

Description

[0001] The invention relates to the field of scissor lifts, in compact particular scissor lift platforms to carry large charges per surface area.

[0002] Scissor lift platforms are well known for selectively raising or lowering objects to facilitate their loading/unloading. For example, in the field of printing, scissor lifts are used inside "stacker units" to facilitate stacking of sheets of paper or booklets. The scissor lift platform is initially inserted in the stacker and receives sheets that have been processed in neighboring connected units (printed, folded, cut,...). The platform of the scissor lift starts in a high position (to be close to the paper output) and slowly lowers down upon accumulation of paper on the pile. Once fully loaded, the scissor lift is extracted from the stacker, and an operator needs to take the pile. For ergonomic reasons, the platform is raised to facilitate this operation. Once emptied, the scissor lift is reinserted in the stacker and a new cycle is started. The vertical movement of the platform is accomplished using a scissor arm mechanism.

[0003] Regarding the general principle of scissor lifts, they comprise at least two elongated members connected together at or close to their midpoint (like scissors) at a pivot point. Generally, two parallel sets are arranged for more stability. The upper end of the elongated members are supporting the platform of the lift, the lower end in a down position, the elongated members are close to a parallel horizontal position, while in the up-position, the elongated members are close to a parallel vertical position. The upper end of the elongated members are supporting the platform of the lift while the lower end of at least one elongated member is connected to a motor arranged to displace horizontally said end of the elongated element. The motor usually drives a horizontal lead screw imparting an horizontal movement to a crossbar (or transversal beam) connected to the lower ends of one of the elongated element, the end of the elongated elements can thereby be drawn closer to each other, which result in the platform lifting, or apart from each other, which results in the platform moving down.

[0004] Each elongated element can be prolonged by a further series of elongated elements also connected in their middle to increase the vertical displacement of the platform without increasing the footprint of the lift. Indeed, such scissor lifts need to be compact to fit into the stacker. As the stacker is one element of a chain of elements along which paper circulates, it is not possible to change its configuration.

[0005] Paper is heavy, and such scissor lifts need to be able to raise heavy loads per unit area. The mechanism of such lift is based on the transformation of a horizontal force into a vertical force. The highest force is required at the initiation of the upward movement. This initiation step is challenging. This is why the lift is never completely collapsed and usually a minimum angle of 8° is maintained.

[0006] Lifts have been developed where the mechanism is split in two phases, a lifting phase, for assisting the initiation phase, and a translation phase for finishing the upward movement.

[0007] US 9,310,753 explains the principle of such lifts mechanisms. To assist the initial phase of the upward movement of the lift, a spring assist assembly is arranged to, when the platform is in its lower position, exert a pulling tension which transmits to an arm equipped with a roller which pushes up the platform. Alternatively, instead of a spring, the assisting arm can be driven (pushed instead of pulled with the spring) by the front of a sliding carriage which imparts force on the assisting arm, up to a point where the rear of the sliding carriage comes into contact with the end of the elongated element and pulls it forward for driving the remaining upward movement of the lift. This assistance mechanism is typically effective for angles between 8° and 21° .

[0008] US2003/0075657A1 discloses a scissors lift with a lead screw (and actuator) connected to a carriage with rollers that are arranged to roll on a ramp while contacting a cam surface mounted on a first arm of the scissors upon the initial phase of the uplifting movement thereby imparting an upward movement to said arm. The carriage also supports a rod for imparting a horizontal movement to the second arm of the scissors. An offset mechanism (sliding link) on the carriage allows the rollers to move first, until reaching an abutment which allows the carriage to then action the rod or crossbar.

[0009] In this device, the carriage is rather bulky and takes significant space at the basis of the system. The position of the cams, close to the pivot points of the arms result in inefficient "lever-effect". Moreover, for high load, the forces exerted on such mechanisms are very high and the multiplicity of pieces result in a multiplication of weakness points of potential failure.

[0010] The applicant has therefore judged necessary to manufacture more robust mechanisms that can be very compact while allowing lifting of heavy charges.

Solution of the invention

[0011] To this purpose the invention relates to a lifting device including a base and platform placed over a scissors mechanism comprising a first arm with a traveling lower end and a second arm pivotably mounted on the base, both arms being joined together around their middle point by a pivot and describing an angle α with the base, and wherein the platform is movable from a down-position to an up-position upon action of an actuator arranged to action means

traversing the lower end of the first arm through a groove having a larger section than said means thereby allowing said means to slide within said groove, said means being pivotably connected to one end of a wedge mechanism having:

- a wedge arm (or wedge pivot linkage) arranged in a parallel plane to the arms of the scissors arms,
- a double-wheel roller being mounted on the other end of the arm,
- a lower cam being arranged on the base on the path of one of the roller wheel, and
- an upper cam being arranged on the second arm of the scissors lift, in the plane of the other roller wheel and substantially facing the lower cam so that the roller can contact both cams simultaneously.

[0012] Preferably, the means traversing the lower end of the first arm can for example be a transversal beam arranged perpendicular to the plane of the scissors arms, said beam being movable horizontally perpendicularly to its axis. The actuator is driving the movement of the traversing means. For example, the actuator is driving a horizontal movement of the transversal beam. The beam is not necessarily long. It could be part of another element, like a protrusion from a leadscrew.

[0013] Preferably, the length of the wedge arm is at least 50% lower than the length of the scissors arms. Preferably, the length of the wedge arm represents less than 35% of the length of the scissors are, preferably less than 30%. This allows an optimized lever effect.

[0014] The fact that the groove in the mobile scissors arm has a section that is larger than the section of the transversal beams allows the creation of an offset between the actioning of the wedge arm and the actioning of the scissors arm at the beginning of the upward movement, thereby creating an assistance mechanism to the lifting. The groove can advantageously have a circular shape or an elongated shape, but any shape, like for example a polygon, can be used, to the discretion of the person skilled in the art. The edges of the groove can be machined to minimize the friction of the transversal beam therein or can be covered with another material. The groove is further designated as "bushing" in the description, whatever its shape and whether it comprises or not an insert or an element of another material.

[0015] The means to prevent friction can alternatively or additionally be installed on the transversal beam.

[0016] The bushing can have any suitable shape, and can for example be circular, or a groove into which the transversal beam can slide. Such a groove can for example be a few millimeters for a few centimeters longer than the section of the bar, and is typically calculated to obtain the desired offset.

[0017] At the beginning of an upward movement of the lift, the scissors arms are close to a horizontal position (typically between 5 and 10° from the horizontal), the transversal bar is in contact with the bushing towards the extremity of the scissors arm, the roller of the wedge arm is resting on the base. Upon action of the actuator, the transversal bar moves horizontally so as to impart a movement to the wedge mechanism: one wheel of the roller rolls up the bottom cam thereby pushing onto the upper cam to impart a vertical upward movement to the second arm. Meanwhile, the transversal bar moves within the bushing without inducing a translation effect to the scissors arm.

[0018] Because the length of the wedge arm is smaller than half of the length of the scissors arms, the cams are positioned close to the extremity of the scissors arms, which allows a powerful lever-effect for lifting up the scissors and the platform. When the wheel of the roller reaches the top of the lower cam, and, preferably simultaneously the lower point of the upper cam, the transversal bar has reached the opposite end of the bushing and starts imparting a translational movement to the extremity of the scissors arm, ensuring a smooth continuation of upward movement of the platform.

[0019] The roller wheels are each free to rotate in both directions, depending on the upward or downward movement of the lift. They can run in opposite directions.

[0020] The lower cam has a substantially triangular profile with a cam base lying onto the base of the lifting device, a rising side arranged for supporting the rise of the rollers during the action of the wedge mechanism upon the lifting phase and a downwards side. One of the roller wheel rolls on the rising side of the cam during the initial phase of lifting.

[0021] The rising side of the lower cam can be linear. Preferably, a linear rising side makes an angle β_1 with the base (horizontal or resting side) comprised between 10 and 65°.

[0022] In an advantageous design, the rising side of the lower cam has a concave curved profile, i.e. a non-linear profile following a function of a least 1st order. This means that the rising side makes an evolutive (increasing) angle β_1 with the base. This angle preferably starts between 1° and 20°, preferably between 5° and 15°. This allows a low rising angle at the beginning of the assistance, to increase efficiency and a higher angle later in the assistance to keep the cam short enough.

[0023] Still more advantageously, the rising side of the lower cam has a hybrid profile comprising a concave curved section prolonged with a linear section. This allows a smoother backward movement of the roller when the lift goes down, and allows a slightly longer assistance than with a curved only profile.

[0024] The upper cam also has a substantially triangular profile pointing downwards with a cam base lying on and along the second arm (fixedly attached to this arm and parallel to it) of the lifting device, a downwards side arranged for contacting the rollers during the action of the wedge mechanism upon the lifting phase and a third side.

[0025] The downwards side of the upper cam can be linear. Preferably, a linear downwards side makes an angle β_2

with the second arm (its resting side) comprised between 10° and 65° .

[0026] In an advantageous design, the downwards side of the upper cam has a concave curved profile or slope, i.e. a non-linear profile following a function of a least 1st order. This means that the downwards side makes an evolutive (increasing) angle β_2 with the second arm. This angle preferably starts between 1° and 20° , preferably between 5° and 15° . This allows a low rising angle at the beginning of the assistance, to increase efficiency and a higher angle later in the assistance to keep the cam short enough.

[0027] Still more advantageously, the rising side of the lower cam has a hybrid profile comprising a concave curved section prolonged with a linear section. This allows a smoother backward movement of the roller when the lift goes down, and allows a slightly longer assistance than with a curved only profile.

[0028] The upper cam and the lower cams are arranged such that one of the roller wheel rolls on the downwards side of the upper cam, while the other of the roller wheel rolls on the rising side of the lower cam. A forward force is applied by the wedge arm onto the roller, which is converted into an upward force when the roller rolls on the rising side of the lower cam and the upward force is transferred to the downwards side of the upper cam and consequently to the second arm of the lift.

[0029] The scissors lift of the invention can be a single or multiple stage lift. This means that a scissors arm can comprise one or more segment, typically arranged in accordions. For example, the lift is a double stage lift, where each arm has two segments pivotally connected at one extremity.

[0030] The platform can be any suitable surface for supporting weights. Depending on the type of weights, it can be a plate, a plate comprising openings, a grid or have any other shape.

Detailed description of the invention

[0031] The invention will be better understood with reference to the detailed description of several modes of realization with reference to the drawing where:

Figure 1 is a schematic view of the basic principle of a scissor lift mechanism as known to a person skilled in the art;
 Figure 2 is a schematic view of a lifting mechanism of the invention;
 Figure 3 illustrates the equations relating to the cam shapes;
 Figure 4 illustrates the α dependency of equation 24 with Figure 4a: Left terms of Eq. 24 vs γ_2 obtained for $\beta_1=\beta_2=29^\circ34'$ and for various values of α_2 ; 4b: Measured evolution of γ_2 vs α_2 during the assistance phase; 4c: Left terms of Eq. 24 vs γ_2 obtained for $\beta_1=\beta_2=55^\circ$ and for the same various values of α_2 ;
 Figure 5 is a side view of a lower cam with a linear rising profile;
 Figure 6 is a side view of an upper cam with a linear downwards profile;
 Figure 7 is a side view of a lower cam with a curved rising profile;
 Figure 8 is a side view of an upper cam with a curved downwards profile;
 Figure 9 is a side view of a scissors lift according to the invention,
 Figure 10 is a detail of Figure 9;
 Figure 11 is a perspective view of the detail of figure 10;
 Figure 12 is a side view of the same scissors lift as figure 9, in an elevated position;
 Figure 13 is a detail of figure 12;
 Figure 14 is a perspective view of the detail of figure 13;
 Figure 15 is a side view of the scissors lift of figures 9 to 11, in a low position, without the wedge arm;
 Figure 16 is a side view of the scissors lift of figures 12 to 114, in a higher position, without the wedge arm;
 Figure 17 is a representation in perspective of a bushing suitable for the scissors lift of the invention;
 Figure 18 is a schematic representation of the forces involved in a two stages scissors lift of the invention
 Figure 19 is a side view of an upper cam with a curved downwards profile;
 Figure 20 is a side view of a guiding plate;
 Figure 21 is a side view of an upper cam with a curved downwards profile;
 Figure 22 is a side view of a scissors lift including a guiding plate;
 Figure 23 is a perspective view of the detail of figure 22.

[0032] Referring to figure 1, a lifting device 1 as known in the art includes a base 11 and platform 12 placed over a scissors mechanism 10 comprising a first arm 13 with a traveling lower end A and a second arm 14 with an end B pivotally mounted on the base, both arms being joined together around their middle point by a pivot 15, and wherein the platform 12 is movable from a down-position to an up-position upon action of an actuator, acting for example on a transversal beam (not represented) arranged perpendicular to the plane of the scissors arms.

[0033] Typically two sets of scissors arm are arranged in parallel such as end A and B of the arms of both sets form

a rectangle on the base 11 of the lifting device. This also means that the upper end of the arms define a rectangle for supporting the platform 12.

[0034] The transversal beam is arranged to, upon actuation, displace the traveling lower end A of the first arm towards or away from the end B of the second arm. For example, the transversal beam is coupled to a lead screw, horizontally arranged on the base of the lifting device, the screw being actioned by a motor.

[0035] The first arm 13 and the second arm 14 describe an angle α with the horizontal base 11. When the platform is completely downward, the angle reached is α_{\min} , and when the platform is at its maximum height, the angle reached is α_{\max} . Both arms have a length 2L. For example, $L=165$ mm and $\alpha_{\min} = 8^{\circ}26'$.

[0036] The weight W_a of the scissor lift 10 itself is acting on its centre of mass, while the load (for example a paper pile with weight $W=46$ Kg) is acting on one half at point 132 (upper end of first arm sequence) and the other half at point 142 (upper end of second arm sequence).

[0037] The total ground reaction $W+W_a$ is acting on one half at point A and the other half at point B.

[0038] The static condition is obtained if an external force F is applied essentially horizontally on the scissor arm pivot at point A.

[0039] The static condition can be derived by evaluating the forces acting on the first scissor arm 13 and by requesting that the total moment induced by those forces around the pivot 15 being zero.

[0040] The three forces acting on the scissor first arm 13 are:

- the ground reaction on point A whose component perpendicular to arm 13 being $\cos(\alpha)$. $(W+W_a)/2$ and thus the induced momentum around pivot 15 being $L \cdot \cos(\alpha) \cdot (W+W_a)/2$;
- the upper force $W/2$ acting on upper point 132 which gives rise to a force having a perpendicular component to arm 13 of $\cos(\alpha) \cdot (W)/2$ and thus a momentum around pivot 15 being $L \cdot \cos(\alpha) \cdot (W)/2$;
- the actuated force F whose component perpendicular to arm 13 being $\sin(\alpha) \cdot F$ inducing a moment around pivot 15 in opposite direction of $-L \cdot \sin(\alpha) \cdot F$.

[0041] Thus:

$$L \cdot \cos(\alpha) \cdot (W+W_a)/2 + L \cdot \cos(\alpha) \cdot (W)/2 - L \cdot \sin(\alpha) \cdot F = 0 \quad (\text{eq. 1})$$

$$\text{So: } F = \cot(\alpha) \cdot (W+W_a)/2 + \cot(\alpha) \cdot (W)/2 \quad (\text{eq. 2})$$

[0042] The required actuator force clearly grows almost exponentially as α is reduced following a law in $1/\tan(\alpha)=\cot(\alpha)$.

[0043] In our particular example, the maximal force required on the actuator without assist at $8^{\circ}26'$ is 6475 N, which is very high and requires a very powerful actuator.

[0044] Also, it is worth noting the value of the momentum around pivot 15 that the force created by the actuator must reach to maintain the static equilibrium. At $8^{\circ}26'$, the component for the force F directed perpendicular to the left bottom arm is $F \cdot \sin(\alpha)$ and the momentum around 15 at this lowest angle is

$$M = F \cdot \sin(\alpha) \cdot L = 950 \text{ N} \times 0.165 \text{ m} = 156.7 \text{ Nm} \quad (\text{eq. 3})$$

[0045] With reference to figure 2, where the elements of figure 1 are all present and pointed to with the same reference numbers, the lifting mechanism 20 further comprises a wedge arm (or wedge pivot linkage) 21 arranged in a parallel plane to the arms 13 and 14 of the scissors arms and having a length of less than half of the length of the scissors arms, i.e. less than L . On end A' of the wedge arm 21 is located close to the lower end A of the first arm 13. A double-wheel roller 22 is mounted on the other end O of the wedge arm 21. The double wheel roller 22 here has a smaller wheel 221 and a larger wheel 222 mounted parallel to each other on the same axis. A lower cam 23 is arranged on the base of the lift device, here on the path of the smaller wheel 221 of the roller, and an upper cam 24, which is arranged on the second arm 14 of the scissors lift, in the plane of the larger roller wheel 222. The upper cam 24 and the lower cam 23 are arranged substantially facing each other meaning that both wheels of the roller 22 can contact the cam located in the same plane simultaneously.

[0046] The end A' of the wedge arm (or wedge pivot) and the end A of the first arm (first arm pivot) of the scissors mechanism are mobile on the same horizontal plane, i.e. at the same height h_w above the base 11. They are also both actuated by the same transversal beam, thanks to an offset mechanism as will be explained in further details below. An xy referential is illustrated on Figure 2. The coordinate of wedge pivot A' and of the arm pivot A, noted $(A'_x; A'_y)$ and $(A_x; A_y)$ are therefore such that $A'_y = A_y$.

[0047] We will also denote $A'x=m$ and $Ax=x$. During a lifting action, starting from the lowest configuration of the platform ($\alpha=8^\circ26'$), the wedge pivot A' will start slightly further away from B than the arm pivot A such that $x>m$ but as it moves faster, A' will merge with A at the end of the assistance phase such that $x=m$, here for example when $\alpha=21^\circ$.

[0048] The wedge arm $A'O$ is essentially almost horizontal when the lift is at its lowest position but as A' moves along the x -axis, the wedge $A'O$ gets inclined and form an angle γ with respect to the horizontal line AB . A single transversal axis is mechanically linked to the actuator and passes by the point A' . During the assistance phase, the arm pivot A is not actioned by the transversal axis (arm pivot A is however not static, action of the wedge mechanism results in lifting up the scissors arms, thereby increasing angle α , which implies a horizontal displacement of arm pivot A). When this assistance stops (assistance phase finishes), arm pivot A and wedge pivot A' are merged and the same transversal axis acts on the arm pivot A .

[0049] A short transversal axis is installed at point O (e.g. with a bearing) and on this axis the first wheel 221 with radius r_1 is in contact with the bottom cam 23 and the second wheel 222 with radius r_2 is in contact with the upper cam 24. When the lift is raised, contact of the two wheels with the two cams is effective during the assistance phase where the wheel 221 with radius r_1 turns clockwise while the wheel 222 with radius r_2 turns counterclockwise. The active portions of the lower and upper cams are essentially the sets of all the points that will contact their respective wheel. The shape of those active parts of the bottom and upper cam can be linear or curved but for the illustration here, those active parts of the lower and upper cams are represented linear.

[0050] The lower linear cam 23 resting on the base is forming an angle β_1 with the horizontal (along the x -axis) that is fixed. The set of all points describing this lower cam can be expressed in the (x,y) referential so their coordinates are $P=(P_x;P_y)$.

[0051] The upper linear cam 24 is installed on the scissor lift second arm 14 and presents a fixed angle β_2 with respect to this line (the line of the arm). The angle of this upper cam with the ground is varying and amounts to $\beta_2+\alpha$. The position of the points Q of contact between wheel 222 and cam 24 in the (x,y) referential are $Q=(Q_x;Q_y)$ but for the sake of simplicity, another referential (u,v) located on the B pivot and rotating with the lift such that the u -axis is always aligned with the second scissors arm 14 will also be used. The Q coordinates in this other (u,v) frame will be denoted $Q=(Q_u;Q_v)$.

[0052] A simple matrixial change of coordinate operation can then be used to obtain the Q coordinates in the (u,v) referential from those in the (x,y) referential:

$$\begin{bmatrix} Q_u \\ Q_v \end{bmatrix} = {}^1_2R \left(\begin{bmatrix} Q_x \\ Q_y \end{bmatrix} + {}^1_2T \right) \quad (\text{eq. 4})$$

where the rotation change matrix from (x,y) referential to (u,v) referential is:

$${}^1_2R = \begin{bmatrix} \cos \alpha & -\sin \alpha \\ \sin \alpha & \cos \alpha \end{bmatrix} \quad (\text{eq. 5})$$

and the translation change matrix from (x,y) referential to (u,v) referential is:

$${}^1_2T = \begin{bmatrix} -2L \\ -h_W \end{bmatrix} \quad (\text{eq. 6})$$

[0053] The wedge length of the segment $A'O$ being L_W , the P and Q coordinates in the frame (x,y) are thus :

$$P_x = m + L_W \cos \gamma + r_1 \sin \beta_1 \quad (\text{eq. 7})$$

$$P_y = h_W + L_W \sin \gamma - r_1 \cos \beta_1 \quad (\text{eq. 8})$$

$$Q_x = m + L_W \cos \gamma + r_2 \sin(\beta_2 + \alpha) \quad (\text{eq. 9})$$

$$Q_y = h_W + L_W \sin \gamma + r_2 \cos(\beta_2 + \alpha) \quad (\text{eq. 10})$$

[0054] Eq. 7 and Eq. 8 are the two equations describing the shape of the bottom cam.

[0055] To obtain the shape of the upper cam thus in the (u,v) frame, combining Eq. 4, Eq. 9 and Eq. 10 gives:

$$Q_u = \cos \alpha [m + L_W \cos \gamma + r_2 \sin(\beta_2 + \alpha) - 2L] - \sin \alpha [L_W \sin \gamma + r_2 \cos(\beta_2 + \alpha)]$$

(eq. 11)

$$Q_v = \sin \alpha [m + L_W \cos \gamma + r_2 \sin(\beta_2 + \alpha) - 2L] + \cos \alpha [L_W \sin \gamma + r_2 \cos(\beta_2 + \alpha)]$$

(eq. 12)

[0056] The purpose of the wedge mechanism is to convert the available horizontal force into forces having substantial vertical component. This occurs when β_1 and $\beta_2 + \alpha$ are close to 45° . However, other design constraints must be included, like having here linear cam as well as favoring the faster movement of point A' compared to the movement of A.

[0057] The wedge mechanism so far described contains a particular degree of freedom that is the value of the γ angle that varies along with the variation of the lift α angle. So, there is a $\gamma = \gamma(\alpha)$ dependence. This increases the robustness of the system since any imperfection between the theoretical shape of the cams vs. the actual one or any imperfect positioning of the various element would simply result in a slightly different orientation γ taken by the wedge compared to the one designed.

[0058] For example, the following parameters can be chosen: $\beta_1 = \beta_2 = 38^\circ$, $r_1 = 14\text{mm}$, $r_2 = 12\text{mm}$, $h_W = 24\text{mm}$, $L_W = 52\text{mm}$ and $L = 165\text{mm}$, to illustrate how to design the wedge cam shapes by following two different approaches.

Designing the wedge mechanism by selecting the cam profile, for example a linear lower cam:

[0059] The first approach linearizes the lower cam 23 by selecting a particular $\gamma = \gamma_1(\alpha)$ law that leads to such a linear characteristic.

[0060] The different speeds of actuation of the pivot points A' and A will be resulting from this linear characteristic and the exact shape of the bushing in which the first arm pivot A moves will then have to be adapted to the particular resulting movement of those points.

[0061] The point O coordinates in the frame (x,y) are given by:

$$O = [m + L_W \cos \gamma_1 ; h_W + L_W \sin \gamma_1]$$

(eq. 13)

[0062] The equation of the line d2 parallel to the slope of the lower cam and passing by the point O is:

$$d_2 := y - r_1 \cos \beta_1 = (x + r_1 \sin \beta_1) + b$$

(eq. 14)

[0063] Where b is a constant.

[0064] Imposing that $O \in d_2$ gives:

$$(h_W + L_W \sin \gamma_1) - r_1 \cos \beta_1 = x + L_W \cos \gamma_1 + r_1 \sin \beta_1 + b$$

(eq. 15)

[0065] Rewriting it for b gives:

$$b = L_W(\sin \gamma_1 - \cos \gamma_1) - x + \text{const.}$$

(eq. 16)

[0066] With the x coordinate measured along the actuator stroke with the origin as depicted on Figure 1 being located at a 2L distance from the scissor lift fixed bottom right pivot, we get:

$$x = 2L (1 - \cos(\alpha))$$

(eq. 17)

[0067] Since x is a function of α according to Eq. 17, for b to be also a constant, a particular variation of γ_1 with α must be imposed:

$$(\sin \gamma_1 - \cos \gamma_1) = \frac{2L(1-\cos \alpha)}{L_W} + \text{const.} \quad (\text{eq. 18})$$

[0068] Then using the formula $\sin(a - b) = \sin a \cos b - \cos a \sin b$ And rewriting

$(\sin \gamma_1 - \cos \gamma_1) = \sqrt{2} \left(\frac{1}{\sqrt{2}} \sin \gamma_1 - \frac{1}{\sqrt{2}} \cos \gamma_1 \right)$, where $\frac{1}{\sqrt{2}}$ is once seen as $\cos \frac{\pi}{4}$ and once as $\sin \frac{\pi}{4}$,
an analytical expression for γ_1 can be derived:

$$\sqrt{2} \sin \left(\gamma_1 - \frac{\pi}{4} \right) = \frac{2L(1-\cos \alpha)}{L_W} + \text{const.} \quad (\text{eq. 19})$$

[0069] And so:

$$\gamma_1 = \text{Asin} \left[\frac{\sqrt{2}L(1-\cos \alpha)}{L_W} + \text{const.} \right] + \frac{\pi}{4} \quad (\text{eq. 20})$$

[0070] The value of the constant can be calculated stating e.g., that $\gamma_1=0$ for $\alpha=8^\circ 26'$, which gives:

$$\gamma_1 = \text{Asin} \left[\frac{\sqrt{2}L(1-\cos \alpha)}{L_W} - 0.7556 \right] + \frac{\pi}{4} \quad (\text{eq. 21a})$$

[0071] To be exact, the shape of the cam and the $\gamma_1(\alpha)$ variation dictate the $m=m(\alpha)$ variation but a good approximation is obtained taking a linear dependance of m with α such as for the example where the wedging pivot linkage has an horizontal displacement of 3 more mm than the scissor lift bottom arm pivot:

$$m = x - 3 \left(\frac{\alpha - 21}{8^\circ 26' - 21} \right) \quad (\text{eq. 21b})$$

[0072] This way, if the γ_1 and m evolution given by Eq. 21a and 21b are chosen, for $\beta_1 = \beta_2 = 38^\circ$, the shape of the bottom and upper cam can then be deduced using Eq. 7-8 and Eq. 11-12. Figure 3 illustrates these shapes, and in particular, Figure 3a represents $Y = \gamma_1(\alpha)$ law according to Eq. 20, Figure 3b illustrates the lower cam active shape, Figure 3c illustrates the upper cam active shape in (x,y) frame and Figure 3d illustrates the upper cam active shape in (u,v) frame.

[0073] The bottom cam is here indeed linear by construction given the γ_1 evolution of Eq. 21a. However, the upper cam is not linear in the (u,v) frame and could be machined directly with this particular shape or the linear approximation of it can also be used.

[0074] Figures 5 and 6 illustrate suitable profiles for a lower can and an upper cam with linear active slope or portion.

Designing the wedge mechanism by selecting the offset between scissors arm and and wedge arm

[0075] The second approach presented will consist in selecting the $Y=Y_2(\alpha)$ law that will lead the desired speeds of movement of point A' and A. If the resulting shape of the lower and upper cam are close to be linear, they will just be linearized to ease the parts fabrication while relying on the mechanism tolerance to a slight shape deviation via adaptation of the actual value taken by γ .

[0076] In this section, a particular $Y=Y_2(\alpha)$ dependance is selected so that the various position and speed of advance of pivots point A' and A have a selected offset as for example $x = m + 3$ at $8^\circ 26'$. As A' moves faster than A, A' will merge with A at the end of the assist such that $x=m$ at 21° .

[0077] The relationship between m and x (or between m and α , given Eq. 17 that relates x to α) will be established. Two points of the mechanism need to be established: first the point R depicted in Figure 2 located at the intersection of

the upper cam slope and the lift right lower arm, which has coordinates in the (u,v) frame of :

$$R = [R_u; 0] \quad (\text{eq. 22})$$

[0078] Eq. 22 implies indeed in a simple way that the point R rotates around the pivot point B, thus making the relation to the scissor lift in an easy way since $R_y=0$.

[0079] Then the point Q that is expressed in the frame (u,v) as $Q=(Q_u;Q_v)$, which makes the link with the wedge mechanism thus requiring Eq. 11 and Eq. 12 to be fulfilled.

[0080] The relationship between the wedge mechanism and the scissor lift can thus be expressed simply by noting the relation between these two points in the rectangle triangle with angle β_2 and vertex R and Q:

$$\frac{-Q_v}{Q_u - R_u} = \tan \beta_2 \quad (\text{eq. 23})$$

[0081] The relation between m and a is then obtained injecting Eq. 11 and Eq. 12 into Eq. 23:

$$\frac{-\sin \alpha [m + L_W \cos \gamma_2 + r_2 \sin(\beta_2 + \alpha) - 2L] - \cos \alpha [L_W \sin \gamma_2 + r_2 \cos(\beta_2 + \alpha)]}{\cos \alpha [m + L_W \cos \gamma_2 + r_2 \sin(\beta_2 + \alpha) - 2L] - \sin \alpha [L_W \sin \gamma_2 + r_2 \cos(\beta_2 + \alpha)] - R_u} - \tan \beta_2 = 0 \quad (\text{eq. 24})$$

[0082] For ease of resolution of Eq. 24, γ_2 is set to zero for the lowest α , which allows to find the R_u value that is useful for positioning the wedge mechanism with respect to the lift, although some setting can be introduced at the cam fixtures level.

[0083] Looking for a linearized cam solution, we only need to find the two extremities of the active cam sections.

[0084] A first prototype was built with wedge and cams corresponding to model values: $\beta_1 = \beta_2 = 29^\circ 34'$ with some minor deviations during fabrication as explained below:

- For $\alpha=8^\circ 26'$, $m=x-3$, which using Eq. 17 lead to $x=3.565\text{mm}$ and $m=0.565\text{mm}$.

$\gamma_2=0$ is set to zero, which allows to extract from Eq. 24 that $R_u=-321.8\text{mm}$.

Eq. 9-10 and Eq.11-12 then gives the theoretical first point for this lowest a value for the two cams:

$$P_x=59.5\text{mm}, \quad P_y=11.8\text{mm}, \quad Q_u=-268.5\text{mm} \quad \text{and} \quad Q_v=-30.2\text{mm}.$$

The values used to actually manufacture the parts where slightly differing and where:

$$P_x=61.2\text{mm}, \quad P_y=13\text{mm}, \quad Q_u=-268.5\text{mm} \quad \text{and} \quad Q_v=-30.2\text{mm}.$$

- For $\alpha=21^\circ$, $m=x$, which using Eq. 17 leads to $m=x=21.9\text{mm}$.

To find the γ_2 value at $\alpha=21^\circ$, The left term Eq. 24 is plotted vs. γ_2 in figure 4a.

There is sometimes no intersect with the γ_2 -axis such as for $\alpha=21^\circ$ so Eq. 24 is not entirely fulfilled. Due to the relative degree of freedom that γ_2 angle can take to cope any little deviation in the theoretical shape, equilibrium is observed on our first prototype before the left terms of Eq. 24 really reaches zero. For instance, for $\alpha=11.4^\circ$ the value of γ_2 measured in Fig 4b is 6.5° while Fig4a predicts about 10° .

The easiest way to find the second point to determine the shape of the two cams for the highest assisting ($\alpha=21^\circ$) value can thus avoid the resolution of Eq. 24 but rather simply fixing a certain γ_2 value at 21° .

Proceeding this way, at $\alpha=21^\circ$, for the first prototype a value of $\gamma_2=26^\circ$ and then using Eq. 9-10 and Eq.11-12 gives theoretically:

$$P_x=75.6\text{mm}, \quad P_y=34.6\text{mm}, \quad Q_u=-246.2\text{mm} \quad \text{and} \quad Q_v=-61.9\text{mm}.$$

The values actually used to manufacture the parts where slightly differing:

$$Px=77.3\text{mm}, Py=35.8\text{mm}, Qu=-248.5\text{mm} \text{ and } Qv=-60.9\text{mm}.$$

[0085] Figures 5 and 6 illustrate the profiles respectively for a lower cam and an upper cam of this prototype. In this example, for the lower cam between the start and the end of the assistance, $\Delta Px=77.3-61.2=16.1\text{mm}$ and $\Delta Py=35.8-13=22.8\text{mm}$. Similarly for the upper cam, $\Delta Qu=20\text{mm}$ and $\Delta Qv=30.7\text{mm}$.

[0086] The actual angle of inclination for the active part of these cam differs from the β_1 and β_2 values due to the movement of the complete mechanism (such as the wedge pivot A' advancing faster than the first arm pivot A of the lift, thanks to the presence of an enlarged bushing around the first arm pivot A through which the beam driving the movement of the wedge pivot A passes).

[0087] But besides this, the wedge mechanism must keep contact with the lift with the help of the cams. At the verticals passing with the contact points of the wedge wheels with the cam (located here for example about 100mm on the lift arms crossing 15), the lift moves up substantially.

[0088] The model above has been described for a simple one-stage lift; However, it is common to have several stages, the arms being arranged articulated, in accordion shape.

[0089] For example, in the case of a 2-stage lift, the height $z=2 \cdot (2L \sin \alpha)$ and using $L=100\text{mm}$, it can be deduced that from $\alpha=8^\circ 26'$ to $\alpha=21^\circ$, $\Delta z=85\text{mm}$. The γ evolution of 26° brings only for $L_W=52\text{mm}$ a contribution of $\Delta z=(L_W+18.4+3\text{mm}) \cdot \tan 26^\circ = 36\text{mm}$ so the remaining $\Delta z=49\text{mm}$ must arise from the cam themselves (in this case $\Delta z=22.8\text{mm}$ for the lower cam and Δz is about 26mm for the upper cam since $\Delta Qv=30.7\text{mm}$). Also, for the lower cam $\Delta x=16\text{mm}$ and for the upper cam $\Delta x=20\text{mm}$, which corresponds to actual physical angles of 55° and 60° respectively.

[0090] As can be seen in Fig. 4c, when $\beta_1=\beta_2=55^\circ$, there exist then always an intersect with the γ_2 -axis and so an exact solution for all the listed α value up to 21° . Thus, following the above procedure, if too small angles for β_1 and β_2 are selected, the actual resulting angles for the slope of the linear cam would have to be substantially higher to maintain a contact between the wedge mechanism and the upper cam during all the assisting phase, thereby proportionally negatively impacting the effective efficiency of the overall wedge mechanism. However, maintaining contact between the wedge and the cams during all the assisting phase is favored by large β_1 and β_2 values. This is the reason why for the second prototype described below, to reach higher assistance efficiency under drastic space constrains, non-linear bottom and upper cams were implemented.

Non-linear cams

[0091] A second prototype was manufactured with a wedge arm and cams where the linear active part of the cams was replaced with parabolas (second order equation). This means the apparent angles β_1 and β_2 are not constant during the assistance phase. The initial β_1 and β_2 angle values of the cams are kept very low to maximize the assistance efficiency for the lowest α angles whereas the cams present larger β_1 and β_2 values at the end of the assistance to maintain contact between the wedge and the lift. More precisely, taking for illustrative purpose $r_1=14\text{mm}$, $r_2=12\text{mm}$, $h_W=24\text{mm}$, $L_W=52\text{mm}$ and $L=165\text{mm}$, the linear slope was first calculated to ease the design, choosing $\beta_1 = \beta_2 = 42^\circ$. Rather than having the wedge contacting the cam slope with such a large angle for all lift positions, the bushing on the first arm (offset mechanism) was here selected with a larger circular groove such that the wedge pivot point A' is located 13mm away from the first arm pivot A at $\alpha=8^\circ 26'$ and then A' will merge with A at $\alpha=21^\circ$. The linear slope of the rising side of the cams is hereby replaced by a parabolic profile, starting with a very low angle β at $\alpha=8^\circ 26'$ that will increase up to $\beta=42^\circ$ angle at $\alpha=21^\circ$.

- For $\alpha=8^\circ 26'$, $m=x-13$, which using Eq. 17 lead to $x=3.565\text{mm}$ and so $m=-9.43\text{mm}$. Note that a linear slope solution that could serve as a guide to obtain the second order one is obtained here using Eq. 24 for $Ru=-301.7\text{mm}$.

The first point (P_{x1} , P_{y1}) of the lower cam parabola is found setting $\gamma_2=0$ at this α angle and by selecting that the wheel of the wedge roller is tangent to this parabola. Thus $P_{x1}=m+L_W=42.57\text{mm}$ and $P_{y1}=h_W-r_1=10\text{mm}$.

The first point (Q_{x1} , Q_{y1}) of the upper cam parabola is found setting $\gamma_2=0$ at this α angle and by selecting that the wheel of the wedge is tangent to this parabola. Thus $Q_{x1}=m+L_W=42.57\text{mm}$ and $Q_{y1}=h_W+r_2=36\text{mm}$. Using Eq. 4, this point in the (u,v) frame has for coordinates $Q_{u1}=-286.1\text{mm}$ and $Q_{v1}=-30.3\text{mm}$.

- For $\alpha=21^\circ$, $m=x=21.92\text{mm}$. The second point (P_{x2} , P_{y2}) of the lower cam parabola is given by $P_{x2}=m+L_W+r_1 \sin \beta_1=83.29\text{mm}$ and $P_{y2}=35.8\text{mm}$ (same height as the end point of the linear slope design).

The parabola for the bottom cam thus has an active part during assistance with equation:

$$y = a(x - P_{x1})^2 + P_{y1}, \quad \text{where } a = \frac{P_{y2} - P_{y1}}{(P_{x2} - P_{x1})^2} = 0.0156 \quad (\text{Eq. 25})$$

The second point (Q_{u2} , Q_{v2}) of the upper cam parabola is found setting i.e. $\gamma_2=26^\circ$ at this α angle in a similar way (it is also the end point of the linear slope design). So $Q_{u2}=-244.1\text{mm}$ and $Q_{v2}=-63.5\text{mm}$.

The parabola for the upper cam being rotated by $8^\circ 43'$ with respect to the u-axis, the equation of its active part is more easily expressed in parametric coordinates. Indeed, setting $x(t)=t-Q_{x1}$ and $y(t)=-b(t-Q_{x1})^2+Q_{y1}$, where

$$b = \frac{Q_{y2} - Q_{y1}}{(Q_{x2} - Q_{x1})^2} = 0.012$$

Using Eq.5 with $\alpha=8^\circ 26'$, the parametric equations for the active part of the upper cam in the (u,v) referential are then:

$$u(t) = (t - Q_{x1}) \cos 8,43 + (b(t - Q_{x1})^2 + Q_{y1}) \sin 8,43 \quad (\text{Eq. 26})$$

$$v(t) = -(t - Q_{x1}) \sin 8,43 + (b(t - Q_{x1})^2 + Q_{y1}) \cos 8,43 \quad (\text{Eq. 27})$$

[0092] Figure 7 depicts the shape of the lower cam and figure 8 illustrates the profile of the upper cam manufactured for this second prototype.

[0093] With reference to figures 9 to 16, the operational application of the wedge mechanism of the invention will be detailed, using the cams of the second prototype above for illustration. A lifting device 101 as known in the art includes a base 111 and platform 112 placed over a scissors mechanism 110, here a two stages mechanism, comprising a first arm 113 with a traveling lower end 123 and a second arm 114 with an end 124 pivotably mounted on the base, both arms having here for example a distance between their pivots of $2L=330\text{mm}$. Total length of arm 114 is for example 356 mm, while total length of arm 113 is for example 366mm (this arm being longer as a consequence to the presence of the bushing). These arms being joined together around their middle point by a pivot 115, and wherein the platform 112 is movable from a down-position (figure 9) to an up-position upon action of an actuator on a transversal beam 116 arranged perpendicular to the plane of the scissors arms.

[0094] The transversal beam 116 is arranged to, upon actuation, displace the traveling lower end 123 of the first arm towards or away from the end 124 of the second arm. For example, the transversal beam is coupled to a lead screw, horizontally arranged on the base of the lifting device, the screw being actioned by a motor.

[0095] The transversal beam 116 is traversing the lower end 123 of the first arm 113 through a bushing 125, which is a hole of elongated section allowing the transversal beam to slide within said bushing. A suitable bushing is for example illustrated in figure 15. The end of the transversal beam 116 is pivotably connected to one end 151 of a wedge mechanism 150 having:

- a wedge arm 152 (or wedge pivot linkage) arranged in a parallel plane to the arms 13 and 14 of the scissors arms and having here a total length of 83mm with a corresponding inter-axis distance $A'O=55\text{ mm}$, indeed less than half of the length of the scissors arms ($L=165\text{mm}$),
- a double-wheel roller being mounted on the other end of the arm 152, with an external wheel 154 and an internal wheel 153,
- a lower cam 155 being arranged on the base 111, on the path of the external roller wheel 154, and
- an upper cam 156 being arranged on the second arm 114 of the scissors lift, in the plane of the internal roller wheel 153. In the lowest lift position, the upper cam and the lower cam are next to each other so that the internal wheel 153 and the external wheel 154 of the roller are in contact with a cam.

[0096] When the platform is at its lowest position, the transversal beam 116 passes through the bushing 125 of the lower end 123 of the first arm, close to the left side of the bushing. To start lifting the platform, the transversal beam 116 is actioned towards the right. At the beginning of the movement (assistance phase), the transversal beam 116 slides freely within the bushing, without imparting any force to the first arm. For example, the bushing allows here the transversal beam to run 13 mm between both extremities of the bushing. When the beam reaches the second bushing extremity on the right, it occupies the pivot position A of the first arm. The transversal beam applies however a lateral force to the wedge arm 152. The external wheel 154 of the roller starts to roll onto the lower cam 155 rising profile. The associated internal wheel 153 of the roller rolls along the upper cam 156, imparting to said upper cam an upward reaction force which leads to the second arm 114 (to which the upper cam is fixed) being lifted up.

[0097] For an angle corresponding to $\Delta\alpha = 21^\circ - 8^\circ 26' = 12^\circ 34'$ the pivot point of the wedge arm (beam 116) will make a course that is here 13mm longer than the course of the pivot A of the first arm. The arc length inside the bushing groove is thus 13mm (as illustrated on figure 17) and the corresponding circle that present such an arc length for an angle of $\Delta\alpha$ has a radius given by: $R_{bushing} = 13\text{mm} / \Delta\alpha = 59.3\text{mm}$. The horizontal projection of this groove arc is $R_{bushing} \cdot \sin(\Delta\alpha) = 12.9\text{ mm}$.

[0098] When the wheels of the roller have reached the top of the cams, the assistance phase is finished, the transversal beam moving forward has reached the pivot point A of the first arm 113 and will now apply a force directly on this arm, according to the classical principle of scissors lifts. The wedge arm and its roller is allowed to roll freely onto the base 111.

[0099] During the lowering phase of the platform 112, the transversal beam 116 controls the first arm 113 during the whole downward movement, except for a first phase corresponding to the beam 116 moving within the bushing 125, from one side to the other. When reaching low α angles, the roller of the wedge mechanism simply rolls over the cam backwards.

[0100] With reference to figure 18, the forces involved for a two stage scissors lift according to the invention will now be detailed.

[0101] The load W is divided into one half (W/2) applied at point E (below one extremity of the platform) and the other half at point F (below the other extremity of the platform). The weight of the scissor lift itself W_a is acting on the lift centre of mass. The total weight (W+ W_a) causes a ground reaction directed upward and distributed for one half, (W+ W_a)/2, at point A' (pivot point of the wedge mechanism associated to the first arm of the scissors lift, extremity of the transversal driving beam) and the other half at point B (pivot point of the second arm) of the. The lift is transmitting a horizontal force F_{By} at the fixed pivot point B, which causes a reaction of equal amplitude but opposite direction. The actuator applies a horizontal force at point A' denoted here F_{screw} . D is the articulation of the two-stages first arm, and G is the articulation of the two-stages second arm. To list all the forces acting on the AD and the BG lower arms, it is also shown that the lift upper stage applies vertical W/2 forces at points D and G, as well as horizontal forces that amount to $W \cot \alpha$. The lift lower stage induces opposite reactions also at D and G, having the same amplitude but opposite force. The wheel with radius r_1 mounted on the transversal axis passing by the O point of the wedge roller is applying a force perpendicular to the lower cam at point P, which is decomposed into a horizontal part F_{Px} and a vertical F_{Py} part. The lower cam reaction on this wheel creates forces equal and opposite in direction. Similarly, the wheel with radius r_2 mounted on the same transversal axis passing by the O point of the wedge roller is applying a force perpendicular to the upper cam at point Q, which is decomposed into a horizontal part F_{Qx} and a vertical F_{Qy} part with its equal and opposite reaction from the upper cam.

[0102] All the reaction forces acting on the wedge alone that is also at static equilibrium are considered.

[0103] The sum of the horizontal forces acting on the wedge being zero, $\Sigma F_x = 0$, so:

$$F_{screw} - F_{Px} - F_{Qx} = 0 \quad \text{and thus } F_{Qx} = F_{screw} - F_{Px} \quad (\text{Eq. } 26)$$

[0104] The sum of the vertical forces acting on the wedge being zero, $\Sigma F_y = 0$, so:

$$-F_{Py} + F_{Qy} = 0 \quad \text{and thus } F_{Py} = F_{Qy} \quad (\text{Eq. } 27)$$

[0105] The geometry of the lower cam also gives:

$$\frac{F_{Px}}{F_{Py}} = \tan \beta_1 \quad \text{and thus } F_{Px} = F_{Py} \tan \beta_1 \quad (\text{Eq. } 28)$$

[0106] And similarly, the geometry of the upper cam gives:

$$\frac{F_{Qx}}{F_{Qy}} = \tan(\beta_2 + \alpha) \quad \text{and thus } F_{Qx} = F_{Qy} \tan(\beta_2 + \alpha) \quad (\text{Eq. } 29)$$

[0107] Combining Eq. 26 and Eq. 28 gives:

$$F_{Qx} = F_{screw} - F_{Py} \tan \beta_1 \quad (\text{Eq. } 30)$$

[0108] Then combining Eq. 27 and Eq. 30 gives:

$$F_{Qx} = F_{screw} - F_{Qy} \tan \beta_1 \quad (\text{Eq. 31})$$

[0109] Finally, combining Eq. 29 and Eq. 31 allows to solve for F_{Qy} :

$$F_{Qy} = \frac{F_{screw}}{\tan \beta_1 + \tan(\beta_2 + \alpha)} \quad (\text{Eq. 32})$$

[0110] Now that F_{Qy} is known, F_{Qx} can be deduced using Eq. 31:

$$F_{Qx} = \frac{F_{screw} \tan(\beta_2 + \alpha)}{\tan \beta_1 + \tan(\beta_2 + \alpha)} \quad (\text{Eq. 33})$$

[0111] The total amplitude of FQ being simply $F_Q = \sqrt{(F_{Qx})^2 + (F_{Qy})^2}$, we can obtain $F_{Q\perp}$ that is the useful component of FQ being directed perpendicularly to the BG lower arm, which is able to induce a momentum around the lift pivot point C. Since by geometry $F_{Q\perp} = F_Q \cos \beta_2$, we can write:

$$F_{Q\perp} = \frac{F_{screw} \cos \beta_2}{\tan \beta_1 + \tan(\beta_2 + \alpha)} \sqrt{1 + (\tan(\beta_2 + \alpha))^2} \quad (\text{Eq. 34})$$

[0112] For the first prototype (linear cam), with $\beta_1 = \beta_2 = 29^\circ 34'$ and at the lowest position $\alpha = 8^\circ 26'$, Eq. 34 gives $F_{Q\perp} = 0.82 F_{screw}$. The useful distance L_\perp at $\alpha = 8^\circ 26'$ for this prototype is measured to be 96.1mm. We need to obtain the same momentum around point C of 156.7 Nm given by Eq. 3 for the case without assistance and with $F_{screw} = 6475\text{N}$. When the wedge mechanism of the invention is present, that momentum amounts to $M = F_{Q\perp} \cdot L_\perp$ and is obtained with $F_{screw} = 1992\text{N}$. A theoretical assistance efficiency at the lowest lift position therefore allows an actuator force reduction by 69%.

[0113] In practice, some approximations having been made, the gain in efficiency can be lower, depending on how much the actual cam profiles differ from the theoretical model.

[0114] Indeed, there is a discrepancy as already stated between the angle β_1 and β_2 used to generate the shape of the cams and their physical real angular dimensions. To derive the real wedge assistance efficiency, the real resulting angular dimensions should be used (those ensuring contact between the wedge, the cams and the lift during all the assistance phase). Thus, for the first prototype, the physical slope of the bottom cam (55°) and the one of the upper cam (60°) as they were fabricated should be used. Eq. 34 then gives $F_{Q\perp} = 0.34 F_{screw}$, L_\perp at $\alpha = 8^\circ 26'$ is still 96.1mm but the momentum of 156.7Nm is then obtained with $F_{screw} = 4745\text{N}$ so a real assistance efficiency at the lowest lift position gives an actuator force reduction of 27% for this first prototype with linear cams.

[0115] Lifting and lowering time were measured on the first prototype and are gathered in Table 1:

Table 1

Wa (kg)	W (kg)	time up (s)	time down (s)
4	6	12,4	11,4
4	14	13,1	10,8
4	16	13,3	10,4

[0116] The time to lift the weight upward is of course slightly larger than to lower it down.

[0117] The first prototype with linear cams allowed to lift weights up to 36 kg.

[0118] It appears from Eq. 34 that to maximize $F_{Q\perp}$ and so the efficiency of the wedge assistance, β_1 and β_2 should ideally be as small as possible. However, if β_1 and β_2 are too small, the wedge will not be able to keep contact with the lift and upper arm until $\alpha = 21^\circ$. Therefore, a non-linear cam was built for the second prototype with the initial portion of

the cam having small β_1 and β_2 values to increase the efficiency at low α value. The cam then evolves towards larger β_1 and β_2 values to always keep contact between the wedge and the upper cam. Such cam profiles allow to reach efficiencies close to the theoretical efficiency calculated above. As a consequence, suitable actuators, for example motors, can be much less powerful than the ones needed for existing scissors lift. This also results in consequent energy savings.

[0119] To combine the properties of linear cams and curved cams, another prototype has been manufactured. With reference to figure 19, an upper cam 180 has a downward slope which is a second order curve as illustrated on figure 8 along a portion of its length (left of the vertical pointed line). The end of the slope (right of the vertical pointed line) is however further machined to a straight line.

[0120] A first beneficial effect observed is a smoother backward movement of the roller wheels when the lift is moving down, especially when the contact between the cams and the wheels is reestablished.

[0121] Lifting tests have been performed under the same conditions as previously described, which further demonstrate a beneficial effect of such a profile of the loading of the platform. Time for lifting up and down various charges with this hybrid profile for the upper cam are gathered in Table 2.

Table 2

Wa (kg)	W (kg)	W+Wa (kg)	time up (s)	time down (s)
4	6	10	12,4	11,5
4	14	18	12,8	11,5
4	16	20	13,5	11,5
4	24	28	14,3	11,5
4	32	36	14,8	11,5
4	34	38	15,2	11,5
4	36	40	15,7	11,5
4	38	42	16,2	11,5
4	40	44	16,5	11,5
4	42	46	17,2	11,5
4	44	48	17,7	11,5
4	46	50	19,8	11,5

[0122] The slight machining of the upper cam has allowed the system to lift total weights up to 50 kg.

[0123] Another prototype has also been manufactured combining also the properties of linear and curved cams as illustrated in figures 20 to 23. While the previous prototype above was offering the largest assistance at the lower position ($\alpha=8^\circ 26'$), this other prototype allows a more distributed efficiency, to cover also the end of the assisting phase ($18^\circ < \alpha < 21^\circ$). Thus, to obtain a large and well distributed assisting efficiency over the all assisted region ($8^\circ 26' < \alpha < 21^\circ$), the same procedure was followed but with an even larger offset mechanism and so a larger circular groove such that the wedge pivot point A' is now located 29 mm away from the first arm pivot A at $\alpha=8^\circ 26'$ and then A' will merge with A at $\alpha=21^\circ$. A linear lower cam 155 was used for the active part during the assistance as shown in Fig. 20. With reference to figure 21, an upper cam 156 has a downward slope which is a second order curve (parabola) as illustrated along a portion of its length (left of the vertical line). The end of the slope (right of the vertical line) is however further machined to a straight line (for a smoother backward movement of the roller wheels when the lift is moving down, especially when the contact between the cams and the wheels is reestablished). Like this, the combination of a larger offset and a linear lower cam facing a parabolic upper cam permits for instance to reach a similar level of good assistance this time over the whole assisted region.

[0124] Also, as a variant, rather than a complex-shaped bushing 125 with a circular groove made of a low-friction material fitting into the arm 123 to have low-friction contact with the axis 116 carrying the wedge mechanism which then becomes the pivot pin of the elevator at the end of the assist, an equivalent situation can and has been achieved to facilitate/simplify the manufacture of the bushing. This involves creating the required circular groove directly within the arm 123 and fitting a simple cylindrical bushing of low-friction material around the axis 116. In this way, the axis 116, surrounded by its bushing, can always move in the circular groove created in the arm 123. This has the advantage of facilitating the manufacture of the bushing 125, which can then be produced by simple turning, postponing the manufacture

of the circular groove in the arm 123, which can be produced on a multi-axis milling machine.

Table 3

Wa (kg)	W (kg)	W+Wa (kg)	time up (s)	time down (s)
4	32	36	13,8	12,2

[0125] As shown in table 3, even faster total lifting time was achieved with no sign of failure even when repeated total up ($\alpha=50^\circ$) and down ($\alpha=8^\circ 26'$) lifting of 36Kg over 50 000 cycles.

[0126] Optionally, and as visible on figures 20, 22 and 23, a guide 158 for the roller wheel can be implemented. Indeed, although the wedge system 152 carrying the wheels 153 and 154 moves in principle parallel to the arm 123, falling back under its own weight at the end of the assist phase and positioning itself correctly in the mechanism described in the previous examples at each level taken up by the elevator, an even more robust and malfunction-free result in the event of heavy loads can also be obtained by adding a guide guiding the wheel 154. Such guide can be designed to delimitate, above and at the vertical of the lower cam, the top curve below which the roller wheel will evolve. This allows to force the roller downwards at the end of its travel over the operational section of the lower cam during lift up, and thereby prevents the unlikely blockage of the roller in a position where it would be problematic during the downward operation of the lift. The guiding groove is always larger than the diameter of the wheel 154, such that this guide does not take up any force, but can help to correctly pre-position this wheel 154 outside the phases when the assistance mechanism is actually in action.

[0127] This guide can further comprise a lateral wall 157, illustrated on figure 23 as a vertical pane bridging the guide and lower cam, at the exterior side of the lift. This lateral wall preferably extends externally to the lift mechanism, in a parallel plane to the cams, over at least a surface covering the path of the roller wheel. This lateral wall 157 protects the roller wheel from dust and further acts as a safety shield to prevent any exterior element from penetrating the lift in this sensitive area, including fingers or hair or any other element. This lateral protection could also be implemented without the guide. These two elements are not necessarily combined.

[0128] Such lifting devices can advantageously be used in stacking machine for paper, as the overall mechanism is very compact.

Claims

1. Scissors lift (1; 101) including a base (11; 111) and platform (12; 112) placed over a scissors mechanism (20; 110) comprising a first arm (13; 113) with a traveling lower end (A; 123) and a second arm (14; 114) pivotably mounted on the base (11; 111), both arms being joined together around their middle point by a pivot (15; 115) and describing an angle α with the base, and wherein the platform is movable from a down-position to an up-position upon action of an actuator arranged to action means (16; 116) traversing the lower end (A) of the first arm (13; 113) through a groove or bushing (125) having a larger section than said means thereby allowing said means to slide within said groove or bushing, the said means being pivotably connected to one end (A') of a wedge mechanism (150) having:

- a wedge arm (21; 152) arranged in a parallel plane to the arms (13, 14; 113, 114) of the scissors arms,
- a double-wheel roller (22; 153, 154) being mounted on the other end (O) of the arm,
- a lower cam (23; 155) being arranged on the base (11; 111) on the path of one of the roller wheels (221; 154), and
- an upper cam (24; 156) being arranged on the second arm (14; 114) of the scissors lift, in the plane of the other roller wheel (222; 153) and substantially facing the lower cam (23; 155) so that the roller can contact both cams simultaneously.

2. Scissors lift according to claim 1, wherein the means traversing the lower end (A) of the first arm (13) is a transversal beam arranged perpendicular to the plane of the scissors arms, said beam being movable horizontally perpendicularly to its axis.

3. Scissors lift according to one of claim 1 or 2, wherein the length of the wedge arm (21) is at least 50% lower than the length of the scissors arms.

4. Scissors lift according to one of the previous claims, wherein the two wheels (221, 222) of the double-wheel roller (22) are parallel.

EP 4 393 863 A1

5. Scissors lift according to one of the previous claims, wherein the two wheels (221, 222) of the double-wheel roller (22) have a different diameter.
- 5 6. Scissors lift according to one of the previous claims, wherein the lower cam (23) and/or the upper cam (24) has, at least in part, a linear active slope.
7. Scissors lift according to claim 6, wherein the linear active slope of the lower and/or the upper cam has an angle of between 10° and 65° with the base of the cam.
- 10 8. Scissors lift according to one of the previous claims, wherein the lower cam (23) and/or the upper cam (24) has, at least in part, a concave curved active slope.
9. Scissors lift according to claim 8, wherein the concave curved active slope following a function of at least the first order.
- 15 10. Scissors lift according to one of the previous claims, where the linear active slope of the lower and/or the upper cam comprises a concave curved segment at lower angles continued by a linear segment.
11. Scissors lift according to one of the previous claims, wherein the groove or bushing (125) is a groove.
- 20 12. Scissors lift as claimed in any of claims 1 to 11, wherein the bushing is a cylinder of low-friction material around the axis (116).
13. Scissors lift according to one of the previous claims wherein wherein the lower cam (23) has a linear active slope and the upper cam (24) has, at least in part, a concave curved active slope.
- 25 14. Scissors lift according to one of the previous claims, wherein the wedge mechanism comprises a guide delimitating, above and at the vertical of the lower cam, the top curve below which the roller wheel will evolve or a lateral wall extending externally to the lift mechanism over at least a surface covering the path of the roller wheel.
- 30 15. Paper stacker comprising the scissors lift of any of claims 1 to 14.

35

40

45

50

55

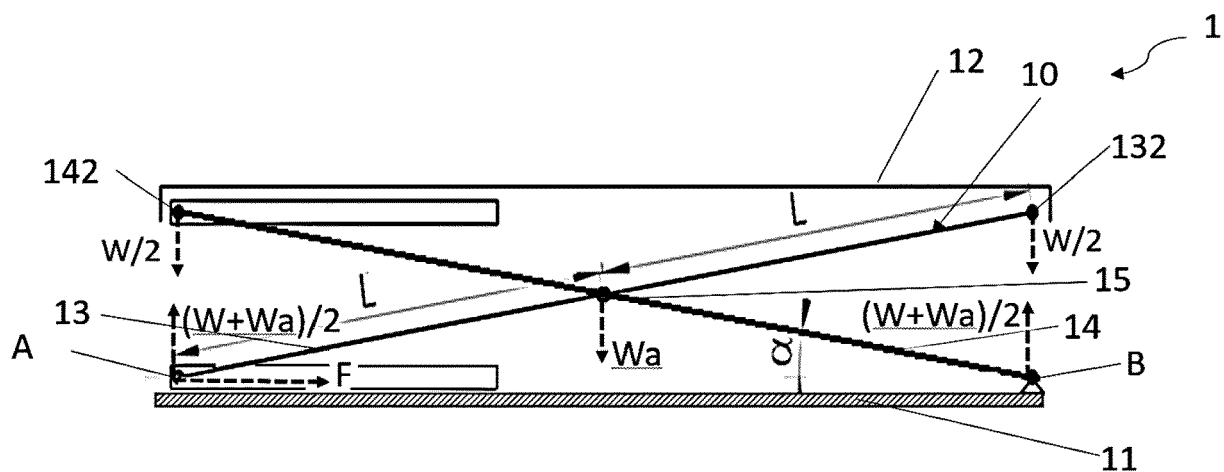


Fig.1

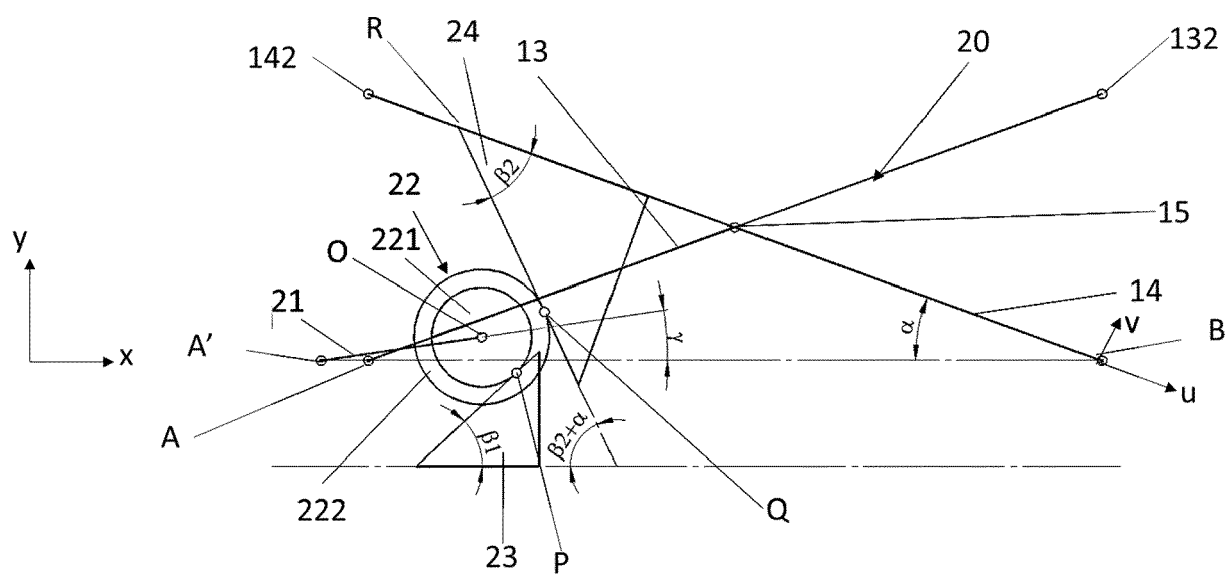


Fig.2

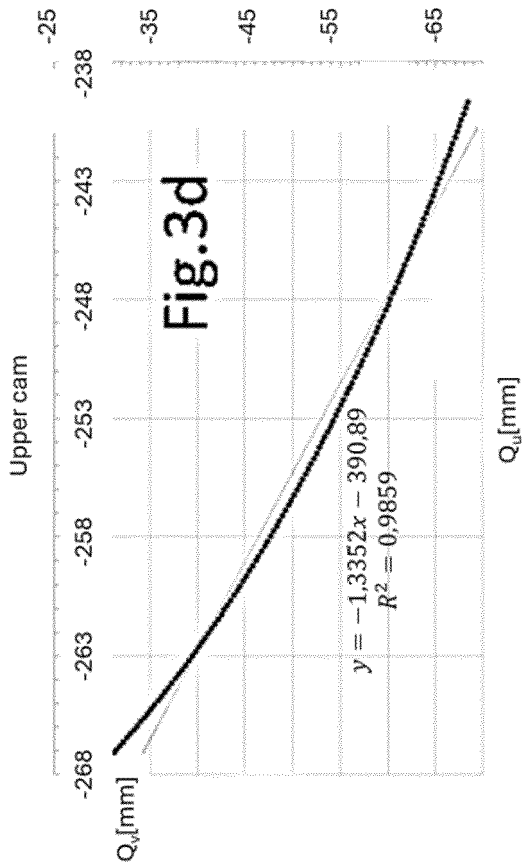
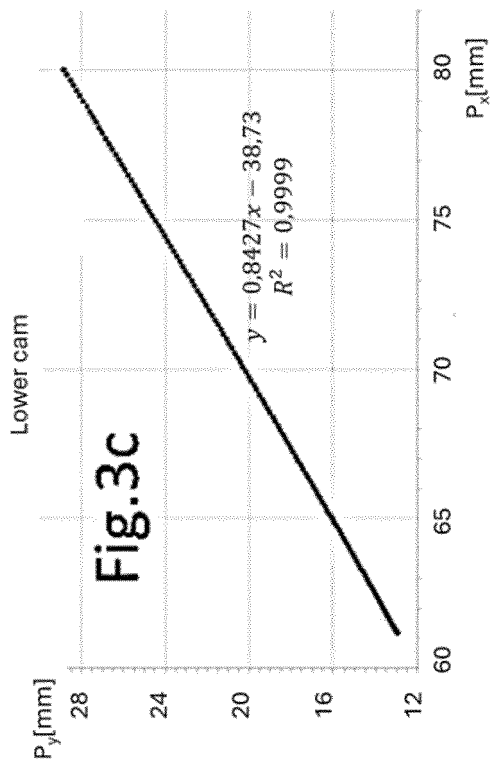
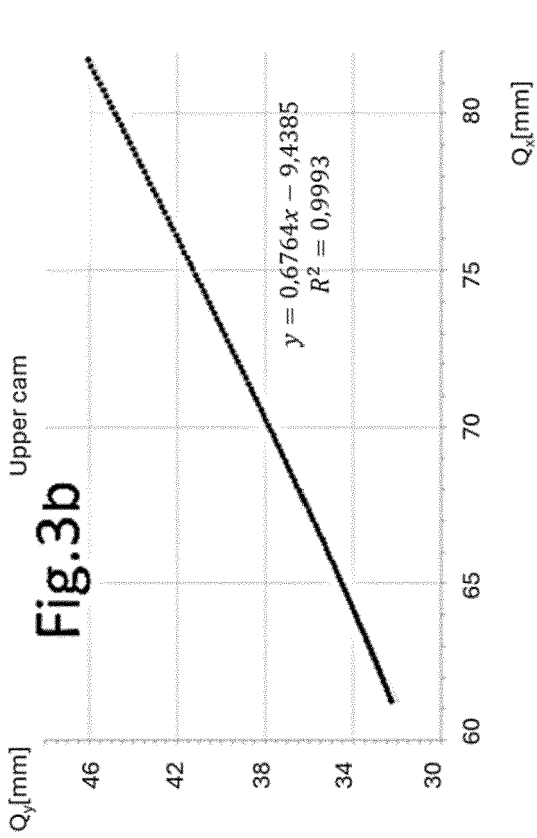
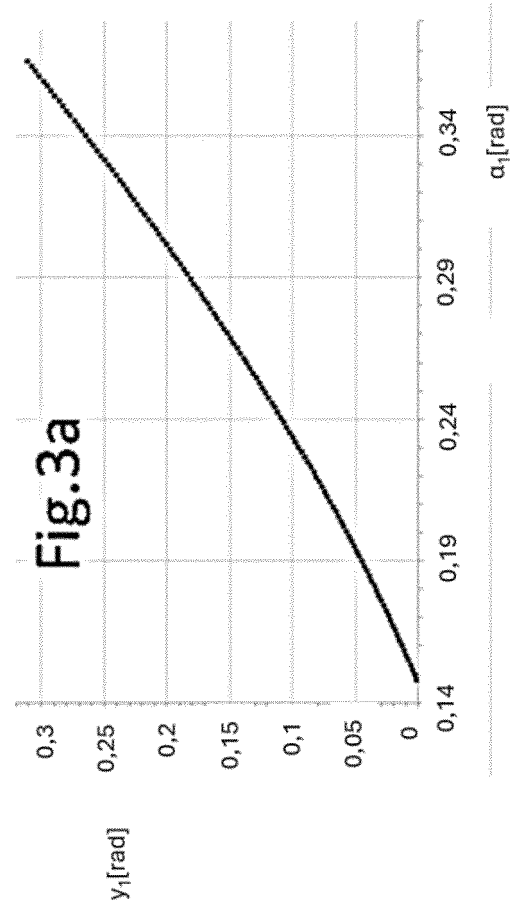


Fig.3

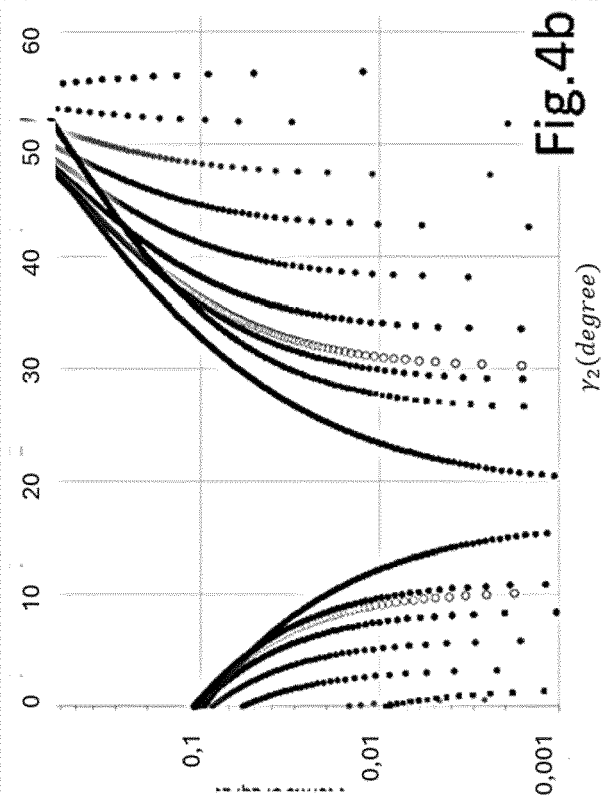
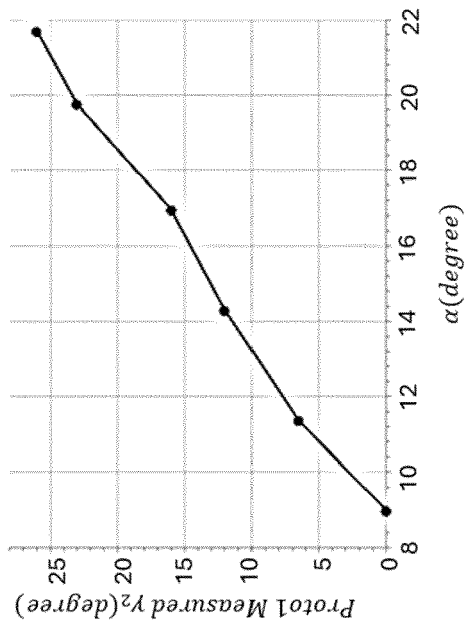
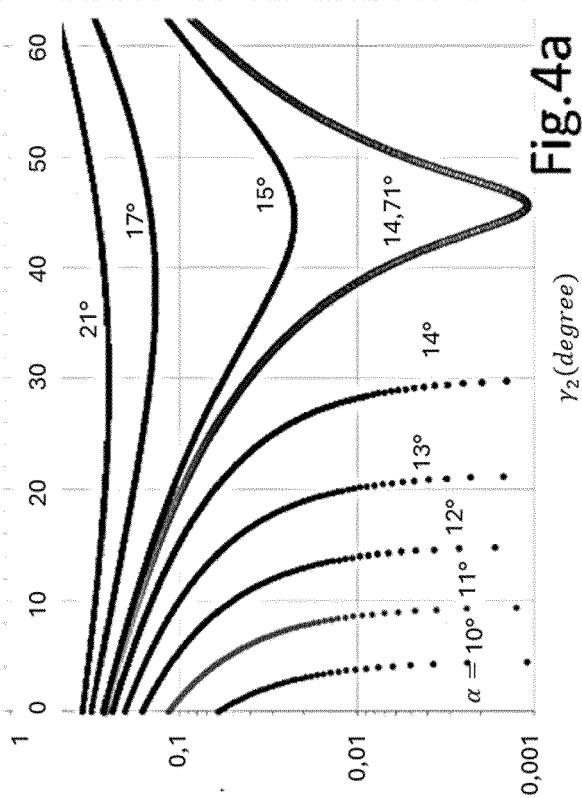


Fig.4

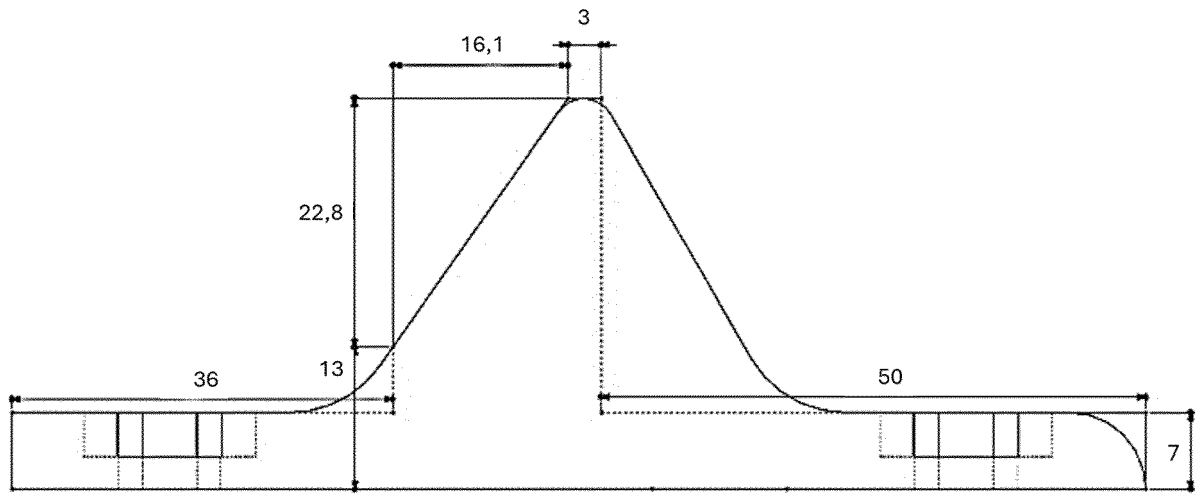


Fig.5

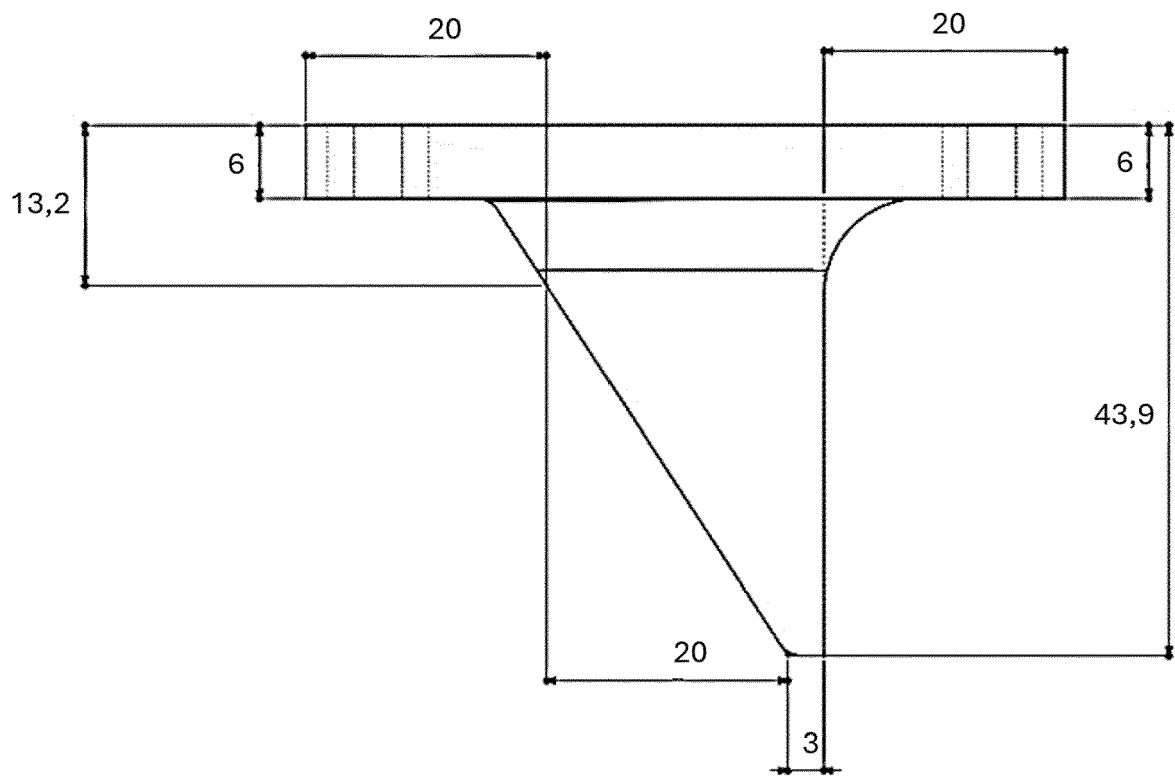


Fig.6

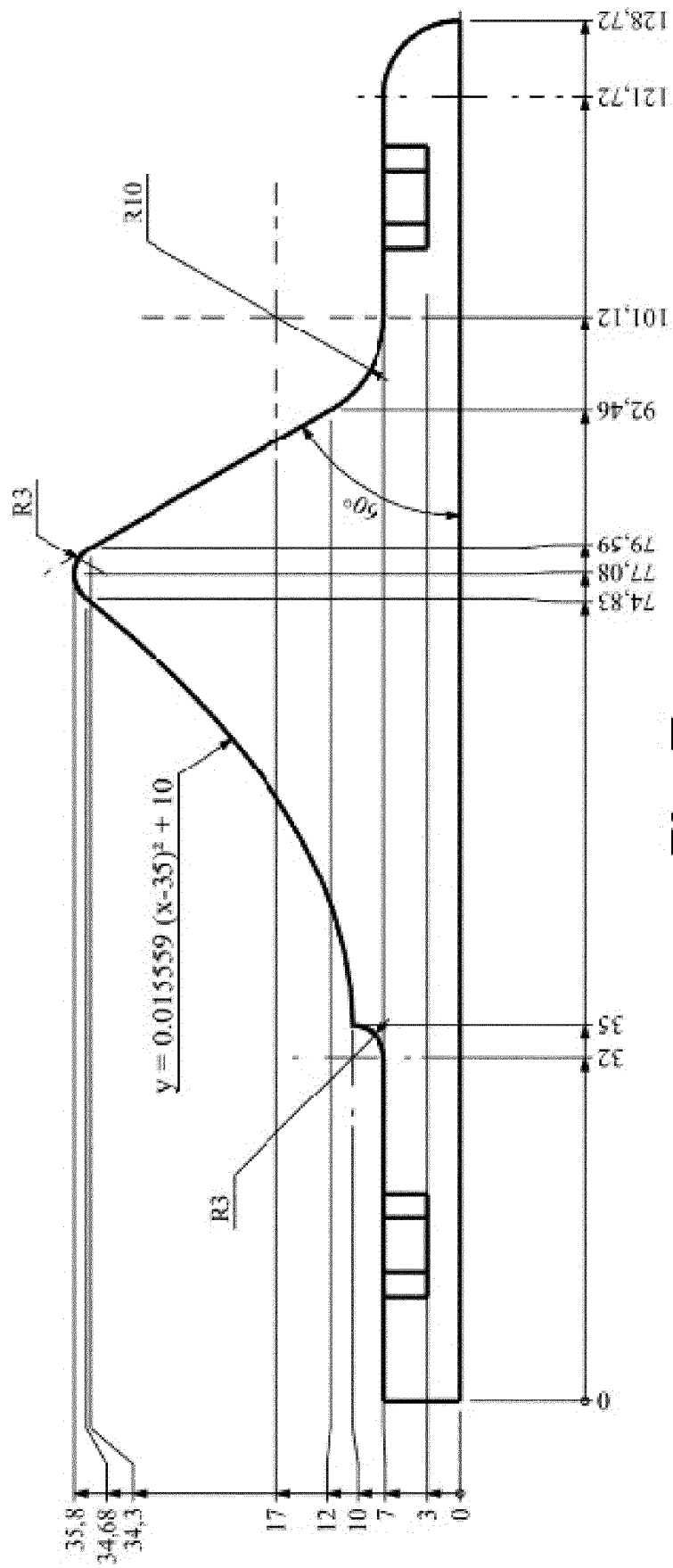


Fig.7

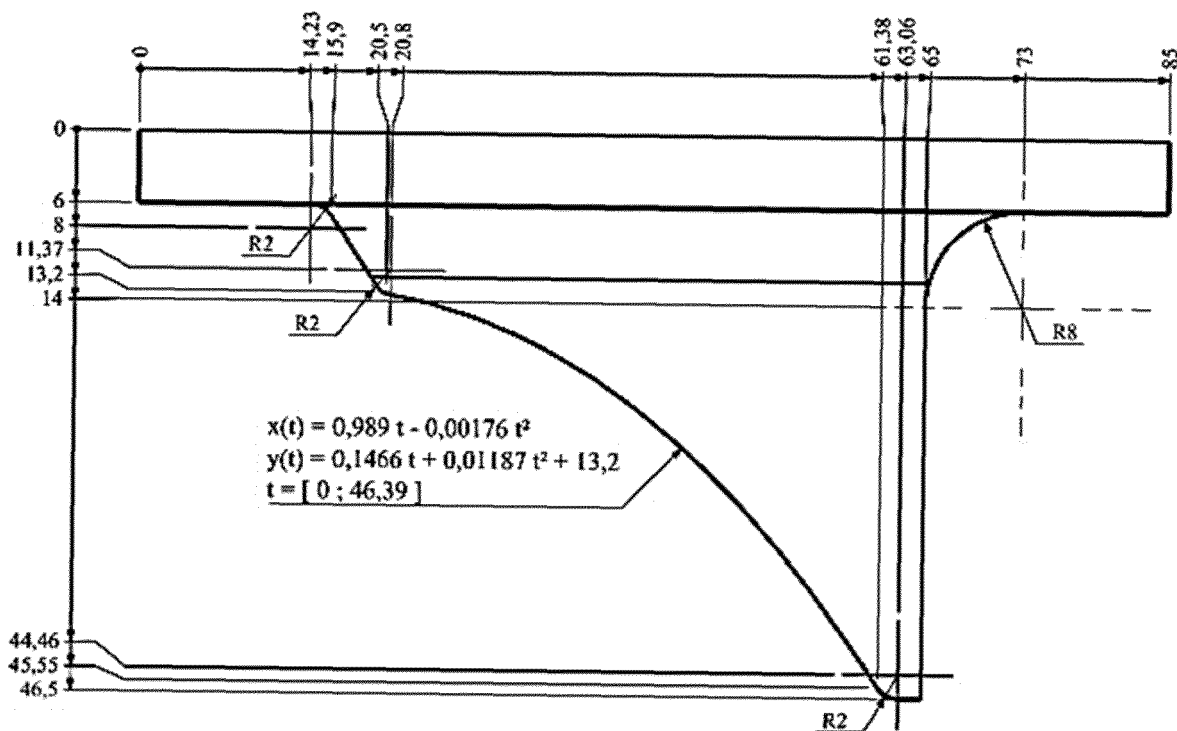


Fig.8

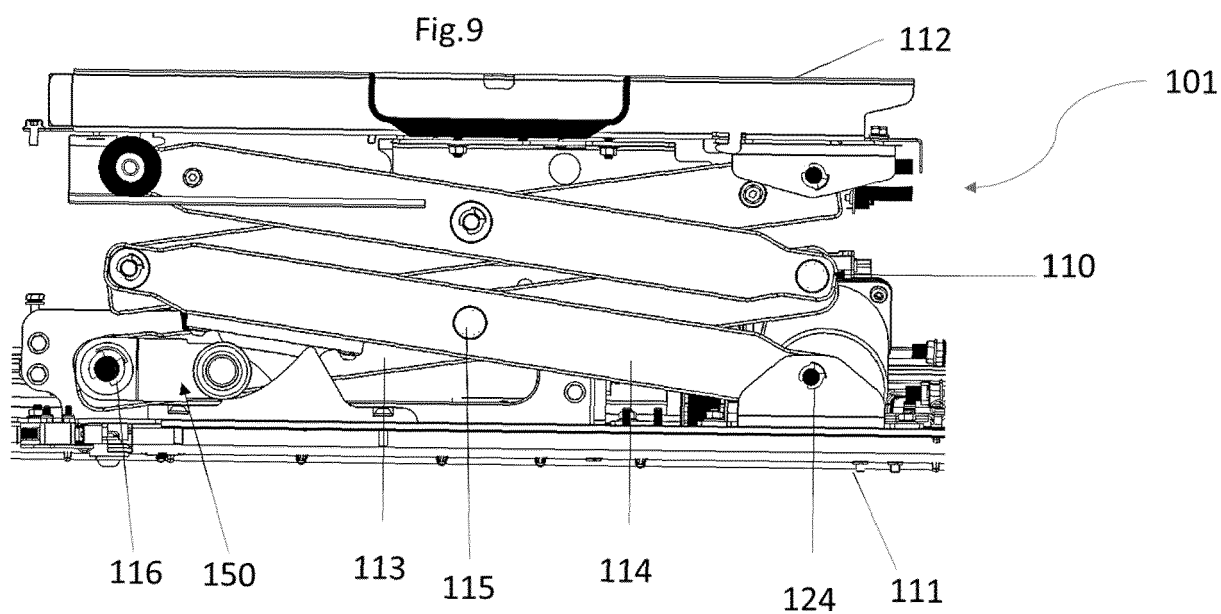


Fig.10

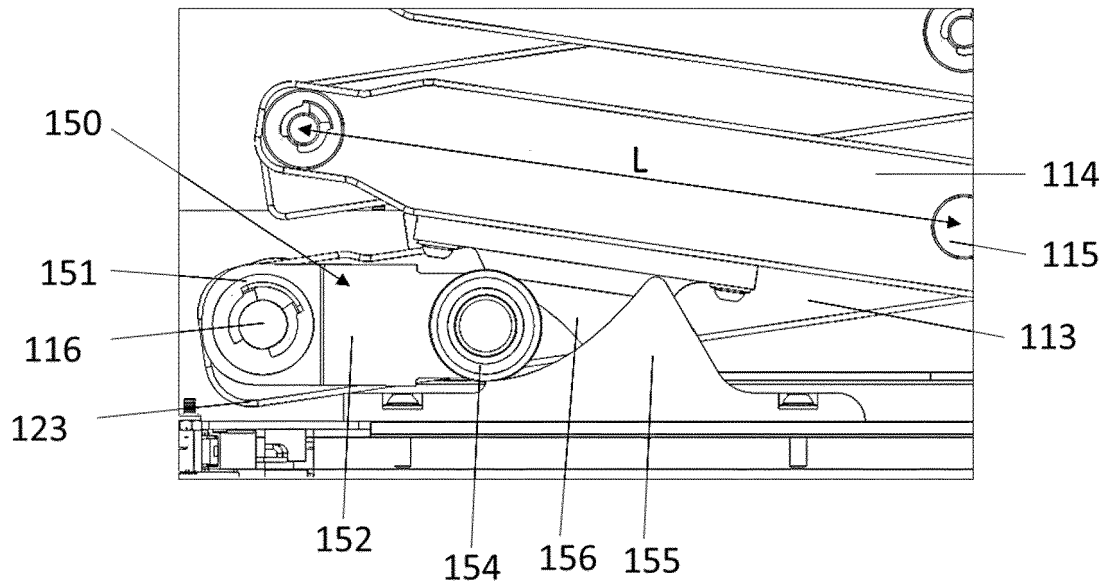


Fig.11

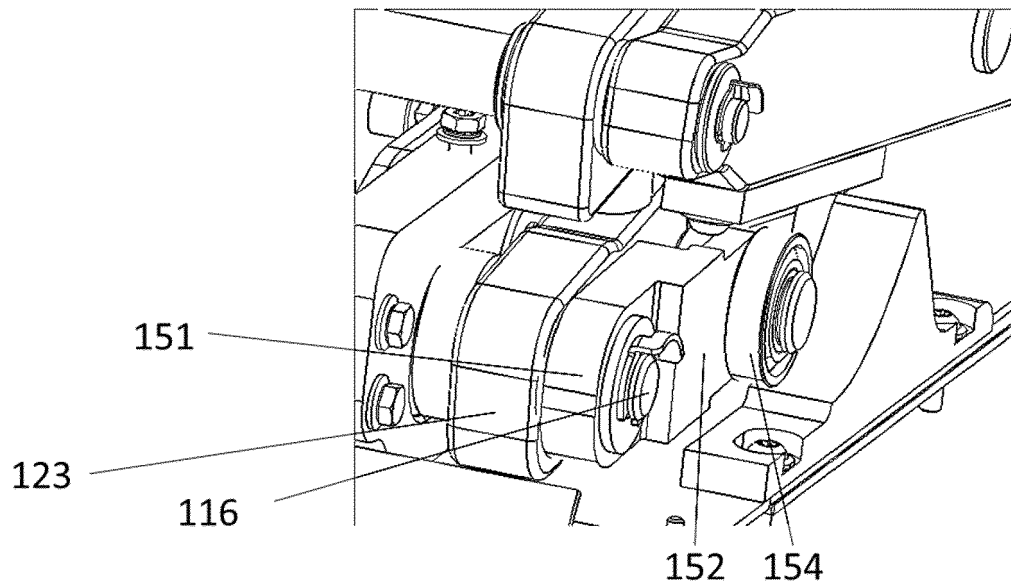


Fig.12

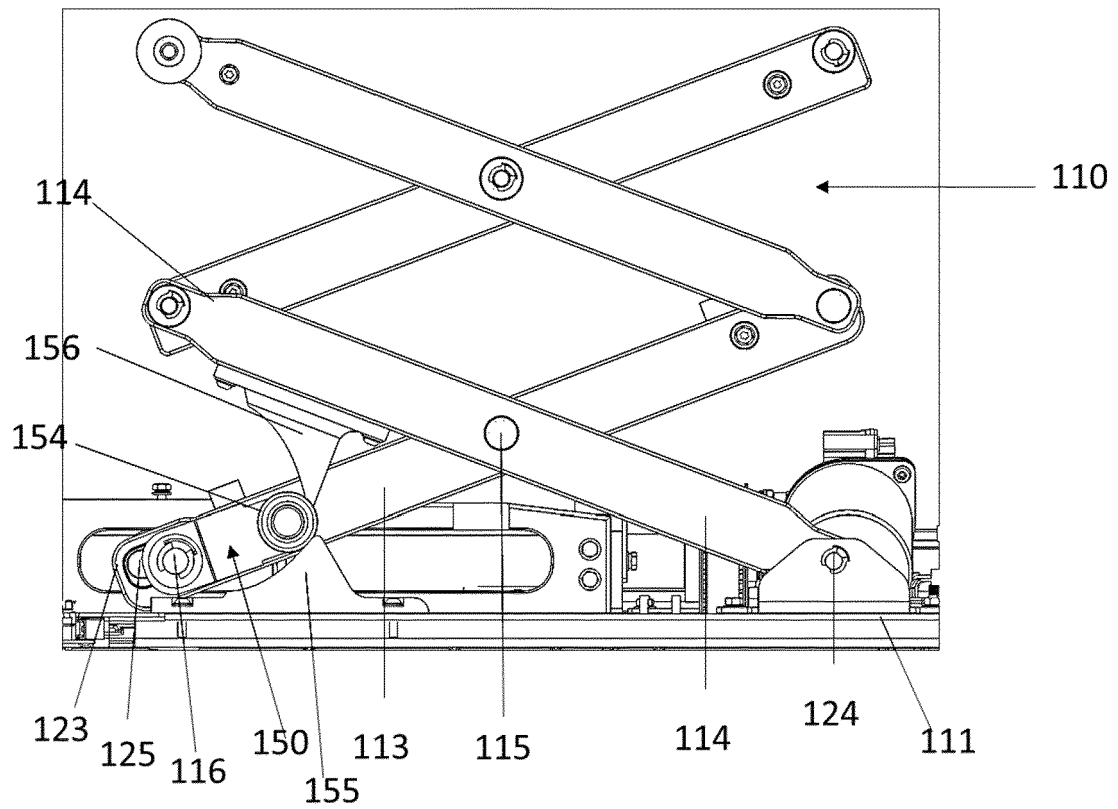
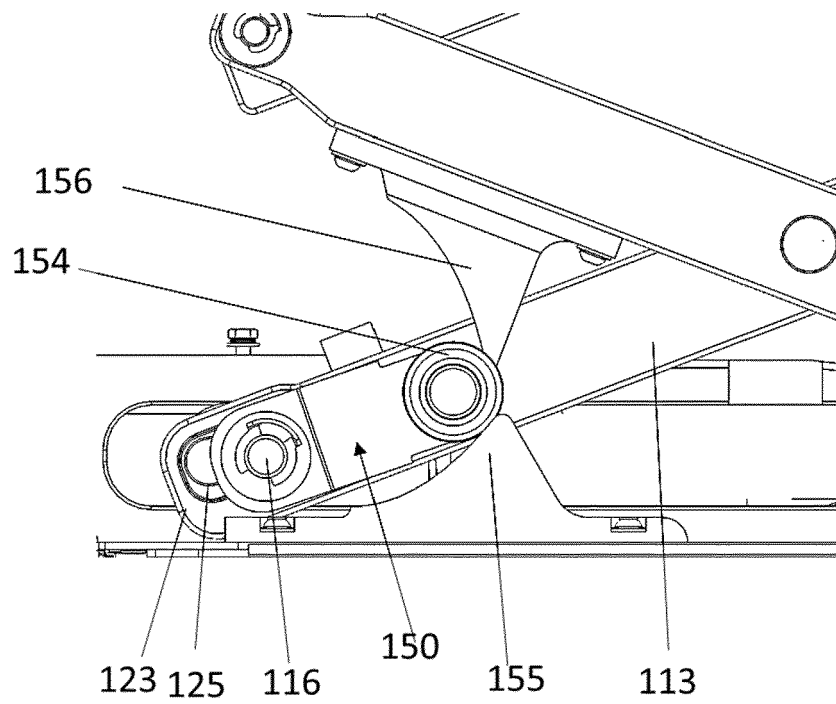


Fig.13



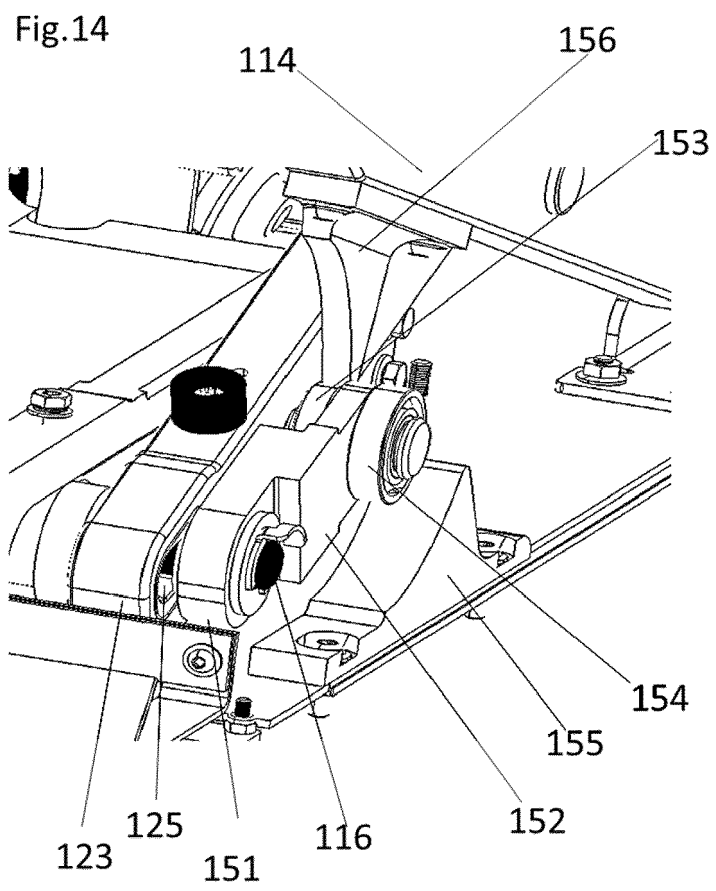


Fig.15

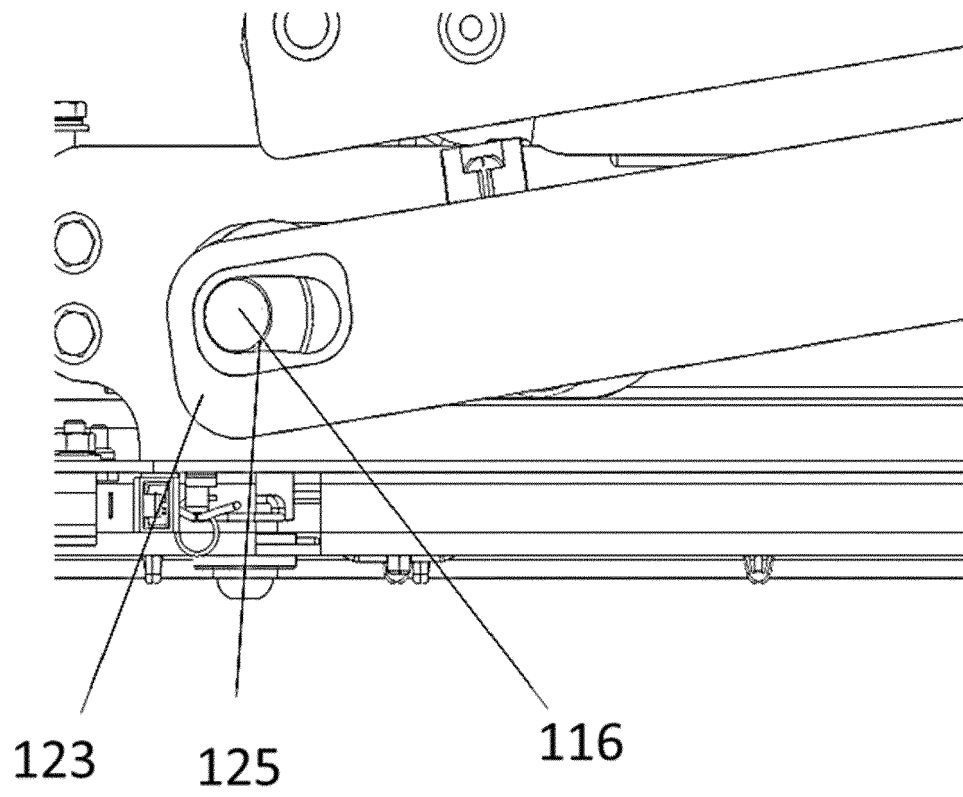
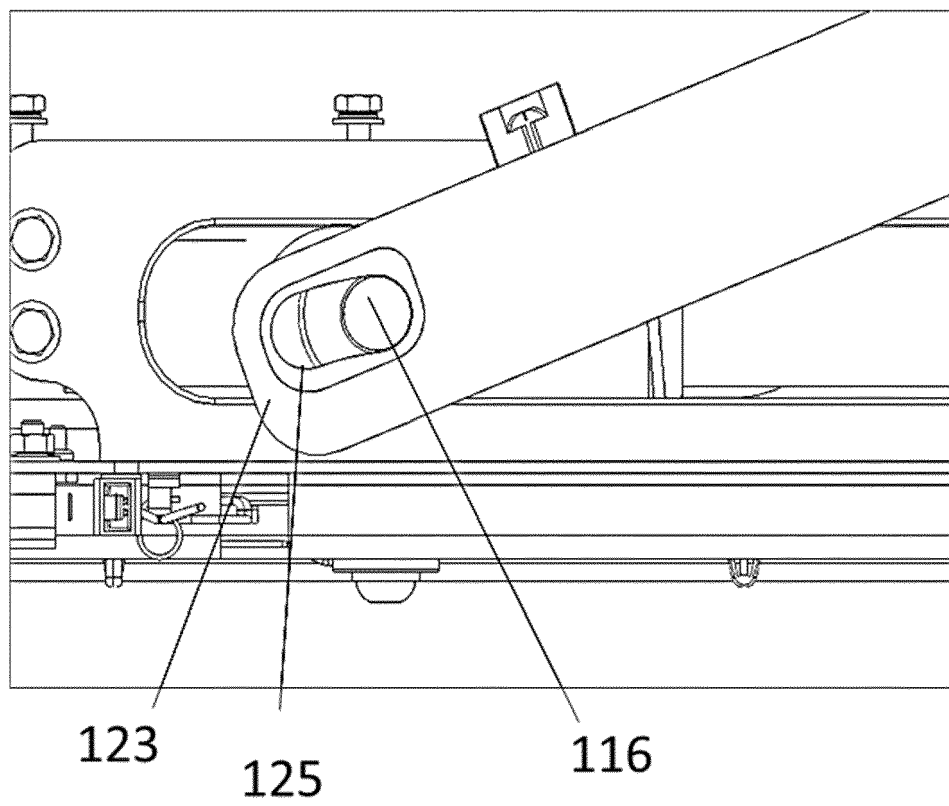


Fig.16



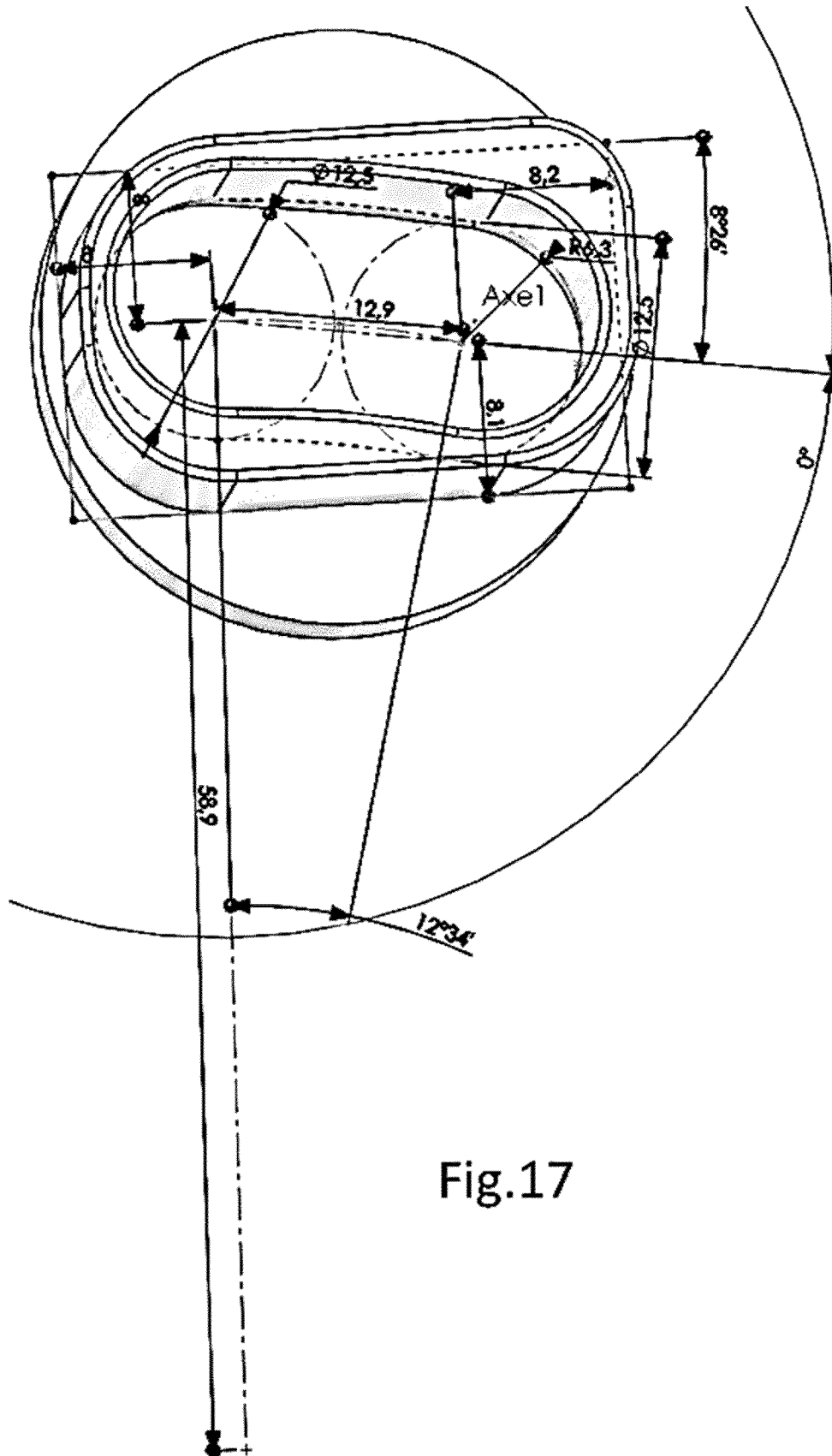


Fig.17

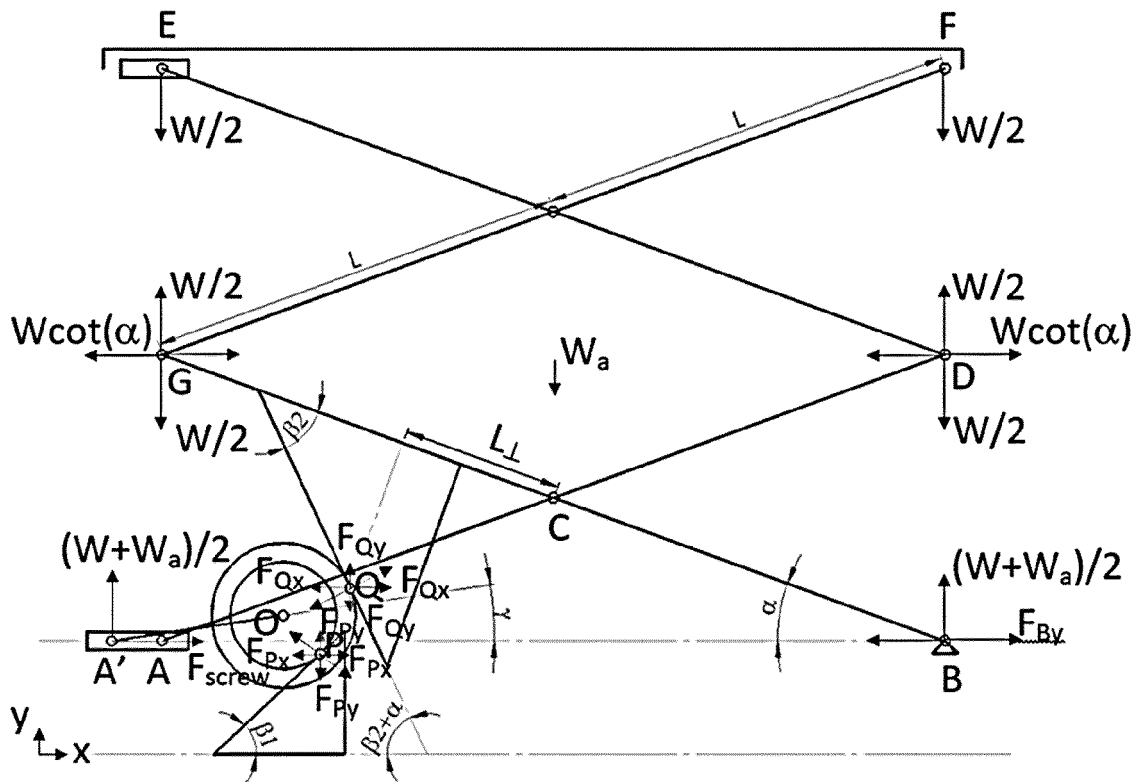


Fig.18

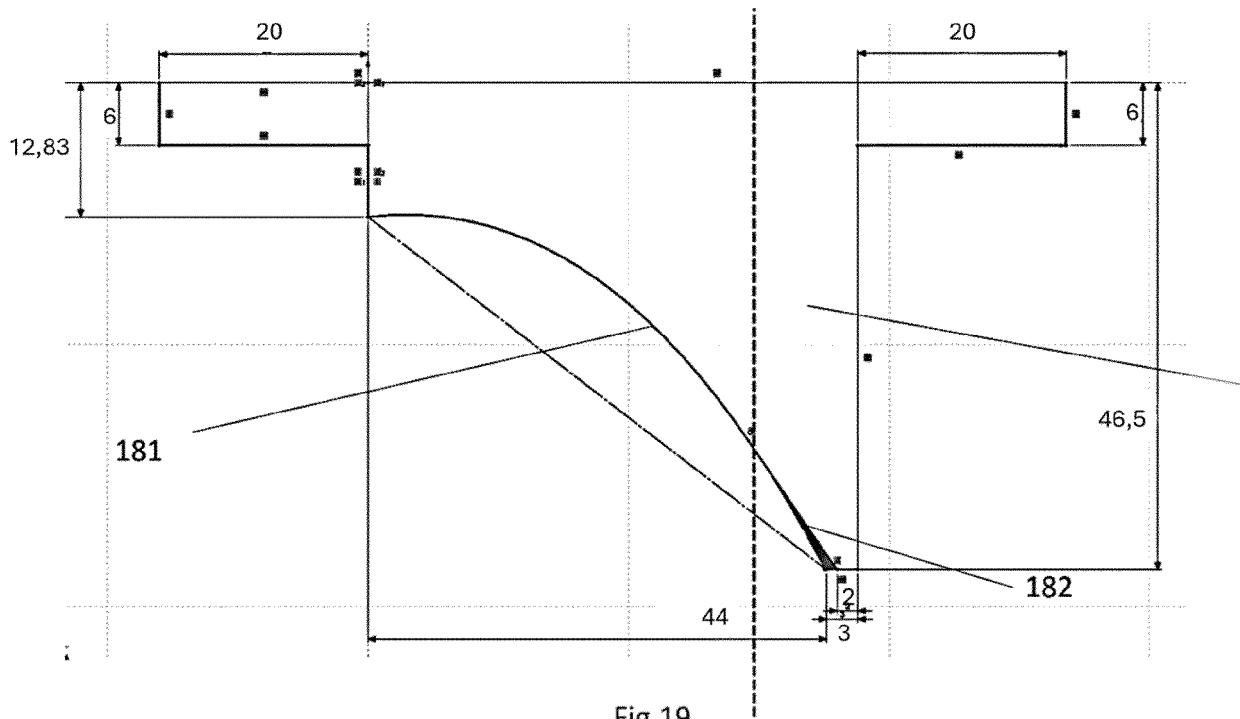
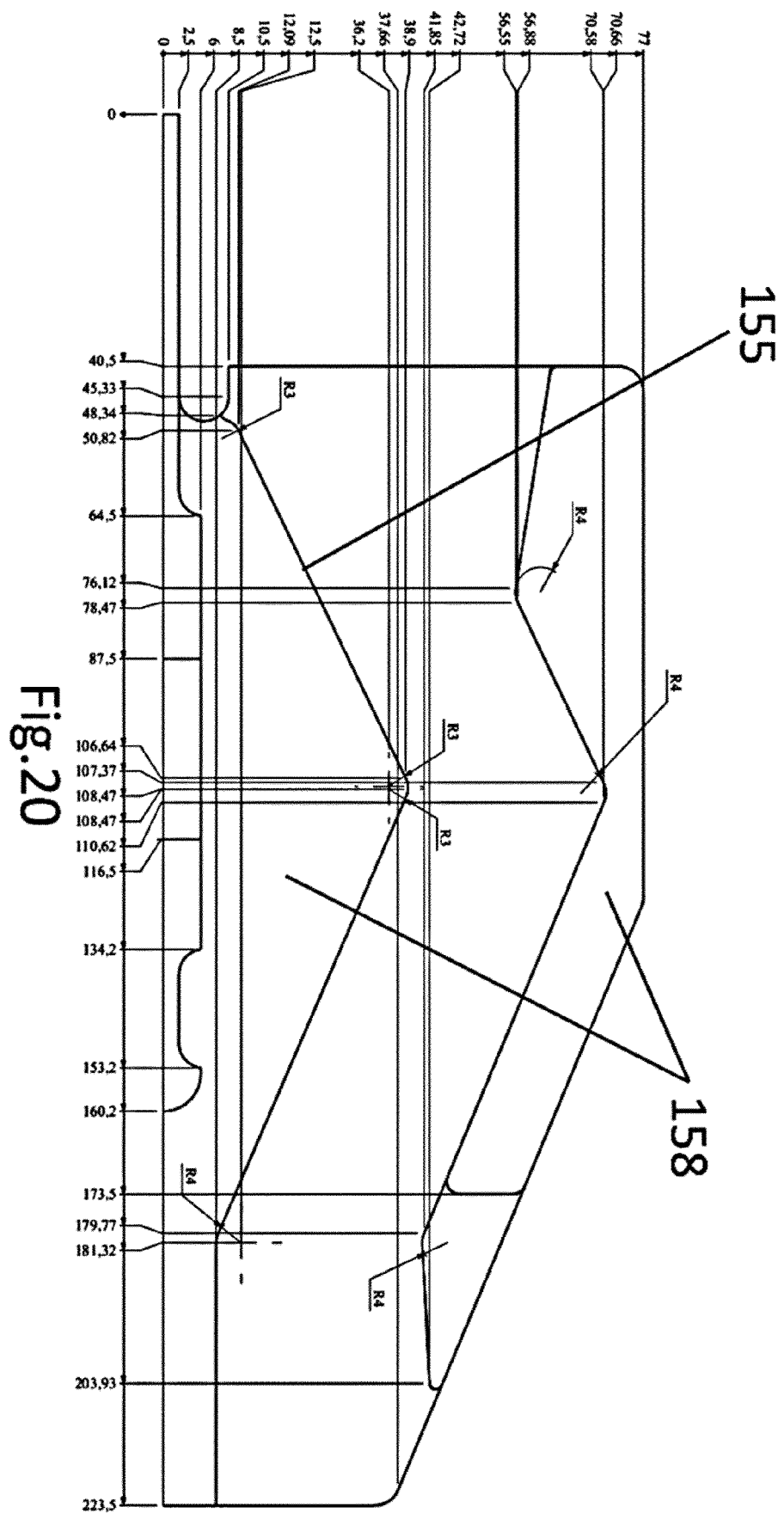
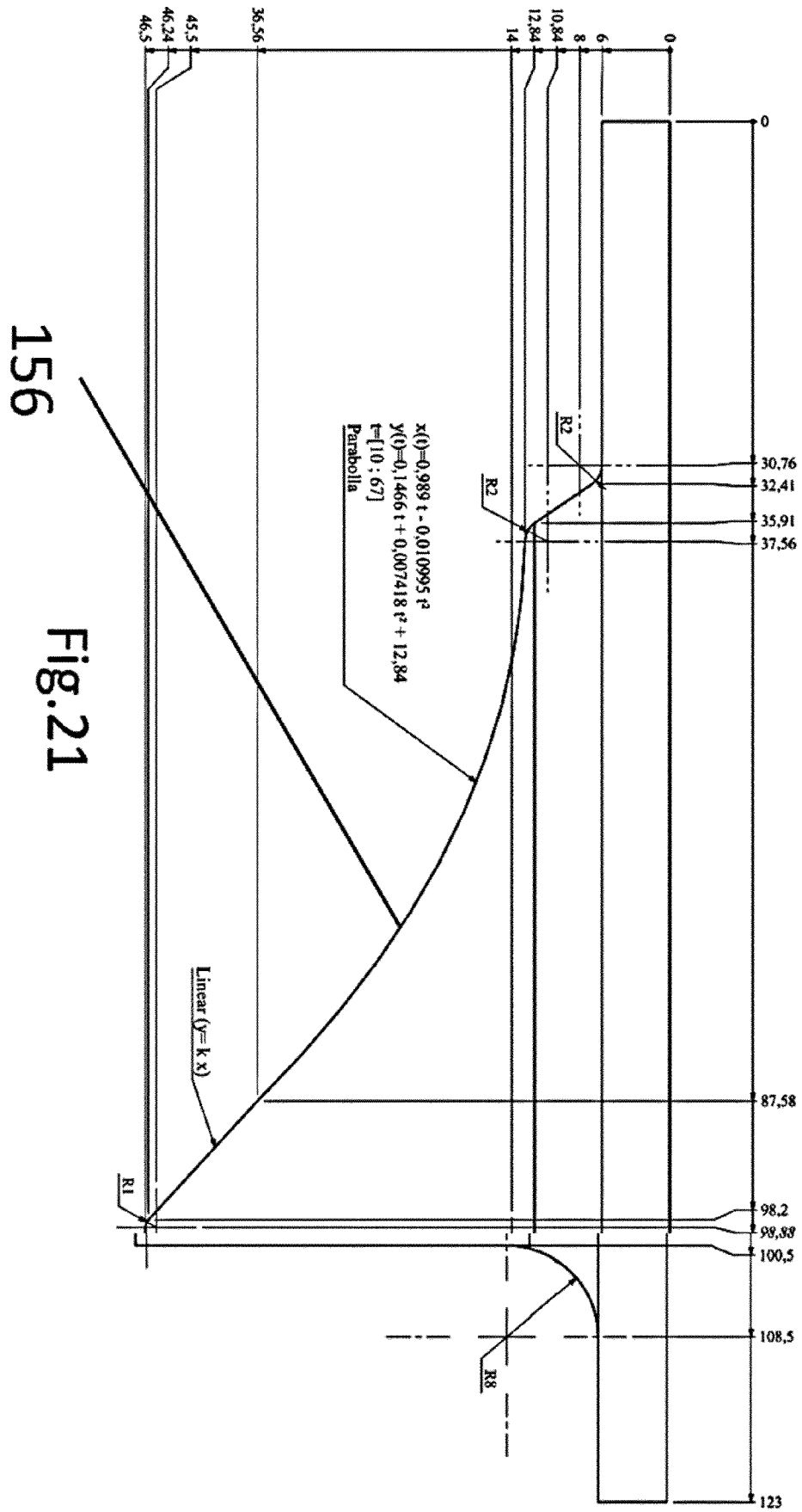
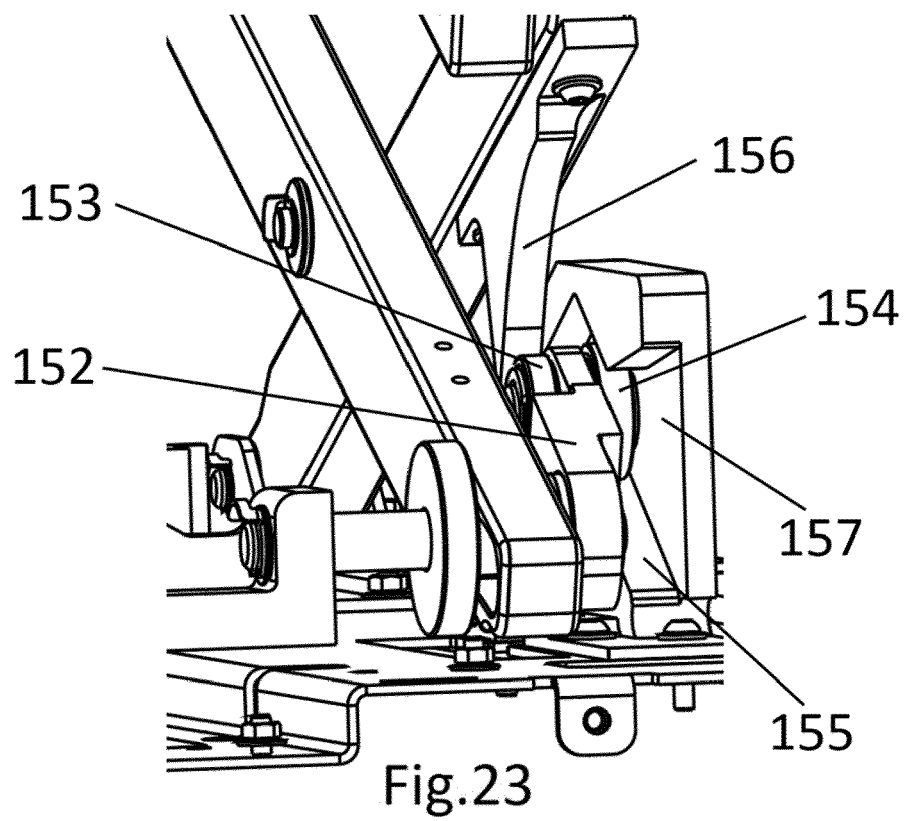
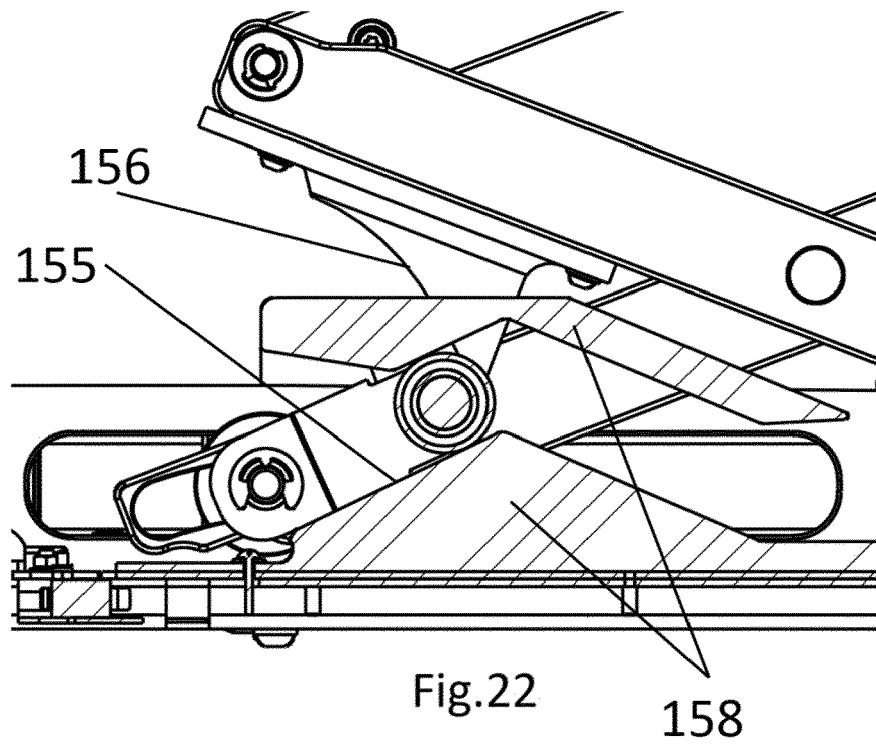


Fig.19









EUROPEAN SEARCH REPORT

Application Number

EP 23 21 6832

5

10

15

20

25

30

35

40

45

50

55

1

EPO FORM 1503 03.82 (P04C01)

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A,D	US 2003/075657 A1 (JOUBERT PIERRE [US]) 24 April 2003 (2003-04-24) * abstract; figures 1-14 * -----	1-15	INV. B66F7/06 B65H31/00
A	US 2 862 689 A (DALRYMPLE PHILIP W ET AL) 2 December 1958 (1958-12-02) * abstract; figures 1-6 * -----	1-15	
A	WO 86/06054 A1 (HYMO AB [SE]) 23 October 1986 (1986-10-23) * abstract; figures 1-4 * -----	1-15	
			TECHNICAL FIELDS SEARCHED (IPC)
			B66F B65H
The present search report has been drawn up for all claims			
Place of search The Hague		Date of completion of the search 9 April 2024	Examiner Özsoy, Sevda
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document		T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document	

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO.

EP 23 21 6832

5 This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

09-04-2024

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US 2003075657 A1	24-04-2003	US 2003075657 A1	24-04-2003
		WO 03033393 A1	24-04-2003

US 2862689 A	02-12-1958	NONE	

WO 8606054 A1	23-10-1986	AU 580874 B2	02-02-1989
		DK 582986 A	04-12-1986
		EP 0252086 A1	13-01-1988
		FI 874541 A	15-10-1987
		JP H05316 B2	05-01-1993
		JP S62502460 A	24-09-1987
		SE 447238 B	03-11-1986
		US 4753419 A	28-06-1988
		WO 8606054 A1	23-10-1986

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- US 9310753 B [0007]
- US 20030075657 A1 [0008]