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RESISTANCES IN ELEVATOR TRANSMISSION SYSTEMS AND THE METHOD FOR THEIR DETERMINATION

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Abstract

The paper defines an experimental procedure for determiningthe efficiency coefficient of elevator facilities depending on the relative load of the cabin in operational conditions. Theexperiment was carried out on a passenger elevator with a rated load of 320 kg and a lifting height of 28 m. An analysis of the resistance in vertical lifting devices was performed, the most significant of which occur and are especially pronouncedin worm gear reducers and on the guiding system of the cabin and counterweight, where the type of guiding elements also has an influence.A particular problem is expressed due to the changing state of the contact surfaces (between the guide rails and the guide shoes) duringexploitation and the eccentric loading of the cabin. Also, the paper analysed the obtained measurement results to determine the dependenceof the efficiency coefficient of the facility concerning the relative load of the cabin.

Keywords*:electric elevators, guide rails and driving mechanism resistances, efficiency determination*

1 INTRODUCTION

Elevators are specific transport machines for verticallifting and inclined transport (most often cargo).However, elevator shafts and inclined elevators arerarely used, i.e., the corresponding mine elevatorsthat work in them. The main reason is that inclinedshafts are longer for the same height, so they aremore expensive to manufacture, and that inclinedshafts require special and more expensive devices forcargo transport [1]. The needs of modern society haveconditioned the development of different elevatorconstructions. Effective vertical mobility is crucial todeveloping and constructing high-rise buildings [2].

The primary goal of this paper is the developmentof a universal experimental method for determiningthe total resistances in electric elevators with wormgear reducers. The goal is that research and thedeveloped method serve to put into practice theassessment of the condition of the existing elevatorswhen periodic inspections are carried out. Also,research and results can serve as a basis for choosingthe optimal parameters of elevators during theirdesign.

Regarding the types of elevators from the pointof view of drive systems, elevators without reducersare usually used for nominal velocities over 2.5m/s. In contrast, reduction devices (gearboxes)must be applied for lower velocities. Spur gearswere occasionally used in the past, but with thedevelopment of design and manufacturing techniques,worm gears have become the accepted standard forelevators with gearboxes. In recent years, leading elevator manufacturers have put into production elevators with gearboxes that use double reduction with high-efficiency helical gears for nominal velocities up to 5 m/s [3]. The elevators are equipped with asynchronous AC motors, and the velocity is controlled by a frequency regulator. For example, in [4], the results of experimental research are given in which a comparison of energy consumption was made in the case of an elevator drive with and without a gearbox.

It should be noted that there is a very limited number of papers and studies in which authors are involved in determining the overall efficiency coefficient for electric elevators with worm reducers. Typically, research is based on the analysis and determination of losses in specific parts of elevator systems, such as guides, driving pulleys, deflecting pulleys, speed governors, reducers, etc.

In [5], power losses and their sources in the worm gear reducer are presented. These losses occur in the coupling of worm teeth and worm gear, in the bearings and seals, and in power losses during oil mixing. The total losses are determined for different values of input revolutions, output torque, and variations in oil types. Factors that significantly influence power losses include, above all, the type of material for gears and the geometry of the worm pair, the type and viscosity of lubricating oil, input revolutions, worm shape, load, temperature, etc. [6].

The drawback of a worm transmission is its relatively low efficiency, especially under extreme operating conditions, primarily related to high speeds. Significant sliding occurs between the worm and worm wheel, resulting in wear on the worm wheel and substantial power losses converted into heat [7] and [8]. The amount of energy transformed into heat is largely determined by the friction coefficient between the gear tooth faces.

In [9], the geometric characteristics of the worm gear reducer, the principle of operation and the theoretical basis for efficiency calculations were discussed. Based on conducted research and experiments, recommendations for improving the efficiency of the 2H-63 type worm transmission were developed.

In [10], self-locking properties have been implemented in a pair of worm gears that are connected to each other in an appropriate mesh pattern. The advantage lies in the simplicity of the system, whereby very high values of efficiencycan be achieved, while conventional systems have significantly lower efficiency.

2 ANALYSIS OF RESISTANCE IN TRANSMISSION SYSTEMS AND GUIDING ELEMENTS IN ELEVATORS

2.1. Analysis of Resistances

The analysis of the lifting system with a driving pulley is very complex [11]. In this paper, based on Fig. 1, expressions for the forces in the hoist ropes as a function of the efficiency factor are provided, allowing us to observe the influence and, consequently, the significance of determining real resistance values.

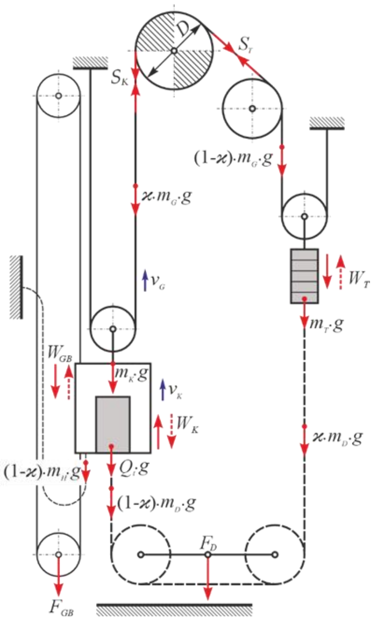

Fig. 1*Generalized schematic representation of the elevator system with drive mechanism located above the shaft, [12, 13]*

The impact of resistances is encompassed by the efficiency coefficient for the gearbox, cabin, and counterweight, as well as all deflection pulleys and sheaves.

In Fig. 1, a generalized schematic representation of the elevator system is provided. The transmission ratio of the elevator shaft, denoted as i_S (which can also be referred to as the suspension ratio), is given as the ratio of the velocity of the supporting rope (hoist rope) v_G to the velocity of the cabin v_K, i.e., $i_S = v_G/v_K$.

The following suspended loads are present in the elevator shaft: Q_t is the mass of the load in the cabin (payload), m_K the mass of the cabin, m_T the mass of the counterweight, m_G the mass of the hoist ropes, m_D the mass of the compensating rope or chain, m_H the mass of the travelling electrical cables, F_D the tensioning force of the compensating rope including the weight of the tensioning devices, and F_{GB} tensioning force of the speed governor.

The following equality applies between the mass of the cabin, the mass of the counterweight and the rated load:

$$m_T = m_K + \beta \cdot Q \tag{1}$$

where Q is rated load for which the elevator is designed, and β coefficient taking account of the percentage of the rated load balanced by the counterweight (counterweight weight factor).

The relationship between payload (current mass of cargo in the cabin) and rated load is:

$$Q_t = \lambda \cdot Q \tag{2}$$

where the λ represents the relative load (cabin load factor).

As shown in Fig. 1, the masses m_G, m_D, and m_H are multiplied by a height factor κ or $(1 - \kappa)$. When the cabin is at the bottom of the shaft, $\kappa = 1$ holds, while when it is at the top, $\kappa = 0$.

The general expressions for the forces in the hoist ropes on the cabin and the counterweight side in stationary operating modes, according to Fig. 1 and [11] and [13], are as follows:

$$S_{Kst} = \left\{ \left[\left(m_K + \lambda \cdot Q \right) \cdot \left(1 + s_v \cdot f_K \right) + \left(m_D + m_H \right) \cdot \left(1 - \kappa \right) \right] \cdot g + \frac{F_D \cdot \eta_{KU}^{1-s_v}}{1+\eta_{KU}^2} + s_v \cdot W_{GB} \right\} \cdot \frac{2}{i_S} \cdot \frac{\eta_{RU}^{\frac{1}{1-s_v}}}{1+\eta_{RU}} - m_G \cdot \kappa \cdot g \cdot \frac{1}{i_S} \tag{3}$$

$$S_{Tst} = \left\{ \left[m_T \cdot \left(1 - s_v \cdot f_T \right) + m_D \cdot \kappa \right] \cdot g + \frac{F_D \cdot \eta_{KU}^{1+s_v}}{1+\eta_{KU}^2} \right\} \cdot \frac{2}{i_S} \cdot \frac{\eta_{RU}^{\frac{1}{1+s_v}}}{1+\eta_{RU}} \cdot \eta_{OU}^{s_v} + m_G \cdot (1-\kappa) \cdot g \cdot \frac{1}{i_S} \cdot \eta_{OU}^{s_v} \tag{4}$$

where S_{Kst} is the stationary force in the hoist ropes on the cabin side, S_{Tst} is the stationary force in the hoist ropes on the counterweight side, s_v index of cabin movement direction, f_K, f_T resistance coefficients of the cabin and the counterweight, η_{OU} efficiency of the deflection pulley, η_{RU}efficiency of the sheaves, and η_{KU} efficiency of the compensating ropes deflection pulley.

In the characteristic phases of the so-called motor operation mode ($Q_{1\text{-}2}$), the values for λ and s_v in Eqs.(3) and (4) are taken as follows: $\lambda = 1$ and $s_v = 1$, for lifting a fully loaded cabin from the lowest station, and $\lambda = 0$ and $s_v = -1$, for lowering an empty cabin from the highest station.

In the characteristic phases of the so-called generator operation mode ($Q_{2\text{-}1}$), the values for λ and s_v in Eq. (3) and Eq. (4) are taken as follows: $\lambda = 0$ and $s_v = 1$, for lifting an empty cabin to the highest station, and $\lambda = 1$ and $s_v = -1$, for lowering a fully loaded cabin to the lowest station.

Based on the forces in the hoist ropes on the cabin and the counterweight side, Eqs. (3) and (4), a general expression for the stationary torque reduced to the motor shaft can be obtained as:

$$M_{Mst} = \left(S_{Kst} - S_{Tst} \right) \cdot \frac{D}{2} \cdot \frac{1}{i_R} \cdot \frac{\left(\eta_{PU} \cdot \eta'_R \right)^{\frac{1-s_v \cdot s_t}{2}}}{\left(\eta_{PU} \cdot \eta_R \right)^{\frac{1+s_v \cdot s_t}{2}}} \tag{5}$$

where s_t is an index of the heavier side ($s_t = 1$ when the cabin side is heavier, $s_t = -1$ when the counterweight side is heavier), i_R transmission ratio of the gearbox, η_{PU} efficiency of the driving pulley, and η_R, η'_R are efficiency of the gearbox during power flow $Q_{1\text{-}2}$, and $Q_{2\text{-}1}$, respectively.

This chapter presents the basic components and elements of the elevator that are the subject of this research, and it will analyse the resistances and the causes leading to their appearance. The most significant resistances occur in the transmission system (gearbox), which has particularly high values in worm gearboxes and on the cabin and counterweight guide rails.

Considerations in [14] indicate that individual losses in lifting systems (driving and deflection pulleys, bearings, guide rails, ropes, etc.) should be more precisely determined for various payloads (besides rated load) for all operating conditions (from acceleration to reaching nominal speed, deceleration (braking), stopping, etc., both ways).

The resistance of the elevator guidance system depends on the applied solution. Sliding shoes or roller guides are most often used to guide the cabin, Fig. 2. Due to the requirements for the straightness of the vertically placed guide rails, their length is limited during production, which means that the elevator cabin will encounter many joints of the guide rails on its path. Those joints are additional sources of resistance [15]. In addition to the guidance system type, the resistance also depends on the method and condition of lubrication of the contact surfaces, assembly, or regulation of the pre-tightening between the sliding shoe and the guide rails. A particularproblem when defining the resistance occurs due to the changing state of the contact surfaces during the exploitation (wear, damage, and influence of the external environment) and the eccentric loading of the cabin (position of the centre of gravity of the load to the cabin suspension point).

Regarding the sliding shoes and their surface contact with the guide rails, highly non-linear hysteretic friction is shown, significantly hindering the transport quality. In [16], an experimental investigation of this type of phenomenon was carried out by measuring the force of contact friction between the sliding guide shoe and guide rail surfaces at different combinations of input parameters.

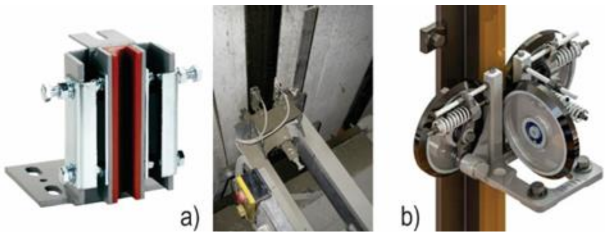

Fig. 2*Presentation of the cabin and counterweight guidance system: a) sliding guide shoes and the real state of the elevator,b) roller guides*

Typically, losses are calculated through efficiency coefficients of transmission. In the literature, they are usually estimated globally based on [11]:

- For elevator worm gear reducer: $\eta_R \approx 0.45$ to 0.80, where higher values of efficiency coefficient correspond to better quality of worm and worm wheel surfacing, smaller transmission ratios, and consequently, higher numbers of worm start. Also, higher values of efficiency coefficient are in the case of rolling bearings, greater forces, and higher worm rotations.
- Pulley losses: $\eta_U \approx 0.96$ to 0.98, where higher values correspond to rolling bearings, ropes with a greater number of strands, thinner wires, and non-metallic rope cores.
- Losses in the elevator shaft, i.e., the cabin and counterweight guide rails (calculated through the overall transmission coefficient of efficiency in the elevator shaft), typically in the range of $\eta_{VO} = 0.70$ to 0.80. Lower values correspond to sliding guide shoes, the presence of compensating ropes or chains, their deflection pulleys or sprockets, and so on.

In addition to the above, the following is noteworthy:

- the accuracy of positioning of cabin and counterweight guide rails in the elevator shaft;
- the variable position of the passenger or freight centre of gravity within the elevator cabin, resulting in varying loads on the guide rails and different resistances.

Resistance on the cabin guide rails due to eccentric loading can be determined based on Fig. 3.

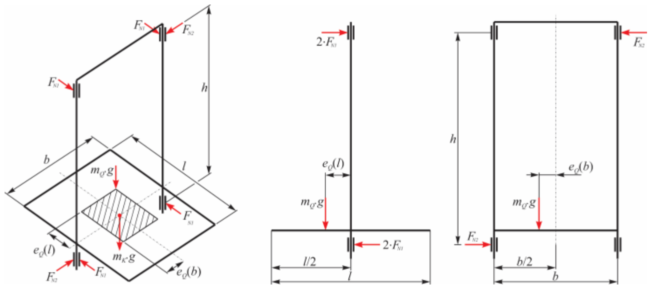

Fig. 3*Frictional forces on cabin guide shoes*

The total resistances that occur with eccentric loading of the cabin are:

$$\sum F_{tr} = 4 \cdot F_{tr1} + 2 \cdot F_{tr2} = 2 \cdot m_Q \cdot g \cdot \frac{e_{Q(l)} + e_{Q(b)}}{h} \cdot \mu_{kl} \quad (6)$$

where:

$F_{tr1} = F_{N1} \cdot \mu_{kl}$ is frictional force on the cabin sliding guide shoe due to the eccentricity of the load position$e_{Q(l)}$,

$F_{tr2} = F_{N2} \cdot \mu_{kl}$ is frictional force on the cabin sliding guide shoe due to the eccentricity of the load position $e_{Q(b)}$,

$e_{Q(l)}$, $e_{Q(b)}$are the largest eccentricity of the load position across the cabin length and width,

$h$is spacing between sliding guide shoes in the vertical direction,

$\mu_{kl}$is guide shoe friction coefficient,

$F_{N1} = m_Q \cdot \frac{e_{Q(l)}}{2 \cdot h}$is the normal force on the sliding guide shoe caused by eccentricity $e_{Q(l)}$,

$F_{N2} = m_Q \cdot \frac{e_{Q(b)}}{2 \cdot h}$is the normal force on the sliding guide shoe caused by eccentricity $e_{Q(b)}$,

Based on the expressions for the frictional force on the guide rails (Eq. (6)), it can be observed that theresistance values can vary from zero (no eccentricity) to some maximum value, which depends on the ratio of the cabin's width to the spacing of the guiding elements on the cabin. It should be noted that these resistances can change during transport, especially in the case of passenger elevators, which further highlights the complexity of determining the total resistance of elevator systems.

2.2. Elevator Efficiency Coefficient

By introducing the term "stationary shaft efficiency coefficient" η_S [17], which would represent the ratio of the load torque M_Q to the torque M_Q+M_S with which the elevator mechanism acts on the drive unit, its value can be written as:

$$\eta_S = \frac{M_Q}{M_Q + M_S} = \frac{1}{1 + \dfrac{M_S}{M_Q}} \tag{7}$$

where $M_Q$represents load torque and $M_S$torque lossin the elevator shaft.
Similarly, the “stationary gearbox efficiency coefficient” η_G can be introduced [17], which at motor operation mode represents the ratio of the torque at the worm wheel to the torque at the worm shaft in the stationary state, i.e., when $d\omega_1/dt = 0$:

$$\eta_G = \frac{M_Q + M_S}{M_Q + M_S + M_G} = \frac{1}{1 + \dfrac{M_G}{M_Q + M_S}} \tag{8}$$

where $M_G$represents torque loss in the transmission.
The overall efficiency is the product of η_S and η_G, that is, the ratio of the torques M_Q and M_M for the motor operation mode:

$$\begin{aligned} \eta = \eta_S \eta_G &= \frac{M_Q}{M_Q + M_S} \cdot \frac{M_Q + M_S}{M_Q + M_S + M_G} = \\ &= \frac{M_Q}{M_Q + M_S + M_G} = \frac{1}{1 + \dfrac{M_S + M_G}{M_Q}} = \frac{M_Q}{M_M} \end{aligned} \tag{9}$$

where $M_M$represents motor torque.
Similarly, the overall efficiency of the generator operation mode is obtained in the same way, and it is given by:

$$\eta' = \frac{M_M}{M_Q} \tag{10}$$

The relative load of the cabin has a significant impact on the efficiency of each elevator system. The efficiency is highest at nominal load and empty cabin, while at half the rated load of the cabin, the value of η drops and may even reach zero, as shown in Fig. 4.
Based on Eqs. (9), and (10), a common (universal) expression can be found that represents the relationship between the transmission efficiency in the generator and motor mode. From Eq. (9), the following equality can be written:

$$1 + \frac{M_S + M_G}{M_Q} = \frac{1}{\eta} \quad \rightarrow \quad \frac{M_S + M_G}{M_Q} = \frac{1}{\eta} - 1 \tag{11}$$

Eq. (10) for generator operation mode gives:

$$\begin{aligned} \eta' &= \frac{M_Q - M_S - M_G}{M_Q} = 1 - \frac{M_S + M_G}{M_Q} \\ &\rightarrow \quad \frac{M_S + M_G}{M_Q} = 1 - \eta' \end{aligned} \tag{12}$$

From Eqs. (11) and (12) follows:

$$\frac{1}{\eta} - 1 = 1 - \eta' \quad \rightarrow \quad \eta' = 2 - \frac{1}{\eta} \tag{13}$$

The last dependency holds true only if the relative load of the cabin (λ) and the vertical position of the cabin in the shaft are the same for motor and generator operation mode while the direction of motion is different, i.e., when all transmission losses are independent of the load.

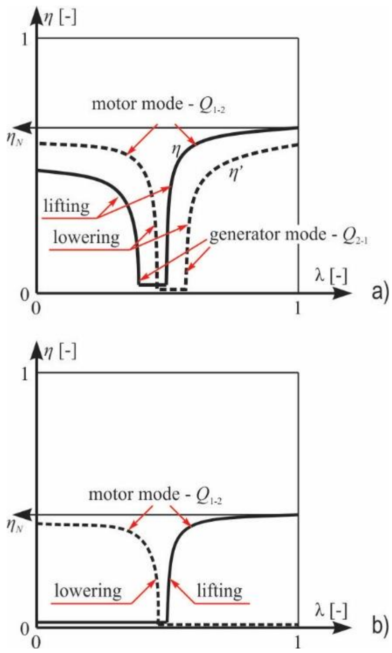

Fig. 4*Principled form of dependence $\eta=f(\lambda)$ with the direction of elevator cabin movement and the direction of power flow through the drive mechanism, as parameters: a) transmission that is not self-locking under nominal load, b) self-locking transmission even under nominal load [11]*

In Fig. 4, it can be observed that the trend of efficiency for upward and downward movement is not identical. Specifically, it cannot be confirmed the commonly used assumption that with a 50 % load compensation ($\beta = 0.5$), the torque resistance of the system at the drive motor shaft is the same whenthe cabin is fully loaded moving upward and when an empty cabin descends downward. In the case of an empty cabin, the losses in the elevator shaft are significantly lower than when fully loaded. Hence, the efficiency of the entire system will be more favourable.
A special problem arises in the case of self-locking transmission (most often worm gear). Generally, for pulleys, gear pairs, and transmissions with high η, there is no difference in the efficiency coefficient depending on the direction of the power flow. At the same time, this must be considered for the worm gear transmission.
Based on the above, it can be observed that there is significant complexity associated with determining losses or the efficiency of vertical lifting systemswith a driving pulley, leading to the need for elevator manufacturers to experimentally determine the values and the character of changes in the efficiency of all transmission elements, especially the worm gear reducer and elements for guiding the cabin and counterweight, or to foresee appropriate experimental measurements on installed facilities in operation. The following section defines an experimental method for determining the magnitude of resistance or the overall efficiency coefficient of elevator facilities.

3 EXPERIMENTAL METHOD FOR DETERMININGRESISTANCE IN ELEVATORS

Determining the resistance in elevators is based on the consequence of frictional resistance in the cabin and counterweight guides, as well as in the drive mechanism. In this manner, the total (equivalent) movement resistance is determined using specially designed measuring equipment for various cabin loads. The measurement procedure involves measuring the resistance magnitude by lifting the cabin to the desired height (position) via the driving electric motor. Subsequently, manual disengagement of the drive is performed (separation of brake shoes using a lever) in the machine room while the cabin is held at the same height via the flywheel. Using a portable electric pipe threader connected to the flywheel via a torque transducer, the elevator cabin is lifted/lowered at a slow speed to a specific height (ten or more revolutions of the flywheel), after which the brake is reactivated. In this manner, the torque transducer measures the value of the total resistance for the current position and load of the cabin.

The dimensions of the measuring equipment and especially the elements for connection with the flywheel or the shaft must correspond to the characteristics of the drive mechanism for the specific elevator. It is necessary to foresee an appropriate method of loading the cabin. Various types of loads can be used to allow variation in the size and eccentricity of the cabin load.

The results obtained through this experiment are used in the field of elevator design, reconstruction, and maintenance. For the computational analysis of elevator driving characteristics, especially concerning the “driving comfort” in passenger elevators, knowing the real resistance values is necessary, as their influence (η_{uk} = 0.45 to 0.80) is of the same order of magnitude as the influence of the rated load of the elevator (Q), and often greater in elevators in operation when the relative cabin load is $\lambda \approx 0.5$.

Thus far, in practice, the results of laboratory tests of resistance, i.e., the efficiency coefficient, which is carried out separately, are most often used, e.g., for part of the drive mechanism in the laboratories of transmission manufacturers and for the elevator shaft with models in the laboratories of elevator manufacturers or specialized institutions.

Fig. 5*Driving machine of passenger elevator*

The experimental method presented in this paper includes measurements performed on a passenger elevator with a rated load of 320 kg, a maximum projected lifting velocity of 1.2 m/s, and a lifting height of 28 m, whose drive mechanism (worm gear reducer coupled with an asynchronous electric motor) is shown in Fig. 5. Other significant characteristics of this elevator can be seen in Table 1. It should be noted that it is a reconstructed elevator whose rated load is reduced compared to the designed load (450 kg). Additionally, mechanical weighing deviceswere installed during the reconstruction to limit the load, and the weights of the cabin and counterweight were modified. Since, in most cases, the elevator is used to transport one person, in order to save energy, elevator technicians have reduced the value of the counterweight weight factor (β) compared to usual values.

Table 1 Elevator technical characteristics

Driving electric motor	Power: 7.5 kW Number of revolutions: 910/210 rpm Nominal torque: 70.2 Nm
Cabin	m_K = 740 kg
Counterweight	m_T = 775 kg
Hoist ropes	z = 4 pieces d = 13 mm (114 wires per cross-section) Unit mass: 0.56 kg/m
Compensation chain	Type: welded chain (with links through which a hemp rope is spliced) z = 1 piece Unit mass: 2.25 kg/m.
Number of floors/entrances:	10/10 lifting height: 28 m.
Diameter of the drive pulley:	D = 650 mm
Cabin dimensions	1030 mm x1420 mm x2200 mm
Cabin guide rails:	⊥ 65 mm x90 mm x14 mm
Counterweight guide rails:	⊥ 50 mm x50 mm x6 mm

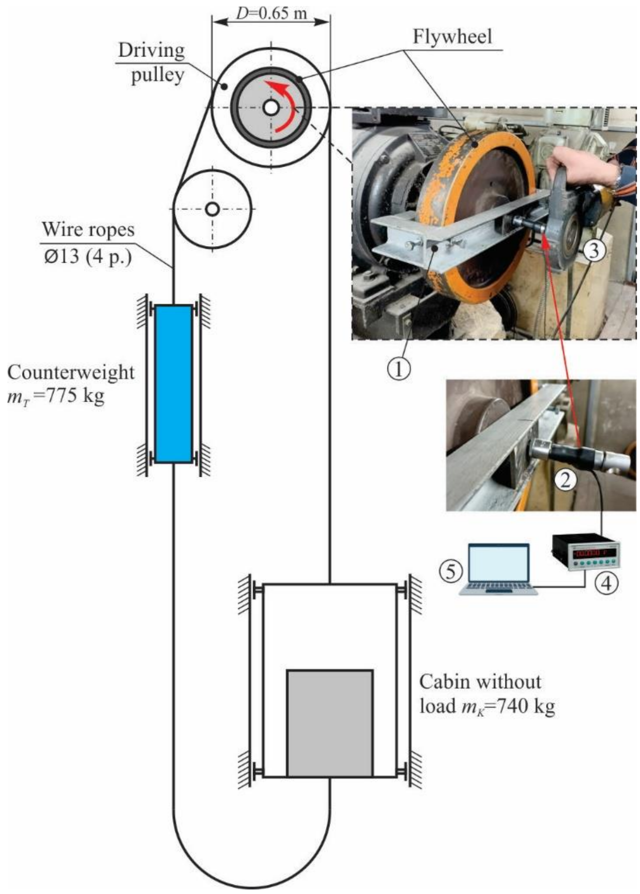

Fig. 6*General arrangement of measuring equipment*

3.1. Measuring Equipment and the Method for Measuring Data Acquisition

For the "stationary" measurements and experimental determination of the values and character of the change in efficiency, or losses, primarily of the worm gear reducer and cabin and counterweight guiding elements, the following equipment was used:

- Measuring lever, position 1 (Fig. 6),
- Rotating torque transducer, measuring range up to 200 Nm, position 2 (Fig. 6),
- Portable electric pipe threader, used for continuous rotation of the flywheel, and thus lifting and lowering of the cabin/counterweight, position 3 (Fig. 6),
- Measuring amplifier HBM MVD2555, accuracy class 0.1, position 4 (Fig. 6),
- Software for acquisition and processing of measuring signals, HBM catmanEasy-AP,
- Laptop for storing the measuring signals (torques) collected from the flywheel of the electric motor and from the force measurement device in the suspension ropes, position 5 (Fig. 6),

The measuring lever is designed and fabricated to allow a simple and sturdy connection with the flywheel of the electric motor and to enable easy and quick attachment of the torque measurement device at its midpoint.

Fig. 7shows the measuring lever with the torque measurement device and the driving element (portable electric pipe threader) during measurement, mounted on the flywheel of the electric motor.

After manual disengagement of the brake lever, the portable electric pipe threader (Fig. 7) is activated, whose torque is transmitted to the flywheel through the torque transducer (position 2, Fig. 6), thereby lifting or lowering the cabin. The measurement signal (change in torque on the flywheel) from the torque transducer is directed to the measurement amplifier HBM MVD2555, and from there, through the data acquisition and processing software HBM catmanEasy-AP, stored in the computer memory.

Fig. 7*Measuring lever mounted on the flywheel of the electric motor*

It should be noted that the measurements were conducted when the cabin was approximately at the midpoint of the lifting height. The cabin was lifted via an electric pipe threader for more than ten revolutions of the flywheel (shaft) of the drive motor and then lowered also for several revolutions of the flywheel.

3.2. Protocols and Results of the Experiment

This experiment and the measurement results demonstrate that determining the resistance in elevators can be conducted on the drive machine in the elevator machine room. This way, it is possible to determine the total (equivalent) movement resistance using specially designed measuring equipment for various loads and cabin height positions. The experiment enables direct measurement of resistance in cases where the load torque is greater than the friction torque, as well as measurement of resistance for the reverse relationship of these torques for the case of the so-called "self-locking" drive.

The results will be shown for cases of lifting the cabin from the midpoint of the lifting height (middle level) for more than 10 revolutions of the drive motor shaft (flywheel), then lowering for twice the number of revolutions of the motor shaft, and then lifting again and returning to the initial position. Upon stopping, the brake lever was simultaneously disengaged. The experiment was repeated for both the empty cabin and the cabin loaded with weights of 80 kg, 160 kg, 240 kg, and rated load of 320 kg.

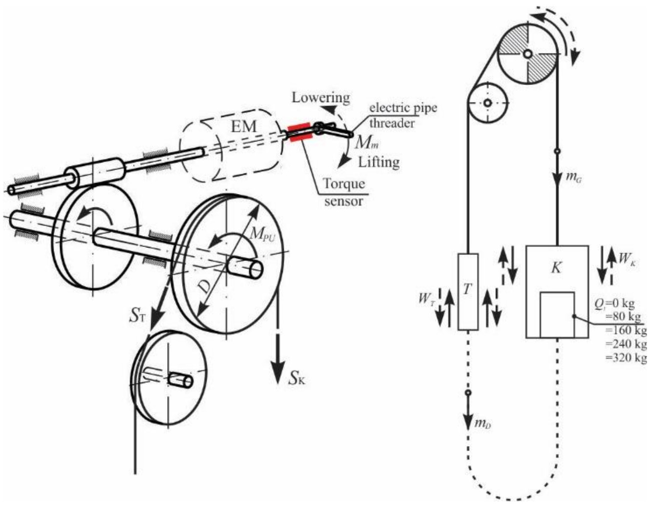

Fig. 8*Passenger elevator parameters relevant for determining total resistances*

As a result of the conducted experimental tests, changes in torque on the shaft (flywheel) of the elevator's drive motor are presented in the following figure.

From the diagrams, it can be observed that when engaging and disengaging the electromotor-driven threading machine (startof movement), there is a sudden peak in the torque value registered on the measuring device (torque transducer), which can be attributed to dynamic influences, i.e., acceleration/deceleration of translational and rotational masses of the elevator system. Under "stationary" conditions, a relatively steady total torque value is reached, which is the result of the sum of lifting "relative" cabin load and total resistances reduced to the motor shaft.

The parts of the diagram related to the elevator's stationary operating regime are used to determine the average values of the total elevator efficiency, as shown in Chapter 3.3.

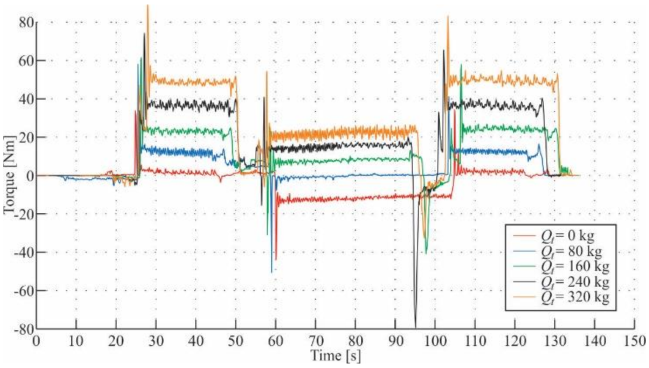

Fig. 9*Measurement of the total torque on the drive motor shaft during the lowering/lifting of the elevator cabinat the middle level for various loads*

Table 2 gives the averaged values of the measured torque on the drive motor shaft, obtained from the diagram shown in Fig. 9 for the steady-state operation, i.e., lifting and lowering the cabin. The measured torque values are shown for five different cabin loads and three different cabin positions.

Table 2 Average values of the measured torque on the drive motor shaft

	M_m [Nm]	
Q_t [kg]	Lifting	Lowering
0	1.725	-11.690
80	12.420	-0.303
160	23.705	7.910
240	36.570	15.150
320	49.035	21.930

3.3. Determination of the Elevator Overall Efficiency Based on the Measurement Results

Based on the measurement results depicted in Fig. 9 and Table 2, it is possible to determine the overall efficiencies of the elevator for different cases of motion and various loads of the cabin in stationary operating modes of the elevator. The theoretical basis for determining the overall efficiencies is provided in Chapter 2.2. The measured values of the torque on the shaft of the drive motor (given in Table 2) for the motor operation mode (Q_{1-2} direction of power flow) represent the sum of the moments M_Q+M_S+M_G. Determining the overall efficiency of the elevator based on the developed experimental method is done according to Eq. (9):

$$\eta = \frac{M_Q}{M_M} \tag{14}$$

The values of the overall efficiency for the generator operation mode, i.e., for Q_{2-1}, are obtained from Eq. (10):

$$\eta' = \frac{M_M}{M_Q} \tag{15}$$

Due to the equal weights per meter of the supporting ropes and compensating chain of the subject elevator (m_G = m_D, Table 1), and due to the 1:1 suspension system of the cabin and counterweight (i_S = 1), equation for determining the load torque (load torque reduced to the shaft of the drive motor) is (m_H can be neglected due to its small value):

$$M_Q = \frac{D}{2}\cdot\frac{1}{i_R}\cdot Q\cdot g\cdot(\lambda-\beta) = \frac{(\lambda-\beta)\cdot Q\cdot g\cdot D}{2\cdot i_R} \tag{16}$$

where D = 650 mm is the diameter of the driving pulley (Table 1), and i_R = 27.7 is the transmission ratio of the reducer.
The counterweight weight factor and the relative load of the cabin (cabin load factor) are determined based on Eqs. (1) and (2).
The calculated values of the overall elevator efficiency coefficients are given in Table 3.

Table 3 Overall elevator efficiency coefficients

Q_t [kg]	λ [-]	M_Q [Nm]	η [-]			
			Q_{1-2}		Q_{2-1}	
			Lift-ing	Lower-ing	Lift-ing	Lower-ing
0	0.00	-4.03	-	0.34	0.43	-
80	0.25	5.18	0.42	-	-	0.06
160	0.50	14.39	0.61	-	-	0.55
240	0.75	23.60	0.65	-	-	0.64
320	1.00	32.80	0.67	-	-	0.67

The measurement results with calculated values of the total efficiency (η), presented in Table 3 can also be depicted graphically in Fig 10. Based on the table and diagram (in addition to numerous values), it can be concluded that the change in the total efficiency of the elevator depending on the cabin load factor is not linear and corresponds to the theoretical considerations provided in the first part of the paper. By increasing the relative cabin load, i.e. by moving the value of the cabin load factor away from the counterweight weight factor, the value of the overall efficiency coefficients of the elevator also increases.

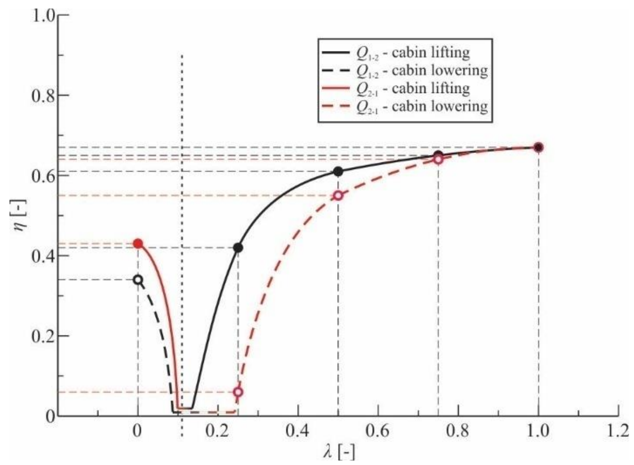

Fig. 10*The change in the total efficiency of the elevatorfor different operating modes*

4 CONCLUSIONS

The measurements and method presented in the paper are based on determining the resistance in elevators due to friction in the cabin, counterweight guide rails, and part of the drive mechanism. In this way, the total (equivalent) movement resistance is determined using a specially-made measuring tool for different cabin loads. The experiment represents a unique method for determining total losses by measuring the torque on the shaft (flywheel) of the drive motor.

This manuscript specifically addresses a method for determining total losses in electric elevators with a worm reducer. Such systems involve two power flows–from the drive motor to the driving pulley and vice versa (depending on whether the motor operates in motor or generator mode). A particular issue arises in the case of self-locking transmission.

This research and the unique method can serve in the future to introduce an assessment of the condition of existing elevators in practice through measurements, for example, during periodic inspections. Based on the large amount of data obtained from these measurements, it is possible to establish a basis for improving the conditions for designing more efficient elevator systems in terms of optimal parameter selection, energy savings, maintenance, etc.

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