URN (Paper): urn:nbn:de:gbv:ilm1-2014iwk-144:7
58th ILMENAU SCIENTIFIC COLLOQUIUM
Technische Universität Ilmenau, 08 – 12 September 2014
URN: urn:nbn:de:gbv:ilm1-2014iwk:3

A WHEG-AXLE-TRACKING MECHANISM FOR PASSENGER TRANSPORT PURPOSES

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1. ABSTRACT

Within the project Silver-Mobility it was ascertained that the market does provide technical assistance to support near-field mobility for the age group 50+, but it is rarely optimally adapted to the user's needs. In particular the ability to overcome obstacles is in need of improvement. For this using whegs, known from mobile robotics, seems to be promising. But in order to use them for passenger transport purposes, the dynamic height offset occurring in wheg-driven mechanisms, called alternation, strictly has to be eliminated. For that purpose a mechanical solution published in a patent specification was taken up and developed. In result there are optimized gearboxes which are capable of reducing the alternation to 8.4 %, respectively 3.3 % assuming an additional electronic speed control of the drive.

2. INTRODUCTION

Accessibility of sites is a growing problem in everyday life, especially in rural areas, where still no area-wide mobility support exists. If a person needs a mobility aid, the lack of suitability with respect to obstacles usually is a major problem. If steps or even stairs have to be managed, mobility aids available quickly reach their limits. To solve those problems technically, several solutions were examined. It became visible that vulnerabilities exist, which only become apparent in everyday practice. A promising idea is to use whegs, known from mobile robotics, for transportation of passengers. But this demand, however, urges to reduce or in the best case to eliminate the dynamic height offset occurring in wheg-driven mechanisms, called alternation. If there is success in minimizing alternation, an easily implementable and safely usable mobility system could be provided, with a feeling like riding on wheels, able to overcome small steps without having to worry about a possible 'stuck' or ruining the mobility aid.

3. APPROACHES

To keep the number of actuators and the expenses for sensor and control technology low (which generally has a positive effect on the cost and reliability of the mobility aid), a force coupled mechanical solution is proposed, which was motivated by the results of a patent search. In [1] a solution was published, which requires only one drive for the movement and height adjustment, called 'compensation of alternation'. However, this original version does not take into account that a certain 'residual' alternation remains, and that a vibration in the direction of travel occurs. The intensity of the disturbing phenomena depends on the type of transmission, as well as the number, dimensions and position of the gear elements. Fig. 1a shows the basic mechanism for height adjustment. In addition to the basic gearbox, a timing belt drive is superimposed to transfer the rotary motion to the wheg.

In the first step exemplary structural dimensions such as link lengths were selected. It was recognized that these dimensions appropriately chosen can lead to optimized design variants of the gearbox, but they also can make it unusable. Additionally, the clever choice of the type and sequence of the transmission stages, as well as the arrangement of all the members to each other generates the opportunity to further minimize alternation.
In the second step the design criteria concretion of possible gearboxes to compensate the alternation was addressed. On the one hand the best possible compensation of alternation, on the other hand, the most simple construction in focus, optimization of the main transmission was performed by systematic variation of the length and position parameters and the type and number of components, illustrated in figs. 2a to 4. The main difference consists in the number of gear stages. Other differences arise from the criteria mentioned. Considering the region of the path of the contact point (12; B) between touchdown and lift-off of the contact point (11), the maximum possible smoothing was worked out through simulations, cp. fig. 3. Taking into account a third attribute, namely the uniform travelling speed, the gearbox illustrated in fig. 4 is the result. But its use in mobility aid, however, is not possible due to the lack of ground clearance.

Fig. 1a: Technical principle of the basic mechanism for height adjustment

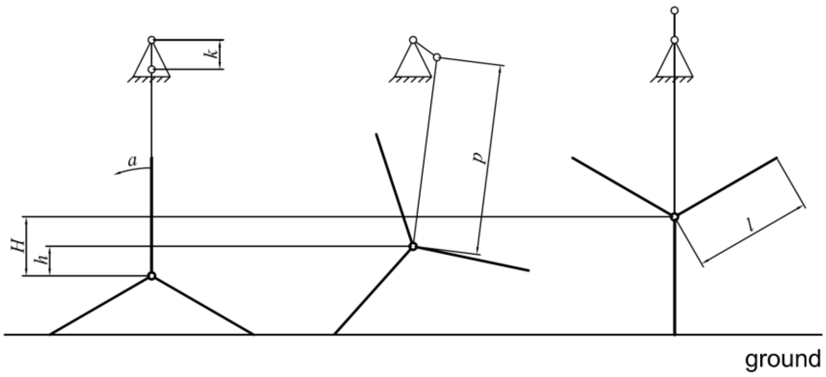
To describe the transmission parameters the equations, illustrated in fig. 1b, underlie.

$$h(\alpha) = H\left|\sin\left(\frac{z}{2}\alpha\right)\right| \qquad \{1\}$$

$$H = l\left|\cos\left(\frac{180°}{z}\right)\right| \qquad \{2\}$$

$$k = H/2 \qquad \{3\}$$

$$K(\alpha) = H - h(\alpha) \qquad \{4\}$$



with h = height offset of the axis depending on the angular position of the wheg, α = angular position of the wheg, z = number of spokes, H = maximum height offset, l = spoke length, k = crank length, p = couple length, K = compensation offset depending on the angular position of the wheg

Fig. 1b: Transmission parameters and equations

Eq. 1 represents the so-called alternation function which describes the curve of the alternation in dependence on the number of spokes z and the maximum height offset H. The maximum height offset H is calculated according to Eq. 2 assuming the knowledge of the spoke length l.

In order to determine the crank length k the maximum height offset H is halved, compare eq. 3. The required, rotating angle-dependent compensation K arises ultimately from eq. 4.

4. RESULTS

Whereas in the single-stage variant (cp. figs. 2a and 2b), the compensation of the alternation could be approximated to 91.6 %, it is 96.7 % (cp. figs. 3 and 4), corresponding to an almost complete compensation of the alternation in the two-stage variant.

Basic principle of all gearboxes is the mechanical positive coupling of the forward motion with the up and down movement, shown in fig. 2a. The impellent engages on the crank (10; K), on which a timing belt pulley (13a1; I) is mounted rigidly, on which again a timing belt (13a3; H) is running, whereby the rotary movement is transferred to a further timing belt pulley (13a2; F) with the z-fold (z: number of spokes) diameter. Timing belt pulley at the input of the gearbox (13a1; I) and timing belt pulley at the output of the gearbox (13a2; F) form, including timing belt (13a3; H) the primary drive, designed as traction drive with translation (13a). In the two-stage variant, which is shown in fig. 3, the function of the primary drive (13) is the change of direction (primary drive, designed as pair of gears to reverse the rotation direction). The necessary translation of the drive motion according to the number of spokes falls to the secondary drive, running as a traction drive with translation (15). Fig. 4 shows, however, the secondary drive, designed as traction drive (15a) as a module for the pure transmission of rotary motion (without translation). The axes of the primary drive are connected with a couple (8; G) which converts the rotary motion into a translatory motion, and allows the movement of the movable part of the linear bearing (9; C), and thus the rotational axis of the Wheg (16; D) and the rigidly associated Wheg (14; E). In figs. 2a and 2b the rotary motion of the crank (10; K) enters on the primary drive, designed as traction drive with translation (13a), in fig. 3 on the primary drive, designed as pair of gears to reverse the rotation direction (13), and the secondary drive, designed as traction drive with translation (15) and in fig. 4 on the primary drive, designed as traction drive with translation (13a) and the secondary drive, designed as traction drive without translation (15a) to the rotational axis of the Wheg (16) and the rigidly associated Wheg (14).

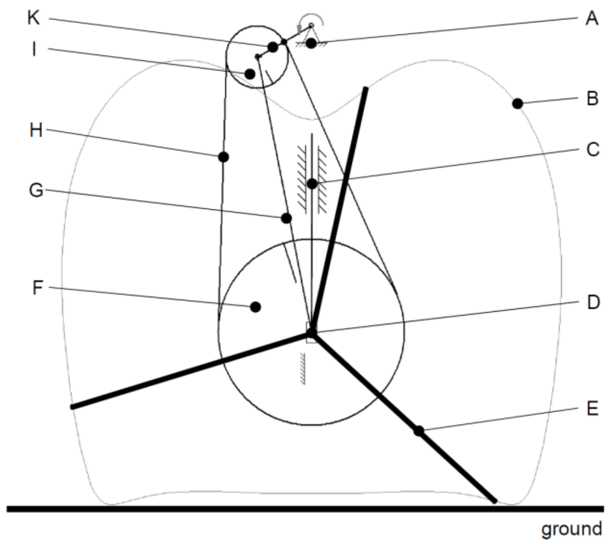
Design guidelines for the synthesis of optimized gearboxes arise as shown in Table 1. In order to generate the capability of overcoming high steps, the largest gap dimension, 120° offset spokes (3-spoke Wheg), was considered for exemplary determination of performance parameters of the optimized gearboxes, see Table 2. For this particular example, the spoke length l is 160 mm, resulting in a crank length k in the basic mechanism for the alternation compensation of 40 mm, see eqs. 2 and 3. The travelling speed amounts to 1 km/h or 278 mm/s.

Table 1: Design guidelines for optimized gearboxes

	gearbox **figs. 2a and 2b**	gearbox **fig. 3**	gearbox **fig. 4**
crank length k	$k = H/2$		
translation	$i = z$		
couple length p in mm	$p = 4{,}5k$	$p = 2{,}4k$	$p = 3k$
rotation direction reversal input/output	no	yes	no
translation located in	primary drive	secondary drive	primary drive

Table 2: Performance parameters of the optimized gearboxes

	Wheg **without** compensation of alternation	gearbox **fig. 2**	gearbox **fig. 3**	gearbox **fig. 4**
alternation respectively residual alternation in % and (mm)	100 (80)	8,4 (6,7)	3,3 (2,6)	8,3 (6,6)
maximum of acceleration in travelling direction in m/s^2	-0,6 to 0,6	-1,3 to 1,3	-2,1 to 2,1	-0,4 to 0,4
number of extrem values in stance phase	1	5	5	5
sufficient ground clearance	yes	yes	yes	no



A = crankshaft bearing
B = path of the contact point
C = linear bearing
D = rotational axis of the wheg
E = wheg
F = timing belt pulley (at the output of the gearbox with diameter $d_2 = z{\cdot}d_1$)
G = couple
H = timing belt
I = timing belt pulley (at the input of the gearbox with diameter d_1)
K = crank

I and K in rigid coupling

Fig. 2a: Technical principle of the coupling between the forward and the up and down movement; the trajectory of the spoke tips in the body frame is shown

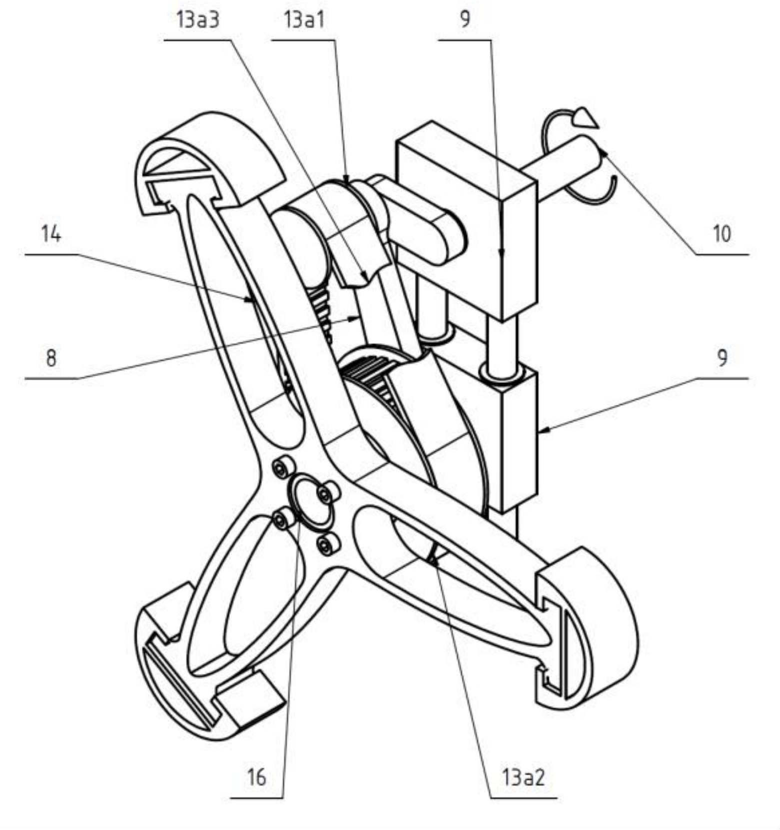


Fig. 2b: Technical design of the one-stage variant

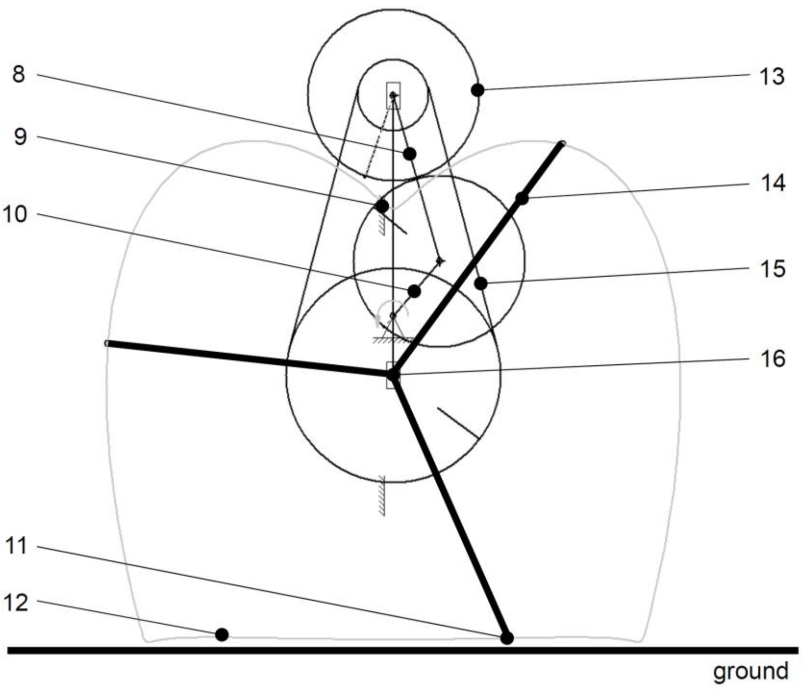


Fig. 3: Technical principle of the two-stage variant;
the trajectory of the spoke tips in the body frame is shown

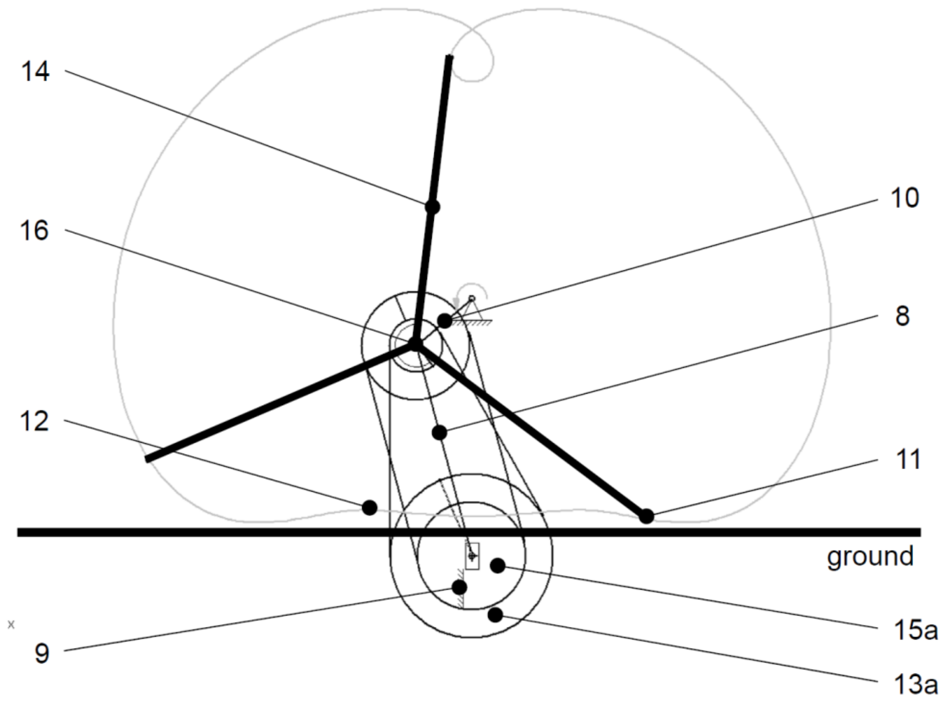


Fig. 4: Technical principle of the two-stage variant including uniform travelling speed; the trajectory of the spoke tips in the body frame is shown

For figures 2b to 4, the following legend applies.

8	=	couple
9	=	linear bearing
10	=	crank
11	=	contact point
12	=	path of the contact point
13	=	primary drive, designed as pair of gears to reverse the rotation direction
13a	=	primary drive, designed as traction drive with translation i=z (z - number of spokes)
13a1	=	timing belt pulley (at the input of the gearbox with diameter d_1)
13a2	=	timing belt pulley (at the output of the gearbox with diameter $d_2 = z{\cdot}d_1$)
13a3	=	timing belt (primary drive)
14	=	Wheg
15	=	secondary drive, designed as traction drive with translation i=z (z - number of spokes)
15a	=	secondary drive, designed as traction drive without translation
16	=	rotational axis of the Wheg

5. DISCUSSION

The gearboxes shown in figs. 2a to 3 have their respective advantages. The decision for the one- or two-stage variant depends on the weighting in reference to the requirements on the quality of alternation compensation, the characteristics of travelling speed and the construction itself. The fact that doubling the speed of travel quadruplicates the maximum acceleration at the contact point, and thus of the entire platform (model of infinite stiffness of the material assumed), invokes the necessity to produce countermeasures in the context of an agile mobility aid. Helpful in this case could be a speed compensation of the impellent (mechanical or electronic) or adequate modules (e.g. rounding of peaks by elasticities). When the desired vehicle speed is rather low and great whegs are possible, the latter elasticities to equalize the travelling speed are sometimes sufficient. With such elasticities, in combination with the single-stage version, which already compensates a great part of the alternation, one obtains a structurally simple, efficient and therefore cost-effective system. With the prioritization of the best possible compensation of alternation, the availability of powerful speed compensation and the related mechanics, the two-stage variant provides the best solution, cp. fig. 3.

6. CONCLUSION

Gearbox solutions were simulated and optimized with respect to various criteria. It became obvious that it is impossible to achieve the necessary running smoothness for the transport of people in both vertical and longitudinal direction only by optimization of the gearbox parameters, if the mobility aid should be agile. Using the separation of functions, that means, exploit the full optimization potential in terms of vertical axis and a separate compensation of the vibration in direction of travel (longitudinal direction), the 'residual' alternation could be reduced to 3.3 %. To compensate the vibration in the longitudinal direction a parallel-operating electronic speed control of the drive is proposed.

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ACKNOWLEDGEMENTS

The authors are members of the research group SilverMobility. This study was supported by the Development Bank of Thuringia and the Thuringian Ministry of Economic Affairs with funds of the European Social Fund (ESF) under grant 2011 FGR 0127.

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