



UNIVERSITY OF MANITOBA

Royal Canadian Mint



ROYAL CANADIAN MINT

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Plating Line Transporter Retrofit

Draft Design Report

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Dear Professor Labossiere:

Please find enclosed our report "*Analysis and Remediation of Plating line Transporter Drive Shaft Failures*". The report has been submitted for evaluation on Monday, December 5, 2011 in accordance with the requirements of the MECH 4680 Engineering Design course. This report contains descriptions, analysis and drawings of each section of the final design, as well as a cost analysis.

We would like to acknowledge Mr. Kashif Khan and the employees at The Royal Canadian Mint for allowing us to visit the site and for providing us with valuable input.

We hope that this report provides you with all necessary details and exceeds the requirements of this project. If you have any questions, please do not hesitate to contact us at the following email addresses:

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Respectfully,

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Table of Contents

Abstract	iv
1. Introduction.....	1
1.1 Customer Needs	2
1.1.1 High Reliability.....	2
1.1.2 Standardized Solution	3
1.1.3 Easily Sourced	3
1.1.4 Rapid Maintenance	3
1.1.5 Serviceable by Current Workforce	3
2 Details of the Proposed Modifications.....	4
2.1 Drive Shaft Redesign	5
2.1.1 Drive Shaft Features	5
2.1.2 Drive Shaft Design Considerations.....	6
2.2 Motor Relocation.....	7
2.2.1 Motor Relocation Features.....	7
2.2.2 Motor Relocation Considerations	11
3 Cost Analysis	13
3.1 Estimated Implementation.....	13
3.1.1 Shaft Redesign	13
3.1.2 Motor Relocation	14
3.2 Annual Cost Savings	15
3.3 Payback Period Method	16
3.4 Present Worth Comparison Method.....	16
4 Conclusion and Recommendation	18
5 References.....	19
6 Appendix A – Drive Shaft Design Logic.....	21
6.1 Visual study.....	21
6.2 Insight from Lead Hands.....	23
7 Appendix B – Drive Shaft Stress Analysis	24
7.1 Drive Shaft Analysis	24

7.2	Endurance Strength Calculation.....	24
7.2.1	Surface Factor	24
7.2.2	Size Factor	25
7.2.3	Load Factor	25
7.2.4	Temperature Factor	26
7.2.5	Miscellaneous Effects Factor	26
7.2.6	Final Material Strengths.....	26
7.3	Strength of the Shear Pin.....	26
7.4	FEA Results.....	27
8	Appendix C – Design Drawings	30

List of Figures

Figure 1 current system.....	2
Figure 2: Visual comparison of flange designs, Old (Left) and New (right).....	5
Figure 3: Convergence data from previous tables illustrating the convergent behaviour.	7
Figure 4: Overall view of the left side of the transporter. The brackets, idlers and belt path are clearly visible.	8
Figure 5: View of the Main Support Bracket.....	9
Figure 6: View of the Secondary Support Bracket	9
Figure 7: Picture of the rear axle and belt.....	10
Figure 8: FEA Simulation of the Bracket Assembly. The solution did not converge, but the local geometry is identical to that currently being used.	11
Figure 9: Approximation of the current lateral stiffener.....	12
Figure 10: Drawing of existing flange	21
Figure 11: Broken shaft component showing crack propagating through welded area into the tube.....	22
Figure 12: Screen cap from SolidWorks Simulator illustrating simulation conditions. Cylindrical surfaces are free to move in the Θ direction. Flange was constrained in the X direction and bolt holes were fixed. Join it pinned and load applied 100Nm.	28
Figure 13: Convergence data from previous tables illustrating the convergent behaviour.	29

List of Tables

Table 1: Bill of Materials and Associated Costs for the Motor Relocation.....	14
Table 2: Cost Analysis Table.....	15
Table 3: Annual Drive Shaft Failure Expenses.....	15
Table 4: Endurance Strengths of Driveshaft Components.....	26
Table 5: Convergence data showing data from the stress bearing surfaces in the flange.....	28
Table 6: Convergence data for highest stress surfaces in the outer tube.	29

Abstract

Team ELEVEN was tasked by the Royal Canadian Mint (RCM) to redesign the drive shafts for the coin plating line transporters. These drive shafts are used on the plating line transporters that transfer the batches of coins between chemical baths and operate continuously with a 12 hour maintenance window per week. The RCM is currently using two different drive shafts for each transporter. The unexpected failure of these driveshafts creates an excessive amount of downtime during the plating process, which costs the RCM \$1550/hour in lost production. The RCM asked our team to redesign a standardized, more durable driveshaft to reduce the unexpected downtime.

To accomplish this, two potential modification to the transporters were developed. First, the flange currently used to connect different sections of the shaft together was redesigned to obtain lower stress concentrations and avoid cracks originating from the welds. Secondly, the transmission system used to transfer power from the motor to the wheels was redesigned to provide power more evenly to the wheels and to standardize the drive shafts.

1. Introduction

The Royal Canadian Mint (RCM) opened the doors of its Winnipeg facility in 1979, and prides itself in the high speed production of secure coinage for both domestic and foreign use. The need to have a secure and protected product has led to the production of multi-layered electroplated coinage. Production lines for these coins have had issues with uptime because of unscheduled breakdowns due to catastrophic and unpredicted driveshaft failure. These drive shafts are located on the plating line transporters that transfer the batches of coins between chemical baths and currently operate continuously with a 12 hour maintenance window per week. Visual inspections are performed on the areas most prone to failure. However, the shafts tend to fail catastrophically during coin production as the visual inspection is not sufficient to catch all cracks.

In the current drives system, the gearbox is attached directly to the front shaft of the transporter, and a belt passing through sheaves on one side of the transporter transmits power to the rear shaft. This results in different front and rear shafts, as well as providing unequal power distribution as seen in Figure 1.

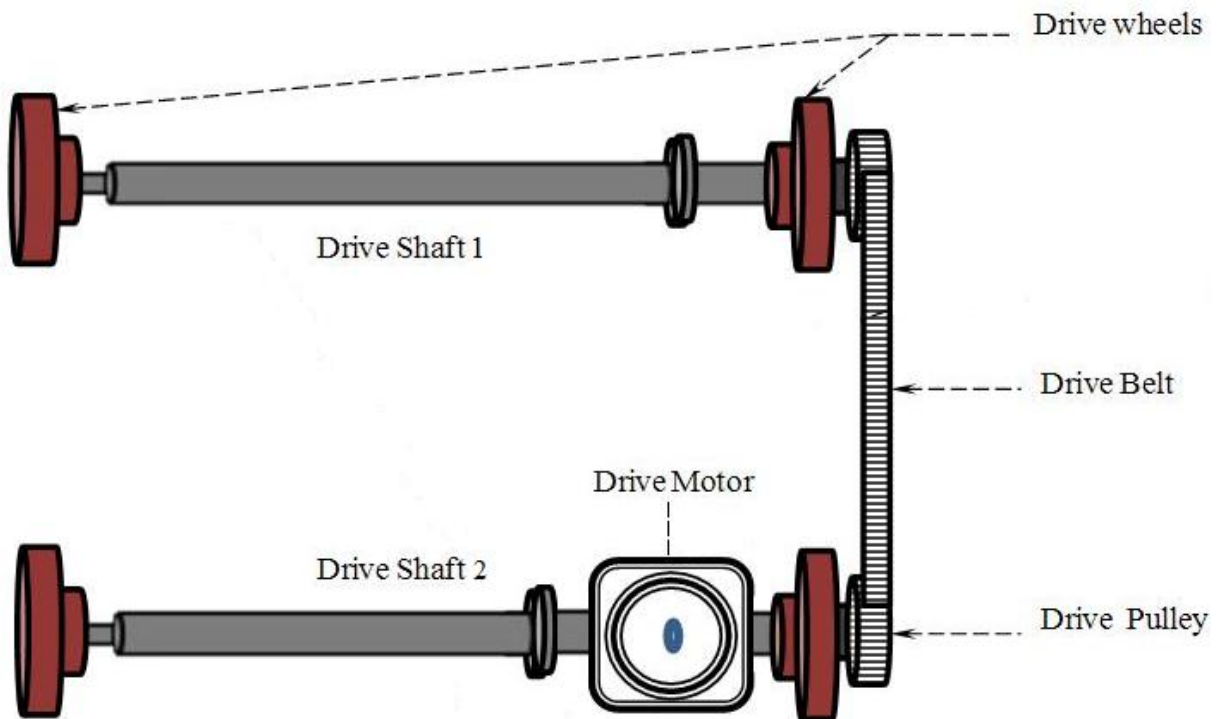


Figure 1: Current system

1.1 Customer Needs

Using the information provided by the RCM, our team identified the customer needs that the new drive shaft must address.

1.1.1 High Reliability

The design shall ensure that work stoppages caused by the failure of the plating line transporters' drive system do not occur. The system currently in use is prone to crack development in the drive axle, causing unscheduled downtime. The design will be capable of extended periods of service without requiring maintenance.

1.1.2 Standardized Solution

The design shall utilize identical shafts for both Line A and Line B transporters, as well as front and rear axles. As available storage space is limited, the number of parts kept available for exchange must be minimized. In addition, a lower number of parts will reduce the costs associated with the manufacture, installation and replacement of these parts.

1.1.3 Easily Sourced

The design shall have a low lead-time, and the components and labour must be obtainable quickly, easily, and from a variety of sources to ensure availability.

1.1.4 Rapid Maintenance

In the event of a maintenance replacement, catastrophic failure, or similar event, the design shall be able to be installed, maintained or replaced quickly. As the opportunity cost of downtime is high, any maintenance or replacement of the design must be rapid to minimized production loss.

1.1.5 Serviceable by Current Workforce

The majority of the work shall be performed by existing personnel. This will ensure a smooth and problem-free implementation and reduce dependence on external sources of labour, which could cause unwanted downtime because of availabilities. The subject plating lines are located in a high-security area, further increasing the need to use the existing workforce.

The area of concern centers on the drivetrain of the transporters. The current system has two plating lines that use four unique driveshaft designs that need to be fabricated each time a transporter fails. The driveshafts are prone to catastrophic failure in welds that attach flanges to the hollow shaft serving as the main axle. Therefore the focus of the group is to design a more robust drive train with more predictable fatigue properties as well as simplify or unify the

designs of the drive shafts

In order to meet the aforementioned customer needs, the design team developed a list of technical specifications to be satisfied by our design. These technical specifications each have a corresponding need to which they provide a solution. The technical specifications of our design are as follows:

1. The overall drivetrain system shall equalize the load between the front and rear drive shafts while preserving the all-wheel drive system already in place.
2. The design shall withstand shear fatigue loading caused by the transporters starting, stopping and changing direction. In addition, the design shall compensate for any minor misalignment of the current system that could induce a cyclic flexural load at the flange end of the current system.
3. The design shall utilize a single drive shaft design to ensure that fewer parts are held in inventory.
4. The design shall withstand the acidic and basic corrosive conditions of the operational plating environment.
5. The design modifications, replacement of the drive shafts, and any associated maintenance shall be within the capabilities of the current workforce. If required, manufacturers shall be sourced to produce the parts for installation.

2 Details of the Proposed Modifications

After considering several ways to eliminate shaft breakage, two main design components were selected. The flanges used to bolt the sections of shafts together had stress risers in the form of

sharp corners and high-stress welds. It was therefore decided to redesign the flange to eliminate these high-stress locations.

2.1 Drive Shaft Redesign

The initial design of the driveshaft was such that flange component at the connection end was prone to failure. The flange incorporates some glaring oversights that lead to failure that the new flange will aim to overcome. The first step in prevention of failure is to know the failure modes involved. Some of these were found from studying the failures in the flange while others came from talking to the maintenance personnel who build the shafts.

2.1.1 Drive Shaft Features

The new design prevents the faults that were associated with the initial driveshaft by manipulating the flange geometry and connection method. A visualization of the old and new flanges is provided in Figure 2 and detailed drawings attached in Appendix C.

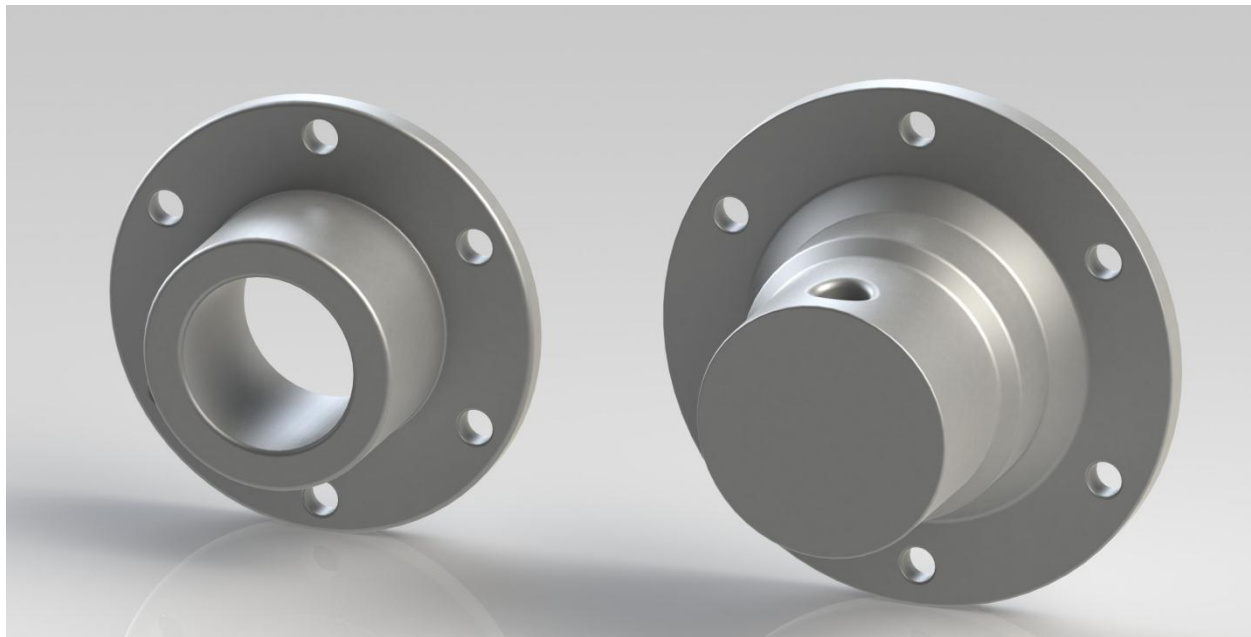


Figure 2: Visual comparison of flange designs, Old (Left) and New (right)

The RCM possesses highly specialized CNC equipment, capable of producing variety of complex components, which allows for freedom in design. As shown in Figure 2, the flange size has been increased considerably and the hollow section has been removed. This significantly increases the strength and lowers the machining time as the flange is formed from solid stock. The transitions in diameter are also chamfered to prevent the stress concentrations in the flange. A pin will be used as a primary connection instead of a weld to prevent the initiation of cracks in the parts. As an added benefit, the bolted connection will allow the RCM to reuse parts that haven't failed if breakdown does reoccur.

2.1.2 Drive Shaft Design Considerations

The design is based on fatigue loading conditions in a torsional system. As such, modifying factors have to include the strength calculation described by the equation below,

$$S_e = S'_e K_a K_b K_c K_d K_e [1]$$

Where S'_e is the endurance strength of material, K_a is the surface condition modifier, K_b is the size modification factor, K_c is the loading modification factor, K_d is the temperature modification factor and K_e is the miscellaneous-effects modification factor [1]. Details on the calculation for each factor are provided in Appendix A. The endurance strength of 304 stainless steel is given as 240 MPa [2, 3] which is reduced to slightly different values for individual shaft sections when all factors are considered with 99% reliability. Designing such that the Von Mises stress in the various parts is below this threshold will ensure that the design will not fail due to fatigue. This is confirmed by the Finite Element Analysis (FEA) discussed in detail in Appendix B. Results from this FEA are shown in the figure below.

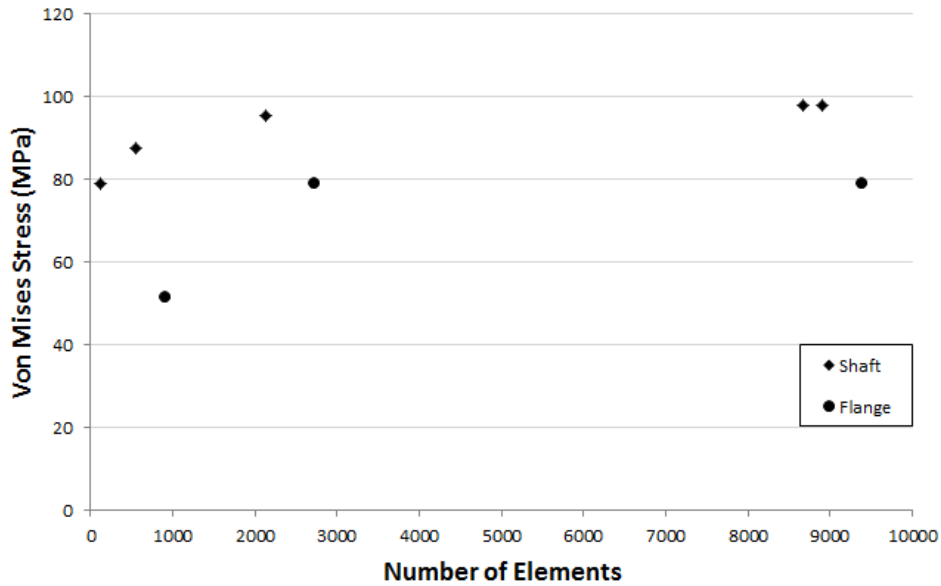


Figure 3: Convergence data from previous tables illustrating the convergent behaviour.

Figure 3 illustrates the convergence in the simulations with respect to the number of elements in the stress bearing surfaces.

2.2 Motor Relocation

As discussed in the introduction, the power is split on the front drive axle, resulting in unequal torque distribution between the four wheels. Relocating the motor allows a better distribution of forces to be achieved, and permits the use of identical front and rear shafts.

For detailed drawings, please refer to Appendix C.

2.2.1 Motor Relocation Features

To relocate the motor while using the same sheaves requires it to be moved vertically. This entails:

- The creation of a bracket to raise the motor, transfer case and bearings
- The replacement of the current single idler with a redirection pulley and an idler
- The creation of a cutout in the lateral stiffener and associated reinforcement doubler.

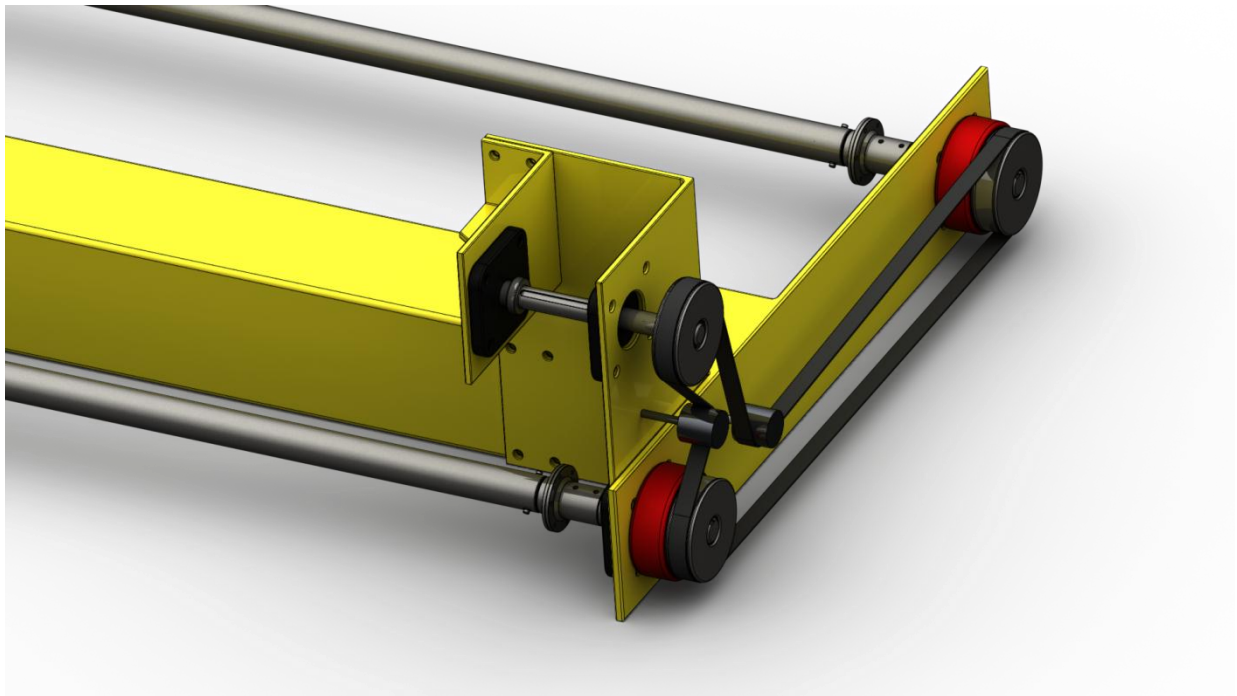


Figure 4: Overall view of the left side of the transporter. The brackets, idlers and belt path are clearly visible.

In the figure above, the motor and transfer case are not shown. They are located between the main support bracket and the secondary support bracket, and attached using current hardware to locations near the outer bearing. The body of the transporter is carbon steel, painted yellow.

The brackets are to be made from 3/8in carbon steel plate, with the main bracket forming an L-shape. The secondary bracket is to be a duplicate of the current design, however it will be adjusted for length.

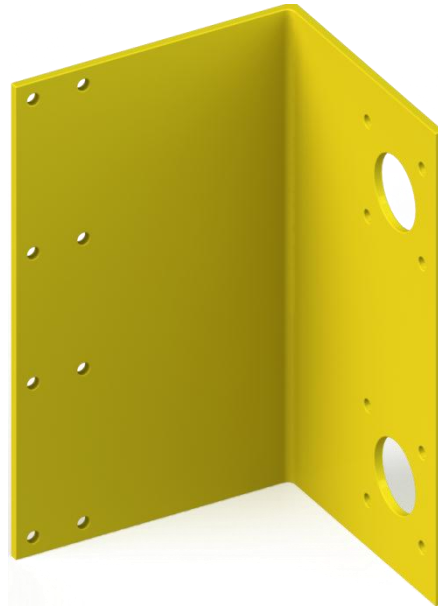


Figure 5: View of the Main Support Bracket



Figure 6: View of the Secondary Support Bracket

Since the current belt part number could not be obtained, the pitch and width of the belt were obtained from the picture below by visual comparison using the knowledge that the wheels have a 6in diameter.

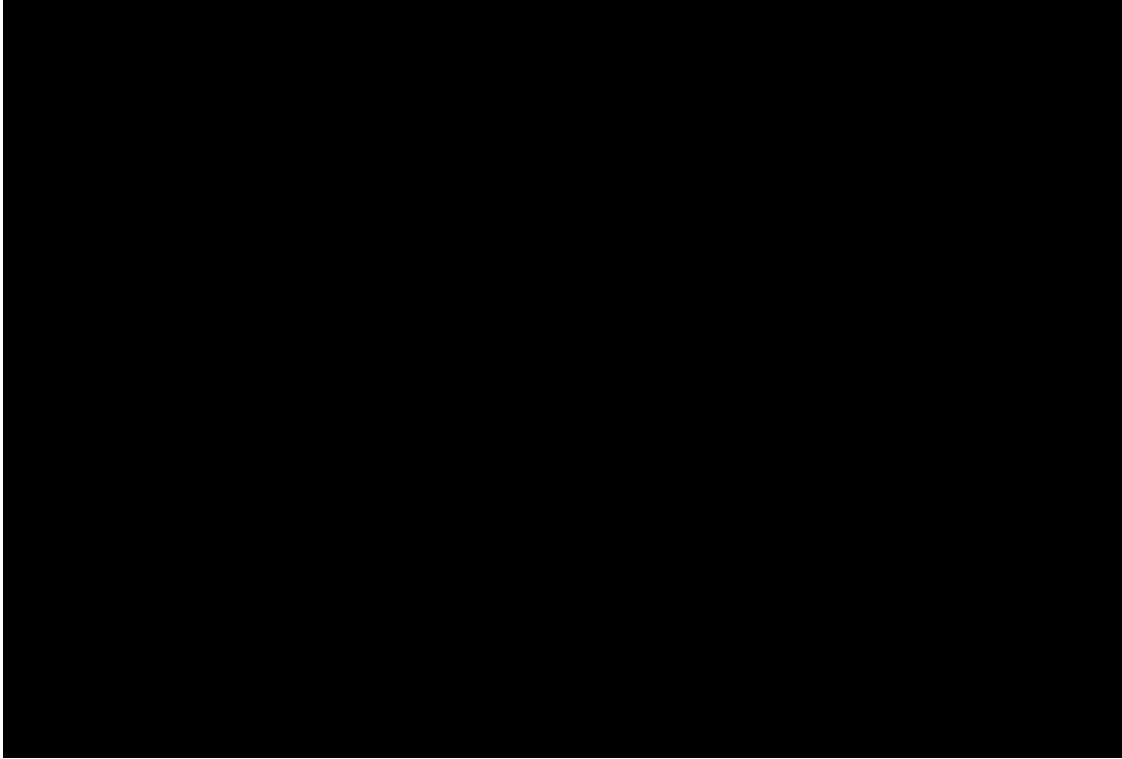


Figure 7: Picture of the rear axle and belt

The belt was determined have a pitch diameter of $\frac{3}{8}$ in (L-pitch belt) with a width of 1in. The distance between the centres of the front and back axles obtained the same way, indicating approximately 36in.

To ensure that the correct wrap angle of above 120° is achieve on all sprockets, two idlers will be required. The idler currently in use will be removed and relocated.

In addition, a cut-out will need to be performed on the side-stiffener to allow the belt to connect to the now-raised motor shaft. This will necessitate a reinforcement doubler on the outer section of the stiffener.

2.2.2 Motor Relocation Considerations

Using input parameters of 2hp, 200RPM, the default service factor of 1.8, equal sheave sizes and total sheave distances of 60in, Baldor's Power Transmission Wizard tool [4] recommended the use of a 3280-8M-20 HTR belt. According to Baldor's HTR Synchronous belt drive catalogue [5], these belts are equivalent to L-pitch belts. As the motor is rated for 2hp, it was assumed that the belt would be able to transmit any load transferred through it.

A model of the support bracket assembly was created in SolidWorks, and loaded using a belt tension of 100N as well as the sideways force generated by the transfer case. Although convergence could not be achieved, the figure below shows that the geometry of the location in question is identical to that currently being used without failure.

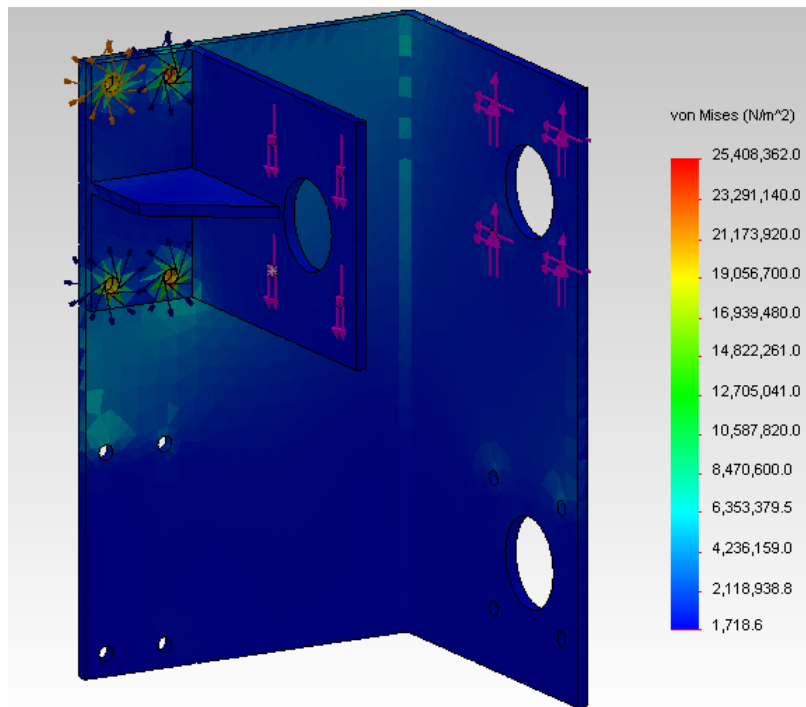


Figure 8: FEA Simulation of the Bracket Assembly. The solution did not converge, but the local geometry is identical to that currently being used.

Since a 1.5in section will need to be cut out of the lateral stiffener, a doubler will be required to retain the same stiffness. The stiffener cross-section was approximated as shown in the following figure:

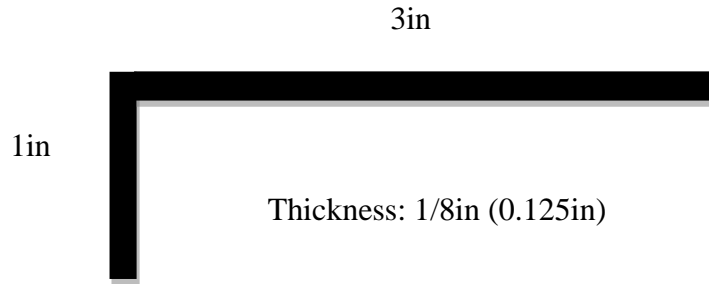


Figure 9: Approximation of the current lateral stiffener

This gives a cross-sectional moment of inertia of $2.25in^4$.

$$I = \sum \left(\frac{1}{12}bh^3 + bhy^2 \right)_n$$

$$I_{original} = \frac{1}{12} \left(\left(\frac{1}{8} \right) (3)^3 + (1) \left(\frac{1}{8} \right)^3 \right) + \left(\frac{1}{8} \right) (3)(1.5)^2 + \left(\frac{1}{8} \right) (1)(3)^2 = 2.25 in^4$$

Removing the outer 1.5in of the horizontal plate yields a moment of inertia of $1.27in^4$.

$$I_{cutout} = \frac{1}{12} \left(\left(\frac{1}{8} \right) (1.5)^3 + (1) \left(\frac{1}{8} \right)^3 \right) + \left(\frac{1}{8} \right) (1.5)(0.75)^2 + \left(\frac{1}{8} \right) (1)(3)^2 = 1.27 in^4$$

This requires a doubler at least 0.11in thick, such as a 1/8in plate, on the vertical section.

$$I_{doubler} = 2.25 = \frac{1}{12} \left(\left(\frac{1}{8} \right) (1.5)^3 + (1) \left(\frac{1}{8} + t \right)^3 \right) + \left(\frac{1}{8} \right) (1.5)(0.75)^2 + \left(\frac{1}{8} + t \right) (1)(3)^2$$

The doubler should be continued past the cut-out to ensure that the stresses are evenly distributed, and could be attached by bolting or welding.

3 Cost Analysis

The deciding factor for implementing a design is ultimately whether or not the implementing entity will profit from it, either directly financially or through processes streamlining. To perform these analyses, it is necessary to obtain values for the implementation cost and the projected cost savings. Both the Shaft Redesign and the Motor Relocation design sections will be evaluated, using the Payback Period Method and Present Worth Method.

3.1 Estimated Implementation

The material cost of each unit was estimated using different parts catalogues and online stores such as Acklands Grainger [6]. When manufacturing parts in-house, an average cost of \$45 per hour was estimated for each participating employee. It should also be noted that some parts were not considered to have a cost as those materials were already in service.

3.1.1 Shaft Redesign

The Shaft Redesign does not require any new materials or techniques, and thus will not incur extra material or time expenses over manufacturing a current-style flange. A one-time cost of **\$400.00** was therefore allocated to account for time lost and mistakes made during the familiarization process.

3.1.2 Motor Relocation

Table 1: Bill of Materials and Associated Costs for the Motor Relocation

DESCRIPTION	QUANTITY	UNIT	PRICE/UNIT	VENDOR	LOCATION
3048-8M-20 HTR belt, p/n 482800	1	ea	123.77 CAD [4]	Dodge Reliance (Baldor)	US
Sheave - Pitch diameter 5.093 in, Part # P32H150- 2517	1	ea	85.63 CAD [7]	Maska Pulleys	US
Four-bolt flanged unit (FY 1 3/16 TF)	3	ea	83.21 CAD [8]	Motion Industries	Winnipeg
REGULAR MILD Steel Rectangular Plate (10.85" L x 20" H x 1" W)	1	ea	325.00 CAD [9]	SCT Laser cut & Welding	Winnipeg
Drive Stub Shaft (16.54" L)	1	ea	406.00 CAD [10]	Standard Machine Works	Winnipeg
Bolt - 0.375" x 3" L	12	ea	1.50 CAD [6]	Acklands Grainger	Winnipeg
Total Cost			\$1,576.30		

Taking into account an estimate 18 work-hours and an additional 6 hours of downtime above the regular maintenance schedule to implement this modification, the table below shows the total estimate of the implementation cost.

Table 2: Cost Analysis Table

MATERIAL COST	\$ 1576.30
DOWN TIME (6hrs)	6* \$ 1550.00 = \$ 9300.00
LABOUR (2 employees)	18* \$ 45.00 = \$ 8460.00
TOTAL	\$ <u>19,336.30</u>

3.2 Annual Cost Savings

It was discovered that there were an average of four catastrophic failures per year. Each breakdown typically lasted four hours, with a loss of production and labour hours corresponding to the downtime.

Table 3: Annual Drive Shaft Failure Expenses

Average Catastrophic Breakdowns/year	4
Average down time per breakdown hours	4
Lost production per hour	\$1550.00
Labour cost per hour (2 personnel)	\$90.00
Total lost per hour (Production+labour)	\$1,640.00
Total Saving Cost Per Annum :	\$26,240.00

For the purpose of this cost analysis, a very conservative estimate was made that implementation of each proposed change would result in a cost savings of approximately 1/3rd of the shaft failure expenses: **\$8747.00**.

3.3 Payback Period Method

One of the simpler and more widespread methods of analysing the benefit of a disbursement of funds is to divide the implementation cost by the net annual savings [11]:

$$\frac{\textit{Implementation Cost}}{\textit{Net Annual Savings}} = \textit{Payback Period}$$

For the adoption of the new shaft design, the Implementation Cost was estimated at \$400.00 and the net annual savings were estimated to be \$8747.00. This modification therefore shows a payback period of:

$$\frac{\$400}{\$8747/\textit{yr}} = 0.046 \textit{ year}$$

The drive system modification's implementation cost was estimated at \$19 336.30, and its net annual savings were also estimated at \$8747.00 due to reduced maintenance downtime and better part standardization. Therefore, the payback period for this method will be:

$$\frac{\$19\,336.30}{\$8747.00/\textit{yr}} = 2.21 \textit{ year}$$

3.4 Present Worth Comparison Method

This method of cost analysis determines the overall value of the project (be it positive or negative) as a value in current money. This is done by converting future savings and disbursements into a current value using a chosen interest rate. For this analysis, the interest rate was chosen at 5% annually.

When an asset is assumed to have an infinite service life, the annual cash flow can be converted to a present value by dividing it by the interest rate:

$$\frac{A}{i} = \text{Current Value of Annuities}$$

Where A represents the annuity value, and i represents the interest rate.

The Capitalized worth can therefore be found by the equation:

$$\text{Capitalized Worth} = P + \frac{A}{i}$$

Where P represents the initial costs. [Engineering Economics, second canadian edition, by Riggs, Bedworth, Randhawa, Khan]

Therefore, given a first cost of \$400.00 and annual cost savings of \$8747.00, the current Capitalized Worth of the conversion to the new flange design was found to be:

$$(-\$400.00) + \frac{\$8747.00}{0.05} = \$174\,540.00$$

For the drive system modification, the initial cost of \$19 336.30 and the annual cost savings of \$8747.00 give a current Capitalized Worth of:

$$(-19\,336.30) + \frac{\$8747.00}{0.05} = \$168\,600.00$$

4 Conclusion and Recommendation

In summation, the modifications proposed in this plating line transporter drive train redesign are cost effective and can be implemented, pending further analysis in the case of the motor relocation. These modifications in this report satisfy all the needs of the customer. They have a single drive shaft design, provide higher reliability, and are implementable with standard parts and using the capabilities of the current RCM workforce.

The stress analysis of the flange and shaft sections has resulted in values well below the modified endurance strengths of each component even under twice the expected design load. This is achieved by stepping the flange and chamfering the diameter changes as well as including a larger bolting plate. The new flange and pin connection design is implementable without further analysis, individually or in conjunction with the motor relocation bracket.

The motor relocation bracket has been included as a reference concept. Due to security clearance issues, the majority of the original drawings were not available to the team. Dimensions were therefore inferred from pictures using visual comparison from known parts and a detailed analysis could not be conducted. Plate steel of the same thickness as is currently in service was used for the preliminary design. If this modification is desired, it is possible to simply change the dimensions to be accurate.

5 References

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6 Appendix A – Drive Shaft Design Logic

This appendix contains the logical progression of design insights leading to the preliminary design of the modifications proposed in body of the report.

6.1 Visual study

From studying the failure in the flange, it was determined that there are two primary modes of material failure; the flange breaking off in the sharp transition zone between the 44mm shank and 63mm circle that is coincident with the center points of the bolts, and the welded connection between the shank and tube.

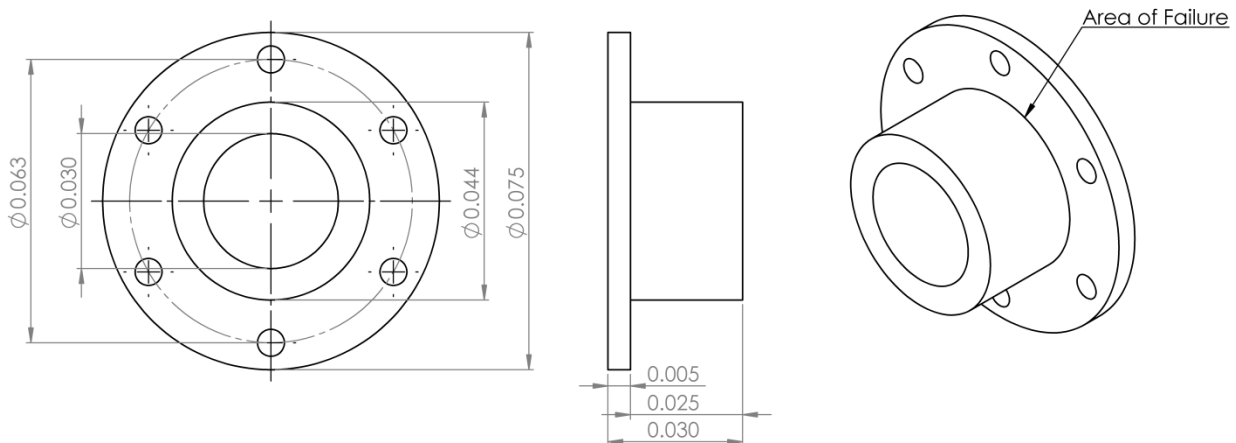


Figure 10: Drawing of existing flange

The issue in the flange stems from the sharp geometrical change which can cause large stress concentrations in torsional loading as well as bending in the case of misalignment. Plastic deformations can readily occur when transition fillets between geometrical changes are not used, which leads to crack propagation. Beach marks on the failed flanges indicate that the parts failed because of fatigue, however a picture was not made available to us.

The second mode of failure occurs between the shank and the tube in a 9mm plug weld that is filled in a single pass. Large weld pools are prone to defects similar in nature to castings, which include gas porosity, inclusions, blowholes, cracks and tensile stresses due to shrinkage [12]. When tensile stress is induced in the surface of a material subjected to fatigue loading, the life is significantly shortened as cracks easily propagate through the material [1]. When this effect is combined with the solid pathway between the weld and part, the crack can propagate into the solid bodies that were welded together. Figure 11 shows evidence of cracks passing through the weld and into the tube section.

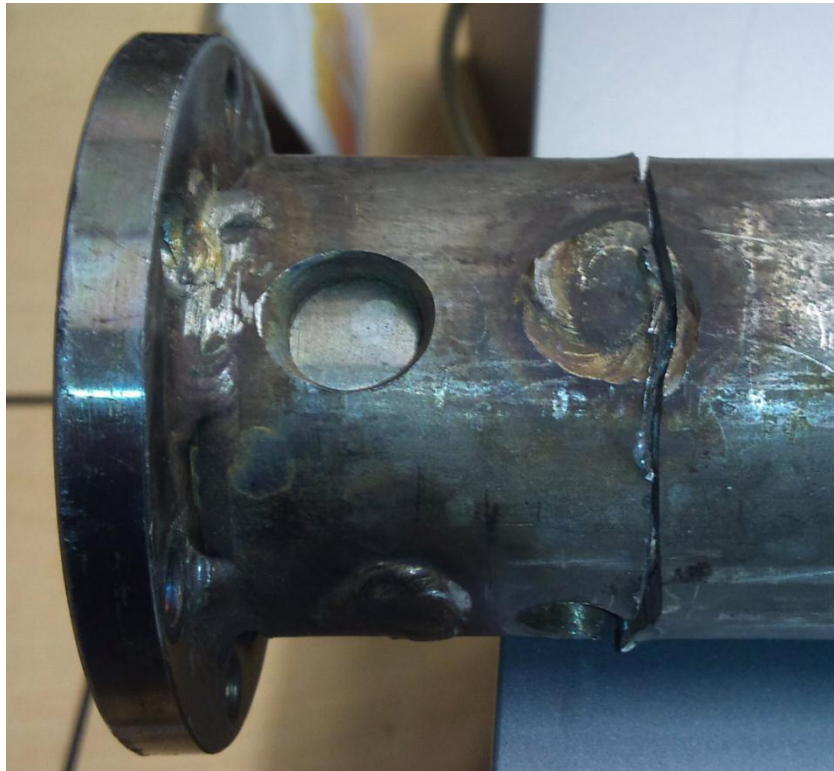


Figure 11: Broken shaft component showing crack propagating through welded area into the tube.

It is also noted that the cracks passes through the area of highest stress concentration at the interface where the weld fuses with the circular hole.

6.2 Insight from Lead Hands

Speaking with the lead hands that maintain and rebuild the plating line transporter when it fails has led to additional considerations for the design. Figure 11 shows weld beads along the perimeter of the interface between the flange and the tube. The figure also shows that there is no weld in the areas below the bolt holes, in contrast with the initial design. This deviation from the initial design is due to clearance issues between the weld bead and the bolt holes. This failure is not due to manufacturing, but from poor design choices.

7 Appendix B – Drive Shaft Stress Analysis

The initial problem from a dynamics point of view was given as a 1000kg mass accelerating a 0.5m/s^2 reaching a top speed of 1m/s. This translates to a torque on each of the 6in wheels of approximately 50Nm. Therefore, the power delivered from the drive axels will have to accommodate this torque. An additional factor of safety of 2 will be applied to ensure that the design will not fail under normal operating conditions. The calculations will have to be broken up into the flange, outer tube and pin.

7.1 Drive Shaft Analysis

The design stress for a torsional system subjected to fatigue loading has to be adjusted to account for different loading scenarios. This ensures that the material will reach its endurance limit. For 304 stainless steel, the endurance strength is 240MPa [2, 3]; therefore calculations have to be conducted modify this number. As well, a factor of safety of two will be applied to the system, increasing the torque to 100Nm.

7.2 Endurance Strength Calculation

The endurance strength is given as 240MPa, however it will have to be modified according to

$$S_e = S'_e K_a K_b K_c K_d K_e [1]$$

Where each K is a modifying factor; K_a is the Surface Factor, K_b is the size factor, K_c is the load factor, K_d is the Temperature factor and K_e is the miscellaneous-effects modification factor.

7.2.1 Surface Factor

The surface factor accounts for any imperfections in the machining process when compared to the highly polished samples. Both the flange and the pin have to be machined, whereas the tube

comes polished from the supplier. Therefore the modification factor is described by the equation below

$$k_a = \alpha S_{ut}^{\beta}$$

Where the ultimate strength the 304 stainless is 515MPa [2, 3], $\alpha=4.45$ and $\beta= -.265$ for machined parts, the modification factor is therefore $k_a = .855$ for the pin and flange.

7.2.2 Size Factor

Various equations are used to describe ranges of different outer diameters in inches and millimeters:

Inner Flange (D=1.811in)

$$k_b = \frac{.947}{1 - .016/D} = .9554$$

Outer Tube (D=2in)

$$k_b = .931 \left[1 + \frac{.014}{1 + D^2} \right] = .934$$

Pin<10mm, the size factor is 1

7.2.3 Load Factor

The drive shafts are a torsional system, therefore the strength will be highly impacted as the parts will fail in shear, described by the equation:

$$k_b = \alpha S_{ut}^{\beta}$$

Where $\alpha = .258$ and $\beta = .125$, $k_b = .5631$

7.2.4 Temperature Factor

The temperature factor only plays a role at severe deviations from standard conditions and is therefore not going to modify the strength of the material.

7.2.5 Miscellaneous Effects Factor

Since the system is not using shrink fits, under hydraulic pressure, is going to be fully annealed with no coatings and excellent corrosion resistance to the environment, these effects can be assumed negligible.

7.2.6 Final Material Strengths

Considering all the factors, the final material endurance strengths are summed up in the table below:

Table 4: Endurance Strengths of Driveshaft Components

	Endurance Strength [MPa]	99% Rel.
Pin	115.55	93.60
Flange	110.39	89.42
Shaft	126.22	102.24

7.3 Strength of the Shear Pin

The shear stress experienced by a pin can be calculated by:

$$\tau_{pin}^2 = \frac{4 T}{D_{shaft} \times \pi \times d_{pin}}$$

Where T is the torque, D_{shaft} is the outer diameter of the inside shaft, and d_{pin} is the diameter of the pin [13].

Torque was assumed to be 100 Nm, the diameter of the inner shaft must be 44mm to match current part geometry, the material was chosen to be 304 stainless steel, with an assumed tensile endurance strength of 240MPa, leading to an endurance shear strength of 90MPa.

The shear equation above can be manipulated to obtain the required minimum diameter of the pin:

$$d = \sqrt{\frac{4 T}{D_{shaft} \times \pi \times \tau_{pin}}} = \sqrt{\frac{4 \times 100Nm}{0.044m \times \pi \times 90MPa}} = 0.00515m$$

This pin however will be upsized to 9mm to ensure that the connection is strong.

7.4 FEA Results

A preliminary analysis was performed using the bolted connections to a fixed plate. The resultant forces in the flange plate were sufficiently far away that they did not influence the results near the pin hole. This justifies the simplification of the loading conditions to speed up the simulations. The simplified loading conditions are highlighted in Figure 12. It was also found that the stresses between the flange plate and shank were minor compared to those in the components around the pin.

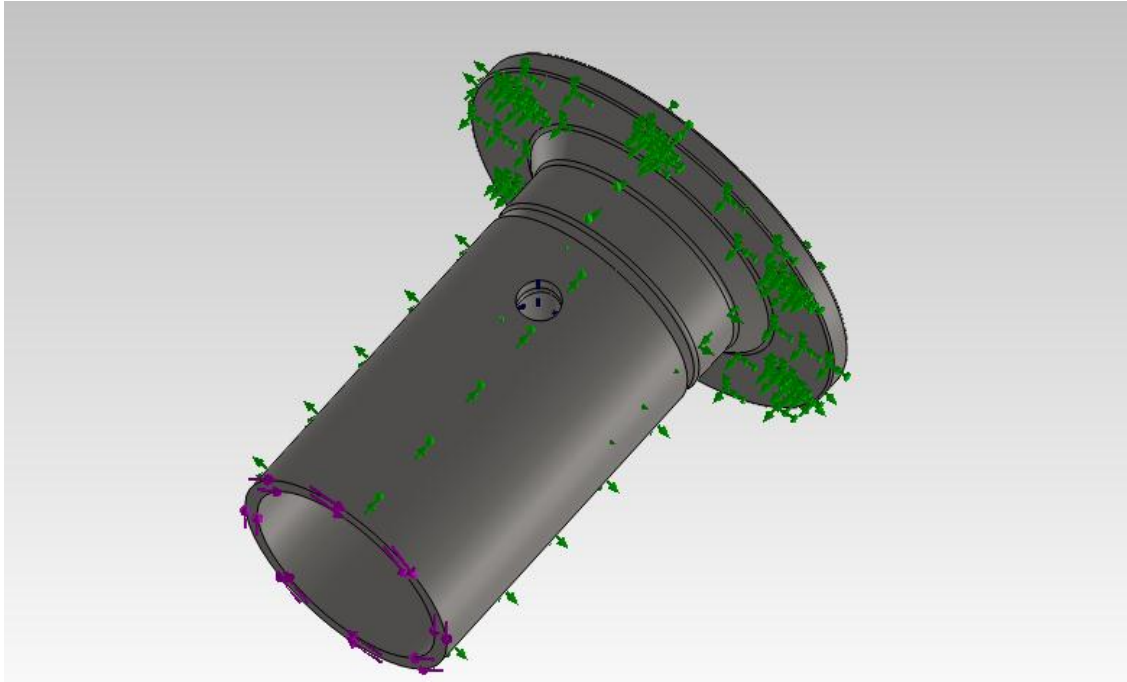


Figure 12: Screen cap from SolidWorks Simulator illustrating simulation conditions. Cylindrical surfaces are free to move in the Θ direction. Flange was constrained in the X direction and bolt holes were fixed. Joint is pinned and load applied 100Nm.

SolidWorks Simulation uses 10 node tetrahedral elements. Mesh controls were then applied around the pin holes until convergence was reached. The convergence data for the flange is given in Table 5, and for the outer tube Table 6. The data is then visualized in the Figure 13.

Table 5: Convergence data showing data from the stress bearing surfaces in the flange

Max Dimension of Element[mm]	Von Mises Stress[MPa]	Percent Difference	Number of Elements in Pin Area
1	51.8	0	889
0.5	79.2	10.46	2707
0.25	79.3	0.03	6927

Table 6: Convergence data for highest stress surfaces in the outer tube.

Max Dimension of Element[mm]	Von Mises Stress[MPa]	Percent Difference	Number of Elements in Pin Area
2	78.8	0	118
1	87.4	2.59	540
0.5	95.2	2.14	2136
0.25	97.6	0.62	8664
0.2	97.9	0.08	8900

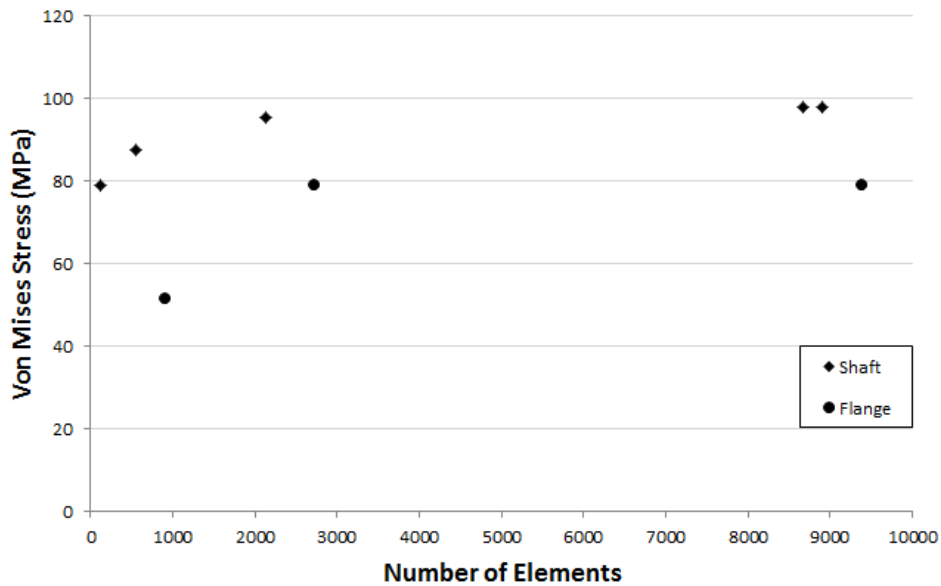
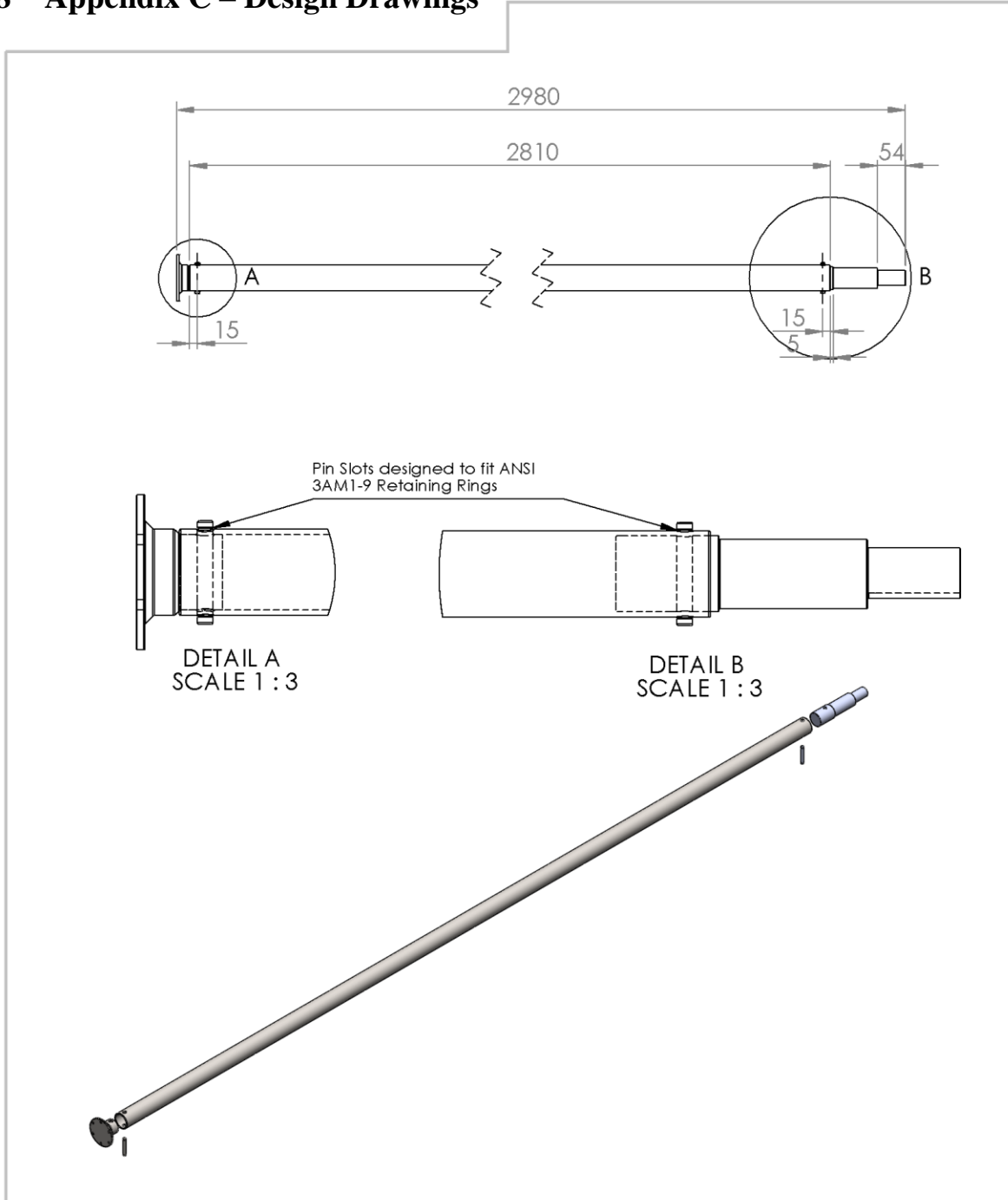


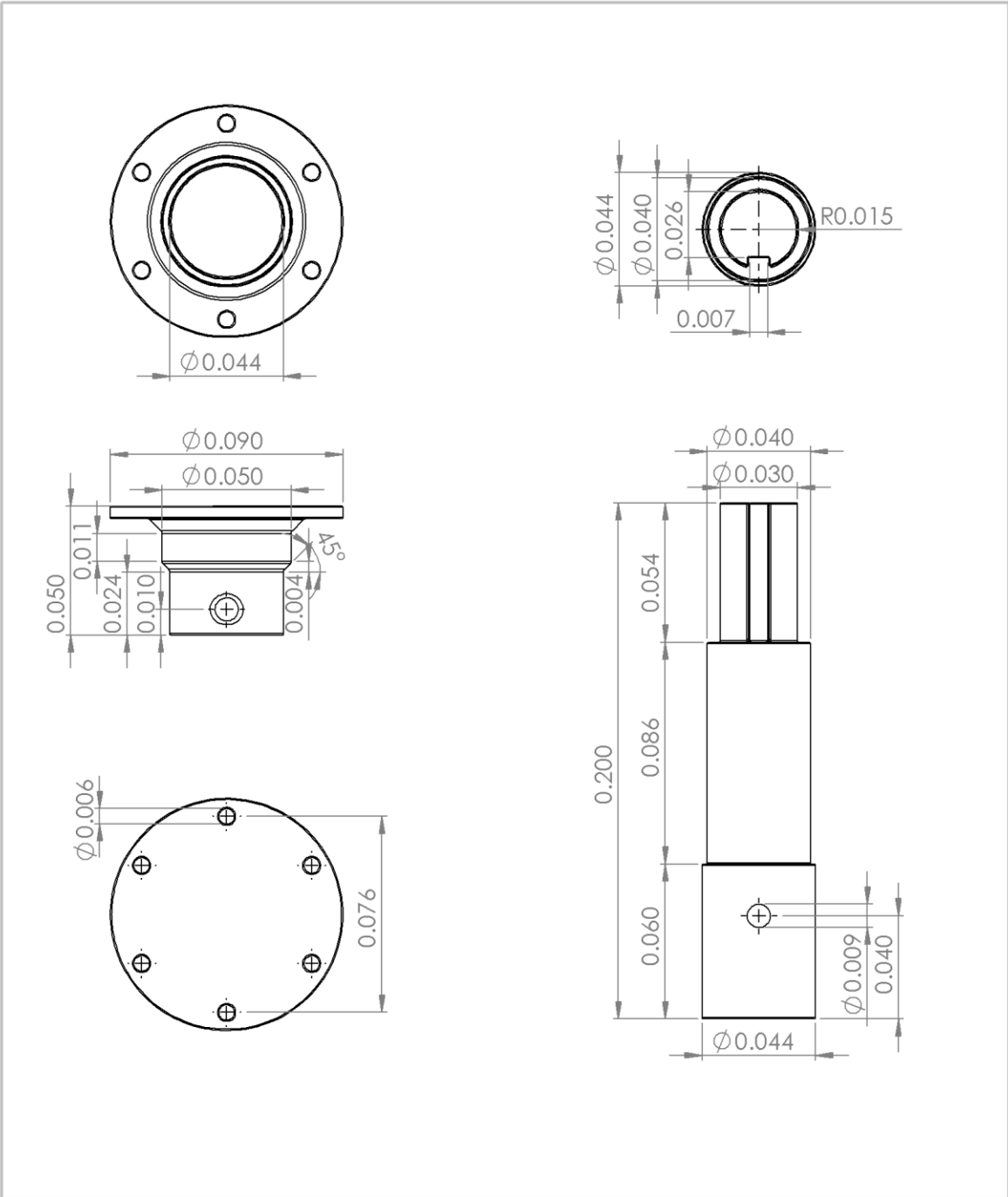
Figure 13: Convergence data from previous tables illustrating the convergent behaviour.

With the factor of safety of 2, the stress in the connection does not exceed those of the design limitations therefore the connection is not likely to fail under normal conditions. The detailed drawings of the drive shaft design are provided in Appendix C.

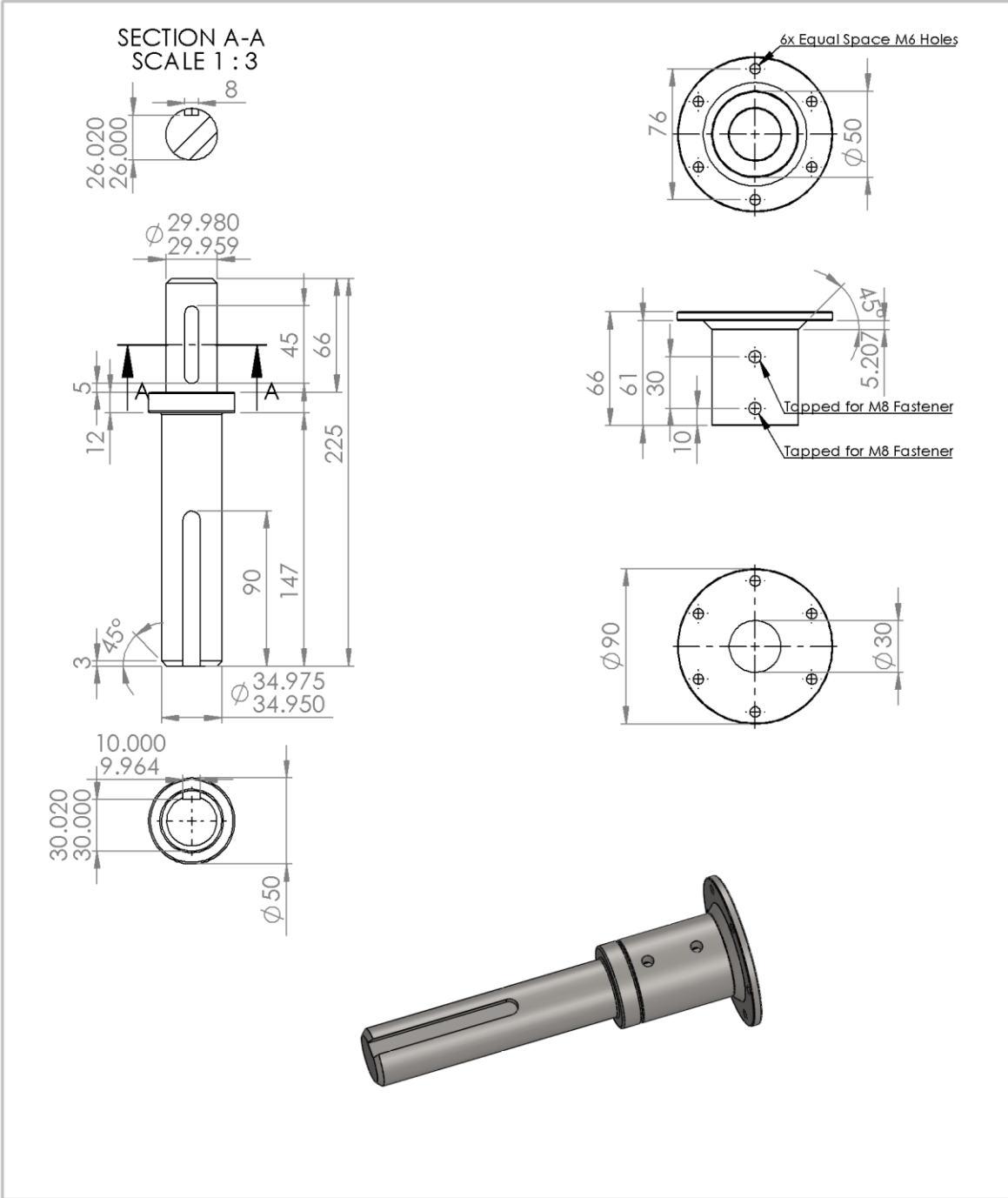
8 Appendix C – Design Drawings



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: POLISHED		FINISH:		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
NAME		SIGNATURE		DATE		TITLE:			
DRAWN: BRENDAN NAGY				NOV 25		Drive Shaft Assembly			
CHK'D						DWG NO. DS1			
APPV'D						A4			
MFG						SHEET 1 OF 1			
Q.A				MATERIAL:					
				304/304L SS					
				WEIGHT:					

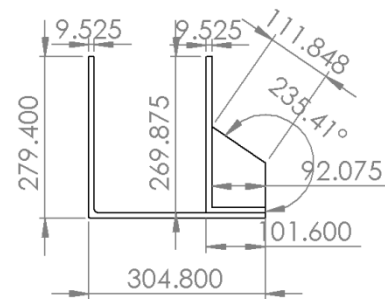
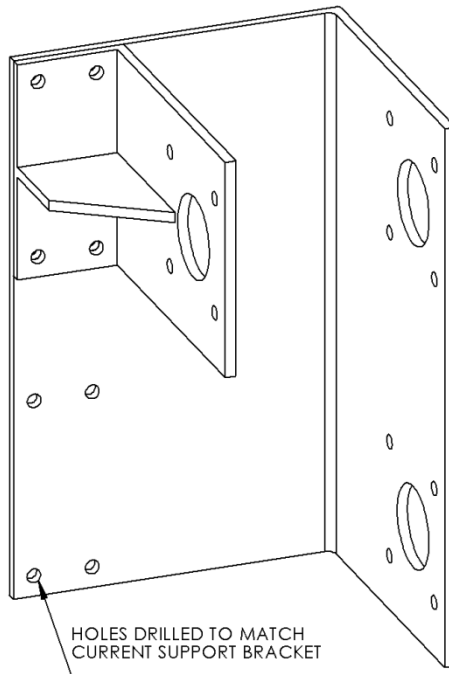
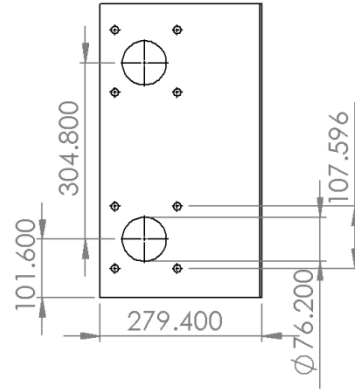
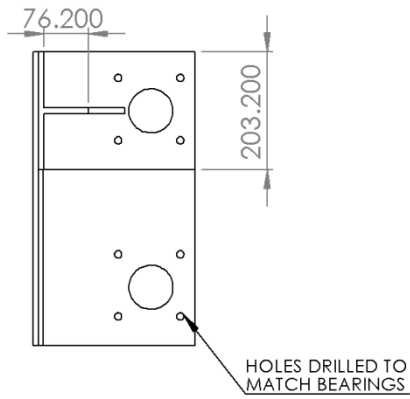


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH:				FINISH:		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
DRAWN: BRENDAN NAGY				SIGNATURE:		DATE: NOV 25		TITLE: DRIVE SHAFT COMPONENTS			
CHK'D:				MATERIAL: 304/304L SS		DWG NO. DS2				A4	
APPV'D:				WEIGHT:		SCALE: 1:2				SHEET 1 OF 1	
MFG:											
Q.A:											



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH:		FINISH: NONE SPECIFIED		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
DRAWN: BRENDAN NAGY		SIGNATURE:		DATE: NOV 25		TITLE: STUB SHAFT FOR TRANSPORTER			
CHK'D:		SIGNATURE:		DATE:		DWG NO. ST1			
APPV'D:		SIGNATURE:		DATE:		SCALE: 1:3			
MFG:		SIGNATURE:		DATE:		SHEET 1 OF 1			
Q.A:		SIGNATURE:		DATE:		MATERIAL: 304/304L SS			
WEIGHT:		SIGNATURE:		DATE:		A4			

DISCLAIMER: ALL DIMENSIONS ARE ESTIMATED DUE TO LACK OF INFORMATION ON TRANSPORTER DIMENSIONS



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH:		FINISH: PAIN ALL SURFACES PRIOR TO INSTALLATION TO PREVENT CORROSION		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
DRAWN: BRENDAN NAGY		SIGNATURE:		DATE: NOV 25		TITLE: MOTOR RELOCATION BRACKET			
CHK'D:		SIGNATURE:		DATE:		DWG. NO.: MRB1		A4	
APPV'D:		SIGNATURE:		DATE:		MATERIAL: MILD STEEL		SHEET 1 OF 1	
MFG:		SIGNATURE:		DATE:		WEIGHT:		SCALE: 1:10	
Q.A:		SIGNATURE:		DATE:		SCALE: 1:10		SHEET 1 OF 1	