# DESIGN AND DEVELOPMENT OF AN ACTIVE WHEELCHAIR WITH IMPROVED LIFTING KINEMATICS USING CFRP-COMPLIANT ELEMENTS

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## Abstract

A growing number of people worldwide are permanently or temporarily dependent on a wheelchair. In order to improve the quality of life for wheelchair-users, it is important that the wheelchair offers the greatest possible comfort on the one hand and on the other hand enables many everyday situations without assistance, while being light enough to be lifted into a vehicle by the user alone. In addition to the weight optimisation of the lift wheelchair, the main objective of the here-presented investigations is to achieve increased comfort for the wheelchair-user through more uniform actuation forces compared to the reference model. This is realised by an optimisation of the lifting kinematic and the use of flexible lightweight construction elements, also known as compliant elements, to overcome the aforementioned challenges.
This paper focusses on the gear synthesis and the parametric simulation study of the compliant elements enabling a profound understanding of the wheelchair components. Based on the derived results, the lifting mechanism was optimised and the new developed wheelchair was thoroughly tested. The experimental studies show a good agreement with the analytical and numerical results and confirmed the chosen design approach and the established development process.

## 1. Introduction

### 1.1. Motivation

In Germany, more than 1.5 million people are permanently or temporarily dependent on a wheelchair [1]. In general, wheelchairs are distinguished between passive and active wheelchairs. An active wheelchair allows people with limited mobility to have a larger range of movement and thus a significant increase in autonomy in the execution of simple everyday situations. With the aim of an operative and self-determined way of living, an active wheelchair improves participation in social life. Especially, wheelchairs with an additional lifting function offer a significant extension of the movement radius and an improvement of the comfort of the wheelchair-user. This technology enables the wheelchair-user to a continuous height adjustment of the sitting position, thereby expanding the action or gripping area into previously unachievable heights. For example, shelves can be reached at workplace and at home, as shown in Fig. 1, everyday situations, such as receiving food in the canteen made possible, and conversations can take place at eye level.
However, the few existing lifting mechanisms that are used are complex and weight-intensive. Due to the heavy mass in particular, wheelchairs with lifting function cannot be loaded independently by wheelchair-users in passenger cars, which in turn leads to reduced mobility. This results in an

application-specific improvement potential, which can be exploited by optimising the lifting mechanisms by i.e. using flexible lightweight construction elements, also known as compliant elements.

### 1.2. State of the art

The group of passive wheelchairs includes multifunctional, transport or positioning wheelchairs, as well as standard manual attendant-propelled wheelchairs. Passive wheelchairs are often equipped with integrated additional functions, such as an electric motor ("powerchairs") or an electrically operated seat unit with a standing or lying function. This makes these wheelchairs comparatively heavy (up to 200 kg).
The group of active wheelchairs (self-propelled manual wheelchairs) includes the adaptive and lightweight wheelchairs. In this case, the rolling resistance and the overall weight are minimised in such a way that an independent movement by the user is possible only with their muscular force.

**Figure 1.** Application of an active lift wheelchair at the workstation (Courtesy of the company Pro Activ Reha-Technik GmbH).

Only a few active wheelchairs are equipped with a lifting function, even if, as already mentioned, this offers a number of advantages in daily life, e.g. for personal well-being and for the inclusion of wheelchair-users in society. The main reason for this is the additional mass of the lifting kinematic, which leads to current lift wheelchairs with a weight over 19 kg having a mechanical lifting mechanism and a further 10-15 kg, with an electrical lifting mechanism. State of the art active wheelchairs are manufactured as tube constructions made of steel or aluminum tubes. Composite materials are only used in a few components such as fenders or fasteners made of carbon fibre-reinforced plastic (CFRP). The use of lifting mechanisms with flexible composite structures is currently not state of the art.
A characteristic feature of current wheelchairs with a mechanical lifting mechanism is the force that the user has to provide to support the lifting process. This force increases with the lifting height, whereby the arm is increasingly stretched. For more user comfort, the aim is to keep the necessary force as constant and low as possible over the entire lifting process. Furthermore, unpleasantly strong accelerations especially at the beginning and end of the lifting process should be avoided. Compliant structures made of FRP are known, e.g. from the automotive industry, where composite sheet springs for trucks have recently been developed [2,3]. The use of elastic FRP compliants with their anisotropic properties and configurable stiffness offers the possibility to improve the lifting mechanism [4-8].

### 1.3. Aim and outline of the paper

Aside from weight optimisation of the lift wheelchair the main goal of the current investigations is the achievement of an increased comfort for the wheelchair-user by means of more uniform actuation forces compared to the reference model. In this way even persons with stronger disabilities can use the mechanical lifting mechanism of the wheelchair. The innovation in the here-presented investigations is the combination of CFRP compliant elements into a four-bar mechanism to improve the lifting function at active wheelchairs. This paper presents the gear synthesis, the parametric simulation study of the compliant elements and the experimental proof-of-concept of the optimised wheelchair.

## 2. Concept and gear synthesis description

The kinematic requirements for the design of the lifting mechanism include horizontal movement of the wheelchair seat with minimal vertical displacement during the lifting process. The seat surface should not rotate during the entire lifting process, i.e. the seat surface may only move parallel to the initial position. Different gear concepts can be used to meet this requirement. For mass-related reasons, two different types of mechanisms are relevant: a four- and a six-bar linkage.
The selection of a four-bar lifting mechanism was based on the following reasons:

- manageable number of parts and small number of joints
- low bearing clearance and good tilting stiffness as well as a relatively simple design.

In order to examine the kinematic movement of the lifting mechanism (Fig. 2), the required objective function was derived mathematically.

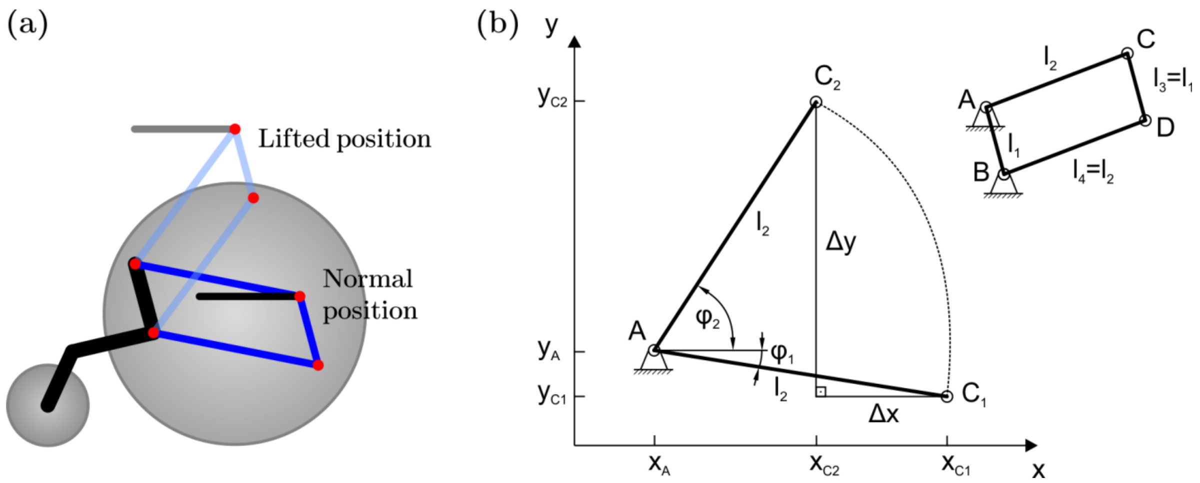


**Figure 2.** (a) Schematic drawing of the lift mechanism in the wheelchair and (b) sketch of the kinematics of the used four-bar linkage.

Having defined as target values the horizontal and vertical offsets $\Delta x$ and $\Delta y$, the trigonometric equations of the system are (variables $x_{C1}$, $x_{C2}$, $y_{C1}$ and $y_{C2}$ as well as the angles $\varphi_1$ and $\varphi_2$ are as defined in Fig. 2b)

$$\begin{aligned} x_{C1}(\varphi_1) &= x_A + l_2 \cos\,\varphi_1 \\ x_{C2}(\varphi_2) &= x_A + l_2 \cos\,\varphi_2 \\ y_{C1}(\varphi_1) &= y_A + l_2 \cos\,\varphi_1 \\ y_{C2}(\varphi_2) &= y_A + l_2 \cos\,\varphi_2 \end{aligned} \tag{1}$$

which leads to

$$\begin{aligned} \Delta x &= l_2\,(\cos\varphi_2 - \cos\varphi_1) \\ \Delta y &= l_2\,(\sin\varphi_2 - \sin\varphi_1)\,. \end{aligned} \tag{2}$$

By dividing the two equations from Eq. 2 by each other, we have

$$\frac{\Delta x}{\Delta y} = \frac{(\cos\varphi_2 - \cos\varphi_1)}{(\sin\varphi_2 - \sin\varphi_1)} \iff -\frac{\Delta x}{\Delta y} = \tan\frac{\varphi_2 - \varphi_1}{2}. \tag{3}$$

Based on Eq. 3, the relation between the rotation angle and the target values $\Delta x$ and $\Delta y$ are

$$\begin{gathered} \Delta\varphi = \varphi_1 + \varphi_2 \\ \Delta\varphi = 2\arctan(-\frac{\Delta x}{\Delta y}). \end{gathered} \tag{4}$$

For the calculation of $\varphi_1$ the following equation is formed, based on Eq. 2 and 4

$$\sin\Delta\varphi\cos\varphi_1 + (-1 - \cos\ \Delta\varphi)\sin\varphi_1 + \left(-\frac{\Delta y}{l_2}\right) = 0. \tag{5}$$

The upper swing arm length $l_2$ remains as a single optimisation parameter for the kinematic analysis. The general form (with generalised parameters *a*, *b* and *c*) of the Eq. 5 is

$$\mathrm{a}\cos\delta + \mathrm{b}\sin\delta + c = 0 \tag{6}$$

with the solution of the form

$$\delta = 2\arctan\frac{\mathrm{b} \pm \sqrt{a^2+b^2-c^2}}{a-c}. \tag{7}$$

Applying Eq. 7 on Eq. 5 gives the objective function for the optimisation as

$$\varphi_1 = 2\arctan\frac{-1-\cos\Delta\varphi \pm \sqrt{(\sin\Delta\varphi)^2+(-1-\cos\ \Delta\varphi)^2-(-\frac{\Delta y}{l_2})^2}}{\sin\Delta\varphi+\frac{\Delta y}{l_2}}. \tag{8}$$

With this objective function possible geometric variants of the four-bar linkage were investigated altering the geometric parameters, such as swing arm length $l_2$, horizontal and vertical offset $\Delta x$ and $\Delta y$, and rotation angle $\Delta\varphi$ within the limits of the available design space.

With the gear synthesis described above, it has been possible to achieve a significant improvement in the necessary vertical force for a large part of the lifting height compared to the previous force. Fig. 3 shows an excerpt from the parameter study about the influence of the swing arm length $l_2$ on the resulting vertical force ($l_{2,B}$ = 1.04 · $l_{2,A}$ , $l_{2,C}$ = 1.08 · $l_{2,A}$). Also the resulting force is as required almost constant up to a lifting height of approx. 250 mm it increases significantly afterwards. It is in this area where the compliant elements are to be used to further optimise the vertical force.

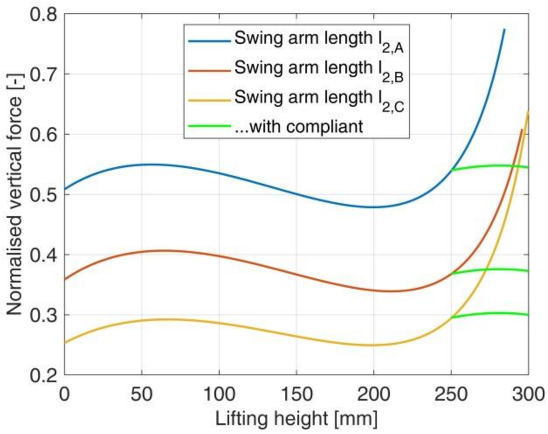


**Figure 3.** Parameter study of the resulting vertical force depending on the swing arm length $l_2$ with the potential advantageous use of CFRP compliant elements to reduce the force peaks at big lifting heights.

## 3. Parametric simulation study of the compliant elements

The compliant element is integrated into the developed wheelchair construction in such a way that it is only subjected to bending stress and thus compensates for excess forces or bending moments above a defined lifting height (Fig. 3). As a result, the element is loaded only in the lifted state of the wheelchair. For better production and assembly of the compliant element, its shape is assumed to be a straight bending beam. The element is designed analytically using the differential equation for pure bending (9) with: $\nu$ – bending line, $M_b$ – bending moment, $E$ – modulus of elasticity and $I$ – area moment of inertia of the bending beam.

$$\nu'' = -\frac{M_b}{EI} \tag{9}$$

The requirement for the compliant element is a one-sided twist of α = 12.5° with the other end supported by a transversal support (Fig. 4a). The parametric mathematical model (10) for the description of bending lines is solved using the software MATLAB with boundary conditions:
1) $\nu_{(x=0)} = 0.5 \cdot l_f \cdot \sin(\alpha)$, 2) $\nu'_{(x=0)} = \tan(\alpha)$ and 3) $\nu_{(x=l_c)} = 0$.

$$\nu(x) = -\iint \frac{M_b(x)}{E(x) \cdot I(x)} \tag{10}$$

The corresponding parameters to be taken into account in the design are: $l_f$ – clamping length, $l_c$ – unsupported bending length, $t_{1\text{-}3}$ – variable thickness of the bending beam (Fig. 4b).
The derivation of the analytical solution of Eq. 10 is dispensed within this publication and the corresponding literature on applied mechanics is referred to.

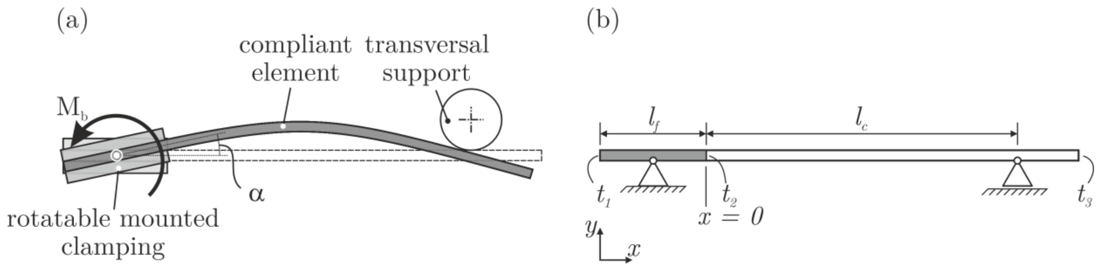


**Figure 4.** Simplified representation of the compliant element: (a) principle structure and (b) sketch of the mechanical model for the mathematical describe of the bending line.

The compliant element is designed so that the maximum material strain is 0.5 %. For a known bending line $\nu(x)$ from Eq. 10, the strain can be calculated as follows:

$$\varepsilon(x) = \frac{\nu''(x)}{(1 + (\nu'(x))^2)^{3/2}} \cdot \frac{t(x)}{2}. \tag{11}$$

The calculated material strain in the outer layer along the length $l_c$ of the compliant elements is shown in Fig. 5. Here are shown: the elongation curves of a bending beam with constant (Fig. 5a) and variable (Fig. 5b) laminate thicknesses. The selected variant with parameters $t_2 = t_1 = 1.05$ mm and $t_3 = 0.45$ mm was also analysed using FEM. This shows that the analytical and numerical solutions are very well matched both qualitative and quantitative. Thus, the analytical solution can be used for the pre-design of such compliant elements.

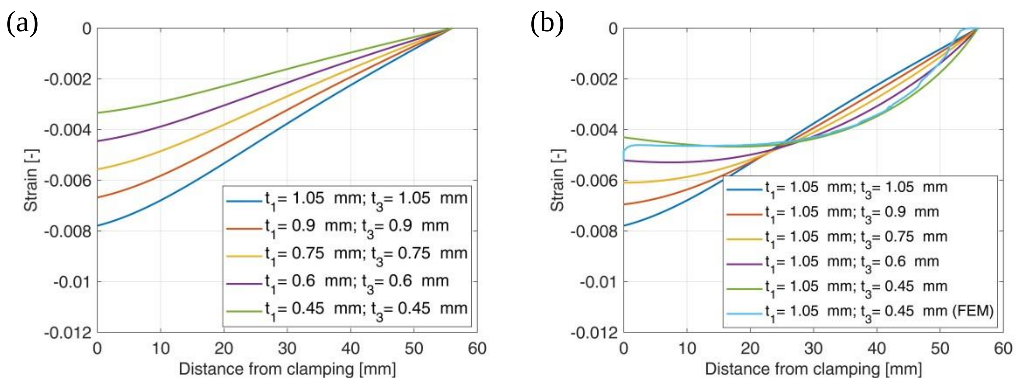


**Figure 5.** Material strain of the bending beam with (a) constant and (b) variable thickness of the bending beam with: $t_2 = t_1$, $l_f = 44$ mm and $l_c = 56$ mm.

As a result of the analytical and numerical optimisation process a non-symmetrical layup with seven layers of carbon fibre unidirectional (UD) fabric was selected for an optimal displacement and stress distribution in the compliant elements. The compliant elements (120 mm x 20 mm x $t$) were manufactured using the vacuum assisted resin infusion (VARI) process. Optimum compression of the fibre layers was guaranteed by a precisely adjusted pressure and temperature profile during production. The targeted fibre volume ratio $\varphi$ is 50 %. A multilayer composite of 0° oriented UD reinforcement layers with a HTS40 fibre and L-epoxy resin was used. The modulus of elasticity of the material used for the design is $E_{(\varphi = 50\,\%)} = 120$ GPa.

The static and dynamic two-point bending tests with the compliant elements were carried out on a universal testing machine (Co. Zwick, Fig. 6a). The clamping of the elements was realised by a force-locking clamping over the entire clamping length of 44 mm. Two strain gauges were applied to the tensile side of the surface; as close as possible to the clamping and at the analytically and numerically determined point of maximum bending stress located 22 mm from the clamping. The fin that moved vertically downwards during the test has a radius of 4 mm. The unsupported bending length between the fin and the clamping was 56 mm. The vertical force was recorded by the 10 kN load cell of the testing machine. The test setup was identical for the static and the dynamic (fatigue) tests. The results of the experimental analysis confirm the results obtained by FEM (Fig. 6b).

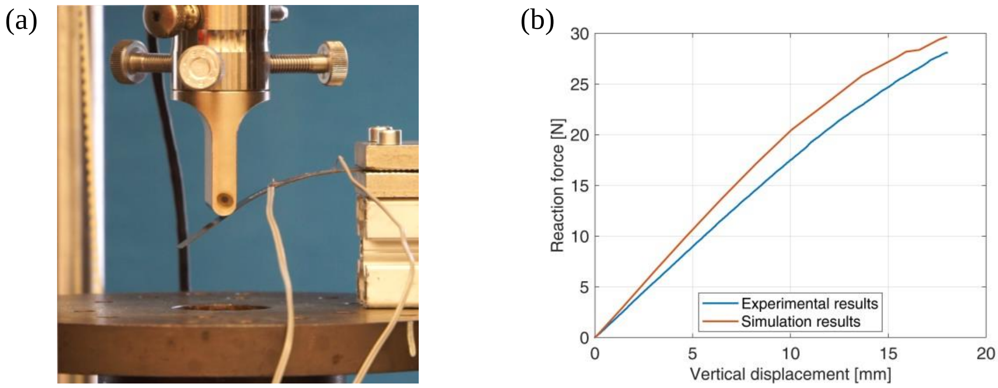


**Figure 6.** Analysis of the force-displacement behaviour of the compliant element: (a) test setup, (b) comparison between experiment and FEM.

## 4. Results – Experimental proof-of-concept

The measurement of the resulting vertical forces during the lifting and lowering process was also carried out with a universal testing machine (Co. Zwick). For this purpose, the frame construction (without drive, support wheels and without seat construction) was fixed in the lower test chamber of the testing machine and rollers were mounted on the seat retainer plates. While the lifting mechanism is actuated these can unroll on a plate mounted on the load cell of the testing machine (Fig. 7a). The machine performs the desired lifting movement under displacement control and the vertical upwards acting force is recorded as a function of the lifting height via the load cell of the testing machine. The displacement was measured by the testing machine, the test-speed was set to 15 mm/s and the vertical force was measured with a 100 kN load cell.

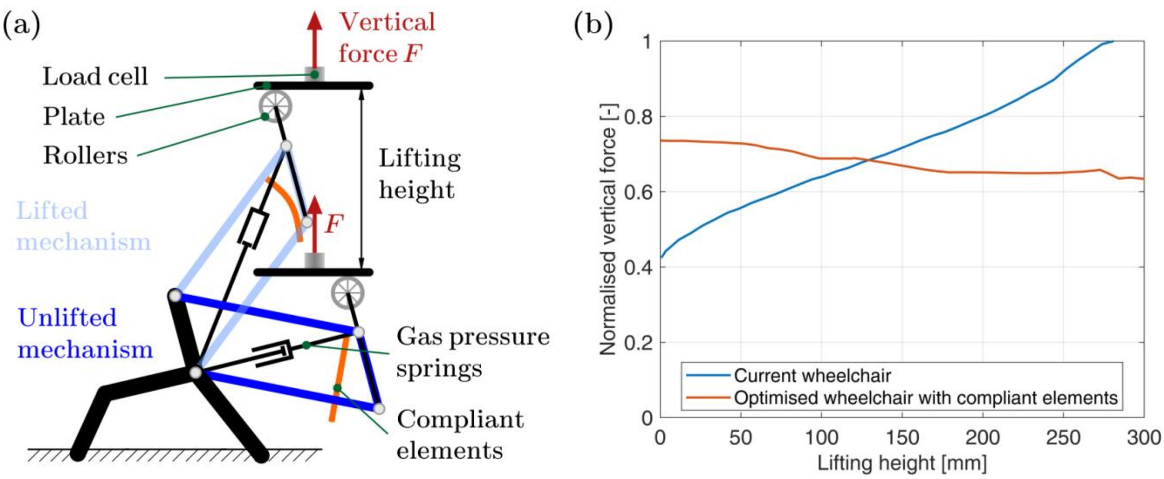


**Figure 7.** Experimental proof-of-concept of the optimised wheelchair: (a) schematic drawing of the test setup and (b) comparison of the vertical force versus lifting height of the current and the optimised active wheelchair with improved lifting kinematics and using CFRP compliant elements.

One of the most significant improvements of the developed new lifting kinematic and the use of CFRP compliant elements is the almost constant vertical force, which the user has to provide to support the lifting process (Fig. 7b). This significantly increases the comfort over the entire lifting height.
Due to component reduction, component substitution and topology optimisation the mass of the lifting kinematic has been reduced by 35 %. With this, the prerequisites were established for the total weight of the wheelchair to reach less than 12 kg. This enables the wheelchair-user to benefit from the advantages of the lift wheelchair without having to move more weight than with conventional wheelchairs. The lifting height is 300 mm with a minimal horizontal offset. These dimensions guarantee a secure standing of the wheelchair (tilt stability). Furthermore, by substituting maintenance-intensive components (bearings etc.), the maintenance interval can be doubled from two to four years.

## 5. Conclusion

The novel combination of CFRP compliant elements with aluminium rigid elements has improved the lifting mechanism by reducing the torque fluctuation range during lifting and lowering the seat. In addition, the comfort has now been increased by constant operating forces over the entire lifting height. All these measures reduce the physical stress on the wheelchair-user. The experimental studies show a good agreement with the analytical and numerical results and confirmed the chosen design approach and the established development process.

## Acknowledgments

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