DEVELOPMENT OF A BIAXIAL LOADING FRAME FOR THIN SHEET CRUCIFORM SPECIMENS

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DEVELOPMENT OF A BIAXIAL LOADING FRAME FOR THIN SHEET CRUCIFORM SPECIMENS

BY

JOSEPH F. WILSON

Baccalaureate of Science, University of New Hampshire, 2011

THESIS

Submitted to the University of New Hampshire
in Partial Fulfillment of
the Requirements for the Degree of

Master of Science
in
Mechanical Engineering

May, 2015
This thesis has been examined and approved in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering by:

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On December 19, 2014

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ABSTRACT

DEVELOPMENT OF A BIAXIAL LOADING FRAME FOR THIN SHEET CRUCIFORM SPECIMENS

by

Joseph F. Wilson

University of New Hampshire, May, 2015

Characterization of the evolving yield loci and forming limit diagrams for sheet materials under biaxial loading is necessary for the development of accurate sheet metal forming process simulations. Biaxial tension testing has been shown to have significant advantages over the current computational and experimental methods for such material characterization; however, the few commercially available loading frames are far too large and expensive to be practical for most metal forming research laboratories. To address this problem, the University of New Hampshire’s Mechanics, Materials, and Manufacturing Lab is working to design a practical servohydraulic biaxial loading frame for such metal forming laboratories. The physical system design, fabrication, and component selection was performed previously by a team of mechanical engineering seniors in collaboration with Greenerd Press and Machine Co. To continue the project, this thesis presents the design, implementation, and validation of a PLC-based control system and LabVIEW graphical interface for operating the biaxial loading frame. Experimental data shows that the displacement control system can accurately maintain equal displacement of opposing actuators to within 0.1[mm] for fixed position, 80[mm/min] ramp, and 0.2[Hz] sinusoidal profiles. The selection and mounting position of the
hydraulic control valves were found to be the major limiting factor in the abilities of the control system. Preliminary uniaxial and biaxial tension tests with Al-6022-T4 show inconsistent stress-strain responses that cause differing force measurements of up to 8[\%] between opposing load cells. The inconsistencies were attributed to the mechanical design of the current frame of the testing machine. Corresponding mechanical, hydraulic, and software/control design improvements are suggested, and plans for the future of the project are discussed.
CHAPTER I

INTRODUCTION AND BACKGROUND

Sheet metal forming processes are vital to industries such as beverage can, automotive, and aerospace. For example, the United States alone deep draws over 300 million aluminum beverage cans per day, or the equivalent of one can per American per day [1]. Sheet metal stamping also accounts for over 75% of the total components in a typical car, of which over 65 million were produced in 2013 [2], [3].

Characterization of yielding and failure in sheet materials experiencing biaxial stretching is necessary for the development of accurate simulations of sheet metal forming processes [4]. If accurate biaxial material data is available, such simulations can reduce empirical, trial-and-error approaches and thus shorten product lead times, significantly reduce tooling costs, and aid in the development of more rigorous and cost-effective designs. For example, accurate failure prediction techniques help die designers and manufacturing engineers to design tooling that brings a material to its highest possible work-hardened state (and thus its maximum strength) while avoiding the onset of plastic instability, i.e., localized necking and tearing failure. Thinner and lighter parts with a higher strength can thus be created. Considering that reducing the mass of a beverage can by 1% can save approximately $20 million annually in aluminum costs [1], the importance of accurate numerical simulations becomes clear. Of particular interest in industry are the evolving yield loci and the forming limit diagrams of the various materials used in component designs.
1.1 YIELD LOCI

Yielding in sheet materials is typically expressed graphically using yield loci, which for biaxial stress states are represented by plots of the major and minor in-plane (i.e., in the plane of the sheet metal) stresses $\sigma_1$ and $\sigma_2$ at yielding, as shown in Figure 1.1.

![Yield loci and evolution](image)

**Figure 1.1: Yield loci and evolution [5].**

The yield locus represents the boundary between elastic and plastic deformation in a material, a boundary which changes as plastic deformation progresses along a given stress path [6]. As seen in Figure 1.1, the yield locus can expand (isotropic hardening, left), translate (kinematic hardening, center), or some combination of the two (mixed hardening, right). In order to accurately and fully characterize yielding in a sheet material, it is therefore necessary to quantify not only the shape of its initial yield locus, but of subsequent yield loci as well.

Initial and subsequent yield loci can be generated analytically for a sheet material by assuming a yield criterion and hardening model and applying the principle of normality [7]. Yield loci can also be measured experimentally from thin-walled tubes under torsion.
and axial tension [8], or from thin-walled tubes with internal pressure and axial tension or torsion [9, 10, 11].

1.2 FORMING LIMIT DIAGRAMS

Failure in sheet metal stamping is traditionally predicted using strain-based forming limit diagrams (FLDs) [12, 13], i.e., plots of the major and minor in-plane principal strains at failure, as shown in Figure 1.2.

![Figure 1.2: Strain-based FLD (left) with directional strain definitions (right) [7].](image)

Failure is said to occur if the measured strain state in a sheet material exceeds the strain Forming Limit Curve (FLC) for that material. Unfortunately, characterizing the failure of individual sheet materials using strain-based FLDs is often difficult in practice due to the
dependence on the inherent deformation path(s) during manufacturing, as shown in Figure 1.3.

The strain-FLD shifts dramatically depending on the prestrain of the material, e.g., uniaxial, plane strain, or equibiaxial. Because different prestrains are associated with each stage in the manufacturing of a component, a new FLD is required not only for every material, but for every prestrain as well. Despite their difficulty to implement in practice, strain-FLDs are still widely used in industry to predict failure in sheet metal forming operations.
In 2001, Stoughton analytically converted the strain-based FLDs seen in Figure 1.3 to stress space, the results of which are shown in Figure 1.4.

![Figure 1.4: Less path dependence of stress-based FLDs [15].](image)

From Figure 1.4, it is seen that most of the strain-based FLDs from Figure 1.3 resolve to nearly a single curve when analytically converted to stress space. In addition, the levels of the stress-based FLDs are far more uniform than those of the strain-based FLDs. Both characteristics indicate that stress-based FLDs are significantly less dependent on the deformation path or the prestrain in the material, supporting the feasibility of their implementation in practice.

Analytical models for predicting FLDs for sheet materials have been developed, the most widely recognized of which is the M-K model developed by Marciniak and Kuczynski [16].
However, this and other similar methods are sensitive to the assumed input parameters, e.g., the yield criterion or hardening model. FLDs therefore are often generated experimentally using modified Marciniak (Figure 1.5) or Nakajima tests, in which metal blanks of varying geometries (Figure 1.6) are deformed to failure [17].

Figure 1.5: Marciniak test setup [18].
The Marciniak test setup shown in Figure 1.5 induces in-plane stretching in the bottom surface of the cup, allowing for a direct measurement of the major and minor strains. By varying the specimen geometry as shown in Figure 1.6, the strain and stress (i.e., deformation) path is altered and thus the FLC is defined.

### 1.3 Biaxial Testing

Biaxial tension testing involves loading a cruciform, or cross-shaped, specimen independently along its perpendicular axes as shown in Figure 1.7.
Biaxial tension testing has significant advantages over the traditional methods for both predicting and measuring yield loci and FLDs. The in-plane stretching of sheet materials represents various real-world metal forming operations more closely than the aforementioned testing of thin-walled tubes or the select geometries used in Marciniak testing. In addition, biaxial testing is simpler than other methods in the sense that the applied loading or deformation in the major and minor directions of the sheet is controlled.
directly. A single specimen geometry for the sheet can produce the various deformation paths required. Thin slots cut into the arms of the specimen improve the uniformity of the stress distribution in the gage section, potentially allowing for axial stress to be calculated simply as force/area. Finally, complicated and time-varying deformation paths can be achieved, a significant advantage over traditional Marciniak or Nakajima testing which have a single path per experiment.

A number of commercial, special order, and custom-built biaxial loading frames exist in practice, notable examples of which are shown in Figure 1.8, Figure 1.9, and Figure 1.10.

![Commercial biaxial frame with axial torsion, MTS Corporation](image)

*Figure 1.8: Commercial biaxial frame with axial torsion, MTS Corporation*
Figure 1.9: Special-order biaxial frame, National Institute of Standards and Technology
Figure 1.10: Custom biaxial frame, Tokyo University of Agriculture and Technology.

Unfortunately, the few commercially available biaxial loading frames are far too large and expensive to be practical for small metal forming research laboratories. For example, the NIST machine pictured in Figure 1.9 costs upwards of $1 million. Considering also that an ISO proposal to standardize the biaxial testing of cruciform specimens has recently been published (ISO 16842:2014), the need for an inexpensive, reliable biaxial loading frame becomes clear [19].

To address the need to generate reliable biaxial material data at a reasonable cost, this thesis presents the design and evaluation of a practical servohydraulic biaxial loading frame for use in general metal forming research laboratories. Chapter II describes the
mechanical, hydraulic, and electrical design of the biaxial loading frame. The control system and corresponding software, developed in-house, are also discussed at length. Chapter III presents experimental data to show that the position control system can accurately maintain equal displacement of opposing actuators to within $0.1 \, [mm]$ for fixed position, $80 \, [\frac{mm}{min}]$ ramp, and $0.2 \, [Hz]$ sinusoidal profiles. Preliminary uniaxial and biaxial material data for Al-6022-T4 is also presented in Chapter III, but inconsistencies in the measured stress-strain responses suggest deficiencies in the mechanical design of the loading frame. Finally, Chapter IV describes a number of suggested improvements to the mechanical and hydraulic design, as well as to the control system and software. Plans for the future of the project are also discussed.
CHAPTER II

EXPERIMENTAL SETUP

The University of New Hampshire collaborated with the National Institute of Standards and Technology to design, fund, and fabricate a tabletop biaxial loading frame for use in metal forming research labs [18]. The major components of the loading frame are shown in Figure 2.1.

The loading frame consists of a 597x597[mm] (23.5x23.5[in]) solid 1018 steel frame mounted to a HSS 177.8x177.8x12.7[mm] (7x7x0.5[in]) structural steel base. The base was designed to limit the vertical end deflection of the frame to less than 25.4[µm] (0.001[in]) with a safety factor of 1.75. Dimensioned top and section views of the biaxial loading frame are shown in Figure 2.2.

Figure 2.1: Diagram of biaxial loading frame components.
Figure 2.2: Dimensioned top view (left) and section view (right) of biaxial loading frame.

Cruciform specimens are gripped by four Qualitest THS427-5 mechanical wedge grips with a maximum specimen opening of 4\,[mm] (0.157\,[in]) thick and 20\,[mm] (0.787\,[in]) wide. Parker 3LX series hydraulic cylinders provide the necessary force for loading the cruciform specimens. Key specifications for the cylinders are provided in Table 2.1.

Table 2.1: Key specifications of Parker 3LX Series hydraulic cylinders.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Metric Value</th>
<th>English Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. Pressure</td>
<td>9.65,[MPa]</td>
<td>1400,[psi]</td>
</tr>
<tr>
<td>Bore Diameter</td>
<td>63.5,[mm]</td>
<td>2.5,[in]</td>
</tr>
<tr>
<td>Stroke</td>
<td>12.7,[mm]</td>
<td>0.5,[in]</td>
</tr>
<tr>
<td>Max. Tensile Force</td>
<td>25.8,[kN]</td>
<td>5800,[lb]</td>
</tr>
<tr>
<td>Max. Compressive Force</td>
<td>30.7,[kN]</td>
<td>6900,[lb]</td>
</tr>
</tbody>
</table>

2.1 HYDRAULIC SYSTEM

The hydraulic system for the biaxial loading frame, shown in Figure 2.3, was designed and built by Greenerd Press and Machine Co. of Nashua, NH.
A Rexroth A10VSO 11.2[kW] (15.02[hp]) axial piston pump moves hydraulic fluid from a 125[L] (33[gal]) tank to the cylinders at a rate of up to 32.2[Lpm] (8.5[gpm]). The pump can produce working pressures of up to 20.7[MPa] (3000[psi]), but a pressure limit can be set using a Rexroth DBETE-5X proportional relief valve.

Figure 2.4 shows the layout of the manifolds and control valves.
Flow to the individual cylinders is controlled by four Rexroth 4WREE6E08 proportional directional valves. On-board electronics control the spool position, i.e., -100% to 100%, in response to a ±10[V] analog command signal. That is, a +5[V] command signal will move the spool to the +50% position. Each axis, or set of opposing cylinders, operates on a separate hydraulic circuit. The “X-axis” cylinders operate on the main circuit, with a pressure limit set using the proportional relief valve shown in Figure 2.3. The “Y-axis” cylinders operate on a reduced-pressure circuit, on which the pressure limit is set to a fixed percentage of the X-axis pressure limit using an Atos RZGO-AE-010 proportional reducing valve (Figure 2.4). For example, if the proportional reducing valve is 50% open, the Y-axis pressure limit is 50% of the X-axis pressure limit.
2.2 CONTROLLER AND ELECTRONICS

Figure 2.5 shows the controller and associated electronic components.

The control circuitry for the biaxial loading frame was also designed and built by Greenerd Press and Machine Co. The controller is an Allen-Bradley 1769-L32E CompactLogix Programmable Logic Controller (PLC). Ethernet communication between a computer and
the PLC allows the user to download programs to the PLC as well as to exchange information with the PLC in real-time. An Allen-Bradley 1769-OF8V 16-bit Analog Output Module provides command voltage signals to the proportional directional, relief, and reducing valves. Two Allen-Bradley 1769*IF4I 16-bit Isolated Analog Input Modules provide control feedback from up to 8 analog sensors (typically force and displacement sensors). For data visualization and acquisition purposes, a National Instruments USB-6341 16-bit Data Acquisition Board (not pictured) can simultaneously acquire measurements from up to 16 analog sensors. A SOLA SDN-24-100P 24[\text{V}] switched-mode power supply provides power to the control valves and the PLC, while an Acopian \pm 10[\text{V}] dual output linear power supply provides power to the various sensors.

The biaxial loading frame is equipped with a number of sensors used for both data collection and control feedback purposes. Four Gefran TK series 0-20.7[\text{kPa}] (0-3000[\text{psi}]) pressure transducers measure fluid pressure after the directional valves on the rod side of each cylinder. Four Transducer Techniques SSM-8K \pm 35.6[kN] (8000[lb]) tension/compression load cells provide force readings for each cylinder. The load cell output signals are amplified to measurable levels using a simple circuit designed and built in-house with INA129P instrumentation amplifier chips. Each cylinder is also equipped with an embedded Parker 0-12.7[\text{mm}] (0-0.5[\text{in}]) linear resistive transducer (LRT) to provide position/displacement measurements. Neglecting noise on the measurements, the resolution of each sensor using the 16-bit data acquisition system is shown in Table 2.2 and is sufficient for DAQ and control purposes.
Table 2.2: Resolution of sensors for 16-bit DAQ system.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
<th>Resolution (Absolute)</th>
<th>Resolution (FS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>0-20.7[MPa]</td>
<td>0.63[kPa]</td>
<td>0.0030 [%]</td>
</tr>
<tr>
<td>Force</td>
<td>±35.6[kN]</td>
<td>±1.6[N]</td>
<td>0.0045 [%]</td>
</tr>
<tr>
<td>Position</td>
<td>0-12.7[mm]</td>
<td>1.4[μm]</td>
<td>0.011 [%]</td>
</tr>
</tbody>
</table>

It should be noted that in-plane major and minor strains can be extracted directly using any number of techniques, including strain gages, a biaxial extensometer, or digital image correlation (DIC). In this thesis, the DIC was chosen for the strain measurements presented, unless otherwise specified.

2.3 CONTROL STRUCTURE

2.3.1 VIRTUAL MASTER-SLAVE CONFIGURATION

Pairs of opposing actuators, i.e., the X- and Y- axes of the biaxial loading frame, operate on a virtual master-slave configuration as shown in Figure 2.6. That is, there is closed-loop control on each individual actuator, or slave, based on a virtual master command signal that is generated by the controller in real-time.
Each slave actuator has a separate 200[Hz] Proportional-Integral-Derivative (PID) control loop as shown in Figure 2.6. Although the capability for full PID control exists, D-control is not used because a noisy analog error signal, i.e., from the force or position sensors, tends to drive the system unstable [20]. Each slave control loop is given some time-varying command signal, typically a force or displacement ramp. The PID loops use feedback from the appropriate sensor to adjust the position of the corresponding directional valve such that the actuator effectively tracks the desired profile. The opposing actuators are effectively slaves to a virtual master command as is typical in motion control applications. The intent is to ensure that the two actuators always displace equally. Displacement matching on each axis is critical in that it prevents planar translation of the center of the specimen, which would introduce undesirable shear loads and ultimately invalidate the test [21]. Although it seems more intuitive to have one actuator track the position of the opposite actuator, the resulting motion of the actuator would be less smooth due to the relatively noisy analog signal from the position sensor. In addition, there is an inherent tracking error during the initial acceleration of the cylinders. Part of
this initial tracking error is a lag caused by expansion of the lengths of hose between the control valves and hydraulic cylinders, as pressure builds to accelerate the actuator. In addition, position is an integrating process, i.e., the actuator position increases continuously for a fixed voltage command to the control valve. The mathematics of using PI control on an integrating process to track a ramp introduces an additional tracking error at the beginning of the ramp [20]. Therefore, forcing a slave actuator to track the position of a master actuator would cause not only a lag between the virtual command signal and the master actuator, but an additional lag between the slave actuator and master actuator. Using a single, smooth virtual command for opposing actuators helps minimize any initial tracking error between the two actuators.

2.3.2 DEADBAND COMPENSATION

The Rexroth 4WREE6E08 proportional valves have a closed-center spool, meaning there is essentially no oil flow and thus no actuator motion for a range of control signals around 0%. To characterize the deadband for the actuators on the biaxial loading frame, commands of -25[%] to 25[%] in 1[%] increments were given individually to each actuator, and the resulting steady-state speed was calculated from the measured actuator position over time. A deadband of about ±10% is seen in the open-loop speed characteristics of each actuator as shown in Figure 2.7.
While the closed-center spool type reduces leakage flow across the valve near the null spool position, the nonlinearity introduced by the deadband severely limits the response and controllability of the valve and actuator. The deadband is particularly undesirable in applications where the valve must be shifted back and forth across the zero point, i.e., quickly routing oil to and from each side of the cylinder to hold a position or force, or changing the direction of actuator motion [22].

To partially counter the negative effects of the deadband on the Rexroth 4WREE6E08 directional valves, a simple deadband compensator was implemented in the custom controller that was created for the biaxial loading frame. The compensator scales the

Figure 2.7: Open-loop speed characteristics for each actuator with no deadband compensation.
output of the PID controller such that the control output immediately steps up to a fixed position in the direction that the spool needs to move, essentially forcing the valve to skip over the deadband. An example of a 20% deadband compensator is shown for a theoretical closed-center valve in Figure 2.8.

![Figure 2.8: 20% deadband compensation for a theoretical linear servovalve with a closed-center spool.](image)

Figure 2.8: 20% deadband compensation for a theoretical linear servovalve with a closed-center spool.
Note that the compensator simply scales the voltage command that is physically sent to the valve, e.g., when a 10% command is specified, actually send 20%. A 10% deadzone compensator was found to significantly improve the controllability of the real system as seen in Figure 2.9.

![Response of one X-axis actuator to a 6[mm], 0.1[Hz] sinusoidal displacement command, with and without 10[%] deadband compensation.](image)

Figure 2.9: Response of one X-axis actuator to a 6[mm], 0.1[Hz] sinusoidal displacement command, with and without 10[%] deadband compensation.

However, the response and controllability of the system are still limited by the nonlinearity of the valves near the spool null position, as well as the ~25[ms] required for the spool to physically pass through the entire deadband.
2.4 PROGRAMMING ARCHITECTURE

The control system for the biaxial loading frame was developed in-house, using RSLogix5000, a proprietary software for programming Allen-Bradley PLCs. The programs were written in a state machine architecture as shown in Figure 2.10.

![State machine architecture for biaxial loading frame control programs.](image)

Figure 2.10: State machine architecture for biaxial loading frame control programs.

State machine architecture consists of various sections of code, each of which executes only while the controller is in the appropriate state. Events, such as a button press or a warning signal, can trigger transitions from one state to another. Referring to Figure 2.10, for example, pressing the jog button while in the Idle/Hold state triggers a transition to the Jog state and thus starts the open-loop motion of the actuator. Similarly, releasing the jog button while in the Jog state indicates that motion is done and thus triggers a transition
back to the Idle/Hold state, where the actuator is held in a fixed position under closed-loop control.

Ladder logic (LAD) and function block diagrams (FBD) were used in RSLogix 5000 to program the PLC, which is advantageous due to the simplicity and speed of execution by the PLC relative to that of a PC running a resource-heavy operating system. Time-sensitive tasks such as the PID control loops and the tasks within the individual machine states were therefore programmed onto the PLC using RSLogix5000. All relevant programs, which are too lengthy to include here, are filed electronically in the Mechanics, Materials, and Manufacturing lab at the University of New Hampshire.

2.5 HUMAN-MACHINE INTERFACE

National Instruments LabVIEW software was used to program the human-machine interface (HMI) for the biaxial loading frame. LabVIEW allows for the creation of graphical user interfaces, which from an operator's standpoint are far more efficient and user-friendly than the ladder logic code used to program the PLC. Unlike ladder logic code, LabVIEW code runs on a computer and generally does not execute quickly enough to manage overly demanding or time-sensitive tasks. However, LabVIEW is well-suited to handling less demanding tasks such as event processing and data acquisition/visualization, making it a good choice for programming the user interface for the biaxial loading frame. National Instruments Industrial Communications drivers were used to communicate between LabVIEW and the PLC over Ethernet, allowing the operator to easily exchange real-time data with the PLC from the user-friendly LabVIEW front panels.
2.5.1 DATA COLLECTION AND VISUALIZATION PANEL

The LabVIEW panel created to record and display real-time test data is shown in Figure 2.11.

![LabVIEW front panel for data collection and visualization.](image)

Referring to Figure 2.11, the three leftmost graphs show the individual load, displacement, and rod-side cylinder pressure measurements for each actuator. The top-right graph shows the load-displacement curves for each axis of the machine. For each axis, the load is the average of the individual actuator loads, while the displacement is the sum of the individual actuator displacements. Numeric indicators in the ‘Sensor Readings’ area show the individual load, displacement, and rod-side cylinder pressure measurements for each
actuator. ‘Zero Disp.’ and ‘Zero Load’ buttons allow the operator to zero-off the displacement or load readings.

### 2.5.2 MACHINE CONTROL PANEL

The cylinders can be jogged manually using the Jog Controls tab as shown in Figure 2.12.

![Figure 2.12: LabVIEW front panel for actuator jogging interface.](image)

Controls on the ‘Jog Controls’ tab allow the operator to manually position the cylinders when setting up a test and installing a specimen. Because the mechanical clamping action of the wedge grips tends to compress the specimen, a zero-load control feature is available to maintain no load on the specimen during clamping. The ‘Sync Pos.’ button moves all actuators to the absolute position specified in ‘Sync To,’ which is useful for setting up material tests. The ‘Cycle’ button is used to cycle the actuators over the full stroke at a frequency of $0.2\,[Hz]$, a procedure that is typically used at startup to warm up the hydraulic system.

Force or displacement ramp tests are controlled with the Ramp Controls tab as shown in Figure 2.13.
Parameters for the ramp test can be set individually for each axis, including the ramp speed, length, units, and direction. For systems with complicated or unpredictable mathematical models, i.e., hydraulic systems, manual tuning of control systems is often more convenient than trying to predict effective controller parameters analytically [20]. With this in mind, the ‘Tuning Controls’ interface shown in Figure 2.14 was created for real-time manual tuning of the PID control loops for each of the hydraulic actuators.
Figure 2.14: LabVIEW front panel for PID tuning interface.
Using the 'Tuning Controls' tab, the operator can select an actuator to tune as well as the type of control (force or displacement). The Jog Controls tab allows the operator to manually position the cylinder to begin tuning. The Tuning Controls tab then allows the user to configure the command signal, i.e., signal type, frequency, and amplitude. The proportional (P), integral (I), and derivative (D) controller gains can then be adjusted in real time until the feedback signal sufficiently matches the command signal as seen on the virtual oscilloscope screen. Note that the system dynamics change for extending and retracting due to the differing areas on each side of the piston as well as the differing lengths of hose between the directional valves and the cylinders. A technique known as ratioed gains is therefore used to scale the controller gains by approximately the ratio of the areas on either side of the piston, as the actuator would for a given command extend faster than it would retract. The user specifies the P, I, and D gains for extending the cylinder, and specifies the Gain Ratio parameter to scale the gains appropriately for retracting the cylinder. Finally, the Mode indicator shows the current state of the program, and contains a large shutdown button to close all the control valves and shut down the system programmatically.

2.5.2.1 DISPLACEMENT CONTROL TUNING
Displacement control tuning was performed manually using a sine wave command signal with amplitude of $0.5\, [mm]$ and a frequency of $1\, [Hz]$. The frequency of the sine wave was chosen to match the frequencies expected for pulsed loading tests (see Section 4.2.3). The amplitude was selected such that the corresponding fluid volume demand was near the fluid displacement limits of the pump. The control loops were tuned with the actuators
moving freely, as rod displacement is most sensitive to the control signal when the actuator is not loaded. The natural frequency and damping ratios were observed in the oscillating actuator response to a triangle wave input as shown in Figure 2.15.

![Figure 2.15: Response of X1 actuator to a triangular input with slope 300[mm/s] and amplitude.](image)

The damped natural frequency $\omega_d$ was calculated from the period $T$ of the oscillations as
The damping ratio $\zeta$ was estimated from the decay of the oscillations between the first peak and the $n^{th}$ peak using the log-decrement method [20],

$$\delta = \frac{1}{n}\ln\left(\frac{x_1}{x_n}\right)$$  \hspace{1cm} (2.2)

$$\zeta = \frac{1}{\sqrt{1 + \left(\frac{2\pi}{\delta}\right)^2}}$$  \hspace{1cm} (2.3)

Note that $x_n$ represents the amplitude of the $n^{th}$ peak. The natural frequency was then calculated as

$$\omega_n = \frac{\omega_d}{\sqrt{1 - \zeta^2}}.$$  \hspace{1cm} (2.4)

Key properties for the resulting tuned position control system are shown in Table 2.3.

<table>
<thead>
<tr>
<th>Actuator</th>
<th>P Gain [%/mm]</th>
<th>I Gain [%/mm-s]</th>
<th>Gain Ratio</th>
<th>Natural Freq. [Hz]</th>
<th>Damping Ratio [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>X1</td>
<td>2.3</td>
<td>.23</td>
<td>0.9</td>
<td>5.43</td>
<td>0.20</td>
</tr>
<tr>
<td>X2</td>
<td>2.2</td>
<td>.22</td>
<td>0.8</td>
<td>5.92</td>
<td>0.21</td>
</tr>
<tr>
<td>Y1</td>
<td>2.5</td>
<td>.25</td>
<td>0.8</td>
<td>5.36</td>
<td>0.29</td>
</tr>
<tr>
<td>Y2</td>
<td>2.6</td>
<td>.26</td>
<td>0.7</td>
<td>6.69</td>
<td>0.27</td>
</tr>
</tbody>
</table>

Note that the % in the P and I gain units refers to the setpoint of the proportional valve, i.e., -100% to 100%. The low natural frequencies are likely due to the low hydraulic
stiffness of the system, discussed further in Section 4.1.2. The damping ratios of 0.2-0.3 are also very low, which increases overshoot and is thus detrimental to the transient (step) response. However, recall that a ramp input is used for standard position-controlled tension tests. The steady-state error of the ramp response of a second order system is inversely proportional to the damping ratio, so a low damping ratio is desirable under normal testing conditions [20]. Derivative control can offer a compromise between transient and ramp responses, but can also drive the system unstable if the feedback signal is too noisy. Because of this, and because the system should never see a step position input under normal operating conditions, the low damping ratio is acceptable.

### 2.5.2.2 FORCE CONTROL TUNING

Because actuator force is a far more dynamic parameter than rod displacement, force control tuning was performed using a square wave command signal with amplitude of $10[kN]$ and a frequency of $1[Hz]$. The amplitude of the square wave was chosen to match the axial load limit of the wedge grips, while the frequency was again chosen to match the frequencies needed for pulsed loading tests (see Section 4.2.3). Because any compressive forces would buckle the specimen when under zero-load control, controller gains were selected to produce a critically damped system, i.e., a damping ratio of 1, which has zero overshoot. To provide a means to build pressure (and thus force) in the cylinders, a steel rod with a $20[mm](0.787[in])$ square cross-section was clamped into the machine. The rod was sized to prevent yielding or buckling under the $10[kN]$ load. The gains for critical damping are shown for each actuator in Table 2.4.
Table 2.4: Force control system properties for each actuator.

<table>
<thead>
<tr>
<th>Actuator</th>
<th>P Gain [%/kN]</th>
<th>I Gain [%/kN-s]</th>
<th>Gain Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>X1</td>
<td>0.76</td>
<td>0.05</td>
<td>0.9</td>
</tr>
<tr>
<td>X2</td>
<td>0.74</td>
<td>0.04</td>
<td>0.8</td>
</tr>
<tr>
<td>Y1</td>
<td>0.76</td>
<td>0.05</td>
<td>0.8</td>
</tr>
<tr>
<td>Y2</td>
<td>0.79</td>
<td>0.06</td>
<td>0.7</td>
</tr>
</tbody>
</table>
CHAPTER III

VERIFICATION TEST RESULTS AND DISCUSSION

3.1 CONTROL SYSTEM VERIFICATION

Several sets of tests were conducted to verify the effectiveness of the control system with respect to tracking various prescribed axial displacement profiles under no-load conditions. Fixed position, triangle wave, and sine wave profiles were investigated for both independent and simultaneous axes. In accordance with ISO 16842:2014, the biaxial, cruciform specimen testing standard, the control system was required to maintain a displacement tracking error of less than $0.1\text{[mm]}$ between opposing cylinders at all times to ensure negligible shifting of the center of a test specimen.

3.1.1 FIXED POSITION PROFILES

Fixed absolute position setpoints of 2, 5, 8, and $11\text{[mm]}$ were prescribed to each actuator. The position of each actuator was recorded for $30\text{[s]}$ to investigate the ability of the control system to hold a fixed position with no load applied. Figure 3.1 and Figure 3.2 show the distribution of the position tracking error for the X- and Y-axes, respectively, during all fixed position tests.
Figure 3.1: Histogram (50 bins) showing the distribution of tracking error between the X-axis cylinders and fixed absolute position setpoints of 2, 5, 8, and 11[mm].
Figure 3.2: Histogram (50 bins) showing the distribution of tracking error between Y-axis cylinders and fixed absolute position setpoints of 2, 5, 8, and 11[mm].

As seen in Figure 3.1, the control system is able to maintain the position of the X-axis actuators to within 1.1[μm] of the setpoint, on average, with a standard deviation of 4.1[μm]. In addition, the position of either cylinder never deviates more than 15.3[μm] from the setpoint. Similarly, Figure 3.2 shows that the control system is able to maintain the position of the Y-axis actuators to within 0.9[μm] of the setpoint, on average, with a standard deviation of 4.6[μm]. The position of either Y-axis cylinder never deviates more than 16.8[μm] from the setpoint. It should be noted that the maximum deviation of any actuator from the fixed position setpoint is on the same order of magnitude as the
resolution of the position sensor, or $1.4[\mu m]$. Based on the maximum allowable displacement error of $0.1[mm]$ taken from ISO 16842:2014, which is the critical design requirement for the system, the performance of the control system is sufficient for maintaining a fixed actuator position.

### 3.1.2 TRIANGULAR PROFILES

Triangular position profiles were prescribed to each axis to simulate tensile and compressive ramp tests at speeds of $1-100[mm/min]$ in $10[mm/min]$ increments. The tracking error between opposing cylinders was monitored in an effort to determine the maximum ramp speed achievable by the system while maintaining a displacement tracking error of less than $0.1[mm]$ between opposing cylinders at all times, again in accordance with ISO 16842:2014 for cruciform specimen testing. Figure 3.3 shows for each axis the maximum observed displacement tracking error between opposing actuators, with respect to ramp speed.
As seen in Figure 3.3, a maximum ramp speed of \( \sim 60 \frac{\text{mm}}{\text{min}} \) is achievable without exceeding a displacement tracking error of \( 0.1 \frac{\text{mm}}{\text{min}} \) on either axis when the actuators are not under load. It is important to note, however, that the maximum displacement tracking error tends to occur during the initial acceleration of the actuators, and levels off to a lesser value for the remainder of the tests. This is likely a result of the slightly differing dynamic responses of each actuator, e.g., due to differing lengths of hose between valve and cylinder. Note also that the sharp discontinuity at the peaks of the triangle wave can drive the system unstable at higher speeds.
3.1.3 **SINUSOIDAL PROFILES**

Sinusoidal position profiles with amplitude 0.5[mm] and frequency 0-1[Hz] were prescribed to each axis to simulate pulsed loading tests. The tracking error between opposing cylinders was monitored in an effort to determine the maximum frequency achievable by the system while maintaining a displacement tracking error of less than 0.1[mm] between opposing cylinders at all times, again in accordance with the proposed ISO standard for biaxial, cruciform specimen testing. Figure 3.4 shows for each axis the maximum observed tracking error with respect to sinusoidal frequency.

![Figure 3.4: Maximum displacement tracking error between opposing cylinders for sinusoidal position profiles of amplitude 0.5[mm] and frequency 0-1[Hz].](image)

Figure 3.4: Maximum displacement tracking error between opposing cylinders for sinusoidal position profiles of amplitude 0.5[mm] and frequency 0-1[Hz].
As seen in Figure 3.4, a maximum frequency of ~0.2 [Hz] is achievable without exceeding a displacement tracking error of 0.1 \( \text{mm/min} \) on either axis when a specimen is not being loaded in a test. It is important to note that the maximum displacement tracking error tends to occur when the actuators change directions, i.e., at the peak of the sine wave, such as seen for a 0.5 [Hz] sine wave on the X-axis actuators in Figure 3.5.

![Graph of X-axis tracking a sine wave with amplitude 0.5[mm] and frequency 0.5[Hz].](image)

Figure 3.5: X-axis tracking a sine wave with amplitude 0.5[mm] and frequency 0.5[Hz].

A noticeable 0.5[s] delay is observed when changing the direction of the actuators, followed by an abrupt jump back to the target path of travel. The delay when changing actuator directions is likely caused by the use of closed-center proportional valves and
the lengths of flexible hose between the valves and cylinders as explained in Section 4.1.2.

### 3.2 UNIAXIAL MATERIAL TESTS

To investigate the ability of the biaxial loading frame to produce accurate material data, rectangular strips of Al-6022-T4 with \( \sim 19 \times 1 \text{[mm]} \) cross-sections were pulled in uniaxial tension at a prescribed \( 10^{-3} \text{[s]} \) strain rate on both axes of the biaxial loading frame. Figure 3.6 compares the resulting stress-strain data to that from a similar test using an MTS Landmark 370 servohydraulic loading frame. The strain for the MTS experiment was acquired with a mechanical extensometer (model MTS 634.12E-24), while for the experiments on the cruciform machine was acquired from 2D Digital Image Correlation (VIC-2D from Correlated Solutions, Inc.). In every case, the stress was determined by dividing the load cell reading with the initial cross-sectional area.
Figure 3.6: Average engineering stress vs. Lagrangian strain data from Al-6022-T4 uniaxial tests at $10^{-3} [1/s]$ strain rate on both axes of the biaxial loading frame and on an MTS servohydraulic loading frame.

It is seen that the stress-strain data acquired from the X-and Y-axes of the biaxial loading frame differ from the MTS stress-strain data by about 3.5[%] and 6[%], respectively, in the plastic range. For the same tests, Figure 3.7 shows further discrepancy between the individual force/stress measurements on opposing actuators.
Figure 3.7: Individual average stress vs. Lagrangian strain data from Al-6022-T4 uniaxial tests at $10^{-3}\,[1/s]$ strain rate, on both axes of the biaxial loading frame and on an MTS servohydraulic loading frame.

Note that some of the curves do not start exactly at (0, 0), as the deadband of the hydraulic valves makes it difficult to hold a steady actuator position while a specimen is clamped. The individual stress, and thus force, measurements on opposing actuators differ by as much 7.4[%] and 4.4[%] in the plastic range on the X- and Y-axes, respectively. However, repeating the test multiple times on the X-axis confirmed that the force (and thus stress) measurements were reasonably consistent from test-to-test, as seen in Figure 3.8.
Regardless, the test results suggest that reliable uniaxial material data cannot be obtained on individual axes of the biaxial machine. Given the fact that the no-load, displacement tracking experiments reported in Section 3.1 indicated that the tracking error at the slow rates tested is rather low, this discrepancy can be attributed to the mechanical design of the frame and the load train, rather than the hydraulic or control systems.

### 3.3 BIAXIAL VERIFICATION TESTS

To investigate the behavior of the loading frame during biaxial testing, Al-6022-T4 cruciform specimens with $\sim 30 \times 30 \times 1 [\text{mm}]$ gage sections were pulled in equibiaxial tension
at a speed of $1.5 \frac{mm}{min}$ per axis ($0.75 \frac{mm}{min}$ per actuator) to approximate a $10^{-3} \frac{1}{s}$ strain rate. The experiments were performed under displacement-control with the two axes receiving the same command, the assumption being that for an isotropic material, equibiaxial strain will result in equibiaxial tension. The specimen geometry was chosen according to the ISO standard (also described in [23]), which specifies the use of 11 slots in the arms to make the stress distribution in the test-section more uniform [19]. The specimen was prepared by laser cutting. Care was taken to ensure that the slots were cut from the outside of the arms inward to the test-section, keeping away from the test section the enlarged hole initially created during the focusing of the beam. The sheet rolling direction (RD) was aligned with the X-axis of the loading frame, while the transverse direction (TD) was aligned with the Y-axis of the loading frame.

The displacement history applied for one test is shown in Figure 3.9.
Figure 3.9: Actuator response for an Al-6022-T4 equibiaxial cruciform test at 1.5[mm/min] per axis (0.75[mm/min] per actuator), or a prescribed $10^{-3}$[1/s] strain rate.

Note that the actuators maintained equal displacements to within 0.89[mm], which is sufficient for biaxial testing [19]. The corresponding individual load-displacement data is shown in Figure 3.10.
Figure 3.10. Individual load vs. displacement data from an Al-6022-T4 equibiaxial cruciform test at $10^{-3}[1/s]$ strain rate.

Much like in the uniaxial tests, the individual force measurements on opposing actuators differ by as much as 7.0[%] and 5.0[%] in the plastic range on the X- and Y-axes, respectively. In addition, the average loads on the X- and Y-axes differ by as much as 16.0[%] in the plastic range. Note that the increased difference between the two axes is likely due in part to the differing material properties in the rolling and transverse directions of the Al-6022-T4 sheet material. The measured properties of Al-6022-T4 can be found in Appendix A.
The measured forces shown in Figure 3.10 were used to determine the average engineering stresses in the two directions. The resulting stress path is shown in Figure 3.11.

![Stress path diagram](image)

Figure 3.11: Average engineering stress path from an Al-6022-T4 equibiaxial cruciform test at $10^{-3}[1/s]$ strain rate.

It is seen that despite the four actuators maintaining equal displacement to within $0.06[mm]$ throughout the test, the measured stress path (blue) deviates from the expected 1:1 path (red) by as much as $66[\%]$.

Despite the questionable force measurements, the in-plane Lagrangian strain maps from the DIC software, shown immediately prior to failure in Figure 3.12, Figure 3.13, and
Figure 3.14, indicate that the strain distribution through the gage section of the specimen is relatively uniform.
Figure 3.12. X-axis (horizontal) strain distribution immediately before failure in an Al-6022-T4 cruciform specimen under equibiaxial tension at $10^{-3} [1/s]$ strain rate.
Figure 3.13: Y-axis (vertical) strain distribution immediately before failure in an Al-6022-T4 cruciform specimen under equibiaxial tension at $10^{-3}$[1/s] strain rate.

Note that the Lagrangian strain $e_L$ is defined as

$$e_L = \frac{1}{2}(\lambda^2 - 1),$$

(3.1)
where $\lambda$ is the ratio of the final length to the initial length. The shear strains shown in Figure 3.14 reveal that very little shear strain develops in the specimen. In every case, the shear strain is less than 1/10 of either of the axial strain components.

Figure 3.14: Shear strain distribution in an Al-6022-T4 cruciform specimen under equibiaxial tension at $10^{-3}[1/s]$ strain rate.
Figure 3.15 shows the measured Lagrangian strain paths at the center of the test section, and at the centers of the four quadrants of the test section.

![Figure 3.15: Measured strain path for an Al-6022-T4 cruciform specimen under equibiaxial tension at 10^{-3}[1/s] strain rate. QI, etc. refer to the strain paths at the centers of the four quadrants of the test section.](image)

Despite the actuators accurately following the prescribed 1.5\([\frac{mm}{min}]\) equibiaxial displacement profile, the measured strain path indicates that the Y-axis strains are consistently higher than the X-axis strains throughout the test. This observation corresponds with the higher stresses observed on the Y-axis as was seen in the stress path in Figure 3.11; the strain path is not 1:1, and the stress path is also not 1:1. The
deviation from the expected strain path is likely caused by differing slip in the grips on the
two axes of the machine; even though the actuators displace by the same amount, the
specimen wouldn’t deform the same amount in both directions. In addition, the data points
appear in clusters, suggesting that the strain increases in more of a stairstep fashion than
in a straight line. This phenomenon is likely related to the limitations of the hydraulic
system; extremely tight control is required to accurately move the actuators under load at
such low speeds.
CHAPTER IV

RECOMMENDED IMPROVEMENTS AND FUTURE WORK

4.1 RECOMMENDED IMPROVEMENTS

Based on the limitations of the equipment suggested by the results of the control system validation and materials tests, a number of upgrades to the system should be implemented. The following outlines the mechanical, hydraulic, and electrical/software changes that would significantly improve the performance of the biaxial loading frame.

4.1.1 MECHANICAL

The calibration of the load cells was confirmed using the MTS Landmark 370 servohydraulic loading frame, suggesting that the discrepancy between individual load cell readings for the same axis are likely due to mechanical deficiencies in the frame itself. Because the load cells are sensitive to bending, any misalignment of the actuators, load train, or specimen can impose non-axial forces on the load cells and result in erroneous force readings. For this reason, a new frame has been designed and is currently being manufactured. The improved design is shown in Figure 4.1.
Figure 4.1: New biaxial loading frame design.

The new loading frame is comprised of both bolted and welded plates, and bolts onto two large welded I-beams (dark gray) for added stiffness. The most significant improvement to the design is the addition of a pantograph system (yellow) to mechanically couple opposing cylinders. The system forces opposing cylinders to move with equal displacement, minimizing undesired shifting of the center of the specimen. The pantograph system is also mounted to guide blocks (gray) that ride on linear tracks (orange), ensuring excellent and continuous alignment of the hydraulic actuators. A new
grip design (purple) allows specimens to be easily installed from above, and an alignment block (green) with precision pins ensures accurate, repeatable alignment of the specimen. The improvements to the frame design are expected to correct the errors observed in the load cell readings during material tests, and improve the ability of the control system to maintain equal displacement between opposing cylinders. The full system is shown during assembly in Figure 4.2.

Figure 4.2: Assembly of the new biaxial loading frame.

4.1.2 HYDRAULIC

Jelali and Kroll note that the use of a high-pressure, metal-reinforced rubber hose between a valve and cylinder can decrease the effective stiffness of the hydraulic system by nearly a factor of 3 when compared to a steel pipe of the same length, as shown in Table 4.1 [24].
It is therefore likely that significant lengths (~8 ft) of flexible hose between the proportional valves and hydraulic cylinders of the biaxial loading frame limit the ability of the system to react quickly to abrupt changes in the position command signal, e.g., the peak of a triangular or sinusoidal wave as observed in Section 3.1. It is therefore recommended that the proportional valves be moved as close to the hydraulic cylinders as possible, and connected to the hydraulic cylinders using the minimum possible length of steel piping. Ideally, and if funding permits, custom manifold blocks should be purchased to allow the proportional valves to mount directly to the hydraulic cylinders.

As discussed in Section 2.3.2, the nonlinearity present in the closed-center Rexroth proportional valves severely limits the response and controllability of the valves, especially near the null spool position. Closed-center valves are particularly problematic in the case of force control, where very small changes in the command signal produce quick, large changes in the output, and in displacement controlled tests where the actuator changes direction, such as pulsed tension tests. In addition, the flow response of the Rexroth proportional valves is only linear for ~20-70% control signals. Given that a 20% control signal would produce $16 \frac{mm}{s}$ of displacement per axis, or a prescribed

Table 4.1. Comparison of effective hydraulic stiffness for rubber hose vs steel pipe [24].

<table>
<thead>
<tr>
<th>Pressure [$MPa$]</th>
<th>Eff. Stiffness w/ Rubber Hose [$MPa$]</th>
<th>Eff. Stiffness w/ Steel Pipe [$MPa$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>500</td>
<td>1460</td>
</tr>
<tr>
<td>9</td>
<td>537</td>
<td>1510</td>
</tr>
<tr>
<td>13</td>
<td>568</td>
<td>1570</td>
</tr>
</tbody>
</table>
strain rate of over $50 \frac{1}{s}$, the linear range of the valves lies outside the typical operating range of the biaxial loading frame. Because the PID control algorithm expects the output of the system to be linear, the control parameters are not necessarily optimal over the full operating range of the system, and tight, effective control cannot be guaranteed in every testing situation. A nonlinear control algorithm could be used to improve the response of the system; however, the response would still be limited by the deadband of the valves. For this reason, it is recommended that the Rexroth proportional valves simply be replaced with proportional or servo valves having a linear flow response and axis-cut (i.e., no deadband) spools, such that a simple, linear PID controller would truly be appropriate.

4.1.3 CONTROLS AND SOFTWARE

As observed in Section 3.1, the largest displacement tracking errors between opposing cylinders tend to occur when the system is responding to a non-monotonic or non-differentiable position command, e.g., at the peak of a sine wave, or the initial acceleration during a ramp test. In addition to the hydraulic design flaws discussed in Section 4.1.2, it is important to consider that a simple PID feedback controller will never act instantly based on the input command; there must be some error present before the controller will react and change the output. Feedforward control, seen in Figure 4.3, is a technique that introduces a predictive element into the controller based on the desired output, such that the system can react instantly to a command signal.
Feedforward control calculations expected control output directly from the command signal, a calculation that is based on a measured or modeled open-loop response of the system [25]. For example, if a hypothetical hydraulic actuator were known to travel at $10 \frac{mm}{min}$ for a 10% command signal and $30 \frac{mm}{min}$ for a 30% command signal, it could be inferred that the actuator speed is directly proportional to the command signal by a factor of 1. The controller could use this information to, for example, determine that 20% is the appropriate control output for a desired speed of $20 \frac{mm}{min}$. The controller could then immediately output a control signal of 20% whenever the desired speed is $20 \frac{mm}{min}$, without having to wait for some error to appear. A similar technique could be used to improve disturbance rejection: a scaling factor or additional gain could be calculated based on the measured force on the actuator, and used to boost the output of the controller as the load or loading rate increases. A feedback controller, e.g., a PID controller, could then be used to compensate for any slight errors caused by inaccuracies in the model, by lag in the
system response, or by unmodeled disturbances to the system; the feedforward controller would be doing a large majority of the work. The combination of feedforward and feedback control is therefore highly recommended, as such a control strategy could greatly improve the response of the system to changing or non-continuous inputs and increase the testing capabilities of the biaxial loading frame.

The use of a PLC for programming the control loops is another limiting factor in the response of the system. PLCs have scan times on the order of milliseconds, and thus expect control loops times on the order of seconds or minutes. Dedicated motion controllers have hardware designed for high speed measurements and calculations with scan times on the order of microseconds, which allows for much tighter control loop times on the order of milliseconds or even microseconds. This is especially critical for hydraulic systems, where very small changes in the control output can yield large, very quick changes in fluid pressure and thus actuator force/speed [24]. In addition, hydraulic motion controllers have a wide array of features that are particularly useful for controlling hydraulic actuators, including feedforward control, valve linearization, and ratioed controller gains. Such features are time consuming to program and difficult to implement correctly in custom control software, yet are readily available in a number of off-the-shelf hydraulic controllers. It is therefore recommended that the PLC be replaced with a dedicated motion controller, specifically designed for hydraulic systems, to increase the ease of implementation, the range of useful features, and the overall performance of the control loops for the biaxial loading frame.
4.2 FUTURE TESTING AND CONTROL STRATEGIES

With the proposed upgrades outlined in Section 4.1, the biaxial loading frame is theoretically capable of running a number of material tests depending on the application of interest, e.g., identification of yield surface, probing of failure limits, etc. The following is an outline of future tests that could be performed using the biaxial loading frame, as well as the associated control strategies.

4.2.1 CONSTANT STRESS RATIO

Industries such as the automotive industry are interested in the ability to generate yield loci of new-age sheet materials used in their latest designs. Points on the yield locus, which is a plot of the major and minor stresses in a sheet material, are generated by loading a material along a constant stress path and recording the stress state at yielding. The process is repeated using different stress paths to generate the full yield locus. Also of interest is the ability to generate stress-based FLDs, or plots of the major and minor stresses in a sheet material at failure. Stress-FLDs are generated by loading a material along a constant stress path and recording the stress state at failure. Again, the process is repeated using different stress paths to generate the full FLD. The ability to perform constant stress ratio tests using the biaxial loading frame is therefore necessary for generating yield loci and stress-FLDs.

Because the axial stresses cannot be measured directly, they must be estimated and controlled using available measurements of related parameters in an approach known as inferential control or “soft-sensing” [25]. The engineering stress ratio can be defined
based on measurable properties of the loaded cruciform gage section as shown in Figure 4.4.

Assuming that the specimen is designed for a uniform stress distribution through the gauge section and that no shear loads are present in the gauge section, the engineering stress $s$ can be approximated for each axis, i.e., $s_x$ or $s_y$, as

$$s = \frac{F}{l_0 t_0},$$

where $F$ is the applied force in the axial direction, i.e., $F_x$ or $F_y$, $l_0$ is the length of the gage section, and $t_0$ is the thickness of the gage section. The engineering stress ratio $s_x/s_y$ is thus

$$\frac{s_x}{s_y} = \frac{F_x}{F_y}.$$
From Equation (4.2), it is seen that the engineering stress ratio $s_x/s_y$ can be directly controlled by controlling the load ratio $F_x/F_y$. The control strategy is to run the X-axis under displacement control following a displacement ramp, and run the Y-axis under load control to follow a certain percentage of the measured load on the X-axis. A constant load ratio, and thus a constant engineering stress ratio, is therefore maintained on the specimen.

The yield locus (especially a subsequent yield locus) of a material can also be generated using the true stress $\sigma$, written for each axis, i.e. $\sigma_x$ or $\sigma_y$, as

$$\sigma = s(1 + e) = \frac{F}{l_0 t_0}(1 + e),$$

(4.3)

where $e$ is the engineering strain in the axial direction, i.e., $e_x$ or $e_y$. The true stress ratio $\alpha$ is thus

$$\alpha = \frac{\sigma_x}{\sigma_y} = \frac{F_x(1 + e_x)}{F_y(1 + e_y)},$$

(4.4)

From Equation (4.4), it is seen that the true stress ratio $\alpha$ can be directly controlled by controlling the load ratio $F_x/F_y$. The control strategy is to run the X-axis under displacement control following a displacement ramp, and run the Y-axis under load control to maintain the necessary force $F_y$ to keep $\alpha$ constant at the measured strain state $(e_x, e_y)$, i.e.,

$$F_y = \frac{F_x(1 + e_x)}{\alpha(1 + e_y)}.$$  

(4.5)
A constant true stress ratio $\alpha$ can therefore be maintained on the specimen if in-situ strain measurements (e.g., from high-elongation strain gages) are available.

There is also potential to measure stresses in-situ using a modified x-ray diffraction technique developed at the National Institute of Standards and Technology [26]. Although the technique is not fast enough to be implemented in process control, it can serve to validate the stress paths and corresponding yield loci calculated based on the applied load and the initial cross section of the cruciform specimens as described in Equation (4.1).

**4.2.2 CONSTANT STRAIN RATIO**

The ability to perform constant strain ratio tests using the biaxial loading frame is necessary for generating strain-FLDs. The strain ratio $\rho$ is calculated as the ratio of the axial true strains $\varepsilon_x$ and $\varepsilon_y$:

\[
\rho = \frac{\varepsilon_x}{\varepsilon_y} = \frac{\ln(1 + \varepsilon_x)}{\ln(1 + \varepsilon_y)}. \quad (4.6)
\]

The control strategy is to run the X-axis under displacement control following a displacement ramp, and run the Y-axis under strain control (using an external sensor for the strain, e.g., strain-gage, biaxial extensometer, optical method/DIC, etc.) to maintain the necessary strain $\varepsilon_y$ to keep $\rho$ constant for a given $\varepsilon_x$. A constant strain ratio $\rho$ can therefore be maintained on the specimen if in-situ strain measurements are available.
4.2.3 PULSE AND LOAD-UNLOAD TESTS

Pulsed loading tests are of interest in tube hydroforming applications, as past experiments of this type suggest an increase in the ductility of the loaded material [27]. Pulsed loading is also very relevant for materials that have a propensity for deformation-induced heating, as well as for stamping with servohydraulic presses. In a pulsed loading test, a material is brought to a certain stress level, at which point the load is cycled about a mean, until failure of the material occurs. An example of a pulsed test on a pressurized tube is shown in Figure 4.5.

![Figure 4.5: Pulse test on a pressurized tube [27].](image)

The mean value about which pulsing occurs is generally constant or linearly increasing as shown in Figure 4.5. In biaxial stretch tests, the control strategy is to place one axis under displacement control following a displacement ramp. When a certain strain or stress level is reached, the axis is switched to force or stress control to follow a command.
that is the sum of a sine wave (pulse) signal and a ramp (mean) signal. For a constant mean value, the slope of the ramp signal is simply set to zero. The second axis can be operated independently in the same fashion, or at a constant stress or strain ratio, as outlined previously.

Load-unload tests are of interest in pulsed tube hydroforming applications, as past experiments of this type suggest an increase in the ductility of the loaded material. An example of a uniaxial load-unload test on DP590 dual-phase steel is shown in Figure 4.6.

![Figure 4.6: Load-unload uniaxial test on DP590 dual-phase steel [28].](image)

In a load-unload test, the material is loaded to a predetermined increment of stress or strain, unloaded, then reloaded to the next increment of stress or strain. The process is repeated until failure of the material occurs. A loading phase can be performed under any type of control, i.e., force, strain, or displacement. In addition, a constant stress or strain ratio can be maintained during loading using the control strategies outlined previously.
The unload phase would then be performed under displacement or strain control until some prescribed force threshold is reached, at which point the next loading phase would begin. The loading path can be fully characterized if the loading profile, unloading stress or strain increment, unloading speed, and unloading force threshold are specified by the user. Note that the unloading speed for the axes would be identical only in a balanced biaxial test.

4.3 BIAXIAL STRESS MEASUREMENT USING X-RAY DIFFRACTION

The National Institute of Standards and Technology has developed and implemented an x-ray stress measurement system to measure residual stresses in sheet materials [26].

Figure 4.7: Modified x-ray diffraction system for in-situ stress measurements in sheet materials [26].
The system uses a modified x-ray diffraction technique to measure the change in interatomic spacing of the material's lattice structure in-situ relative to that of the unstressed material (typically a powder); constitutive laws are then used to calculate the residual principal stresses. The stress measurement occurs at a point, so multiple measurements can provide a full map of the in-plane stress distribution in the sheet material. The open-top design of the biaxial loading frame allows the device to be used in conjunction with NIST’s (or a similar) x-ray diffraction system to measure the stresses and ultimately the yield loci of sheet materials under biaxial loads.
REFERENCES


A. ANISOTROPY DATA FOR AL-6022-T4

Figure A.1: Rolling direction stress-strain data for Al-6022-T4.
Figure A.2: Transverse direction stress-strain data for Al-6022-T4.