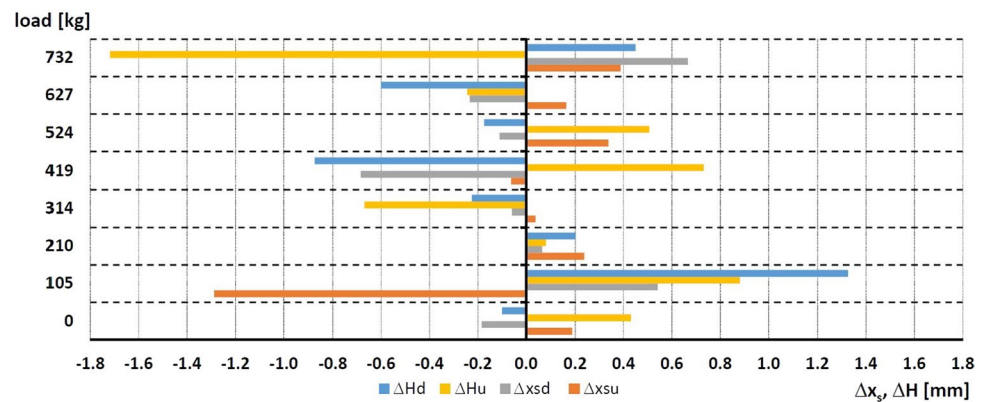


Table 5 The precision of car and piston rod positioning using limit switches

Payload [kg]	End position				Position difference			
	x_s [m]		H [m]		x_s [m]		H [m]	
	Lifting	Lowering	Lifting	Lowering	Δx_{su}	Δx_{sd}	ΔH_u	ΔH_d
0	0.4031	0.0016	1.6087	0.0084	0.0002	-0.0002	0.0004	-0.0001
105	0.4016	0.0023	1.6091	0.0098	-0.0013	0.0005	0.0009	0.0013
210	0.4032	0.0018	1.6083	0.0087	0.0002	0.0001	0.0001	0.0002
314	0.4030	0.0017	1.6076	0.0082	0.0000	-0.0001	-0.0007	-0.0002
419	0.4029	0.0011	1.6090	0.0076	-0.0001	-0.0007	0.0007	-0.0009
524	0.4033	0.0016	1.6088	0.0083	0.0003	-0.0001	0.0005	-0.0002
627	0.4031	0.0015	1.6080	0.0079	0.0002	-0.0002	-0.0002	-0.0006
732	0.4033	0.0024	1.6065	0.0089	0.0004	0.0007	-0.0017	0.0004

Fig. 9 The precision of car and piston rod positioning using limit switches



respectively. The indexes u and d used in Table 5 and Fig. 9 indicate lifting and lowering, respectively.

A high level of accuracy and repeatability of the piston rod and the car positioning was achieved. The deviation from the average position value at the end of the lifting and lowering cycle did not exceed 1.7 mm, regardless of the payload.

The high accuracy of positioning and repeatability of the indirect elevator's car is based on a simple quasi-open loop control system. Thanks to the continuous measurement of the position of the car, it is possible to stop the elevator in the assumed position. The control system will operate with

a similar effect also when typical limit switches are used. Thanks to deceleration to creep speed (150 rpm) and short car deceleration time (0.2 s), the stop is instantaneous when the set position value is detected. Under such conditions, the inertia of the system is strongly limited and the impact of the load on the stopping position is negligible.

The most common hydraulic and indirect elevators have much greater inaccuracies in the positioning of the cabin (up to 10 mm and more), which vary depending on the weight of the load being transported. In special solutions, when there is a need to increase the positioning accuracy of the

cabin, special positioning systems are used. Their task is to check whether the difference in position is greater than expected after the cabin reaches its destination. If so, the system restarts and raises or lowers the car at creep speed to the detent position. In the proposed (in this paper) drive system, this procedure takes place during a typical lift cycle.

5 Conclusion

The main goals of this research have been achieved. These are:

- a new hydraulic system of an indirect (hydraulic) elevator with a special control system was designed and implemented,
- the device is characterized by a simple structure and the main role is played by a fixed displacement pump driven by an electric motor fed by a frequency inverter,
- to control the motor speed, an original control system was used that allows the free shaping of various operational characteristics.

Experimental tests carried out on the presented stand confirmed good operational parameters of the drive such as:

- the same speed course was obtained using a simpler structure than classical systems based on a combination of valves operation,
- the usage of limit switches ensures the accuracy and repeatability of car positioning at a level of less than 1.7 mm,
- the use of some FI functions (vector control, overload limitation, etc.) increases the safety of the drive,
- setting appropriate starting ramps ensures smooth movement of the car and reduction of vibrations in the rope,
- the hydraulic system can significantly reduce oil heating by eliminating the throttle valves,
- the load while lowering generates potential energy that can be collected by the return energy module.

The proposed new solution provided safe operating conditions among others, thanks to the possibilities of free adjusting of the acceleration and deceleration times. The new system can be implemented in both new and modernized indirect rope elevators or hydraulic elevators. The proposed control system may be the beginning of further research on:

- reduction of vibrations of steel ropes through the developed system of time ramp adjustment in FI, depending on the transported cargo,
- the possibility of energy recovery to the power grid,

- developing the most effective work cycle to reduce energy consumption,
- the possibility of using industrial elevators in automated systems.

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Declarations

Conflict of interest The authors have no relevant financial or non-financial interests to disclose.

Research involving human participants and/or animals This research did not involve human participants and/or animals.

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