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Conceptual Design for Replaceable Paddle for the Saltstone Production Facility (SPF) Mixer

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EXECUTIVE SUMMARY

The primary objective of this task was to provide a recommendation for a replaceable paddle that could be incorporated into the Saltstone Production Facility (SPF) process. The replaceable paddle concept was proposed given the time it took to replace the existing paddle designs was approximately a week of downtime. Two different types of replaceable paddles were considered, a split paddle and a replaceable tip paddle. The split paddle design was developed; however, a review of the drawings by maintenance personnel determined that changeout of the new design would take about the same time/effort as replacing the existing paddles. Five different replaceable tip designs were considered: side attachment, V neck, straight notch, angled notch, and dovetail. The recommended replaceable tip design was the angled notch, which includes a central hub with two angled notches, where the replaceable tips are secured to the hub using two ½" – 13 Unified National Coarse Threads (UNC) socket head cap screws (SHCS) (Figure 4-1). Design drawing for both the flat and helical paddle blades are complete and are included in Appendix A. A structural calculation (M-CLC-Z-00137) was performed on this design which determined the design is structurally adequate for normal or extreme loads that would be encountered in the SPF READCO™ mixer. This calculation is included in Appendix C. Based on the calculation, mechanical failure will not limit the life of the tip or tip attachment. The quantity of erosion will be the primary driver dictating tip replacement schedules.

This design was assessed for fitness of service and was determined to satisfy all the performance and maintenance requirements. Tools may need to be made so that any accumulation of grout on the shell can be removed during maintenance, during the removal or during replacement of the tips.

Erosion testing was performed. The Astralloy V and Ultimet materials were placed into simulant saltstone grout and tested to measure erosion rates versus speed. Testing showed that erosion rates are a function of a power law relationship and the velocity exponents were 2.229 and 2.814 for Astralloy V and Ultimet, respectively. This finding suggests that reducing the speed is highly beneficial in reducing the erosion rate and extending the life of wear components that have similar conditions. Based on these velocity exponent values, reducing the speed by 50 percent would reduce the erosion rate by a factor 4.7.

The following are recommendations from this work.

(1) Use the recommended replaceable tip design if the decision is to use paddles in the 1 through 6 paddle locations in the SPF READCO™ mixer.
(2) Reduce the SPF READCO™ mixer speed using a variable frequency drive. Reduction of speed will reduce erosion rates as well as the applied loads. SPF should determine the minimum speed based on processing experience using premix and water.

Future work.

(1) Procure hardened materials that could be used as the replaceable tips. Materials would be assessed using the Miller machine and corrosion testing using representative simulants.
(2) Show that EDM technology can be used to fabricate the replaceable tip design.
(3) Fabricate four pairs of flat and four pair of helical paddles and 12 replacement tips (six of each) for additional fitness of duty testing in spare SPF READCO™ mixer using Astralloy V. Testing should also be used to determine tool design necessary to remove grout buildup on the barrel.
(4) Measure the material properties of the Astralloy V paddles after fabrication to determine consistent material properties.

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LIST OF ABBREVIATIONS

EDM	Electric Discharge Machine
ELAWD	Enhanced Low Activity Waste Disposal
ELN	Electronic Laboratory Notebook
gpm	gallons per minute
MOC	material of construction
RPM	revolutions per minute
rs/min	reciprocating strokes per minute
SAR	Surface Abrasivity Response
SDU	Saltstone Disposal Unit
SHCS	Socket Head Cap Screws
SPF	Saltstone Production Facility
SRNL	Savannah River National Laboratory
SRR	Savannah River Remediation
TTR	Task Technical Request
UNC	Unified National Coarse Threads
w/p	water to premix ratio

1.0 Introduction

A primary component in the Saltstone Production Facility (SPF) is the 10-inch READCO™-Kurimoto co-rotating dual shaft continuous mixer. One of the salient features of the READCO™ mixer is that it is self-cleaning given the tight tolerance (gap) between the feed screws/paddles and the barrel. The READCO™ mixer operates at a single shaft speed of 238 revolution per minute (RPM). The READCO™ mixer blends dry premix (10 wt. % Portland cement, 45 wt. % blast furnace slag, and 45 wt.% Class F fly ash) with the decontaminated salt solution at a water to premix ratio (w/p) between 0.59 to 0.61. The dry premix is fed into feed screws (Figure 1-1, left side of drawing) that conveys the dry premix into the paddles where the salt solution is then added from the top approximately five to six paddles downstream of the feed screw/paddle interface (i.e., see "Water Port" in the figure), where the two streams are blended to make grout. Figure 1-1 does not show all the paddles/screws that are downstream of Paddle 10. The present targeted premix rate is 30 tons/hour and salt solution flow rate of approximately 78 gallons per minute.

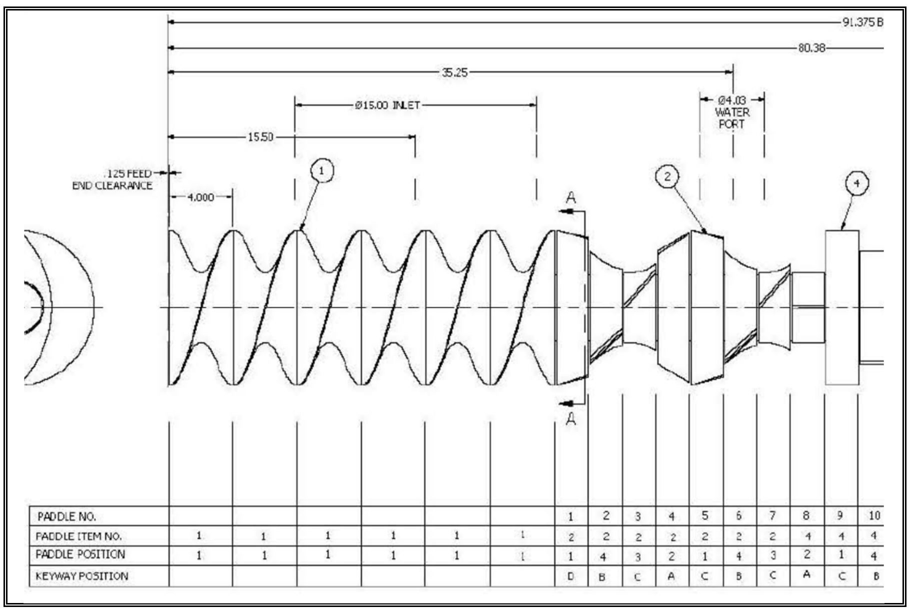


Figure 1-1. 2014 Side View - SPF Feed Screw and Paddle Arrangement on a Single Shaft [Ref. 1]

One of the primary issues with the READCO™ mixer is the erosion of the mixing paddles upstream of the liquid injection point, a region where wetting/mixing of the dry premix and spattering of (limited) salt solution (or water) as it enters the mixer first occurs [Ref. 2]. The grout in this region is very viscous due to the lack of liquid that mixes with the dry premix. This mixing results in an abrasive grout product that erodes the metal. Additionally, the grout in this region cannot be sufficiently conveyed forward given the original designs of the paddles, which were all flat faced. When the mixer is stopped, this region hardens, filling the gap between paddle and barrell with abrasive media. The region of erosion is at the tips of the rotating paddles. Over time, as erosion progresses on the paddle, the gap between the paddle and barrel increases, resulting in accumulated hardened grout in this region, thus creating an orifice. As this orifice grows, it can reduce the rate at which the premix can be processed by the READCO™ mixer, hence causing the premix to backup or causing a reduction in the overall processing rate.

As part of the Enhanced Low Activity Waste Disposal (ELAWD) program, a systematic approach in understanding different paddle configurations and erosion rates was executed. In 2012, the first set of paddles/feed screws and new mixer were installed, where Paddles 2 through 28 were flat paddles (item 4 in Figure 1-1) and Paddle 1 was an helical paddle (Item 2 in Figure 1-1) [Ref. 3]. Paddle 1 was configured such that the start of the helix was aligned with the discharge of the feed screw, for continuity. All the paddles were fabricated from Astralloy V, in place of stainless steel. Astralloy V was selected given it has good erosion characteristics [Ref. 4] and is machinable. The feed screws (Item 2 in Figure 1-1) were made of stainless steel and the tips on the augers were hardened with Stellite 12. This material selection has been maintained. The key dimensions of the feed screw and paddles and the clearances when placed into the READCO™ are provided in Table 1-1, there are no tolerances specified for the dimensions. This paddle configuration was modified in December 2014, where the first seven paddles were helical, and the paddles keyed to minimize the discontinuity between paddles. This configuration was used until December 2017. In January 2018, six helical paddles were replaced with three feed screw sections (Feed Screws 7, 8 and 9) and the remaining paddle configuration was maintained [Ref. 3] as shown in Figure 1-2, hence a total of nine feed screws are used to deliver the premix. The remaining seventh helical paddle [Item 5 in Figure 1-2] was aligned with the discharge of the feed screw for continuity. The liquid in this case enters the final feed screw section and helical paddle section of the READCO™ mixer.

Table 1-1. Original Dimensions and Clearances For 10-Inch SPF READCO™ mixer [Ref. 3]

Component		Dimension (inches)
Paddle	Diameter	9.750
	Width	2.000
Feed Screw	Diameter	9.750
	Width	4.000
Paddle to Paddle or Paddle to Feed Screw		0.095 to 0.155
Feed Screw to Feed Screw		0.020 min
Screw & Paddle to Barrel Radius (gap)		0.110 to 0.140 (nominal 0.125)

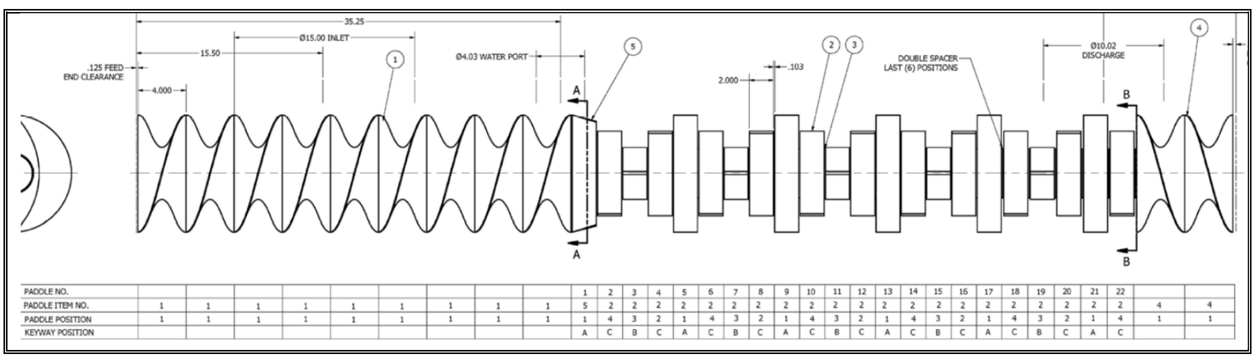


Figure 1-2. 2018 Side View - SPF Feed screw and Paddle Arrangement on a Single Shaft [Ref. 3]

As part of the ELAWD program, periodic inspections were performed measuring the distance between the tips of the paddle to the inside of the barrel. The inspection date, the quantity of salt solution processed between inspection dates, the cumulative amount of salt solution processed, and the average distance between the paddle tips and barrel are provided in Table 1-2.[1] Up to four paddle tips to barrel measurements were made, two on each tip. A total of 3,612,265 gallons[1] of salt solution was processed using the flat paddle, though no final gap measurements were obtained. Independent of the paddle selection, paddle locations 4 and 5 resulted in the greatest wear. Based on wear (inches) per gallon of salt solution processed that can be derived from the information in Table 1-2, the helical paddles had lower rates at paddle locations

[1] Satish Shah, SRR, Melter & Saltstone Engineering, provided Excel spreadsheet of the collected ELAWD erosion data for both the flat and helical paddle configurations in the READCO™ mixer. This Excel file is saved in the ELN [Ref. 5].

4 and 5 as compared to the flat paddles, but saw a higher rate for paddle locations 6 and 7. The calculated wear rates are included in the Electron Laboratory Notebook [Ref. 5]. In general, as more gallons of salt solution were processed, the rate at which the gap grew decreased. The maximum gap reported in the Excel spreadsheet provided by SRR is 1.688 inches. When the existing paddles require replacement, this maintenance activity takes approximately a week to replace the overhaul the READCO™ mixer.

Table 1-2. Measured Gap (inches) Between Paddle and Barrel in READCO™ Mixer

Inspection Date	Flat Paddle (ELAWD Installed) (9/1/12 - 9/30/2014)			Helical Paddle (Change Out: 12/2014) (1/28/2015 - 12/20/2017)			
	1/24/2013	4/9/2013	5/12/2013	9/8/2015	3/17/2016	9/19/2016	1/24/2018
Processed Salt Solution (gallons)	1,269,571	482,469	189,664	828,128	420,470	1,085,540	185,780
Cumulative Sat Solution (gallons)	1,269,571	1,752,040	1,941,704	828,128	1,248,598	2,334,138	2,519,918
Paddle 1 (inches)							
East Side Paddle	0.125	0.125	0.125	0.111	0.125	0.125	0.125
West Side Paddle	0.125	0.156	0.156	0.176	0.219	0.172	0.156
Paddle 2 (inches)							
East Side Paddle	0.125	0.125	0.125	0.120	0.125	0.141	0.156
West Side Paddle	0.312	0.313	0.375	0.381	0.594	0.438	0.333
Paddle 3 (inches)							
East Side Paddle	0.125 - 0.250	0.125	0.250	0.380	0.438	0.438	0.594
West Side Paddle	1.000	1.000	1.000	0.597	0.812	0.750	0.854
Paddle 4 (inches)							
East Side Paddle	0.687 -1.125	1.250	1.375	0.543	0.609	0.828	0.844
West Side Paddle	1.062	1.062	1.062	0.332	0.469	0.875	0.875
Paddle 5 (inches)							
East Side Paddle	0.375 -1.125	1.250	1.375	0.392	0.563	1.453	0.875
West Side Paddle	0.187 - .250	0.250	0.250	0.149	0.265	0.359	0.609
Paddle 6 (inches)							
East Side Paddle	0.125	0.250	0.250	0.236	0.266	0.516	0.656
West Side Paddle	0.187	0.187	0.187	0.142	0.187	0.156	0.219
Paddle 7 (inches)							
East Side Paddle	0.125	0.125	0.125	0.189	0.250	0.422	0.656
West Side Paddle	0.125	0.125	0.125	0.158	0.187	0.156	0.203

In January 2019, after 511,287 gallons of salt solution was processed using the feed screw configuration, an inspection was performed.[2] The average gap between Feed Screws 7 through 9 and the barrel was 1/8" to 3/16" as provided by SRR. Feed Screw 9 showed signs of flattening at the point where the liquid entered and three gouges. No information was provided on the helical paddle adjacent to Feed Screw 9. Future inspections are planned to further assess the performance of the feed screw configuration.

Due to the erosion observed on the paddle configurations tested during ELAWD, a Task Technical Request (TTR), M-TTR-Z-00017 [Ref. 6] was issued from SRR to Savannah River National Laboratory (SRNL) to develop a paddle design where paddles upstream of Paddle 6 do not have to be removed during maintenance

[2] Satish Shah, SRR, Melter & Saltstone Engineering, provided a power point presentation, "Saltstone Mixer Paddle Wear", 2/06/2019 with data. This presentation is saved in the ELN [Ref. 5].

activities. The task identified in the TTR are as follows and will be described in more detail in subsequent sections in this report.

(1) Assessment of material for use (MOC for current paddle is Astralloy V, refer to SRNL-L3100-2013-00047 for Recommended Acceptance Criteria for Paddle MOC). Material will be assessed for erosion using the Miller machine.
(2) Engineering 'replaceable paddle' design drawings.
(3) Calculations to support structural integrity of the design of 1 year of continuous operations (TBD). Calculations to performed per manual E7.
(4) Procure materials and fabricate paddles for testing.
(5) Evaluate selected design for fitness for service in the mixer

2.0 Design and Testing Efforts

The effort to support the tasks stated in the TTR were broken into five activities as stated below. Each of these tasks will be discussed in more detail in their separate sections.

(1) Designs of Replaceable Paddles
(2) Fabrication of Replaceable Paddles
(3) Fitness of Duty
(4) E7 Calculation to Support Paddle Selection
(5) Erosion Testing

2.1 Design of Replaceable Paddles

The detailed physical dimensions of the helical and flat paddles were not provided by the vendor. To obtain the dimensions, drawings [Ref. 3,7] and actual paddles provided by SRR were used. Salient features such as outside paddle diameter, shell inside diameter, shaft diameter, and locations of the key cutouts were obtained from drawing. The other dimensions, paddle inside diameter, paddle key cutout dimensions, paddle shape, paddle tip thickness, and twist angle for the helical paddle were obtained from the physical samples. The key dimensions of the flat and helical paddle designs are provided in Table 2-1.

Table 2-1. Key Physical Dimension of READCO™ Flat and Helical Paddles

Dimensions	Flat	Helical
Paddle Width (inches)	2.000	2.000
Tip Width (inches)	0.250	0.250
Concave Radius (inches)	6.625	6.625
Tip to Tip (inches)	9.750	9.750
Twist angle (degrees)	0.0	27.5
Inside Hub Diameter (inches)	3.250	3.250
Maximum side to side (inches)	4.528	4.528
Four Hub Notched Centers Relative to the Centerline of Tip to Tip (degrees)	0, 90, 135, 225	0, 90, 135, 225
Notch Width (inches)	0.750	0.750
Notch Height (inches)	0.250	0.250

Two different replaceable paddle designs will be considered; (1) a replaceable paddle where two parts are secured around the shaft using bolting material, referred to as the split paddle design and (2) a solid central hub secured to the shaft with replaceable tips that are secured using bolting material, referred to as the replaceable tip design. The visual details are provided in Section 3.1.

The objective of this task is to provide final construction drawings of flat and helical replaceable paddles that satisfies the fitness of duty requirements and are structurally analyzed using Manual E7, Procedure 2.31 [Ref. 8] to be suitable for service.

2.2 Fabrication of Replaceable Paddles

The original objective of this task was to fabricate one full scale paddle set of the selected design out of Astralloy-V, either a set of flat or helical paddles. This objective was changed as the project progressed. Fabrication of 3-D plastic models of the full-scale paddle designs was deemed acceptable by SRR.[3] 3-D models were used to assess the various proposed designs by Engineering and Maintenance personnel to support selecting the final design. 3-D modeling was also used to make a mixer jig that contained 4 pairs of paddles to assess fitness for duty requirements. If a paddle was to be fabricated from Astralloy-V, the material properties [Ref. 9] must be satisfied. These Astralloy-V properties are presently prescribed by SRR when procuring fabricated paddles from READCO™. Any variance in the properties would be evaluated by SRR and SRNL.

2.3 Fitness of Duty

The fitness of duty objectives includes dimensional and physical requirements so that the final design satisfies the requirements that the replaceable paddles perform in the same manner as existing paddles and the paddles downstream of paddle 6 are not required to be removed when performing maintenance on paddles 1 through 6. Table 2-2 provides the list of performance and maintenance requirements for the final replaceable paddle design.

Table 2-2. Fitness of Duty Requirements for Replaceable Paddle

Performance	Maintenance
Dimensionally the same as the paddles provided by READCO™.	Ability to remove the replaceable paddle without moving the paddles downstream of paddle 6.
Co-located paddles do not interfere with each other during rotation.	Ability to access and clean cured grout that has accumulated on the barrier. In this case, access will be assessed.
Structurally suitable for the application. Suitable for at least one year of operation.	Ability to access bolting locations.

2.4 E7 Calculation to Support Paddle Selection

Manual E7, Procedure 2.31 [Ref. 8] was used to assess the structural integrity of the selected replaceable paddle design. The calculation uses the base material, Astralloy V and the bolting material identified in the selected design for analysis. This calculation also addresses if the components will last more than one year of continuous service. The calculation provides additional engineering requirements if identified. The calculation includes approval from the originator, verifier, checker, and design authority, and hence constitutes an approved calculation. The calculation is attached to this document (with signatures removed) as Appendix C.

[3] Email communication with Charles Whitehurst, "RE: SCHEDULE UPDATE", January 22, 2019 and re-affirmed on email dated January 29, 2019. Emails will be saved in the ELN [Ref. 5].

2.5 Erosion Testing

The primary purpose of this erosion testing was to determine how speed could impact the rate of erosion. It was concluded in SRNL-STI-2014-00406 [Ref. 10] that the READCO™ mixer speed could be reduced by a factor of two without impacting product quality (of grout entering the SDU vault) due to the additional mixing that occurs in the grout hopper and transfer line prior to being discharged into a Saltstone Disposal Unit (SDU). This testing is to determine if speed could be a contributor to erosion rate and the potential impact/saving through the reduction of speed and extension of service life of the wearing components.

For this activity, the baseline material Astralloy V was tested with other commercially available wear materials identified by SRNL as potential candidates that could be used in this wearing application. Due to the availability and long lead times of such materials, it was agreed by SRR that SRNL test what was available in stock. The available materials were Astralloy V and Ultimet. The Astralloy V wear samples were obtained from paddles provided by SRR. The Ultimet wear samples were obtained from an Ultimet plate [Ref. 11] maintained by SRNL, which was the same material tested in SRNL-STI-2012-00379 [Ref. 4] and is the plate material that has not been work hardened as discussed in the referenced document.

The two principle wear mechanisms for the READCO™ mixer were identified as three body abrasion and erosion at the paddle transition [Ref. 4]. This reference also identified ASTM G75 [Ref. 12] as the most representative wear testing method that simulates wear in the SPF READCO™ mixer. ASTM G75 specifies use of the Miller Machine to perform the Miller Test. The Miller Machine contains four erosion troughs and Figure 2-1 provides a cross-sectional view of a single trough. The basic operations for obtaining erosion data is to set the block lifting cam consistent for all the troughs, attach the wear block (coupon) to the wear holder, attach the wear holder to the reciprocating arm, load slurry (approximately 220 mL) into each trough, lower the wear block into the slurry and engage the drive mechanism to provide a horizontal reciprocating harmonic motion. The function of the lifting cam is to raise the reciprocating arm to refresh the slurry between the wear block and lap. There is an additional five-pound weight attached to each reciprocating arm. The reciprocating stroke length in one direction is eight inches, therefore one revolution is 16 inches. Detail operations of the Miller machine and can be found in their operating manual [Ref. 13] or ASTM G75.

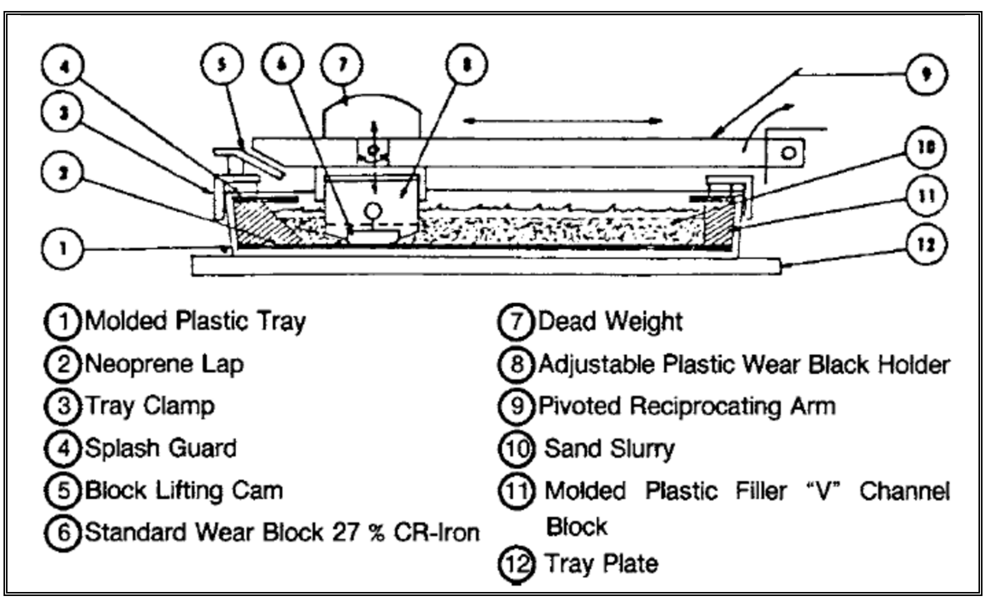


Figure 2-1. Miller Machine Slurry Trough Cross-Section [Ref. 12]

The wear blocks were cut using an electric discharge machine (EDM) located at SRNL. The wear blocks were cut to the dimensions shown in Figure 2-2 and the bottom of the wear blocks (tapper ends) were further prepared by using 600 grit sand paper, pulling the wear block in one direction when sanding. The wear blocks were cleaned as specified in the ASTM G75 prior to weighing activities.

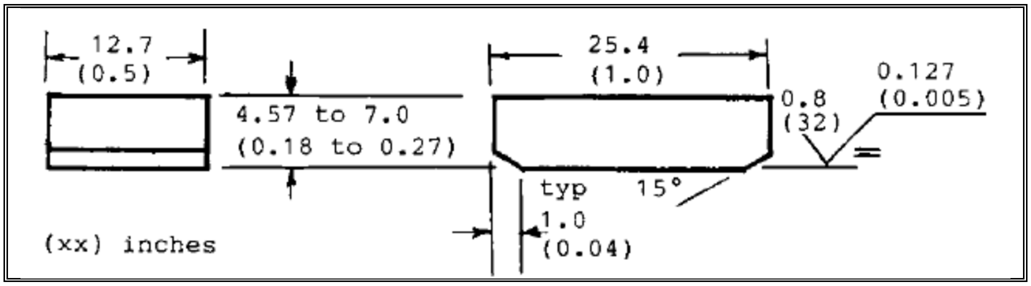


Figure 2-2. Wear Block Dimensions [Ref. 12]

ASTM G75 specifies a reciprocating rate of 48 reciprocating strokes per min (rs/min) and mass loss measurements are made at two-hour intervals up to a total time of six hours. The mass loss rate data is fitted to a power law function (equation (1)) and the Surface Abrasivity Response (SAR) number is calculated (equation (2)). The higher the SAR value, the more abrasive the slurry for a specific material. The mass loss rate is determined by taking the derivative of equation (1) and used to calculate the SAR (equation (2)). In this test, the ASTM sand (150 grams of sand and 150 grams of DI water), two 27 wt.% Cr-iron, one Astralloy V, and one Ultimate wear blocks were tested to obtain additional operational experience with the instrument. This data is reported; but is not directly representative of the actual conditions observed in the READCO™. Obviously, this simulant is not representative of saltstone grout. The ASTM sand particle size distribution is sieved material, between 70 to 50 mesh or 270 to 368 microns respectively. This sand is much larger than the premix materials SRR uses in the SPF process, where the volumetric mean particle size of the largest component, Fly Ash, is around 50 microns [Ref. 14]. The SAR data for various materials tested by SRNL using simulant salt supernate and premix are reported in SRNL-STI-2012-00379 [Ref. 4], including the Astralloy V and Ultimet plate tested in this effort. In SRNL-STI-2012-00379, the Ultimet had a lower SAR value and lower weight loss as compared to Astralloy V.

$$M = A \cdot t^B \tag{1}$$

where: M = cumulative mass loss (mg)
t = time (hr)
A = curve fit (mg/hr^B), and
B = power law fit (unitless)

$$SAR = 18.18 \cdot A \cdot B \cdot t^{(B-1)} \cdot \frac{7.58}{SG_{WM}} \tag{2}$$

where: SAR = Surface Abrasivity Response (unitless), and
SG_{WM}= Specific Gravity of wear block material (unitless)

The primary purpose of this erosion testing is to use the Miller Machine to assess the influence of speed relative to erosion rates. The Miller Machine reciprocating rate can be adjusted between speed settings of 0 to 10 using a dial as shown in Figure 2-3. To relate the rs/min to dial speed setting, baseline data was obtained from five-minute trails with Astralloy wear coupons placed into troughs containing water and various speed settings and the number of reciprocating strokes recorded. This relationship was used to target the rs/min for the ASTM G75 and this effort.

Figure 2-3. Miller Machine Speed Control

It is a known that during Miller testing, both erosion of the wear block and attrition of the particles in the slurry occurs. In general, as the test progresses, the wear rate decreases due to the attrition of particles. This was observed in SRNL-STI-2012-00379 for both the Astralloy V and Ultimet, where the mass loss decreased for each successive two-hour measurement. To help minimize the influence of attrition, the slurry was refreshed after a set number of reciprocating strokes. In this task, the maximum speed (rs/min) was used as the basis to determine the number of strokes in a two-hour period, at which time the slurry was replaced, for a total of three slurry batches for this condition. A second point at 2/3rd the maximum speed was used, and the slurry replaced every three hours, for a total of two slurry batches, maintaining the number of strokes for a batch. Finally, a speed 1/3 of the maximum was used and the slurry batched one time. When new batches of slurry are added, the mass of the wear block was measured prior to restarting the test. Each speed settling is targeted for a total of six hours of test time. The total number of strokes and dial setting are inputs into the Miller machine; hence both stroke reading and time were recorded for each test dial setting. The speed of the reciprocating arm can be determined using equation (3). The Miller machine has four troughs, hence two wear blocks of Astralloy V and two wear blocks of Ultimet were used and their respective data averaged.

$$V_{ra} = \frac{TSL \cdot \omega_{rs}}{12 \cdot 60} \tag{3}$$

where: V_{ra} = velocity of reciprocating arm (ft/s)
TSL = total stroke length (inches), and
ω_{rs} = average reciprocating strokes per min (rs/min)

For comparison purposes, the maximum tip speed of the READCO^{TM} mixer can be determined using equation (4). The rotational speed of the READCO^{TM} mixer is 238 RPM [Ref. 15]. The radius is provided in Table 2-1 and is ½ that of the tip to tip distance.

$$V_{TS} = \frac{2\pi \cdot \omega \cdot R_{paddle}}{12 \cdot 60} \tag{4}$$

where: V_{TS} = Tip speed of SPF READCO^{TM} paddle (ft/s)
ω = rotational speed of mixer shaft (RPM), and

R_{paddle} = radius of paddle (inches)

The mass loss erosion rates for the various speeds is determined using equation (5). The volumetric loss erosion rates can be determined by dividing the mass loss erosion rates by their respective density.

$$\dot{m}_{LM,i} = \frac{\sum_{i=1}^{n} m_i}{\sum_{i=1}^{n} t_i} \tag{5}$$

where: $\dot{m}_{LM,i}$ = mass loss erosion rate $\left(\frac{mg}{hr}\right)$
m_i = mass loss between slurry batches (mg), and
t_i = testing time of batch (hours)

As stated above, the total number of stokes is specified for a batch of grout. The speed dial setting was fixed for the duration of testing. During the test, the number of strokes and time were measured to determine the rs/min. During the six hour runs, the actual speed slightly increased over time, even though the dial setting was fixed, and an average speed was then determined which was then used to determine the time of testing. The average rs/min value determined were then used to determine the duration of the test. These calculated times were used to determine the erosion rates.

Erosion rates have been proposed to exhibit an empirical power law relationship with the erosive particle velocity [Ref. 16]. In the case of the Miller machine, the particles are stationary and the wear block moves. Literature reports that the velocity exponent varies between 0.34 to 4.83 depending upon the particles, material properties, and condition of the test. The higher the velocity exponent, the more sensitive erosion rate is to speed. The mass erosion rate versus speed was plotted and fitted with a power law relationship, equation (6), with the results compared to each other. The same data reduction approach was used for volumetric loss erosion rates. JMP Pro Version 11.2.1 was used to generate the statistical analysis of data using the Nonlinear modeling platform in JMP [Ref. 17].

$$\dot{m}_{LM} = C \cdot V_{ra}^{D} \tag{6}$$

where: $\dot{m}_{LM}$ = mass loss rate $\left(\frac{mg}{hr}\right)$
C = curve fit $\left(\frac{mg}{hr} \cdot \left(\frac{s}{ft}\right)^{D}\right)$, and
D = power law fit (unitless)

To support this testing, a salt solution simulant based on the average of the third quarter 2017 [Ref. 18] and first quarter 2018 [Ref. 19] Tank 50 samples was used. The composition for one liter of the Tank 50 salt simulant is shown in Table 2-3, where species greater than one weight percent by mass in the averaged data as well as phosphate [Ref. 20] were considered as part of the salt simulant. The density and solids content of the resulting simulant were measured and used for batching purposes, these values are listed in Table 2-3. The chemical composition was not measured.

Table 2-3. One Liter of FY17/FY18 Tank 50 Salt Simulant

Species	Base Chemicals	Mass Addition (g)/Units
Aluminate ($Al(OH)_4$)	$Al(NO_3)_3$-$9H_2O$	63.37
Carbonate (CO_3)	Na_2CO_3	27.11
Free Hydroxide (OH^-), 50 wt. %	NaOH	201.86
Nitrate (NO_3)	$NaNO_3$	116.62
Nitrite (NO_2)	$NaNO_2$	39.59
Phosphate (PO_4)	Na_3PO_4	0.37
Sulfate (SO_4)	Na_2SO_4	5.60
DI Water	H_2O	770.58
Density	1.2257	g/mL
Solids fraction	0.2672	g-solids/g-supernate

The saltstone grout used in this erosion testing was modified to reduce the effect of particle attrition, to provide more mass of solids so measurable wear data could be obtained, and for the grout not to set during testing. The w/p ratio in the facility is between 0.59 to 0.61 with no admixtures added. In this case, the w/p was reduced to 0.45. Additionally, 0.0125 grams of Daratard 17 per gram of premix was added so the grout would not set in the Miller trough when performing the six-hour erosion run. This quantity of Daratard was used in all batches. The premix materials were provided by SRR; (1) The SEFA group, 3Q18 Fly Ash, Lot# 2018-IR-05-1297, (2) LeHigh, 2Q18 Slag, Lot# 2018-IR-05-1299, and (3) Holcim Holley Hill Plant, Type I/II Portland cement, Lot # 2018-IR-05-1666 and combined as 45, 45, and 10 wt.% respectively. The 0.45 w/p batch for each test is provided in Table 2-4. During batching, the Tank 50 salt solution was added to the mixing vessel, followed by Daratard 17 and finally the premix. During this blending/mixing activity, a small vortex was maintained during the premix addition and for an additional 5 minutes upon complete of the premix addition, to properly wet the premix with the salt solution/admixture prior to adding the grout to the troughs. Approximately 396 g of the grout were added to each trough, targeting 220 mL.

Table 2-4. 0.45 w/p Grout Formulation for One Batch to Provide Feed for the Four Troughs

Component	Mass Addition (g)
Tank 50 Salt Simulant	663.2
Premix	1079.9
Daratard 17	13.5

2.6 Quality Assurance

This work was requested via a Technical Task Request [Ref. 6] and directed by a Task Technical and Quality Assurance Plan [Ref. 21]. The functional classification of this task is Production Support. This task is not waste form affecting and does not need to follow the quality assurance requirements of RW-0333P. Microsoft Excel and JMP Pro Version 11.2.1 was used to support this work. Data are recorded in the PerkinElmer E-Notebook under experiment C9827-00219-04 [Ref. 5]. Requirements for performing reviews of technical reports and the extent of review are established in Manual E7, Procedure 2.60 [Ref. 22]. This document, including all Microsoft Excel and JMP calculations, was reviewed by a Design Check. SRNL documents the extent and type of review using the SRNL Technical Report Design Checklist

contained in WSRC-IM-2002-00011, Rev. 2 [Ref. 23]. The approved engineering calculation [Ref. 8] in Appendix C is not part of this review, but is attached as an appendix to support the Design Check.
.

3.0 Results and Discussion

The results and discussion for each subsection in Section 2.0 will be discussed individually.

3.1 Design of Replaceable Paddles

Two different replaceable paddle designs were proposed; split paddle or replaceable tip. The split designs that were considered are shown in Figure 3-1. Upon review by SRR maintenance personal, any split design would be unacceptable given the paddles downstream of paddle 6 would either need to be removed or shifted such that the split design could be replaced and installed. The reason is due to the paddles being compressed on the shaft during installation and this compression would have to be relieved to replace the paddles and, once replaced, recompressed, essentially taking the same time/effort as replacing the existing paddle design. An additional complication with the split paddle design was that the notches could not be installed in one design, leading to multiple paddle designs so as to maintain the internal paddle configuration (Figure 1-1), this is shown for the slant split design in Figure 3-1 where the bolting is located at different locations.

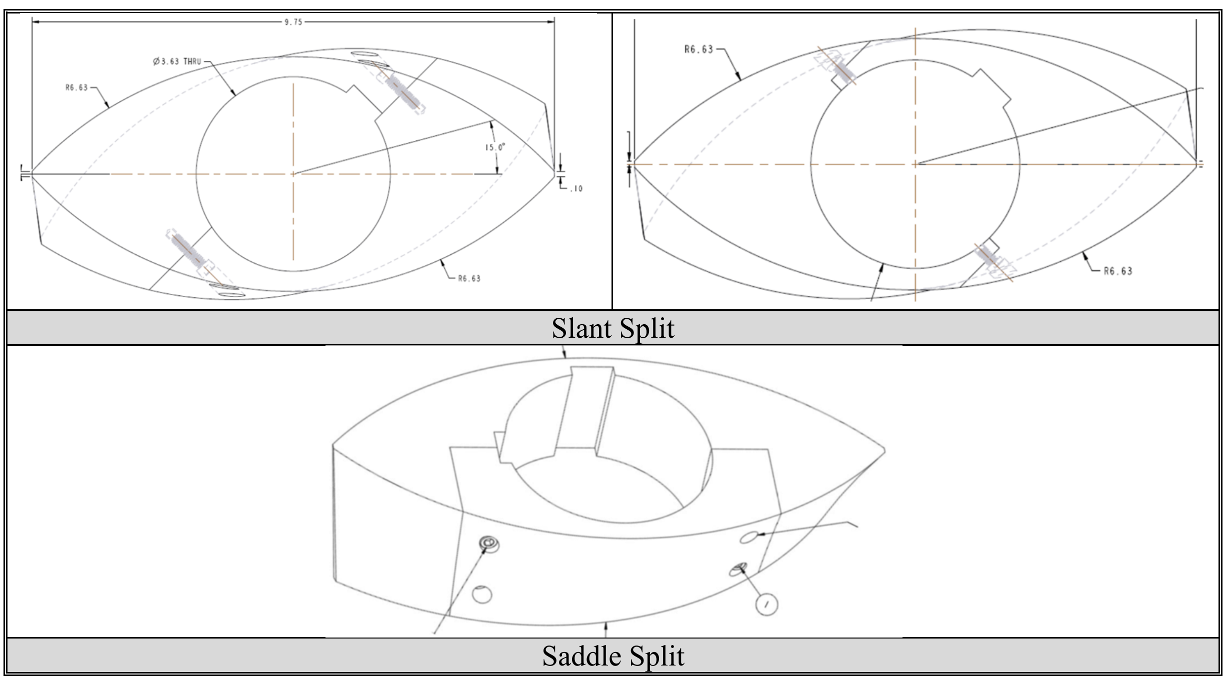


Figure 3-1. Split Paddle Design

The replaceable tip designs consisted of a central hub that contains all the notches so that a single hub could maintain the internal paddle configurations, whereas only the tips had to be replaced. The hub to replaceable tip configuration is different for a given replaceable tip design as shown in Figure 3-2. The side attachment replaceable tip was excluded for consideration due to the bolts not being accessible for the removal/attachment of the replaceable tips when configured in the 45^{o} off-set as shown in Figure 3-3. The V-neck design was excluded due to the bolting needing to absorb all the applied forces such as shearing and lifting loads. The dovetail design is the most mechanically stout of the designs but is more complex to fabricate when using the EDM. The design could be further simplified to a single dove tail connection if the replaceable tip could be molded. The dovetail design was the only design requiring a single bolt to secure the replaceable tip. If grout got in between the dovetails or if galling occurs due to applied loads on the dovetails, it might be very difficult to remove the tip from the hub. To correct the dovetail guides on the hub to accept a new tip could be both cumbersome and time consuming, defeating the purpose of easily replaceable tips. This design was not selected due to such complications.

The two designs that remain are the notched design. Both have large mating surfaces, allowing for the tip to be removed from the hub. Of these two designs, the angle notch was considered superior due to the notch absorbing more applied load than the straight, hence reducing the load on the bolts. The angle notch is designed to absorb the lifting loads, unlike the straight notch design.

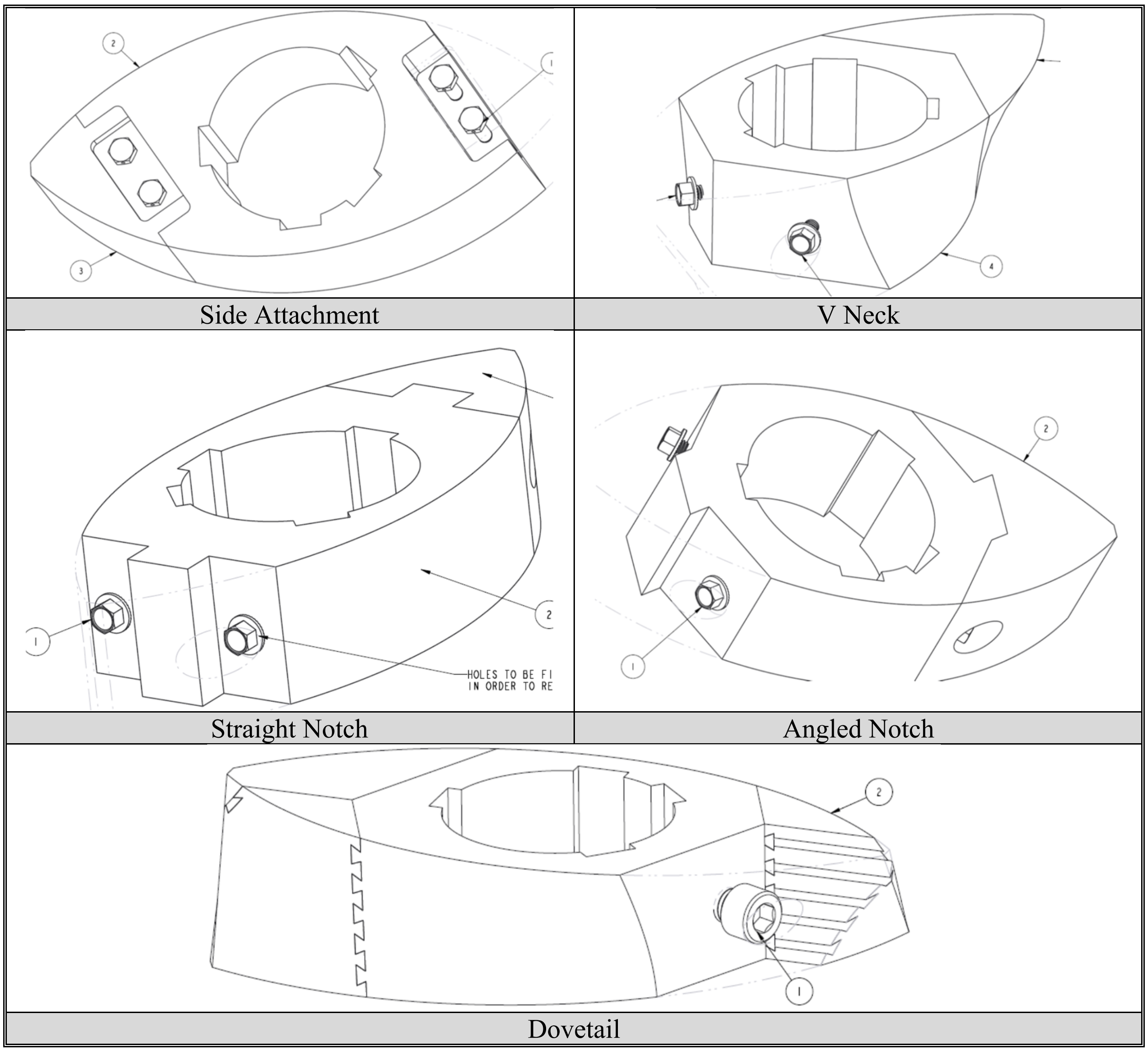


Figure 3-2. Central Hub Designs

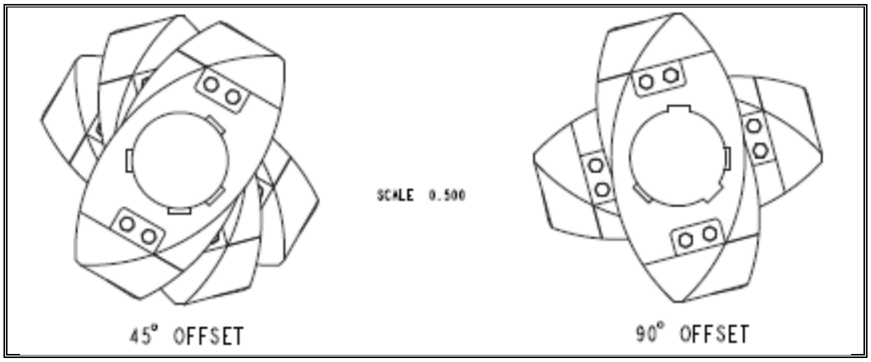


Figure 3-3. Central Hub - Side Attachment Layout on Shaft

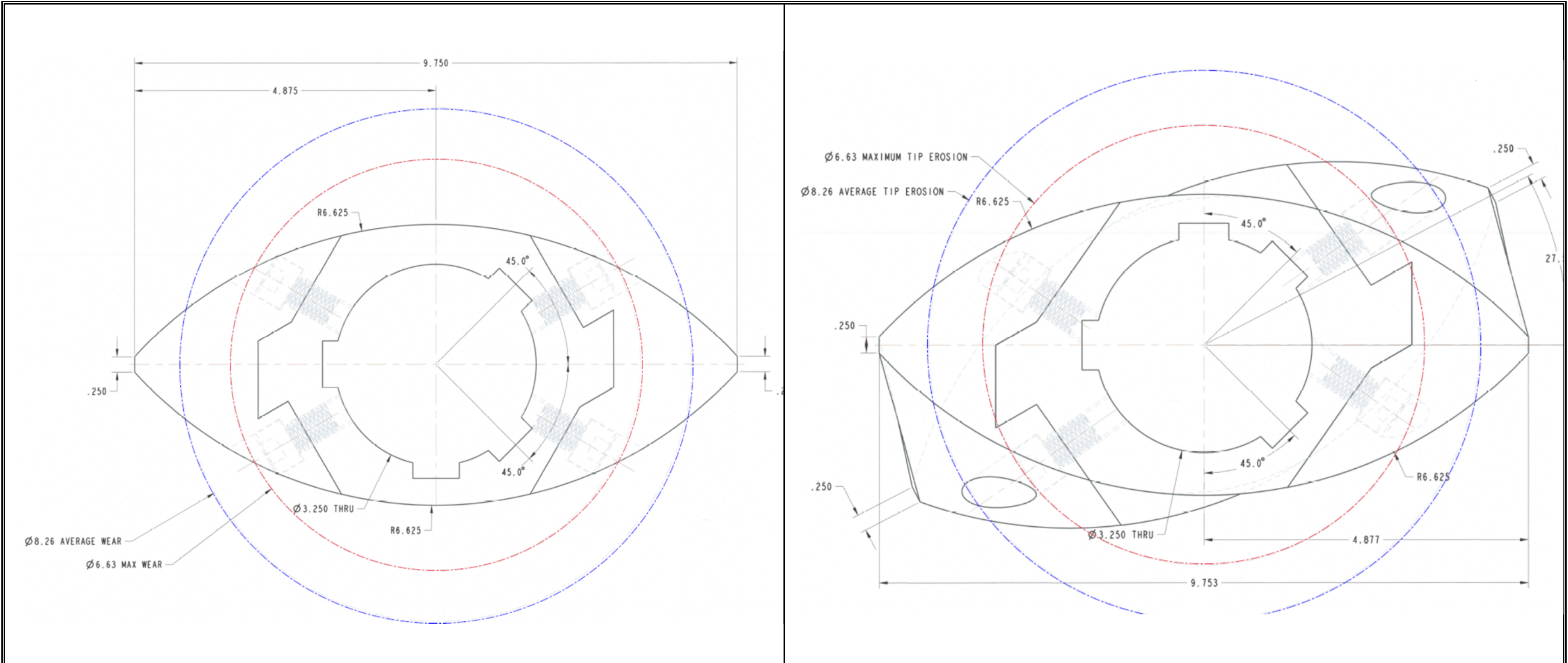
Appendix A contains the approved construction drawings for the recommended replaceable tip angled notch hub design for both the flat [Ref. 24] and helical paddles [Ref. 25]. These drawings are used in the structural analysis.

Appendix B contains the approved construction drawing of the replaceable tip straight notch hub design for both the flat and helical paddles. These drawings were provided per request from SRR and have not been analyzed structurally; only the selected design was analyzed.

A further consideration was the location of the bolt caps such that they would be within the maximum single wear data point reported by SRR. These regions are shown in red in Figure 3-4 for both the flat and helical design.

3.2 Fabrication of Replaceable Paddles

The fabrication of a set of replaceable paddles made from Astralloy V was substituted using 3-D plastic printing technology models, given 3-D models were shown to be effective tool to downselect designs. In place of the Astralloy V paddles, four sets of helical paddles were fabricated, and a mixer jig designed/fabricated to hold the four sets of paddles as though they were installed in the mixer (Figure 3-5). The yellow and white paddles are the angled notched hub design and the red and blue paddles are the V-neck design. The jig included simulated shaft with key way and a lower barrel section like that of the READCO™ mixer internals. The shafts were connected to gears and a hand crank that allows the paddles to co-rotate as in the actual application. This jig was also used to help assess fitness of duty. A single dovetail 3-D helical paddle was fabricated and provided to SRR.



Flat	Helical

Figure 3-4. Potential Wear Region

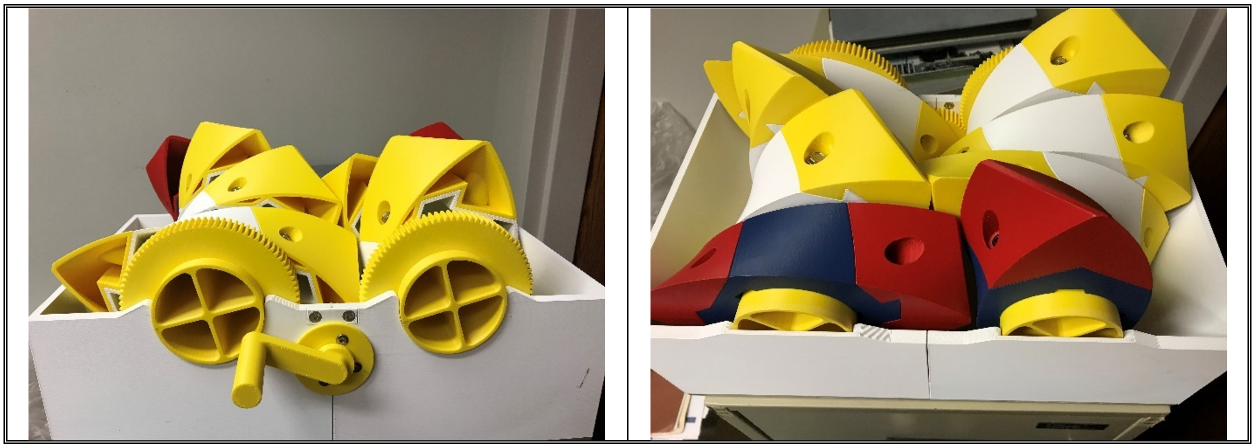


Figure 3-5. Four Sets of Helical Paddles Installed in SPF Mixer Jig

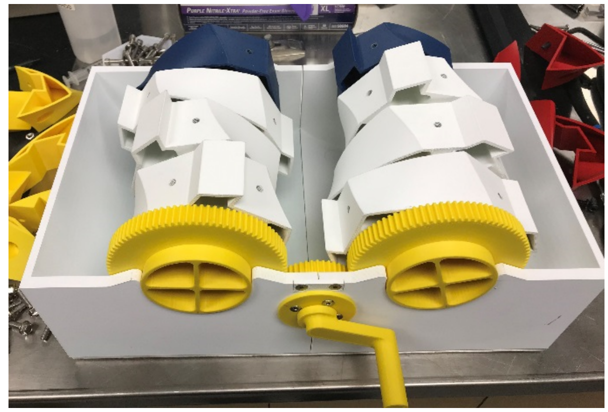
3.3 Fitness of Duty

The fitness of duty requirements covers both performance and maintenance and are summarized in Table 3-1. All the requirements have been satisfied. Using the 3-D model, it was identified that tools might have to be made or modified to properly clean the barrel when the tips are replaced.

Table 3-1. Summary for Fitness of Duty Requirements

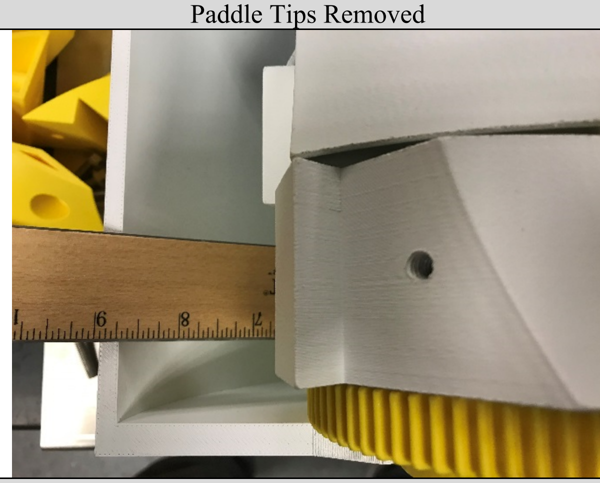
Requirement		Summary
Performance	Dimensionally the same as the paddles provided by READCO™.	Appendix A contains the approved construction drawings for the recommended replaceable tip angled notch hub design for both the flat and helical paddles. They are dimensionally the same as the READCO™ paddles provided by SRR.
	Co-located paddles do not interfere with each other during rotation.	Four sets of replaceable tip helical paddles were installed in a SPF Mixer Jig and rotated without interfering with each other rotation.
	Structurally suitable for the application. Suitable for at least one year of operation.	E7 Calculation (Appendix C) supports the bolting material and replaceable tip design for use in the READCO™ mixer.
Maintenance	Ability to remove the replaceable paddle without moving the paddles downstream of paddle 6.	The selection of the replaceable tips satisfies this requirement. The shaft will have to be manually rotated to replace tips.
	Ability to access and clean cured grout that has accumulated on the barrier. In this case, access will be assessed.	Mixer Jig was used to determine accessibility when all the removeable tips were removed. Tools might have to be modified with bends to properly access and clean the barrel. The shaft will have to be rotated for proper access. (Figure 3-6)
	Ability to access bolting locations.	Replaceable tips can be removed by rotating the shaft such that the bolting material can be accessed. The Mixer Jig facilitated the assessment.

Easy Access to Bolts



Paddle Tips Removed

Access to shell, between the paddles



Access to shell from the side

Figure 3-6. Access to the READCO^{TM} Shell Using 3-D Model Mixer Jig

3.4 Structural Calculation to Support Paddle Selection

An structural calculation of the SPF mixer paddle replaceable tip design was performed, reviewed, confirmed and approved. The calculation, M-CLC-Z-00137 [Ref. 26] is attached in Appendix C. The structural analysis was performed on the design drawings provided in Appendix A. The material of construction for the hub and tip was Astralloy V and the tips were attached to the hub with two ½" - 13 Unified National Coarse Threads (UNC) 304 stainless steel socket head cap screws (SHCS).

The calculation addressed both the helical and flat paddle design, which could be arranged in any order along the mixer shaft, to be structurally adequate for normal and extreme loads encountered in the SPF READCO^{TM} mixer. The issue of fatigue was also addressed, which indicate time of service (e.g., one year of operation) is not a limiting factor. Erosion will be the limiting factor and inspections such as those presently being performed will be necessary to assess erosion and replacement of the tips.

The attached calculation recommends that the ½" – 13 UNC SHCS be installed and torqued to a value of 25 ft-lb ($\pm$ 3 ft-lb).

3.5 Erosion Testing

The results using the 50 wt. % ASTM 50-70 test sand and 50 wt. % DI water and running the Miller test per ASTM G75 [Ref. 12] are provided in Table 3-2. The rs/min target per the ASTM is 48 rs/min. The rs/min versus dial speed curve fit from five-minute readings was used to determine the 48 rs/min setting (Figure 3-7). Based on the curve fit (Figure 3-7), the dial speed setting would be approximately 8 (and was used), but given the data, the estimated speed was 50.6 rs/min. The reported SAR values for the 27 wt. % Cr. Iron in the ASTM G75 was an average of 153.00, a 95% within-lab repeatability of $\pm$19.38, and a 95% reproducibility between labs of $\pm$52.21. The measured value in this testing falls within the upper range of the 95% reproducibility between labs. The difference between the ASTM and the data reported in Table 3-2 could be due to slight differences in speed at which the measurement was performed. The results in Table 3-2 show that the Astralloy V and Ultimet have erosion rates much greater than that of the 27 wt. % Cr. Iron with by far the Ultimet faring the worse. As stated in Section 2.5, this is informational only of SAR values.

Table 3-2. SAR Values Using ASTM Sand for Materials Tested

Sample time	Cumulative mass loss (g)			
	27 wt. % Cr Iron		Ultimet	Astralloy V
1st 2 hr.	0.0250	0.0249	0.3060	0.1155
2nd 2 hr.	0.0434	0.0464	0.5602	0.1917
3rd 2 hr.	0.0641	0.0628	0.7841	0.2480
SAR	191	194	2142	713

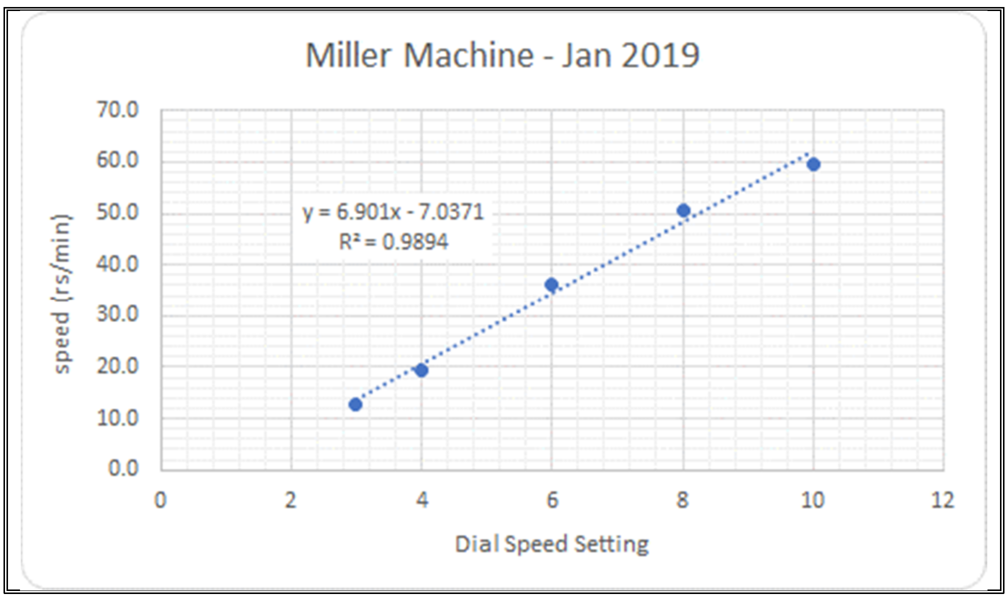


Figure 3-7. Miller Machine Speed (rs/min) versus Dial Speed Setting

For the saltstone grout tests, the total number of reciprocating strokes was determined for a 6-hour test based on the maximum measured speed setting and the lower values related to the curve fit in Figure 3-7.

Based on the maximum measured rs/min of 59.6, the total number of reciprocating strokes for six hours is 21,462. This value is divided by three to obtain the target 7154 that was used for each two-hour trial for the maximum speed, for each three-hour run for the 2/3rd of maximum speed and for the single run at 1/3rd of maximum speed. During testing the number of strokes and time differences were measured and used to determine the actual rs/min. The speed setting used, the rs/min, linear speed, total reciprocating strokes, and the total time for the measurement are reported in Table 3-3.

Table 3-3. Actual Speed Values, Strokes, and Total Time for the Three Different Speeds Setting Used for the Miller Machine

Cycle	Speed Setting	rs/min	Linear speed - V_{ra} (ft/sec)	Total reciprocating strokes	Total Time (hr.)
Maximum	10+	61.5	0.68	21462	5.81
2/3rd Max	6.8	43.2	0.48	14308	5.52
1/3rd Max	3.9	19.3	0.21	7154	6.19

The average mass and volumetric losses for the Ultimet and Astralloy V wear blocks are provided in Table 3-4 for the different cycles. Based on these results, the Astralloy V wears faster than the Ultimet and the difference between the two gets larger as the cycle increases from 1/3rd max to maximum. This data is consistent with prior testing [Ref. 4] where the Astralloy V lost more mass than Ultimet.

Table 3-4. Mass and Volumetric Losses for the Different Cycles

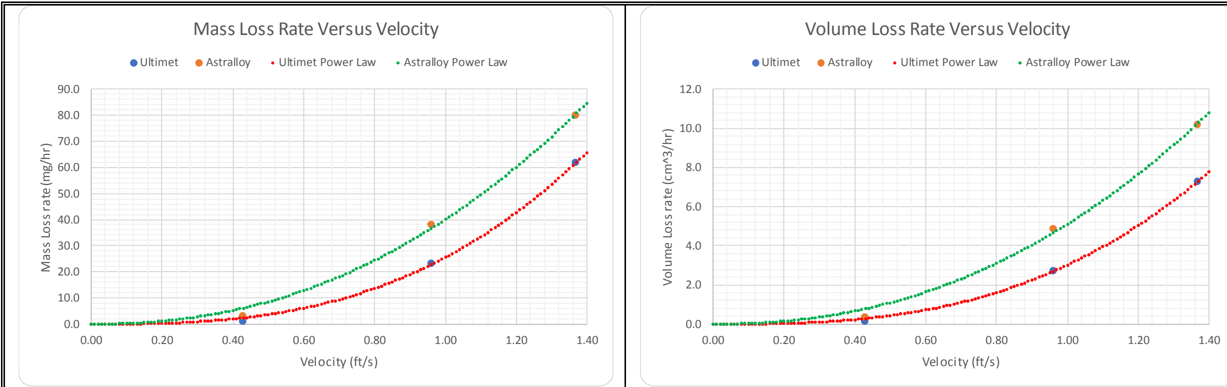
Cycle	Average Mass Loss (g)		Average Volumetric Loss (cm^3)	
	Ultimet	Astralloy V	Ultimet	Astralloy V
Maximum	0.3593	0.4663	0.0424	0.0594
2/3rd max	0.1277	0.2112	0.0151	0.0269
1/3rd max	0.0692*	0.1456	0.0082	0.0185

* One measurement report. 2nd wear block was not installed properly during measurement.

The mass and volumetric erosion rates are provided in Table 3-5. The mass and volumetric erosion rate data and speed are plotted and shown in Figure 3-8. These figures include the plotted power law fits to the mass and volumetric rate data and the equations. The statistical analysis of this data, using JMP Pro Version 11.2.1, is provided in Appendix D. The results from these equations show that by reducing the velocity a factor of two would reduce the erosion rates by a factor of 7.0 and 4.7 for the Ultimet and Astralloy V respectively. As stated in Section 2.5, reducing the speed by a factor of two did not impact the quality of the grout and this reduction was used as the example.

Table 3-5. Mass and Volumetric Loss Rates for the Different Cycles

Cycle	Mass Loss Rate (g/hr.)		Volumetric Loss (cm^3/hr.)	
	Ultimet	Astralloy V	Ultimet	Astralloy V
Maximum	61.81	80.22	7.30	10.22
2/3rd max	23.11	38.23	2.73	4.87
1/3rd max	1.32	3.00	0.16	0.38



Power Law Fit		Power Law Fit	
Ultimet	$\dot{m} = 25.5 \cdot V_{ra}^{2.814}$	Ultimet	$\dot{V} = 3.01 \cdot V_{ra}^{2.814}$
Astralloy V	$\dot{m} = 40.0 \cdot V_{ra}^{2.229}$	Astralloy V	$\dot{V} = 5.10 \cdot V_{ra}^{2.229}$
Mass Erosion Rate		Volumetric Erosion Rate	

Figure 3-8. Mass and Volumetric Erosion Rates

The maximum tip speed of the READCO™ mixer at 238 RPM is 10.1 ft/sec. This is approximately an order of magnitude greater than the maximum speed obtained from the Miller machine. There are also physical differences between the READCO™ mixer and Miller machine, one being a constant load is always applied by the Miller machine whereas the load varies for the READCO™ mixer. In addition, the composition of the w/p is highly variable in the region where the premix initially mixes with the supernate in the READCO™ mixer but constant in the Miller test. Hence, the Miller test cannot be used as a direct estimate of wear rates during operation.

4.0 Conclusions

The primary objective of this task was to provide a recommendation for a replaceable paddle that could be incorporated into the Saltstone Production Facility (SPF) process. The replaceable paddle concept was proposed given the time it took to replace the existing paddle designs was approximately a week of downtime. Two different types of replaceable paddles were considered, a split paddle and a replaceable tip paddle. The split paddle design was developed; however, a review of the drawings by maintenance personnel determined that changeout of the new design would take about the same time/effort as replacing the existing paddles. Five different replaceable tip designs were considered: side attachment, V neck, straight notch, angled notch, and dovetail. The recommended replaceable tip design was the angled notch, which includes a central hub with two angled notches, where the replaceable tips are secured to the hub using two ½" – 13 Unified National Coarse Threads (UNC) socket head cap screws (SHCS) (Figure 4-1). Design drawing for both the flat and helical paddle blades are complete and are included in Appendix A. A structural calculation (M-CLC-Z-00137) was performed on this design which determined the design is structurally adequate for normal or extreme loads that would be encountered in the SPF READCO™ mixer. This calculation is included in Appendix C. Based on the calculation, mechanical failure will not limit the life of the tip or tip attachment. The quantity of erosion will be the primary driver dictating tip replacement schedules.

This design was assessed for fitness of service and was determined to satisfy all the performance and maintenance requirements. Tools may need to be made so that any accumulation of grout on the shell can be removed during maintenance, during the removal or during replacement of the tips.

Erosion testing was performed. The Astralloy V and Ultimet materials were placed into simulant saltstone grout and tested to measure erosion rates versus speed. Testing showed that erosion rates are a function of a power law relationship and the velocity exponents were 2.229 and 2.814 for Astralloy V and Ultimet, respectively. This finding suggests that reducing the speed is highly beneficial in reducing the erosion rate and extending the life of wear components that have similar conditions. Based on these velocity exponent values, reducing the speed by 50 percent would reduce the erosion rate by a factor 4.7.

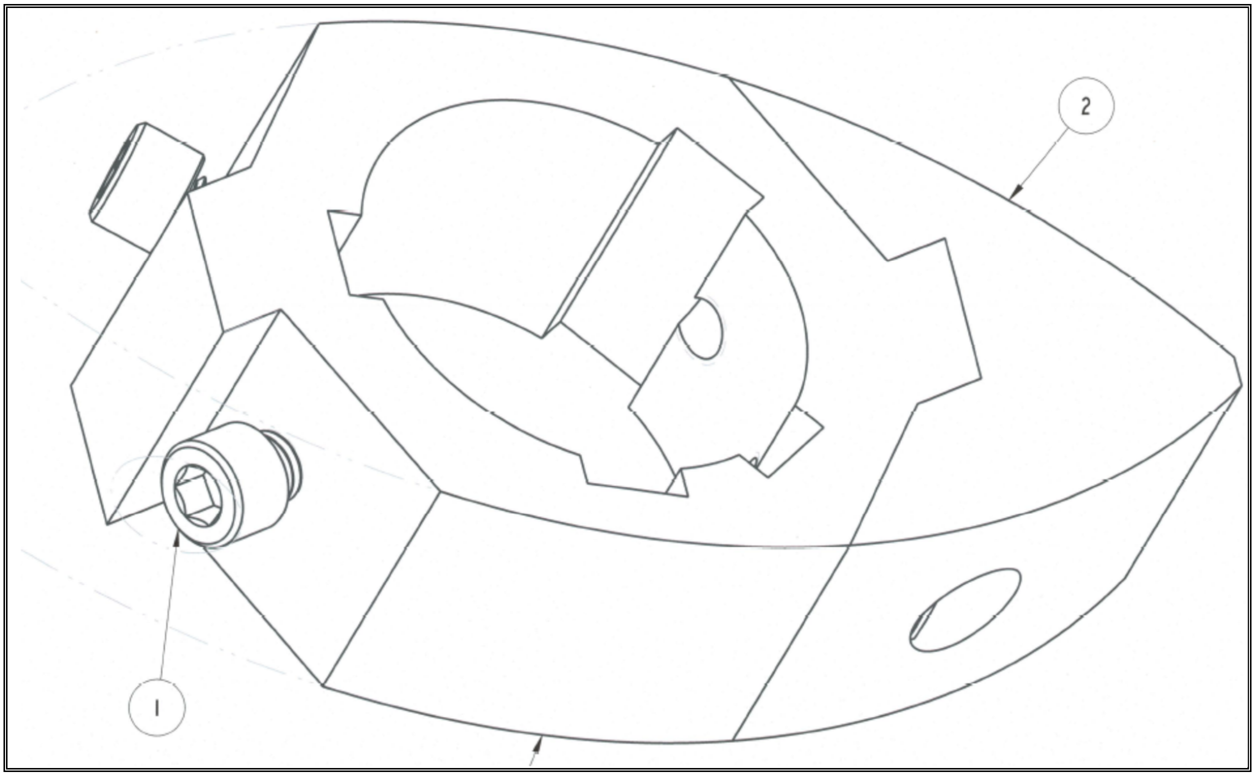


Figure 4-1. Recommended Replaceable Tip, Flat Paddle Design

5.0 Recommendations and Future Work

The following are recommendations from this work.

(1) Use the recommended replaceable tip design if the decision is to use paddles in the 1 through 6 paddle locations in the SPF READCO^{TM} mixer.
(2) Reduce the SPF READCO^{TM} mixer speed using a variable frequency drive. Reduction of speed will reduce erosion rates as well as the applied loads. SPF should determine the minimum speed based on processing experience using premix and water.

Future work.

(1) Procure hardened materials that could be used as the replaceable tips. Materials would be assessed using the Miller machine and corrosion testing using representative simulants.
(2) Show that EDM technology can be used to fabricate the replaceable tip design.
(3) Fabricate four pairs of flat and four pair of helical paddles and 12 replacement tips (six of each) for additional fitness of duty testing in spare SPF READCO^{TM} mixer using Astralloy V. Testing should also be used to determine tool design necessary to remove grout buildup on the barrel.
(4) Measure the material properties of the Astralloy V paddles after fabrication to determine consistent material properties.

6.0 References

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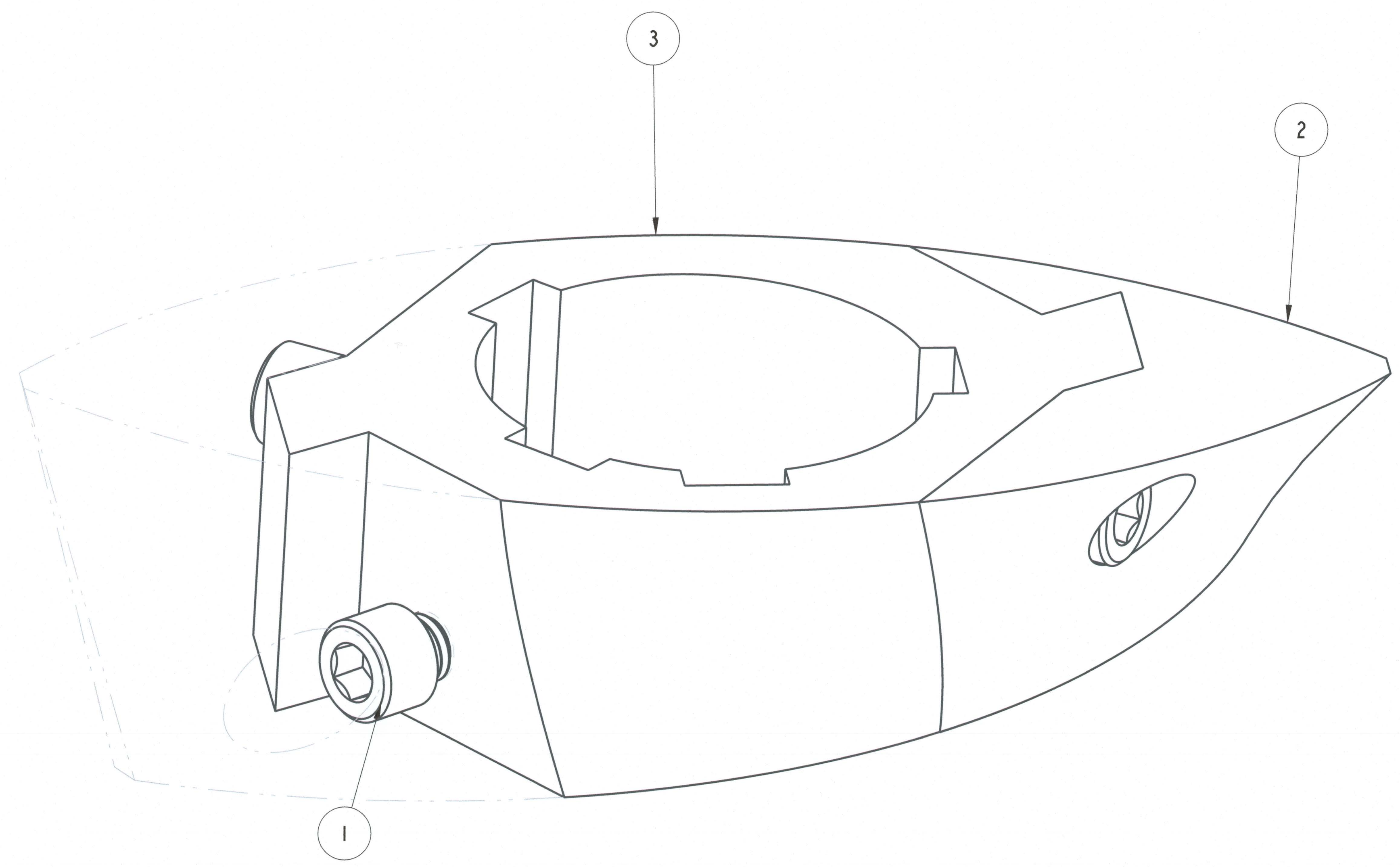
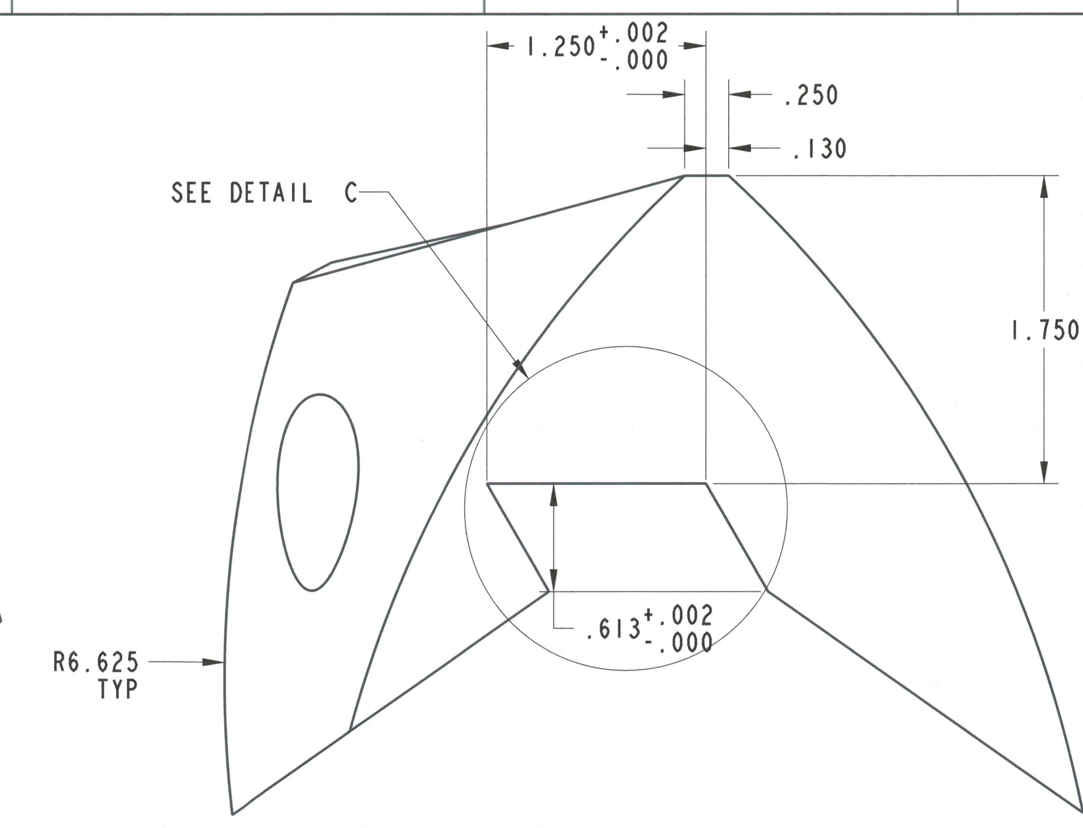
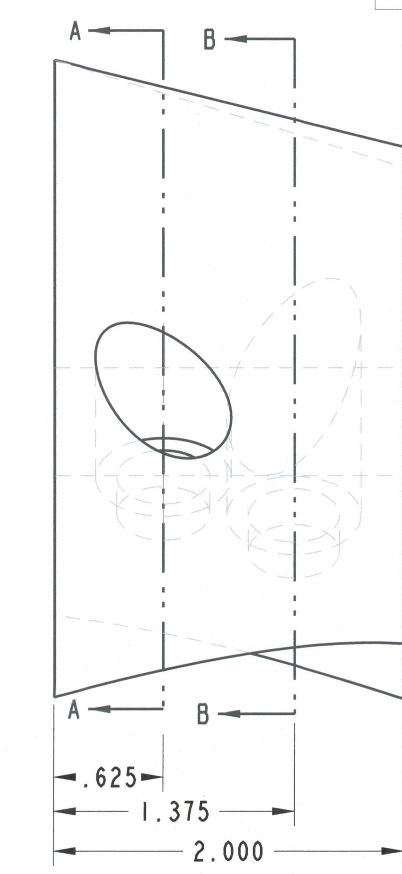
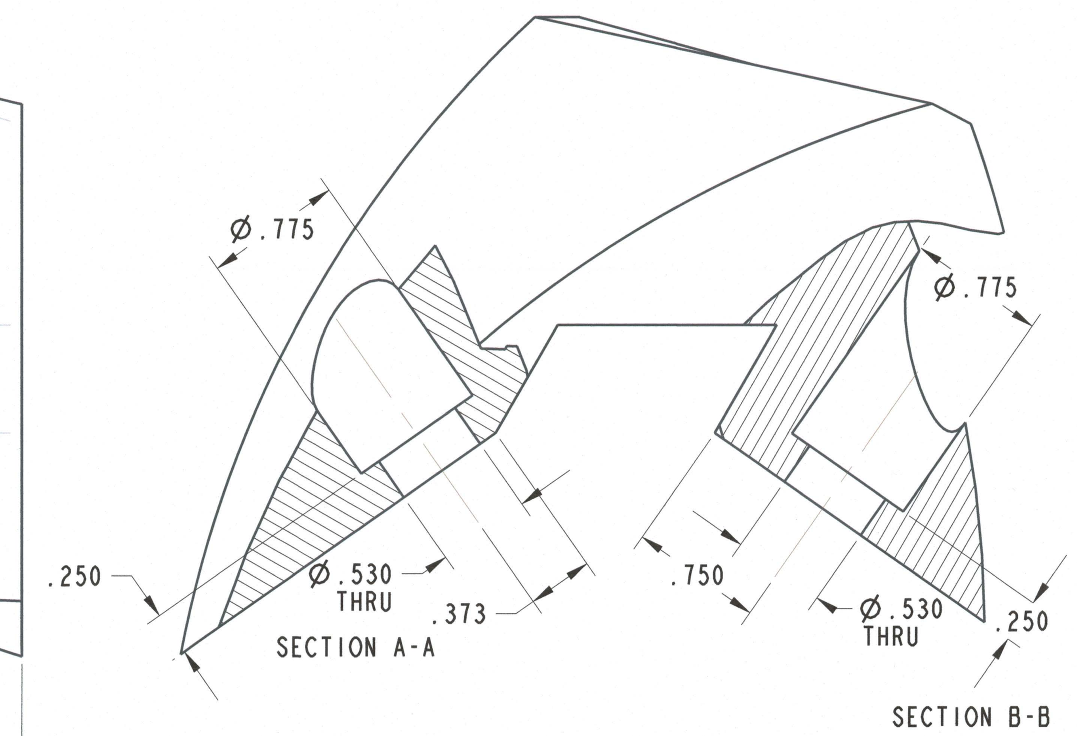
Appendix A. Recommended Replaceable Tip Angled Notch Hub Design

PROJ	REV	DATE	REVISION DESCRIPTION	PRPD	ENGR	CHKD	DEMR	CUSTOMER OTHER	CUSTOMER OTHER
N/A	A	12/13/2018	ISSUED FOR CONSTRUCTION	WTL	WTL	JSB			









TIP
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R-R4-Z-00013-B
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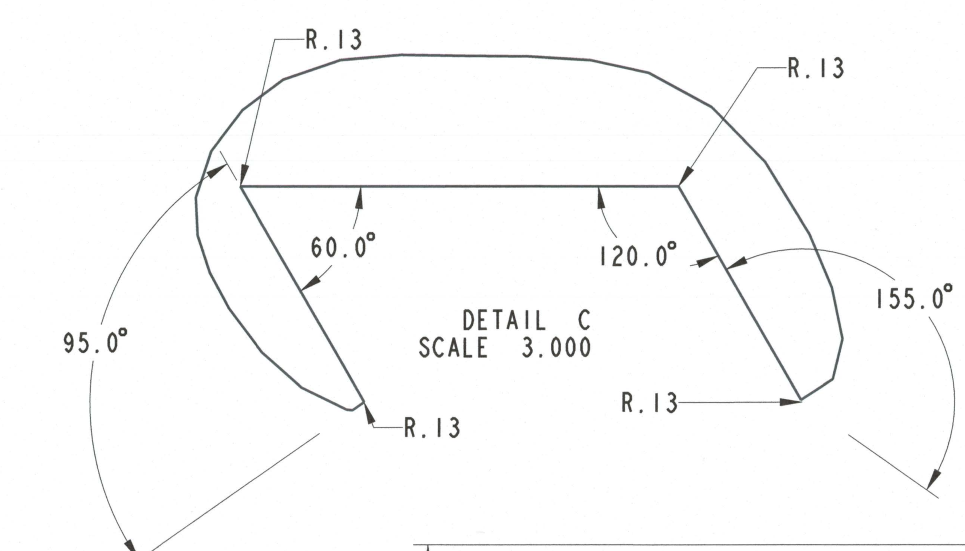
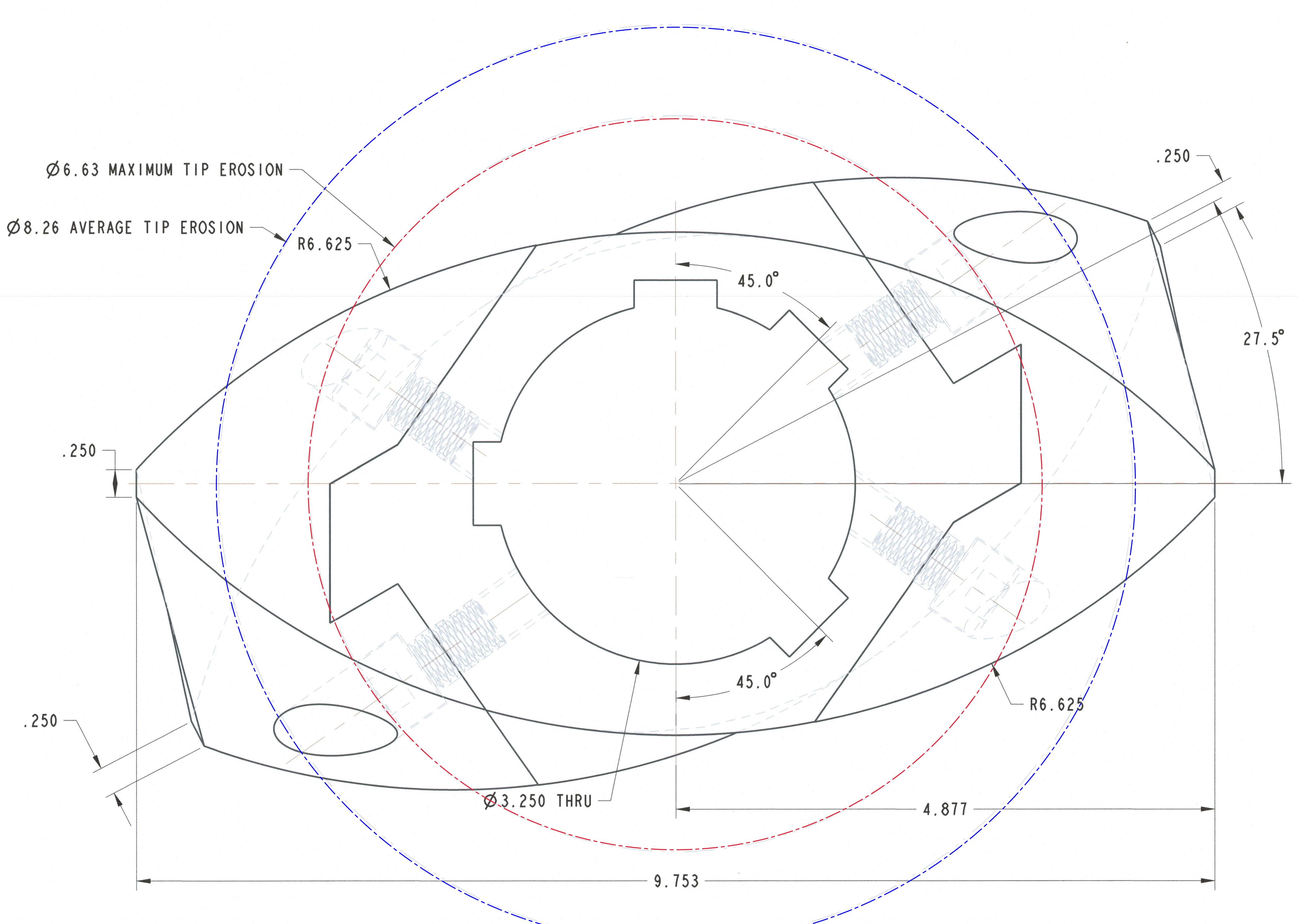
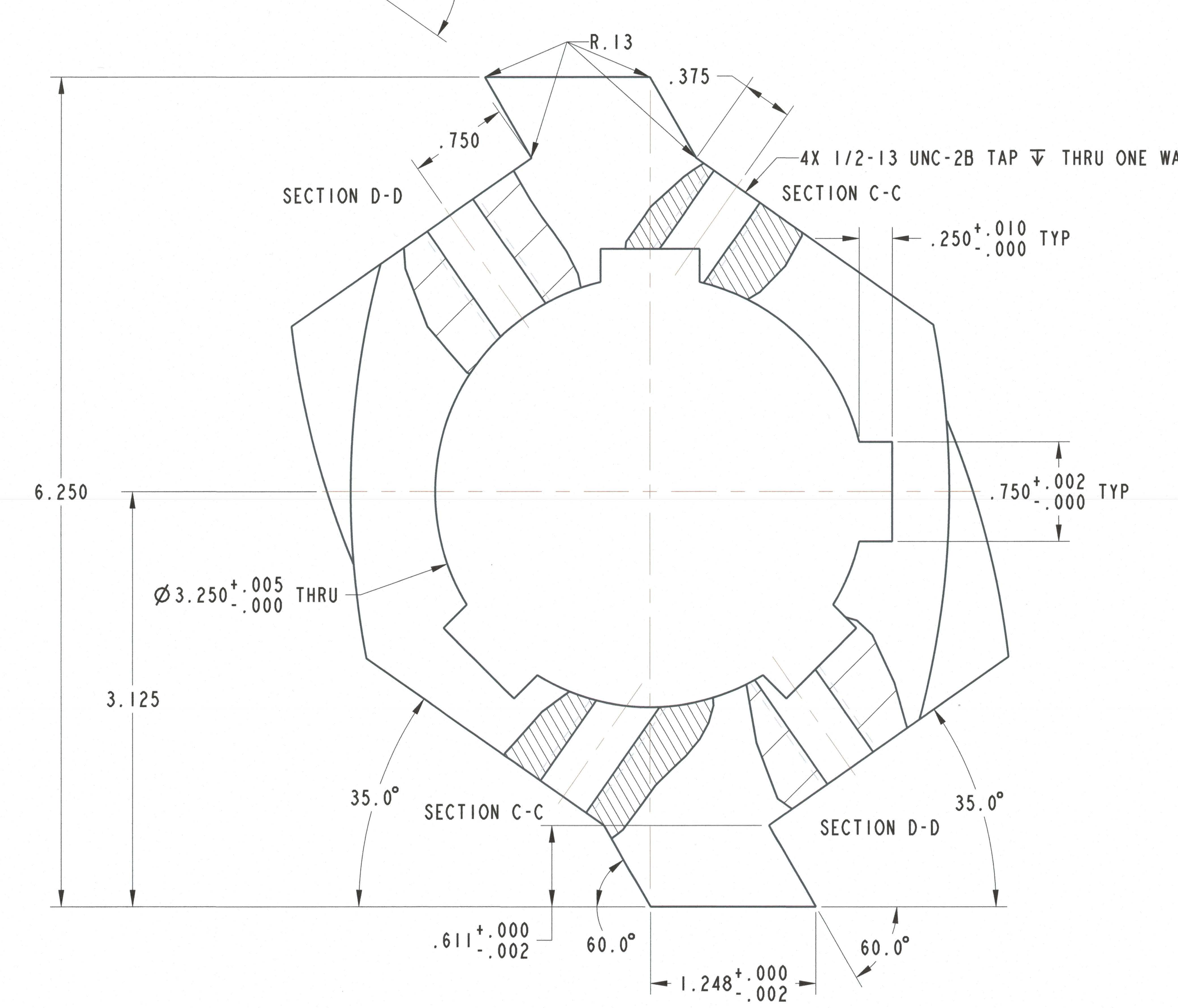
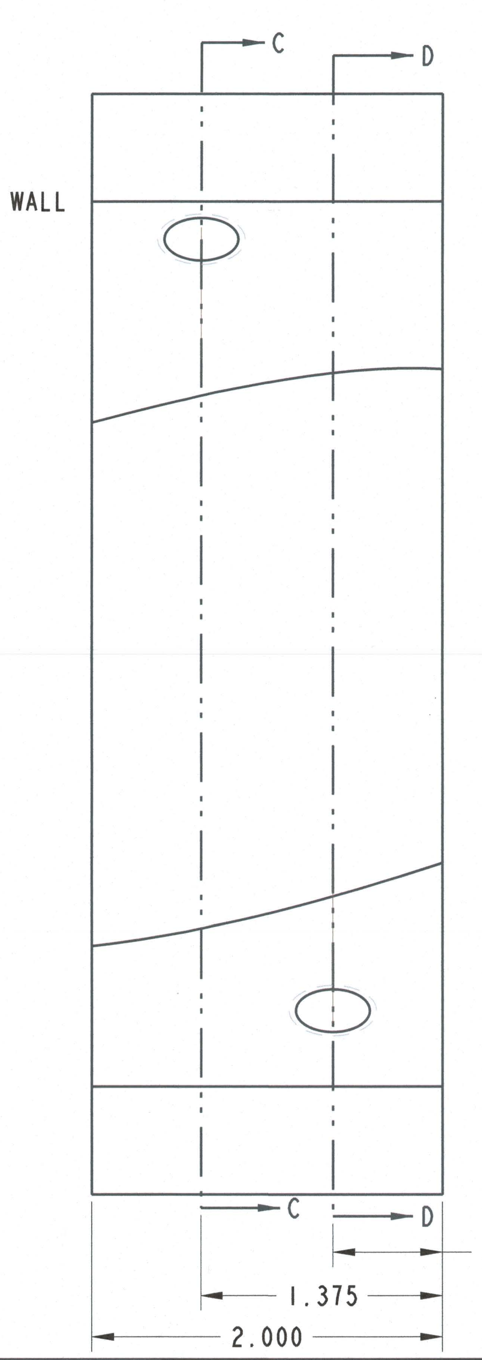


ITEM	PART AND DRAWING NO.	REQD PER ASSEMBLY	DESCRIPTION
3	R-R4-Z-00013-C	1	HUB
2	R-R4-Z-00013-B	2	TIP
1	COML	4	1/2-13 SST SHCS W/ THREAD LOCKING NYLON PATCH, MCMASTER-CARR #: 93705A637



HELICAL PADDLE, INTERCHANGEABLE TIPS, SLOTTED
R-R4-Z-00013-A
SCALE: 18" = 1'-0





HUB
MATL: ASTRALLOY "V" REQD: 1
R-R4-Z-00013-C
SCALE: 18" = 1'-0

NOTCH AND HOLE FEATURES ARE MIRRORED ACROSS PART

TOLERANCES ARE: FRACTIONAL ±1/16"; DECIMAL .X ±0.03, .XX ±0.01, .XXX ±0.005; ANGULAR ±.5°
THIRD ANGLE PROJECTION
UNLESS OTHERWISE SPECIFIED: DO NOT SCALE DRAWING; BREAK ALL SHARP EDGES; FINISH ALL SURFACES 63
INTERPRET DIMENSIONS AND TOLERANCES PER ASME Y14.5M

RESPONSIBLE ENGINEER	W. LOZIER
ENGINEERING MANAGER	J. KINNEY
DRAWING ORIGINATOR	W. LOZIER

REFERENCE DRAWINGS

474795	PDMLINK DRAWING
474708	PDMLINK ASSEMBLY

REFERENCE STANDARDS

SITE CLEARANCE NO.	BUILDING NUMBER	DRAWING NUMBER	REV
N/A	704-Z	R-R4-Z-00013	A
DESIGN AUTHORITY	DESIGN GROUP	SCALE / JOB NUMBER	SHEET
N/A	SRNL R&DE	SHOWN / 23694	1 OF 1

UNITED STATES DEPARTMENT OF ENERGY
SAVANNAH RIVER SITE

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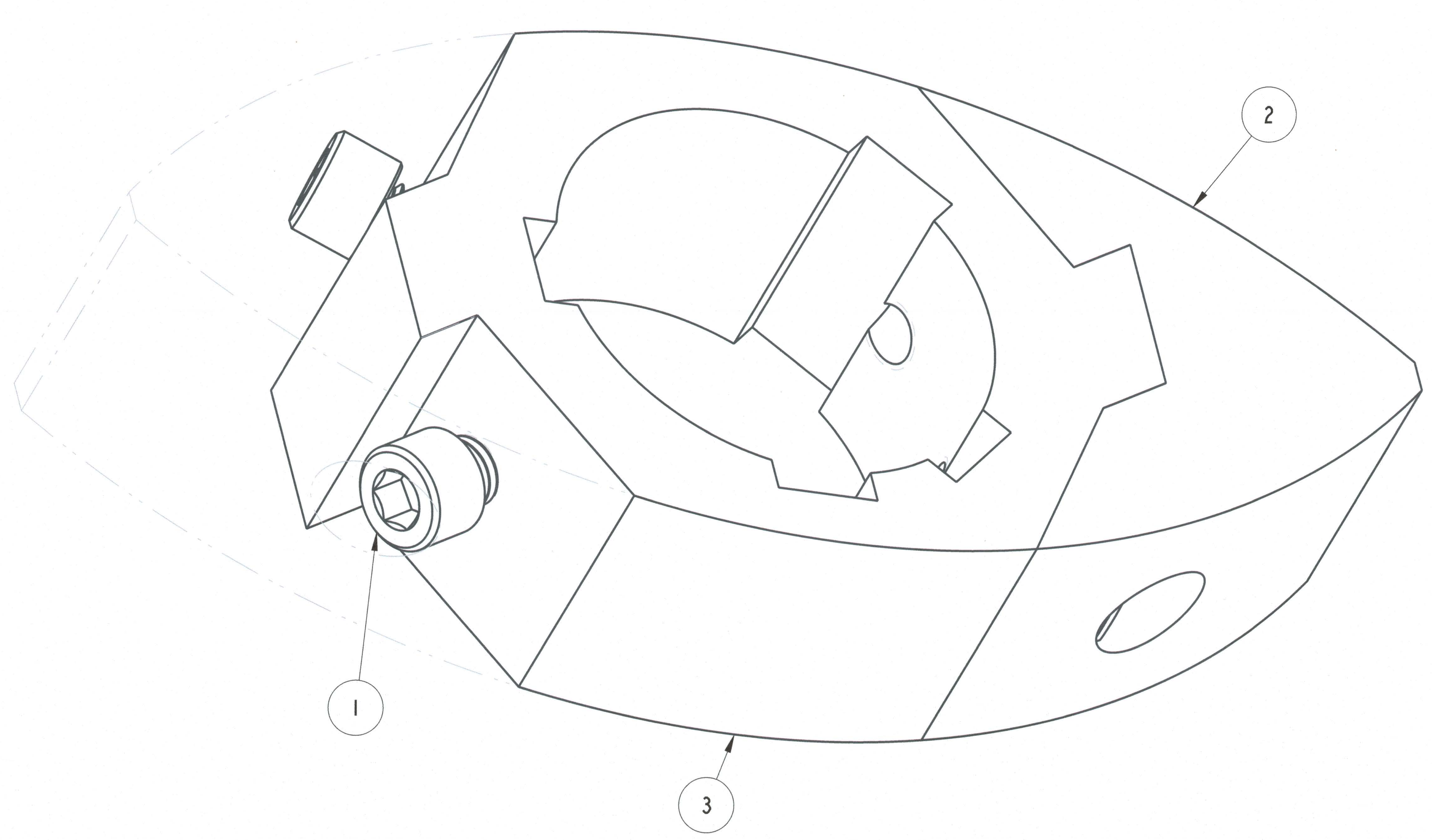
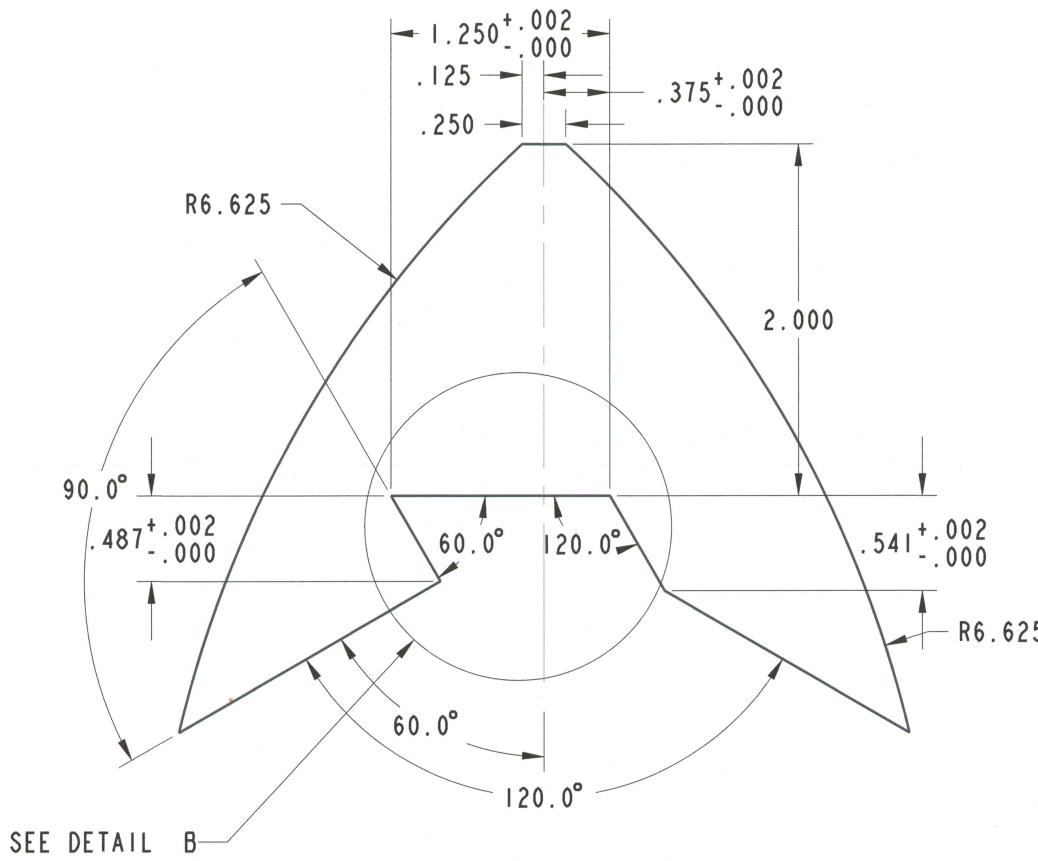
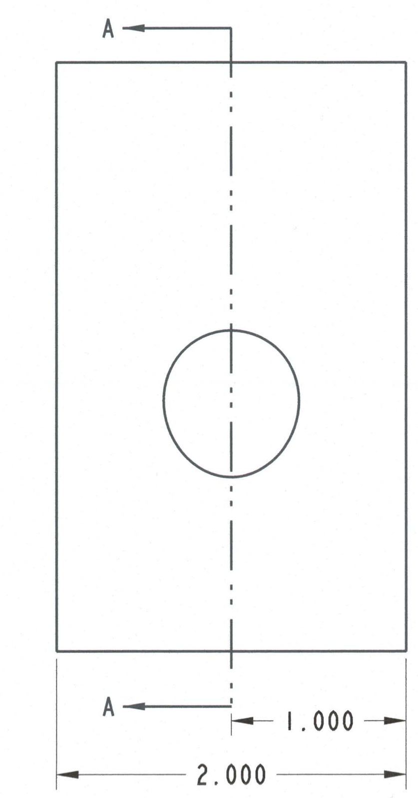
TITLE
SALTSTONE
HELICAL PADDLE,
INTERCHANGEABLE TIPS
ANGLED NOTCH DESIGN
ASSEMBLY AND DETAILS

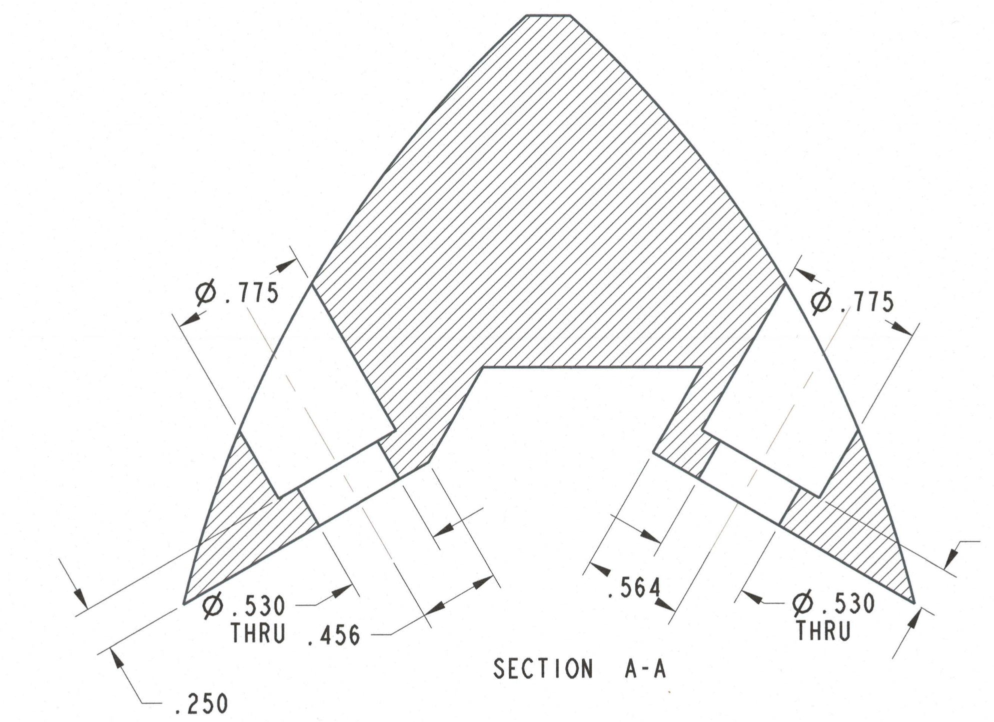
PROJ	REV	DATE	REVISION DESCRIPTION	PRPD	ENGR	CHKD	DEMR	CUSTOMER OTHER	CUSTOMER OTHER
N/A	A	12/13/2018	ISSUED FOR CONSTRUCTION	WTL	WTL	AJB			



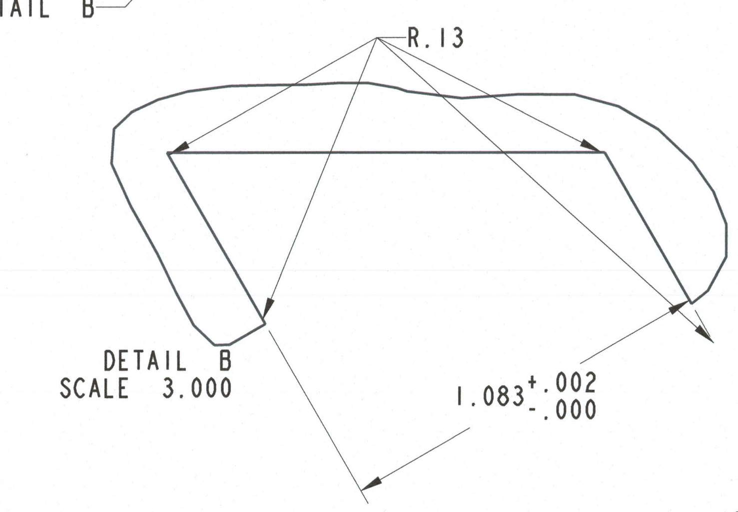
ITEM	PART AND DRAWING NO.	REQD PER ASSEMBLY	DESCRIPTION
3	R-R4-Z-00016-C	1	HUB
2	R-R4-Z-00016-B	2	INTERCHANGEABLE TIP
1	COML	4	1/2-13 SST SHCS W/ THREAD LOCKING NYLON PATCH, MCMASTER-CARR #: 93705A637





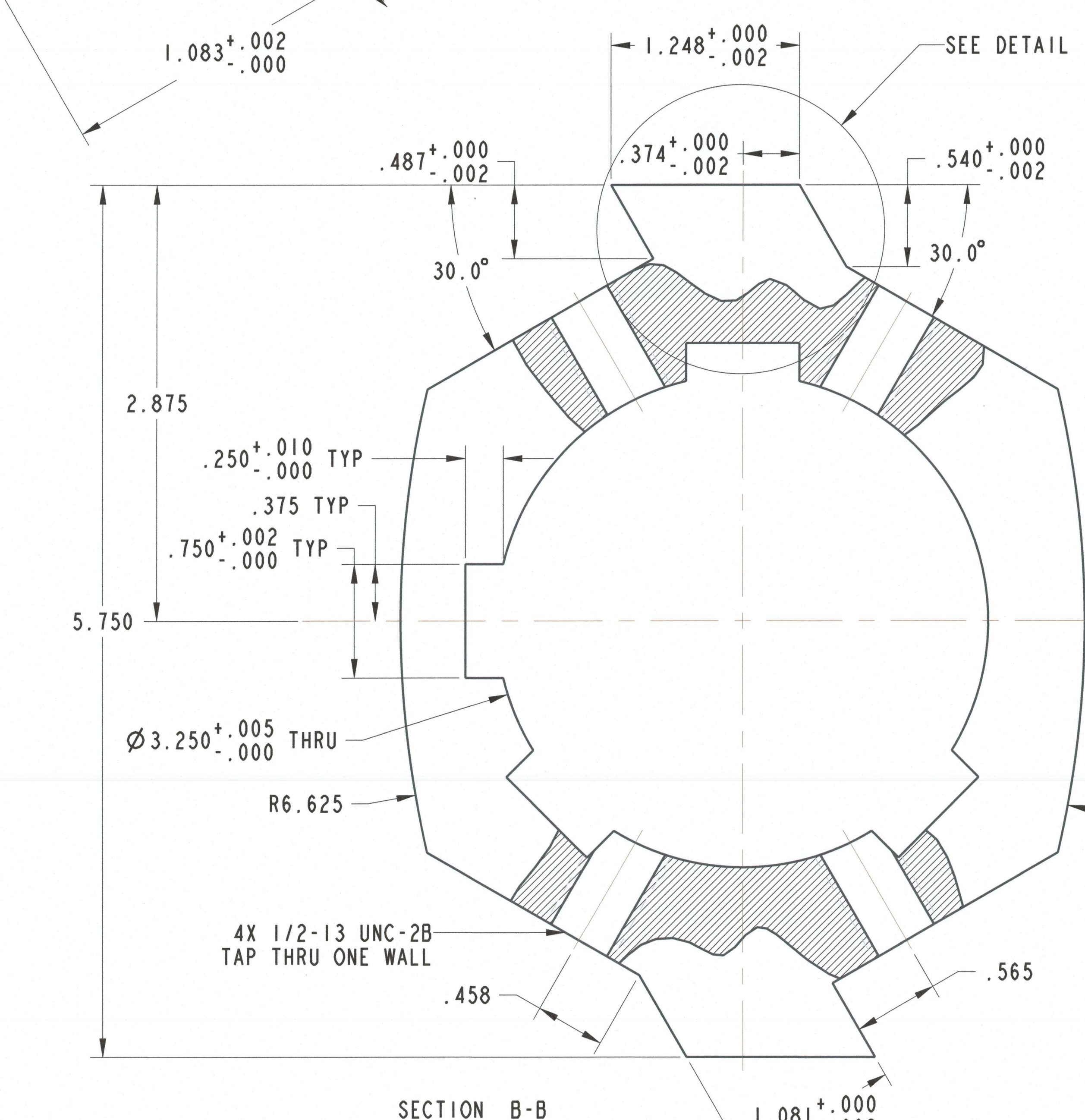


SECTION A-A

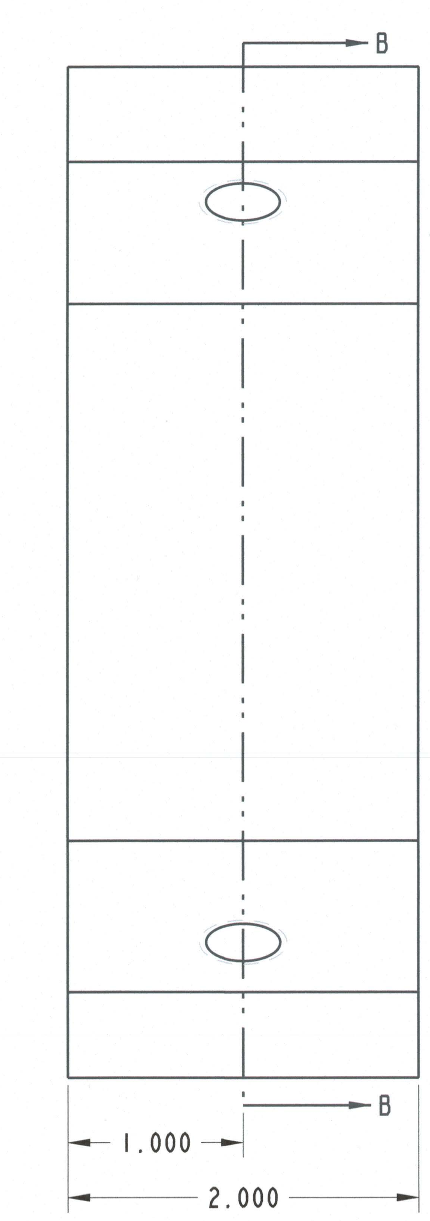


DETAIL B
SCALE 3.000

INTERCHANGEABLE TIP
MATL: ASTRALLOY "V" REQD: 2
R-R4-Z-00016-B
SCALE: 18" = 1'-0

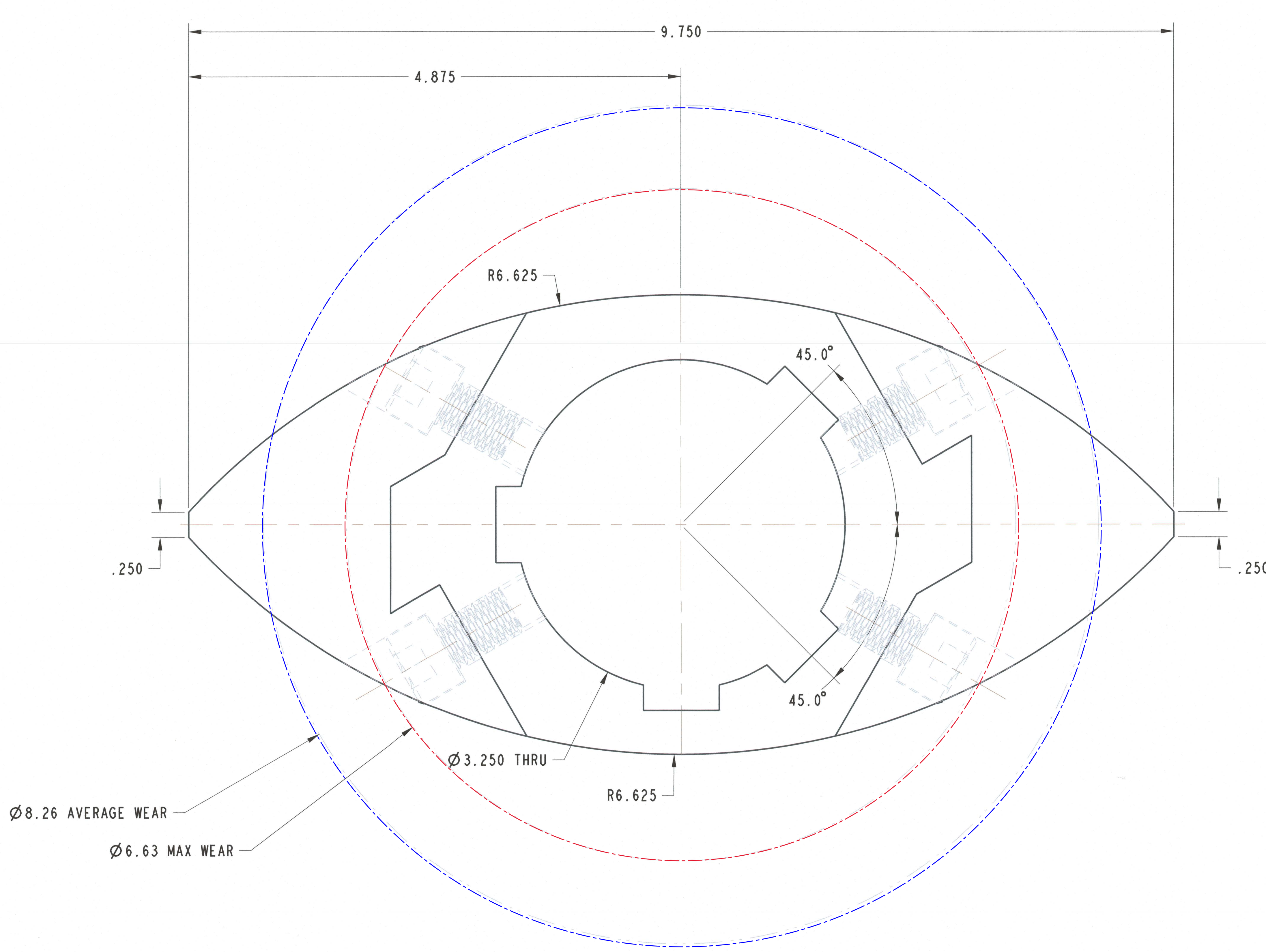


SECTION B-B

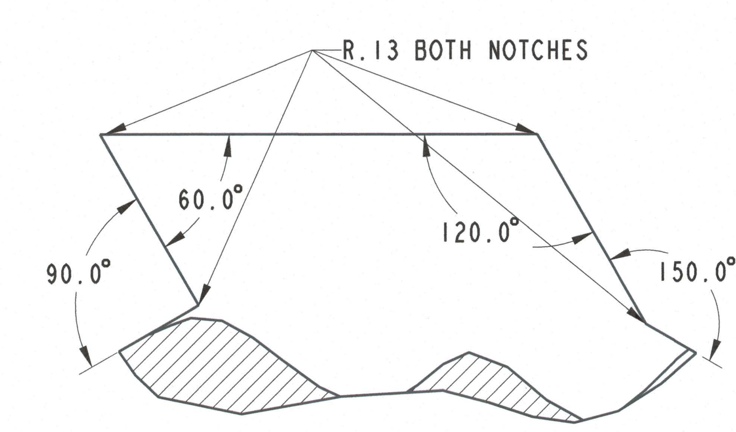


NOTCH AND HOLE FEATURES ARE MIRRORED ACROSS PART

HUB
MATL: ASTRALLOY "V" REQD: 1
R-R4-Z-00016-C
SCALE: 18" = 1'-0



FLAT PADDLE, INTERCHANGEABLE TIPS, NOTCHED
R-R4-Z-00016-A
SCALE: 18" = 1'-0



DETAIL A
SCALE 3.000

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	DESIGN AUTHORITY N/A	DESIGN GROUP SRNL R&DE	SCALE SHOWN	JOB NUMBER 23694	SHEET 1 OF 1

TOLERANCES ARE: FRACTIONAL ±1/16" ANGULAR ±.5° DECIMAL .X ±0.03 .XX ±0.01 .XXX ±0.005
THIRD ANGLE PROJECTION
UNLESS OTHERWISE SPECIFIED: DO NOT SCALE DRAWING, BREAK ALL SHARP EDGES, FINISH ALL SURFACES: 63
INTERPRET DIMENSIONS AND TOLERANCES PER ASME Y14.5M

RESPONSIBLE ENGINEER	W. LOZIER
ENGINEERING MANAGER	J. KINNEY
DRAWING ORIGINATOR	W. LOZIER
REFERENCE DRAWINGS	
474855	PDMLINK DRAWING
476888	PDMLINK ASSEMBLY
REFERENCE STANDARDS	

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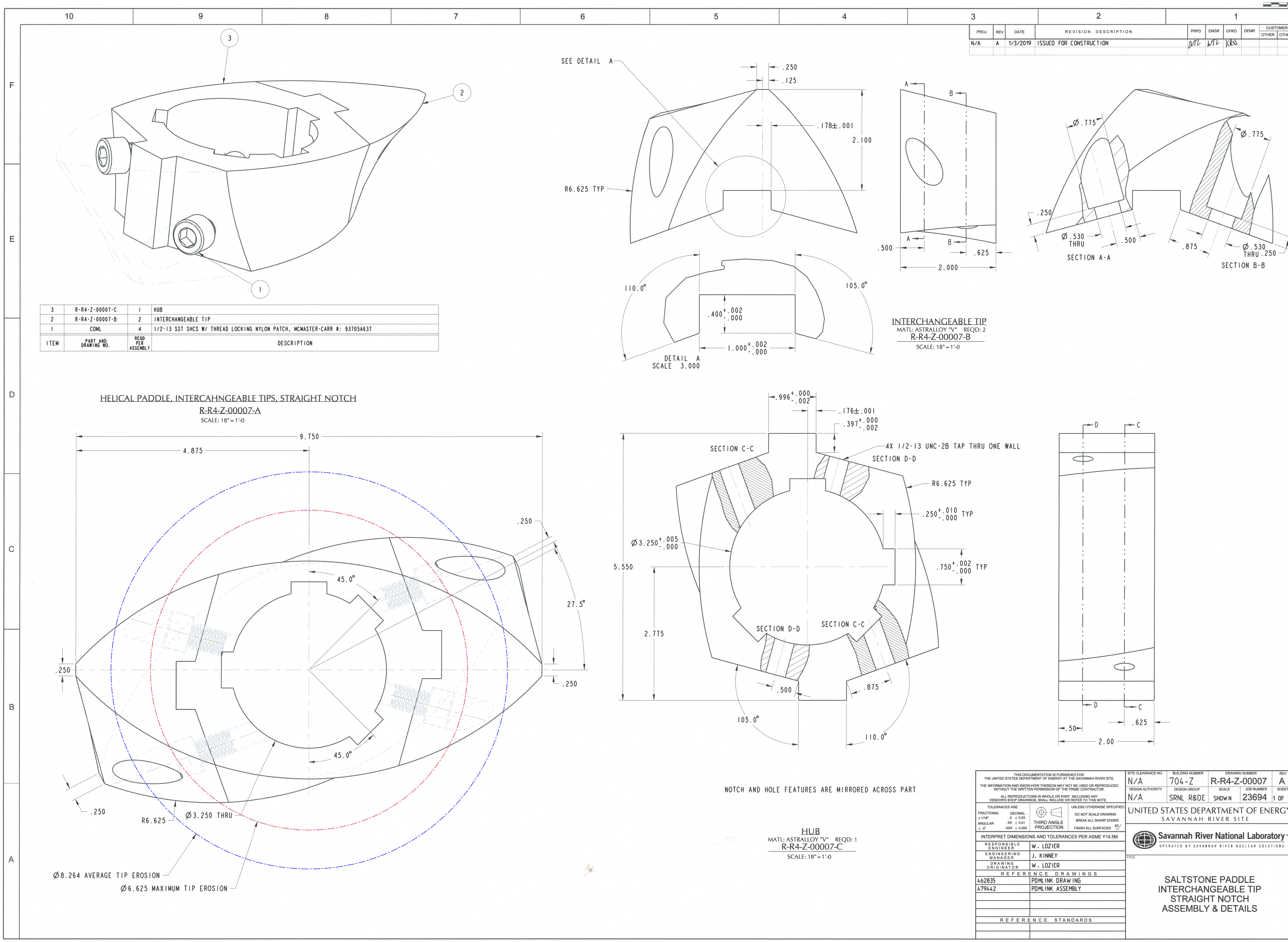
TITLE
SALTSTONE
FLAT PADDLE WITH
INTERCHANGEABLE TIPS, ANGLED
NOTCH
ASSEMBLY AND DETAILS

Appendix B. Recommended Replaceable Tip Angled Notch Hub Design


HELICAL PADDLE, INTERCAHNGEABLE TIPS, STRAIGHT NOTCH
R-R4-Z-00007-A
SCALE: 18" = 1'-0
3 | R-R4-Z-00007-C | 1 | HUB
2 | R-R4-Z-00007-B | 2 | INTERCHANGEABLE TIP
1 | COML | 4 | 1/2-13 SST SHCS W/ THREAD LOCKING NYLON PATCH, MCMASTER-CARR #: 93705A637
ITEM | PART AND DRAWING NO. | REQD PER ASSEMBLY | DESCRIPTION
SEE DETAIL A
R6.625 TYP
DETAIL A
SCALE 3.000
INTERCHANGEABLE TIP
MATL: ASTRALLOY "V" REQD: 2
R-R4-Z-00007-B
SCALE: 18" = 1'-0
SECTION A-A
SECTION B-B
Ø.530 THRU
Ø.775
SECTION C-C
SECTION D-D
4X 1/2-13 UNC-2B TAP THRU ONE WALL
R6.625 TYP
Ø3.250 THRU
Ø8.264 AVERAGE TIP EROSION
Ø6.625 MAXIMUM TIP EROSION
NOTCH AND HOLE FEATURES ARE MIRRORED ACROSS PART
HUB
MATL: ASTRALLOY "V" REQD: 1
R-R4-Z-00007-C
SCALE: 18" = 1'-0
N/A | A | 1/3/2019 | ISSUED FOR CONSTRUCTION
THIS DOCUMENTATION IS FURNISHED FOR THE UNITED STATES DEPARTMENT OF ENERGY AT THE SAVANNAH RIVER SITE.
INTERPRET DIMENSIONS AND TOLERANCES PER ASME Y14.5M
THIRD ANGLE PROJECTION
RESPONSIBLE ENGINEER W. LOZIER
ENGINEERING MANAGER J. KINNEY
DRAWING ORIGINATOR W. LOZIER
REFERENCE DRAWINGS
462835 PDMLINK DRAWING
479442 PDMLINK ASSEMBLY
REFERENCE STANDARDS
704-Z
R-R4-Z-00007
SRNL R&DE
SHOWN
23694
1 OF 1
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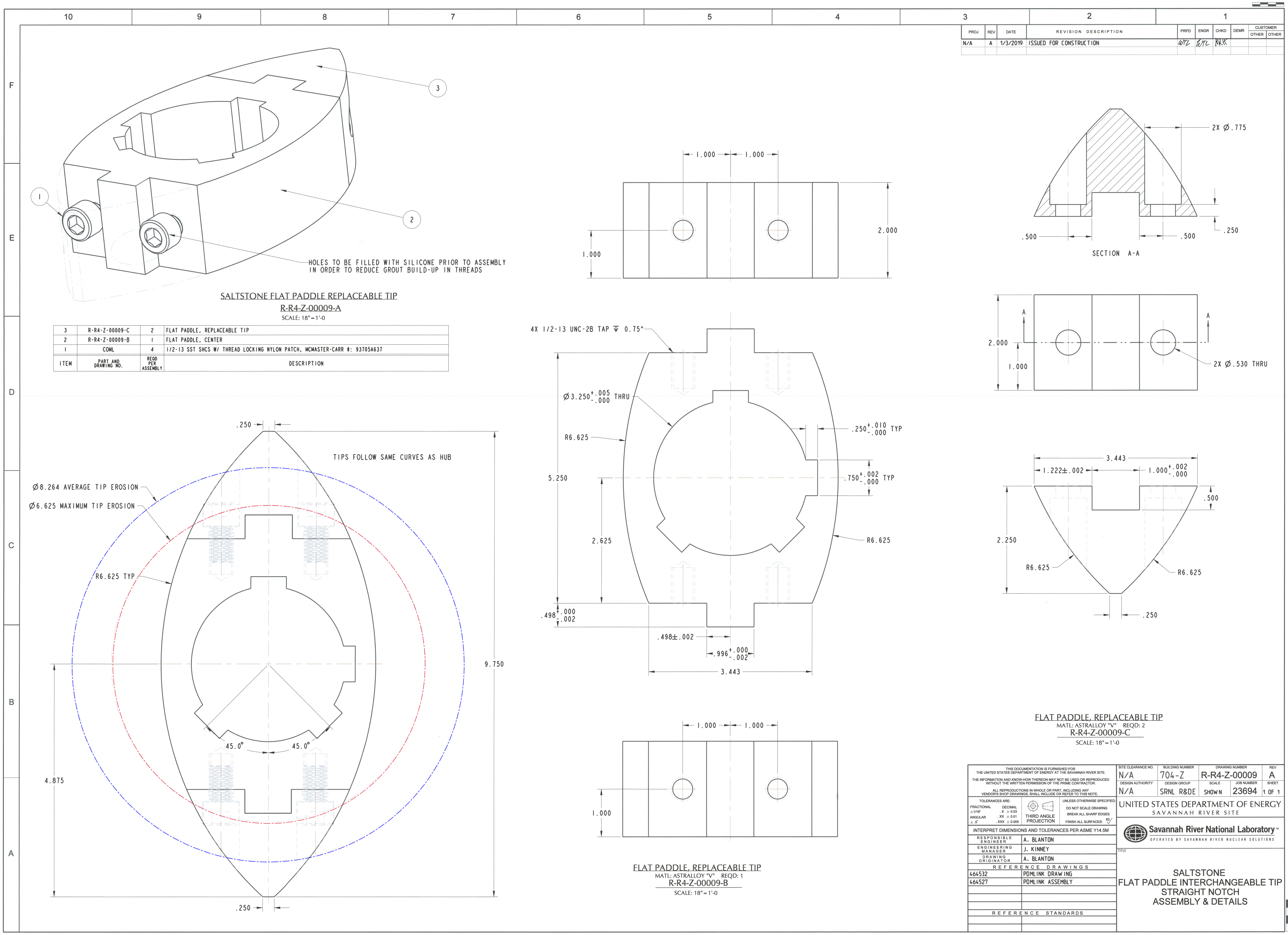
SALTSTONE PADDLE
INTERCHANGEABLE TIP
STRAIGHT NOTCH
ASSEMBLY & DETAILS


SALTSTONE FLAT PADDLE REPLACEABLE TIP
R-R4-Z-00009-A
SCALE: 18" = 1'-0
HOLES TO BE FILLED WITH SILICONE PRIOR TO ASSEMBLY IN ORDER TO REDUCE GROUT BUILD-UP IN THREADS
3 R-R4-Z-00009-C 2 FLAT PADDLE, REPLACEABLE TIP
2 R-R4-Z-00009-B 1 FLAT PADDLE, CENTER
1 COML 4 1/2-13 SST SHCS W/ THREAD LOCKING NYLON PATCH, MCMASTER-CARR #: 93705A637
ITEM PART AND DRAWING NO. REQD PER ASSEMBLY DESCRIPTION
TIPS FOLLOW SAME CURVES AS HUB
Ø8.264 AVERAGE TIP EROSION
Ø6.625 MAXIMUM TIP EROSION
R6.625 TYP
4X 1/2-13 UNC-2B TAP 0.75"
Ø3.250 +.005 -.000 THRU
R6.625
.250 +.010 -.000 TYP
.750 +.002 -.000 TYP
.498 +.000 -.002
.498±.002
.996 +.000 -.002
3.443
2X Ø.775
SECTION A-A
2X Ø.530 THRU
1.222±.002
1.000 +.002 -.000
FLAT PADDLE, REPLACEABLE TIP
MATL: ASTRALLOY "V" REQD: 2
R-R4-Z-00009-C
SCALE: 18" = 1'-0
FLAT PADDLE, REPLACEABLE TIP
MATL: ASTRALLOY "V" REQD: 1
R-R4-Z-00009-B
SCALE: 18" = 1'-0
N/A A 1/3/2019 ISSUED FOR CONSTRUCTION
RESPONSIBLE ENGINEER A. BLANTON
ENGINEERING MANAGER J. KINNEY
DRAWING ORIGINATOR A. BLANTON
REFERENCE DRAWINGS
464532 PDMLINK DRAWING
464527 PDMLINK ASSEMBLY
REFERENCE STANDARDS
704-Z R-R4-Z-00009 A
SRNL R&DE SHOWN 23694 1 OF 1
UNITED STATES DEPARTMENT OF ENERGY
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SALTSTONE
FLAT PADDLE INTERCHANGEABLE TIP
STRAIGHT NOTCH
ASSEMBLY & DETAILS

Appendix C. Calculation M-CLC-Z-000137, Rev. 0

Calculation Cover Sheet

Project/Task	Calculation Number	Project/Task No.
M-TTR-Z-00017	**M-CLC-Z-00137**	**N/A**
Title Structural Analysis of Saltstone Mixer Paddle Replaceable Tips	Functional Classification PS Discipline Mechanical	
Calculation Type ◉ Type 1 ○ Type 2	Type 1 Calculation Status ○ Preliminary ◉ Confirmed	
Computer Program No. ABAQUS ☐ N/A	Version/Release No. V2017	
Purpose and Objective This calculation performs a structural evaluation of a Saltstone Mixer Paddle replaceable tip design.	DC/RO Date _________ NA for SRR work	

Summary of Conclusion

The analysis shows that replaceable tips manufactured from Astralloy "V" and bolted to a hub using two ½-13UNC SHCS per tip are structurally adequate for normal and extreme loads encountered in the Saltstone Mixer. The analysis addresses flat paddles and helical paddles, arranged in any order along the mixer shaft. The recommended installation torque for the bolts is 25 ft-lbs (+/- 3 ft-lbs).

REVISIONS

Rev No.	Revision Description
0	Initial Issue

SIGN OFF

Rev No.	Originator (Print) Sign/Date	Verification/Checking Method	Verifier/Checker (Print) Sign/Date	Manager (Print) Sign/Date
0	Charles McKeel 2/20/2019	☐ Design Check (GS/PS Only) ☒ Document Review ☐ Qualification Testing ☐ Alternate Calculation ☐ Operational Testing	Joshua Flach 2/20/2019	Robert Watkins 2/20/2019
		☐ Design Check (GS/PS Only) ☐ Document Review ☐ Qualification Testing ☐ Alternate Calculation ☐ Operational Testing		

Additional Reviewer (Print Name) James Chan	Signature	Date 2/21/2019
Design Authority - (Print Name) Satish Shah	Signature	Date 2/25/2019
Release to Outside Agency - (Print Name) NA	Signature NA	Date NA

Security Classification of the Calculation

UNCLASSIFIED

RECORD OF REVISION					
REV. NO.	PAGES SUPERSEDED	PAGES ADDED	PAGES DELETED	PAGES REVISED	DESCRIPTION OF REVISIONS
0					Initial Issue

Open Items: NONE

Table of Contents

1.0 Purpose

This calculation performs a structural evaluation of a Saltstone Mixer Paddle replaceable tip design.

2.0 Background/Scope

The 210-Z (Saltstone) mixer (Z-2100000-SSMT-MIX-001) consists of a 75 HP electric motor with a 7.59:1 single ratio transmission driving a 10 inch READVO-Kurimoto co-rotating dual shaft mixer. Each shaft is 3.25 inch diameter by 92 inch long and the two shafts are spaced 10 inch apart. The first 20 inches of shaft length within the mixer consists of a feed auger, followed by 7 helical design mixing paddles and 21 flat design mixing paddles. Each paddle is 2 inch width and made from Astalloy V. The paddles in positions 3-6 experiences excessive tip wear erosion and periodically requires replacement. This requires opening the mixer from the reversing augers end and removal of augers and mixing paddles. This is time and labor intensive to support the SWPF high demand throughputs. The scope of this analysis is a structural analysis of a multi-piece paddle where the eroded regions of a paddle can be replaced without full disassembly [Ref. 2].

3.0 Conclusions

The analysis shows that the Saltstone mixer replaceable tip paddle design described in drawings R-R4-Z-00016 [Ref. 16] and R-R4-Z-00013 [Ref. 20] manufactured from Astralloy "V" and bolted to a hub using two ½-13UNC SHCS are structurally adequate for normal and extreme loads encountered in the Saltstone Mixer. The analysis addresses flat paddles and helical paddles, arranged in any order along the mixer shaft. The recommended installation torque for the bolts is 25 ft-lbs (+/- 3 ft-lbs).

4.0 Inputs and Assumptions

4.1 <u>Description of Mixer</u>

The saltstone mixer (Z-210000-SSMT-MIX-001) is classified as Production Support (PS), per U-FCD-Z-00027, Rev 2 [Ref. 6]. The system is required to blend liquid and dry materials and convey them to the disposal vault. The Saltstone Mixer mixes grout composed from precise amounts of premix dry feed (fly ash, slag and cement) with the salt solution from the SFT and/or SSRT System [Ref 8]. The average Saltstone formulation consists of 47 wt% salt solution and 53 wt% cement/slag/flyash blends [Ref. 9]. The mixer output maximum is 180 gpm. The mixer control logic alarms when mixer speed drops below 200 rpm and at 100 rpm, or if the required motor power consumption exceeds 50 Hp.

Drive System

Motor: 75 Hp, 760 rpm to 2200 rpm capability

Drive Unit: Dodge reducer, 7.59-to-1 gear reduction

Transmission: Dual output Readco transmission.

The system operates at 238 rpm, as the motor is driven at a constant 1800 rpm [Ref. 9]

Rotary Mass Properties (Information Only, not used in calculation, info from T-CLC-Z-00019 [Ref 23]):

Paddle Shaft and transmission: 13,000 lb-in^2

Dodge Reducer: 1,400 lb-in^2

Motor: 864 lb-in^2

The basic layout is shown in Figure 4-1 and 4-2.

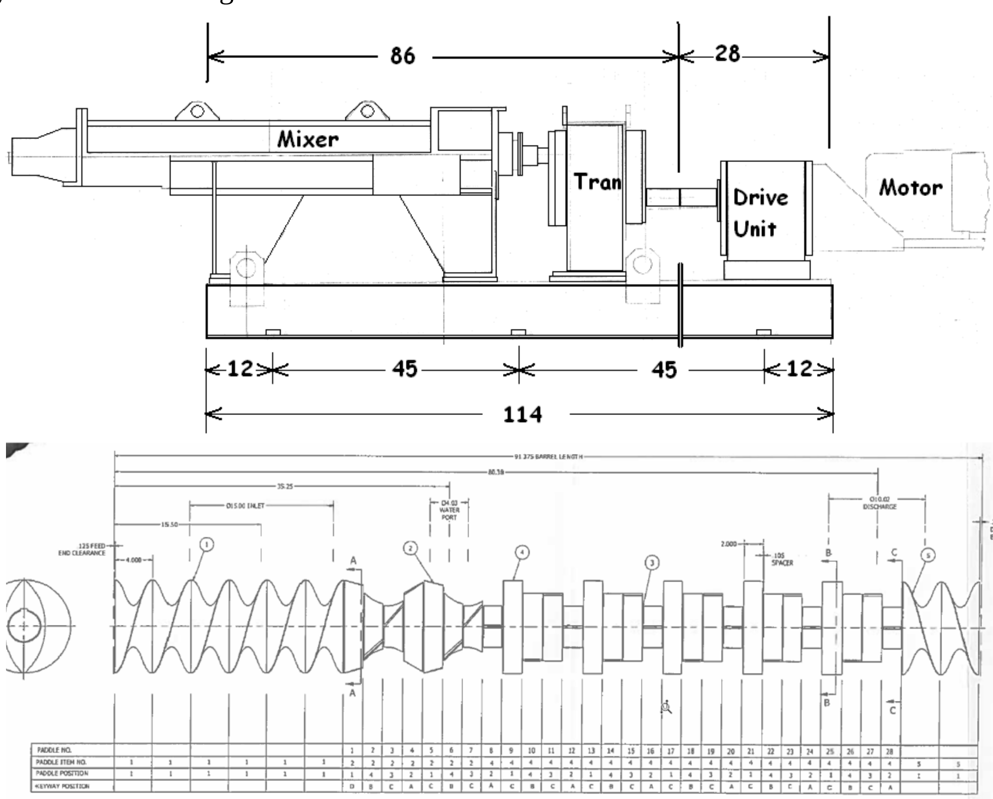


Figure 4-1 Elevation View of Mixer (from T-CLC-Z-00019, Ref. 23), and Shaft [Ref. 2]

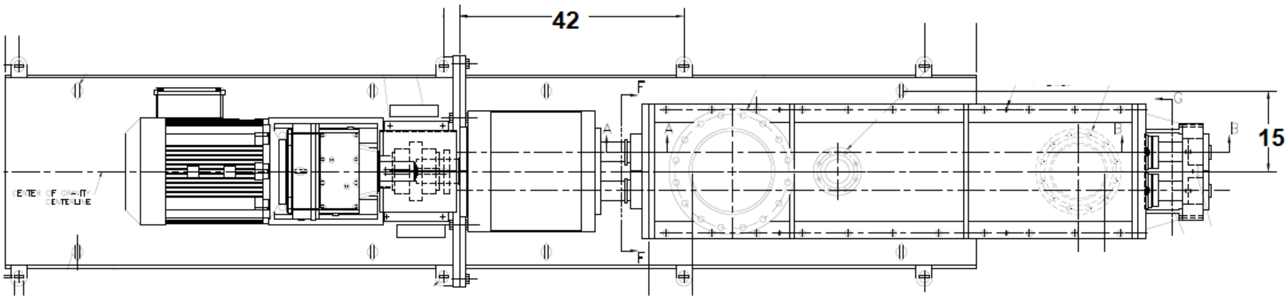


Figure 4-2 Plan View of Main Glovebox (modified image from READCO KURIMOTO, Sketch Supplied by Design Authority Satish Shah)

4.2 Description of Paddles

The paddles evaluated in this analysis include the flat paddle (Figure 4-3) and the helical paddle (Figure 4-4). The helical paddles are also called the twisted paddle within this calculation   The paddles are made from Astralloy V material and consist of a hub section and two tips. The tips are bolted on using ½ -13UNC SHCS.

Attachment Screws
Size = 1/2-13 SST SHCS, MC # 93705A637
Nominal Unthreaded area $=A_b = \pi*0.5^2/4 = 0.196$ in^2
Tensile Area = 0.142 in^2 [Ref. 10, Table 6]
Shear Stress Area = 0.126 in^2 [Ref. 10, Table 6]
Material = 18-8 SST (304) [Ref. McMaster-Carr]
Dimensions: Per ASME B18.3 [See Attachment B]

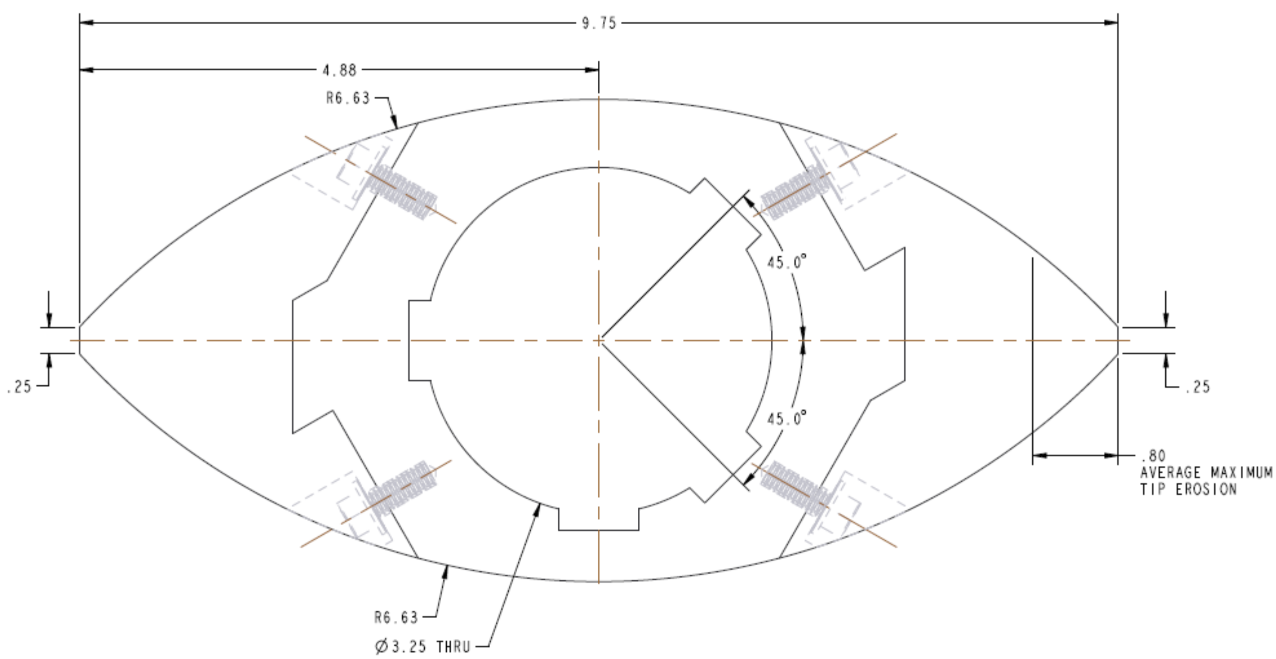


Figure 4-3 Mixer Paddle with Replaceable Tip (Flat Paddle), Per Reference [16]

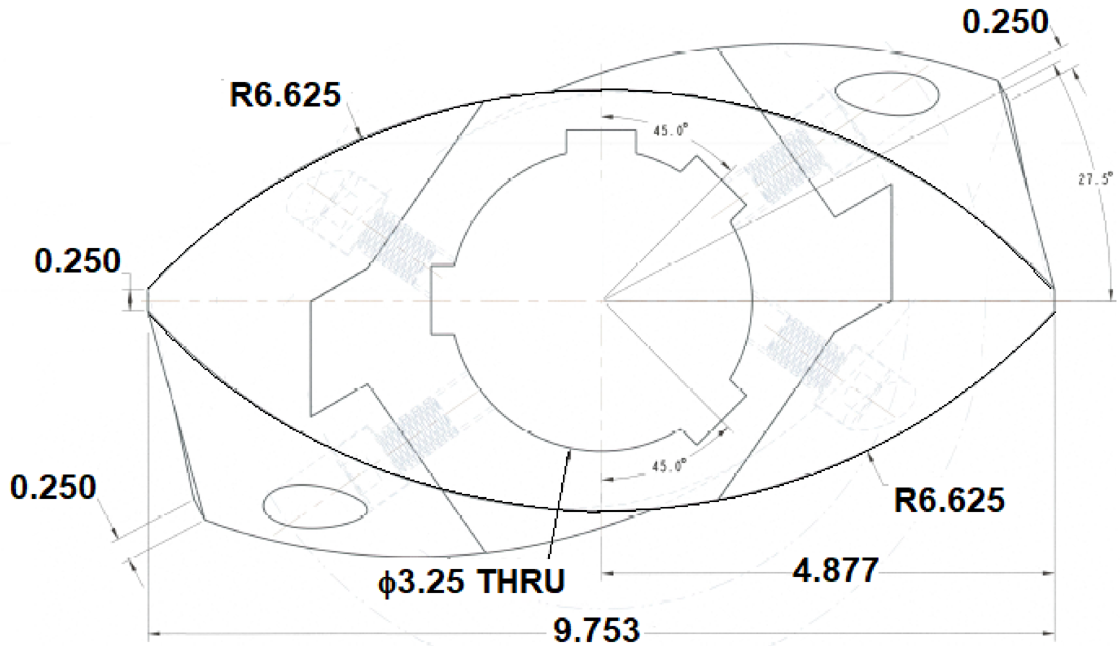


Figure 4-4 Mixer Paddle with Replaceable Tip (Helical Paddle, eg, "Twisted")

4.3 Material Properties

4.3.1 Paddle Hub and Tip, Astralloy "V"

E_{astly} = Young's Modulus = 29.0×10^6 psi [Ref. 12*]

For Plate Material

Yield Stress =	Sy = 157,000 psi	@ 70°F	[Ref. Attachment A]
Tensile Stress =	St = 241,000 psi	@ 70 °F	[Ref. Attachment A]
Charpy	=22 ft-lb	@ 70 °F	[Ref. Attachment A]
Elongation	= 12% (for size 2 in)		[Ref. Attachment A]

For Bar Material

Yield Stress =	Sy = 153,000 psi	@ 70°F	[Ref. Attachment A]
Tensile Stress =	St = 175,000 psi	@ 70 °F	[Ref. Attachment A]
Charpy	=27 ft-lb	@ 70 °F	[Ref. Attachment A]
Elongation	= 15% (for size 2 in)		[Ref. Attachment A]
Thermal Expansion	$\alpha = 6.6^{e-6}$ in/in/°F		[Ref. 12*, Figure 2.3.1.0]

Astralloy V features uniform hardness throughout the bar (340 BHN) to resist abrasion and promote slidability, and toughness to absorb impact without cracking.

- Maintains toughness and hardness without becoming brittle at temperatures down to less than minus 40°F (-40°F).
- Hardenability will reach a range of 550 BHN by impact or sliding action, without deformation or brittleness. * Astralloy-V's small oxide surface quickly disappears, exposing a slick, durable surface.

*Elastic Modulus and Thermal expansion properties are from AISI 4330 and 4340, which have similar chemistry and fine-grain structure.

4.3.2 Socket Head Cap Screw

The screws are socket head cap screws. 18-8 Material, per McMaster-Carr. The following properties are use in this analysis, and are deemed valid across the full actual operating range of the mixer.

$E_{SST\text{-}bolt}$ = Young's Modulus	$= 28.3\times10^6$ psi	[Ref. 3 Table TM-1]
Yield Stress Basis =	Sy = 30,000 psi	[Ref. 3 Table 3]
Tensile Stress =	St = 75,000 psi	[Ref. 3 Table 3]
=	St = 70,000 psi	[per McMasterCarr, see att B]
Elongation	ε = 30%	[Ref. 17, Table 2]
Thermal Expansion	α = 8.6e-6 in/in/°F@ 100°F	[Ref. 3 Table TE-1]
	α = 8.8e-6 in/in/°F@ 150°F	

4.3.3 Grout Mixture

Based on assumption #1, the cement/slag/flyash with hydration has a mechanical behavior typical of common construction mortar.

Density = 130 pcf [minimum density, per Ref. ASCE 7-2010 [Ref 27] Table C3-2]

A maximum value of 150 pcf is used for analysis.

ρ = 150 pcf = 0.0868 lb/in^3

Viscosity, μ= 0.15 to 0.46 N-s/m^2 [Ref. 4, 7]

Yield = 5 to 36 N/m^2 < 0.005 psi (1 N/m^2 = 0.000145 psi) [Ref.7]

4.4 Assumptions and Analysis Basis

The analysis basis for this calculation is:

(1) The grout mixture of 47 wt% salt solution and 53 wt% cement/slag/fly ash blend is consistent with common construction type cement, such that standard mechanical properties from Civil Construction practices can be used. This assumption allows a referenceable basis for density and viscosity. The overall results will be shown not sensitive to this basis.

(2) The pre-mix (cement/slag/flyash) is a controlled, homogeneous-like mixture free from large metallic debris (like bolts, nuts). This allows the exclusion of non-compressible debris (hardened metal rods, large rocks, or similar) under normal operating conditions that could cause a single paddle to experience full motor torque. This assumption is valid based on the controls cited in reference [6]. Hard debris will be considered as a one-time accident condition loading.

5.0 Analytical Methods and Acceptance Criteria

5.1 Methods

The loads on a mixing paddle are determined based on a combination of methods. The best estimate of load is based on the actual paddle speed and the resistance properties of the mixture. Other methods are used which consider full motor power being directed to select paddles, or basing the single paddle load on the strength of the hub connection.

Capacity of the bolted and keyed paddle tip is based on a combination of hand calculations, empirical relations and finite element analysis. The FEA is performed using ABAQUS, version 2017 [Ref. 11].

5.2 Acceptance Criteria

The following acceptance criteria is derived with a goal to demonstrate adequate design life (based on stresses, not erosion. Erosion addresses elsewhere) for the paddle tip and for the bolting. This includes normal ASME code stress criteria, fatigue criteria, and fracture mechanics.

Allowable Stress for Paddle

Allowable stresses are developed based on the stress analysis methods of ASME III-NB, Figure NB-3222-1 [Ref. 21]. The Astralloy material and stress allowables are not listed in ASME-Section II Part D, so the allowable stresses are developed based on ASME II, Appendix 2 [Ref. 3]. ASME determines the allowable stress intensity of a material as a function of temperature based on the minimum value from three different criteria. These are:

(1) 1 /3 of the material tensile strength at room temperature
(2) 1.1 /3 of the material tensile strength at above design temperature
(3) 2/3 of the room temperature yield strength.

For the Astralloy material, rule #1 controls. Sm = 1* 175 ksi / 3 = 58.3 ksi. Stress Criteria for normal conditions is shown in Table 5-1. For accident conditions, the allowable stress is 70% of ultimate for primary stresses, and no limits imposed on secondary or peak, other than per fracture mechanics.

Table 5-1 Acceptance Stress Criteria for Normal Operation Conditions

	Stress Type Classification[note a]			
	Pm	Pl + Pb	Pl+Pb +Q	Peak
Stress Limit	Sm = 58,300 psi	1.5Sm = 87,500 psi	3*Sm = 175,000 psi	Per Fatigue

a. P_m is the primary membrane stress. P_L is the local primary membrane stress. P_b is the primary bending stress. Q is the secondary stress intensity, as defined in ASME III-NB.

Allowable Stress in Bolting

The basic stress limits for service loadings on bolting will be per ASME III-NF (Supports) [Ref. 22]. Preload is excluded from these limits, per ASME PCC-1, Section 10. For preload, a maximum of 80% of yield is common.

Tensile Limit = Ft $< \frac{Su}{3.33} = \frac{70{,}000\ psi}{3.33} = 21{,}000\ psi$ ASME III-NF-3324.6(1)

Shear Limit = Fv $< \frac{0.62*Su}{5} = \frac{0.62*70{,}000\ psi}{5} = 8{,}700\ psi$ ASME III-NF-3324.6(2)

$$\text{Combined Tension and Shear} \quad \left(\frac{\text{ft}}{Ft}\right)^2 + \left(\frac{\text{fv}}{Fv}\right)^2 < 1$$

5.2.1 Fatigue

The primary stress limits are sufficiently low that typical peak stresses that exist will not create a fatigue issue. When a part has structural discontinuities that do produce high peak stresses, or when the high number of cycles are specified, then additional fatigue analysis is needed.

Stainless Steel Bolt
Per ASME III-1, Mandatory Appendix I, the endurance stress limit for the 304 stainless steel used for the bolts is 13,600 psi. A stress concentration factor of 4.0 will be used (threads). This stress limit is applicable to the alternating stress component imposed on the bolt during operation.

Astralloy V Material
The fatigue concern in the paddle tip is at the notched features used to lock the tip to the hub. This notch is similar to the existing key-way notch used to lock the paddle hub to the mixer shaft. The keyway notch at the tip to hub connection is angles, thus potentially presenting a slightly sharper notch. However, a relief radius of 0.13 inch (3.3 mm) is used to alleviate stress concentration. Per Shigley [Ref.18], this results in a notch sensitivity factor of 0.95. Thus Kf ~ Kt (no reduction). This fact will be used in evaluating the FEA stress output. Since the actual part with the actual relief radius is modeled, the above information indicates that no additional stress intensification should exists, beyond what is represented within the adequately meshed FEA model.

The performance goal for fatigue life evaluation is > 10E6 cycles. Per Shigley [Ref. 18] and Juvinall [Ref. 19], the endurance strength of alloy steels for infinite life is at least 40% of the material's ultimate strength.

S'e = 0.4* 175 ksi = 70 ksi

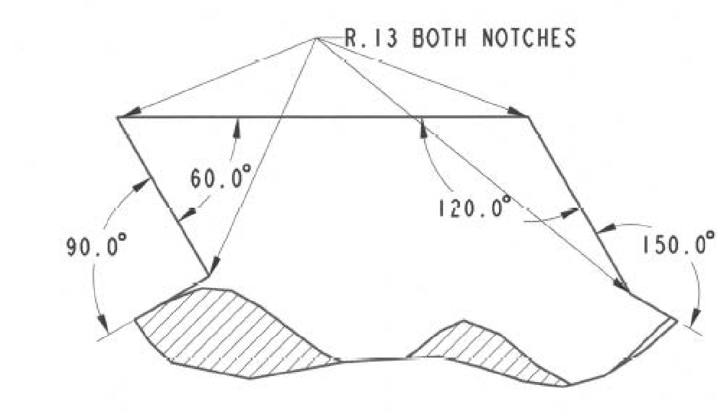


Figure 5-1 Detail of Notch Relief

To apply the above endurance limit, both Reference 18 and 19 have modification factors to address material effects, surface condition, stress concentration, environment, size, shape, and loading speed. These are accounted in the Marin equation shown below:

$$Se = k_a k_b k_c k_d k_e k_f S'_e$$

Where: S'e = 70 ksi (per above)

These terms are quantified below:

Ka = 1.34 * 240$^{(-0.085)}$ = 0.84	1.0 is for smooth surface, use 0.84 for ground surface, per EQ (6-19) of Ref [18].
Kb = 1.0	Both Kb and Kc deal with the type of loading and the size of the component. Specifically, these relate to how quickly the peak stress decays with distance (relative to the total distance of the part or potential fracture plane) compared to stress distribution across the section of the rotating beam test specimen used to develop fatigue data. For this case, the region of high stress (at 95% of max) is a much smaller region vs the rotating beam specimen used for baseline fatigue strength (see Section 6.4.3)
Kc = 1.0	
Kd = 1.0 (70°F)	
Ke = 0.7 (99.99% reliability)	Fatigue data is given as "median" strength. ASME Code allowables are 95 percentile. For maximum reliability, a 99.99% reliability factor is used (Table 5 of Ref 18].
Kf	Notch concentration factor. = 1.0 for this analysis, since actual stress at notch determined via FEA.

Combining all terms, the endurance stress level used for acceptance criteria is:

Se = 0.84 x 1.0 x 1.0 x 1.0 x 0.7 x 1.0 x 70,000 psi

Se = 41,000 psi

5.2.2 Fracture Strength (Info Only)

The fracture toughness of the Astralloy can be determined from its Charpy V-notch data.

$$\left[\frac{Kic}{\sigma_y}\right]^2 = 5\left[\frac{CVN}{\sigma_y} - 0.05\right]$$

$\sigma_y = 153\ ksi$

Minimum Charpy is for Bar Material: CVN = 22 ft-lb

$$\left[\frac{Kic}{153}\right]^2 = 5\left[\frac{22}{153} - 0.05\right] \Rightarrow Kic = 105\ ksi\sqrt{in}$$

The fracture toughness is computed for reference only. With the design of the paddle tip key way not having sharp notches, the fracture stress demand (Ki, shown below) will be low compared to the Astralloy-V toughness.

Fracture Stress Intensity demand is expressed in the form:

$$Ki = Stress\sqrt{\pi a}$$ [Ref API-579, Appendix C]

where stress is the computed stress at a hypothesized crack site, and "a" is depth of crack. Based on the relatively high fracture toughness, and the absence of any crack-like features, fracture is not a significant concern.

6.0 Analysis

6.1 Operating Loads on Paddle Tip

The mixer operates at approximately 180 rpm (100 to 290 rpm, by a VFD). Looking south, the paddles spin clockwise. The paddles are spinning through the grout media, which causes a pressure load to be applied to the leading face of each paddle. There are 28 paddles on each shaft, with each shaft end also including a feed auger (or reversing auger at opposite end). The majority of the wear occurs on the first 7 paddles, where the water is initially added.

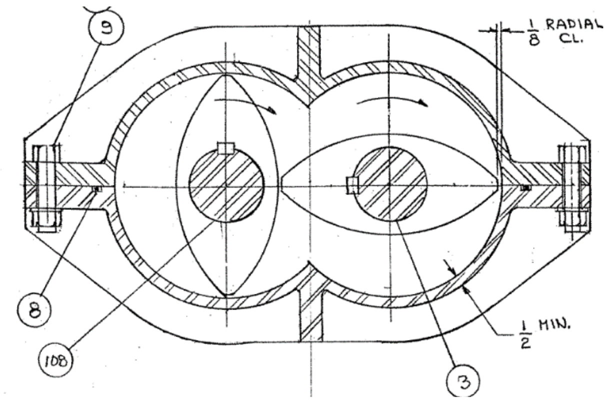
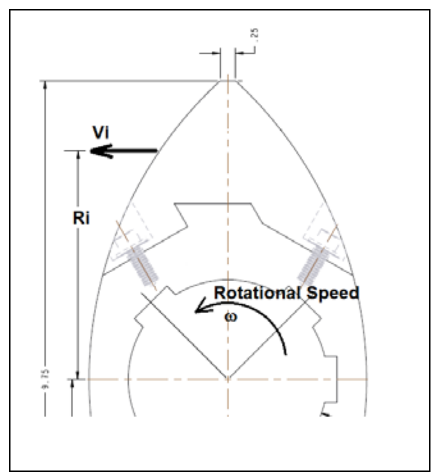


Figure 6-1 Schematic of Mixer Paddles

6.1.1 Paddle Rotation and Linear Speed



As the mixer shaft rotates, the paddle velocity varies with radius by the following relation

Vi = ω·Ri

Where:

Vi = velocity of paddle at position Ri, (inch/sec)

ω = rotational speed (radians/sec)

Ri = Distance from shaft centerline (inch), from 0 to 4.875"

Convert speed in rpm to rad/sec

$$\omega = 238\frac{rot}{min} \cdot \frac{2\pi rad}{1\,rot} \cdot \frac{1\,min}{60\,sec} = 24.92 \text{ rad/sec}$$

Radius	Rotational Speed	Linear Velocity Equation	Linear Velocity
(3.25 inch)/2 = 1.625 inch	238 rpm	=24.92 r/sec*1.625 inch	40.5 in/sec
2.275 inch	= 24.92 rad/sec	=24.92 r/sec*2.275 inch	56.7 in/sec
3.25 inch		=24.92 r/sec*3.25 inch	81.0 in/sec
3.575 inch		=24.92 r/sec*3.575 inch	89.1 in/sec
4.875 inch		=24.92 r/sec*4.875 inch	121.5 in/sec

6.1.2 Paddle Forces Based on Mixing Load

The augers at the front end of the mixing process push the dry feed into the paddle section. The liquid (salt solution) is added from the top after approximately three to five paddles downstream to make the grout mixture. The first few paddles have a slant on the front and back faces (helical paddles) and the remainder are flat paddles. Each of these paddles will experience a pressure on the leading face, as the paddle pushes its way through the grout mixture. The force from this pressure is the summation of the frontal drag forces and the viscous friction forces.

<u>Drag Forces</u>

The basic equation for drag force is:

$$\mathrm{Fd} = \frac{1}{2} C_d \cdot \rho \cdot A \cdot V^2$$

Where:
- Cd = drag coefficient
- ρ = density of media flowing around plate
- A = frontal area
- V = local velocity difference between plate and media

At very low Reynolds number (less than ~ 3), the resistance is proportional to velocity, which means the drag coefficient is inversely proportional to the Reynolds number. Above a Reynolds number of at least a few hundred, the drag coefficient is a fixed value, between 1.0 and 2.0.

$$\mathrm{Re} = \frac{Inertial\ Resistance}{Viscous\ Resistance} = \frac{\rho V b}{\mu}$$

For a 2 inch plate width, and a representative velocity of 100 in/sec (prev page), the Reynolds number is:

$$\mathrm{Re} = \frac{0.0868 lbm/in^3 \cdot\ 100\ in/s\ \cdot 2\ inch}{0.46\ N \cdot s/m^2} * \frac{1N}{0.22481 lbf} * \left(\frac{39.37 in}{m}\right)^2 * \frac{lbf \cdot s^2}{386.4 lbm} = 670$$

For this flow regime, the force is determined based on the drag equation:

Cd = 1.2 for flat plate moving at Reynolds number > 100, use 2.5 (see discussion below) [Ref 5].

V = velocity, varies from center out to tip,

$$\mathrm{F} = \int_{r\_inner}^{router} \frac{1}{2} C_d \cdot \rho \cdot width \cdot V^2 dr$$

$$\mathrm{A} = \int_{r\_inner}^{router} \mathrm{width} \cdot dr$$

Because the paddles are arranged in a manner the prevents grout from freely flowing around both edges of the front face, the drag coefficient on the front face is taken as 2.0, which is consistent with a complete change in the grout momentum [Ref. 5]. To address negative pressure on the back face of the paddle, an additional 0.5 is added, for Cd = 2.5.

The torque required at the shaft is determined based on the integration of the drag force on the incrementalized paddle face and the radial position of each increment. This is shown in Table 6-1.

Table 6-1 Computation of Single Paddle Forces at 238 rpm Shaft Speed.

Radius Coord (inch)	increment (inch)	velocity (in/sec)	area (in2)	Force (lbs)	Torque (in-lb)
1.625	0.217	40.50	0.433	0.200	0.324
1.842	0.217	45.90	0.433	0.256	0.472
2.058	0.217	51.30	0.433	0.320	0.659
2.275	0.217	56.70	0.433	0.391	0.890
2.492	0.217	62.10	0.433	0.469	1.169
2.708	0.217	67.50	0.433	0.554	1.502
2.925	0.217	72.90	0.433	0.647	1.892
3.142	0.217	78.30	0.433	0.746	2.344
3.358	0.217	83.70	0.433	0.852	2.863
3.575	0.217	89.10	0.433	0.966	3.454
3.792	0.217	94.50	0.433	1.087	4.120
4.008	0.217	99.90	0.433	1.214	4.868
4.225	0.217	105.30	0.433	1.349	5.701
4.442	0.217	110.70	0.433	1.491	6.623
4.658	0.217	116.10	0.433	1.640	7.641
4.875	0.217	121.50	0.433	1.796	8.757
Total				14.0 lb	53.3 in-lb

There are 28 paddles and two shafts. The auger sections add approximately 10% (augers comprise 17% of the length and mixes dry feed, but forces are more due to viscosity vs drag force). Therefore, the total torque is:

Torque at 238 rpm = 53.3 in-lbs * 2 shafts * 28 paddles* 2 tips * 110% = 6,567 in-lb = 547 ft-lb

As a check of this value, the required motor power to sustain the conservatively computed load is:

Shaft Torque = 547 ft-lb
Shaft Speed = 238 rpm
Horsepower = Torque * rpm/5252 = 24.8 Hp

Since the actual motor power is 75 Hp and the system alarms above 50 Hp, the forces computed for a single paddle represents a typical value. Therefore, the force estimate is confirmed, but not bounding. To develop a bounding design, the force on a single paddle is multiplied by 10.

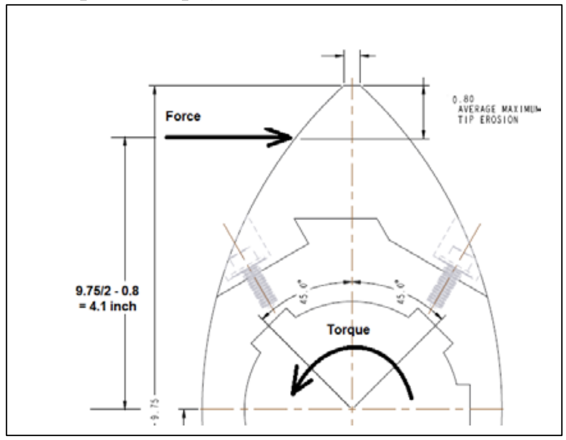
Force on Paddle Tip = 140 lbs

The effective location of this force is determined by equation force to torque:

Effective Location = 533 in-lb/140 lbs = 3.8 inch
(use 4.1 inch radius for force center, conservatively on high side)

6.1.3 Paddle Tip Load Based on Motor Torque

For this case, the entire motor torque is computed as if reacted by a pressure force acting at the tip of just one paddle. The center of the pressure force is taken near the end of the paddle tip (at Distance=4.1 inch). Because the paddles operate in opposing pairs, the minimum number paddles involved is two (2)



Motor power = 75 Hp
Speed = 759 to 2200 rpm.

Motor Torque $= \dfrac{HP * 5252}{rpm} = \dfrac{75 * 5252}{759} = 520\ ft - lb$

Reduction Ratio = 7.59
Geared Torque at Paddle Shaft
= 7.59 * 520 ft-lb = ~ 48,000 in-lb

Moment Arm to Force = 9.75 inch / 2 – 0.8 = 4.10 inch
2 paddles * F * 4.10 inch = 48,000 in-lbs
<u>F = 5,850 lbs</u>

By comparison to the 5,850 lbs, a typical point-load compressive strength of a very high strength rock would be ~ 10 Mpa, = 1450 psi [Ref. 14]. Assuming the maximum size of a stray rock was on the order of 2”x 1”, the maximum single point source load would be less than 3000 lbs (1450 psi * $(2in)^2$).

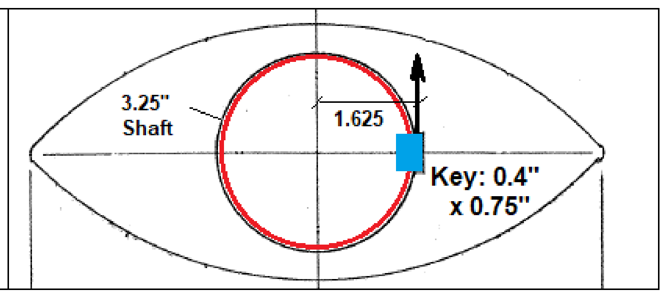
6.1.4 Torque to Shear a Key

A second approach to determine the maximum force acting on a single paddle is to look at the shear force required to shear the shaft key. Various materials have been used over the life of the mixer, and in all cases, the shaft-to-paddle key has withstood the operational loads. In the original system, the bar, paddle, and key materials were (SRS vendor document 13239-NH-21182-1(1)-5) [ref 24] :

Shaft: 316SS Bar (Yield=30,000 psi, Tensile = 75,000 psi)
Paddle: ASTM A743-C8FM (Yield=30,000 psi, tensile=70,000)
Key: 316 SS (per AC37391A-002A), same properties as shaft

Moment Arm = 3.25 inch/2 = 1.625 inch
Shear would occur at the minimum shear plane, which is the plane through the key that is parallel with the paddle sides

Width of Key = W = 0.75 inch (slot radial direction)
Height of Key = H = 0.40 inch (taken as 2 times the 0.20 inch depth of the key into the paddle)



Consider the 70 ksi nominal tensile strength equates to 100 ksi upper bound for actual material (conservative) and that the conversion from tensile to shear is 75% (this is conservative compared to 1/sqrt(3) used in Mises stress theory, since at or near failure, the ultimate shear is generally known to increase above the 1/sqrt(3) for ductile materials)

Limit Shear Stress = 100,000 psi * 0.75 = 75,000 psi
Number of Shear Planes = 2 (= for failure, key must fail on both sides of the paddle)

Max Key Shear Force = Area * Stress = (2 planes * W * H) * 75,000 psi =
= 2 * 0.75 inch x 0.4 inch x 75,000 psi = 45,000 lbs
Maximum Torque at paddle = 45,000 lb * 1.625 = 73,125 in-lb
Force at 4.1 inch = 45,000 lbs * 1.625/4.1 = 17,800 lbs

The operating experience of the mixer shows no history of key damage or failure. This includes the initial operations prior to filtering of the inlet media. Additionally, this load is less than motor capacity, thus not considered as a design load.

6.1.5 Load for Analysis

Based on Fluid Forces:	Force = 140 lbs, Torque = 533 in-lb
Based on Max Motor Torque:	Force = 5,850 lbs
Based on Key Shear:	Force = 17,800 lbs (exceeds max motor capacity)

The maximum motor torque applied to a single set of paddle tip is shown overly conservative, and exceeds the strength to crush a stray rock by at least a factor of 2. The paddle force computation based on shear key damage was shown to exceed the motor torque capacity, thus is discredited.

The 140 lbs per paddle computed from drag force was shown to require 248 HP from the motor if that load were to occur on each paddle face consistently. Since the motor power is only 75 Hp, the 140 lbs force is deemed a conservative, but reasonable design bases. This equates to all motor power being imposed onto 30% of the paddles (8 to 9 paddles). To account for sudden starts and stops, the computed force is rounded to 150 lbs and then doubled.

Therefore, a force value of 300 lbs will be used for design. An accident condition case of 3000 lbs will be used for a one-time accident load condition.

<u>Summary</u>

Normal Operating Force on Paddle Tip = 300 lbs (force center acting at 4.1 inch radius position)
Accident Condition Force = 3000 lbs

For the helical paddles, the 300 lb force will be applied in two directions. (in plane and axial)

6.2 Bolted Connection

The replaceable paddle tips are connected to the paddle hub using two ½-13UNC SST socket head cap screws. The bolted connection works in parallel with the half-dovetail slot. The analysis is performed considering the bolts carry all load.

6.2.1 Estimated Service Load on Bolt

The design load on the paddle tip is used to establish a suitable target preload. The 300 lbs pressure load on the face of the paddle will be considered to the acting perpendicular to the paddle shaft.

Force on Paddle Face = 300 lbs

Prying Moment = 300 lbs * (4.1"-1.524")
= 773 in-lb

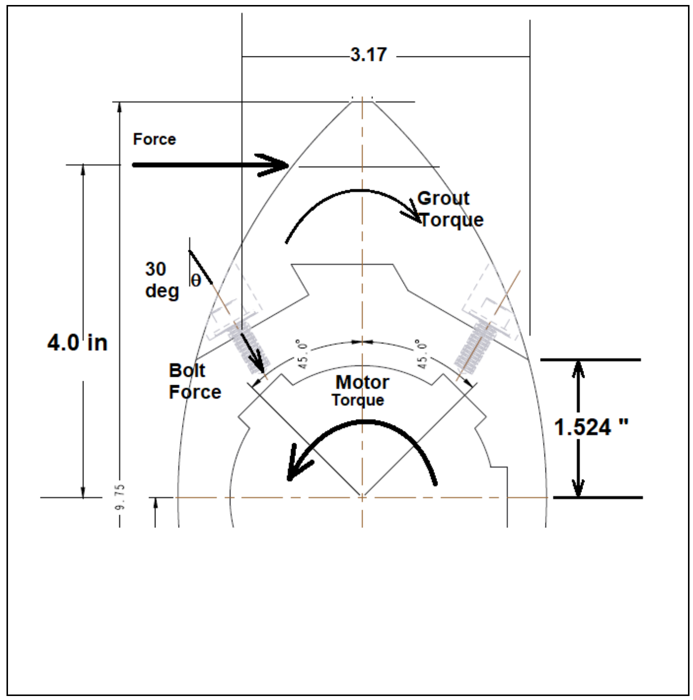
Leverage Moment Arm = 3.17 inch (see Figure)
Required Bolt Reaction (ortho to Force)
= 773 in-lb/3.17 inch = 244 lbs
Bolt Angle = θ = 30 degrees

TENSION

Required Bolt Tensile Force = 244 lbs/cos(θ)
$\boxed{\text{Ft} = 282 \text{ lbs}}$

SHEAR

Bolt Shear Force = Consists of direct shear
+ shear from prying reaction
Direct Shear = 300 lbs*cos(θ)/2 bolts = 130 lbs
Shear Component of Pry Force = 244 lbs*sin(θ)
= 122 lbs
$\boxed{\text{Fv} = 130 \text{ lbs} + 122 \text{ lbs} = 252 \text{ lbs (Flat paddle)}}$



For the helical paddles, an additional force of 300 lbs, in the shaft direction, is distributed on two bolts

Direct shear = $\sqrt{(300 * \cos(30)/2)^2 + (300/2 \text{ bolts})^2}$ = 200 lbs
Shear From Prying = 122 lbs
$\boxed{\text{Fv} = 200 \text{ lbs} + 122 \text{ lbs} = 322 \text{ lbs Helical Paddle}}$

6.2.2 Bolt Stress from Service Loads

The 282 lbs bolt tensile load and the 322 lbs shear load are used to compute stress levels. Bolt tensile and shear stress areas are per ASME B1.1 for the ½ inch bolt.

Bolt Tensile Area = At = 0.142 in^2
Shear Stress Area = Av = 0.126 in^2

$$\sigma t = \frac{Ft}{At} = \frac{282 lb}{0.142 in^2} = 1986\, psi$$

$$\sigma v = \frac{Fv}{Av} = \frac{322\ lbs}{0.126 in^2} = 2556\, psi$$

$$\sigma = \left(\frac{\sigma t}{All\_t}\right)^2 + \left(\frac{\sigma v}{All\_v}\right)^2 = \left(\frac{1986}{21{,}000}\right)^2 + \left(\frac{2556}{8{,}700}\right)^2 = 0.096 \ < 1, \text{ therefore acceptable}$$

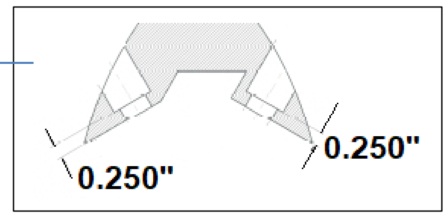
6.2.3 Bolt Preload

The goals of preload are to:

- Ensure sufficient preload on the joint to overcome the expected service loads, such that cyclic loads on the bolt are avoided, accounting for thermal expansion.
- Ensure sufficient thread preload and initial bolt stretch to preclude bolt loosening, but not to exceed yield.

<u>Required Preload for Thermal Expansion</u>
Differential thermal expansion could cause the bolt preload to be lost, leading to the bolt loosening. In the case of the mixer, no temperature delta is expected between the bolt and paddle. Therefore, only differential expansion coefficients between the two components is considered. The effective free length of the bolt includes the ¼ inch section of the paddle tip plus approximately one complete thread (0.0769 inch).

Expected Maximum Temp = 150°F
Expected Minimum Temp = 30°F
Temperature Change = 120°F
Bolt Free Length = 0.25 in + 0.0769 = 0.33 inch
Thermal Expansion Coeff. of Bolt = 8.8 e^{-6} /°F (use 150F)
Thermal Expansion Coeff. of Astralloy = 6.6 e^{-6} /°F
$\Delta\alpha = 2.2\ e^{-6}$



Bolt Length Change = $\alpha\ L\Delta T = 2.2\text{e}-6\ \text{in}/(\text{in°F}) * 0.33\ \text{inch} * 120\text{°F} = 0.00009\ \text{inch}$

Require Bolt Load = $A_{bolt} \cdot E \cdot \varepsilon = A_{bolt} \cdot E \cdot \frac{\Delta L}{L} = 0.142\text{in}^2 \cdot 28.3E6 \cdot \frac{0.00009}{0.33}$ = 1,100 lbs

<u>Minimum Required Preload for Service Conditions</u>
The target preload to overcome service load = 282 lbs tension (use 300 lbs, see section 6.2.1). To avoid slip under the 322 lbs of shear per bolt, a target service load of 975 lbs is used. (0.33 friction coeff, which is 50% of expected for Hard steel on Hard Steel ([Ref 15], page 16).

<u>Maximum Preload, to Avoid Yield</u>
Target Preload to preclude loosening = 80% Yield = 0.80* 30,000 psi * 0.142in^2 = 3,400 lbs

A torque corresponding to 80% of yield should not be performed without a lubricant, or seizing of the soft stainless could occur.

Target Minimum Preload = service load requirement + thermal requirement
Target Minimum Preload = 975 lbs + 1100 lbs = 2,075 lbs

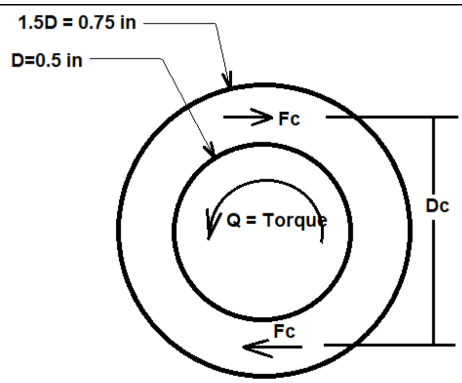
6.2.4 Required Installation Torque

The required torque, Q, to achieve a preload, Fa, is given by [per Ref 19]:

$$Q = \frac{F_a \cdot D_b}{2}\left(\frac{f\pi D_b + L\cos\alpha_n}{\pi D_b \cos\alpha_n - fL}\right) + \frac{F_a \cdot f_c \cdot D_c}{2}$$

where:

f = bolt thread friction = 0.16 +/- 0.02 [Ref. 13]
fc = Bolt Head contact friction,
L = lead angle = 1/13 for 13 thrds/inch = 0.0769 inch
α_n = thread angle = 30 deg [Ref 10]
Dc = mean dia of bolt head



Thread Friction (*f*):
f = 0.16 +/- 0.02 [Ref. 13]

Bolt Head Friction (fc)
For same conditions, this value is ~ 2/3 of thread
[Per Ref. 13]
fc = 0.11
Dc = sqrt avg(0.75^2, 0.5^2) = 0.64 inch*
*The effective location of friction force is weighted toward the OD, approximated by the above.

Lower Bound Torque (lowest preload, low friction)
Use Fa = 2,075 lbs
f = 0.14, fc=0.10

$$\mathrm{Q} = \frac{F_a \cdot D_b}{2}\left(\frac{f\pi D_b + L \cdot cos\alpha}{\pi D_b \cdot cos\alpha - f \cdot L}\right) + \frac{F_a \cdot f_c . D_c}{2}$$

$$= \frac{2211 \cdot 0.5}{2}\left(\frac{0.14\pi 0.50 + 0.0769 \cdot cos30}{\pi 0.50 \cdot cos30 - 0.14 \cdot 0.0769}\right) + \frac{2211 \cdot 0.10 \cdot 0.64}{2}$$

$$\mathrm{Q} = \frac{F_a \cdot 0.0531}{1} + \frac{F_a \cdot 0.032}{1} = \frac{2075lb \cdot 0.0851}{1}$$

Q = 180 in-lb (15 ft-lbs)

Upper Bound Torque (Highest preload, High friction)
Use Fa = 80% Yield = 0.80*30,000* 0.142 in^2 = 3400 lbs
f = 0.18, fc=0.11

$$\mathrm{Q} = \frac{F_a \cdot D_b}{2}\left(\frac{f\pi D_b + L \cdot cos\alpha}{\pi D_b \cdot cos\alpha - f \cdot L}\right) + \frac{F_a \cdot f_c . D_c}{2}$$

$$= \frac{3400 \cdot 0.5}{2}\left(\frac{0.18\pi 0.50 + 0.0769 \cdot cos30}{\pi 0.50 \cdot cos30 - 0.18 \cdot 0.0769}\right) + \frac{3400 \cdot 0.11 \cdot 0.64}{2}$$

$$\mathrm{Q} = \frac{F_a \cdot 0.0649}{1} + \frac{F_a \cdot 0.0352}{1}$$

Q = 340 in-lb (28.4 ft-lbs)

Therefore, an installation torque of 25 ft-lbs will be prescribed, which should provide more than the 2,075 lbs of minimum requires preload.
(10% variation on normal torque wrench allows for 21 to 28 ft-lbs)

<u>Check Max Spec Torque (28 ft-lbs = 336 in-lb), Minimum Friction (per equation above)</u>

$336\mathrm{in} - \mathrm{lb} = \frac{F_a \cdot 0.0851}{1} \Rightarrow F = 3{,}948\ lbs$, Stress=3,948;bs/0.142in2 = 28ksi, less than yield, OK

[Note: Above equation results are rounded, acknowledging the high uncertainty associated with bolts]

6.2.5 Bolt Thread Shear

The Socket Head Cap Screws have a Class 3A thread fit. In terms of thread shear, the external threads on the 30 ksi yield strength bolts are controlling vs the internal threads on the 153 ksi strength Astralloy-V. ASME B1-1, Appendix B is used to evaluate the threaded connection between lid and body.

Nominal Size and Threads/in.	Series Designation	Class	External [Note (1)] Major Diameter Max. [Note (2)]	Major Diameter Min.	Major Diameter Min. [Note (3)]	Pitch Diameter and Functional Diameter [Note (4)] Max. [Note (2)]	Pitch Diameter Min.	Pitch Diameter Tolerance [Note (5)]	UNR Minor Diameter, Max. [Note (6)] (Ref.)	Internal [Note (1)] Minor Diameter Min.	Minor Diameter Max.	Pitch Diameter and Functional Diameter [Note (4)] Min.	Pitch Diameter Max.	Pitch Diameter Tolerance	Major Diameter, Min.
½ – 13 or 0.5000 – 13	UNC	3A	0.5000	0.4891	0.4822	0.4500	0.4463	0.003700	0.4084	0.4170	0.4284	0.4500	0.4548	0.0048	0.5000

<u>For External Thread on Bolt:</u>

Shear Area:

$$AS_s = 3.1416\left(\frac{1}{P}\right)(LE)(D_{1max})\left[\frac{P}{2} + 0.57735\left(d_{2,min} - D_{1,max}\right)\right]$$

P = Thread Pitch (inches per thread) = 1/13 = 0.0769 inch

LE = Length of Engagement = 1 inch - 0.25 inch – 0.077 inch thread relief = 0.67 inch

d_{min} = min major diameter of external = 0.4891 inch

d_{2min}=min pitch diameter of external = 0.4463 inch

D_{1max} = Max Minor diameter of internal = 0.4284 inch

$$AS_s = 3.1416(13)(0.67)(0.4284)\left[\frac{0.0769}{2} + 0.57735(0.4463 - 0.4284)\right]$$

$AS_s = 0.57 in^2$

When the available thread shear area is greater than the bolt tensile area (0.142 in^2) by more than 1/0.577, the thread depth is sufficient to reach the full tensile strength of the bolt.

The minimum Required Thread Depth = 0.67 inch $* \frac{0.142/0.577}{0.57} = 0.29\ inch$

Actual Thread Depth = 0.67 inch Therefore, acceptable.

6.3 FEA Analysis of Split Paddle and Bolts

The mixer paddle tip, paddle hub, and bolted joint were modeled using ABAQUS version 2017. The goal of this model was to evaluate stresses at the paddle tip to hub interface, quantify stress levels at the stress discontinuities between the two parts, and determine the bolt loads during operational conditions. The FEA model consisted of the hub section, the tip section, and the two bolts.

6.3.1 FEA Model Geometry and Mesh

An outline view of the FEA model is shown in Figure 6-2. The FEA model consists of a half section of the hub and the associated tip section. A symmetry boundary condition is applied at the hub cut-plane (at center plane of mixer shaft). Per the drawing (Figure 5-1), the hub-tip keyway corners are nominally radiused at 0.13 inch. In the FEA model, the concave corners are radiused with a minus tolerance, and the convex tips are radiused with a plus tolerance to ensure the two parts have a good fit (Figure 6-3, Figure 6-4).

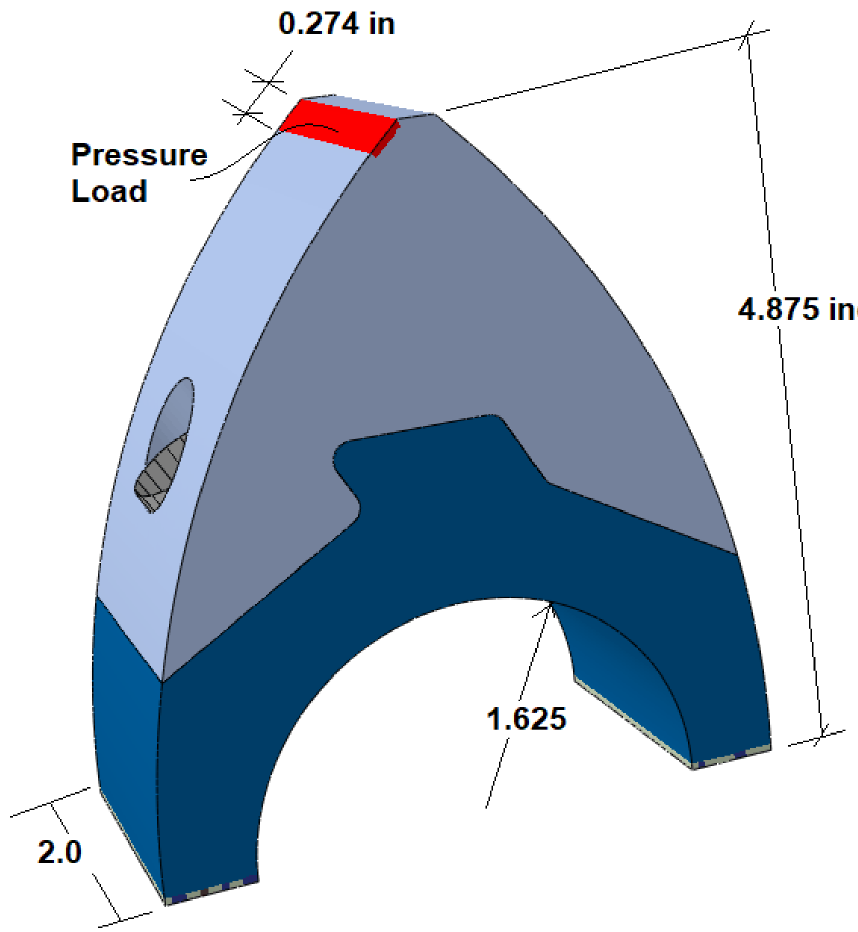


Figure 6-2 FEA Model of Flat Paddle, Showing Dimensions for Hub and Tip Assembly

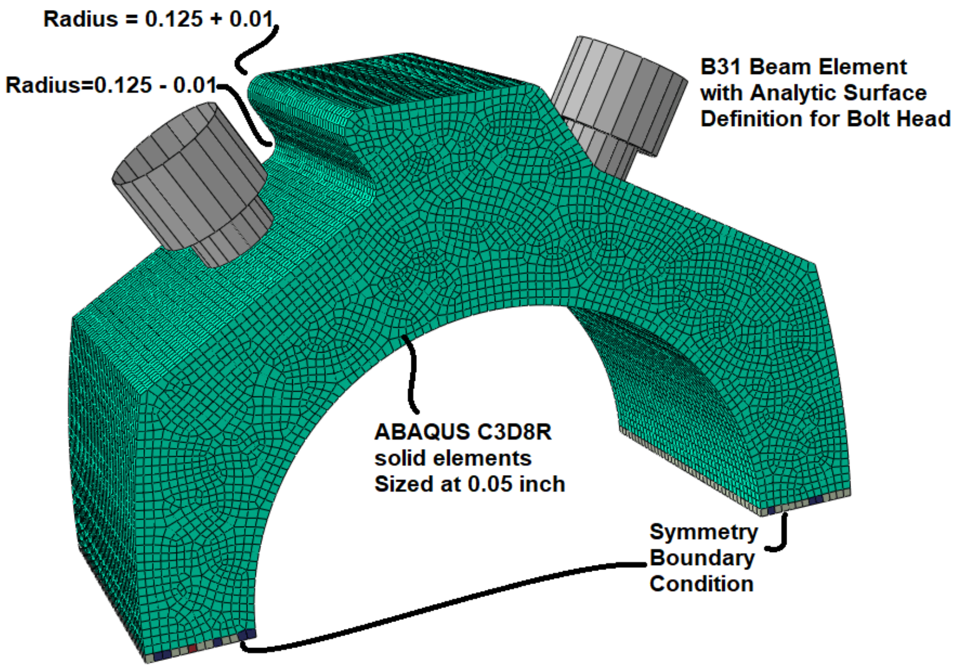


Figure 6-3 FEA Model of Hub Section and Bolts, Showing Element Meshing and Boundary Conditions

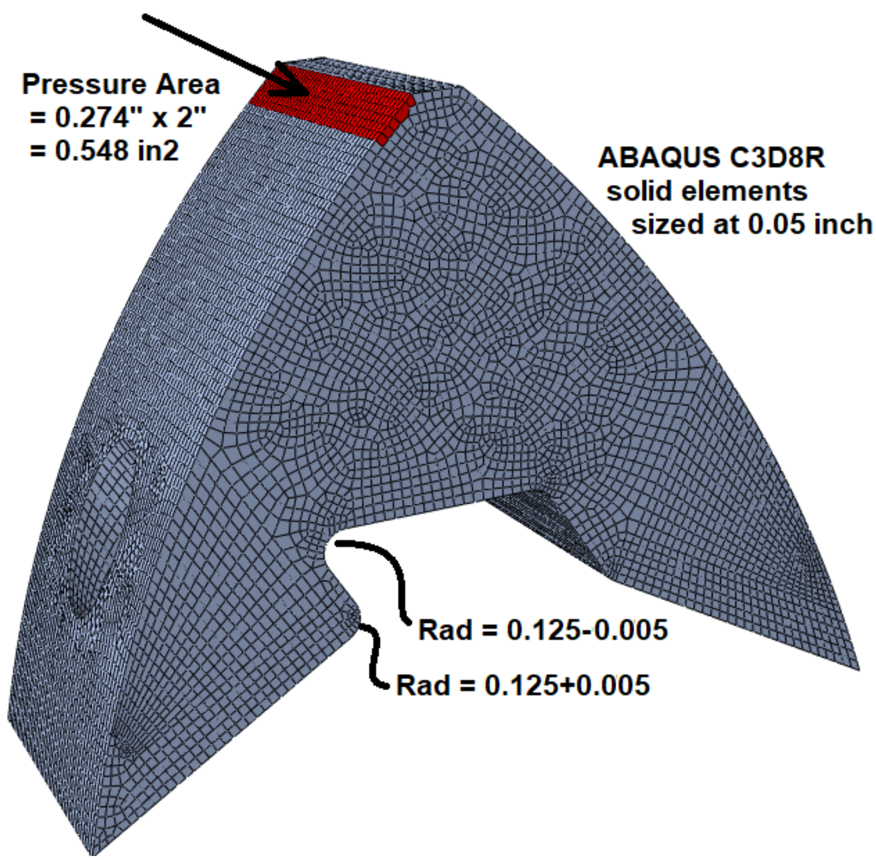


Figure 6-4 FEA Model of Tip Section, Showing Element Meshing and Boundary Conditions

6.3.2 FEA Model Load Inputs

The FEA model loads consists of sequential application of bolt preload followed by paddle pressure load.

Bolt Preload

The preload is applied by modeling the initial position of the bolt where the bolt head is slightly above the contacting surface on the paddle tip. The other end of the bolt is anchored into the hub. Bolt preload is then imposed by applying a thermal contraction to each of the bolts. As the bolt shrinks, all clearances are taken up between the parts and then bolt tension begins to build.

The target preload for the FEA model is based on the 282 lbs of direct service tensile load (see 6.2.1). It's not important that the FEA model exactly achieve this target preload, as the goal is to show how actual service conditions change the bolt load, once the preload is achieved. Figure 6-5 shows the input load history.

Mixing Pressure Load

To maximize the moment and stress reaction from the applied load, the target mixing pressure on the paddle face is concentrated onto a small area near the paddle tip. The computed target load is 300 lbs per paddle face (section 6.1.5). For the FEA model, the applied load is 1600 lbs, to ensure a bounding condition.

Area of Paddle With Applied Load (see Figure 6-4)

Paddle Width = 2 inch (per drawing)

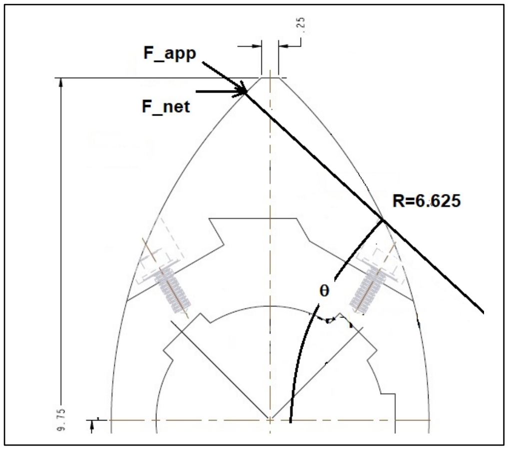
Length of section pressurized = 0.274 inch (see Figure 6-4)

Area = 2" x 0.274 inch = 0.548 in^2

Angle of Paddle Section Pressurized

Distance to Region Center = 4.875 inch – 0.274/2 ~ 4.75 inch

Angle = θ = asin(4.75/6.625) = 45.8 degree



Imposed Pressure = 4265 psi (via *dsload in ABAQUS)

Resulting load = 4265 psi * 0.548 in^2 = 2337 lbs

Fnet = 2,337 lb * cos(45.8) = 1629 lbs

(thus bounding 1600 lbs target)

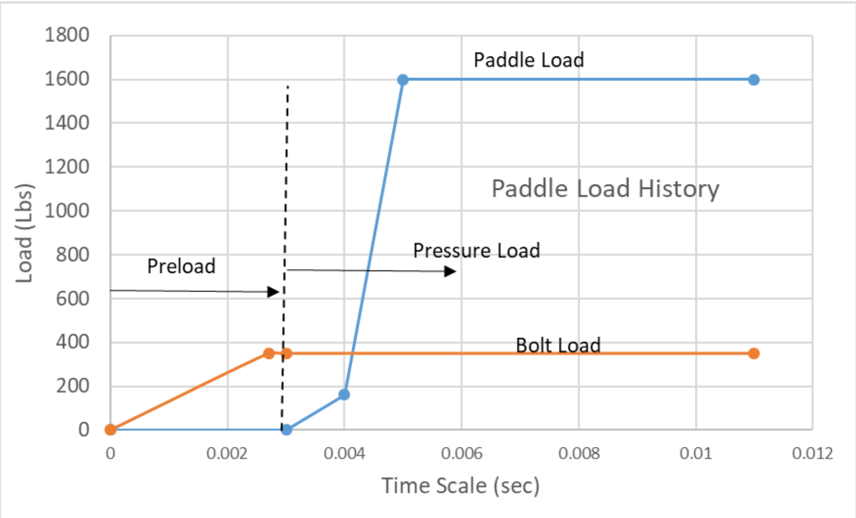


Figure 6-5 FEA Input Load History, Showing Preload followed by Pressure Load

6.3.3 Bolt Load Results

Figure 6-6 shows the bolt tensile load history during the condition of initial pressure and subsequent pressure load. The applied thermal contraction resulted in a preload of approximately 350 lbs on the leading bolt, and 225 lbs on the backside bolt. The difference is due to minor differences in initial clearance between the bolt head and the mating paddle surface. The important result is how the bolt load changes during the subsequent mixing load application.

The load history shows only a 50 lbs load increase on the front side bolt and essentially no change in the bolt tension for the back side bolt. From 0.003 seconds to 0.004 seconds, the paddle load is ramped up to 160 lbs with Figure 6-6 showing zero increase in bolt load. The 50 lbs increase occurs as the paddle face pressure load is ramped from 160 lbs to 1600 lbs. The 50 lbs bolt load change is minor compared to the 3400 lbs load capacity of the bolt (per 80% proof load). Scoping studies also showed that a higher preload would result in essentially no change in bolt load during operation. Therefore, the minimum required preload for these bolts is established at 25 ft-lbs, which will provide a minimum of 2,200 lbs of preload.

Figure 6-7 shows that preload, friction (modeled at 10% friction coefficient in FEA model), and the key slot was sufficient to preclude significant shear loads being imposed on the bolt.

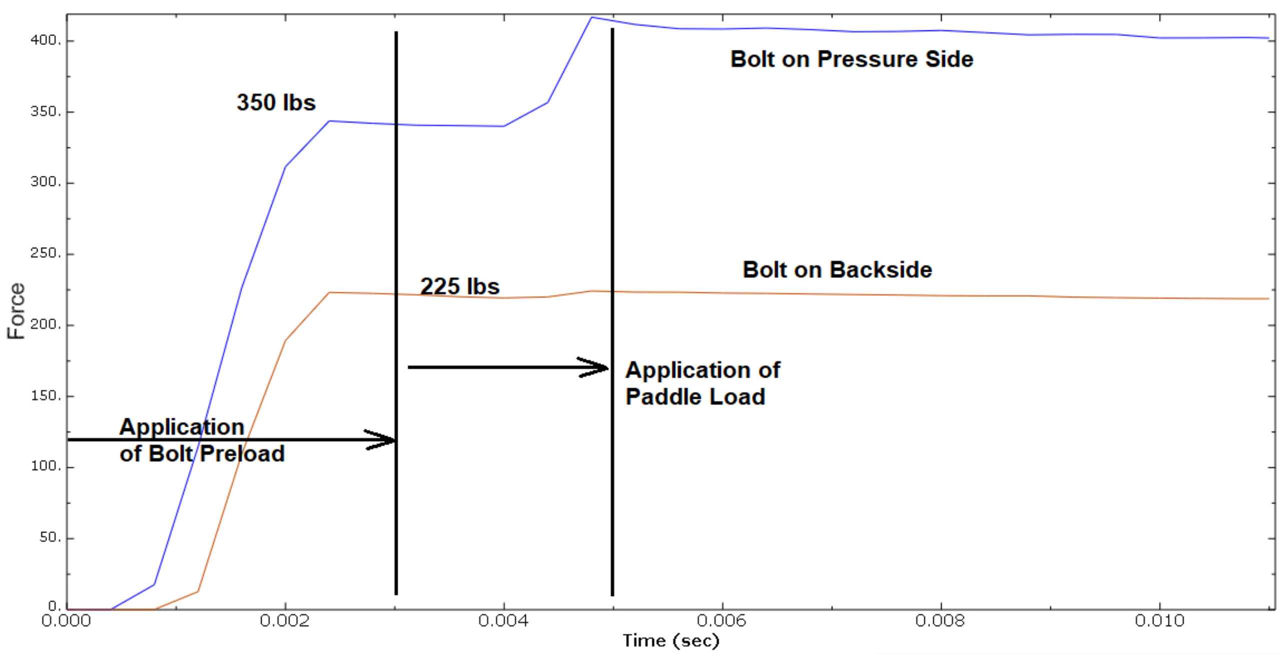


Figure 6-6 Bolt Tensile Load History During Preload and Subsequent Service Load, Showing Essentially no Change in Bolt Tensile Load During Service Condition.

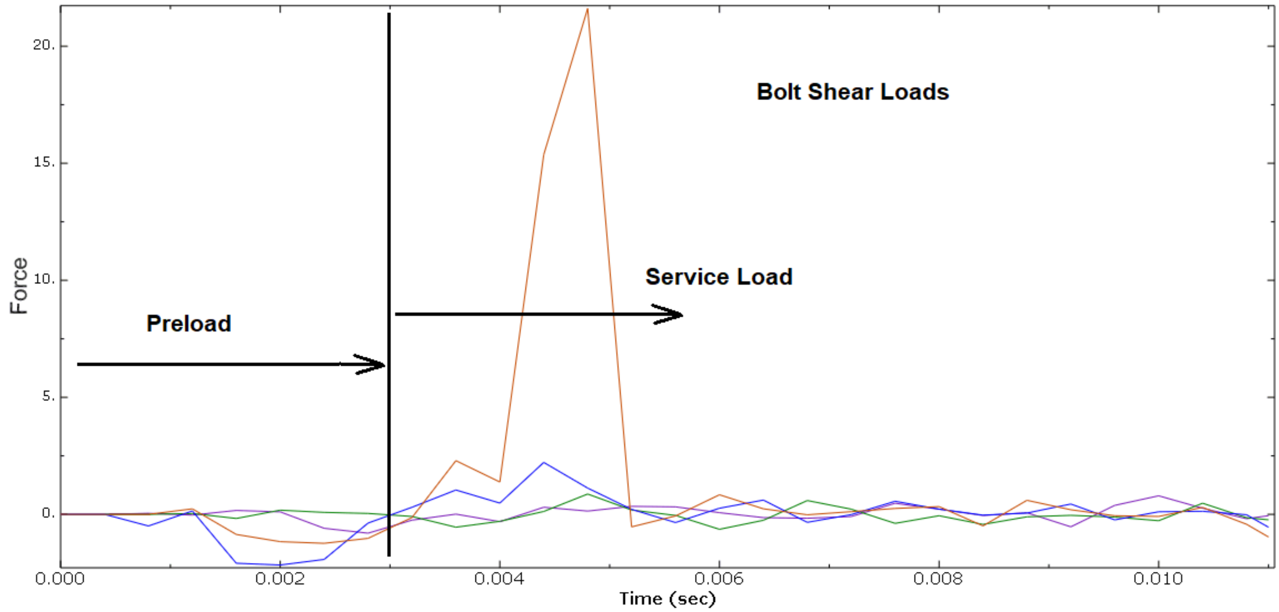


Figure 6-7 Bolt Shear Load History During Preload and Subsequent Service Load, Showing essentially no shear load.

6.4 Paddle Stress

The basic paddle shape has a decade plus history of successful use as a one-piece paddle design. Therefore, the basic elements such as the shaft key interface, the cross-section of the paddle at the hub, and the load capacity of the overall profile is bounded by this successful service. The introduction of the split paddle design, therefore, only requires analysis of the paddle tip to paddle hub interface. The results of interest are the stresses around the bolt hole and the peak stress levels in the notched interface. This interface is a slanted key-way, sloped in manner such that the pressure load during mixing locks the tip to the hub, thus creating a stressed part.

6.4.1 Stresses at Bolting Location

Figure 6-8 show the stresses around the bolt hole are less than 3000 psi. Based on adequate FEA meshing, this includes peak stresses. As confirmed in the bolt load history (no load change during operation), this stress will not vary during the operational loads.

Maximum Stress at Bolt Hole: 3000 psi

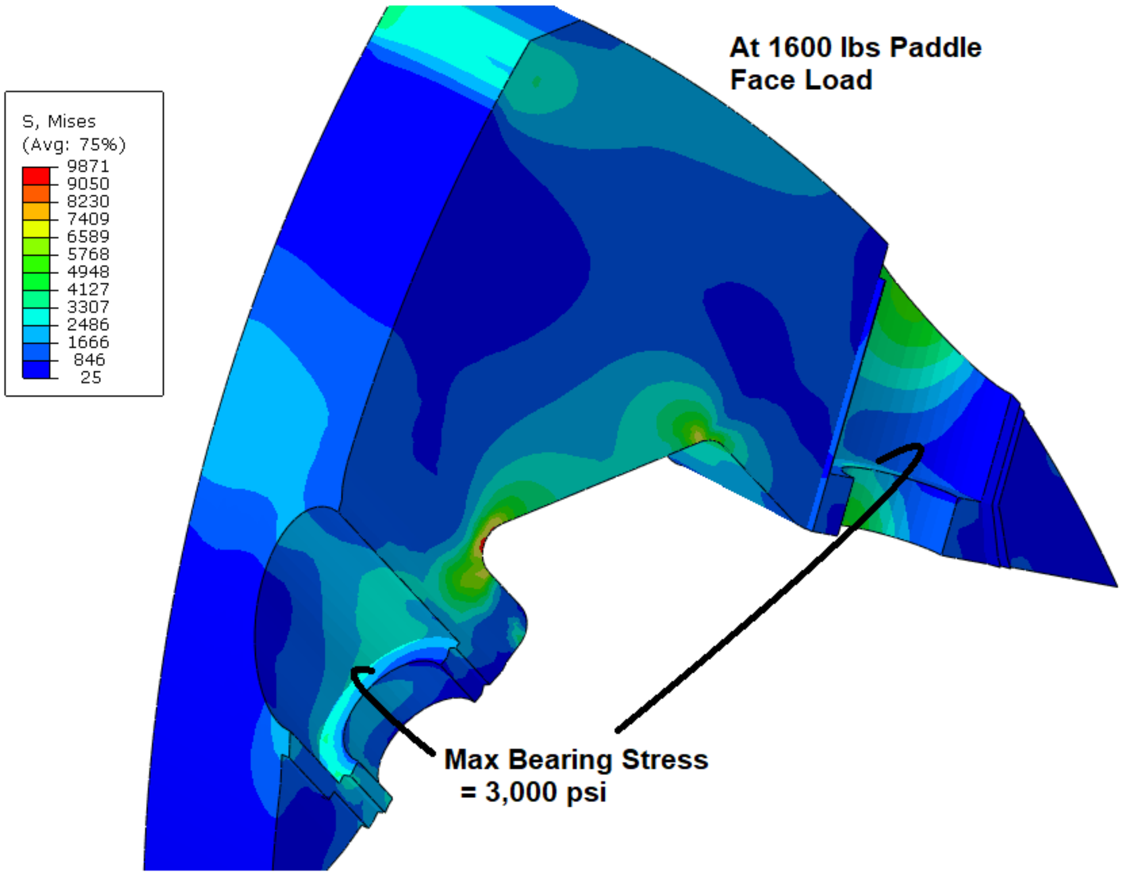


Figure 6-8 Stress at Bolt Hole for 350 lbs Bolt Preload, 1600 lbs Load Mixing Load on Paddle

6.4.2 Paddle Stress at Notch

Theoretical Primary Stress

The forces on the paddle face result in a pressure load on the front face of the hub-to paddle key. In this design, the prying from the 4.1 inch moment arm is reacted by a force coupe between the leading edge of the key and the back side reaction pressure.

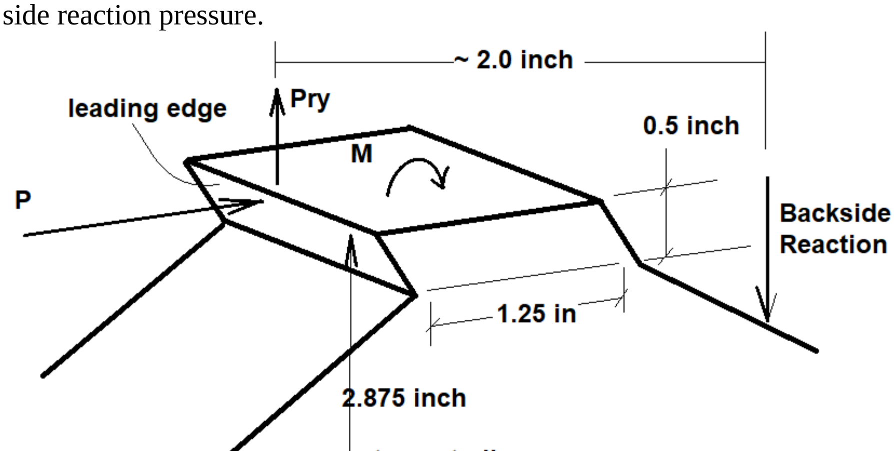


Figure 6-9 Schematic of Locking Key

Mixer Load = F = 1600 lbs, acting at $\geq$ 4.1 inch from center line (per conservative design load)
P on Leading Edge = F = 1600 lbs (the shear load is simply transferred to the leading face)
Pry = 1600 lbs * (4.1 inch - 2.875) / 2.0 inch = 980 lbs

Moment on Key cross section (1.25 inch x 2.0 inch)

M = 1600 lbs * 0.5 inch + 980 lbs (1.25/2) = 1,413 in-lbs

Section Modulus = $\frac{bt^2}{6} = \frac{2 \cdot 1.25^2}{6} = 0.52\text{in}^3$

Area = $\text{b} \cdot \text{t} = 2 \cdot 1.25 = 2.5\text{in}^2$

Tensile Stress = σt = Pry/A + M/S = 980lbs/2.5 + 1,413 in-lb / 0.52in^3 = 3109 psi
Shear Stress = σv = P/A = 1600 lb/2.5in^2 = 640 psi.

Combined Stress Equivalent: $\sigma = \frac{\sigma t}{2} + \sqrt{\left(\frac{\sigma t}{2}\right)^2 + \left(\frac{\sigma v}{1}\right)^2} = 3{,}236\ psi$ vs 58,300 psi allowed

Because of the shape of the sloped key, higher stress levels will occur compared to the primary stresses computed. The increase will be due to stress concentrations at the corners and uneven stress distribution across the various cross-sections.

Peak Stress per FEA – Operating Loads
Figure 6-10 shows the maximum stress in the paddle tip is 9,871 psi, occurring at the notched corner of the interfacing slot between the hub and the tip. Figure 6-11 shows the maximum stress in the hub is 8,750 psi, also occurring at the interfacing slot. These stress levels are cyclic, varying from near zero with no mixing load and up to the indicated value at the mixing load.

Recall that the modeled load was 1600 lbs, vs the bounding value service load of 300 lbs expected during operation. The stress levels, scaled to actual conditions are summarized below:

Table 6-2 Paddle Stress Output Summary

	At 1600 lbs	Reference	At 300 lbs operating (scaled)	Allowable
Notch Primary Stress Primary	3,236 psi	Hand Calc	607 psi	58,300 psi
Paddle Tip Peak Stress	9,871 psi	FEA, Figure 6-10	1,851 psi	87,500 psi
Hub Peak Stress	8,750 psi	FEA, Figure 6-11	1640 psi	87,500 psi

Accident Condition
The accident condition load was set at 3,000 lbs, or 10 times the normal operating load. Based on the results for 300 lbs, the maximum stress at 3000 lbs would be 18,510 psi. As a conservative comparison, this accident condition stress is less than the normal condition allowable.

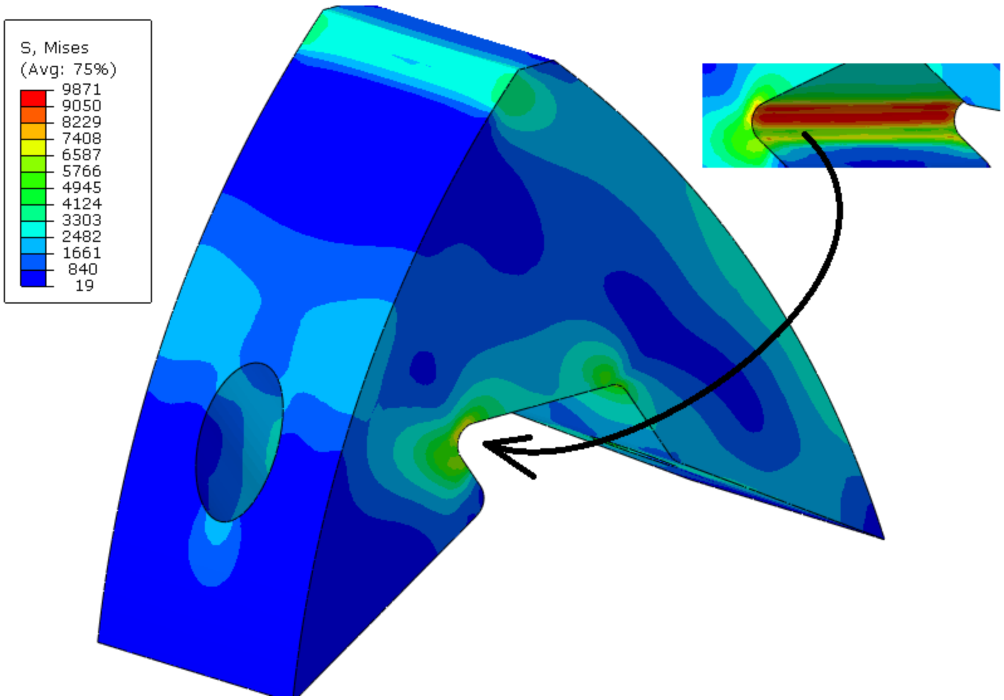


Figure 6-10 Stress Output for Paddle Tip, Showing 9871 psi

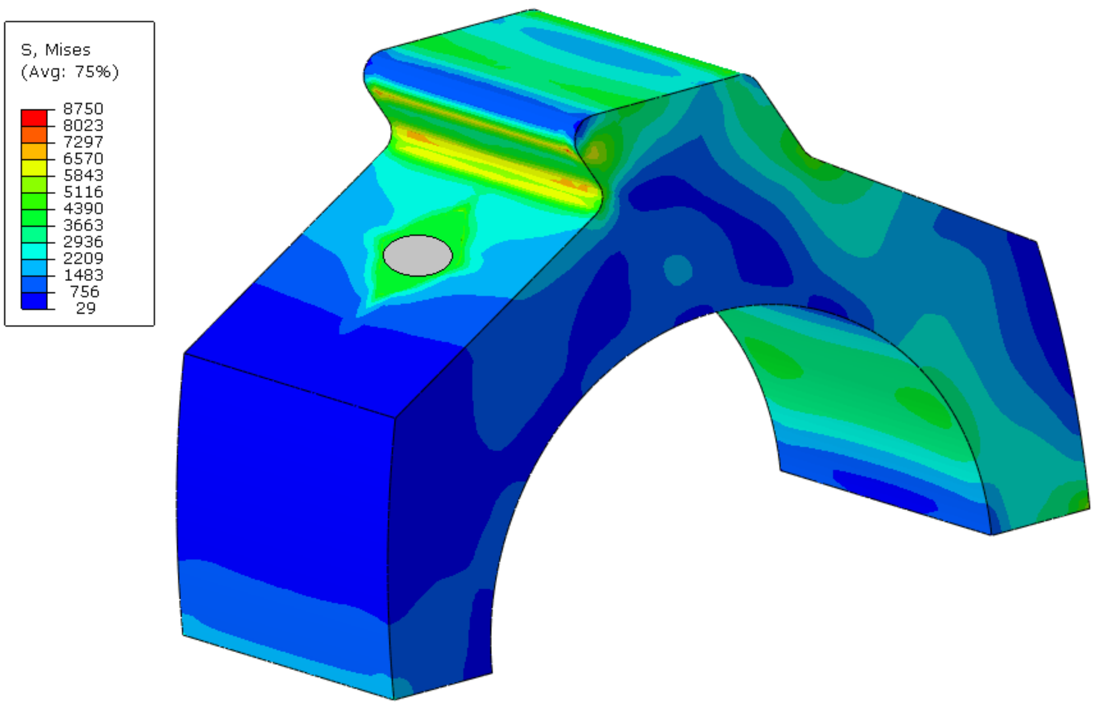


Figure 6-11 Stress Output for Hub Key at 1600 lbs, Showing 8750 psi.

6.4.3 Fatigue Evaluation

<u>Fatigue Factors At Hub Key</u>
Max Stress = 8,750 psi (Figure 6-11, Figure 6-12)

Linear Distance in which Stress is within 95% of Maximum = 0.045 inch (Figure 6-12)

Effective Full Size = 0.045 inch /(1-0.95) = 0.9 in   Per Ref 18 definition
Effective Diameter = 37% of 0.90 inch = 0.33 inch   Per Equation 6-24 of Reference 18

$$K_b = \left(\frac{d}{0.3}\right)^{-0.107} = 0.99$$

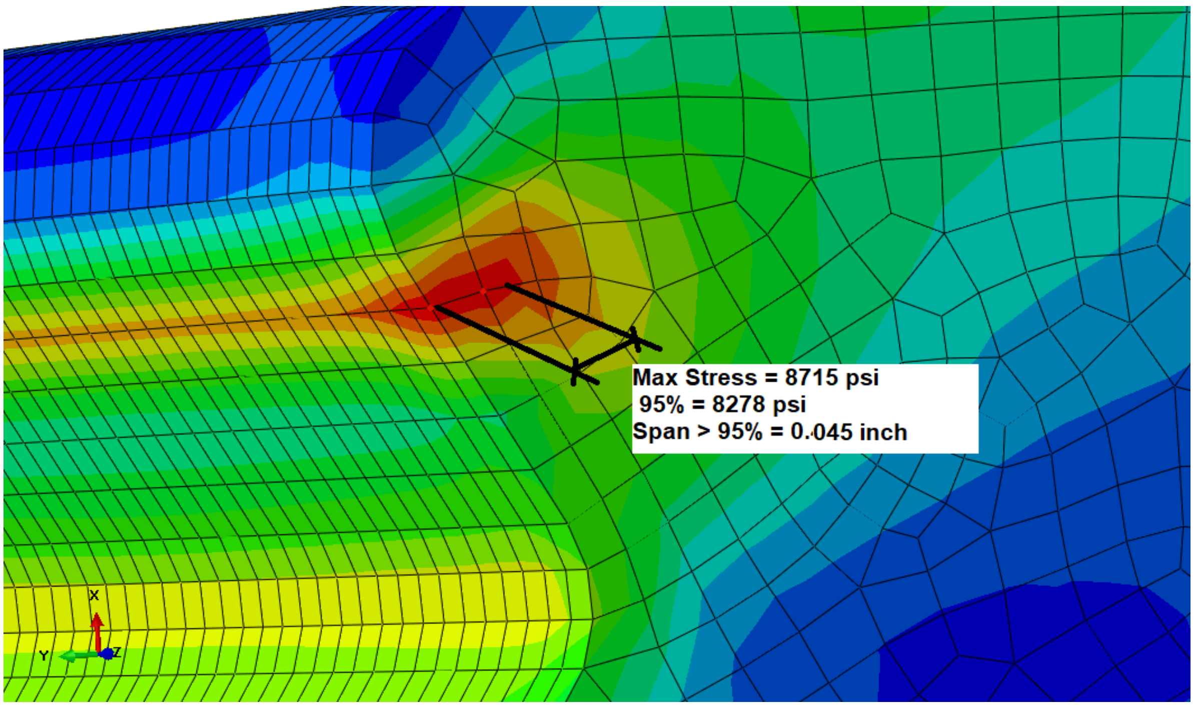


Figure 6-12 Detail of Hub Key Stress at Paddle Design Load, Showing 0.045 inch Span at 95% of Maximum stress.

<u>Fatigue Factors At Paddle Tip</u>
Max Stress = 9,782 psi (Figure 6-10)

Linear Distance in which Stress is within 95% of Maximum < 0.03 inch inch

Effective Full Size = 0.03 inch /(1-0.95) = 0.6 in   Per Ref 18 definition
Effective Diameter = 37% of 0.60 inch = 0.22 inch   Per Equation 6-24 of Reference 18

$$K_b = \left(\frac{d}{0.3}\right)^{-0.107} = 1.0$$

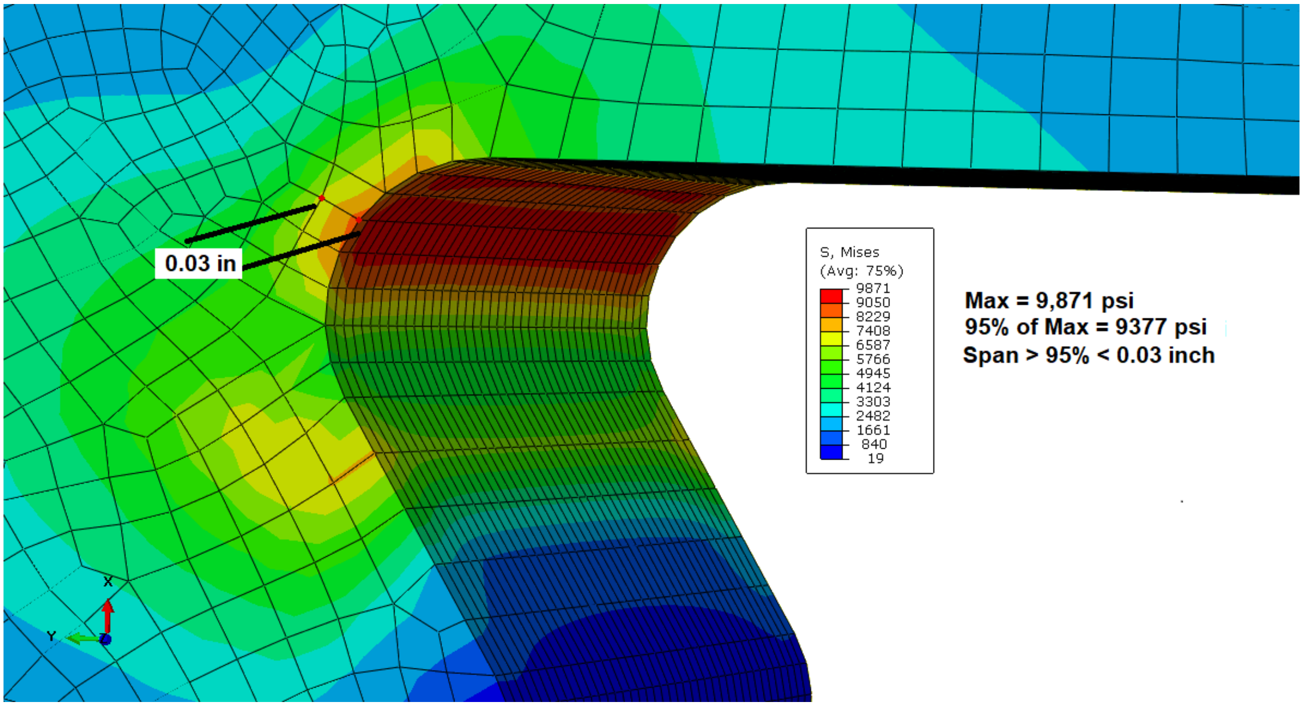


Figure 6-13 Detail of Tip Key Stress at Paddle Design Load, Showing < 0.03 inch Span at 95% of Maximum stress.

Evaluate Cyclic Stresses

Bolts

Recall the service condition stress level from section 6.2.2:

$$\sigma t = \frac{Ft}{At} = \frac{282lb}{0.142in^2} = 1986\ psi \qquad \sigma v = \frac{Fv}{Av} = \frac{322\ lbs}{0.126in^2} = 2556\ psi$$

Maximum Combined Stress: $\sigma = \frac{\sigma t}{2} + \sqrt{\left(\frac{\sigma t}{2}\right)^2 + \left(\frac{\sigma v}{1}\right)^2} = 3{,}735\ psi$

With the preload per 25 ft-lbs, the bolt loads are shown to not change during load operation, thus the cyclic load is near zero. Even if preload were lost, the bolts load would cycle from 0 psi to the 3750 psi.

Sa_before SIF = 3735 psi/2 = 1870 psi

SIF For Bolts = 4.0 (ASME Commonly cites SIF=4 for bolts)

Sa = 1870 psi x 4.0 = 7,500 psi

The endurance limit for the stainless steel bolt is 13 ksi. Therefore, bolt fatigue is not a threat.

Paddle Tip

The maximum stress in the paddle tip is 9,871. This is a localized stress at the notch of the paddle tip. The stress during operation would be expected to cycle between 0 ksi and 9,871 psi.

Cyclic Stress Component = 5 ksi

Endurance Limit = 41 ksi (see Section 5)

Therefore, fatigue is not an issue.

6.4.4 Fracture

The areas of fracture concern would be at the corners of the slanted key slot. The rounded corners on this feature and the overall low stress demand are sufficient to preclude a fracture threat.

7.0 Summary of Results

Bolts
Size = 1/2-13UNC 3A SST SHCS x 1 inch length, MC # 93705A637
Recommended Preload = 25 ft-lbs (+/- 3 ft-lbs)
Operating Condition Tensile Stress = 1986 psi (vs 21,000 psi allowed)
Operating Condition Shear Stress = 2,556 psi (vs 8,700 psi allowed)
Elliptical Interaction = 0.095
Required Thread Depth = 0.29 inch (vs 0.67 inch actual for 1 inch bolt length)
The FEA analysis shows no change in bolt load during mixer operational, compared to the installation preload.

Paddle
The following stresses are at 1600 lbs actual load on a single paddle. The actual upper bound operating load is 300 lbs, thus the 1600 lbs values are conservative. The values shown at the maximum stresses across any cross-section cut through the structure.

Hub Section = 8,750 psi Paddle Key = 9,871 psi (at key slot tips)	Allowable normal condition stress Primary Membrane = 58,300 psi Membrane + Bending = 87,500 psi
Fatigue Life = Meets endurance limit	
Fracture: No fracture expected.	

8.0 References

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Attachment A – Material Data for Astralloy V

A-1 Astralloy "V" Vendor Sheets

Astralloy
a NUCOR company

Astralloy-V®

Abrasion and Impact Resistant, Air Hardened Wear Steel Plate

Chemical Composition* – % Weight							
C	Mn	P	S	Si	Ni	Cr	Mo
.29	1.20	0.015	.010	.40	4.00	2.00	.50

Physical Properties – Typical Values at 68°F				
BHN Hardness	Tensile Strength	Yield Strength	Elongation in 2"	Charpy Test Toughness Index
418 – 512	241 ksi	157 ksi	12%	22 ft. lbs. @ RT

Comparative Benefits	
Astralloy-V Air Hardened	Quenched and Tempered Wear Steel
1. Hardness combined with toughness	1. Hardness with less toughness
2. Work hardenability up to 550 BHN	2. No work hardening ability
3. Lower coefficient of friction	3. Higher coefficient of friction

Astralloy-V® is a unique, deep-air hardened steel that is rich in chemical composition and physical properties. It is through-hardened and unsurpassed in resistance to impact and abrasion.

With continuous impact and abrasion, Astralloy-V can reach a hardness in excess of 550 BHN without brittleness.

Astralloy-V®

Round Bars

Chemical Composition* – % Weight							
C	Mn	P	S	Si	Ni	Cr	Mo
.29	1.05	.015	.010	.35	3.90	1.75	.45

Physical Properties – Typical Values at 68°F				
BHN Hardness	Tensile Strength	Yield Strength	Elongation in 2"	Charpy Test Toughness Index
320 – 388	175 ksi	153 ksi	15%	27 ft. lbs. Longitidinal @ RT

Attachment B – Information Sources (info Only)

Fatigue Strength Support Data (Ref. epi-eng.com (http://www.epi-eng.com/images/MechBasics/BAS-Fatigue2.JPG)

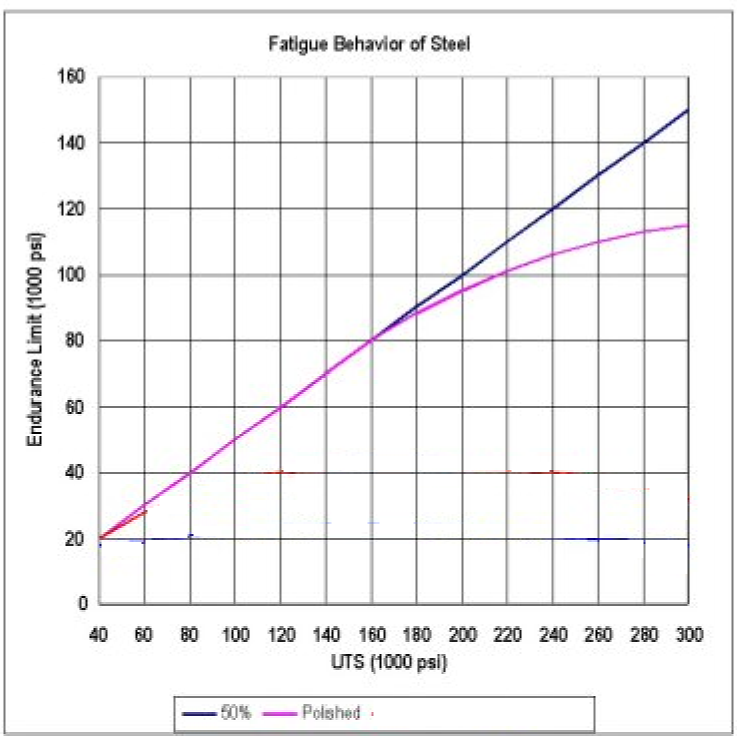


Figure B-1 Graph Showing Fatigue Strength as Function of Ultimate Strength, Showing > 70 ksi for 175 ksi Ultimate)

The endurance limit of steel displays some interesting properties. These are shown, in a general way, in the preceding graph, and briefly discussed below.

It is a simplistic rule of thumb that, for steels having a **UTS** less than 160,000 psi, the endurance limit for the material will be *approximately* 45 to 50% of the **UTS** *if the surface of the test specimen is **smooth** and **polished***.

That relationship is shown by the line titled "50%". A very small number of special case materials can maintain that approximate 50% relationship above the 160,000 psi level.

However, the **EL** of most steels begins to fall away from the 50% line above a **UTS** of about 160,000 psi, as shown by the line titled "Polished".

For example, a specimen of SAE-4340 alloy steel, hardened to 32 Rockwell-C (HRc), will exhibit a **UTS** around 150,000 psi and an **EL** of about 75,000 psi, or 50% of the UTS. If you change the heat treatment process to achieve a hardness of about 50 HRc, the UTS will be about 260,000 psi, and the EL will be about 85,000 psi, which is only about 32% of the UTS.

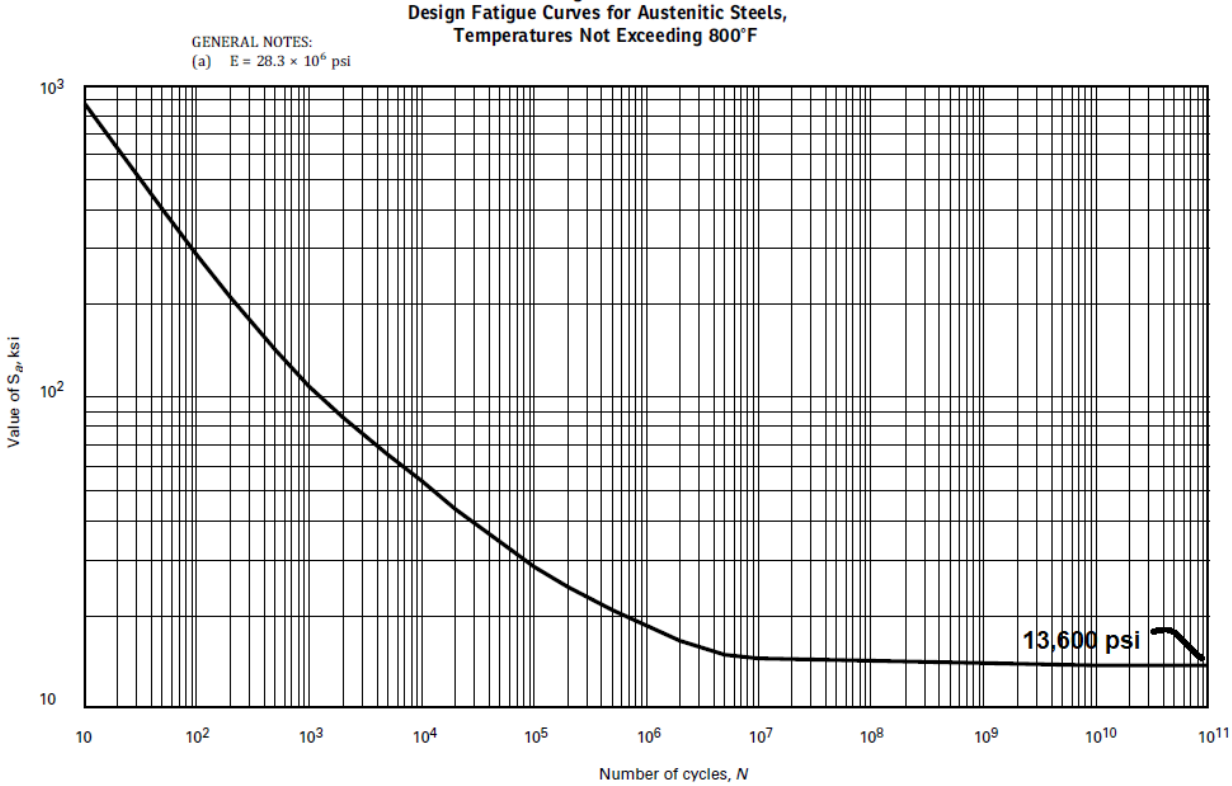
Several other alloys known as "ultra-high-strength steels" (D-6AC, HP-9-4-30, AF-1410, and some maraging steels) have been demonstrated to have an EL as high as 45% of UTS at strengths as high as 300,000 psi. Also note that these values are EL numbers for fully-reversing bending fatigue. EL values for hertzian (contact) stress can be substantially higher (over 300 ksi).

$$\textit{Real-World Allowable Cyclic Stress} = k_a * k_b * k_c * k_d * k_e * k_f * EL$$

a. **Reliability (k_e):** This factor accounts for the scatter of test data. For example, an 8% standard deviation in the test data requires a ke value of 0.868 for 95% reliability, and 0.753 for 99.9% reliability.

<u>ASME Fatigue Curve for 304 Stainless Steel</u>

Figure I-9.2
Design Fatigue Curves for Austenitic Steels, Temperatures Not Exceeding 800°F



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18-8 Stainless Steel Thread-Locking Socket Head Screw

1/2"-13 Thread Size, 1" Long

Packs of 1

ADD TO ORDER

In stock
$2.17 per pack of 1
93705A637

Thread Size	1/2"-13
Length	1"
Threading	Fully Threaded
Head Diameter	0.75"
Head Height	0.5"
Drive Size	3/8"
Material	18-8 Stainless Steel
Hardness	Rockwell B70
Tensile Strength	70,000 psi
Screw Size Decimal Equivalent	0.500"
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 3A
Thread Direction	Right Hand
Head Type	Socket
Socket Head Profile	Standard
Drive Style	Hex
Locking Type	Thread Locker
Thread Locker Type	Nylon Patch
Thread Locker Temperature Range	-70° to 250° F
Specifications Met	ASME B18.3
System of Measurement	Inch
RoHS	Compliant

Attachment C – Select Sections of ABAQUS FEA Model

```
*Heading
 The paddle with rounded edges
** Job name: thetip12 Model name: Model-1
** Generated by: Abaqus/CAE 2017
*Preprint, echo=NO, model=NO, history=NO, contact=NO
**  The tip
*Node
      1,    2.0844295,   -1.61961544,           0.
|
|   nodes and element cords ommitted
|
*beam section, section=circ, material=mbolt, elset=ecapscr
0.16
0,1,0
*beam section, section=circ, material=mbolt, elset=ecapscr2
0.16
0,1,0
*Material, name=mbolt
*Density
 0.00078,
*Elastic
 2.95e+07, 0.3
*Expansion
9.2e-06
************ bolts heads
*surface, name=sbhead1,type=revolution
2.2613, 1.3133, 1.0,3.1273, 1.8133, 1.0
start, 0.375, 0.5
line, 0.375, 0.0001
line, 0.25, 0.0001
line, 0.25, -0.24
**** the head surface is at 30 degrees. the hole is at 15
*surface, name=sbhead2,type=revolution
2.3315, -1.195, 1.0,3.1975, -1.695, 1.0
start, 0.375, 0.5
line, 0.375, -0.001
line, 0.25, -0.001
line,0.25, -0.24
*rigid body, ref node=190301, analytical surface=sbhead1
*rigid body, ref node=190311, analytical surface=sbhead2
*********************************************
*********************************************
*surface interaction,name=behavebolt
*friction
0.10,
*surface behavior,pressure-overclosure=tabular
0.,0.
100, 0.0001
1000,0.0002
20000,0.0005
****************************
*Elset, elset=esbolt1_S1,gen
 100497,  100623,       1
*Surface, type=ELEMENT, name=sbolt1
esbolt1_S1, S1
*Elset, elset=esbolt2_S1, gen
 105427,  105552,       1
*Surface, type=ELEMENT, name=sbolt2
esbolt2_S1, S1
```

```
        ** MATERIALS
        **
        *Material, name=msteel
        *Density
         0.0007,
        *Elastic
         2.9e+07, 0.3
        *initial conditions,type=temperature
        nbolt1,0.0
        nbolt2,0.0
        **************************** Constraints
        *mpc
        beam, 190315, 256529
        beam, 190305, 256615
        *nset,nset=norotate
        190315, 190305, 256529, 256615
        *Boundary
        Set-1, PINNED
        13937,3,3
        12770,3,3
        norotate, 3,6
        **190315,1,6
        **190305,1,6
        *contact
        *contact inclusions, all exterior
        *contact property assignment
        sbolt1,sbhead1,behavebolt
        sbolt2,sbhead2,behavebolt
        *Amplitude, name=PRELOAD, definition=SMOOTH STEP
        0., 0., 0.0027, 1., 0.003, 1.
        *Amplitude, name=forceit, definition=SMOOTH STEP
        0., 0., 0.003, 0., 0.004, 0.1, 0.005, 1.
        0.011, 1.0
        ** -----------------------------------------------------------------
        *Step, name=preload
         Pre-load
        *Dynamic, Explicit
        , 0.011
        *Bulk Viscosity
        0.06, 1.2
        *temperature, amp=preload
        nbolt1, -820
        nbolt2, -820
        *** Pressure area = 2" x 0.27" = 0.54 in^2
        ***** load = 1600 lbs, angle= 47 pressure=1600/.54 / sin43.8
        *dsload , amp=forceit
        spressure,p, 4265
        **
        ** OUTPUT REQUESTS
        **
        *Restart, write
        **
        ** FIELD OUTPUT: F-Output-1
        **
        *Output, field, variable=PRESELECT
        *Output, history,variable=preselect
        *Output, history,time interval=0.0004
        *element output,elset=ebolt
        sf
*End Step
```

Appendix D. Statistical Analysis of Mass and Volumetric Erosion Rates

Nonlinear Fit
Response: Ultimet (mass erosion rate, mg/hr), Predictor: b1 V^b0

Control Panel

Converged in Gradient

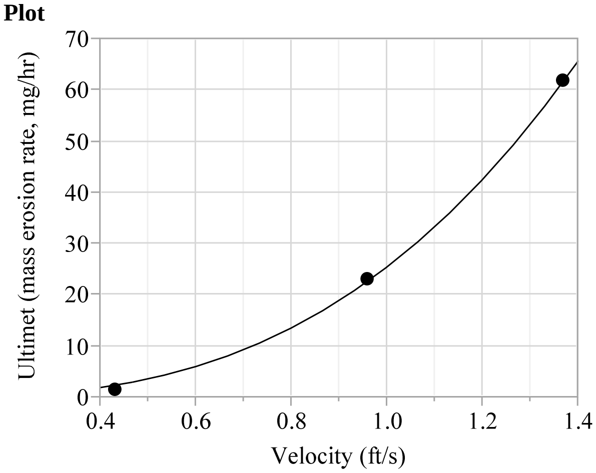
Criterion	Current	Stop Limit
Iteration	10	60
Obj Change	9.477898e-12	1e-15
Relative Gradient	2.1004825e-7	0.000001
Gradient	9.7552803e-7	0.000001

Parameter	Current Value	Lock
b0	2.8140173047	[ ]
b1	25.525203068	[ ]

SSE=1.2475245692
N=3

Edit Alpha=0.050
Convergence Criterion=0.00001Goal SSE for CL
.

Plot



Parameter	Estimate	Low	High
b0	2.8140173047	0.5	1.5
b1	25.525203068	0.5	1.5

Solution

SSE	DFE	MSE	RMSE
1.2475245692	1	1.2475246	1.1169264

Parameter	Estimate	ApproxStdErr
b0	2.8140173047	0.13884768
b1	25.525203068	1.05323415

Solved By: Analytic Gauss-Newton

Correlation of Estimates

	b0	b1
b0	1.0000	-0.9120
b1	-0.9120	1.0000

Nonlinear Fit
Response: Astralloy (mass erosion rate, mg/hr), Predictor: b1 V^b0

Control Panel

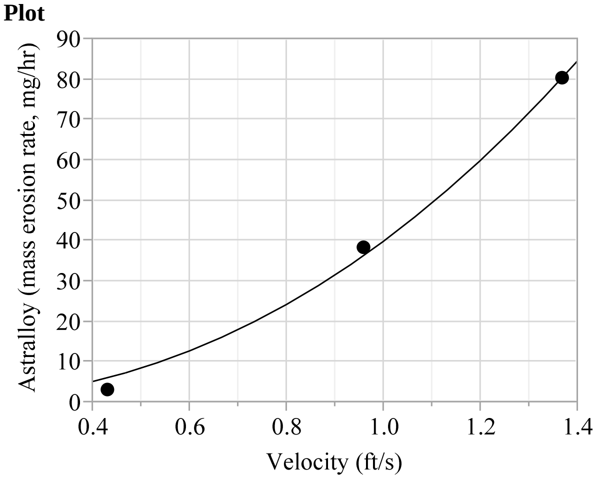
Converged in Gradient

Criterion	Current	Stop Limit
Iteration	11	60
Obj Change	2.844959e-12	1e-15
Relative Gradient	3.1456075e-7	0.000001
Gradient	2.4693167e-6	0.000001

Parameter	Current Value	Lock
b0	2.229140537	[ ]
b1	40.027372494	[ ]

SSE=12.748991553
N=3

Edit Alpha=0.050
Convergence Criterion=0.00001Goal SSE for CL
.
Plot



Parameter	Estimate	Low	High
b0	2.229140537	0.5	1.5
b1	40.027372494	0.5	1.5

Solution

SSE	DFE	MSE	RMSE
12.748991553	1	12.748992	3.570573

Parameter	Estimate	ApproxStdErr
b0	2.229140537	0.26260611
b1	40.027372494	3.07372783

Solved By: Analytic Gauss-Newton

Correlation of Estimates

	b0	b1
b0	1.0000	-0.8521
b1	-0.8521	1.0000

Nonlinear Fit
Response: Ultimet (Volumetric loss rate, cm^3/hr), Predictor: b1 V^b0

Control Panel

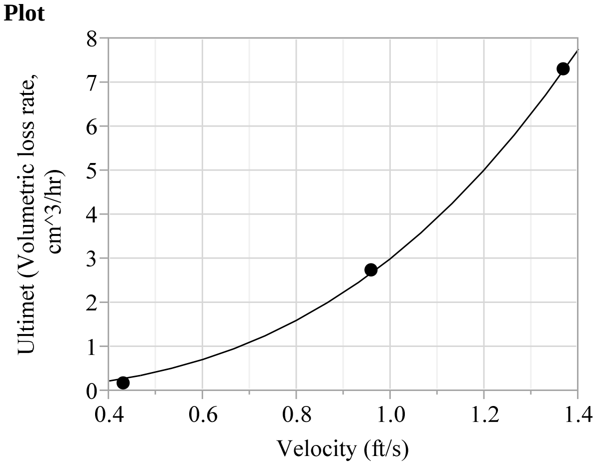
Converged in Gradient

Criterion	Current	Stop Limit
Iteration	7	60
Obj Change	8.04716e-12	1e-15
Relative Gradient	1.9354287e-7	0.000001
Gradient	1.061356e-7	0.000001

Parameter	Current Value	Lock
b0	2.8139911973	[ ]
b1	3.0136345738	[ ]

SSE=0.0173895519
N=3

Edit Alpha=0.050
Convergence Criterion=0.00001Goal SSE for CL
.
Plot



Parameter	Estimate	Low	High
b0	2.8139911973	0.5	1.5
b1	3.0136345738	0.5	1.5

Solution

SSE	DFE	MSE	RMSE
0.0173895519	1	0.0173896	0.1318695

Parameter	Estimate	ApproxStdErr
b0	2.8139911973	0.13884681
b1	3.0136345738	0.12434927

Solved By: Analytic Gauss-Newton

Correlation of Estimates

	b0	b1
b0	1.0000	-0.9120
b1	-0.9120	1.0000

Nonlinear Fit
Response: Astralloy(Volumetric loss rate, cm^3/hr), Predictor: b1 V^b0

Control Panel

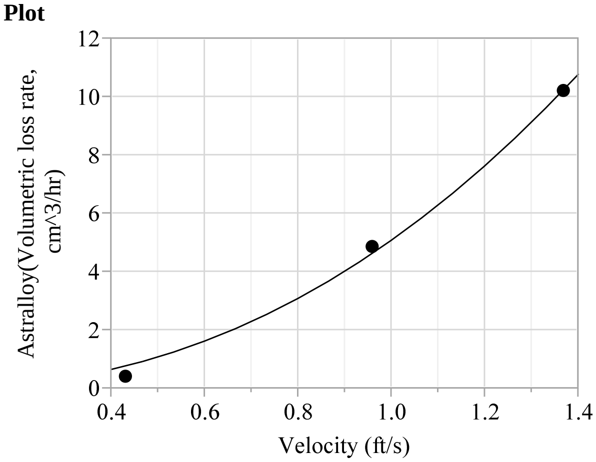
Converged in Gradient

Criterion	Current	Stop Limit
Iteration	8	60
Obj Change	2.114233e-12	1e-15
Relative Gradient	2.7100035e-7	0.000001
Gradient	2.7100318e-7	0.000001

Parameter	Current Value	Lock
b0	2.2291326154	[ ]
b1	5.0990243952	[ ]

SSE=0.2068518861
N=3

Edit Alpha=0.050
Convergence Criterion=0.00001 Goal SSE for CL
.
Plot



Parameter	Estimate	Low	High
b0	2.2291326154	0.5	1.5
b1	5.0990243952	0.5	1.5

Solution

SSE	DFE	MSE	RMSE
0.2068518861	1	0.2068519	0.4548097

Parameter	Estimate	ApproxStdErr
b0	2.2291326154	0.26258265
b1	5.0990243952	0.39152234

Solved By: Analytic Gauss-Newton

Correlation of Estimates

	b0	b1
b0	1.0000	-0.8521
b1	-0.8521	1.0000

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