Project Title: Redesign of a Rail Bending Test Machine

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We would also like to thank Steel Dynamics Inc. for making this project possible. We are especially grateful for the time and energy which Andrew Weber and Drew Seely of Steel Dynamics dedicated to helping us with the project and providing us with information on standards and specifications.
In this report, work for the Senior Design II project titled “Redesign of a Rail Bending Test Machine” will be reported; the sponsor company for this project is Steel Dynamics Inc. (SDI). The focus of this project and result was to have designed and assembled an apparatus to verify the calibration process for a high force output in their Bend Test Machine, which verifies weld strength in rails, that’s currently being operated in the SDI Rail Division in Columbia City. SDI initially requested to generate a greater force which exceeds the current machines capability. This however was held off because to advance with any modifications, this would void the manufacturer’s warranty. This would place the responsibility of maintaining and calibrating the machine to SDI. Hence, the calibration process was deemed top priority. Though the scope of the project has changed from applying a new hydraulic cylinder to designing a support platform, the team have adapted and managed the change in direction, to complete the new requirements of the project.

The ability to calibrate the machine is crucial for validating the weld process. An SDI customer may audit the manufacturing process and require calibration data for the bend test machine. If this would be the first time a calibration process is to be implemented. As stated above, this process is important because due to the modifications, if any SDI customer audited the manufacturing process, the calibration data for the machine would be necessary for verification. Due to these new requirements, a brand new design phase, as well as the testing phase was completed in one semester. The result is a modified machine that can operate in rail bend testing mode, and calibration mode under a direct force of 484,450 lbf, up to 600,000 lbf, and proved the calibration method was correct.
Section 1: Problem Statement
1.1 Modified Problem Statement

With implementation of larger hydraulic cylinder in the Rail Bend Test machine (recommendation to SDI from previous semester), SDI loses calibration guarantee from machine manufacturer. To maintain manufacturing process control, and validate the testing method to customers, SDI requires the capability to calibrate the Rail Bend Test machine. SDI has requested a re-design to the existing machine to support a calibration cell under full hydraulic load from the current cylinder.

1.2 Requirements & Specifications

Future upgrades/modifications are planned for the Rail Bend Test machine, which includes a hydraulic cylinder with a larger force than what is currently installed. The first design requirement is to create a support robust enough to handle the larger force of the future cylinder. Material restrictions were also put in place due to the excess stock of ASTM A36 that was owned by SDI. In order to reduce new costs to SDI, ASTM A36 is the working material for the majority of the design. To aid in the manufacturing of the re-designed frame, individual components of the design are required to be less than 500 lbs. An overhead crane is to maneuver components in place and has a max capacity of 500 lbs. Finally, the calibration cell, which has a height of 12 inches, must make contact with the hydraulic cylinder, i.e. the platform must be in a location that when the calibration cell is inserted, the top of it falls within the stroke length.

1.3 Design Variables

The design variables consisted of the size, location, and orientation of the support structure. Because the frame has to disperse the force internally, the size (mass) of the support structure is critical. The location of the components of the support structure may also vary in response to where the highest stress is found. Orientation of individual components can vary to maximize the ability to bend and disperse force.

1.4 Limitations & Constraints

It is desired to minimize the changeover time as well as labor when the machine is transitioned between calibration mode and rail bend test mode. The maximum desired time for this changeover was set at 5 minutes. The original machine frame is to remain intact; no material may be removed. This is to ensure a baseline of strength, and only material addition to the frame is allowed.
Section 2: Conceptual Designs
With the new scope of the project, brainstorming was done to come up with a new set of conceptual designs to go about designing the calibration support. These designs were comprised of two key ideas.

2.1 Frame Mount Support

This structure was designed to mount directly to the existing frame. It would attach to the front and rear sections of the frame, as shown in Figure 1. This design could be easily installed and removed from the frame; however, ensuring structural integrity would be difficult due to the distance between the mounting points.

![Frame mount design for load cell support.](image)

**Figure 1:** Frame mount design for load cell support.
2.2 Ground Support

The second design shown in Figure 2 would be placed within the machine and supported by the ground. This structure would carry the required load more easily than the frame mount design. The concrete floor supporting the structure would need to be analyzed to ensure stability. Unlike the previous design, this would be a permanent fixture on the frame.

![Figure 2: Ground support design for load cell.](image)
The frame mount and ground support designs were flawed. The frame mount design could not be used due to the stresses being distributed on to its surface. The A36 steel could not withstand the force of the load cell. The ground support design could not be used either because the downward force of the hydraulic cylinder would be able to lift the machine frame off the ground. The thought of adding a permanent fixture on the frame was utilized because it did not require a changeover process from the bend test to the calibration process. By utilizing the successes and failures from the initial conceptual designs, the following options were considered:

2.3 Frame Mount Alternate Design 1

By taking an alternate direction of the first design and implementing a fixture from the second design, this hybrid was conceptualized. The frame mount goes downward into the frame and lies on top of an I-beam which is fixed to the frame and takes the majority of the outputted load. The cone structure’s purpose was to distribute the stress of the load cell to a greater area. This design failed due to the stresses found on the top surface of the cone surface failing and the I-beam structure failed from bending.

Figure 3: Frame mount alternate design 1.
2.4 Frame Mount Alternate Design 2

The frame mount alternate design 2 was an option given to us by Andrew Weber. The frame mount design intrigued him and he did not mind a fixture of sorts underneath the T and L beams. This design has 2 cylinders, extending from the inner faces of the front and back plates. A hook design is then mounted from the cylinder to the bottom of the frame mount to distribute the stresses. Other features were added to the frame mount to distribute the stresses as well. This design failed due to stresses. Another alteration of the frame mount design was analyzed but also failed. The material specifications given by SDI were then questioned.

Figure 4: Frame mount alternate design 2.

Another alteration of the frame mount design was analyzed but fails once again. It was at this point where we had realized that the restrictions on the projects were causing failures. We had to think outside forces and concentrated on the materials we were working with.
2.5 ASTM A36

As can be found under the project requirements, the material for the load cell support was to be ASTM A36 steel. A36 steel has a yield strength of 36,300 psi. This as well as other mechanical properties of A36 can be found in Table 1.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Metric</th>
<th>English</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>7.85 g/cc</td>
<td>0.284 lb/in²</td>
</tr>
<tr>
<td>Ult. Strength</td>
<td>400 - 550 MPa</td>
<td>58000 - 79800 Psi</td>
</tr>
<tr>
<td>Yield Strength</td>
<td>250 MPa</td>
<td>36300 Psi</td>
</tr>
<tr>
<td>Young’s Modulus</td>
<td>200 GPa</td>
<td>29000 ksi</td>
</tr>
<tr>
<td>Poissons Ratio</td>
<td>0.26</td>
<td>0.26</td>
</tr>
<tr>
<td>Shear Modulus</td>
<td>79.6 GPa</td>
<td>11500 ksi</td>
</tr>
</tbody>
</table>

In order to determine if A36 steel was able to withstand the stress beneath the load cell, this stress was calculated using Equation 1. The height and diameter of the load cell as well as the maximum force output of the hydraulic cylinder were known. The load cell height was 12” and the diameter was 4”. A maximum stress of 45,693 PSI was found, which exceeds the yield strength of A36. Therefore, a different material had to be used beneath the load cell.

\[
\sigma = \frac{F}{A} = \frac{F}{\pi \times r^2} = \frac{lbf}{\pi \times 2^2 \text{ in}^2} = \frac{45,693 \text{ psi}}{574,200} = \text{45,693 psi}
\]
2.6 AISI 4140 Normalized

Since the yield strength of ASTM A36 was less than what was required to support the stress produced by the hydraulic cylinder onto the load cell, a different material was chosen. Through discussions with Drew Seely, a metallurgist for SDI, AISI 4140 normalized was chosen. As can be seen in Table 2, this high strength steel has the capability of withstanding the estimated stress applied.

Table 2: Mechanical properties of AISI 4140 normalized steel.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Metric</th>
<th>English</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>7.85 g/cc</td>
<td>0.284 lb/in²</td>
</tr>
<tr>
<td>Ult. Strength</td>
<td>972 MPa</td>
<td>141000 Psi</td>
</tr>
<tr>
<td>Yield Strength</td>
<td>635 MPa</td>
<td>92100 Psi</td>
</tr>
<tr>
<td>Young’s Modulus</td>
<td>205 GPa</td>
<td>29700 ksi</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.29</td>
<td>0.29</td>
</tr>
<tr>
<td>Shear Modulus</td>
<td>80 GPa</td>
<td>11600 ksi</td>
</tr>
</tbody>
</table>

The safety factor for AISI 4140 was found by dividing the yield strength by the maximum stress, which is shown in Equation 2.

\[
FOS = \frac{Yield \ Strength}{Max \ Stress} = \frac{92100}{45693} = 2.02
\]  

(2)

The normalized 4140 steel can withstand the force but it is still unknown if the A36 can and it will depend on the type of stresses that it will experience and the cross-sectional area of the normalized 4140 steel. It is known that the thickness of this buffer will be 3” to ensure the top of the load cell is within the length of the stroke of the hydraulic cylinders piston.
2.7 Hybrid Design 1

Taking into account what was learned about the previous designs and material properties, a new design was proposed. Instead of using an I-beam to absorb the majority of the force, a block of A36 will be used. Instead of a cone to distribute the force and give the load cell the leverage required to make contact with the loading ram, a cylinder made of the normalized 4140 steel will be used. Instead of using the rods to hook on to frame mount, another hook will be used to grab the bottom of the block and extend the forces out to the lower section of the frame.

![Figure 5: Hybrid design 1 is shown.](image)

This design incorporates ideas from previous designs and new features such as the block and cylinder. However, this too failed due to stress concentrations found on the lower section of the frame.
SECTION 3: Final Design
The final design chosen was an alternate to the hybrid design which eliminated the stress concentrations that were located in the lower section of the frame where the feet are in contact with the front and back plates.

Figure 6: Cross section of the final design.

The Final Design is composed of 4 parts, each with a specific purpose. Those parts are described as follow:

- The Buffer
- The Block
- The Vertical Column Support
3.1 The Buffer

Made of AISI 4140 normalized steel, its purpose is to withstand the maximum pressure transferred from the load cell and to distribute the force along a greater area such that the A36 block would not fail.

The cylindrical shape was chosen to reduce the area, thus reducing cost. Its thickness of 3” was chosen to ensure that the load cell, when placed on top of the buffer, would be in length of the stroke of the hydraulic cylinder. The diameter of 8” was chosen so that the force was applied over a greater area, reducing the stress on the A36 block.

Figure 7: The buffer is shown highlighted in red.
3.2 The Block

Made of A36 steel, its purpose is to endure the majority of the distributed force from the buffer. The block has dimensions of 19.5” in length, 12” in width, and 8” in depth. To place the block on to the frame, the T-beam and L-brackets were cut out in the center. This did not violate our requirements of extensively modifying or removing parts from the frame because these 3 pieces serve a minor role. Their purpose is to simply hold wooden blocks which are meant to absorb and ensure the frame is safe from the rupturing rails during the bend test. The Block is to be welded on the inner surfaces of the front and back plates of the frame, and from where the block contacts the T-beam and L-brackets.

![Figure 8: The block is shown highlighted in red.](image)

The block experiences 3 different stresses. It experiences a normal compressive stress from the load cell, bending stress due to the moment caused from the frame, and shear stress at the contact point on the fixed rigid ends. Each of these stresses is calculated to determine the principal stresses in this subassembly part.
The principal stresses must be found because since this particular part is undergoing these different types of stresses, the maximum stress has to be determined to ensure that it passes. Equation 1 shows how to calculate the principal stresses ($\sigma_{1,2}$). There are no forces acting along the horizontal direction, thus the stress in the x-direction ($\sigma_x$) is 0. The normal stress and bending stress both act along the vertical direction, thus the stress in the y-direction ($\sigma_y$) is the summation of these stresses. The shear stress acts along both sides of the block ($\tau_{xy}$).

\[
\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}
\]  

(3)

Where,

\[
\sigma_x = 0, \quad \sigma_y = \frac{F}{A_{circ}} + \frac{M * c}{I}, \quad \tau_{xy} = \frac{3 * V}{2 * A_{rect}}
\]  

(4)

To determine the moment and shear force the block undergoes, an online calculator was used to draw the Free Body, Shear Force, and Moment Diagrams. This program was found on the “Science and Engineering Café on the Net” website.

![Free Body Diagram](Figure 9: The free body diagram is shown.)

By inputting the correct information regarding the force, dimensions, and mechanical properties of the block, the program generates plots and calculates the required information.
**Figure 10:** The left box shows the inputted information & the right shows the shear diagram.

**Figure 11:** The left box shows the results & the right shows the moment diagram.
\[ \sigma_y = \frac{574,200 \text{ lbf}}{\frac{\pi}{4} \times 8^2 \text{ in}^2} + \frac{1,165,833 \text{ lbf} \times \text{ in} \times 4 \text{ in}}{\frac{12 \times 8^3}{12} \text{ in}^4} = 17,655 \text{ psi} \]  

(5)

\[ \tau_{xy} = \frac{3 \times 253,360 \text{ lbf}}{2 \times 12 \times 8 \text{ in}^2} = 3,969 \text{ psi} \]  

(6)

\[ \sigma_{1,2} = \frac{0 + 17,655}{2} \pm \sqrt{\left(\frac{0 - 17,655}{2}\right)^2 + 3,969^2} \]  

(7)

\[ \sigma_1 = 18,506 \text{ psi}, \quad \sigma_2 = -851 \text{ psi} \]  

(8)

By determining the maximum principal stress (\(\sigma_1\)), the Factor of Safety for the Block can be determined.

\[ FOS = \frac{\text{Yield Strength}}{\text{Max Stress}} = \frac{36,300}{18,506} = 1.96 \]  

(9)

With the factor of safety above the 1.5 requirement, the dimensions for the buffer and the block were proven to be adequate. The calculations shown above were confirmed with the SolidWorks, finite element analysis results.
3.3 The Vertical Column Support

Made of A36 steel, its purpose is to reduce the bending moment to relieve the stresses on the contact surfaces of the block.

Figure 12: The vertical column support is shown highlighted in red.
3.4 The Bottom Lip

Made of A36 steel, its purpose is to increase the flexural stiffness, remove the stress concentrations, and to support the vertical column support.

Figure 13: The bottom lip is shown highlighted in red.
3.5 Finite Element Analysis

The final design was modeled using SolidWorks, which has a simulation package for FEA. Using the software, multiple tests were simulated for the calibration process and bend test. Both 11” and 12” bores were used in these tests, which generated forces of 484,200 lbf and 574,200 lbf, respectively. Minimum safety factors were found for each of the four tests.

- Calibration for the 11” bore outputted a FOS of 2.01 with a maximum stress of 18,060 psi located at the surface of the block.

![Figure 14: The calibration process was simulated with a force of 484,200 lbf](image)
Calibration for the 12” bore outputted a FOS of 1.7 with a maximum stress of 21,353 psi located at the surface of the block.

**Figure 15:** The calibration process was simulated with a force of 574,200 lbf
- Bend Test for the 11” bore outputted a FOS of 2.49 with a maximum stress of 19,277 psi located at the location where the hydraulic cylinder is mounted to the frame.

**Figure 16:** The bend test was simulated with a force of 484,200 lbf
• Bend test for the 12” bore outputted a FOS of 2.1 with a maximum stress of 22,857 psi located at the location where the hydraulic cylinder is mounted to the frame.

**Figure 17:** The bend test was simulated with a force of 574,200 lbf

The minimum factor of safety found from the FEA results was 1.7 for the calibration process with the 12” bore. This exceeds the safety factor requirement of 1.5.
Section 4: Building Process
Before components are assembled in place, it is necessary to determine the tools available/needed, the maneuverability of the frame, as well as the order in which components are to be installed. The tools necessary are an arc welder and an overhead lift to maneuver the individual parts into place. SDI has the capability to maneuver the frame on its side for easy access to install and weld components. The order of installation begins with the bottom lip, then the vertical support column, and finally the block which is sectioned in three pieces.

The build process begins with removing the hoses that connect the pumps to the hydraulic cylinder as well as disconnecting the DAQ lines to the PLC program. The bottom lip then has two eye bolts welded on opposite ends. Chains and industrial carabiner are then hooked to the eye bolts and to the overhead crane. The bottom lip is then maneuvered into place and clamped. Every connection between the bottom lip and the existing frame are arc welded per industry standards.

Once the bottom lip is installed, the vertical column support is then placed directly center underneath the hydraulic cylinder. To hold the support in place, the outside of the top and bottom flange are welded to the bottom lip. This ensures no slipping under heavy load. It is not necessary to weld all edges as the weld is not critical to the strength of the column.

With the vertical column support installed, all three sections have an eye bolt welded to them. The first section is maneuvered into place with the help of the overhead crane. Each section has a chamfer on both sides that come in contact with the front and back of the frame to ensure deep weld penetration. With the section centered on the vertical column support, it is welded into place. The section void of material where the chamfer was cut is then filled in with weld. When this section cools the slag is removed and the weld grinded down so the next section lays flat on top of it. The same process is repeated for the remaining two sections of the block.

The machine is then painted along the new parts for aesthetics. Hydraulic lines and DAQ lines are reconnected. Air is purged out of the hydraulic lines and normal operation may resume.
The following are images of the parts in place and the welds:

**Figure 18:** Front view of inner frame.

**Figure 19:** Side view of inner frame.
Figure 20: Side view of the block.

Figure 21: Side view of all new parts in the machine.
**Figure 22:** Rear view of the machine showing the bottom lip welds.

**Figure 23:** Front view of the machine showing the bottom lip welds.
Section 5: Testing
With the hand calculations and FEA results collected, the testing procedure began. Two different tests were conducted, the calibration simulation and bend test.

5.1 Calibration Simulation

To test the calibration process, we would need the loading ram to contact the load cell which sits above the buffer. However, the load cell was not inserted in the machine, but the force and stresses the frame would undergo must be simulated as if it were present in the machine. To do so, the loading ram would make contact with a section of 136RE rail. The 136RE was not tall enough to contact the loading ram, so 4 small plates were placed underneath it.

![Diagram of Calibration Test Set-up]

Figure 24: The Calibration Test Set-up is shown.

This set-up will accurately simulate the calibration process with the substitution of the shown materials.
After conducting this test twice, the following results were found:

**Figures 25a & 25b:** The calibration simulation both generated the above plots.

From inspecting the above figures, both of the calibration simulations output a maximum force of 2154 kN and a displacement of 0.19".
5.2 Bend Test

To test the rail bend machine under its normal function, the bend test for two rails was performed. A section of rail, with the weld in the center, was placed on the simple supports and the machine is run until the hydraulic cylinder completely extends or the rail ruptures.

![Image of rail bend machine](image-url)

**Figure 26:** The rail bend machine is tested as it normally would be.

The above figure shows a section of 136RE rail making contact with the loading ram.
After conducting this test twice with different rails, the following results were found:

**Figures 27a & 27b:** Generated plots from the bend tests.

From inspecting the above figures, the maximum force and displacement outputs were 2000 kN and 0.59” for Figure _a and 2154 kN and 1.12” for Figure _b.
Section 6: Evaluations
In order to verify that the theoretical model of the design was acceptable, the FEA results were compared with experimental data collected from the testing of the implemented design. All testing was performed under conditions which simulated calibration. During testing, the total displacement was measured to be 0.19”. The displacement measured was the total distance of extension of the piston, which included the downward displacement of the load cell support as well as the upward displacement of the hydraulic cylinder due to the reaction force.

A finite element analysis was performed on the design under the same conditions as testing. The total displacement in FEA was found to be 0.078”. This displacement was found by summing the displacement of the cylinder, the load cell support, and the compression of the material between the loading ram and load cell support. Excluding the 0.019” due to the compression of the spacing material, the displacements of the cylinder and load cell support can be found in Figure X. The 0.019” displacement was found using a separate FEA simulation.

![Figure 28: Theoretical displacement of the hydraulic cylinder and load cell support.](image)
The difference between the theoretical and experimental displacements was found to be 83.9%. There are many possible sources of error, including: elastic deformation of the loading ram and imperfect contact due to surface roughness and insufficient flatness.

The surface of the loading ram which contacted the 136RE rail during the calibration simulation was softer than the rail, and therefore deformed during the first calibration test. A portion of this deformation was plastic deformation, causing a permanent indentation in the loading ram surface. The material was hardened due to this plastic deformation. In subsequent tests, there was no evidence of further plastic deformation; however, there may have been elastic deformation in this area. This would account for a portion of the discrepancy between the theoretical and experimental displacements.

All plates used during testing were machined, and therefore had a significant surface roughness. This roughness may cause imperfect contact between the plates. This imperfect contact could cause elastic deformation due to the reduced area of contact, which would account for a small portion of the difference in the displacements.

There are many factors which may affect the flatness of the parts used in the load cell support. The flatness of the parts was not measured prior to their installation, leaving the possibility of imperfections due to manufacturing. Another source of error could have been warping due to the welding process. During welding, the material is significantly heated. As the material cools, it tends to shrink, possibly causing gaps between the contact surfaces. These gaps could result in a higher deflection, which could account for part of the greater displacement during testing.
Section 7: Cost Analysis
The total cost of the project, as specified by Steel Dynamics, was not to exceed $30,000. In order to determine if the project met this requirement, a detailed cost analysis was performed. The cost was divided into 4 subgroups: buffer disk, hydraulic cylinder, labor, and material. The buffer disk was made from normalized AISI 4140 steel, which Steel Dynamics ordered from a different manufacturer. A hydraulic cylinder with a 12” bore as well as its installation was quoted from Tri-Tech Engineering Inc. Labor cost included the manufacturing and installation of the load cell support and was provided by Steel Dynamics. The material cost covered the ASTM A36 steel which was the primary material used for the load cell support which was considered excess and incurred no cost to Steel Dynamics. As seen in Table X, the total cost of the project was $19,327 which was approximately 64% of the allotted budget.

Table 4: Detailed cost analysis of rail bend test machine project.

<table>
<thead>
<tr>
<th>Parts</th>
<th>Descriptions</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Buffer Cylinder</td>
<td>8&quot; Diameter x 3&quot; Thick 4140 Pre-hardened</td>
<td>$417.00</td>
</tr>
<tr>
<td>Hydraulic Cylinder</td>
<td>12&quot; Bore x 8&quot; stroke cylinder, 7&quot; piston rod, no rod bumper</td>
<td>$18,490.00</td>
</tr>
<tr>
<td>Labor</td>
<td>Manufacturing &amp; Welding by in house operators</td>
<td>$420.00</td>
</tr>
<tr>
<td>Material (Block, Bottom Lip, Column)</td>
<td>SDI excess</td>
<td>$0.00</td>
</tr>
<tr>
<td>Total cost for both designs:</td>
<td></td>
<td>19,327.00</td>
</tr>
</tbody>
</table>
Section 8: Conclusions & Recommendations
The experimental test was run 4 times with no problems. Since the calibration will be performed once per year, this testing was comparable to 4 years of calibration. During the experimental testing of the load cell support, a total displacement of 0.19” was found. A theoretical displacement of 0.078” was found in the FEA simulation. This equates to a difference of 83.9%. The discrepancies between the theoretical and experimental values can be explained by elastic deformation of the loading ram and imperfect contact due to surface roughness and insufficient flatness.

A total cost estimate for the project of $19,327 was found which is well within the allotted project budget of $30,000. This cost estimate included: the buffer disk, hydraulic cylinder, labor, and material for the load cell support.

It is recommended that a hydraulic cylinder with a 12” bore be installed in order to sufficiently test the higher strength rail that Steel Dynamics plans to produce. Once installed, this hydraulic cylinder should be calibrated in order to ensure the accuracy of the measured force.
References


The Appendix will include the following:

- Dimensions of the created parts
  - The Block with the Vertical Column Support
  - The Bottom Lip
  - The Buffer
- Final Design
The Block with the Vertical Column Support

Name: Mid Block
Material: A36
The Bottom Lip
The Buffer

Name: The Buffer
Material: AISI 4140 Normalized