Sheet Metal Transfer Device
ME 493 Capstone: Final Report
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Streimer Sheet Metal Works, Inc.

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Executive Summary

Streimer Sheet Metal Works, Inc. is a Portland, Oregon-based company that specializes in the supply of sheet metal products, systems, and services. Streimer commissioned a capstone group to develop a device that would assist in a shop floor process - the transfer of rough-cut metal sheets from a conveyor to a CNC plasma-cutting table for final dimensioning. The current transfer process is manually driven and labor intensive, making it undesirable and unsafe. Thus, operator safety is the principal factor driving the customer’s need for an alternative means of transfer. Secondary considerations include but are not limited to ergonomics, efficiency, and simplicity of design.

Following the proper design process, the team developed several potential solutions through internal and external research, eventually vetting these rough concepts through a series of decision making criteria. The team selected and refined a design for a transfer apparatus that would meet the customer’s needs, with the additional benefits of an ergonomic and hands-off process. The completed device integrates a boom-mounted vacuum lift system with a transfer cart, reducing the operator effort in transferring a sheet from the cart to the CNC.

After an extensive design review process, the team and Streimer agreed to a set of deliverables including a portion of the completed, custom-machined components, a complete set of mechanical drawings detailing assembly of components, bill of materials, and all CAD model files associated with the device.
Table of Contents

1.0 Introduction ................................................................................................................. Page 3
2.0 Mission Statement ........................................................................................................... Page 4
3.0 PDS Requirements ......................................................................................................... Page 4
4.0 Alternative Design Concepts/Selection Process ............................................................. Page 5
5.0 Final Design ..................................................................................................................... Page 7
6.0 Evaluation ....................................................................................................................... Page 10
7.0 Conclusion and Recommendations ................................................................................ Page 13
Appendix A - PDS and Decision Making Matrices ............................................................... Page 16
Appendix B - Calculations ................................................................................................... Page 20
Appendix C - FEA Modeling and Reporting ....................................................................... Page 28
Appendix D - Trolley Manufacturing .................................................................................. Page 32
1.0 Introduction

Streimer Sheet Metal Works Inc. is a Portland company that fabricates and installs a variety of sheet metal products. The company specializes in commercial and industrial HVAC systems, architectural and clean room technologies, as well as duct accessories. Streimer’s production facility utilizes a CNC plasma cutter to cut individual pieces of sheet metal into final dimensions for later assembly. Prior to being placed on the CNC table, rolls of sheet metal are decoiled, sheared into rough blanks, and fed along a conveyor which feeds the sheets onto a transfer cart. At times the stack of sheet metal loaded onto the transfer cart can weigh in excess of 3000 pounds. After stacking, the cart is pushed by hand to a CNC plasma cutter where the operator manually unloads the sheets onto the table (Figure 1). The capstone team was tasked with designing and fabricating a device to optimize the transfer of sheet metal from the decoiler conveyor to the CNC plasma table.

Following a site visit to explore the shop and better understand the scope of the problem, the team performed extensive research in order to develop a list of product design specifications (PDS). The development of the PDS was a collaborative effort between the team and Streimer, ultimately to provide design benchmarks that the team would work toward during the design process.
2.0 Mission Statement

The purpose of this project was to design and fabricate a device that will allow for a safe, hands-free transportation of sheet metal at Streimer Sheet Metal Works production facility. Per Streimer’s design requirements and subsequent PDS, the device would need to accommodate sheets ranging from 26 to 16 gauge, and 36” x 36” to 120” x 72” in area. Streimer also emphasized that operator safety was their driving concern for proposing the project - the primary motivation in developing the device was to make the transfer process as hands free and ergonomic as possible. In addition to the aforementioned design targets, the apparatus would need to be easy to operate and robust enough to handle the demands of industrial production. Upon completion of an agreed-upon design, the team would initiate fabrication and provide all necessary drawings and supporting documentation to the sponsor.

3.0 Product Design Specifications

An extensive list of specifications that the final design needed to meet were developed through a series of meetings between the team and Streimer (Appendix A). To efficiently define and address all needs for the project, the team implemented a PDS chart prioritizing the requirements. As stated above, safety was the greatest priority. Other high priority design criteria included performance, quality, and reliability. Some of the specifics of these high-priority requirements included:

- Presence of emergency shut-offs and fail safe devices
- No unguarded pinch points
- Capability to lift double the weight of the largest sheet
- Ability to match or reduce current process cycle time of 30 seconds

Considerations such as cost of production, service life and weight were determined to be medium priority. Additionally the team identified low-priority criteria that had the least effect on the customer’s stated objectives for the design.

Though minor adjustments were made after the PDS Review with the customer, the PDS remained relatively unchanged as the project progressed. The final design met
all PDS criteria, and implemented numerous design considerations such as limit switches, emergency stops, and a vacuum loss alarm system to ensure that the apparatus adequately addressed Streimer’s safety concerns.

4.0 Alternative Design Concepts/Selection Process

Initially, the team developed eight rough concepts with a wide range of design elements ranging from fully automated carts with docking stations, to carts with pneumatic or hydraulically-powered mechanisms for assisted unloading. The concepts were evaluated using a decision matrix and were ranked based on how closely they met the high priority requirements of the PDS (Appendix A). Upon completing the matrix it was apparent that the eight concepts could be divided into three groups based on common means in which the sheet metal was delivered to the CNC table: overhead, gravity fed, or docked roller fed.

After further discussion and refinement, the team condensed the three groups of concepts into three representative ideas that could be presented to Streimer for feedback. The overhead option consisted of a mobile cart with a column-mounted telescoping boom and vacuum pad lifter (Figure 2). The gravity fed concept was a moveable cart with a top that was able to tilt in a controlled fashion to allow one sheet to slide off of the stack onto the CNC table (Figure 3). The final concept, representing the docked roller fed group, was a moveable cart along with a moveable dock. The cart would engage with the dock to which the dock would use a vacuum pad lifter and roller assembly to unload sheets onto the CNC table (Figure 4).
Options
- Powered or unpowered vacuum lifters
- Powered or unpowered cart movement
- Column raises - could be hydraulically operated: powered or hand-actuated
- Hoist raises sheets - could be hand-operated or powered
- Sliders on boom to extend horizontal reach

Figure 2 - Option I: Cart with integrated vacuum unloading

Options
- Powered or unpowered lifting mechanism - pneumatic, hydraulic, hand-pumped
- Powered or unpowered cart movement
- Several tray layers to allow for segregation of sheets by size

Figure 3 - Option II: Cart with gravity-assisted unloading

Options
- Powered cart moves unloader when docked
- Unloader with drive wheel
- Unloader/docked cart movement guided by track
- Compatibility with existing carts

Figure 4 - Option III: Cart with powered unloading dock
The three concepts were put through a simplified selection matrix administered by the team. The second matrix used weighted PDS criteria to determine the optimum concept (Figure 5). From this process, it was clear that the cart with the integrated vacuum lift frame best fit the specified requirements for the device. At a concept review meeting with Streimer, the three concepts were presented, along with the simplified selection matrix that outlined the team’s concept ranking process. After discussion, it was agreed that the overhead concept would become the finalized design.

<table>
<thead>
<tr>
<th>Criteria</th>
<th>I (Overhead)</th>
<th>II (Gravity)</th>
<th>III (Dock)</th>
<th>Criteria weight</th>
</tr>
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<tbody>
<tr>
<td>Safety</td>
<td>2</td>
<td>1</td>
<td>3</td>
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</tr>
<tr>
<td>Ease of Use</td>
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<td>3</td>
<td>0.2</td>
</tr>
<tr>
<td>Cost</td>
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<td>3</td>
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<td>0.1</td>
</tr>
<tr>
<td>Manufacturability</td>
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<td>2</td>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td>Size (Footprint)</td>
<td>3</td>
<td>2</td>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td># Components</td>
<td>2</td>
<td>3</td>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td>Total</td>
<td>2.3</td>
<td>1.8</td>
<td>1.9</td>
<td></td>
</tr>
</tbody>
</table>

Figure 5 - Simplified decision making matrix

5.0 Final Design

The final design built upon the functionality described in the overhead vacuum unloading option. Utilizing both custom fabricated parts and off-the-shelf components, the apparatus expanded upon many early design elements, with several important improvements. The device consists of a moveable cart with a ball-screw driven carriage assembly attached to a horizontally telescoping boom with an attached vacuum lift frame (Figure 6).
The boom is composed of two H-beams secured together and guided by a system of three trolleys, custom designed to travel along the beams. Stepper motors interfaced with a toothed, rack-and-pinion style belt system are present in both the lower and upper trolleys, while a third trolley on the edge of the top beam uses only idler casters for guiding motion. Controlled by a PLC, the stepper motors extend the H-beams simultaneously for maximum efficiency of movement, made possible by a calculated velocity ratio between the motors. The trolleys incorporate both concentric and eccentric steel, high capacity flanged rollers to provide adjustability and safeguard against bearing failure. Within the trolleys, nearly 20 individual components are used to ensure controlled rolling along the beams, along with proper belt tensioning and motor mounting (Figure 7). Bumper plates were also added as a safety measure to provide a hard stop in the event of a slipped belt. Attached to the lower trolley is the vacuum lifter, which has eight vacuum pads that can be repositioned or disengaged entirely to accommodate all applicable sheet sizes.
The complete boom assembly is mounted to the carriage, which moves up and down a vertically positioned H-beam column via a ball screw mechanism. The carriage assembly is made up of four primary plates, rollers, clevises, and the boom mount plates. The carriage was designed to withstand and absorb the moment generated by the boom assembly and to ensure that the ball screw wouldn’t be subjected to lateral forces. The carriage design, particularly the plate sizing and shape, as well as number and orientation of rollers, was validated through calculations (Appendix B). The ball screw assembly was selected after considering the weight of the boom loaded with the heaviest possible sheet, projected service life of one million cycles, and cost. The ball screw assembly consists of a mounting plate attached to the column, upper/lower mounts for the ball screw, and a mounting plate for the ball screw nut to interface with the carriage assembly. The base of the column is attached to the cart. 

Lastly, the cart itself consists of a frame, table top, four 8” casters, and ergonomic handles. The vacuum generator, controls enclosure and operator panel are
all incorporated into the cart and positioned to act as a counterweight to the boom. A thorough tipping analysis was done during the design of the cart to ensure that at its most extreme moment-arm condition, the device would remain rooted to the ground (Appendix B). Details such as the placement of the vacuum generator on the cart helped offset the tip conditions that the device will see during normal operation. Additional cart design considerations included maximizing strength while minimizing weight and ease of manufacturability.

6.0 Evaluation

During the design process, several concerns arose regarding tip, and the failure of the boom at several locations. While a proof of concept was created at the end of the project in the form of a boom assembly mounted to a stationary column, complete assembly and testing with a fully loaded cart or vacuum frame was not possible due to the project timeframe. As such, the majority of concept evaluation and validation was done via calculations and modeling.

With respect to the boom arm, FE modeling was performed in abaqus at the interface between the the trolley rollers and H-beams, as well as the bolted connection between the column mounting plates and telescoping boom assembly. At the request of Streimer, a report on the results of the FE modeling at the trolley-beam interface was generated. The analysis showed that the connection was well within safe limits; a safety factor of 3.5 for yielding was reported along the smaller, lower beam given a conservative loading scenario (Figure 8). Full text of this report is available in Appendix C. At the trolley-column interface, FEA demonstrated that the initial bolt selection of ¼” was insufficient, and informed the decision to switch to an array of larger, grade 8, ½” steel bolts at the interface.
A center of mass calculation for the system as a whole was needed in order to assure the safe operation of the machine. For this analysis, the worst case scenario was determined to be the telescoping boom fully extended, in the upper most position on the column, with the heaviest load at the end of the boom and no load on the cart. The total weight of the load consists of the sheet and the vacuum lifter assembly. The mass of the upper and lower boom consist of the steel H beam members and the trolley assemblies. The mass of the column assembly includes the carriage, ball screw, and mounting plates. The mass of the cart includes the frame weldment, and the mass of the machinery includes the vacuum generator. The general schematic showing the positions of these components relative to the front wheels and their respective center of mass is shown in figure 9.
The center of mass of the system is located at 15.5 inches height and 21.5 inches towards the center of the cart. The reaction forces on the front wheel and back wheel are 1279 lbs. and 871 lbs. respectively. This indicates that the cart will be stable under these conditions.

FEA analysis of the cart frame was performed. A standard deformable beam model with 2 node linear beam elements was constructed. A line load of 3500 lbs evenly distributed across the upper beams of the cart surface and a line load across the upright beams where the column connects was considered. The cart frame was constrained at the corners where the wheels attach in all directions. This resulted in a maximum deflection of 0.114 inches located at the upper cross members of the frame (Figure 10).
7.0 Conclusion and Recommendations

The capstone team was tasked with devising and implementing a solution to meet Streimer’s need for a more automated and therefore safer process of transferring sheet metal at their facility. By following a methodical design process, the team developed reasonable PDS targets, generated various concepts that met those targets, and selected the concept that best satisfied the agreed-upon requirements. The final concept was further developed into a robust design that successfully met all of the PDS requirements that were originally outlined, including the medium and low priority items. Through thorough analysis, the design was validated prior to fabrication. A functional proof of concept, consisting of full-scale design components in motion, provided further evidence that the team’s analysis was indeed sound. Notably, cycle time (which the team defined as the time the operator grabs a sheet from the transfer cart to the time the sheet is in place on the CNC plasma cutting table) was reduced from 30 seconds to
22 seconds, a huge improvement in efficiency when measured over the course of a three-shift workday.

While the final design was a success when measured against the PDS requirements, the design process was not without challenges. Having some perspective, the team now recognizes potential design improvements that could be made during future design revisions or if a redesign were to occur.

Material selection for the load-bearing device components proved to be challenging given the stringent factor of safety requirements outlined by Streimer. As such, the final device is quite a bit heavier than the current process transfer cart. Some weight increase was expected given the addition of the column-boom superstructure but in hindsight, the team would have done well to challenge the necessity of such a high factor of safety. A robust, functional design could have been attained at less of a weight burden while still remaining safe for the application. In terms of a specific example, had the factor of safety numbers been slightly less conservative, aluminum beams could have been used, reducing overall boom weight, thereby reducing the stress due to the moment that the boom/column interface would be subject to. Reduced stress at this critical juncture would have reduced the need for much of the stabilizing and supporting brackets that were specifically designed to counteract those stresses at a high factor of safety.

One possible retrofit of the existing design could incorporate motors on the cart itself in order to automate that portion of the process. Considerations were made regarding the overall weight of the system and maneuverability of the cart - large diameter, high-capacity urethane casters were specced. Yet the fully loaded cart may still prove unwieldy. Future cart-moving motors could easily adapt to the open frame design of the cart and would eliminate even more risks to operator safety.

Finally, while not necessarily a recommendation for future design iterations, but more a suggestion for future capstone teams, the team would have benefitted from more time allotted for fabrication. The team had six months to go from concept to finished product, a very narrow window, especially given the particular complexity of this
device. Reduced fabrication time was in part due to overly-ambitious scheduling by the team but primarily it was the result of setbacks that occurred because of the need for an extended design phase. This extension was contributed to by regular design review meetings, though constructive, the preparation for them cost the team valuable time, eventually triggering the abandonment of the original Gantt schedule altogether. Challenges aside, the team was able to provide Streimer with an effective and complete solution.
## Appendix A - PDS and Decision Making Matrices

### PDS, High Priority Criteria

<table>
<thead>
<tr>
<th>Rank</th>
<th>Criteria</th>
<th>Customer</th>
<th>Requirement</th>
<th>Metric</th>
<th>Target</th>
<th>Basis</th>
<th>Verification</th>
</tr>
</thead>
<tbody>
<tr>
<td>High</td>
<td>Safety</td>
<td>Streimer</td>
<td>No sharp corners or trip hazards</td>
<td>Number of hazards</td>
<td>0</td>
<td>Group Decision</td>
<td>Inspection</td>
</tr>
<tr>
<td>High</td>
<td>Safety</td>
<td>Streimer</td>
<td>Design for 50th - 90th percentile male</td>
<td>elbow to center of grip [in]</td>
<td>14 - 15</td>
<td>Streimer Requirement</td>
<td>Design</td>
</tr>
<tr>
<td>High</td>
<td>Safety</td>
<td>Streimer</td>
<td>No unguarded pinch points</td>
<td>Number of pinch points</td>
<td>0</td>
<td>Group Decision</td>
<td>Inspection</td>
</tr>
<tr>
<td>High</td>
<td>Safety</td>
<td>Streimer</td>
<td>Emergency shut off and safe fail design.</td>
<td>NA</td>
<td>NA</td>
<td>Group Decision</td>
<td>Design</td>
</tr>
<tr>
<td>High</td>
<td>Performance</td>
<td>Streimer</td>
<td>Lift 2x max sheet capacity</td>
<td>lbs lift</td>
<td>300</td>
<td>Streimer Requirement</td>
<td>Testing</td>
</tr>
<tr>
<td>High</td>
<td>Performance</td>
<td>Streimer</td>
<td>Meets current process cycle time or better</td>
<td>seconds</td>
<td>≤ 30 sec.</td>
<td>Group Decision</td>
<td>Testing</td>
</tr>
<tr>
<td>High</td>
<td>Performance</td>
<td>Streimer</td>
<td>Ability to transfer max sheet size</td>
<td>[in] x [in]</td>
<td>120 x 72</td>
<td>Streimer Requirement</td>
<td>Testing</td>
</tr>
<tr>
<td>High</td>
<td>Performance</td>
<td>Streimer</td>
<td>Ability to transfer min sheet size</td>
<td>[in] x [in]</td>
<td>36 x 36</td>
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<td>Testing</td>
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<td>High</td>
<td>Performance</td>
<td>Streimer</td>
<td>Ability to transfer from stack</td>
<td>Stack height [in]</td>
<td>2.5</td>
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<td>Testing</td>
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<tr>
<td>High</td>
<td>Performance</td>
<td>Streimer</td>
<td>Handle range of sheet metal thicknesses</td>
<td>sheet metal gauge</td>
<td>26-16ga</td>
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<td>Testing</td>
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<td>High</td>
<td>Performance</td>
<td>Streimer</td>
<td>Duty cycle</td>
<td>days of normal use</td>
<td>1</td>
<td>Group decision</td>
<td>Testing</td>
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<tr>
<td>High</td>
<td>Cost of production</td>
<td>Streimer</td>
<td>Total project budget for design, fab</td>
<td>[$]</td>
<td>10-15k</td>
<td>Streimer Requirement</td>
<td>Accounting</td>
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<td>High</td>
<td>Quality and</td>
<td>Streimer</td>
<td>Built with new,</td>
<td>NA</td>
<td>NA</td>
<td>Group</td>
<td>Design</td>
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<tr>
<td>Reliability</td>
<td>readily available components.</td>
<td>decision</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>-----------------</td>
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<td>----------</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>High Quality and Reliability Streimer</td>
<td>Maximum number of missed sheet transfers. (Non safety related)</td>
<td>failure rate</td>
<td>&lt; 1%</td>
<td>Group decision</td>
<td>Testing</td>
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**PDS, Medium Priority Criteria**

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<th>Rank</th>
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<th>Metric</th>
<th>Target</th>
<th>Basis</th>
<th>Verification</th>
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<tbody>
<tr>
<td>Medium</td>
<td>Ease of Use</td>
<td>Streimer</td>
<td>Hands free plasma table loading</td>
<td>NA</td>
<td>NA</td>
<td>Streimer</td>
<td>Inspection</td>
</tr>
<tr>
<td>Medium</td>
<td>Life in Service</td>
<td>Streimer</td>
<td>Reasonable fatigue life</td>
<td>Cycles</td>
<td>1 million</td>
<td>Streimer Requiremen t</td>
<td>Calculation</td>
</tr>
<tr>
<td>Medium</td>
<td>Size and Shape</td>
<td>Streimer</td>
<td>operates within specified space</td>
<td>[ft] x [ft] x [ft]</td>
<td>TBD</td>
<td>functionality</td>
<td>Testing</td>
</tr>
<tr>
<td>Medium</td>
<td>Ease of Use</td>
<td>Streimer</td>
<td>Fully loaded cart can be pushed/pulled with relative ease</td>
<td>max force [lbf] exerted per MIL-HDB K-759B, 1992</td>
<td>70</td>
<td>Streimer Requiremen t</td>
<td>Review of standards, Design analysis/calc s</td>
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<tr>
<td>Medium</td>
<td>Weight</td>
<td>Streimer</td>
<td>Minimum Cart load rating</td>
<td>Lbs</td>
<td>3500</td>
<td>existing cart load max.</td>
<td>Calculation</td>
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<td>Medium</td>
<td>Testing</td>
<td>Streimer</td>
<td>Appropriate safety factor</td>
<td>Multiple</td>
<td>Safety factor of 5 (excludes lifting mechanism).</td>
<td>Group Decision with Streimer Input</td>
<td>Computer modeling, FEA, Design Analysis</td>
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<tr>
<td>Medium</td>
<td>Environment</td>
<td>Streimer</td>
<td>Design has potential to adapt to alterations to manufacturing environment</td>
<td>NA</td>
<td>NA</td>
<td>Streimer Requiremen t</td>
<td>Design</td>
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<tr>
<td>Medium</td>
<td>Environment</td>
<td>Streimer</td>
<td>Compatible with available voltage.</td>
<td>voltage</td>
<td>120/240</td>
<td>Streimer Requiremen t</td>
<td>Design</td>
</tr>
<tr>
<td>Medium</td>
<td>Environment</td>
<td>Streimer</td>
<td>Compatible with available air pressure.</td>
<td>air pressure [psi]</td>
<td>90-140</td>
<td>Streimer Requiremen t</td>
<td>Design</td>
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<tr>
<td>Medium</td>
<td>Company Constraints and Procedures</td>
<td>Streimer</td>
<td>Maintain forklift access to plasma table.</td>
<td>NA</td>
<td>NA</td>
<td>Streimer Requirement</td>
<td>Design</td>
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<td>---</td>
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## PDS, Low Priority Criteria

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<th>Target</th>
<th>Basis</th>
<th>Verification</th>
</tr>
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<tr>
<td>Low</td>
<td>Maintenance</td>
<td>Streimer</td>
<td>Low maintenance</td>
<td>Frequency of service</td>
<td>twice per year</td>
<td>Group Decision</td>
<td>Design</td>
</tr>
<tr>
<td>Low</td>
<td>Installation</td>
<td>Streimer</td>
<td>TBD Based on Design</td>
<td>NA</td>
<td>NA</td>
<td>NA</td>
<td>NA</td>
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<td>Low</td>
<td>Environment</td>
<td>Streimer</td>
<td>Indoor operation only</td>
<td>NA</td>
<td>NA</td>
<td>Streimer Requirement</td>
<td>Design</td>
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<td>Low</td>
<td>Aesthetics</td>
<td>Streimer</td>
<td>Aesthetically pleasing</td>
<td>NA</td>
<td>NA</td>
<td>Group Decision</td>
<td>Inspection</td>
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<td>Streimer</td>
<td>Installation Instructions</td>
<td>NA</td>
<td>NA</td>
<td>Group Decision</td>
<td>Completion</td>
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<td>Streimer</td>
<td>Maintenance Instructions</td>
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## Concept Screening Matrix

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<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
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<td>0</td>
<td>0</td>
<td>0</td>
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<td>5</td>
<td>8</td>
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<td>4</td>
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</table>
Appendix B - Calculations

Below is a sample of calculations performed during the design phase:

Center of mass calculations are based on the center of mass equation

\[ R = \frac{1}{M} \sum m_i r_i, \]

for the coordinates of the center of mass relative to the front wheel of the cart (F_r0). The reaction forces at the wheels are a summation of the moment of the total mass about the front wheel and a summation of the forces in the z direction.

<table>
<thead>
<tr>
<th>part</th>
<th>var</th>
<th>value</th>
<th>Center of mass x Dimensions</th>
<th>Center of mass z Dimensions</th>
<th>part</th>
<th>var</th>
<th>Weights</th>
</tr>
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<td></td>
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<td>15</td>
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<tr>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Total Mass 1816</td>
</tr>
</tbody>
</table>

\[ R_x = 15.51652 \]
\[ R_z = 21.4978 \]
\[ F_{r0} = 1279.367 \]
\[ F_{r1} = 871.6333 \]
TELESCOPING BOOM
DRIVE MECHANISM

3/3/2016
Jon Petersen

SHAFT B TRAVEL = \( \chi B \)
SHAFT C TRAVEL = \( \chi C \)
REQUIRED GEAR RATIO = \( \frac{\chi B}{\chi C} = \frac{5}{3} \)

\[
5 \frac{\chi C}{\chi B} = 3 \frac{\chi B}{\chi C} \Rightarrow \frac{\chi C}{\chi B} = \frac{3}{5} \frac{\chi B}{\chi C}
\]

\[
\chi C = \chi B \left( \frac{3}{5} + 1 \right)
\]

\[
\chi C = \frac{8}{5} \chi B
\]

SECTION A-A

UPPER TROLLEY
BELT TO DRIVE MOTOR
ATTACHED TO VACUUM FRAME

SECTION B-B

LOWER TROLLEY
BELT TO THE BEAM

TOP BELT MOTION
\[
\omega_a = \omega_c \cdot \frac{r_c}{r_b} \Rightarrow \omega_a = \omega_c \cdot \frac{r_c}{r_b} 
\]

\[
\omega_b = \omega_c \cdot \frac{r_c}{r_b} 
\]

\[
\omega_B = \omega_c \cdot \frac{r_c}{r_b} 
\]

BOTTOM BELT MOTION

\[
\omega_C = \omega_b \cdot \frac{r_b}{r_c} 
\]

\[
\omega_b = \omega_c \cdot \frac{r_c}{r_b} 
\]

\[
\omega_B = \omega_c \cdot \frac{r_c}{r_b} 
\]

\[
\omega_C = \omega_b \cdot \frac{r_b}{r_c} + \omega_a \cdot \frac{r_c}{r_b} 
\]
Motor sizing:


\[ v = r \omega \quad a_t = \alpha r \]

relative velocity

\[ v_{BA} = v_B - v_A \]

time for cycle

\[ t_{cycle} = 5 \text{s} \]

bottom trolley speed \( v_c \) and drive belt speed \( v_b \)

(note belt speed equals upper trolley speed \( v_b \) since pulleys will be the same)

\[ \frac{v_c}{v_b} = \frac{8}{5} \]

required trolley speed to achieve cycle time

\[ v_c = \frac{\text{top} \cdot \text{bottom}}{t_{cycle}} = 17.6 \text{ in/s} \quad v_c = 88 \frac{\text{ft}}{\text{min}} \]

\[ v_b = \frac{5}{8} v_c = 11 \text{ in/s} \quad v_b = 55 \frac{\text{ft}}{\text{min}} \]

motor pulley radius and rotational velocity

\[ \rho_A = 0.375 \text{in} \quad (20 \text{ tooth drive pulley}) \]

\[ \omega_A = \frac{v_b}{\rho_A} = 4.669 \frac{\text{rev}}{\text{s}} \quad \omega_A = 280.113 \text{ rpm} \]

assume it takes \( x_{accel} \) inches of linear travel to achieve constant linear speed

\[ x_{accel} = 5 \text{ in} \]

\[ s = r \theta \]

\[ \theta_{accel} = \frac{x_{accel}}{r_A} = 763.944 \text{ deg} \]

\[ \alpha := \frac{\omega_A}{2 \theta_{accel}} = 32.267 \text{ rad/s} \]

\[ t_{accel} = \frac{\omega_A}{\alpha} = 0.909 \text{s} \]

LUMPING ALL MASS IN A DISK, FOR FLYWHEEL EFFECT

\[ I = \frac{1}{2} m r^2 \quad m_{total} := 385 \text{ lb} \]

\[ I := \frac{1}{2} m_{total} r_A^2 = 27.07 \text{ lb-in}^2 \]

\[ T_{accel} := I \alpha = 2.262 \text{ in-lbf} \]
Friction from rolling (0.005 is e for “dirty tram rails” from engineering toolbox)

\[ F_{\text{fric}} := 0.005 \cdot m_{\text{total}} \cdot g = 1.925 \text{ lbf} \]

\[ T_{\text{fric}} = F_{\text{fric}} \cdot r_A = 0.722 \text{ in lbf} \]

torque to overcome additional loading from deflection

\[ T_{\text{defl}} := m_{\text{total}} \cdot g \cdot \sin(2 \text{deg}) \cdot r_A = 5.039 \text{ in lbf} \]

service factor (from Gates design manual) \( SF := 1.7 \)

\[ T_{\text{design}} = \left( T_{\text{accel}} + T_{\text{fric}} + T_{\text{defl}} \right) \cdot SF = 13.639 \text{ lbf \cdot in} \]

\[ T_{\text{design}} = 218.222 \text{ ozf \cdot in} \]

\[ HP_{\text{design}} := T_{\text{design}} \cdot \omega_A = 0.061 \text{ hp} \]

\[ \frac{T_{\text{design}}}{SF} = 128.365 \text{ ozf \cdot in} \]
Ball screw force on boom:

\[ F_L = \sum M \cdot a_y + (\sum W) = (9.66 + 5.75 + 2.18) \cdot 24 + \left(150 + 185 + 70\right) \]

\[ F_L = 707.16 \text{ lbs} \]

Using ball screw spec of 960 lbs as max dynamic load.
Moment force about column:

\[ \Sigma M_0 = L \left( W_{\text{sheet}} + W_{\text{frame}} \right) + \frac{1}{2} \left( W_{\text{boom}} \right) + L_e \left( F_L \right) - 2L_R \left( F_R \right) \]

\[ \Sigma M_0 = 0, \text{ solve for } F_R \]

\[ F_R = \left[ L \left( W_{\text{sheet}} + W_{\text{frame}} \right) + \frac{1}{2} \left( W_{\text{boom}} \right) + L_e \left( F_L \right) \right] / 2L_R \]

\[ F_R = \left[ 13.5 \left( 150 + 185 \right) + 13 \left( 70 \right) + L_e \left( 960 \right) \right] / 2L_R \]

\[ F_R = \frac{4995 + 960L_e}{2L_R} \text{ for } L_e = 0.5 \text{ ft}, L_R = 0.5 \text{ ft} \]

\[ F_R = 5470 \text{ lbs} \]

\[ M_0 = 2F_R \left( L_R \right) \]

\[ = 2 \left( 5470 \right) \left( 0.5 \right) \]

\[ M_0 = 5470 \text{ lb-ft} \]
This table compares reaction forces of multiple roller couple arrangements. Multiple roller pairing models were solved using a simple setup in an FEA software, a singular beam with a moment applied and roller contact points at the spacings shown in the table below.

<table>
<thead>
<tr>
<th>Roller Spacing</th>
<th>3&quot;</th>
<th>6&quot;</th>
<th>9&quot;</th>
</tr>
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<tbody>
<tr>
<td>Roller Pair Reaction Force</td>
<td>F.1</td>
<td>F.2</td>
<td>F.3</td>
</tr>
<tr>
<td>Single Roller Pair</td>
<td>10,940.00</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Single Roller Pair</td>
<td>-</td>
<td>5,470.00</td>
<td>-</td>
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<tr>
<td>Single Roller Pair</td>
<td>-</td>
<td>-</td>
<td>3,646.67</td>
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<tr>
<td>Two Roller Pair</td>
<td>-</td>
<td>1,195.31</td>
<td>2,786.46</td>
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<tr>
<td>Three Roller Pair</td>
<td>1,198.25</td>
<td>546.22</td>
<td>2,819.77</td>
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</tbody>
</table>

The Two Roller Pair is the best choice among the options listed in the table. The chosen roller has a vendor rated capacity of 3820 lbs meaning a roller pair capacity of 7640 lbs.
Column bending stress calculation:

**Outer Fiber Tensile Stress**

\[ \sigma = \frac{M}{I} \]

To satisfy factor of safety of \( \frac{5}{3} \)

\[ 500 < \sigma_{\text{yield}} \]

\[ \sigma_{\text{yield}} > \left( \frac{M}{I} \right) \frac{5}{3} \]

**I-Beam Section Modulus**

\[ S = \frac{I}{c} \]

\[ \sigma_{\text{yield}} > \left( \frac{M}{S} \right) \frac{5}{3} \]

\[ 54,700 \text{ lb} \cdot \text{ft} \cdot \frac{12 \text{ in}}{\text{ft}} = 656,400 \text{ lb} \cdot \text{in} \]

Chosen I-beam has section modulus \( S = 8.5 \text{ in}^3 \)

\[ M = 656,400 \text{ lb} \cdot \text{in} \]

\[ \sigma_{\text{yield}} = 40,000 \text{ psi} \]

\[ \sigma_{\text{yield}} > \left( \frac{M}{S} \right) \frac{5}{3} \Rightarrow 40,000 \text{ psi} > \left( \frac{656,400}{8.5 \text{ in}^3} \right) \frac{5}{3} \]

\[ \Rightarrow 40,000 \text{ psi} > 38,611.8 \text{ psi} \]
Appendix C - FEA Modeling and Reporting

The Model:

Figure 1 shows the telescoping boom under examination. The main objective is to determine the stresses at points of roller contact. For this reason only the boxed region of the boom was modeled. The actual finite element model is shown in Figure 2, due to the symmetric nature only half of the boom was modeled. A 183 lb load was applied to the model, which is slightly more than half the 350 lb limit load. This point was placed at the end of the full bottom boom length. Similarly, a second point was coupled to the top boom to and fixed to simulate the column mount. Note that both the moment arm and load are slightly more than actual, which makes for a more conservative analysis.

The bottom boom is aluminum, all other parts are steel. Both booms and rollers were modeled with solid 8-node brick elements. The roller shafts were modeled with beam (space frame) elements. The trolley plates were modeled with a combination of 3 and 4-node plate elements. Figure 3 shows the arrangement of rollers and trolleys included in the FEM, note the top two rollers on the upper trolley were neglected in this analysis since they provide a negligible reaction in this load case.

Contact was modeled between the roller surfaces and beam flanges with an assumed 0.3 coefficient of friction.

Figure 1 - Area of the telescoping boom under analysis
Figure 2 - Actual FEM. Load and boundary conditions were applied to points that were coupled to beam segment faces to induce shear and moment forces.

Figure 3 - View of the FEM showing rollers and trolley plates included in the analysis – trolleys are hidden on the left. Trolleys were modeled with plate elements and were assumed to be fixed to their respective boom flanges.
Perimeter nodes at holes for the roller and trolleys were each coupled to a single roller shaft node. The roller shaft was modeled with beam (space frame) elements.

**Results:**

FEA results for the top and bottom boom is shown in Figures 5 and 6. Maximum stress of 6.6 ksi occurs in the top beam on the underneath side of the bottom flange at the rear roller contact area. For the bottom beam a maximum stress of 11.4 ksi occurs on the underneath side of the top flange at the front roller contact area. Using yield strength values of 50 ksi for steel and 40 ksi for aluminum, FOS against yield is 7.6 and 3.5 for the top and bottom beams respectively.

![Figure 5 – FEA results for region of roller contact on the top steel beam. Von Mises equivalent stress [psi].](image)
Figure 6 – FEA results for the region of roller contact on the lower aluminum beam. Von Mises equivalent stress [psi].
Figure D1 - First step of trolley fabrication wasmachining sideplates on a CNC mill.
Figure D2 - After pressing in welding bushings, the side plates went back on the mill to finish machine the bore of the bushings for a precise slip fit of the roller shafts. The bushings with the larger ID are for eccentric rollers, used to accommodate for variations in beam flange thickness.
Figure D3 - Side plates after machining demonstrating the fit up of the turned ground and polished shafting used for fixturing during welding.
Figure D4 - The final product, assembled trolleys.
Figure D5 - Beam flange was thicker than expected, eccentric rollers didn't have enough travel to compensate. After machining 0.040" off the boom beams the trolleys fit nicely.