Design of a spring-loaded downforce system for a no-till seed opener

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Gratton, J., Chen, Y. and Tessier, S. 2003. Design of a spring-loaded downforce system for a no-till seed opener. Canadian Biosystems Engineering/Le génie des biosystèmes au Canada 45: 2.29 - 2.35. Constant downforce on a seeder disk opener is essential for maintaining a constant depth when seeding in no-till conditions. This study focused on the development of a mathematical optimization approach for the design of a no-till opener downforce system where the downforce would not change for more than ±10 and ±25% when the opener encounters minor (±50 mm) and major (±100 mm) micro-relief changes in a field, respectively. Two design alternatives, a spring-loaded single linkage and spring-loaded parallel linkage, were considered for the replacement of a hydraulically loaded downforce system. The prototype spring-loaded parallel linkage was tested in laboratory and in field conditions. When the opener unit ran over constructed obstacles representing a micro-relief change of ±50 and ±100 mm, the downforce (target values of 1023, 1360, and 1780 N) was varied by approximately ±13 and ±21%, respectively. Compared to the hydraulically loaded parallel linkage, the spring system resulted in approximately 50% smaller changes in downforce under the micro-relief changes studied. Field tests revealed a faster response of the spring-loaded system. The design approach of minimising the change in opener downforce is an effective method to ensure that the design criteria for downforce variation are met. Keywords: downforce, spring, opener, no-till, micro-relief, design, hydraulic.

INTRODUCTION

Single disc openers and double disc openers are the dominant design for no-till seeding of cereal grains and row crops. A no-till disc seeder usually consists of seed furrow openers, gauge wheels, and press wheels. Gauge wheels are ideal for controlling seeding depth to ensure uniform plant emergence. Press wheels are used to aid in seed-to-soil contact by closing and/or compacting the furrow.

Disc openers for no-till seeding require an effective downforce to cut through surface residues and penetrate hard soil to a specific depth (Payton et al. 1985). This force typically ranges between 700 and 2300 N (Schaaf et al. 1979) depending on the field conditions encountered. Janelle et al. (1993) studied seed placement of two double disc openers under three different downforces (669, 1338, and 2007 N). They reported that the smallest downforce resulted in insufficient seedling emergence and consequently poor crop emergence. Increasing downforce generally increases seedling emergence (Janelle et al. 1993). Maintaining a constant force on openers while seeding will help to achieve a uniform seedling emergence under a uniform soil condition. Constant seeding depth is important to achieve uniform seedling emergence (Choudhary et al. 1985; Chen et al. 2002).

Many studies have been conducted on seeding depth (Lawrence and Dyck 1990), opener types (Choudhary et al. 1985), opener design (Tessier et al. 1991b), press wheel configurations (Morrison 1989), and drill types (Tessier et al. 1991a). However, little work has been done on downforce system design, and there has been little success with the attempts made to minimise downforce variations on openers in laboratory and in field conditions as discussed below.

Springs, air cylinders (Fink and Currence 1995), or hydraulic cylinders (Morrison 1988b) have all been used to transfer force downward onto openers. In existing spring systems, the spring is normally set in a position where the vertical displacement of the opener alters the spring length resulting in a significant increase or decrease in the opener downforce. Micro-relief in a field varies due to clumps of crop residue, traffic tracks, and undulating soil surface. Changes in the opener downforce affect the mean depth of seeding across the seeder by their effect on individual opener penetration (Choudhary et al. 1985). Reducing the increase in downforce when the disc trips over an obstacle will also result in less wear and tear on the opener system. Therefore, the change in downforce should be minimised.
Several studies have been performed on monitoring the change in downforce by letting openers travel on constructed obstacles to simulate micro-relief changes in field conditions. Morrison (1988b) tested a downforce system consisting of a single compression spring. The change in downforce was determined under a static situation by lifting the opener with a load cell and reading the output. An increase in force by approximately four times was observed when the opener underwent an elevation change of 120 mm. The performance of a pneumatic system was evaluated in a laboratory situation by running an opener unit over a wooden obstacle 76 mm high with a 1.1 m flat section (Fink and Currence 1995). The downforce increased to approximately three and a half times the normal force when the opener traveled over the obstacle. Morrison (1988a) tested a hydraulic downforce system over an obstacle 38 mm high and 1.2 m long flat surface with 30° ramps. The downforce from running over the obstacle was 30% greater than the normal force.

The Ponik opener (ST AgriTech, Winnipeg, MB) is comprised of a drag arm carrying an offset double disc (Janelle et al. 1995a). The small disc (380 mm diameter) is oriented vertically whereas the large disc (460 mm diameter) is angled relative to both the direction of travel and the vertical plane. The gauge wheel (410 mm diameter by 100 mm wide) is located beside the small disc. A steel press wheel (360 mm diameter by 13 mm wide) follows behind the discs. The downforce is applied hydraulically with one cylinder for each pair of two openers (Janelle et al. 1995b). This opener has been used in drills for no-till conditions and renovation of forage stands in Central and Eastern Canada. This research was aiming at extending the use of the Ponik opener to large airseeders for Western Canadian conditions. Because hydraulic downforce systems are difficult and expensive to adapt to a large number of large airseeders, spring downforce systems were investigated.

The objective of this study was to apply an optimization approach for the design of a downforce system that allows the Ponik opener to adapt to an airseeder. The specific objectives were: (1) to develop a mathematically based approach for the design of a spring downforce system which minimises changes in the opener downforce in a field, (2) to validate this approach through laboratory and field tests on a prototype, and (3) to compare the performance of the spring downforce system with a hydraulic downforce system under laboratory and field conditions.

**DESIGN OF SPRING DOWNFORCE SYSTEM**

The target downforces for soil cutting with the Ponik opener’s discs were to be adjustable at 1023, 1360, and 1780 N. Target micro-relief changes for optimization purposes were set at ±50 and ±100 mm. Over these target micro-relief changes, the downforce system should maintain the downforce change within ±10 and ±25%, respectively, of the target downforce. As well, the downforce system must account for the likely encounter of obstacles as high as ±200 mm.

Two design alternatives were investigated: a spring-loaded single linkage and a spring-loaded parallel linkage. The spring-loaded single linkage was selected because it could easily adapt to the original arm of the Ponik opener, and the spring-loaded parallel linkage was chosen as it provides considerable compensation for the change in spring angle and maintains a constant relationship between the gauge wheel and the discs. For each alternative, an objective function was developed with the change in opener downforce, ΔDF, being the dependent variable. The independent variables in the objective function included the critical dimensions of the linkage, the spring constant, the position of the spring within the respective linkage configuration, and the opener assembly’s weight. The objective function was then minimised using a computer algorithm to obtain the feasible values for the independent variables.

**Spring-loaded single linkage**

The original arm of the Ponik opener was easily transformed to a spring-loaded single linkage (Fig. 1). The arm was connected to a support bracket at a pivot point O1. In the normal operating position, the angle between the arm and the horizontal plane was set at 30°. The weight of the modified opener assembly provided an equivalent force of 580 N applied 607 mm behind and 616 mm below the pivot point O1; this force was assumed as constant. The downforce (DF), which includes the force resulting from the opener’s weight, and the horizontal force (HF), acting on the opener disc at point D, create a moment (M_{O1}) about the pivot O1

\[ f_1 = M_{O1} = DF \cdot L \cos \theta + HF \cdot L \sin \theta \]  

where:

- L = distance between points D and O1, and
- \( \theta \) = angle to horizontal plane.

In normal operation, \( \theta \) is approximately 45° and deviates about 10° from that angle as a result of the major target micro-relief assumed for the design. The parameter L is assumed constant, as the arm and the disc mounting assembly form a rigid member. The horizontal translations in the actual contact point

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**Fig. 1. Configuration of the spring-loaded single linkage system and the position change of the linkage at the micro-relief change, Δh; DF = downforce; HF = horizontal force on opener; M_{O1} = moment about the point O1; L and \( \theta \) = distance between points D and O1 and the angle to the horizontal plane, respectively.**
Fig. 2. Diagram showing the dimensions of the links and the spring position relative to the arm and pivot in the spring-loaded single linkage system.

Fig. 3. Configuration of the spring-loaded parallel linkage system and the position change of the linkage for the micro-relief change (Δh); DF = downforce; HF = horizontal force on opener; M02 = moment about pivot point O2; and l = distance between the joint points E and O2.

between the disc and the soil at elevations other than the original soil datum is neglected for simplicity as it is small.

If the cutting depth of a tool is well controlled, the HF of the tool can be assumed as constant under a given travel speed. As the gauge wheel of the Ponik opener is close to the small disc, good depth control can be achieved (Morrison and Gerik 1985). Therefore, the HF of the Ponik opener in Eq. 1 was assumed to be constant.

If a compression spring (Fig. 2) is introduced between point A on the inclined arm and point B on the bracket, static equilibrium of \( M_{O1} \) by the compressive force (C) of the spring can be expressed as:

\[
f_3 = M_{O1} = C \cdot d \sin \delta + C \cdot e \cos \delta
\]  

where:
\( \delta = \text{angle between the axis of the spring and the arm, or} \)

\( \delta = \sin^{-1}\left(\frac{b}{c} \sin \phi\right) - \tan^{-1}\left(\frac{e}{d}\right), \) and

\( d, e = \text{dimensions of the bracket and arm (Fig. 2).} \)

By taking the derivatives of both Eqs. 1 and 2 with respect to \( \theta \), noting that:

\[
\frac{df_2}{d\theta} = \frac{df_2}{d\phi} = \frac{d\delta}{d\phi} \frac{df_2}{d\delta}
\]  

a new function, \( f_3 \), can be formed.

\[
f_3(DF, C) = \frac{df_1}{d\theta} \frac{df_2}{d\theta} d\theta dC
\]  

The function \( f_3 \) can be minimized by taking the derivative of Eq. 4:

\[
\frac{df_3}{d\theta} = \frac{\partial f_3}{\partial DF} d(DF) + \frac{\partial f_3}{\partial C} dC
\]  

and setting \( df_3 = 0 \). This results in the objective function:

\[
d(DF) = -1 \frac{1}{L \sin \theta} \left[ \left(\frac{b}{c} \cos \phi \right) \right] \left[ \frac{b \cos \phi}{\sqrt{1 - \left(\frac{b}{c} \sin \phi\right)^2}} \right] d\cos \delta - e \sin \delta dC
\]  

The compression force change, \( dC \), can be obtained from Hooke’s Law as:

\[
dC = k \cdot dc
\]  

where:
\( k = \text{spring constant, and} \)
\( c = \text{length of spring} \)
\[ = \sqrt{a^2 + b^2 - 2ab \cos \phi} \]

Spring-loaded parallel linkage

The spring-loaded parallel linkage (Fig. 3) included a vertical arm, two parallel links, and a vertical bracket to be fixed to the seeder frame. The mass of the parallel linkage provided an equivalent force of 880 N applied at 508 mm behind and 582 mm below the lower front pivot point of the parallel linkage. The force DF includes the opener assembly’s weight and the downforce applied to it. The two links remain in a horizontal position when the opener does not experience any change in micro-relief. This position is defined as the normal position where \( \theta = 90^\circ \). The moment \( M_{O2} \) can be expressed as:

\[
f_4 = M_{O2} = DF \cdot l \cos \theta + HF \cdot l \sin \theta
\]  

where: \( l = \text{distance between joint points E and O2 (Fig. 3).} \)

HF in Eq. 8 is assumed to be constant. If a tension spring is fixed to the positions on
the top link and the vertical bracket at distances a and b, respectively (Fig. 4), the tension in the spring, $T$, balances out $M_{O2}$ to give:

$$f_5 = M_{O2} = T \cdot a \sin \left( \tan^{-1} \left( \frac{\sin \theta}{a \cos \theta} \right) \right)$$ \hspace{1cm} (9)

Similar to the single linkage, a new function $f_6$ is formed as:

$$f_6(DF, T) = \frac{df_1}{d\theta} - \frac{df_5}{d\theta}$$ \hspace{1cm} (10)

The function $f_6$ is minimized in a manner similar to the minimizing of $f_1$ to give a second objective function:

$$d(DF) = \frac{-a}{l \sin \theta} \cos \left( \tan^{-1} \left( \frac{\sin \theta}{v} \right) \right) \frac{v \cos \theta - \sin^2 \theta}{v^2 + \sin^2 \theta} dT$$ \hspace{1cm} (11)

where $v = \frac{a}{b} \cos \theta$.

Once again, using Hooke’s law:

$$dT = k \cdot dc$$ \hspace{1cm} (12)

In this case, $c = \sqrt{a^2 + b^2 - 2ab \cos \theta}$

Selection of design parameters

As the final design was a compromise between the geometric limitations for fabricating the opener mounting linkage and the ideal dimensions that would minimise the downforce change, a trial and error approach was used where the objective functions were made as small as possible within a range of independent variables that remained practical. This was accomplished with a minisation function, constr in Matlab (The MathWorks Inc., 24 Prime Park Way, Natick, MA). The constr function uses a Sequential Quadratic Programming (SQP) algorithm to form a search direction in a line search procedure. Users can specify maximum and minimum constraints that must be in a continuous format, and the objective function must return a real value. The constraint was set as the maximum range of micro-relief: $\Delta h \leq 400$ mm (including both positive and negative micro-relief: $\pm 200$ mm). Each linkage was initially set at its normal operating position. The output of the minimising process did not converge to specific dimensions of the linkages. Instead, feasible trends (as described below) for the selection of the design parameters of linkage dimensions were deduced.

Spring-loaded single linkage

Point A should be as close as practical to the inclined arm when determining the dimension $e$ (Fig. 2); the dimension $d$ should be decided by placing point A as close as practical to the point O1. According to these criteria, the dimensions were determined and adjusted so that the compression spring with the selected spring constant, $k$, would fit in the available space. If the selected spring did not have the sufficient strength to achieve the target downforce, the variable $k$ was increased and the dimensions were adjusted again. This procedure was repeated until sufficient spring strength was identified. A spring constant of 53 N/mm was found to be most suitable for this design alternative. Practical dimensions for the linkage were selected while maintaining the downforce variations (values of the objective function) at ±12 and ±24% for $\Delta h$ of ±50 and ±100 mm, respectively.

Spring-loaded parallel linkage

The variable $l$ (Fig. 4) returned from the minimisation was the maximum value made possible by the constraint specified. The values of $a$ and $b$ returned were the minimum values made possible by the constraint and the two variables had to maintain specific proportions. The procedure of selecting $k$ was the same as for the spring-loaded single linkage and a spring constant of 17 N/mm was found to be most suitable for this design alternative. Practical dimensions for the linkage were selected, and the corresponding downforce variations were ±12 and ±23% for $\Delta h$ of ±50 and ±100 mm, respectively.

MATERIAL and METHODS

Prototype

Spring-loaded parallel linkage

A prototype of the spring-loaded parallel linkage (Fig. 5) was fabricated in accordance with the minimisation results. The spring could be connected to the top link at three positions, which allowed for three initial downforce settings (DF1 = 1023, DF2 = 1360, and DF3 = 1780 N). An operator would have the option of selecting different operating downforces to match different soil conditions.

Hydraulically-loaded parallel linkage

A hydraulic cylinder was also used in lieu of the spring in the prototype of the spring-loaded parallel linkage to compare the effectiveness of the spring to that of a hydraulic cylinder. The hydraulic system was constructed according to the information presented by Janelle et al. (1995b). The hydraulic cylinder had a 16 mm rod, a 25 mm bore, and a 152 mm stroke. The cylinder was connected at both ends in a closed loop attached to a pressure accumulator pressurized with nitrogen gas at 2800 kPa. A check valve was incorporated into the system and hydraulic oil was pumped into the system to achieve the desired hydraulic pressure. The hydraulic oil pressure was the same on both sides of the cylinder’s piston.

Laboratory test

The prototype of the spring-loaded and the hydraulically-loaded parallel linkage systems were tested in an indoor soil bin located...
in the Department of Biosystems Engineering at the University of Manitoba. The soil bin was 15 m long, 1.5 m wide, and 0.6 m deep. A firm test track was created on the soil surface by first levelling the soil and then compacting the soil with multiple passes of a steel roller.

**Experimental design** A factorial experiment was designed with the three initial downforce settings (DF1 = 1023, DF2 = 1360, and DF3 = 1780 N) and two ranges of micro-relief changes: $\Delta h_1 = \pm 50$ and $\Delta h_2 = \pm 100$ mm. The micro-relief change corresponding to $\Delta h_3 = \pm 200$ mm was excluded from the test because at this vertical displacement, the stroke of the cylinder was over its limit. The hydraulic pressures were adjusted to 2800, 6600, and 11300 kPa to achieve the corresponding target downforce settings of the spring system. Each treatment was replicated three times.

**Experimental procedure** Wooden obstacles (Fig. 5) were constructed to provide the desired positive and negative elevations. Both bumps and holes had 45° bevelled edges and a 400 mm flat surface section. For each test run, one bump and one hole were carefully placed on the track so that the edges were level with the surface of the track. To preserve the integrity of the track so that repetitive trials could be performed, the opener disc was removed and only the opener’s gauge wheel maintained ground contact (Fig. 5). The travel speed of the opener unit was kept constant at 5 km/h.

**Measurements** The opener unit was attached to the soil bin carriage through a dynamometer to measure the vertical force. Signals of vertical force were recorded with a data acquisition system at a sampling rate of 200 Hz. Weight of the opener was added to the outputs of the dynamometer. Values of downforce change, $\Delta DF$, were determined as the differences of the measured maximum and minimum forces on the opener unit, and was corrected for the mass of the absent discs.

**Field test**

**Site and equipment description** Field tests were conducted to examine the ability of the spring-loaded parallel linkage system to contour the soil profile of a field as compared with the hydraulic system. The field featured clay soil and cereal stubble and was located at Glenlea Research Station, University of Manitoba. The unit (Fig. 6) for field tests consisted of the opener unit (1) mounted on a toolbar frame (2).

**Experimental design** A completely randomized experimental design with three treatments, represented by the three initial downforces, was used. Four replicates of each treatment were performed. Each plot was approximately 4 m wide and 40 m long. Tractor travel speed was 3.25 km/h. Higher speeds were not used because of the excessive vibrations of the toolbar frame and opener unit.

**Measurements** A Conductive-Plastic Potentiometer (CPP) (3) (LWG450, Novotechnik, Southborough, MA) was installed between a cross bar (4) and the axle of the gauge wheel (5) to measure the vertical displacements of the opener unit (Fig. 6). A light profile-wheel (6) was mounted to the bar directly in front of the gauge wheel. A second CCP (7) was installed between the bar and the axle of the profile-wheel to register the surface datum. Displacement signals were recorded using a data acquisition system at 200 Hz and filtered using a 60 kHz digital filter, as the unit was operated in the field. Each test run consisted of 20 s of data recording.

**Data analysis**

To obtain time lags, $t_{lag}$, between the gauge wheel and the profile-wheel, the vertical displacement of the profile-wheel was used as an input signal into a linear system and that of the gauge wheel was used as the output of the same system (Fig. 7). The cross-correlation between these two signals represents the degree of correlation between them as a function of time interval between the occurrences of similar “events”. The cross-correlation function was obtained between the input and output signals using the function $cra$ in Matlab. The time corresponding to the maximum cross-correlation
Fig. 7. A segment of field signals for the vertical displacements of the profile-wheel and opener gauge wheel, showing the time difference of the two signals (Δt) and the soil deformation (d).

value for each set of data was taken as the time difference, Δt, between the two corresponding maximum displacements of those two signals (Fig. 7). From the apparatus operation speed, the time to travel the fixed distance l (Fig. 6) between the gauge wheel and the profile-wheel was determined and was subtracted from Δt, and the remainder was approximated as the tlag. The difference in vertical displacement between the corresponding maximum values of the profile-wheel and the gauge wheel signals was considered as soil deformation (d) associated with the impact of the gauge wheel. Analysis of variance was performed on the data. Means between the treatments were compared using Duncan’s multiple range tests at a significance level of P < 0.05.

RESULTS and DISCUSSION

Soil bin tests
Both initial downforce setting and micro-relief changes significantly affected the downforces and their variation, while their interactions were not significant. The downforce DF of the spring-loaded parallel linkage varied from approximately 835 to 2046 N over the entire soil bin test track, and that of the hydraulically-loaded parallel linkage varied over a notably wider range (Table 1). Reduced values of maximum, average, and minimum DF (Table 1) were observed at a lower initial downforce for both spring and hydraulic systems, due to the change in position of the spring and the cylinder. Both systems provide sufficient downforce adjustment range, considering the target downforce range.

Although the average forces were at similar levels for both spring and hydraulic systems, ΔDF of the latter system were much higher due to the fact that the bump resulted in a pressure surge that was not able to equilibrate quickly in the hydraulically loaded system. For both systems, the maximum downforces increased and the minimum downforces decreased with increased Δh, while the average forces were not affected by Δh. The amplitude of the downforce changes of the spring-loaded parallel linkage mounting, ΔDF, represented respectively ±13 and ±21%, at Δh1 and Δh2, of the average downforce. These values are in agreement with the calculated corresponding values (±12 and ±23%) of the objective function and the corresponding targeted values (±10 and ±25%). The ΔDF of the hydraulic system was twice that of the spring system.

Field tests
As compared to the spring system, a significant greater time lag, tlag, existed for the hydraulic system at the DF3 (Table 2), indicating a slower response of the opener as a result of the slower response of the hydraulic cylinder at a higher initial downforce. The value of tlag at the DF1 for the spring system was not consistent. The reason for this was unknown. Soil deformations (d) occurred along the path of the gauge wheel, implying that the opener may smoothen the original surface profile in a field condition. This was not taken into consideration in the soil bin tests where artificial surface profiles with rigid bumps and holes were used.

The d resulting from the opener gauge wheel increased with larger initial downforces in both spring and hydraulic systems (Table 2), as expected. The hydraulic system favoured higher d, especially at DF3, compared to the spring system. This trend was true for the other treatment levels, although not significant. The damping effect of oil friction in the hydraulic system was a

Table 1. Maximum, average, and minimum downforces and the change in downforce (ΔDF) measured from the soil bin tests.

<table>
<thead>
<tr>
<th>Treatment*</th>
<th>Spring-loaded parallel linkage</th>
<th>Hydraulically-loaded parallel linkage</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum (N)</td>
<td>Average (N)</td>
</tr>
<tr>
<td>DF1</td>
<td>1161 c**</td>
<td>909 c</td>
</tr>
<tr>
<td>DF2</td>
<td>1537 b</td>
<td>1208 b</td>
</tr>
<tr>
<td>DF3</td>
<td>2046 a</td>
<td>1632 a</td>
</tr>
<tr>
<td>Δh1</td>
<td>1458 C</td>
<td>1269 A</td>
</tr>
<tr>
<td>Δh2</td>
<td>1511 B</td>
<td>1257 A</td>
</tr>
</tbody>
</table>

* Initial downforce settings: DF1 = 1023, DF2 = 1360, DF3 = 1780 N; ranges of micro-relief change: Δh1 = ± 50 mm; Δh2 = ± 100 mm
** Means followed by the same lowercase letter or the same uppercase letter in each column are not significantly different according to Duncan’s multiple range test (P>0.05).
Table 2. Field results in time lags (\(t_{\text{lag}}\)) and soil deformations (d), 2001.

<table>
<thead>
<tr>
<th>Initial down force*</th>
<th>(t_{\text{lag}}) (ms)</th>
<th>d (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean(^1) SD(^{1\dagger})</td>
<td>Mean(^1) SD(^{1\dagger})</td>
</tr>
<tr>
<td>Spring-loaded parallel linkage</td>
<td></td>
<td></td>
</tr>
<tr>
<td>DF1</td>
<td>14.3 b 0.48</td>
<td>17.4 d 3.6</td>
</tr>
<tr>
<td>DF2</td>
<td>11.3 c 1.09</td>
<td>21.8 bc 1.1</td>
</tr>
<tr>
<td>DF3</td>
<td>14.0 b 0.75</td>
<td>23.4 b 2.3</td>
</tr>
<tr>
<td>Hydraulically-loaded parallel linkage</td>
<td></td>
<td></td>
</tr>
<tr>
<td>DF1</td>
<td>7.7 d 0.71</td>
<td>19.7 cd 0.7</td>
</tr>
<tr>
<td>DF2</td>
<td>10.1 c 1.43</td>
<td>22.5 bc 2.7</td>
</tr>
<tr>
<td>DF3</td>
<td>16.1 a 0.41</td>
<td>31.5 a 1.7</td>
</tr>
</tbody>
</table>

* Initial downforce settings: DF1 = 1023; DF2 = 1360; DF3 = 1780 N
\(^1\) Means followed by the same letter in each column are not significantly different according to Duncan’s multiple range test (\(P<0.05\)).
\(^{1\dagger}\) SD = Standard deviation

a likely cause for the slower system response to micro-relief changes, resulting in a higher peak force on the opener as it was forced to change position.

CONCLUSIONS

A design approach based on minimising the objective function developed from the mathematical description of the linkage and forces can be used to select practical design parameters for links and spring to meet set criteria for downforce changes of a disc opener in a field. For average downforces ranging from 900 to 1600 N, the downforce change measured in the soil bin for the spring-loaded parallel linkage system was maintained to approximately \(\pm 13\) and \(\pm 21\%\) for micro-relieves of \(\pm 50\) and \(\pm 100\) mm, respectively. Considering the corresponding target downforce changes of \(\pm 10\) and \(\pm 25\%\), this mathematically based design approach is very effective in meeting design objectives with complex spring-loaded linkage systems.

The soil bin and field tests results showed that a low initial downforce setting might be selected for a smaller downforce change and quicker response when the opener encounters a micro-relief change. As compared to the hydraulic system, the spring-loaded parallel linkage system had comparable or better performance. The spring-loaded parallel linkage will be commercialized for mounting openers for an airseeder.

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