DESIGN OF A RECONFIGURABLE DYNAMIC TESTBED FOR CO-DESIGN METHOD VALIDATION

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ABSTRACT

Physical testing as a technique for validation of engineering design methods can be a valuable source of insights not available through simulation alone. Physical testing also helps to ensure that design methods are suitable for design problems with a practical level of detail, and can reveal issues related to interactions not captured by physics-based computer models. Construction of physical and testing of physical prototypes, however, is costly and time consuming so it is not often used when investigating new design methods for complex systems. This gap is addressed through an innovative testbed presented here that can be reconfigured to achieve a range of different prototype design properties, including kinematic behavior and different control system architectures. Thus, a single testbed can be used for validation of numerous design geometries and control system architectures. The testbed presented here is a mechanically and electronically reconfigurable quarter-car suspension testbed with nonlinear elements that is capable of testing a wide range of both optimal and sub-optimal design prototypes using a single piece of equipment. Kinematic suspension properties can be changed in an automated way to reflect different suspension linkage designs, spring and damper properties can be adjusted in real time, and control system design can be changed easily through streamlined software modifications. While the specific case study is focused on development of a reconfigurable system for validation of co-design methods, the concept extends to physical validation using reconfigurable systems for other classes of design methods.

1 Introduction

Developing physical engineering testbeds for validating design methods is challenging in that testbeds must accommodate evaluation of a wide variety of designs, spanning both optimal and sub-optimal solutions (potentially for distinct design contexts). Creation of a testbed for a dynamic system entails additional challenges. In this article we present an innovative design of a reconfigurable dynamic quarter-car suspension testbed that has been developed specifically to test co-design solutions as a strategy for co-design method validation. Co-design [1–4] is a class of design methods that supports simultaneous optimization of physical (plant or artifact) and control systems [5] to generate system-optimal solutions for actively-controlled dynamic systems. A prominent feature of co-design is the ability to traverse design and control spaces simultaneously to capitalize on design coupling between physical and control systems. This supports engineers in efforts to achieve new levels of performance or new types of functionality.

Most existing hardware test platforms for dynamic systems, such as the Quanser Active Suspension\(^1\), support control system design changes with limited or no flexibility for physical design changes. We assume here that design method validation—and also the validation of design-appropriate predictive models that support design changes—requires tests of multiple distinct physical designs. Fabricating a prototype for each different physical design can be costly and time consuming. An alternative strategy is to develop a single testbed system that can be reconfigured physically to test a wide variety of designs and control

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\(^{1}\)http://www.quanser.com/products/active_suspension
architectures, which is the underlying motivation for the work reported here. Focus is on the development of an innovative new testbed through application of existing methods for reconfigurable systems and co-design. Ongoing work is addressing systematic methods for co-design method validation; detailing these validation methods is outside the scope of this article.

Testbed design for control systems has been well-explored. A sample of these systems include a control system testbed for critical infrastructure protection systems [6], a bipedal robot specifically designed to test a variety of control algorithms for stable-legged locomotion [7], and a platform for testing controls on a small cubesat-scaled satellite system [8]. Moreover, testbeds specifically designed for vehicular suspensions are also reported in the literature [9–11]. The main objective of these testbeds is to evaluate the performance—using metrics such as road handling or passenger comfort—for different control algorithms. A full-size passive suspension test rig is reported in Ref. [9] for testing suspension behavior up to 70 Hz excitation. A robust controller test rig is described in Ref. [10] for a 2-D linear suspension system. A suspension testbed for evaluating nonlinear sliding mode controllers is described in Ref. [11]. Kinematic and compliance testing is another type of study that is routinely performed during the suspension design process. An academic testbed for such study is described in [12] and the commercially available testbeds can be found in [13–15]. All of these testbeds however use non-re-configurable, fixed physical systems.

Reconfigurable engineering systems have also been explored with significant interest across several industries, including manufacturing. Reconfigurable, and to some extent flexible, manufacturing systems can produce a wide variety of parts. These systems are ideal for low quantity production where high variation is present. Flexible manufacturing systems eliminate the need for numerous separate manufacturing systems, reducing cost in many cases [16, 17]. In addition to improving manufacturing systems, reconfigurable capability can extend the capabilities of many other engineering systems. For example, aerospace systems can operate in a wider range of conditions by leveraging reconfigurability. This design strategy may also help reduce system mass. Reconfigurable airplane wings help tailor aerodynamic properties for different flight conditions, and can help improve fuel efficiency [18, 19]. Robotic systems can utilize modular [20] or continuous [21] reconfigurability to improve adaptability by completing a wide variety of tasks with a single robotic system. Non-mechanical systems can also benefit from reconfigurability. For example, microprocessors can automatically reconfigure electronic connectivity to enhance efficiency for distinct computational tasks [22, 23].

A recurrent theme appears in the motivation for reconfigurable systems: adaptability, which is key to achieving improvements in cost, robustness, and operating efficiency. These improvements, however, require compromise. For example, a single reconfigurable system generally cannot perform as well at individual tasks as unique systems designed to perform them optimally. In addition, for continuously reconfigurable systems, increasing the range of reconfiguration (such as link length adjustment) improves adaptability, but increases cost, mass, and volume of the system. A critical element, therefore, of reconfigurable system design is to determine how to allocate strategically reconfigurable capability (e.g., adjustment range for continuously reconfigurable systems) across the system to maximize overall utility [21]. Since the motivation for creating this testbed is to validate a wide range of optimal and suboptimal suspension designs [24], as well as supporting experiment-in-the-loop design optimization (EILDO) studies, the utility of the proposed system can be captured by the effectiveness of reconfigurability with respect to testbed purpose. More precisely, simply increasing reconfigurability level may not always improve testbed usefulness. Enhancing reconfigurability in a way that does improve testbed usefulness should be the driving design objective.

Here we seek to balance the usefulness of the reconfigurable system as a testbed with cost and other considerations. The suspension testbed must support variations in physical and control design using a single physical system that enable tests of specific designs that are important for co-design method validation. It should also support rapid, automated reconfiguration and automated dynamic performance evaluation to increase the rate at which unique designs can be tested. A core question is how to best enhance system adaptability in a tailored manner to yield a testbed that can evaluate a useful range of different geometric system configurations while limiting system cost. The novelty of the proposed testbed lies in its ability to change physical design within a strategic attainable plant design domain, while also supporting a variety of different control algorithms.

The structure of the rest of the paper is as follows: Section 2 provides the background on trailing-arm based automotive suspension, followed by discussion of suspension co-design in Section 3. Section 4 details testbed design, and the paper is concluded in Section 5 with the identification of future work. The core contribution of this article is the presentation of a novel approach for testbed development, both for design method validation and EILDO studies, specifically for active automotive suspension systems. The work presented here lays a foundation for studying fundamental aspects of design theory and methods for actively controlled systems. The contribution here is characterized more aptly as a design innovation supporting research, as opposed to a fundamental scientific contribution.

2 Active Automotive Suspension Design

A well-known design problem used in co-design studies is based on a quarter-car model of an active automotive suspension system. Allison et al. [3] solved the active suspension co-design problem for the first time using a comprehensive mechanical design model, and illustrated how optimal system design changed during the transition from passive to strongly-active
control by varying control authority bounds. To test these resulting co-design solutions, we would not only need the ability to test different control schemes, but also test different physical designs. Rather than fabricate a new suspension testbed for each co-design result, a reconfigurable trailing-arm type suspension testbed was developed that can evaluate the dynamic performance of a wide variety of active suspension system designs. A simplified schematic of this testbed is illustrated in Fig. 1.

![Simplified schematic of testbed](image)

**FIGURE 1.** Reconfigurable testbed schematic

While the studies of Ref. [3] involved a nonlinear relationship between plant design variables and model parameters and design constraints, the suspension was modeled as a linear dynamic system. This testbed builds upon this model by incorporating nonlinear trailing arm suspension kinematics. The kinematic design is specified by the geometric reconfiguration variables \([x_{p1}, \ldots, x_{p4}]\). In addition to kinematic design changes, the testbed supports adjustments to spring stiffness \(K_{\text{lin}}\) and damping rate \(c_s\). Note that even if the spring and damper components are linear, effective stiffness and damping behavior will be nonlinear due to nonlinear kinematics if deflections are large enough to extend beyond linear operation.

Figure 1 shows the road profile elevation \((z_0)\) that changes spatially and with time, corresponding to changes in road elevation as the vehicle moves longitudinally with velocity \(v\). The unsprung mass corresponds to everything that moves with the wheel/tire to follow the road surface, and its position is \(z_{us}\). The sprung mass is everything that moves with the vehicle body, and its position is \(z_s\). Unsprung mass acceleration \(\ddot{z}_{us}\) may be large as the tire approximately follows \(z_0\). Difference between \(z_{us}\) and \(z_0\) is due only to tire compliance, which is small since typically \(K_{\text{lin}} > K_t\). Tire stiffness \(K_t\) is assumed not to be adjustable here.

Reducing tire contact force variance helps improve vehicle handling. This force is approximately proportional to tire deflection: \(z_{us}(t) - z_0(t)\) (assuming that tire damping forces are small in the vertical direction). Acceleration experienced by vehicle occupants \((\ddot{z}_s)\) should be minimized to improve ride comfort. This simple acceleration metric has been found to approximate more sophisticated comfort measures accurately [25]. An ideal suspension thus minimizes both \(\ddot{z}_s\) and the variance in tire deflection. Active suspension control can help improve both metrics simultaneously compared to passive system performance [3].

### 2.1 Suspension Performance Quantification

Design changes that improve handling (e.g., stiff suspensions, lightweight/stiff low-profile tires) often degrade comfort, and design changes that improve comfort (e.g., soft suspensions and tires) often degrade handling. The following approximate metrics are used to quantify this tradeoff:

- **(ride quality)**: \(J_1 = \int_{t_0}^{t_f} \ddot{z}_s(t)^2 \, dt\)
- **(road handling)**: \(J_2 = \int_{t_0}^{t_f} (z_{us}(t) - z_0(t))^2 \, dt\),

where \(t_f - t_0\) is the test duration. These simple metrics have been found to fairly represent passenger comfort and handling, respectively [26–28]. Lower overall sprung mass acceleration corresponds to improved passenger comfort \((J_1)\). Low normal force variance at the contact patch typically improves vehicle handling due to better tire contact \((J_2)\). The conflict between comfort and handling design objectives can be analyzed using standard methods for multi-objective optimization, and understood conceptually by visualizing the tradeoff by plotting the corresponding Pareto set (not shown).

One method to generate a Pareto set is to minimize one objective while constraining the other, and then repeating the procedure with different bounds on the constrained objective. Another approach is to form a scalar weighted sum: \(J = r_1 J_1 + r_2 J_2\), where \(0 \leq \{r_1, r_2\} \leq 1\) are the objective weights, and \(r_1 + r_2 = 1\). Weight values can be varied to find different points on the Pareto set; these points correspond to different designs that are among a larger set of designs that are desirable to evaluate experimentally using the testbed for co-design method validation. This method of Pareto set generation is limit in ability to handle non-convex function spaces, however, is presented here as an example method to generate a Pareto set.

### 3 Co-design Optimization Studies

To determine an appropriate range of candidate optimal and sub-optimal systems, preliminary co-design studies were performed. An example co-design problem for the suspension system, including a description of the dynamic equations of motion in the state-space form, is presented in Eqn. (1):

\[
\begin{align*}
\min_{x_{p, u}(t)} & \int_{t_0}^{t_f} \left( r_1 \ddot{z}_s^2 + r_2 \dddot{z}_s^2 \right) \, dt \\
\text{subject to:} & \quad \dddot{z}_s = A(x_p, \dddot{z}_s) + B_1 q + B_2 F(u) 
\end{align*}
\]
where $\xi$ is the vector of system states. $K_r$ and $C_{l}$ are the tire stiffness and damping rates. $K_r(x_p, \xi)$ represents the dependence of stiffness on state ($\xi$) due to nonlinear kinematics, and on physical design ($x_p$). This kinematic mapping is derived a-priori and incorporated into the system dynamic model (specifically, the system matrix in Eqn. (1e)), resulting in a linear state-dependent model. $M_s$ and $M_{ss}$ are the sprung and unsprung mass values, respectively, and $q(t)$ is the road disturbance that is determined by a spatial road profile and the vehicle velocity $v$. The physical design vector $x_p = [x_{p1}, x_{p2}, x_{p3}, x_{p4}, K_{lin}]^T$ combines the design physical variables that are illustrated in Fig. 1. $K_{lin}$ is the physical stiffness of the spring in its linear operation. The physical constraint matrix $A_p$ ensures that the length $x_{p4}$ does not exceed $x_{p1}$ and $x_{p2}$ does not exceed $x_{p3}$.

The objectives in this formulation are as described in the previous section, based on the state vector defined in Eqn. (1f). $F(u)$ is the force generated by control action (by an actuator in the active case, by the magneto-rheological (MR) damper in semi-active and passive cases). The results obtained in this specific case study are based on the simulated rough road input that has an International Roughness Index (IRI) of 7.37 m/km, corresponding to a maintained unpaved road [25, p.170]. Other road profiles may be simulated using the road disturbance emulator mechanism that is described in Section 4. System parameters listed in Table 1 were used.

**TABLE 1.** Fixed parameters for a representative co-design study

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M_{ss}$</td>
<td>65</td>
<td>kg</td>
</tr>
<tr>
<td>$M_s$</td>
<td>325</td>
<td>kg</td>
</tr>
<tr>
<td>$K_r$</td>
<td>$2.32 \times 10^5$</td>
<td>N/m</td>
</tr>
<tr>
<td>$C_{l}$</td>
<td>0</td>
<td>N/m/s</td>
</tr>
<tr>
<td>$t_F$</td>
<td>5</td>
<td>s</td>
</tr>
<tr>
<td>$v$</td>
<td>10</td>
<td>m/s</td>
</tr>
<tr>
<td>$r_1$</td>
<td>0.5</td>
<td>–</td>
</tr>
<tr>
<td>$r_2$</td>
<td>0.5</td>
<td>–</td>
</tr>
</tbody>
</table>

The co-design problems for this study were solved using direct transcription with trapezoidal collocation [3, 29–31]. Solution accuracy was verified using a high-order simulation. Figures 2–4 present the results obtained for passive, semi-active, and active co-design optimization problems, respectively. Table 2 lists the corresponding optimal physical design variables and $J(\cdot)$ values. The active system performs best, but the semi-active system improves upon the passive system significantly. These results are corroborated by the trajectories shown in Figs. 2–4; oscillation amplitudes for $z_s$, $\dot{z}_s$, and $\ddot{z}_s$ are reduced when transitioning from passive, to semi-active, and finally, to active system designs, thereby reducing $J(\cdot)$. This is an expected behavior from these systems, and future work will include validation of these and other behaviors on the testbed.

**TABLE 2.** Optimal physical design and objective function values for passive, semi-active, and active systems.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Passive</th>
<th>Semi-active</th>
<th>Active</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x_{p1}$</td>
<td>m</td>
<td>0.62</td>
<td>0.43</td>
<td>0.57</td>
</tr>
<tr>
<td>$x_{p2}$</td>
<td>m</td>
<td>0.09</td>
<td>0.10</td>
<td>0.22</td>
</tr>
<tr>
<td>$x_{p3}$</td>
<td>m</td>
<td>0.17</td>
<td>0.11</td>
<td>0.58</td>
</tr>
<tr>
<td>$x_{p4}$</td>
<td>m</td>
<td>0.18</td>
<td>0.37</td>
<td>0.22</td>
</tr>
<tr>
<td>$K_{lin}$</td>
<td>N/m</td>
<td>23,841</td>
<td>27,634</td>
<td>20,094</td>
</tr>
</tbody>
</table>

**Objective** $- 7.40 \times 10^3$ $3.72 \times 10^{-4}$ $2.06 \times 10^{-4}$

4 Design of Dynamic Reconfigurable Suspension Testbed

Using the preliminary results obtained in the previous section, a range of designs to validate experimentally, both optimal and suboptimal, is determined. Physical tests can be conducted to evaluate system performance by first reconfiguring the system according to a specified plant and control system design, simulating the road disturbance using the shaker table, recording the dynamic response of the system, and assessing results. Associated design specifications based on testbed needs are presented in Table 3.

In this section we describe the process used to design the dynamic reconfigurable suspension testbed system that meets the design specifications. The recurring theme in each design process stage (Fig. 5) is the strategic maximization of reconfigurability to accommodate a larger, more effective range of designs that can be tested.

4.1 Testbed Design Planning

Early in the process, specific required system functionality is identified. Here we assume a system architecture (i.e. trailing-arm suspension), but other elements of testbed architecture are not. A critical question to address during design planning is the spectrum of studies that the testbed should be capable of performing. Both system-optimal and suboptimal designs are of interest here to support performance comparisons between co-design and conventional design methods. The first step in the planning process is to assemble a set of representative co-design
formulations with a range of different objective functions, constraints, control strategies, and design variable sets. Design formulations that do not produce system-optimal results (such as sequential design) should be added to the set of representative sets. Parametric studies are then performed using these design optimization formulations to explore the range of design variable values (both physical and control) that should be achieved through reconfiguration. For example, the range of actuation forces, damping and spring rates, and geometric parameter values observed through these studies can serve as a quantitative basis for forming reconfigurability range objectives.

There is an inherent trade-off between reconfigurability and system cost [21]. Increasing either the number of system elements that are reconfigurable, or the range of element reconfigurability, improves system utility, but incurs additional cost. The challenge then is to provide efficient reconfigurable capability within cost requirements in a way that provides the greatest testbed value. Enhancing the range of reconfigurability for
TABLE 3. Reconfigurable suspension testbed design specifications

<table>
<thead>
<tr>
<th>Description</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum power</td>
<td>1800W (Wall outlet: 120V, 15A)</td>
</tr>
<tr>
<td>Testbed Mobility</td>
<td>Must fit through a standard doorway</td>
</tr>
<tr>
<td>Reconfigurable Control</td>
<td>Spring, damping, and control force must be variable</td>
</tr>
<tr>
<td>Reconfigurable Geometry</td>
<td>Spring, damper, control force locations must be variable</td>
</tr>
<tr>
<td></td>
<td>Suspension control arm length must be variable</td>
</tr>
<tr>
<td>Automated Reconfiguration</td>
<td>All reconfiguration to be performed via software GUI and command line</td>
</tr>
<tr>
<td>Material Cost</td>
<td>Less than $15,000</td>
</tr>
</tbody>
</table>

some combinations of elements may be more important than others. Eigenvector analysis and sensitivity studies may be used to identify what individual and combinations of reconfiguration elements provide the greatest impact on testbed value. The result of this process is a decision regarding what system elements should be reconfigurable, what the desired reconfigurability ranges are, and identification of interactions between reconfigurable elements.

After analysis of co-design study results, the reconfigurable system elements for the suspension testbed were selected as illustrated in Fig. 1. Desired ranges were also identified. These quantities were important for testbed concept development, particularly for selecting appropriate technology for reconfiguration and actuation.

4.2 Testbed Concept Development and Selection

Based on the findings of preliminary studies, three different concepts were developed (Fig. 6). The concepts are distinguished by active control mechanism. Concept 1 uses a linear actuator in plane with the spring and damper to provide active force control between spring and unsprung masses. Concept 2 uses a linear motor offset from the suspension to provide active control. Concept 3 uses a rotary actuator at the revolute joint to impart an active control torque. Each of the concepts was evaluated and assigned a rating on: 1) the range of achievable reconfiguration, 2) cost of the concept, and 3) energy requirements. The highest weight (0.5) was assigned to reconfigurability—defined as force/torque range and accommodation of variable kinematics—the most significant design consideration for this particular testbed. The remaining metrics were considered equal in importance and assigned weights such that the sum of all weights is 1. The cost in Table 4 is not the cost of reconfigurability but the estimated monetary cost of a particular concept. For example, concept 2 requires a costly linear motor to provide active control; hence, its cost rating is worst.

Based on the table, Concept 3 (ref. Fig. 7) provides the best alternative for active control strategy according to the weighted criteria. The rotary actuator does not inhibit reconfiguration as the linear actuator would, and is low cost and has relatively low energy requirements.

4.3 System Design

With the final design concept (Concept 3) selected, system-level design can proceed. The final concept contains four unique primary dynamic components: a spring, a damper, a rotary motor, and a linear actuator. Based on the range of spring stiffness, damping rates, and tire contact forces identified through preliminary studies, the selection of these components is described now.

4.3.1 Damper A range of desired damping coefficients was identified from design study data. Off-the-shelf damping devices that support variable damping rates were assessed, and two important candidates are presented in Table 5. Electro-rheological (ER) and magneto-rheological (MR) dampers are attractive as they provide a continuous adjustability of damping properties as a function of applied control current. ER and MR...
dampers respond very quickly to changes in input commands, supporting their use in real time feedback control systems. MR dampers are particularly attractive over ER dampers due to low energy requirements and inherent bounded-input bounded-output (BIBO) stability [33]. Based on our specific adjustability needs, the LORD 8041-1 MR damper was selected for the testbed.

The LORD MR Damper has a total stroke of 2.91 in and operates between 0 and 2A (peak) of current. Experiments were conducted using an MTS810 Test Frame and an Instron 8500 controller to characterize the behavior of the damper. The damper was forced to follow a sinusoidal position profile on the test frame. Three different experiments were conducted with sinusoidal frequencies of 0.39, 0.79, and 1.59 Hz. At each sinusoidal frequency the damper behavior was tested while varying the input current from 0 to 1A in increments of 0.1A. Damper hysteresis from the 0.79 Hz tests is evident in Fig. 8, as is the increase in damping force with MR damper field current. The flat spots (where force is approximately zero irrespective of the velocity) reflect the play in the damper mounting system on the test frame. Test results helped to identify the maximum damper reconfigurability range and were used in the creation of damper models used in control and simulation.

In simplified damping models damping force depends linearly on damper velocity. This is not the case in real dampers. Allison, Guo, and Han presented a design strategy for achieving near-linearity for viscous dampers [3], but for most systems this behavior is undesirable. The force-velocity relationships illustrated in Fig. 8 are more typical of realistic damper force-velocity relationships where slope is higher at low velocities. That said, the MR damper in the testbed could be controlled in such a way to mimic a linear damper if required for specific design tests that are limited to linear systems. Alternatively, a co-design problem may be formulated where the range of available nonlinear damping properties are given by the data in Fig. 8.

FIGURE 6. Concepts for active control mechanism.

TABLE 4. Concept selection for reconfigurable system (rating of 3 is best, 1 is worst).

<table>
<thead>
<tr>
<th>Selection Criteria</th>
<th>Concept 1 Linear Actuator</th>
<th>Concept 2 Linear Motor</th>
<th>Concept 3 Rotary Motor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight</td>
<td>0.50</td>
<td>0.25</td>
<td>0.25</td>
</tr>
<tr>
<td>Rating</td>
<td>1</td>
<td>3</td>
<td>1</td>
</tr>
<tr>
<td>Score</td>
<td>0.50</td>
<td>0.75</td>
<td>0.50</td>
</tr>
<tr>
<td>Rating</td>
<td>2</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>Score</td>
<td>1.00</td>
<td>0.25</td>
<td>0.50</td>
</tr>
<tr>
<td>Rating</td>
<td>3</td>
<td>3</td>
<td>0.75</td>
</tr>
<tr>
<td>Score</td>
<td>1.50</td>
<td>1.75</td>
<td>2.75</td>
</tr>
<tr>
<td>Total Score</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rank</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
</tbody>
</table>

TABLE 5. Damper Comparison

<table>
<thead>
<tr>
<th>Damper</th>
<th>Stroke</th>
<th>Adjustability</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>DHX Air 5.0</td>
<td>3 in</td>
<td>Discrete</td>
<td>$411</td>
</tr>
<tr>
<td>LORD 8041 MR Damper</td>
<td>2.91 in</td>
<td>Continuous</td>
<td>$400</td>
</tr>
</tbody>
</table>

FIGURE 8. Force vs. velocity data for damper at different current inputs showing hysteresis

2http://www.lordmrstore.com/lord-mr-products/rd-8041-1-mr-damper-long-stroke
4.3.2 Spring Choosing a specific damper places restrictions on the geometry of a feasible spring as the damper has a fixed stroke length and defined size. Several types of mechanical springs were evaluated for their mechanical, reconfigurable, and other properties. Two important spring candidates are presented in Table 6.

<table>
<thead>
<tr>
<th>Spring Type</th>
<th>Cost</th>
<th>Size</th>
<th>Stroke</th>
</tr>
</thead>
<tbody>
<tr>
<td>Automotive Air Spring</td>
<td>$39</td>
<td>3.5&quot; x 10&quot;</td>
<td>4&quot;</td>
</tr>
<tr>
<td>Pneumatic Cylinder</td>
<td>$179</td>
<td>3&quot; x 17&quot;</td>
<td>4&quot;</td>
</tr>
</tbody>
</table>

A pneumatic cylinder with a controllable pneumatic pressurization system was selected to provide adjustable spring stiffness (specifically, a Speedaire pneumatic cylinder with a 100 mm (3.94 in) stroke and pressure range of 0–250 psi).

Using the same test equipment as the damper, experiments were conducted to characterize the response of the spring under different chamber pressures. The cylinder was forced into constant rate compression from 0 mm to -70 mm, and then into a constant rate of tension from -70 mm to 0 mm. This experiment was conducted across a range of chamber pressures (0–100 psi), and the results presented in Fig. 9 were used to develop a nonlinear spring stiffness model.

4.3.3 Road Disturbance Emulator A road disturbance emulator is developed to mimic different road profiles as shown in Fig. 10. This emulator can manage up to 80 lbs of vertical thrust load at speeds of up to 40 in/sec. The emulator is used to provide a wide range of road disturbance profiles, such as a sinusoidal road disturbance with unbiased white noise, as described by Eqn. (2).

\[ z_0(t) = Z_0 \sin(\omega t) + \mathcal{N}(0, \sigma) \]  

Here \( Z_0 \) is road disturbance amplitude, \( \omega \) is disturbance frequency, and \( \mathcal{N}(0, \sigma) \) is a random process with zero mean and standard deviation \( \sigma \). Figure 10 shows the road disturbance emulator sub-assembly developed for the suspension testbed. This sub-assembly consists of a JVL MAC400 servomotor\(^3\) integrated with an high speed Tolomatic ERD10\(^4\) lead screw\(^7\). The lead screw moves the vertically movable platform that supports the suspension testbed wheel. The vertical motion of the platform approximates the vertical motion experienced by the wheel as the vehicle moves across a road surface.

4.3.4 Active Control Actuator A harmonic drive provides active control actuation for the testbed. Active control can improve both sprung mass isolation from road disturbances, and improve road-tire contact. Rotary actuation is preferred over linear actuation for its compactness, wide range of torque, low cost, and accommodation of a wide range of kinematics. Linear actuator options would further limit kinematic capabilities of the reconfigurable testbed. The specific actuator chosen here is

\(^3\)http://www.jvl.dk/703/mac400-integrated-servo-motor
the FHA-32C-100-US250A actuator by Harmonic Drive LLC. This actuator provides a peak torque of 398 N/m and a maximum speed of 40 rpm. Please see Ref. [34] for a detailed dynamic model of this harmonic drive. Note that the rotary actuator attachment mount on upper platform is also electronically adjustable (refer Fig. 7) to support reconfiguration.

4.4 Detail Design

Detail design of the testbed was performed once concept selection and system-level design phases were complete. This involves detailed geometric design of components, strength analysis using finite element analysis, characterization of sensors and actuators, and mechatronic system integration and embedded system programming. While most elements of detail design process are omitted here for brevity, we will discuss the reconfiguration mechanism, suspension mounting bracket, and the approach used to design the mounting brackets for the spring and damper. This design has direct impact on the range of reconfigurability for many kinematic elements of the system. Optimization was used to ensure that reconfigurability range and effectiveness was maximized. The resulting design is optimal only with respect to the specific damper and spring components selected.

4.4.1 Reconfiguration Mechanism

The geometric reconfiguration of the system is achieved by lead screw mechanism with stepper motors. The reconfigurable trailing arm assembly is presented in Fig. 11. The lower spring and damper pivots are placed on a linear bearing for structural support and are positioned by lead screw actuation. Trailing arm length can also be adjusted through lead screw actuation. Linear bearings connect nested rectangular aluminum tubing to form a sliding joint that enables this length change.

4.4.2 Suspension Bearing Mounts

To ensure that the suspension does not bind under any loading condition, the 2:1 manufacturers suggested ratio of bearing distance to load distance was observed. Specifically, the center-to-center distance between the lower and upper bearings, $L$, is at least half the distance between the bearings and the center loading, $D$ (i.e., $2L \geq D$) as shown in Fig. 12.

4.4.3 Mounting Bracket Design Optimization: Maximum Reconfigurability

The objective of this task was to design the geometry of the spring and damper mounting brackets to maximize reconfigurability range, given spring and damper travel and other kinematic aspects of the system. The design objective is to maximize the function

$$J(x) = (\Delta m(x))^2 + (\Delta q(x))^2,$$

where $\Delta q$ and $\Delta m$ are the maximum achievable reconfigurability range on the lower control arm and upper platform, respectively (Fig. 13). A poor bracket design has the potential to severely constrain adjustment range for these bracket locations. The mounting bracket designs were parameterized using the variables $x = (x_1, x_2, x_3, x_4, x_5)$, as illustrated in Fig. 13. The optimization formulation is:

$$J(x^*) = \max_x (\Delta m(x))^2 + (\Delta q(x))^2$$

subject to:

$$q_{\text{max}} := \max_x \Delta q$$

$$q_{\text{min}} := \min_x \Delta q$$

$$m_{\text{max}} := \max_x \Delta m$$

$$m_{\text{min}} := \min_x \Delta m$$

$$x_L \leq x \leq x_U,$$

$$2L \geq D.$$
Figure 13 illustrates graphically how the control arm impacts where we can feasibly locate the lower bracket on the mounting bracket design problem. Graphical representation of the optimal solution to the optimization problem was in essence to find a bracket design that maximized the area between the two curves in the feasible zone.

4.5 Final Testbed Design

Based on the results of numerous design studies, the suspension testbed was constructed. A CAD rendering and the physical testbed are illustrated in Fig. 4.5. The geometric reconfiguration of the system, specified by $[x_1, x_2, x_3, x_4]$, is driven by lead screw mechanisms developed to maximize linear adjustment range within a given space. Note that the horizontal location of the contact patch changes with suspension configuration. To account for this lateral displacement and ensure that the contact patch is near the center of the shaker table, the road disturbance emulator assembly can be repositioned horizontally using a lead screw driven track. Both the MR Damper and air spring were selected to support testing of a variety of other control systems. The stiffness of the damper is modified directly by a variable current supply. The stiffness of the spring is modified via air pressure adjustments, which are controlled using a pneumatic systems consisting of a controlled pressure regulator, pneumatic solenoid valves, and pressure sensors.

To support feedback control and model validation, the testbed was fitted with several data-collecting sensors. The upper and lower platforms were equipped with linear encoders to measure $z_1$ and $z_2$ directly. The upper platform is equipped with an accelerometer, to measure sprung mass acceleration $\ddot{z}$, directly. The trailing arm is equipped with an inertial measurement unit (IMU), to collect both acceleration and orientation data. They are used to determine the position of the wheel, $\dot{z}_{ax}$. Sensor outputs are read via Simulink, filtered and utilized in the control interface.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Optimal Value</th>
<th>Variable</th>
<th>Optimal Value</th>
<th>Unit</th>
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<td>$x_4$</td>
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<td>in</td>
</tr>
<tr>
<td>$J$</td>
<td>54.24</td>
<td></td>
<td></td>
<td>in²</td>
</tr>
</tbody>
</table>

Table 7. Optimal design vector for the mounting bracket design optimization problem.

4.6 Control Interface

Both reconfigurability and data logging is controlled by MATLAB/Simulink® via the Arduino Support Package. Important features are combined into a graphic user interface (GUI) as a convenient interface. Parameters guiding the geometry of the system can be adjusted, and commands are sent to the Arduino® when an update button is clicked. The testbed can also be controlled through a command line interface to support sets of automated tests or experiment-in-the-loop design optimization.

The testbed supports all three control schemes. When passive mode is selected, both the the spring and damper rates are...
fixed. The system can control the damper in a way that emulates a linear damper. When semi-active mode is selected, the system controls the MR damper in real-time. In the default implementation, full-state feedback gain values for the damper are obtained a priori through control design optimization. The active system controls the harmonic drive to improve dynamic system performance. Control designs for both the MR damper and the harmonic drive may also be specified directly, which is particularly important for testing co-design results from a candidate co-design method.

Mechanical reconfiguration and control system specification are both pre-experiment procedures to adjust the system before performing dynamic tests. The final step is to select a road disturbance profile. A GUI supports suspension reconfiguration, road profile selection, test execution, and results analysis. Experimental results can then be compared to co-design simulation data for method validation. Moreover, three different control schemes with different levels of control authority can be tested using this testbed: active control, by using the rotary actuator, and passive and semi-active control, by swapping out rotary actuator for a revolute joint. An additional use of this reconfigurable testbed has been to provide engineering undergraduate students with an opportunity to evaluate experimentally the outcomes of suspension design projects [24].

5 Conclusions and Future work

A novel testbed for validating co-design methods and conducting EILDO studies was developed that supports automated physical testing of a wide range of physical and control system designs. Automated reconfiguration enables rapid testing of systems with distinct physical and control system designs without the need to construct new physical prototypes. Co-design and parametric studies were performed to identify a desired range of system properties that the testbed should access. Recently developed methods and principles for design of continuously reconfigurable engineering systems were used to develop an efficiently reconfigurable system design.

The use of automated reconfigurability in the development of a testbed for design method validation is a novel concept. Existing design methods for reconfigurable systems were useful, but development of design method testbeds presents unique challenges – enabling the ability to test wide ranges of non-optimal and optimal designs. This motivates further investigation and advancement of design frameworks for reconfigurable systems, including study of utility functions for assessing the value of specific reconfiguration ranges in the context of specific design testbed needs. The work presented here is limited to the process used and the resulting design method. Ongoing work is addressing theory and methods for using physical tests in design method validation studies, setting up experiment in the loop design optimization studies using the testbed, to extract insights not accessible through simulation alone.

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References


