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TECHNICAL PAPER

Synthesis and optimization of an eight-bar linkage mechanism for seat suspensions

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Abstract
The work presents the analysis and synthesis of an eight-link mechanism for industrial seat suspension, followed by a numerical optimization of the design under specific technical requirements. Even though the eight-bar linkage mechanism is a well-known architecture, it has few applications in seat suspensions, and it shows interesting features compared with more traditional scissor systems. It is possible to exploit this mechanism, provided that is correctly designed, to achieve a seat suspension system, which grants a large quasi-perfect vertical motion, without the need of prismatic couplers, which are often a weak point in off-highway application due to their maintenance issues and cost. Firstly, the problem was tackled analytically in order to obtain the general set of equations and verified numerically with a multibody solver. Secondly, to choose the best solution for the application, we carried out an optimization of the system, aimed at minimizing the horizontal displacement of the seat along with granting the desired vertical travel needed. The results are both a general design procedure to optimize under specific constraints the eight-bar linkage for seat suspensions problems and a detailed design and possible embodiment of a seat suspension systems with application in the agricultural field.

Keywords Seat suspensions · Eight-bar linkage · Mechanical design · Kinematic equations · Vibrations

List of symbols

a, b	Alternative horizontal distance between the spring and the lower left pin
L_i	Length of member i
β	Angle between members 5 and 9
θ_i	Anticlockwise angle between an horizontal line and the member *i*
ω_i	Angular velocity of member *i*
F_m	Force of the spring
F_p	Force due to passenger weight
$Fx\ ij$	Horizontal force exchanged between i and j when only two members insists on a pin
$Fy\ ij$	Vertical force exchanged between i and j when only two members insists on a pin
$FxP\ ij$	Horizontal force between Pin *i* and member *j* when three members insists on a pin
$FyP\ ij$	Vertical force between Pin *i* and member *j* when three members insists on a pin
x_m	Horizontal distance between the spring and the lower left pin

1 Introduction

Seat suspensions are a key element in agricultural and off-highway machines, since the health of the operator largely depends on this component. The seat suspension connects the body of the operator to the frame of the machine, which is subjected to several types of vibrations, either coming from the harsh environment, such as rough, uneven terrain, or from the engine and the equipment connected to the system. The operator must withstand these vibrations for a very long time, without having consequences in terms of muscle stress and/or other work-related pathologies. To date, vibrations are one of the most important ergonomic factors affecting workers' health and therefore global efficiency of the production. Almost all the workers using mechanized systems are exposed to mechanical vibrations while working [1]. The most common effects of vibration exposure are: relaxation, increase in nervous tension, occupational

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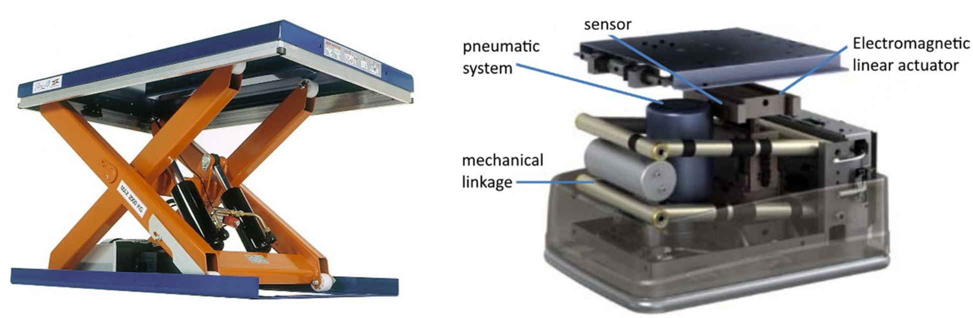
problems [2], depending on vibration of the system. Vibrations therefore should be avoided or at least decreased since they affects human health, working comfort, work productivity, work quality, and work safety [3]. One of the most used parameter to measure the exposure to vibrations is the whole-body vibration (WBV) which could lead to health issues, including spinal column problems and lower-back pain, depending on the magnitude, frequency, direction, duration, and distribution of the vibrations on the human body [4–6]. Workers from many occupations are exposed to WBV in road or off-road vehicles, but one of the most important application where WBV affects the operators is the farm tractor world [7]. Typically, modern tractors have front-axle suspensions and elastomeric anti-vibration mounts or a suspension system between the cabin and the chassis [8–14], but, since the vibration levels in tractors are higher than in other road vehicles [15], the role of the seat suspension is particularly important. In order to be more conservative for the worker health and safety, the design of a seat suspension does not consider the presence of other vibration suppression systems on the machine which will add an additional comfort to the machinery.

Agricultural vehicle WBV levels must be measured in accordance with the ISO 5008:2002 [16], where the tractor is driven over a smooth predefined path for 100 m and for the last 35 m on rougher track at a prescribed speed. During each cycle, the acceleration levels are measured simultaneously in three directions (X – longitudinal, Y – transverse and Z – vertical). The seat suspension must be rigid in X and Y direction, while should effectively suppress the vibration in Z direction. The typical structure used in seat suspension applications, therefore, is a 1 d.o.f. mechanism which allows only the vertical movement of the seat platform, with no rotation along the three axes (roll, pitch, yaw should be prevented).

The most common technical solution to this problem is the classic scissor configuration (Fig. 1a) [17], widely analysed in the literature [18] and quite popular among the main seat manufacturers [19–21]. This classical mechanical solution has some advantages in terms of stability, simplicity of construction, robustness to harsh environments, but it shows some drawbacks as well, since it prevents the space under the seat to be fully exploited, and it has sliders on top and bottom of the structure, which need maintenance (lubricant) to achieve a smooth behaviour. The scissor configuration (Fig. 1a), as shown in [18, 22] has two main components which are in charge of suppressing the vibration: a spring and a damper. Typically, high-performance seat suspensions exploit air springs [18] and increase the damping capability by using a magnetorheological (MR) damper. The MR damper is able to produce a force which is proportional to the applied current, which grants improved performances in case of uneven terrain [23–28]. Air springs are typically preferred to mechanical spring since they provide adjustable height of the seat, according to the operator comfort, thanks to pressure calibration inside the spring. In order to overcome the drawback of the scissor suspension, Bose proposed a simple Grashof four-bar linkage active mechanism (Fig. 1b) [29, 30], which has an active electromagnetic system that compensate most of the vibrations especially for truck seats, where the speeds are very high but the displacement of the vehicle due to the external excitations smaller than in agricultural and off-highway. Since the seat is connected to a link of the four-bar mechanism, one of the limitations is that the vertical displacement of the system is coupled with a horizontal motion, which could be small for truck applications, but becomes too large for agricultural ones, where the displacements are larger due to the uneven terrain. The typical displacement range required for an agricultural seat suspension is above 100 mm up to 180 mm. The vibration frequencies are typically low in the normal working condition, but the seat must be able to prevent shocks due to impacts in rough terrain as well. The traditional scissor system as well as the Bose suspension provide a good damping behaviour in case of low frequency vibrations, but suffers in case of shocks, when the suspension hits the rubber stops. In order to prevents shocks and harsh impact of the seats against the hard stops of the structure having a seat with longer travel is an advantage, since it gives the spring, always present in the system, the chance to intervene with a stronger force, or better it provides an adequate reaction force with less rigid spring, increasing the comfort of the



(a) (b)

Fig. 1 Scissors seat suspension general example with shock absorber and air spring [17] **a** and four-bar linkage **b** as reported in the Bose white paper [29] ride. This consequence is clearly visible in many suspension applications such as down-hill mountain bikes or off-road enduro motorbike systems as well as in seat suspension for marine and military applications [31].

A more detailed explanation of how the seat travel affects the performance is reported in [32, 33] where three seats with different travel were compared and the results shows how the seat with longer vertical stroke provides a better performance in terms of SEAT (seat effective amplitude transmissibility) and VDV (vibration dose value).

The present work deals with the design of a seat suspension able to provide an almost pure vertical and large stroke while keeping small dimensions. An additional requirement is to avoid the sliders and retain the necessary under-seat volume for spring and dampers, which excludes the state-of-the-art solution, i.e. the scissor seat suspension. In dirty environments typical of off-highway machines, hinges are widely preferable than sliders thanks to their intrinsic reliability, lower wear and absence of lubrication, easiest manufacturing, and assembly. The present work does not consider any dynamic analysis and control of the suspension but only a kinematic analysis; the reader interested in those topics can refer to the specific literature [34, 35].

According to the technical literature, many kinematic systems, which ensures an almost pure vertical motion, were already proposed and analysed. Considering sliders, we can find the Evans' linkage and several multibar expanding grilles [36]. The Watt's mechanism [36] presents a configuration with a single point of a link, which moves along a vertical line, as well as the Peaucellier straight line mechanism [36] or the D-Drive [36]. These three mechanisms simply use pinned connections. Other more complex mechanisms can be found in the literature, mostly based on the four-bar linkage couple with several ternary rigid members, which are too complex and thus costly to be implemented in an industrial environment [36]. A promising mechanism is the eight-bar linkage (8BL) in Fig. 2, initially developed for scales: it shows an almost vertical motion of a platform and can be designed to be used in a seat suspension application. Regarding this mechanism, scarce information can be traced in technical literature, therefore this work focuses on the analysis and the optimization of the system, up to the design of a seat suspension digital twin with a detailed embodiment suitable for agricultural applications.

2 Kinematic analysis of the eight-bar linkage for vertical motion

The application of the 8BL to the suspension problem requires the development of an analytical solution to the kinematic closure equations of the mechanism. The aim is to design and optimize the linkage described in Fig. 2 to obtain the required quasi-vertical movement of the output platform as a function of the motion of the input member. The problem is tackled first by providing a kinematic analysis of the system through a set of analytical equations, and then, an optimization is carried out under a set of prescribed dimensions and constraints which come from the potential application of the system. After that the embodiment of the suspension is provided, the rigid links dimensions are established to sustain the forces and stresses due to the load, and the spring and damper elements are placed under the seat compartment. The description of this methodology is provided in the subsequent sections.

This configuration can be studied as a planar mechanism, while a real application for a suspension will consists of two twin 8BL mechanisms at a given distance (i.e. the two 8BL lay on two parallel planes), in order to increase the robustness of the system and to provide a stable support for the operator seat.

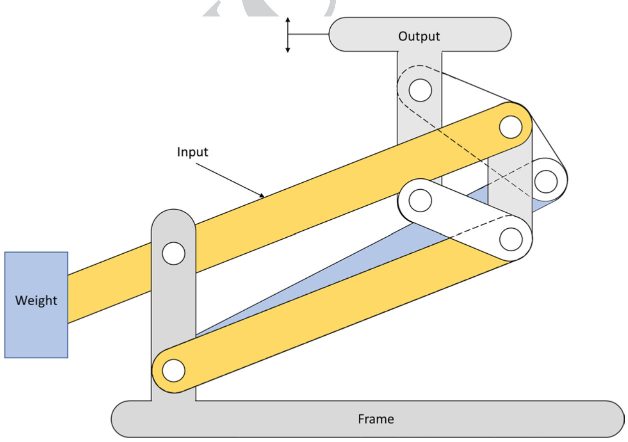


Fig. 2 Eight-bar linkage inspired by [36]

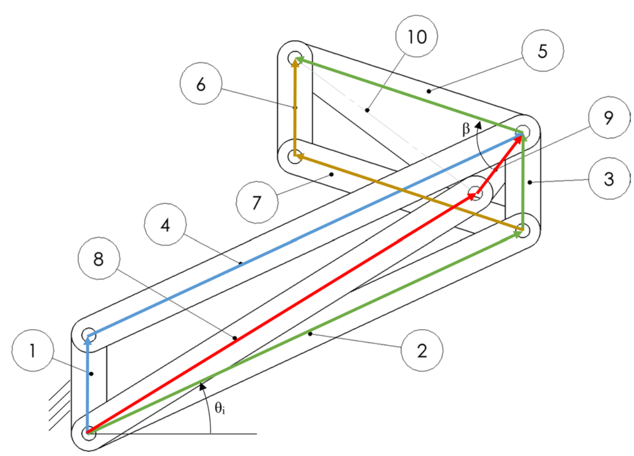


Fig. 3 Eight-bar linkage planar schematic used for the kinematic analysis

Figure 3 shows the nomenclature for the 8BL, the peculiar dimensions of each members in the figure are identified by a number, from L_1 to L_{10} which coincides with the member itself except the triangular rigid rocker arm, defined by the three distances of each hinge (5–9–10) and by the angle β comprised by 5 and 9. Member 1 is assumed fixed and for each member of the mechanism a specific angle θ_i, measured always starting from the horizontal line in anticlockwise direction, as shown in Fig. 3, defines the position of the axis of each member. The vectors show the kinematic chains used in the next closure equations.

Initially, no assumptions are provided for the 8BL, in order to be more general as possible. Considering the four-bar linkage 1–4–9–8, it is possible to write the following relationships:

$$\begin{cases} L_4 \cos(\theta_4) = L_8 \cos(\theta_8) + L_9 \cos(\theta_9) \\ L_1 + L_4 \sin(\theta_4) = L_8 sin(\theta_8) + L_9 sin(\theta_9) \end{cases} \tag{1}$$

The angles θ_9 and θ_4 are taken as dependent variables, while the angle θ_8 is the independent one. Through a series of analytical and algebraic simplifications, reported in Appendix, it is possible to explicitly write the angles θ_9 as a function of the distances of the links and of the angle θ_8:

$$\frac{1}{cos(\theta_9)} = -\frac{2L_9^2\left(L_1^2 - 2L_1^2 L_8 \sin(\theta_8) + L_8^2\right)}{L_8 L_9 \cos(\theta_8)\left(L_1^2 - 2L_1^2 L_8 \sin(\theta_8) - L_4^2 + L_8^2 + L_9^2\right) - \sqrt{-L_9^2\left(L_1 - L_8 \sin(\theta_8)\right)^2}} \tag{2}$$

From trigonometric consideration, we have as well:

$$\theta_4 = \arccos\left(\frac{L_8 \cos(\theta_8) + L_9 \cos(\theta_9)}{L_4}\right) \tag{3}$$

Once the expression of the angles is known, it is possible to derive the system of equations, reported in 1 in order to obtain the linear equations of the velocities ω_4 and ω_9 as a function of ω_8:

$$\begin{cases} \omega_4 = \frac{L_8 \omega_8 \text{cosec}(\theta_4 - \theta_9) \sin(\theta_8 - \theta_9)}{L_4} \\ \omega_9 = \frac{L_8 \omega_8 \sin(\theta_4 - \theta_8) \text{cosec}(\theta_4 - \theta_9)}{L_4} \end{cases} \tag{4}$$

Considering the rigid triangular member formed by the bars 5, 9, and 10, it is obvious that they have the same angular velocity, and the angles are constrained to form a triangle:

$$\begin{aligned} &\omega_9 = \omega_5 = \omega_{10} \\ &\theta_5 = \pi + \theta_9 - \beta \end{aligned} \tag{5}$$

The kinematic of the remaining members can be easily derived from the closure equations of the four-bar linkages 1–2–3–4 and 3–5–6–7 and their time-derivative to get velocity equations:

$$\begin{cases} L_4 \cos(\theta_4) = L_2 \cos(\theta_2) + L_3 \cos(\theta_3) \\ L_1 + L_4 \sin(\theta_4) = L_2 \sin(\theta_2) + L_3 \sin(\theta_3) \\ L_3 \cos(\theta_3) + L_5 \cos(\theta_5) = L_7 \cos(\theta_7) + L_6 \cos(\theta_6) \\ L_3 \sin(\theta_3) + L_5 \sin(\theta_5) = L_7 \sin(\theta_7) + L_6 \sin(\theta_6) \end{cases} \tag{6}$$

$$\begin{cases} L_4 \sin(\theta_4)\omega_4 = L_2 \sin(\theta_2)\omega_2 + L_3 \sin(\theta_3)\omega_3 \\ L_4 \cos(\theta_4)\omega_4 = L_2 \cos(\theta_2)\omega_2 + L_3 \cos(\theta_3)\omega_3 \\ L_3 \sin(\theta_3)\omega_3 + L_5 \sin(\theta_5)\omega_5 = L_7 \sin(\theta_7)\omega_7 + L_6 \sin(\theta_6)\omega_6 \\ L_3 \cos(\theta_3)\omega_3 + L_5 \cos(\theta_5)\omega_6 = L_7 \cos(\theta_7)\omega_7 + L_6 \cos(\theta_6)\omega_6 \end{cases} \tag{7}$$

Member 6 can be considered as the end-effector of the 8BL mechanism, i.e. where the driver's seat will be mounted on. The aim of the kinematic analysis is to write the displacement and the speed of the seat both in horizontal and vertical direction, in order to optimize the system. The vertical travel must be maximized, since a high vertical stroke leads to better vibration suppression [33], while the horizontal one must be minimized, or possibly prevented. By deriving the horizontal displacement of member 6 and substituting the expression of the angles (Eq. 5) and the angular velocities (Eq. 4) we can write the horizontal speed of member 6.

$$v_{6h} = \frac{L_8 \omega_8 (\text{cosec}(\theta_4 - \theta_9) \sin(\theta_4 - \theta_8)(L_5 \sin(\beta - \theta_9) + L_9 \sin(\theta_9)) - L_9 \sin(\theta_8)}{L_9} \tag{8}$$

By comparing the horizontal speed of member 6 and the angular speed of member 8 (ω_8), we can write the final expression to be minimized.

$$\frac{v_{6h}}{\omega_8} = \frac{L_8 (\text{cosec}(\theta_4 - \theta_9) \sin(\theta_4 - \theta_8)(L_5 \sin(\beta - \theta_9) + L_9 \sin(\theta_9)) - L_9 \sin(\theta_8)}{L_9} \tag{9}$$

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This expression can be optimized by using an optimization algorithm, under the desired constraint of prescribed height in stored position and minimum vertical travel. Even though Eq. 9 is general, the problem can be simplified by considering that the dimensions of the link are constrained by the required vertical displacement, i.e. four-bar linkages 1–2-3–4 and 3–5-6–7 must be articulated parallelograms. Therefore, we prescribe the following constraints:

$$\begin{aligned} & L_1 = L_3 = L_6 \\ & L_2 = L_4 \\ & L_5 = L_7 \end{aligned} \tag{10}$$

which means, according to Figure 3, that the corresponding angles are:

$$\begin{aligned} & \theta_1 = \theta_3 = \theta_6 = \pi/2 \\ & \theta_2 = \theta_4 \\ & \theta_5 = \theta_7 \end{aligned} \tag{11}$$

In order to limit the horizontal size of the mechanism in the close configuration, the triangle 5–9-10 must have $\beta \leq \frac{\pi}{2}$.

3 Numerical optimization procedure

The synthesis of the mechanism is solved numerically by means of an optimization procedure. First, the seat suspension constraints (Eqs. 8–9) were applied to the general problem. Assuming that member 6 (i.e. the seat) is the output of the suspension, the following dimensional requirements are the constraints that must be achieved:

1) The horizontal displacement of member 6 during the vertical travel should be within the range ± 6 mm.
2) The seat suspension height in closed position should be less than 130 mm.
3) The overall vertical travel should be more than 120 mm.
4) The global horizontal dimension should be less than 380 mm.

These requirements grant an innovative suspension compared to the traditional scissor solution with a lower seat index point (as defined by ISO5353) and a much more compact structure. In order to solve the numerical optimization problem, these requirements should be quantified in numbers and other geometrical constraints should be added, mainly based on the mechanic of the seat suspension and on the available space. Table 1 reports a detailed list of all the constraints applied to the optimization problem.

We implemented the optimization problem through the Matlab software [37], by using the equations presented in the previous section and the constraints in Table 1. These constraints consider the average space typically allowed for the seat suspension in agricultural equipment and are comparable with some of the seats of commercial manufactures [18–20].

To identify the minimum of the objective function (i.e. Eq. (7)), we used the "*fmincon*" solver [37], which calculates the constrained minimum of a scalar function, can manage several variables and can be used with a sequential quadratic programming algorithm. The maximum number of iteration allowed was 1000, and the tolerance of the solution was 1.0 E -6.

To identify the absolute value of the minimum of the objective function, we ran the procedure from different starting points. The geometrical constraints on the link lengths are imposed by the 8BL architecture, while the last constraints are needed to maintain the system where it has a minimum drift of the horizontal displacement.

Table 1 Optimization problem constraints

Enforced constrains in the optimization problem	Motivation
$L_1 = 70\,\mathrm{mm}$	Maximum horizontal displacement
$L_2 = 345\,\mathrm{mm}$	Horizontal space availability
$L_1 = L_3 = L_6$	Geometrical constraint
$L_2 = L_4$	Geometrical constraint
$L_5 = L_7$	Geometrical constraint
$50\,\mathrm{mm} < L_5 < 340\,\mathrm{mm}$	Maximum horizontal displacement
$50\,\mathrm{mm} < L_8 < 340\,\mathrm{mm}$	Horizontal space availability
$50\,\mathrm{mm} < l_9 < 100\,\mathrm{mm}$	Maximum horizontal displacement
$0 < \beta \leq \pi/2$	Geometrical constraint
$0 < \theta_8 < \pi/3$	Geometrical constraint
$L_8\,\mathrm{sen}[\theta_8] + L_9\,\mathrm{sen}[\theta_9] - L_5\,\mathrm{sen}[\theta_9 - \beta] = 60\,\mathrm{mm}$	Maximum horizontal displacement

Table 2 Optimal configuration for the seat suspension under the constraints in Table 1

Geometrical variable	Output results
$L_1 = L_3 = L_6 = 70\,\mathrm{mm}$	Horizontal displacement of member 6: ± 5.35 mm
$L_2 = L_4 = 345\,\mathrm{mm}$	Vertical displacement of member 6: 12.2 mm
$L_5 = 71.75\,\mathrm{mm}$	Minimum height of the suspension: 125.2 mm
$L_8 = 332.38\,\mathrm{mm}$	Maximum height of the suspension: 250.3 mm
$L_9 = 51.80\,\mathrm{mm}$	Overall travel of member 6 > 125 mm
$\beta = 1.218$ rad	

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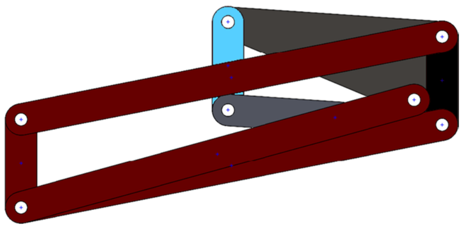


Fig. 4 Solidworks Motion simplified model of the optimized configuration (not to scale)

4 Numerical optimization results

The results of the numerical optimization, based on the analytical method described by Eqs. (1–9), led to the following link length for the 8BL mechanism, reported in Table 2. All the constraints are met, and the performance of the suspension satisfies the system requirements.

5 Numerical kinetostatic validation

In order to verify numerically the selected solution for the suspension we developed a Solidworks Motion model, using the dimensions reported in Table 2 to validate the motion equations for the 8BL. The model is reported in Fig. 4, and it is used to estimate the behaviour of the system in term of horizontal (x) displacement of member 6 (in light blue), the vertical link which is connected to the passenger's seat and the horizontal displacement of member 3 (in black), the vertical link of the parallelogram connected to the chassis, which provides a direct comparison to a four bar linkage system (Fig. 1b) The input of the system is applied to the passenger seat as a sinusoidal vertical (z) travel from the lowermost to the uppermost position.

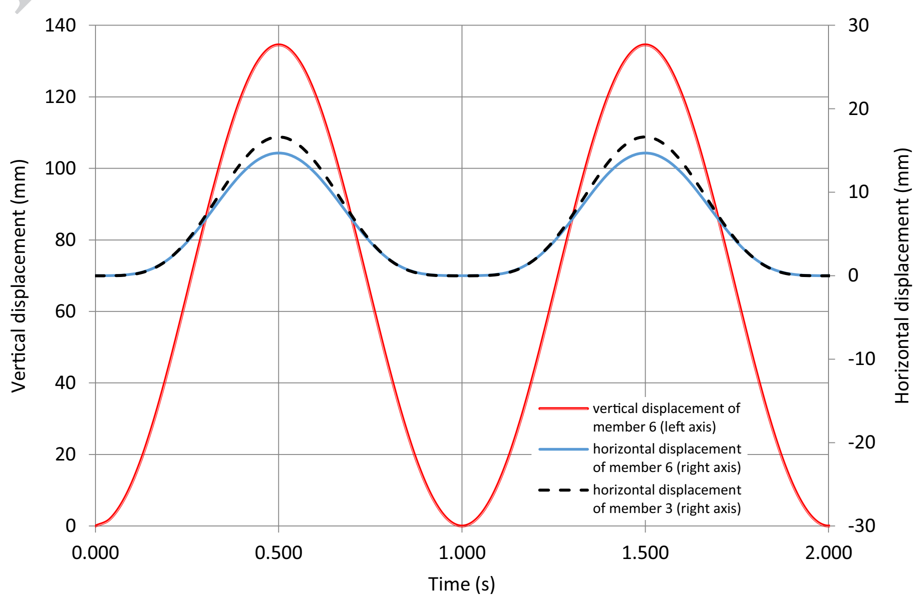
Figure 5 depicts that the 8BL mechanics provides a low horizontal displacement (within ± 6 mm) with a quite high vertical travel (more than 130 mm) as the system moves from its closed to its open position. Figure 5 reports the horizontal displacement of the member as well, 3 in order to show that the simple four bar linkage 1–2–3–4 displays a 14% larger horizontal displacement, compared to the proposed solution. This result confirms that the mechanical design meets the constraints and provides the desired performance for the seat suspension.

6 Design and Structural analysis of the optimized mechanism

Once the optimization procedure and the numerical simulations confirmed the feasibility of the solution, the next step was to provide a suitable embodiment for the 8BL concept, which must be tailored to be applied to a real seat suspension. First, we designed a structure that switch from planar to 3D, by means of several rigid links that couples two parallel 8BL together. Then, structural and manufacturing constraint could be considered, especially regarding the base of the seat suspension and the member 6, which is designed to be the housing of the proper seat. The detailed design of every component is not reported in this paper for the sake of brevity, but we report only synthetic considerations about the structural stability and strength of the main members.

Figure 6 reports the scheme used for the static structural analysis of the system: if we assume an input load F_p on

Fig. 5 Temporal plot of the horizontal displacement of member 6 (blue solid line) and member 3 (black dashed line) to be read on the right axis and vertical displacement of member 6 (red solid line) to be read on the left axis



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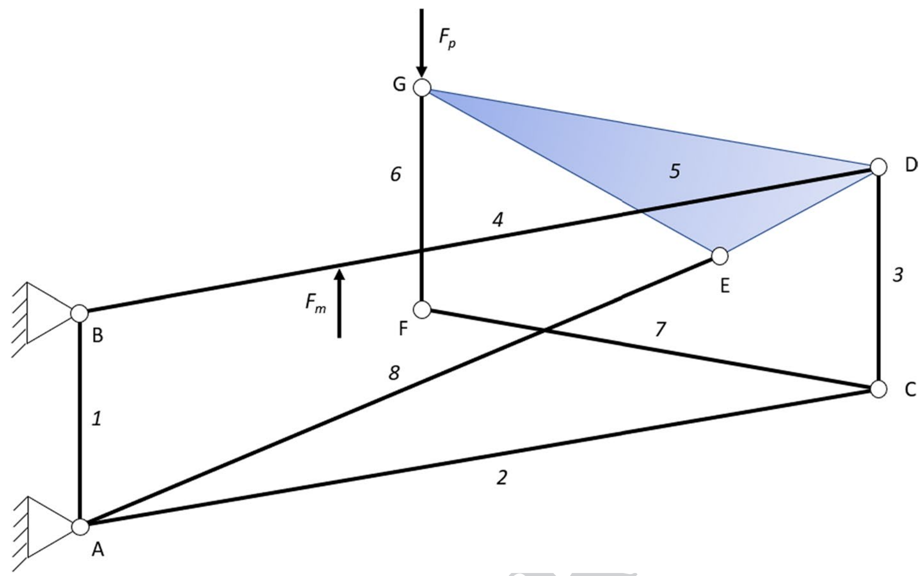

Fig. 6 Schematic of 8BL members used to determine the forces of acting on the system

member 6 (i.e. the passenger payload), to enforce the static and dynamic equilibrium of the system a spring is applied on member 4, with an arm $x_m = 220\ mm$ from member 1, which is the frame, and a line of action as the force, *Fm*, in Fig. 6. Starting from this, we wrote the 29 equilibrium equations, not reported here for the sake of brevity, that describe the forces acting on each pin and thus on each member of the mechanism. This system of equations was solved for each equilibrium point, since the distances between the hinges are completely defined depending only on one Lagrangian parameter, the angle θ, as reported in Fig. 3.

To determine the forces on the members of the suspensions and on the connecting pins, we used the following notation: F_{ij} is the force exchanged between members *i* and *j* when only two members insists on a pin, while when the pin connects three members the force on the pins, which works under shear and bending are defined as F_{PAj}, where *P* stands for pin, *A* is the pin name as defined in Fig. 6 and *j* is the member that exchanges a force with the pin. The force on each member is reported in the three important positions of the 8BL suspension: i) the closed position (Fig. 7a), ii) the maximum extension position (Fig. 7b), iii) the nominal position, in the middle between the other two (Fig. 7c). Table 3 reports the force in the three positions in horizontal (x) and vertical direction (y) normalized with respect to the force due to the passenger gravity, *Fp,* except the components that are null. The force acting on the triangular rigid member are identified by number 5. The force of the spring below the seat is *Fm*. It is worth noting that the 8BL suspension with the proposed design exploits a spring which provides a force around twice the target force and conversely grants a travel for the seat which is two times the stroke of the spring.

The final embodiment of the seat suspension is reported in Fig. 7. Two 8BL are coupled in parallel by rigid links, which help the system stability and prevent unwanted lateral displacement. The links were designed to sustain the desired load. The base of the system was designed to host all the components needed to provide the correct suspension of the passenger,

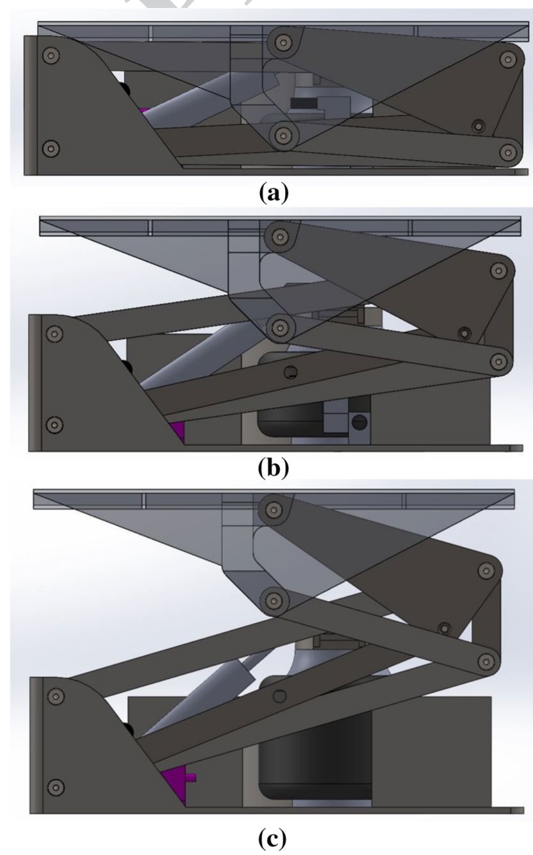

Fig. 7 Embodiment of the suspension in fully closed position (**a**) in nominal position (**b**) and fully open position, (**c**)

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Table 3 – Forces acting on the 8BL members and hinges, normalized over the passenger weight, F_p

	Normalized Force (dimensionless)		
	Closed Position	Nominal Position	Open Position
*Fx*58	−3.42	−3.94	−4.34
*Fy*58	−0.21	−0.88	−1.38
FxP D5	3.42	3.94	4.34
FyP D5	1.20	1.86	2.37
Fx 14	3.42	3.94	4.34
Fy 14	−0.70	−0.18	0.20
Fx P D4	3.42	3.94	4.34
Fy P D4	1.20	1.86	2.37
Fx PA8	−3.42	−3.94	−4.34
Fy PA8	−0.21	−0.88	−1.38
Fx 12	−3.42	−3.94	−4.34
Fy 12	−0.21	−0.88	−1.38
Fy P G5	−0.98	−0.98	−0.98
Fm	1.89	2.04	2.17

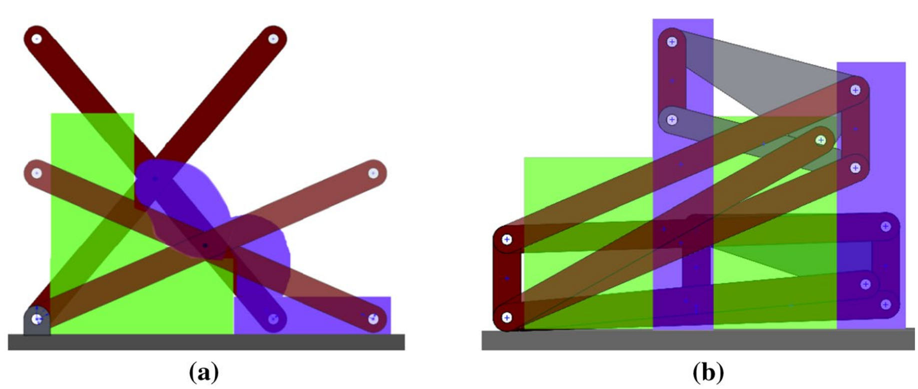
such as springs, dampers, sensor, control system, and air flow control devices. On the left part of Fig. 7, it possible to see that the available space under the seat is used to host a damper and the control electronic with pneumatic valves used to control the pneumatic spring (in black) attached to member 4. The pneumatic spring, which was accounted in the analysis, posed one of the major constraints, since this element is typically commercial, and the design has to consider its stiffness and its maximum and minimum travel. The spring used in this application is a Contitech SZ 51–5 [38], with a travel of ± 30 mm. The selected architecture allows this spring to be connected to a member in a position which suits the spring stroke limitations and provides the desired reaction force (which is dependent on the pressure applied by the pneumatic circuit): at 5 Bar the spring stiffness is roughly 83 N/mm and the force in central position is around 3500 N.

7 Global discussion of the proposed solution

Since most of the 8BL components works only in tension and compression, the system is able to equip lighter members and save some weight compared to the scissor system and grants a very large internal space which can be exploited to provide additional systems which enhance the passenger comfort, such as dampers, larger pneumatic springs, sensors and control systems. Especially for smaller off highway vehicles, space constraints in the cabin are tight and providing additional room for the auxiliary system as in the 8BL gives an important advantage over the traditional suspension systems. Another strong merit is to exploit only pinned connection between the moving members, which are much rugged and durable than linear couplings. All the links are joined together by means of a commercial socket head shoulder screw [39] which carries the load and acts as radial bushings with no need of complex lubricants systems due to its high hardness.

In order to discuss the merits of the proposed system over other commercial solution, it is useful to recall the main functions it has to provide. The suspension system is in charge of transferring the loads from the seat to the chassis, while providing the lateral stability and the desired vertical travel thanks to its kinematic degrees of freedom. Second, it has to deal with dynamic loads and the vibration suppression combined with the spring, so there must be enough space to include a damper in parallel to the spring. The traditional scissor like suspension, which are by far the most commons solution (Fig. 1a) leaves little space under the central pin for the spring, so the constraints on the spring dimensions are quite tight. A comparison of the proposed solution in terms of availability of space under the seat is shown in Fig. 8 by comparing it to the traditional scissor system (Fig. 8a) and the 8-link solution (Fig. 8b), with the same space constraints and the same suspension travel. The possibility of displacing springs and other useful elements such as dampers or active controllers under the seat compartment (shown in green)

Fig. 8 Traditional scissor system (**a**) and 8-link system (**b**) in open (shaded) and closed configurations. The space available for the springs and the other elements under the seat depicted in green, while unavailable space is shown in purple



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is larger for the proposed solution. Moreover, in the proposed solution there are many elements to which the spring could be connected as well as the possibility to choose the position of the spring almost freely in the under-seat volume. Only the central portion and the rightmost portion of the volume (shown in purple) are not available, but the same happens for the scissor suspensions, due to the central pins used to join the two sides of the system. The scissor system, which shows comparable compactness and similar vibrational behaviour, needs a spring with larger strokes and stiffness since the spring must be positioned not far from the leftmost hinge, due to the spatial disposition of its members.

Hence, the scissor suspension is mainly used when the sliders are not an issue or when the available horizontal space for the seat is not a tight constraint. To increase the space available for the components other solutions have been envisioned, such as systems like the Bose one (Fig. 1b) These systems, unfortunately, couples the vertical motion with the horizontal one, which is annoying for the driver especially for long strokes. Conversely, the 8BL grants a quite large space under the seat a small horizontal drift and multiple places where the spring can be placed, according to the manufacturer needs. Specifically, it can be attached either to bar 2, 8, or 4 (Fig. 9). This freedom is crucial for the designer which can easily provide the desired dynamic response since the force and the deflection of the spring to be selected are dependent on its distance from member 1 (Fig. 9).

Another important features worth mentioning are the rubber hard stops which prevent the seat to hit the base and limit the upper stroke to avoid possible damages of the structure: these elements are considered in the suspension system, but not reported in the 3D model in Fig. 10 to avoid confusion.

8 Conclusion

This research shows the methodology used to provide the analysis and the synthesis of an eight-link mechanism for agricultural and off-highway seat suspensions. A numerical optimization of the system is applied to ensure that the final design meets the technical specification and provides applicable results even compared with the traditional scissor systems which are the state of the art in this field. The methodology shown overcomes two issues of the traditional systems, i.e. the small space under the seat for the auxiliaries, which becomes more important as the functions of seat increases to satisfies ergonomics and safety aspects, and the presence of linear prismatic couplers, which are always delicate elements due to presence of lubricants and friction elements. First a set of analytical equations of the kinematics are developed in order to describe the proposed system, then a numerical optimization procedure is provided. The first aim is minimizing the horizontal displacement of the seat, which is annoying for the driver and the second aim is providing the desired vertical motion which leads to good vibration mitigation especially in harsh terrain. The large vertical displacement prevents the suspension to hit the hard stops and therefore the driver to undergo dangerous shocks, as reported in the Introduction section. The main advantages of the proposed

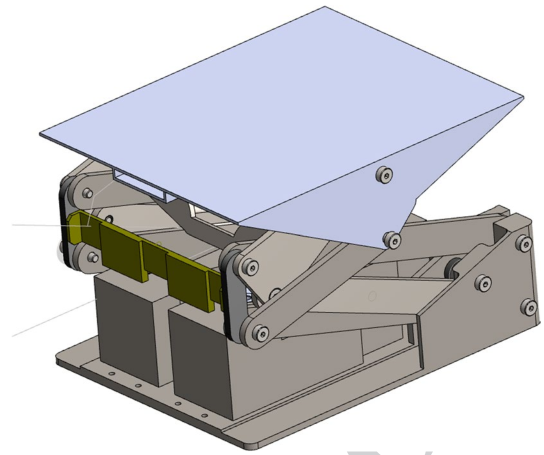

Fig. 10 3D assembly of the final suspension system

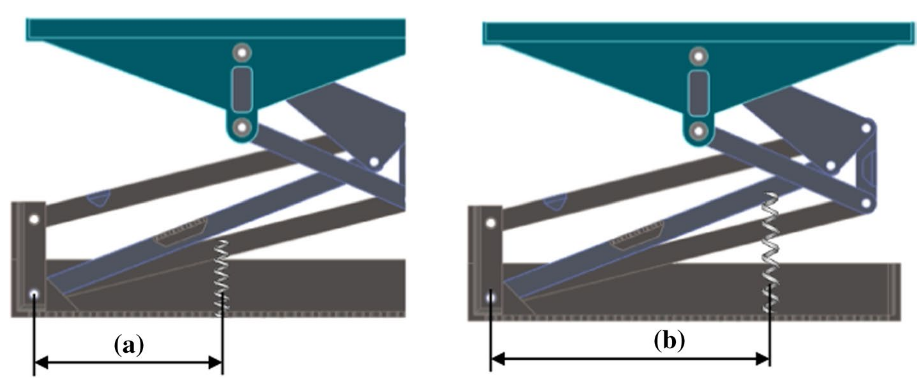

Fig. 9 Possible positions for the spring element. The spring is represented as mechanical spring for clarity, but in the application it was used the air spring described in the text. Position **a** linked to member 2, is suitable for small and rigid springs and position **b** linked to member 8, is for more compliant springs

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solution are the compactness (380 by 300 footprint with around 125 mm of height in closed position) and the low seat index point, a very high vertical travel (more than 130 mm) which is more than a 100% extension and moderate horizontal travel of the seat, within ± 6 mm. Moreover, the proposed embodiment grants a quite large space under the seat, more than half of the total suspension volume, which can be conveniently filled the auxiliaries and with a wide range of commercial pneumatic springs and dampers according to the dynamic needs of the vehicle, with several possible combinations. To conclude, the general design procedure of the analysis and synthesis of an eight-bar linkage mechanism and its optimization under given constraints for seat suspensions was shown in detail for a possible application in the agricultural field, but the same methodology is widely applicable to other applications and industrial contexts.

Appendix

Considering the same notation reported in Fig. 3, Eq. 1 can be rewritten as:

$$\begin{cases} L_4 c_4 = L_8 c_8 + L_9 c_9 \\ L_4 s_4 = L_8 s_8 + L_9 s_9 - L_1 \end{cases}$$

where $c_i = \cos\left(\theta_i\right)$ and $s_i = \sin\left(\theta_i\right)$. It is possible to square both equations and sum them together:

$$L_4^2 = L_8^2 + L_9^2 + L_1^2 + 2L_8L_9c_8c_9 + 2L_8L_9s_8s_9 - 2L_1L_8s_8 - 2L_1L_9s_9$$

Introducing the parametric formula: $s_\alpha = \frac{2t}{1+t^2}$, $c_\alpha = \frac{1-t^2}{1+t^2}$ where $t = \tan\left(\frac{\alpha}{2}\right)$ we have:

$$\left(L_1^2 + L_8^2 + L_9^2 - L_4^2 - 2L_1L_8s_8\right) + 2L_8L_9c_8\frac{1-t_9^2}{1+t_9^2} + 2(L_8s_8 - L_1)L_9\frac{2t_9}{1+t_9^2} = 0$$

Rearranging the members it becomes:

$$\left(L_1^2 + L_8^2 + L_9^2 - L_4^2 - 2L_1L_8s_8\right)\left(1+t_9^2\right) + 2L_8L_9c_8\left(1-t_9^2\right) + 4(L_8s_8 - L_1)L_9t_9 = 0$$

This is a second-degree equation in the term t_9:

$$At_9^2 + 2Bt_9 + C = 0$$

where

$$A = L_1^2 + L_8^2 + L_9^2 - L_4^2 - 2L_1L_8s_8 - 2L_8L_9c_8$$

$$B = 2(L_8s_8 - L_1)L_9$$

$$C = L_1^2 + L_8^2 + L_9^2 - L_4^2 - 2L_1L_8s_8 + 2L_8L_9c_8$$

So we have the well-known solution:

$$t_9 = \frac{-B \pm \sqrt{B^2 - AC}}{A} = f\left(\theta_8\right)$$

After that follows a sequence of algebraic simplifications made by the software Mathematica [40] leads to Eq. (2).

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