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Analysis of Power Losses and Experimental Method for Determining Resistance in Electric Elevators

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To be able to carry out a high-quality analysis of the operation and design of elevator systems with a driving pulley, it is necessary to determine the size of the losses that occur in the drive mechanism and elevator guide rails. The paper defines an experimental procedure for determining the losses and the efficiency coefficient of elevator facilities depending on the relative load of the cabin in operational conditions. The experiment 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, i.e., the source of power losses in vertical lifting devices, was performed, the most significant of which occur and are especially pronounced in 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) during exploitation and the eccentric loading of the cabin. Also, the paper analysed the obtained measurement results to determine the dependence of the efficiency coefficient of the facility concerning the relative load of the cabin.
Keywords: electric elevators, guide rails and driving mechanism resistances, efficiency determination

Highlights
- *The complexity of designing and analysing elevator systems with driving pulley is manifested, among other things, in the occurrence of a large number of loss sources and the problem of determining their values.*
- *Resistance values in elevators are particularly significant when eccentric cabin loading occurs and because they change during elevator operation, specifically changes in the vertical position of the cabin and counterweight.*
- *The developed experimental method represents a unique procedure for determining total losses by measuring the torque on the drive motor shaft.*
- *The specificity of the developed experimental method for determining total losses in electric elevators with worm gear reducers lies in the fact that a special problem arises in the case of self-locking transmission.*

0 INTRODUCTION

Elevators are specific transport machines for vertical lifting and inclined transport (most often cargo). However, elevator shafts and inclined elevators are rarely used, i.e., the corresponding mine elevators that work in them. The main reason is that inclined shafts are longer for the same height, so they are more expensive to manufacture, and that inclined shafts require special and more expensive devices for cargo transport [1]. The needs of modern society have conditioned the development of different elevator constructions. Effective vertical mobility is crucial to developing and constructing high-rise buildings [2]. For buildings with many floors are being built, where elevators are necessary to transport passengers to a great height. There are many sources of power loss in vertical lifting devices. Fig. 1 shows some kinematic solutions of electric elevators; arrows show where the most considerable losses occur. Friction occurs in all places where there is relative movement of the contact surfaces of the elevator elements [3]. If, in addition to different kinematic solutions, the application of different transmissions as part of the drive mechanism is taken into account, it can be stated that for each specific case, it is necessary to perform an analysis and definition of the total resistance or the efficiency coefficient of the facility.

The primary goal of this paper is the development of a universal experimental method for determining the total resistances in electric elevators with worm gear reducers. The goal is that research and the developed method serve to put into practice the assessment of the condition of the existing elevators when periodic inspections are carried out. Also, research and results can serve as a basis for choosing the optimal parameters of elevators during their design.

Regarding the types of elevators from the point of view of drive systems, elevators without reducers are usually used for nominal velocities over 2.5 m/s. In contrast, reduction devices (gearboxes) must be applied for lower velocities. Spur gears were occasionally used in the past, but with the development of design and manufacturing techniques, worm gears have become the accepted standard for

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elevators 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 [4]. The elevators are equipped with asynchronous AC motors, and the velocity is controlled by a frequency regulator. For example, in [5], 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.

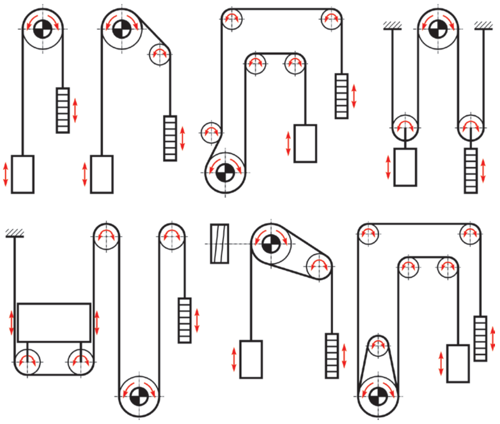


Fig. 1. *Places where friction occurs for different kinematic solutions in elevators*

Depending on the applied kinematic solution in electric elevators (Fig. 1), where different total resistance values occur, there are techniques for the previous selection of the drive mechanism with a gearbox. In [6], the author specifies the conditions that the torque characteristic of a potential drive motor must meet. Among other requirements, it is stated that there should be a guaranteed excess of the locked-rotor torque over the maximal load resistance moment at rest. Additionally, the presence of an excess of the motor's maximal torque over the maximal torque required from the motor is emphasized. Also, the author lists the steps the designer must follow when choosing and gives general terms for different concepts (overhead machine, basement machine room, elevators without a machine room, etc.).

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 [7], 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. [8].

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 [9] and [10]. The amount of energy transformed into heat is largely determined by the friction coefficient between the gear tooth faces.

In [11], 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 [12], power losses and their sources in worm gear reducers are presented. Considering that the operation of a worm transmission involves the line contact of coupled elements accompanied by significant sliding, the power losses in the coupling of the worm and worm wheel have the highest value. Formulas are provided for calculating individual power losses and the efficiency of the transmission. Furthermore, the issues of worm gear reducers were addressed in [13], in which models with concave-shaped contact surfaces of the worm and convex-shaped contact surfaces of the worm wheel teeth were analysed. The key characteristics of the proposed solution for the worm gear reducer lie in the fact that, due to the concave-convex contact, the entire side surfaces of the worm coil and the worm wheel teeth are involved in power transmission. Therefore, improved power transmission and lubrication properties can be expected, resulting in lower energy losses and less wear and tear.

In [14], 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 efficiency can be achieved, while conventional systems have significantly lower efficiency.

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

The analysis of the lifting system with a driving pulley is very complex [15]. In this paper, based on Fig. 2, 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.

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. 2, 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. 2, 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, $\kappa = 0$.

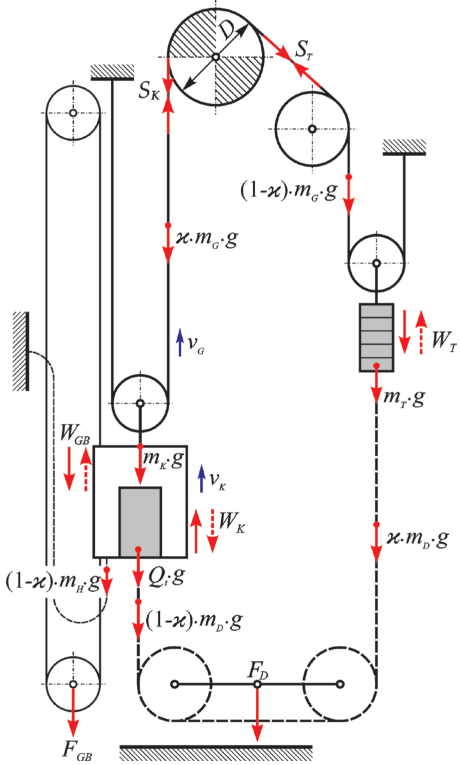


Fig. 2. *Generalized schematic representation of the elevator system with drive mechanism located above the shaft,* **[16]** *and* **[17]**

The general expressions for the forces in the hoist ropes on the cabin and the counterweight side in stationary operating modes, according to Fig. 2 and [15] and [17], are as follows:

$$S_{Kst} = \Big\{\Big[\big(m_K + \lambda \cdot Q\big)\cdot\big(1 + s_v \cdot f_K\big) + \big(m_D + m_H\big)\cdot\big(1-\kappa\big)\Big]\cdot g$$

$$+\frac{F_D \cdot \eta_{KU}^{1-s_v}}{1+\eta_{KU}^2} + s_v \cdot W_{GB}\Big\}\cdot\frac{2}{i_S}\cdot\frac{\eta_{RU}^{\frac{1}{1-s_v}}}{1+\eta_{RU}} - m_G\cdot\kappa\cdot g\cdot\frac{1}{i_S}, \tag{3}$$

$$S_{Tst} = \left\{\Big[m_T\cdot\big(1 - s_v \cdot f_T\big) + m_D\cdot\kappa\Big]\cdot g + \frac{F_D\cdot\eta_{KU}^{1+s_v}}{1+\eta_{KU}^2}\right\}$$

$$\cdot\frac{2}{i_S}\cdot\frac{\eta_{RU}^{\frac{1}{1+s_v}}}{1+\eta_{RU}}\cdot\eta_{OU}^{s_v} + m_G\cdot\big(1-\kappa\big)\cdot g\cdot\frac{1}{i_S}\cdot\eta_{OU}^{s_v}, \tag{4}$$

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:

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

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_{2-1}, 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 [18] 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. 3. 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 [19]. 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 particular problem 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).

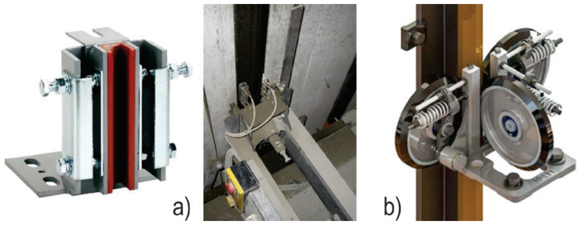


Fig. 3. *Presentation of the cabin and counterweight guidance system; a) sliding guide shoes and the real state of the elevator; b) roller guides*

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 [20], 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.

Typically, losses are calculated through efficiency coefficients of transmission. In the literature, they are usually estimated globally based on [15]:

- 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. 4.

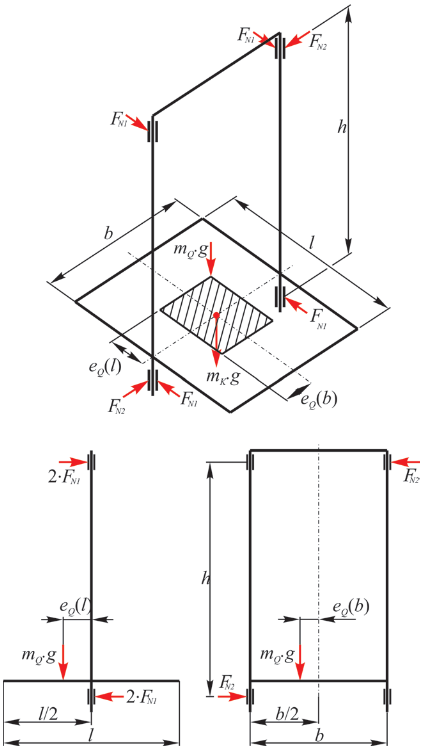


Fig. 4. *Frictional forces on cabin guide shoes*

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

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

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 $e_{Q(l)}$, $F_{tr2}=F_{N2} \cdot \mu_{kl}$ frictional force on the cabin sliding guide shoe due to the eccentricity of the load position $e_{Q(b)}$; $e_{Q(l)}$, $e_{Q(b)}$ the largest eccentricity of the load position across the cabin length and width, h spacing between sliding guide shoes in the vertical direction, μ_{kl} guide shoe friction coefficient, $F_{N1} = m_Q \cdot e_{Q(l)} / (2 \cdot h)$ the normal force on the sliding guide shoe caused by eccentricity $e_{Q(l)}$, and $F_{N2} = m_Q \cdot e_{Q(b)} / h$ 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 the resistance 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.

3 POWER (TORQUE) LOSS IN ELEVATOR SYSTEMS AND TRANSMISSION EFFICIENCY

According to the resistance analysis discussed in the previous chapter, po wer transmission losses in the drive and transmission mechanisms of elevators and vertical transport machines occur primarily in

- Gearbox (most commonly present in the system), at:
  - points of conjugate gear action, such as worm and worm wheel,
  - shaft bearings, due to friction,
  - points of friction between gears and oil inside the gearbox,
  - places where power is taken for auxiliary drives (e.g., lubrication pumps, etc.) if they exist.
- Pulleys (driving pulley, deflection pulley, cabin and counterweight pulley in the case of 2:1 suspension, other types of sheaves or sprocket for compensation rope or chain if they exist, pulleys for governor, etc.), at:
  - pulley bearings, due to friction,
  - at locations of rope bending [21] due to the rope stiffness.
- Cabin and counterweight guide rails.

3.1 Load Torques and Losses in Elevator Systems

Fig 5 shows a schematic representation of an elevator system with power transmission using ropes. A distinction must be made between the high-speed part (System 1), which includes the motor M and the part of a gearbox G (worm), and the low-speed part (System 2), which includes the part of a gearbox (worm gear), the shaft mechanism S, and the load Q.

All torques and moments of inertia are reduced to the high-speed parts shaft of the electric motor (angular velocity ω_1). The sum of the moments of inertia of the high-speed system components masses is denoted as J_1, J_{21} represents the reduced moment of inertia of the masses of the worm gear and the driving pulley, while J_{22} represents the reduced moment of inertia of the (translational and rotational) moving parts masses which are located in the elevator shaft. The sum of J_{21} and J_{22} can be denoted as J_2: $J_2=J_{21}+J_{22}$.

The differential equation of motion of the elevator system, according to Fig. 5, can be expressed in general form as [22]:

- for motor operating mode Q_{1-2}:

$$M_M - M_G - M_S - M_Q = (J_1 + J_2)\cdot\frac{d\omega_1}{dt}. \qquad (7)$$

- for generator operating mode Q_{2-1}:

$$-M_M - M_G - M_S + M_Q = (J_1 + J_2)\cdot\frac{d\omega_1}{dt}, \qquad (8)$$

where M_M is motor torque, M_G torque loss in the transmission, M_S torque loss in the elevator shaft, and M_Q load torque.

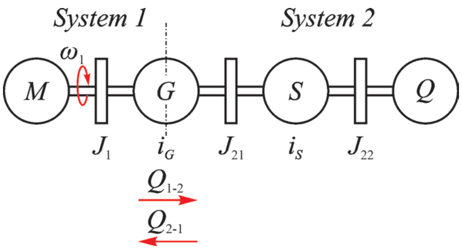


Fig. 5. *Schematic representation of elevator systems with power transmission through friction* [22]

All torque values are reduced to the motor shaft.

The energy balance can also be separately written for Systems 1 and 2:

$$M_M - M_G - M_2 = J_1\cdot\frac{d\omega_1}{dt}, \qquad (9)$$

$$M_1 + M_S + M_Q = J_2\cdot\frac{d\omega_1}{dt}. \qquad (10)$$

In Eq. (9), M_2 represents the torque acting at the meeting point of both systems. The same applies to M_1, where $M_1 = -M_2$, considering that the sum of Eqs. (9) and (10) must give Eq. (7). Therefore, M_2 is the torque acting on the shaft of the worm gear, which can be expressed from Eq. (10) as:

$$\text{s}\, M_2 = M_Q + M_S - J_2\cdot\frac{d\omega_1}{dt}. \qquad (11)$$

Depending on the magnitude of individual terms in this expression, M_2 can be greater or less than zero. If M_2 acts in the opposite direction of rotation, energy is transferred from system 1 to system 2, meaning that the power flow is from system 1 to system 2, symbolically denoted as Q_{1-2}. If M_2 acts in the direction of rotation, energy is transferred from system 2 to system 1, and this power flow is symbolically denoted as Q_{2-1}. This means that the direction of power flow is precisely defined at each moment, and it holds that: for Q_{2-1} $\operatorname{sgn} M_2 = -\operatorname{sgn}\omega_1$, and for Q_{2-1} $\operatorname{sgn} M_2 = +\operatorname{sgn}\omega_1$.

The weights located in the elevator shaft create a torque on the driving pulley (load torque), according to the forces in the ropes on the side of the cabin/counterweight and Eqs. (3) to (5), which, when reduced to the motor shaft, has the following magnitude:

$$M_Q = \frac{D}{2}\cdot\frac{1}{i_R\cdot i_S}\cdot\left[m_K - m_T + \lambda\cdot Q + \right.$$
$$\left. + m_D\cdot(1-2\kappa) + m_H\cdot(1-\kappa) + m_G\cdot(2\kappa-1)\right]\cdot g. \qquad (12)$$

By substituting Eq. (1) into Eq. (12), the final value is obtained in the form:

$$M_Q = \frac{D}{2}\cdot\frac{1}{i_R\cdot i_S}\cdot\left[Q\cdot(\lambda-\beta) + m_D\cdot(1-2\kappa) + \right.$$
$$\left. + m_H\cdot(1-\kappa) + m_G\cdot(2\kappa-1)\right]\cdot g. \qquad (13)$$

By introducing the term "nominal load torque":

$$M_{Qn} = \frac{D}{2}\cdot\frac{1}{i_R\cdot i_S}\cdot Q\cdot(1-\beta), \qquad (14)$$

the load torque becomes:

$$M_Q = M_{Qn}\cdot\left(\frac{\lambda-\beta}{1-\beta} - \sigma + \delta\cdot\kappa\right), \qquad (15)$$

where: $\sigma = \dfrac{m_G\cdot i_S - m_D - m_H}{(1-\beta)\cdot \mathrm{Q}}\cdot g$,

and $\delta = \dfrac{2\cdot m_G\cdot i_S - 2\cdot m_D - m_H}{(1-\beta)\cdot \mathrm{Q}}\cdot g$.

Losses in the elevator shaft are caused by friction on the cabin and the counterweight guides, in the bearings of the deflection pulleys and sheaves, as well as due to the bending of the ropes over the driving and deflection pulleys and sheaves. The assumption is that the torque loss (resistance) increases linearly with increasing load, i.e., that the coefficient of friction remains independent of the magnitude of the load. This assumption cannot always be fully applied to the friction on the guides since the position of the load in the cabin can play a role; that is, additional forces can occur due to the occurrence of asymmetric loading of the cabin.

In general, the coefficient of friction will undoubtedly change with the corresponding sliding velocity (the cabin velocity). The sliding velocities at different locations have different magnitudes at any given moment, so determining them would overly complicate the calculation. Therefore, a multiplier v_S dependent on the velocity is introduced for the torque loss, whose course of change (values) still requires detailed examination.

Some type of total friction coefficient can be calculated for the elevator shaft. The forces acting in the shaft, multiplied by the total friction coefficient $\mu_S \cdot v_S$, result in the friction force in the shaft, which would include the influence of individual friction coefficients for each deflection pulley, sliding guide, etc., for which there is already some empirical data.

Here, d'Alembert's apparent force originating from the masses on the counterweight side has retained a negative sign. This is because, for example, when $d\omega_1/dt > 0$, the apparent force loads the contact surfaces on the cabin side while it unloads them on the counterweight side. J_T denotes the moment of inertia of the masses on the counterweight side reduced to the shaft of the electric motor.

The total torque loss of the elevator shaft has the form:

$$M_S = -\left\{ M_C v_S + \mu_S v_S \cdot \left[M_{Qn} \frac{\lambda + \zeta \cdot \chi}{1-\beta} + \right.\right.$$
$$\left.\left. + \left(J_K - \chi J_T\right) \frac{d\omega_1}{dt} \right]\right\} \cdot \operatorname{sgn} \omega_1, \quad (16)$$

where:

$$M_C = M_{Qn} \mu_{ST} \frac{\chi\beta + \vartheta}{1-\beta},$$

$$\vartheta = \frac{1}{Q} \cdot \left[\chi m_G g i_S + m_D g + m_H g + \left(1+\chi\right)\left(m_K g + \frac{F_D}{2} \right) \right],$$

$$\zeta = \frac{1}{Q} \cdot \left[\left(1-\chi\right)\left(m_G g i_S - m_D g\right) + m_H g \right].$$

χ is the ratio of total moments of inertia in the elevator shaft on the side of the counterweight and on the side of the cabin.

Considering that roughly estimated, $J_K \approx J_T$ and $\chi = 1$, it follows:

$$J_K - \chi J_T \approx 0. \quad (17)$$

Since the values for J_K and J_T are already small compared to the total moment of inertia of the elevator system, the difference $J_K - \chi J_T$ multiplied by the factor $\mu_S \times v_S$, which is usually less than 0.1, becomes negligibly small. With this justified neglect, the torque loss of the elevator shaft becomes:

$$M_S \approx -\left(M_C v_S + \mu_S v_S M_{Qn} \frac{\lambda + \zeta \cdot \chi}{1-\beta} \right) \cdot \operatorname{sgn} \omega_1. \quad (18)$$

The torque loss of the elevator shaft is practically independent of the moments of inertia of the masses located within the shaft.

Losses in the transmission consist of losses due to gear meshing, losses due to friction in the bearings and losses in the so-called "idling".

Losses due to gear meshing occur on the contact surfaces between the worm and the worm wheel. The torque loss due to meshing can be expressed as a function of the torque on the worm wheel shaft M_2:

$$M_{VZ} = -\Omega \cdot \left|M_2\right| \cdot \operatorname{sgn} \omega_1, \quad (19)$$

where: $\Omega \approx \left(\frac{1}{i_R} \frac{d_R}{d_S} + i_R \frac{d_S}{d_R} \right) \cdot \mu$, and d_S is the pitch diameter of the worm, the d_R pitch diameter of the worm wheel, and μ coefficient of friction.

The value of the friction coefficient depends not only on the quality of the surface and the viscosity of the oil but also on the sliding velocity between the friction surfaces, i.e., from the number of revolutions.

The torque loss of worm pairs, according to Eq. (19), is linear only for small values of M_2, namely when viscous friction is present. In the presence of dry and viscous friction, the torque loss increases exponentially, i.e., the factor Ω is, in a sense, also dependent on $|M_2|$.

The torque loss due to friction in the bearings can also be expressed as a function of the torque M_2:

$$M_{Lag} = -C_{Lag} \cdot \left|M_2\right| \cdot \operatorname{sgn} \omega_1, \quad (20)$$

where C_{Lag} is constant dependent on the geometric dimensions of the worm gear.

Losses in the so-called "idling" occur due to resistance when parts move through oil in the gearbox. These losses depend on the geometric dimensions of the gearbox, the viscosity of the oil, and the rotational speed and can be expressed as:

$$M_0 = -C_0 \cdot \sqrt[3]{n_1}, \quad (21)$$

where n_1 is the number of worm rotations, and C_0 is constant, dependent on the dimensions of the gearbox.

The total torque loss of the gearbox is the sum of three losses given by Eqs. (19) to (21).

For this study, it is important to mention that the magnitude of the torque M_2 also depends on the angular acceleration J_2, according to Eq. (11). However, the experimental method applied in this study is related to determining the total losses in the stationary operating mode of elevators.

3.2 Elevator Efficiency Coefficient

By introducing the term "stationary shaft efficiency coefficient" η_S [22], 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:

$$\eta_S = \frac{M_Q}{M_Q + M_S} = \frac{1}{1+\dfrac{M_S}{M_Q}}. \tag{22}$$

Similarly, the "stationary gearbox efficiency coefficient" η_G can be introduced [22], which at $Q_{1\text{–}2}$ (Fig. 5) represents the ratio of the torque at the worm wheel to the torque at the worm shaft in the stationary state, i.e., when $d\omega_1/dt = 0$:

$$\eta_G = \frac{M_Q + M_S}{M_Q + M_S + M_G} = \frac{1}{1+\dfrac{M_G}{M_Q + M_S}}. \tag{23}$$

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 $Q_{1\text{–}2}$ direction of power flow.

$$\eta = \eta_S \eta_G = \frac{M_Q}{M_Q + M_S} \cdot \frac{M_Q + M_S}{M_Q + M_S + M_G} = $$
$$= \frac{M_Q}{M_Q + M_S + M_G} = \frac{1}{1+\dfrac{M_S + M_G}{M_Q}} = \frac{M_Q}{M_M}. \tag{24}$$

Similarly, the overall efficiency of the $Q_{2\text{–}1}$ direction of power flow is obtained in the same way, and it is given by:

$$\eta' = \frac{M_M}{M_Q}. \tag{25}$$

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. 6. As already mentioned, the power flow through the drive mechanism also has a certain impact on the magnitude of the efficiency.

Based on Eqs. (7), (8), (24), and (25), a common (universal) expression can be found that represents the relationship between the transmission efficiency in the generator ($Q_{2\text{–}1}$) and motor mode ($Q_{1\text{–}2}$). From Eq. (24), the following equality can be written:

$$1+\frac{M_S + M_G}{M_Q} = \frac{1}{\eta} \quad \rightarrow \quad \frac{M_S + M_G}{M_Q} = \frac{1}{\eta} - 1. \tag{26}$$

Eq. (8) and Eq. (25) for generator operation mode ($Q_{2\text{–}1}$) give:

$$\eta' = \frac{M_Q - M_S - M_G}{M_Q} = $$
$$= 1 - \frac{M_S + M_G}{M_Q} \quad \rightarrow \quad \frac{M_S + M_G}{M_Q} = 1 - \eta'. \tag{27}$$

From Eqs. (26) and (27) follows:

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

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 $Q_{1\text{–}2}$ and $Q_{2\text{–}1}$ while the direction of motion is different, i.e., when all transmission losses are independent of the load.

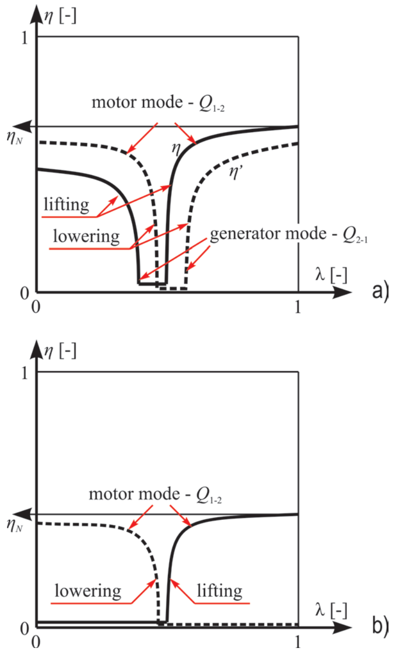


Fig. 6. *Principled form of dependence η=f(λ) with the direction of elevator cabin movement and the direction of power flow through the drive mechanism, as parameters; a) transmission that is not self-locking under nominal load; b) self-locking transmission even under nominal load* [15]

In Fig. 6, 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 (β = 0.5), the torque resistance of the system at the drive motor shaft is the same when the 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. Therefore, based on Fig. 5, the following characteristic cases of power flow may occur [15]:

- from the drive electric motor to the cabin or counterweight, Fig. 7a;
- from the cabin, i.e., the counterweight to the drive electric motor, Fig. 7b;
- from the drive electric motor to the transmission and from the cabin, i.e., the counterweight to the transmission (case of self-locking), Fig. 7c.

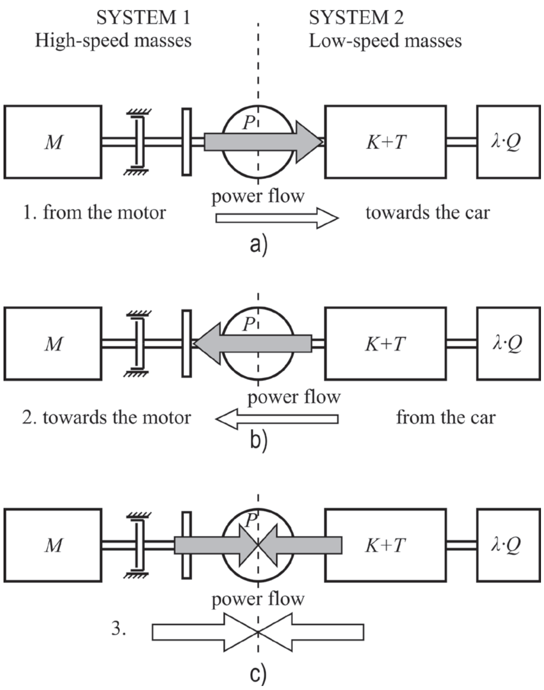


Fig. 7. *Power flow in elevators: a) from the drive electric motor to the cabin or counterweight, b) from the cabin or the counterweight to the drive electric motor, and c) from the drive electric motor to the transmission and from the cabin or the counterweight to the transmission*

Based on the above, it can be observed that there is significant complexity associated with determining losses or the efficiency of vertical lifting systems with 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.

4 EXPERIMENTAL METHOD FOR DETERMINING RESISTANCE 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.

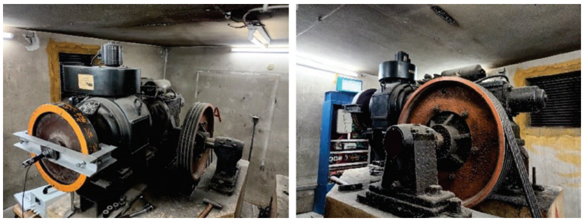


Fig. 8. *Driving machine of passenger elevator*

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. 8. 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 devices were 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
Diameter of the drive pulley:	D = 650 mm
Cabin dimensions:	1030 mm × 1420 mm × 2200 mm
Cabin guide rails:	⊥ 65 mm × 90 mm × 14 mm
Counterweight guide rails:	⊥ 50 mm × 50 mm × 6 mm

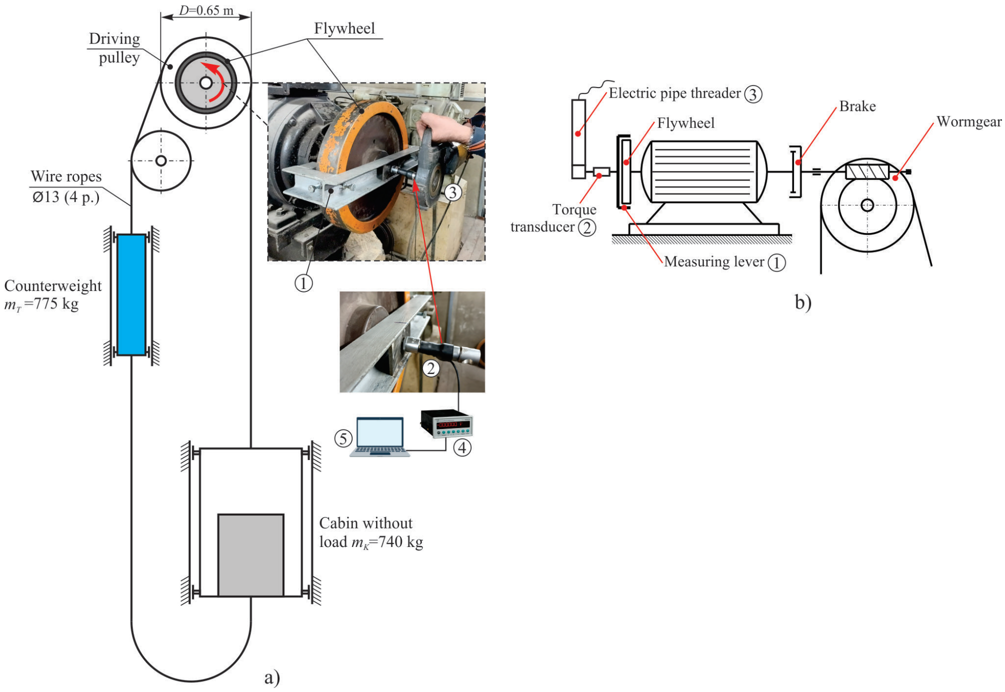


Fig. 9. *Arrangement of measuring equipment; a) general arrangement of measuring points, and b) detailed view of the kinematic chain*

4.1 Measuring Equipment and the Method for Measuring Data Acquisition

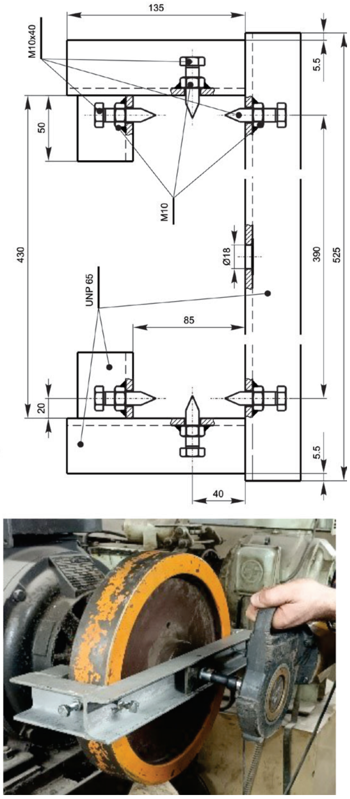
For the "stationary" measurements and experimental determination of the values and character of the change in efficiency, or losses, primarily of the worm gear reducer and cabin and counterweight guiding elements, the following equipment was used:

- Measuring lever, position 1 (Fig. 9),
- Rotating torque transducer, measuring range up to 200 Nm, position 2 (Fig. 9),
- Portable electric pipe threader, used for continuous rotation of the flywheel, and thus lifting and lowering of the cabin/counterweight, position 3 (Fig. 9),
- Measuring amplifier HBM MVD2555, accuracy class 0.1, position 4 (Fig. 9),
- 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. 9),
- Rope force measuring device of S-type HBM S9M, measuring range 0.5 kN to 50 kN,
- Universal 8-channel measurement amplifier HBM SPIDER 8.

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.

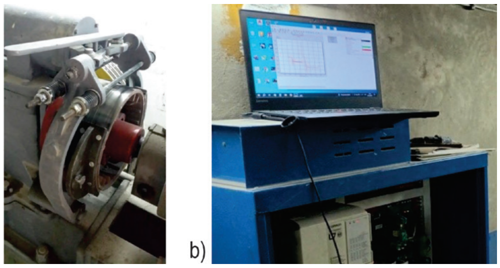
Fig. 10a shows the design of the measuring lever in the form of a technical drawing as a basis for its fabrication, while Fig. 10b depicts the measuring lever with the torque measurement device and the driving element (portable electric pipe threader) during measurement, mounted on the flywheel of the electric motor.

After manual disengagement of the brake lever (Fig. 11a), the portable electric pipe threader (Fig. 10b) is activated, whose torque is transmitted to the flywheel through the torque transducer (position 2, Fig. 9), 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. 11b).



a) b)

Fig. 10. *Measuring equipment used for the "stationary" measurements; a) technical drawing of measuring lever, and b) lever mounted on the flywheel of the electric motor*



a) b)

Fig. 11. *Measuring point: a) brake with the release lever, and b) part of measuring equipment*

Before the beginning of the experiment, i.e., before determining the total resistances based on torque measurement on the shaft of the drive electric motor, the weight of the cabin and counterweight was measured (checked) using a force measuring device on the ropes, as shown in Fig. 12a. The measurement was performed by mounting the device on the suspension ropes (from the first to the fourth) directly above the cabin. The device was connected to the universal measurement amplifier SPIDER 8; then the measurement data were collected in the memory of the laptop, position 5 (Fig. 9). The values of forces (weights) in individual ropes were read, and their sum provided the exact (actual) weight of both the cabin and the counterweight.

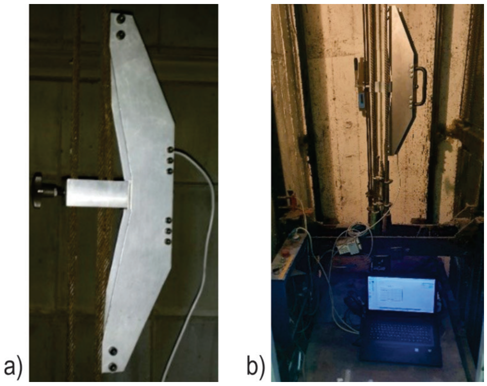


Fig. 12. *Measuring rope tension force: a) device for measuring (checking) the mass of the cabin and counterweigh, b) the position of the mounted device.*

It should be noted that the measurements were conducted for cases when the cabin was at the bottom level (ground floor), approximately at the midpoint of the lifting height (14 m), and at the top level (9th floor). For all these cases, 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. In the case where the cabin was at the top level, lowering was performed first, followed by lifting.

4.2 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.

In Fig. 13, a schematic representation of a passenger elevator with marked levels (floors) is shown.

The measurement results and determination of the total elevator efficiency for stationary conditions in this study are presented for three characteristic cases (cabin positions), with different loads in the cabin and without any load. The results will be shown for cases of lifting and lowering the empty cabin, and the cabin loaded with weights of 80 kg, 160 kg, 240 kg, and with the rated load (320 kg), based on which the total resistances will be determined in the form of the coefficients of efficiency. The characteristics of the selected experimental cases are as follows:

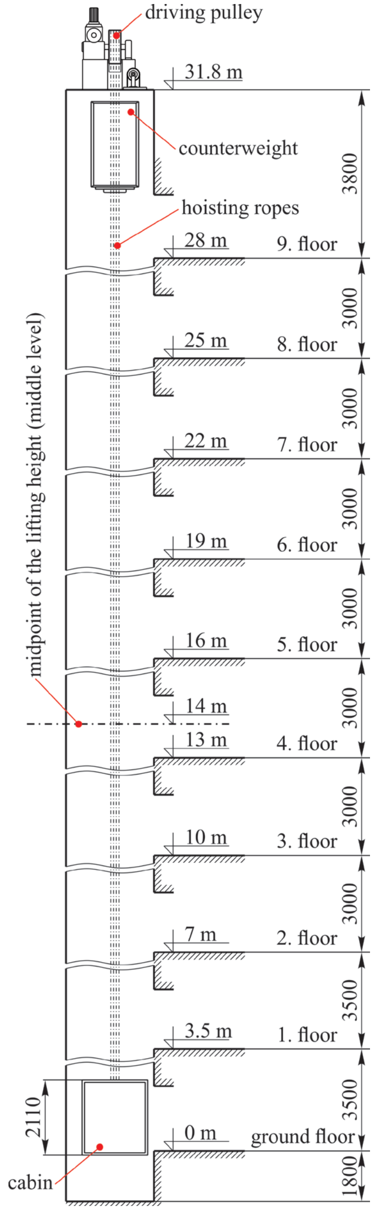


Fig. 13. *Schematic representation of the elevator shaft*

I) Lifting the cabin from the ground floor (bottom level) for more than 10 revolutions of the drive motor shaft (flywheel), then lowering for the same value and returning to the initial position. Upon stopping, the brake lever was simultaneously disengaged. The experiment was repeated for

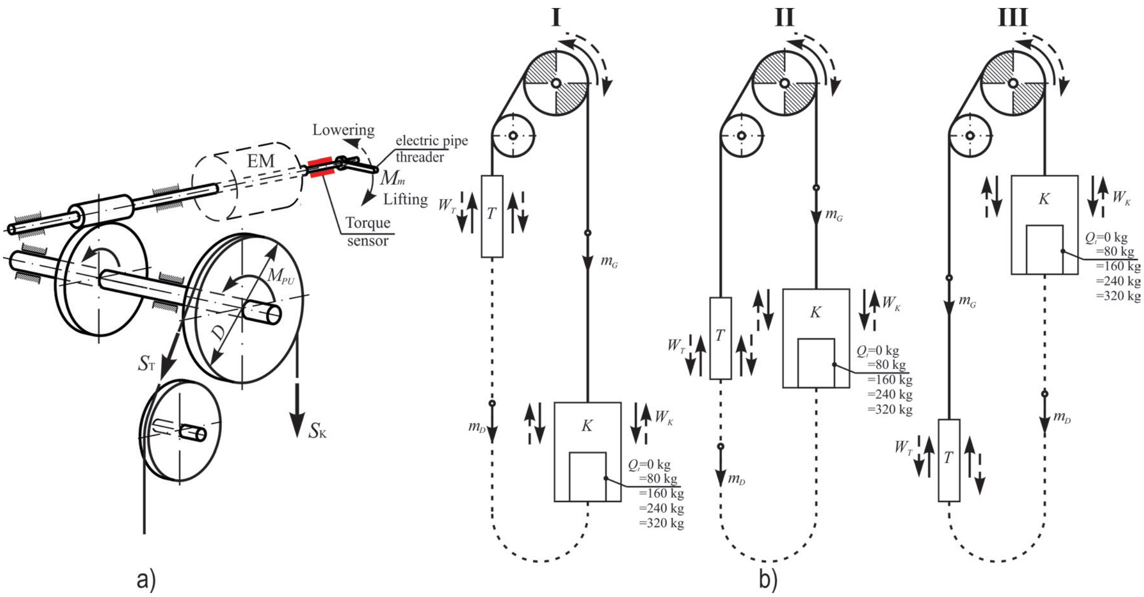


Fig. 14. *Passenger elevator parameters relevant for determining total resistances: a) elevator drive mechanism, and b) forces and different positions of the cabin*

both the empty cabin and the cabin loaded with weights of 80 kg, 160 kg, 240 kg, and 320 kg.

II) 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 as in case (I) for both the empty cabin and the cabin loaded with weights of 80 kg, 160 kg, 240 kg, and 320 kg.

III) Lowering the cabin from the 9th floor (top level) for more than 10 revolutions of the drive motor shaft (governor), then lifting for the same value and returning to the initial position. Upon stopping, the brake lever was simultaneously disengaged. The experiment was repeated as in cases (I) and (II) for both the empty cabin and the cabin loaded with weights of 80 kg, 160 kg, 240 kg, and 320 kg.

The activity flow during experimental measurements can also be illustratively presented through the diagram given in Fig. 15.

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 Figs. 16 to 18.

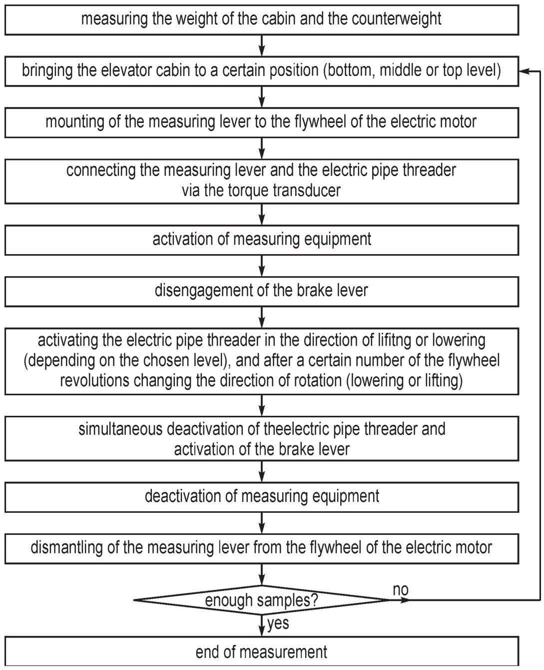


Fig. 15. *Activity flow diagram for experimental measurement*

The results correspond to all three cases of movement, i.e., lifting/lowering of the cabin at the bottom level (ground floor), at the midpoint of the lifting height, and at the top level (9th floor), for five different load values. From the diagrams, it can be observed that when engaging and disengaging the electromotor-driven threading machine (start

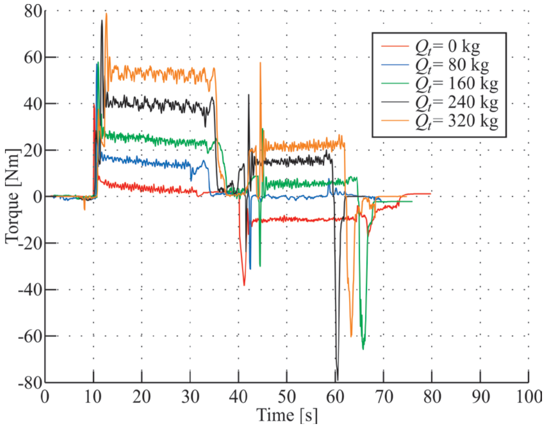


Fig. 16. *Measurement of the total torque on the drive motor shaft during the lowering/lifting of the elevator cabin at the bottom level for various loads (Case I)*

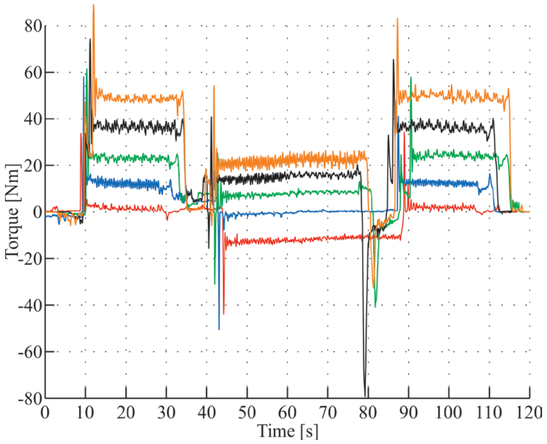


Fig. 17. Measurement of the total torque on the drive motor shaft during the lowering/lifting of the elevator cabin at the middle level for various loads (Case II)

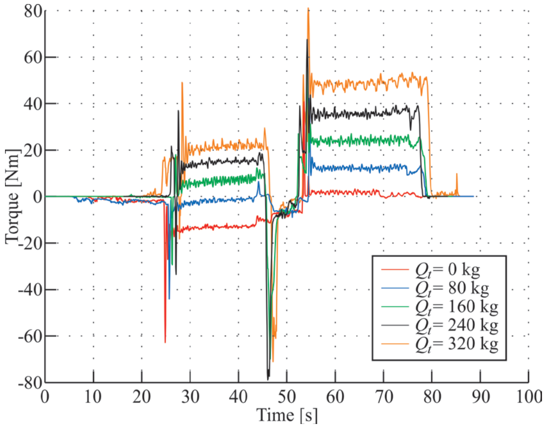


Fig. 18. Measurement of the total torque on the drive motor shaft during the lowering/lifting of the elevator cabin at the top level for various loads (Case III)

of 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 4.3.

Table 2 gives the averaged values of the measured torque on the drive motor shaft, obtained from the diagrams shown in Figs. 16 to 18 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*

Q_t [kg]	M_m [Nm]					
	Bottom level		Middle level		Top level	
	Lifting	Lowering	Lifting	Lowering	Lifting	Lowering
0	2.95	–11.14	1.72	–11.69	–11.48	1.43
80	14.78	–0.28	12.42	–0.30	–1.69	12.26
160	24.45	5.71	23.71	7.91	6.95	24.17
240	39.29	14.00	36.57	15.15	14.84	35.83
320	52.76	21.87	49.03	21.93	21.51	48.85

4.3 Determination of the Elevator Overall Efficiency Based on the Measurement Results

Based on the measurement results depicted in Figs. 16 to 18 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 3.2. 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. (24):

$$\eta = \frac{M_Q}{M_M}.$$

The values of the overall efficiency for the generator operation mode, i.e., for Q_{2-1}, are obtained from Eq. (25):

$$\eta' = \frac{M_M}{M_Q}.$$

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 ($i_S = 1$), Eq. (13) for determining the load torque (load torque reduced to the shaft of the drive motor) becomes (m_H can be neglected due to its small value):

$$M_Q = \frac{D}{2} \cdot \frac{1}{i_R} \cdot Q \cdot g \cdot (\lambda - \beta) = \frac{(\lambda - \beta) \cdot Q \cdot g \cdot D}{2 \cdot i_R}, \quad (28)$$

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.

Table 3. *Overall elevator efficiency coefficients at the bottom level*

Q_t [kg]	λ [-]	M_Q [Nm]	η [-]			
			Q_{1-2}		Q_{2-1}	
			Lifting	Lowering	Lifting	Lowering
0	0	–4.03	-	0.36	0.73	-
80	0.25	5.18	0.35	-	-	0.05
160	0.50	14.39	0.59	-	-	0.40
240	0.75	23.60	0.60	-	-	0.59
320	1.00	32.80	0.62	-	-	0.67

Table 4. *Overall elevator efficiency coefficients at the middle level*

Q_t [kg]	λ [-]	M_Q [Nm]	η [-]			
			Q_{1-2}		Q_{2-1}	
			Lifting	Lowering	Lifting	Lowering
0	0	–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

Table 5. *Overall elevator efficiency coefficients at the top level*

Q_t [kg]	λ [-]	M_Q [Nm]	η [-]			
			Q_{1-2}		Q_{2-1}	
			Lifting	Lowering	Lifting	Lowering
0	0	–4.03	-	0.35	0.35	-
80	0.25	5.18	0.42	-	-	0.33
160	0.50	14.39	0.60	-	-	0.48
240	0.75	23.60	0.66	-	-	0.63
320	1.00	32.80	0.67	-	-	0.66

The counterweight weight factor and the relative load of the cabin (cabin load factor) are determined based on Eqs. (1) and (2).

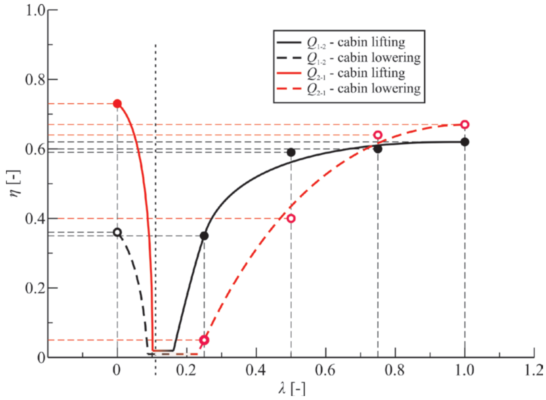


Fig. 19. *The change in the total efficiency of the elevator for different operating modes and cabin movements at the bottom level*

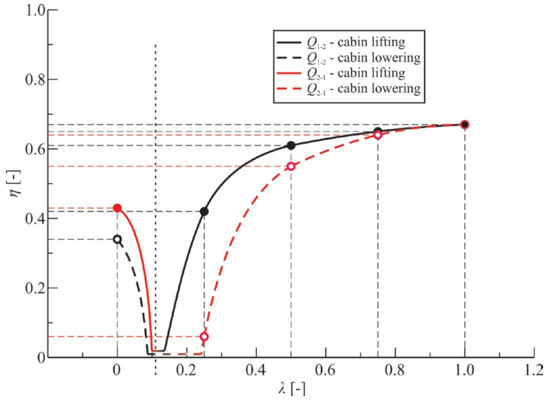


Fig. 20. *The change in the total efficiency of the elevator for different operating modes and cabin movements at the middle level*

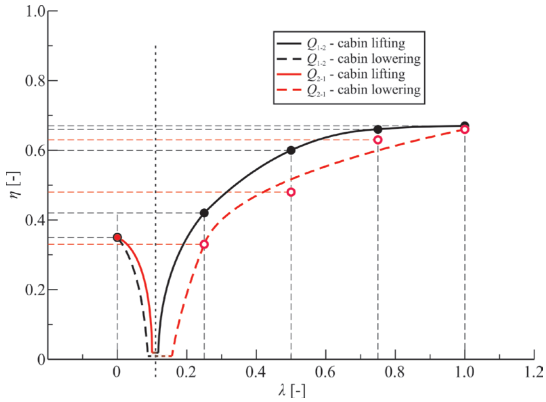


Fig. 21. *The change in the total efficiency of the elevator for different operating modes and cabin movements at the top level*

The calculated values of the overall elevator efficiency coefficients are given in Tables 3 to 5.

The measurement results with calculated values of the total efficiency (η), presented in Tables 3 to

5, can also be depicted graphically in Figs. 19 to 21. Based on the tables and diagrams (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.

5 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.

The total resistance in elevators depends on numerous factors, such as the elevator's design, the type of transmission in the drive mechanism, the relative velocity of the contact surfaces, the type of cabin and counterweight sliders, the size and eccentricity of the cabin load, lubrication, wear and tear, the condition of the contact surfaces, temperature, etc. Through the presented procedure and experimental research, it is possible to determine the real values of these parameters for static conditions.

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.

6 ACKNOWLEDGEMENTS

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