Determining of the drive power of a transport machine for disabled persons using a computational model

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Abstract. The article deals with calculation of a stair lift drive and its design. The aim is the determining the necessary power of the used electric motor depending on the variable input values such as a passenger weight, a gradient of the seat lift etc. Calculating the required power for operation of the equipment is essential for selecting the optimal drive motor. In practice, the mathematical-physical principle of solving a problem such as drive calculation is often neglected. This results in the use of excessively powerful propulsion equipment, and thus in excessive financial cost in mass production. Consequently, by designing a drive-train, it will be possible to design the entire drive mechanism, i.e. motor, gearbox, gear module, etc. A significant part of the overall solution will be the design of the shaft for the drive pinion and its analytical dimensional calculation in comparison with the numerical solution. This will fulfill the prerequisites for optimizing the current shaft design by reducing its mass from the current value of 1.185 kg while the safety and reliability conditions of the construction are still met.

1. Introduction

One of the social aspects of a modern advanced society is its ability to help those in a difficult situation. One subset affliction of people is a restriction of a person’s mobility. Several devices have been contrived for people with disabilities to facilitate surmounting various obstacles. A stair seat can also be found between these devices, which is also the subject of the issue solution (Figures 1 and 2). Since there are stairways of different shapes and dimensions, stair seats cannot be uniformly specified. In production of each stair lift, its design modifications must be done according to the specific requirements of each customer [1], [2].

Such a handling machine is usually made of more kind of material to reach suitable mechanical properties of particular components in combination with required lifetime and acceptable price. There are most often used combination of standard steel, high strength steel, light metals, mainly such as aluminium alloys, plastics etc. [3-5]. The latest types of stair lifts have the automatic process of tilting and they dispose of a remote control [6]. The drive-train is composed of an electric motor, a gearbox and a gear wheel, which is in the mesh with a rack.

Previous steps of solving this issue [7], [8] cover: boundary and load conditions of the seat, structure material properties and force effects calculations related to the seat track.

Subsequently, the acquired results will be used to design the necessary power of the electric motor used as a seat drive, since determining the power of the electric motor for driving is an inevitable part of the design of each mechanism. Thanks to the calculation, it is possible to equip the device with an optimum engine that is powerful enough to handle the operation, but at the same time its power is not unnecessarily oversized. From the previous calculations [7], [8], it is clear that quantities of resistance can be expressed in the form of force or moment. The same principle applies at calculating the engine
power when the desired speed of the moving device and the diameters of the contact circles of the wheel and the track are known. Calculation with considering resistances as moments of forces is appropriate in the case of known shaft speeds, the starting point being the approximate speed of the device and the pinion geometry. The speed calculated based on the approximate speed should be corrected in terms of a suitable gear ratio between speed of the engine and the pinion.

Figure 1. A real model of SA Alfa stair lift (left), Schematic change of geometry when the carriage is tilted due to the ascent of the track (right).

2. Theoretical aspects of electric motor power calculation

The calculation of the engine power of a stair seat consists of multiple variables and constant values. Therefore, it is necessary to state the boundary conditions when the resistive forces acting against the movement of the seat will be maximum. There has been found out, that from all resistances influencing the needed power of the used motor just the transported person weight is the most variable quantity. From this point of view there is necessary to consider different conditions in the contact of the driving wheel and the track [9-13].

From a structural point of view, a track with a maximum inclination of 52° is possible, which is the first boundary condition for the calculation. The second is the maximum load capacity that is in this case 130 kg. The third condition is the speed of the device, taking 0.1 ms⁻¹. From the three stated boundary conditions, the track inclination (or pitch) angle α and the weight of a person are the variables. Hence, it is suitable to illustrate the dependence of the required power on these variables. Resistance forces are dependent on the type of material and the machining contact areas in relative motion at mechanism operation.

Value of this resistance is determined by the normal force transmitted by the wheel, friction coefficient between the pin surface and material of the wheel, and the friction circle of radius [14], [15].

The staircase track (Figure 2) is made of steel S235JRG1 (STN 11 373) with powder coating comaxit. This ensures a surface resistance to mechanical damage and excellent durability and chemical resistance. Since the applied coating is hardened and its thickness is negligible, it can be assumed that it will not have a significant impact on the rolling resistance. A part of the track is also a gear rack made of steel C16E (STN 12 020) with modulus \(m = 3.5\) (mm).

The components that come into contact with the track are made of polyamide er talon 6 PLA. Its usage in industry ranges from the production of simple parts to gears and plain bearings. Its basic mechanical properties include high strength, stiffness, hardness and toughness, high fatigue strength, high mechanical damping ability, good sliding properties, abrasion resistance and good machinability. The wheels of the upper carriage are pressed on a composite plain bearing placed on a steel pin. The lower carriage pulley carrying the vertical load moves axially on the shaft. There is also located the pinion gear that transmits the power from the output shaft on the gear rack. The horizontal load bearing...
Polyamide rollers of the lower carriage are mounted on steel quenched and polished pins. All connections between the rotating guide member and the pin are lubricated to reduce the coefficient of friction between the components. Output shaft of the gearbox is made of steel 17MnCr5 (STN 14 220). Because of the location of the rolling bearings on the shaft for the lower carriage, the shaft part is cemented and hardened to 60 HRC. The power flow is directed from the motor to the worm gear on the output shaft, from where it is transmitted to the track by means of a pinion on the track gear rack. The pinion is made of zinc galvanized steel E335 (STN 11 600). Since the pinion teeth engage more often than the teeth on the gear rack, the hardness of the pinion teeth needs to be higher due to wear reduction.

![Figure 2. A curved path of the track for the designed stair lift.](image)

3. Design of a drive motor

To calculate the necessary power of the used electric motor, a large number of input parameters are needed, the design of which was solved in [8]. The values of parameters obtained in previous calculations are stated in Table 1.

| no. | Input quantity                                      | Value of quantity                      |
|-----|----------------------------------------------------|----------------------------------------|
| 1   | \( G \) - weight of the seat                       | \( G = 559.17 \text{ N} \)             |
| 2   | \( Q \) - weight of a person                       | \( Q_{\text{max}} = 1275.3 \text{ N} \) |
| 3   | \( \alpha \) - slope of the staircase              | \( \alpha_{\text{max}} = 52^\circ \)  |
| 4   | \( v \) - seat speed                               | \( v = 0.11 \text{ ms}^{-1} \)        |
| 5   | \( m \) - gear rack and pinion module              | \( m = 3.5 \text{ mm} \)              |
| 6   | \( r_{\text{cav}} \) - radius of pin friction of lower guide rollers | \( r_{\text{cav}} = 4 \text{ mm} \) |
| 7   | \( r_{\text{cak}} \) - radius of pin friction of the pulley carrying the vertical load | \( r_{\text{cak}} = 17.5 \text{ mm} \) |
| 8   | \( r_{\text{ch}} \) - radius of pin friction of upper guide rollers | \( r_{\text{ch}} = 5.95 \text{ mm} \) |
| 9   | \( f_{\text{oh}} \) - friction coefficient of upper guide wheels | \( f_{\text{oh}} = 0.04 (-) \) |
| 10  | \( f_{\text{oa}} \) - friction coefficient of lower guide wheels | \( f_{\text{oa}} = 0.1 (-) \) |
| 11  | \( e_{\text{av}} \) - the coefficient of rolling resistance of the lower guide rollers | \( e_{\text{av}} = 0.021 \text{ mm} \) |
| 12  | \( e_{\text{ak}} \) - the coefficient of rolling resistance of the lower guide pulley | \( e_{\text{ak}} = 0.07 \text{ mm} \) |
| 13  | \( e_{h} \) - the coefficient of rolling resistance of the upper guide rollers | \( e_{h} = 0.03 \text{ mm} \) |
| 14  | \( a \) - distance (figure 1)                      | \( a = 49 \text{ mm} \)              |
Table 1. Calculation results of input variables for calculation of electric motor performance (continuation).

| no. | Input quantity                        | Value of quantity |
|-----|--------------------------------------|-------------------|
| 15  | \( b \) - distance (figure 1)        | \( b = 67.5 \text{ mm} \) |
| 16  | \( h \) - distance (figure 1)        | \( h = 236 \text{ mm} \) |
| 17  | \( \eta_1 \) - worm gear efficiency   | \( \eta_1 = 0.66 \) |
| 18  | \( \eta_2 \) - rolling bearing efficiency | \( \eta_2 = 0.98 \) |
| 19  | \( g \) - gravity acceleration       | \( g = 9.81 \text{ m}\cdot\text{s}^{-2} \) |
| 20  | \( \beta \) - the angle between the normal reaction of the upper guide wheels and the vertical axis | \( \beta = 32.7^\circ \) |
| 21  | \( d \) - pitch diameter of the drive gear pinion | \( d = 66.5 \text{ mm} \) |
| 22  | \( F_s \) - the force from the seat ascent | \( F_{s\text{max}} = 2570 \text{ N} \) |
| 23  | \( x \) - the distance between the reactions \( R_{av} \) and \( R_{by} \) | \( x = 212.977 \text{ mm} \) |
| 24  | \( n \) - shaft speed of the electric motor | \( n = 28.72 \text{ min}^{-1} \) |
| 25  | \( \eta_c \) - the overall efficiency of the propulsion mechanism | \( \eta_c = 0.634 \) |

According to the relation (1), the power \( P \) (W) of the used engine depending on the resistive forces \( F_o \) (N), the speed of the seat movement (m/s\(^{-1}\)) and the overall efficiency of the mechanism \( \eta_c \) (-) is:

\[
P = \frac{\sum F_o \cdot v}{\eta_c}
\]

(1)

By adding the appropriate relations of all the considered resistive forces to equation no. 1 for power \( P \) we get:

\[
P = \frac{\sum (F_{ob} + F_{on} + F_{ok} + F_s) \cdot v}{\eta_c}
\]

(2a)

\[
P = \frac{2 \cdot (G \cdot 119 + Q \cdot 180)}{x} \left( \frac{e_b + r_{ob} \cdot f_{ob}}{d_b \cdot \sin \beta} + \frac{e_{av} + r_{om} \cdot f_{om}}{d_{av}} \right)
\]

(2b)

\[
+ \frac{(G + Q) \left[ \frac{2 \cdot (e_{ak} + r_{ok} \cdot f_{ok})}{d_{ak}} \right] \cdot \cos \alpha + \sin \alpha}{\eta_c} \cdot v
\]

Figure 3. The drive engine TTN-442-15 with the worm gearbox CM 040 (left), along with the designed seat mounts (right).
According to the relation (2), the power $P$ (W) of the used motor depending on the resistive moments $M_o$ (N·mm), angular speed of the engine rotor $\omega$ (rad·s⁻¹) and the overall efficiency of the mechanism $\eta_c$ (-) is:

$$P = \frac{\sum M_o \cdot \omega}{1000 \cdot \eta_c}$$  \hspace{1cm} (3)

By adding the appropriate relations of all the considered resistance moments to equation no. 2 for power $P$ we get:

$$P = \frac{\sum (M_{ob} + M_{av} + M_{ok} + M_s) \cdot \omega}{1000 \cdot \eta_c}$$

(4a)

$$P = \frac{2 \cdot (G \cdot 119 + Q \cdot 180)}{x} \left( \frac{e_b + r_{sb} \cdot f_{sb} + e_{av} + r_{ca} \cdot f_{ca}}{\sin \beta} \right) +$$

$$+ \frac{(G + Q) \left[ (e_{ob} + r_{bo} \cdot f_{ob}) \cdot \cos \alpha + \frac{d}{2} \sin \alpha \right]}{1000 \cdot \eta_c} \cdot \omega$$  \hspace{1cm} (4b)

By substituting the appropriate values in (1) and (2) it is possible to obtain the value of the necessary power of the propulsion engine to meet the above requirements. The resulting power of the propulsion engine proposed for nominal equipment parameters is calculated as $P = 256.677$ W. The engine that will drive this equipment must therefore have the closest higher power in order to ensure the operation of the equipment without the risk of permanent overload. Therefore, the TTN-442-15 engine is proposed for drive. Its nominal power is $P_n = 270$ W, supply voltage $U = 24$ V, current $I = 16$ A, torque $M_n = 1.4$ Nm, speed $n = 1,800$ min⁻¹ and weight $m = 5.5$ kg. The worm gearbox of gear ratio $i = 60$ (-) marked CM040 will be used. The entire drive unit assembly is shown in figure 3.

4. Design of drive mechanism
The transmission of torque from the motor into the system is ensured by two keys DIN 6885 A. The keys are located one behind the other on the shaft (figure 4 on the left). In this part of the shaft, the M8 thread is concentrically drilled. A bold with special washer is screwed there, when the shaft is mounted to the gearbox. This ensures the shaft against axial movement. In the middle of the shaft there is a space for the flange, seat cover, holder, washer, needle bearings and a movable plastic roll. A toothed pinion is mounted on the shaft from the other side. Between the pinion and the shaft there is a DIN 6885 A key, which transmits the torque. To achieve the minimum possible length of the key, the keyseat is located specifically extending into the side of the shaft cylinder. Similarly to the front of the shaft, an M8 thread is drilled into the shaft end, into which the screw is placed. Its purpose is to cover the tooth system by means of a washer and to axially secure the toothed pinion.

![Figure 4. Shaft (left), plastic roll with groove (in the centre) and sprocket with pentagon flange (right).](image-url)

Plastic roll (figure 4 in the centre) is clearance fit on the shaft so that it can move axially and copy the path shape mainly in curves. A special groove is formed on the roll, which is part of the safety
mechanism. When the permitted seat speed is exceeded, a mandrel is lowered to the groove to secure the seat in place. In addition to the plastic roll, the gear also has a special adjustment (figure 4, right). The pentagonal flange is a part of the safety system. The bearing, which is placed on the stirrup moves on the flange. The mechanism controls the speed of the unit as well as the release of the safety mandrel.

![Figure 5. 3D model of the drive mechanism with the back of the skeleton and safety mechanism (left) and mechanism with safety covers (right).](image)

The drive mechanism is firmly attached to the seat frame to carry all reaction forces and moments. A flange is screwed onto the gearbox and the bolts are secured with spacers. The flange thus fastened is further screwed by means of bolts, washers and tabs to the back of the seat frame (figure 5). Pins with rollers are placed in the bearing housing. These help to move, stabilize the seat and transfer a portion of the forces acting on the lower carriage of the seat.

5. Expressing the change in the required engine power with respect to human weight
Of all the factors that affect the size of the resistance, and hence the performance of the drive motor, the weight of a man is the most variable value. The coefficients associated with passive resistances depend on the choice of material and the geometry of the components used, so their value is unchanged until the design of the device changes. Certain parameters are subjected to regulations and standards e.g. STN EN 81-40, therefore, the speed of a stair seat of various manufacturers is mostly 0.1 ms⁻¹.

| \( m_c \) [kg] | \( Q \) [N] | \( M_{ob} \) [N·mm] | \( M_{oav} \) [N·mm] | \( M_{oak} \) [N·mm] | \( M_o \) [N·mm] | \( \Sigma M_o \) [N·mm] | \( P \) [W] |
|---|---|---|---|---|---|---|---|
| 52 | 510.1 | 368.9 | 313.0 | 1,198.1 | 28,016.8 | 29,986.9 | 148.2 |
| 58 | 569.0 | 393.5 | 334.0 | 1,264.1 | 29,559.1 | 31,550.7 | 156.4 |
| 64 | 627.8 | 418.2 | 354.9 | 1,330.0 | 31,101.3 | 33,204.5 | 164.6 |
| 70 | 686.7 | 442.9 | 375.9 | 1,396.0 | 32,643.5 | 34,658.3 | 172.8 |
| 76 | 745.6 | 467.6 | 396.8 | 1,462.0 | 34,185.7 | 36,512.0 | 181.0 |
| 82 | 804.4 | 492.3 | 417.8 | 1,527.9 | 35,727.9 | 38,165.8 | 189.2 |
| 88 | 863.3 | 516.9 | 438.7 | 1,593.9 | 37,270.1 | 39,819.6 | 197.4 |
| 94 | 922.1 | 541.6 | 459.6 | 1,659.8 | 38,812.3 | 41,473.4 | 205.6 |
| 100 | 981.0 | 566.3 | 480.6 | 1,725.8 | 40,354.5 | 43,127.2 | 213.7 |
| 106 | 1,039.9 | 591.0 | 501.5 | 1,791.7 | 41,896.8 | 44,781.0 | 221.9 |
| 112 | 1,098.7 | 615.6 | 522.5 | 1,857.7 | 43,439.0 | 46,417.4 | 230.1 |
| 118 | 1,157.6 | 640.3 | 543.4 | 1,923.6 | 44,981.2 | 48,088.5 | 238.3 |
| 124 | 1,216.4 | 665.0 | 564.4 | 1,989.6 | 46,523.4 | 49,742.3 | 246.5 |
| 130 | 1,275.3 | 689.7 | 585.3 | 2,055.5 | 48,065.6 | 51,396.1 | 254.7 |

It follows that, while maintaining the design of the stair seat, the necessary power changes only in relation to the change in human weight and the slope angle. In the calculation for the power of the drive motor, two forces, namely force \( G \) and force \( Q \), are exerted. The force \( G \) is formed by the weight of the stair seat and its size is fixed, while the force \( Q \) is the result of the weight of the person sitting on the seat.
Figure 6. Dependence of resistive moments on passenger weight at constant inclination angle $\alpha = 52^\circ$.

Given that the weight of the occupants of the stair seat differs, the value of the necessary power will also be different. The starting weight of a human is 52 kg because it does not make sense to think of a human weight of zero or very low weight, since the primary purpose of the device is to transport a person. The change in the magnitude of the resistive moments is quantified in table 2 and graphically represented in figure 6 and figure 7.

Figure 7. Dependence of the required power of the drive motor depending on the weight of the passenger at a constant inclination angle of the stair seat path $\alpha = 52^\circ$.

6. Numerical verification of shaft safety
Numerical calculation of stress in shaft was performed by means of the finite element method, which is common used method for standard analyses of structures of technical parts, assemblies and also whole constructions [16], [17]. In our research we have used Ansys Workbench. After starting the program, input material data was set in the EngineeringData section. The shaft model in format STEP was imported from Autodesk Inventor, and the shaft surfaces were split to simulate the surfaces under the pinion, plastic roll, bearings and gearbox. In addition, a surface has been created by dividing the front cylindrical surface resting on the gearbox in two halves. The larger radius of this area has been reduced to 15 mm, which corresponds to the radius of the contact portion of the gearbox. On the prepared surface it was possible to simulate the shaft-to-gear contact. All these actions were performed in the Geometry
section. The model mesh was then defined in the Model section. The mesh was created with 2 mm long elements and was formed by 53,684 nodes and 31,234 elements (figure 8).

Figure 8. FEM mesh of the shaft.

After meshing the shaft, the necessary degrees of freedom have been provided to the prepared surfaces under the gearbox and the front cylindrical surface of the shaft. The forces and moments have been assigned to the area under the bearings, the area under the plastic roll, and the area on which the pinion is located. Under these conditions, the equivalent (von Mises) stress was selected as output and then the calculation was started. The calculation results are shown in figure 9.

Figure 9. Stress distribution in the shaft.

The shaft material used is steel 1.7131 (16MnCr5, STN 14 220) with tensile strength \( R_m = 780 - 850 \) MPa and with the minimum yield strength \( R_y \) \( = 550 \) MPa. The maximum stress value determined by the simulation is \( \sigma_{\text{max}} = 463.42 \) MPa. The safety \( k \) of the shaft to a non-elastic deformation is then (5):

\[
k = \frac{R_{\text{min}}}{\sigma_{\text{max}}}
\]

By adding values to the relationship (5) we get:

\[
k = \frac{R_{\text{min}}}{\sigma_{\text{max}}} = \frac{550}{463.42} = 1.187
\]

The future research will be focused to analyse effects resulting from the structure dynamics. For these purposes, advanced dynamical model has to be developed in MBS software [18]. Solutionists will make an effort for creating a computational model of the solved lifting seat as a multibody model [19], [20] even including flexible model, which will aid to identify the most loaded elements of the mechanical system and further optimise them. By means of further simulations and analyses it will be possible to find out values of accelerations in individual components of the lifting seat. Accelerations are main
dynamical outputs [21], based on which we are able to analyse other dynamical effects influencing comfort [22], [23] during operation of the seat (acceleration, deceleration, climbing, kneeling, moving along a curved track on a sharpened staircase).

7. Conclusion
The article brings a summary of the state of the art of handling technology design for disabled people. This is the stair lift of the type with high availability for the civil sector. Along with previous calculations, article solves the design of the required power of the used electric motor, which was set to \( P = 256.667 \text{ W} \). From this point on, when a dimensional power calculation was made, it was possible to select a particular type of electric motor used along with gearbox.

Whole engine set TTN-442-15 with gearbox CM 040 will be delivered by Italian company Transtecno. To determine the instantaneous value of engine power depending on the weight of the passenger at the maximum inclination angle of the seat, a clear graph illustrating the given linear dependence was made. This seems logical, since human weight was considered as the only change in the parameters entering the calculation. Consequently, after determining the geometry of the engine, it was possible to design the seat frame into which the force effects are transferred from the operation of the device. Pinion drive shaft design and its numerical dimensional calculation were performed.

The analysis proved the suitability of the used shaft for the specified operation, but also revealed the space for its optimization because of the minimum stress values in the bulk of the shaft volume. Also, the safety factor \( k = 1.187 \) (-) does not correspond to the generally accepted fatigue damage prevention value to \( k_{\text{min}} = 1.2 \) (-). Therefore, it will be appropriate to optimize the shaft, which will be the main goal of further addressing this issue. It will also be necessary to perform security-oriented calculations for other parts and components used to complete the mechanism (figure 10).

![Figure 10. 3D visualization of the stair lift.](image)

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