

**HEAVY MOVABLE STRUCTURES, INC.
SIXTEENTH BIENNIAL SYMPOSIUM**

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**Safeco Field Movable Stadium Roof
Bogie System Repair**
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INTRODUCTION

Configuration of Retractable Roof

Safeco Field (home of the Seattle Mariners) is a stadium with a movable retractable roof that allows for open air events but also has the ability to close during inclement weather. The retractable roof system at Safeco Field is comprised of three (3) independently operable roof panels. Each panel opens and closes in the “east-west” direction; the two smaller panels (Panels 1 and 3) nest below a single large panel (Panel 2) when the roof is retracted, and are staggered to enclose the playing field and stands when extended. Each movable panel is supported at its north and south edge by single-line wheeled trucks called bogies, which drive along parallel rails and are supported by reinforced concrete runways. Each of the concrete runways are supported by a steel framework which consist of braced columns and flexural trusses that span the adjacent rail yard to the east of the stadium.

Four (4) eight-wheel bogies (2 on the north runway and 2 on the south runway) support Panels 1 and 3. Eight (8) eight-wheel bogies (4 on the north runway and 4 on the south runway) support Panel 2. Panels 1 and 3 bogies share common north and south rails, while Panel 2 travels on separate rails outboard of the Panel 1 and 3 rails. Panels 1 and 3 are each approximately half the width of Panel 2.

Each bogie assembly consists of four (4) two-wheeled truck assemblies connected to the roof structure by a series of equalizer pins and equalizer beams. The system equalizes the vertical loads along a line of bogies while allowing for slight variations in rail elevation. Six of the eight wheels on each bogie are powered. A gear motor a single pinion gear which directly powers each driven wheel. Each wheel is fit on and keyed to an axle, which is supported by two spherical roller bearings.

The spherical roller bearings which support the axle are mounted in cartridge housings. The bogie structural frame rests upon the cartridges which are connected by clamping bolts which pass through the end caps and bogie structural side plates. The cartridge end caps retain the bearing and resist axial loads.

Year long trouble free operation of the stadium’s roof panels is critical for the Seattle Mariners. While the primary priority is the Major League Baseball Season (April through September), many other events are scheduled in the off season (including corporate and private events, concerts, etc.).



Safeco Field – Open Air Stadium with Retractable Roof

CHAPTER 1 – INITIAL AXLE FAILURE

On January 26, 2012, the 3rd wheel axle of the Panel 1 North Northwest Bogie # 1 (designation PIN-B1-W3) experienced a catastrophic axle failure during a routine maintenance roof move. The failed axle is pictured below.



Fortunately for the Mariner's, the axle failure happened during the off-season of baseball. After an emergency repair construction contract was issued, the failed wheel assembly was replaced with one of the two spare wheel assemblies.

Following the emergency replacement of the wheel assembly, an effort was undertaken to determine the cause of failure, evaluate the remaining in-service wheel assemblies and develop long and short term corrective measures.

CHAPTER 2 – TESTING, ANALYSIS, ACTION PLAN

Working closely with the Mariners, Hardesty & Hanover developed and recommended the following action plan:

- Perform forensic testing on the failed axle to determine the failure mechanism.
- Perform ultrasonic and magnetic particle testing of the remaining in-service axles to determine if there was a system wide problem with the axles.
- Analyze the as-built axle design details and perform a fatigue analysis.
- Perform a "Fitness for Service Evaluation" for the wheel axles.
- Prioritize short term axle replacements and develop interim axle repair details.
- Develop design upgrade/phased work sequence working within the time and budget constraints.

Forensic Testing

Once removed, the failed axle was sent to Northwest Laboratories in Seattle, WA. Testing and evaluations included: visual examination, hardness testing, fractographic and metallographic examination, mechanical property testing and chemical analysis. In short order, the following was determined:

- The axle failure was the result of *low-stress/high-cycle fatigue*, (i.e., the axle failure was progressive in nature and occurred over a period of time).
- *The fatigue failure originated at the toe of the locking collar groove* and then progressed across the shaft thickness until crack instability resulted in final failure.
- Approximately 95% of the fracture face was covered by fatigue, suggesting that a *relatively low nominal stress had propagated the failure (Stage II of fatigue)*.



Initial Axle Failure at Locking Collar Groove

Ultrasonic Examination of the In-Service Roof Wheel Axles

Ultrasonic examination of the remaining 127 wheel axles was performed by Wiss, Janney Elstner Associates. The evaluation identified over forty axles that produced indications of crack initiations. Based on the examination of the fractured axle, with advanced crack growth that exceeded 60 percent of the cross section, none of the detected indications represented conditions that warranted immediate replacement of the remaining axles.

Magnetic particle testing was also performed on a small sample of the axles exhibiting UT indications, confirming the presence of cracks propagating from the locking collar and snap ring grooves.

All indications were believed to represent cracks propagating from the two types of wheel retention device grooves (locking collar or snag ring). Apparent cracks were detected in 37 axles with locking collars and 4 axles with snap rings.

Later in 2012, subsequent fine tuned ultrasonic testing determined that the depths of the cracks in 15 wheels (10 driven wheels and 5 idler wheels) warranted axle replacement in the short term. These wheels

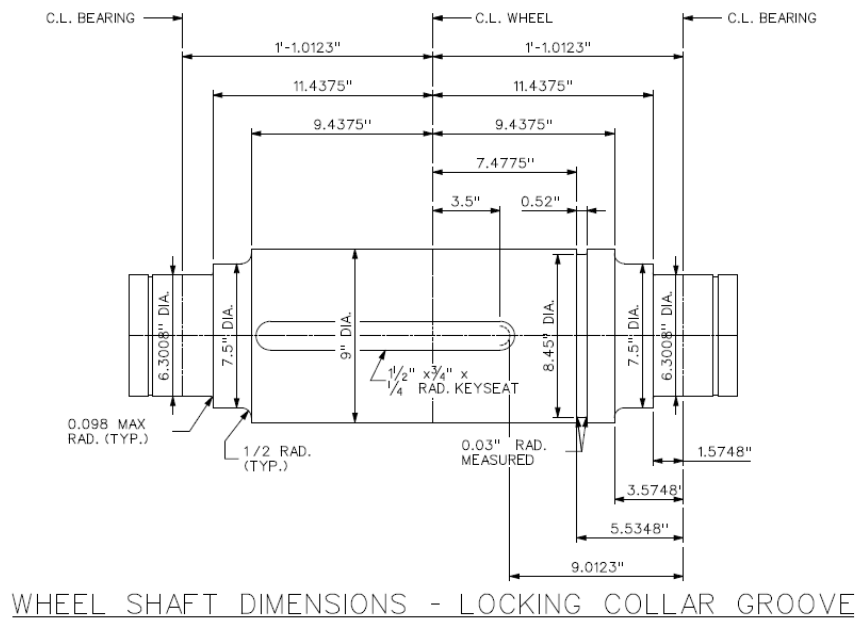
exhibited circumferential cracks radiating from the locking device grooves, with crack depths estimated to be between 1/8 inches and 1/2 inches.

Axle Analysis

Wheel Assembly Arrangement

Each wheel assembly is comprised of a 36 inch tread diameter, double flanged wheel mounted on forged alloy steel axles. The axle is supported by spherical roller bearings at each end. The bearings are housed in a clamp-type cartridge housing that connects to the bogie truck frame. Wheel assemblies transmit roof dead load, live loads and operating loads from the roof structure to the running rail support system.

Each wheel assembly is locked axially by a snap ring or by a locking collar that fits into a circumferential groove machined in the body of the wheel axle. Early revisions of the wheel assembly drawing do not show details of the locking collar, however, Safeco personnel indicate that during the original commission of the roof panels the wheels were observed to be moving axially on the axles. To prevent the wheels from moving on their respective axles, the locking collar and snap ring details were added to the wheel assemblies. To seat these collars, grooves were field machined into the axles during the construction period.



Wheel Axle Fit on Axle

Safeco personnel provided the shop fabrication drawings for review to better understand the design of the wheel assemblies, in particular the design of the wheel axle. These fabrication drawings indicate a fit between the wheel and axle as follows:

Wheel Bore (inch)	Axle Diameter (inch)	Fit Range (inch)
8.9958	9.0000	-0.0042
8.9978	8.9990	-0.0012

The above dimensions indicate the fit between the wheel and axle to be an American National Standards (ANSI) LN3 Locational Interference Fit. ANSI B4.1-1967, Revision 1979, Preferred Limits and Fits for Cylindrical Parts, provides guidance and recommendations for fits and tolerances of mating components. ANSI B4.1 describes Locational Interference fits as:

Locational Interference fits are used where accuracy of location is of prime importance and for parts requiring rigidity and alignment with no special requirements for bore pressure. Such fits are not intended for parts designed to transmit frictional loads from one part to another by virtue of the tightness of the fit, as these conditions are covered by force fits.

Original design loads indicate the presence of horizontal loads at columns that would be resolved as lateral loads at the wheels and rail support system (this is further supported by the fact that flanged wheels were utilized). Based on the above commentary, the wheel assembly fit is not appropriate for transferring lateral loads from the bogies through the wheel assemblies. It is likely that because of this fit, the locking collar and snap ring details were necessary to maintain the wheel position on the axle. While this detail did in fact solve the issue of lateral movement of the wheels, these locking details have had a significant detrimental effect on the service life of the wheel axles.

Axle Fatigue Analysis

Laboratory examination of the failed wheel axle indicates a low-stress/high-cycle fatigue failure. This failure mode is characterized by the initiation of a crack during service, crack propagation as service continues, and ultimately results in fracture of the component. Low-stress/high-cycle fatigue failure is commonly associated with components that exhibit service load stresses that are in excess of the component's endurance limit, its fatigue strength, which results in a finite life. In general, fatigue life is distinguished by two regions, a finite-life region, less than 1,000,000 wheel cycles and infinite-life region, more than 1,000,000 wheel cycles. A cycle of the wheel axle is considered one (1) full rotation of the axle under service loads. At the time of failure, the Panel 1 wheel assembly had been subject to approximately 240,000 wheel cycles during its service life.

The endurance limit of a component is determined by physically testing a "perfect" polished, unnotched test specimen of the same material in a reversed bending. Test data shows that for steel material with an ultimate strength similar to that of wheel axle, the endurance limit is 50% of the ultimate tensile strength or 79 ksi in the case of the failed axle. This value is considered the unfactored endurance limit of the material, as it is representative of a "perfect" polished and unnotched test specimen.

The wheel axles used in the Safeco Field wheel assemblies are a complex design containing multiple shoulders, grooves, threaded stake holes, and a key seat. These design features result in geometric changes that cause localized high stresses called stress concentrations. Because stress concentrations can have tremendous impact on the performance and service life of components, it is common practice to perform a fatigue analysis that accounts for not only the in service loads, but other influencing factors such as stress concentrations, manufacturing processes, material properties, component size and service reliability. To determine the fatigue life of a component, these influencing factors are applied to the unfactored endurance limit.

A traditional fatigue analysis of both the locking collar grooved wheel axle and the snap ring grooved wheel axle was performed in accordance with Shigley's Mechanical Engineering Design and Peterson's Stress Concentration Factors. At the Panel P1-B1 and B2, there are two (2) wheel axle designs that utilize either a locking collar or a snap ring fit in a groove. All other stress risers are otherwise similar for each axle design. Given the system arrangement, all bogies distribute load equally to wheel assemblies using an equalizer system. At Panel P1, the design loads for the northwest bogie are as follows:

Panel 1 North Bogie 1 - Design Vertical Loads

	Total Load (kip)	Wheel Load (kip)
Dead Load	1897	237
Moving Load	2065	258
Maximum Static Load	2622	328

The above noted design dead load closely matched the actual lift-off load measured during jacking of the bogie for replacement of the failed wheel axle.

The fatigue analysis was only performed for bending loads and was applied in cyclic reversed direction for both the locking collar groove axle and the snap ring groove axle under only dead load. Shown below are the modifying factors used for calculating the wheel axle endurance limit and the wheel axle service life. Each modifying factor can be thought of as a reduction factor. For example, the surface finish factor reduces the axle unfactored endurance limit by **29%**. As shown below, the stress concentration induced by the design of the locking collar groove and snap ring groove significantly reduce the wheel axle unfactored endurance limit, both **79%** and **82%** respectively.

Wheel Axle - Summary of Modifying Factors

	Locking Collar Groove		Snap Ring Groove	
	Modifying Factor	Percent Reduction	Modifying Factor	Percent Reduction
Surface Finish Factor	0.706	29%	0.706	29%
Size Factor	0.651	35%	0.649	35%
Load Modification Factor	1	0%	1	0%
Temperature Factor	1	0%	1	0%
Reliability Factor	0.814	19%	0.814	19%
Stress Concentration Factor	0.215	79%	0.177	82%
Miscellaneous Effects Factor	1	0%	1	0%

As previously noted, the unfactored endurance limit for the wheel axle material is 79 ksi. Applying the above factors to the unfactored endurance limit of the axle material yields the following factored endurance limit and corresponding life estimates for each geometric stress concentration on the axle:

Wheel Axle - Factored Endurance Limit, Bending Stress, Life at Geometric Stress Concentrations

	Factored Endurance Limit ksi	Bending Stress ksi	Estimated Life Wheel Cycles
Snap Ring Groove	5.2	11	200,000
Locking Collar Groove	6.4	11	275,000
Key Seat Fillet	13.9	14.9	800,000
Axle Fillet 1 (1/2" Rad. fillet)	16.5	10.2	3,500,000
Axle Fillet 2 (@ bearing)	11.7	7.6	5,100,000

As demonstrated above, applying the modifying factors in the wheel axle analysis yields a significant reduction to the axle endurance limit. The incorporation of the grooves in the axle reduced the estimated life of the axle from the next highest stress riser, the key seat fillet, by as much as 75%.

Fitness for Service Evaluation

A Fitness for Service Evaluation (FFS) was performed by Dr. Robert J. Connor for the axles containing the collar groove used on the Safeco Field movable roof structure. In its most general definition, fitness-for-service is defined as the ability to demonstrate the structural integrity of an in-service component containing a flaw or damage. Today, many industries with large steel structures have spent tremendous resources developing such guidance. Some of these industries include oil and gas pipeline, pressure vessel, power, offshore, and ship structure. Two of the most common specifications and most closely applicable to bridges structures are the American Petroleum Institute's API-579 "API Recommended Practice 579, Fitness for Service" and British Standard BS-7910 "Guide to Methods for Assessing the Acceptability of Flaws in Metallic Structures". For this evaluation, BS7910 was used.

It is important to emphasize the objective of this analysis was not to establish the cause of the axle failure. Rather, the following analysis was intended to provide data needed to establish a rational path forward regarding inspection and repairs, while ensuring reliable and safe operation. Hence, the specific calculations were made using conservative, yet reasonable procedures and assumptions.

Approach

The assessment of flaws for fracture potential is guided by the construction of what is known as a Failure Assessment Diagram (FAD) (see Figure 1). In its simplest form, the FAD is a method that graphically illustrates the potential for fracture failure, ranging from brittle fracture to plastic collapse. The vertical (x) axis of the FAD represents susceptibility to brittle fracture while the horizontal (y) axis is a measure of susceptibility to failure through plastic collapse. The FAD curve is derived from specific properties of the materials in question, primarily toughness, strength, and ductility (stress-strain). A Level One curve does not consider the interaction of those properties, while higher order FAD curves (Levels Two and Three) do respond to the interaction of the material properties and material behavior under load to derive a more accurate envelope of acceptability using linear elastic or elastic-plastic theories of material mechanics. A Level Two analysis was used for this assessment. The derivation of the FAD curve can be thought of as analogous to the "resistance" side of LRFD design approach for strength of a member, but without the application of factors of safety at higher levels of analysis (level two and three). Therefore, factors of safety are typically applied by the user based on a quantitative or qualitative risk assessment.

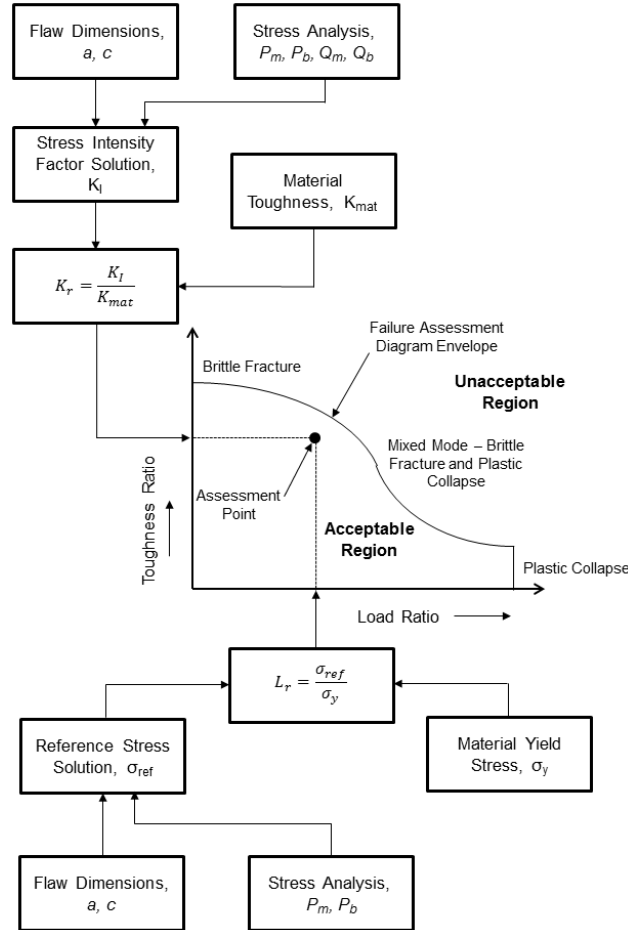


Figure 1: Flow Chart and Overview Illustrating FAD (From API 579)

Calculations for the given flaw being assessed result in the determination of a single value or point representing the flawed condition. This point is then plotted against the envelope of acceptable values. A value falling within the curve indicates that the flaw is acceptably safe from fracture failure while one falling outside the curve would be unacceptable at that level of assessment and with the input values assumed.

While the above flow chart was used as the basis for the evaluation contained herein, the specific steps were as follows:

1. Determine the critical circumferential crack size (a_{cr}) at which failure occurs using the FAD. The critical crack size is then used as the final crack size (a_f) in the fatigue life calculations as this is the largest tolerable crack.
2. Determine the initial crack size (a_i) to be used as the starting point for the fatigue life calculations. For this evaluation, the smallest initial crack size is conservatively set to be equal to the depth of the groove in the shaft. However, based on the results of the UT conducted to date, it appears some cracks may actually extend into the shaft an additional distance. For those axles, other, larger initial crack sizes were also evaluated and tabulated. Larger initial crack sizes result in much lower estimated lives.

3. Calculate the number of cycles (N) required to propagate the crack from the initial size (a_i) to the final size (a_f). As stated in #2, various lives were calculated assuming different values for the initial crack size (a_i).
4. Once the number of cycles to failure has been calculated, a recommended inspection interval/retrofit program can be rationally established. Obviously, this interval must be less than the time to failure. In order determine if it is appropriate to manage the issue through periodic inspection or by preemptively retrofitting a given axle, there are several factors to consider. For example, questions include:
 - a. What is the smallest detectable crack that can be found with high confidence?
 - b. How reasonable does the FFS model represent a given axle?
 - c. Is routine inspection feasible and practical?
 - d. What is the estimated crack growth rate for a given axle?
 - e. What is the existing condition of a given axle (i.e., have cracks been detected to date)?
 - f. What is the consequence of failure in terms of safety and operation of the facility?

To perform the FFS evaluation, data pertaining to loading, material properties, and crack size are required

Table 1 – Material Property Assumptions

Data	Value	Source	Comment
Stress Range	11.2 ksi	Hardesty & Hanover	FFS assumes pure bending, ignoring shear and any potential axial loading.
Cycles per Opening	55	Hardesty & Hanover	Assumes worst case axle. Others will be less since they travel a shorter distance.
Toughness	150 ksi-in ^{0.5}	Estimated and based on results of literature review	AFGROW Material Library ¹
Yield Strength	139 ksi	Samples obtained from failed axle	Obtained from the failed wheel shaft: Northwest Laboratories RTW Axle Failure Report February 15, 2012 Mechanical Tests
Tensile Strength	158 ksi	Samples obtained from failed axle	Obtained from the failed wheel shaft: Northwest Laboratories RTW Axle Failure Report February 15, 2012 Mechanical Tests
Initial Crack Size	0.275 in	Drawings of axle	Conservatively assumes the initial crack size is equal to the depth of the collar groove.
C_p -	1.7 x 10 ⁻⁹	Estimated and based on results of literature review	Paris law constants for A4340 steel from AFGROW Material Library ¹
M_p	2.7	Estimated and based on results of literature review	Paris law constants for A4340 steel from AFGROW Material Library ¹

1. AFGROW is a Damage Tolerance Analysis (DTA) software used to analyze crack initiation, fatigue crack growth, and fracture to predict the life of metallic structures. It was originally developed by The Air Force Research Laboratory

Calculation of the Critical Crack Size (a_{cr})

To model the cracking observed in the axle, a solid circular shaft of radius 'r' and a circumferential crack of size 'a' was selected from the family of geometries included in BS7910. The model is capable of including the effects of primary axial and bending stresses, though only bending stresses were included. Further, no residual stresses were included, though the model is capable of accommodating as secondary stresses. Figure 2 illustrates the model conceptually. The axle that was evaluated contains the locking collar groove detail.

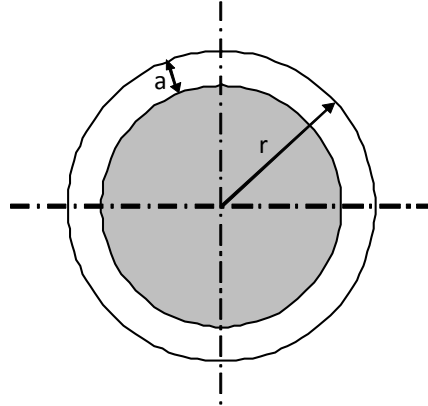


Figure 2 - Model used to calculate the critical crack size

The model assumes the crack is circumferential. Evaluation of cracks that have extended partially around the circumference is not possible in this model. Further, the crack size (a) is uniform around the entire axle. Lastly, the threaded dowel holes that were found to exist are not included in the model.

Using the data above, the critical crack size was calculated to be 2.6 inches. Note this is a circumferential crack of 2.6 inches leaving a core of steel that is about 3.8 inches in diameter. At this size, only about 17% ($11.3 \text{ in}^2/63.6 \text{ in}^2$) of the cross sectional area remains assuming only bending loading.

As stated, a_f is then set equal to the a_{cr} for purpose of the fatigue life calculations.

Selection of the Initial Crack size (a_i)

As stated, accurately establishing the *initial* crack size (a_i) is essential in the analysis. The reasons are illustrated in Figure 3, which shows schematically the change in crack length 'a' vs. the applied number of cycles. As is readily apparent, as the number of cycles increase, the crack size also increases, but at an increasing rate. In other words, the growth rate is not linear. At the initial stages of crack growth, there is relatively little crack extension with increasing cycles (N) and much more time (i.e., cycles) is spent growing the crack from the initial size (a_i). However, toward the right hand of the plot (especially near fracture), it can be seen that for the same number of applied cycles, there is much greater crack growth.

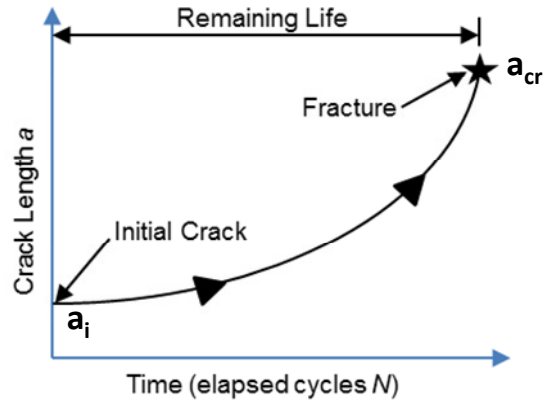


Figure 3 – Illustration of crack growth rate as a function of crack size.

Figure 3 also reveals that the selected value of a_i has a major influence on the total estimated life. The implication of this observation is most important in axles where cracks are already believed to extend out of the locking collar groove. In such cases, much of the life has already been exhausted, in contrast to those axles where there appears to be no evidence of cracking. This also emphasizes the need for UT procedures that have been calibrated such that they consistently result in data of high confidence.

The data obtained thus far appears to confirm the cracks all originate at the machined grooves in the axle shaft. One of the reasons for this is due to the fact that at the base of the groove there is a high stress concentration factor (SCF) which amplifies the nominal stress (σ_{nom}). The SCF due to the groove was estimated to be on the order of 4.5 to 4.7 or even higher. However, it is well known that the effect of the stress concentration decreases with the distance from the groove, as illustrated schematically in Figure 4. Eventually, the stress state returns that consistent with the nominal stress.

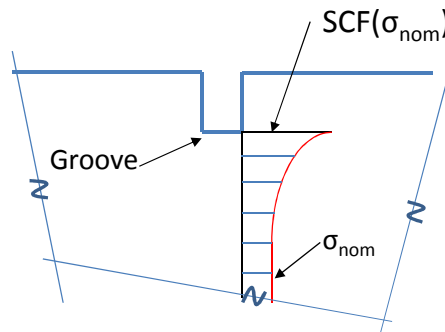


Figure 4- Illustration of the Effect of SCF at the Base of Groove

Examination of Figure 4 makes it clear that cracks growing out of the groove are initially subjected to much higher stress ranges due the influence of the SCF. However, as these cracks propagate, the effect of the SCF diminishes and crack growth slows.

To avoid the need for the rigorous finite element analysis, the associated assumptions, and the need for very accurate measurements of the crack size at the base of the groove, a conservative model was developed. This model simply assumes the initial crack size is equal to the depth of the notch. This alleviates the need to calculate the stress concentration effect at the notch as it is inherently included in the crack model. Of course, this leads to a conservative estimate as the initial crack size (a_i) is assumed to

be large, on the order of 0.25 inches and ignores crack initiation life and the life associated with growing a small crack. Nevertheless, the approach is reasonable and eliminates the need for several assumptions.

As a result, the initial crack size (a_i) was set to be equal to the depth of the collar groove, or 0.275 inches.

Fatigue Life Calculation (Number of Cycles to Grow from a_i to a_f)

The estimated number of cycles to failure was calculated using the well-known Paris Law. Although there are many other crack growth approaches available, Paris law provides conservative estimates and requires the least number of assumptions when specific material property data are not available. As stated, the initial crack size (a_i) was set at 0.275 inches and the final crack size (a_f) was set at 2.6 inches based on the results of the FAD analysis.

Based on the analysis, the estimated number of cycles to failure is 199,766 cycles. Considering the scatter in fatigue data and the fact that a simplified model was used, this is actually in very good agreement with the actual number of cycles seen by the failed wheel shaft (about 250,000 cycles).

However, it is important to recognize that the above assumes that the initial crack size is equal to the depth of the groove. As previously stated, there are several axles where it is believed that cracks extend beyond the base of the groove. As shown in Figure 3, larger assumed values for the initial crack size result in significant decreases in the total calculated life (N). In order to illustrate the effects of existing cracks which extend below the grooves, various initial cracks sizes were evaluated, the results of which are tabulated in Table 2 and plotted in Figure 5.

Crack Length Beyond Base of Groove	Depth of Groove	Total Initial Crack Length (a_i)	Calculated Life (N)
0.0	0.275	0.275	199,766
0.125	0.275	0.400	134,920
0.25	0.275	0.525	95,694
0.375	0.275	0.650	69,546
0.5	0.275	0.775	51,161
1.0	0.275	1.275	15,072
1.5	0.275	1.775	3,812

Table 2 – Influence of the initial crack size (a_i) on total fatigue life (N)

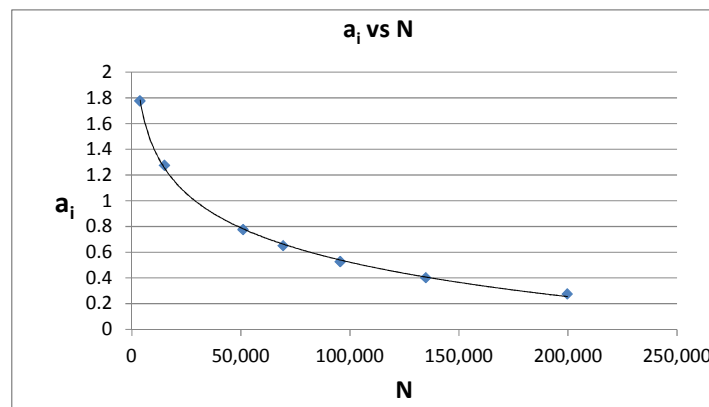


Figure 5 – Plot illustrating the influence of increasing the initial crack size (a_i) on total fatigue life (N)

It is clear that increases in the initial crack size have large effects on the total estimated life, especially as the initial crack size increases. The implication is that at locations where cracks extend beyond the base of the groove, even if only say 0.25 inches, there is a significant decrease in the calculated life.

Interpretation of Results

Using the above information, a strategy for axle replacement along with a rational inspection interval was established. There was high confidence that ultrasonic testing would be able to assess the presence and depth of cracks in the axles. Two cases were considered: 1) axles that show no sign of cracking and; 2) those where cracks are found.

Obviously, the inspection interval must be set so that it occurs well before the failure in order to be of value. Hence, it is not appropriate to use the estimated number of cycles at failure (i.e., 199,766 cycles) when setting the interval. Basically, there are two approaches that can be taken.

One approach is to apply a safety factor to the number of cycles at failure (e.g., a factor of 3) and perform the inspection at that interval. For example, for an axle where UT indicates no cracking has extended beyond the groove, one could set the inspection interval to be at one third the estimated life of 199,766 cycles or about 66,600 cycles. Based on the calculations, this corresponds to a crack that grew from 0.275 inches to total crack depth of about 0.5 inches. After 66,600 cycles, the axle would be re-inspected, any crack growth noted and the axle either replaced or the inspection interval maintained (assuming no growth).

The other approach is to select a detectable crack length and set the interval based on the corresponding number of cycles. In this case, one could select a crack size at which inspection would be performed and easily capable of finding the crack. For example, the inspection could be performed when the crack is estimated to have grown to a depth of 1.0 inch. Based on the analysis, the number of cycles to grow a crack to that depth is about 170,000 cycles. Inspection would be conducted based on that interval.

However, as stated earlier, it is important to recognize that crack growth is non-linear, as was shown in Figure 3. As seen, as the crack gets larger, it grows at an increasing rate. The practical implication is that although the crack may have only grown to a depth that is half of the critical crack size, perhaps 90% of the life may have been exhausted. Note this approach also requires there be high confidence in the UT crack sizing capability. Therefore, caution must be exercised when setting the inspection interval based on the crack size.

Conclusions and Recommendations

- *Ultrasonic examination indicated cracking in 41 of the 128 in-service wheel axles. Of the 41 axles with crack indications, 15 axles exhibited crack depths warranting replacement in the short term.*
- *Fatigue analysis indicates a finite service life for the wheel axles, with an estimated design service life of 200,000 to 275,000 wheel cycles.*
- *Fitness for Service Evaluation establishes a crack depth of 1/2", at any point, beyond the groove depth as the maximum size crack that should be tolerated in service.*
- *Develop a comprehensive short (Interim Repair with improved fatigue details) and long term (Upgrade Design with improved or infinite fatigue life) rehabilitation plan to ensure continued reliable operation of the retractable roof system.*

CHAPTER 3 – INTERIM AXLE REPLACEMENT

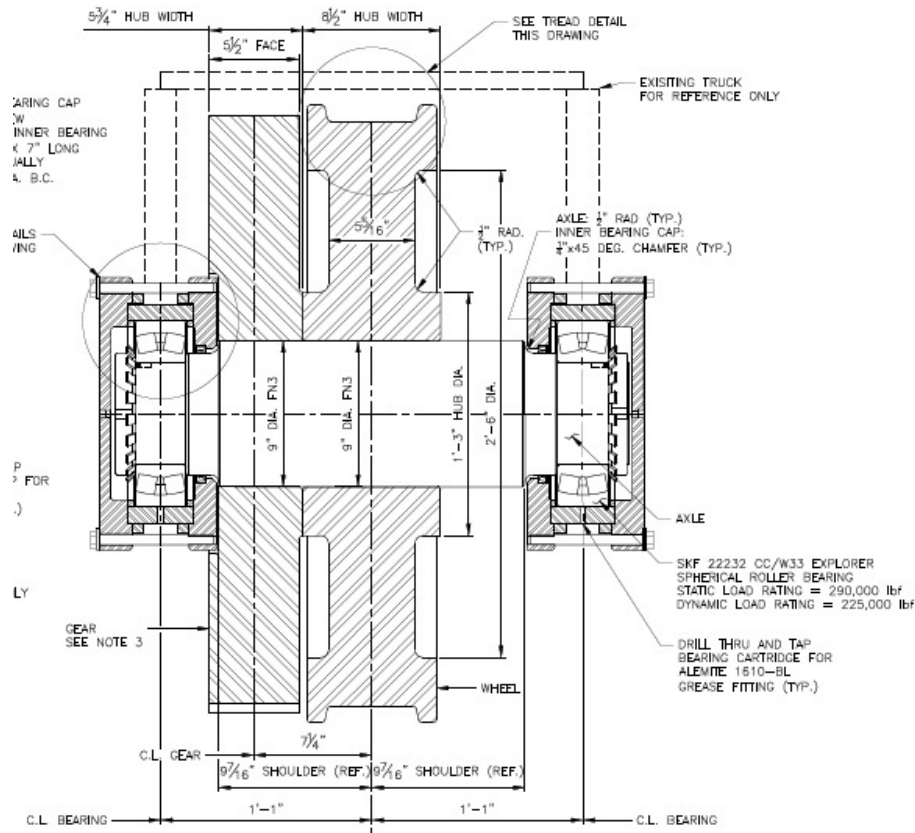
Interim Axle Design Improvements

Fortunately for the Mariners, the initial shaft failure happened in late January. Late January and February are the two months of the year when stadium events are infrequent and minimal roof panel moves are required. Come March, the grounds crew is diligently working on the baseball field's grass in preparation of the April home opener and start of each baseball season.

It was imperative to the Mariners to minimize the risk of another shaft failure during the baseball season. As such, the short term action plan was to replace the 15 wheel assemblies that exhibited the worst axle cracks as soon as possible.

Hardesty & Hanover developed a “drop in” replacement wheel assembly design with the following improvements:

- Eliminated the 1-1/2 x 3/4 inch key between the axle and wheel/gear
- Eliminated the locking collar and snap rings as a axial wheel restraint
- Utilized a “**Heavy Drive FN3**” force fit between the axle and wheel/gear to provide lateral wheel restraint and transmit driving torque



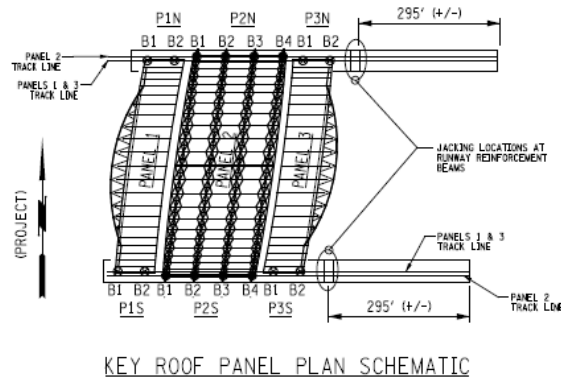
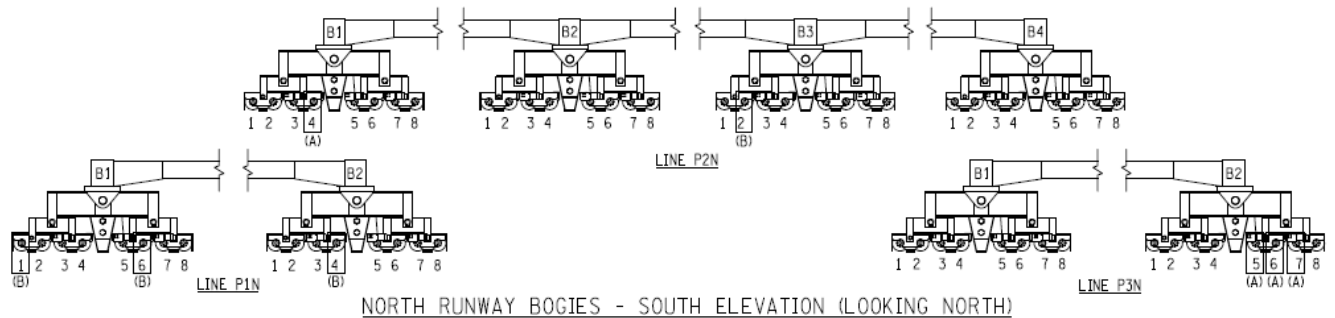
Interim Wheel/Axle Assembly with Improved Details

Safeco Field Bogie System Repair

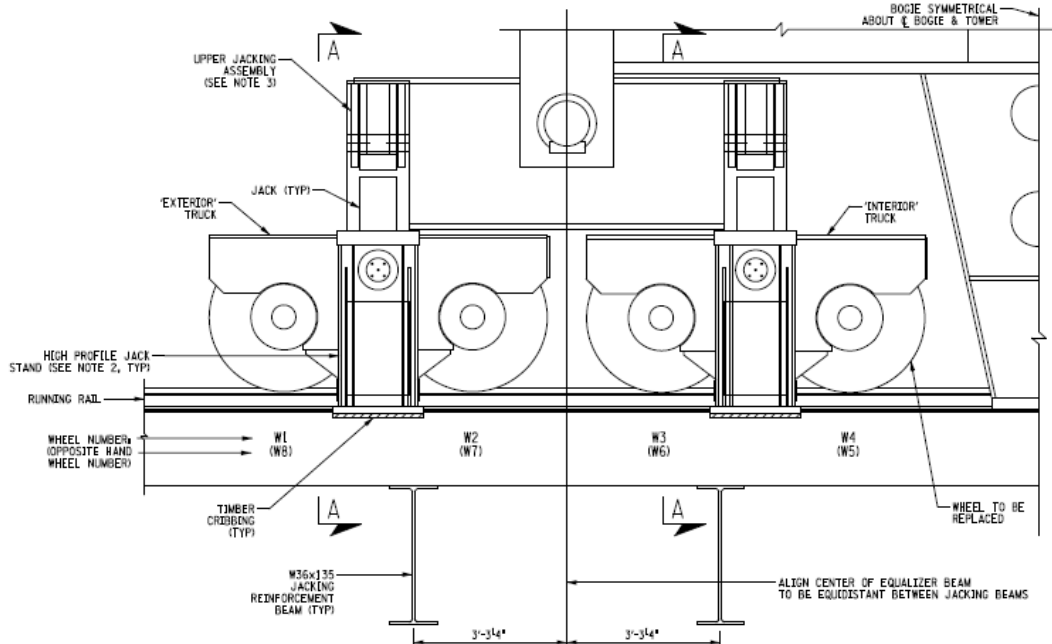
In the fall of 2012, the Mariners issued a procurement contract to fabricate the 15 interim wheel replacement assemblies.

Subsequently, an installation plan for the Interim replacement was developed. The 15 most severely cracked axles were scattered about the entire roof system. The roof system was not equipped with adequate jacking points to lift the roof panels and relieve the loads to facilitate wheel change outs. A jacking system consisting of welded on tabs, pinned jacking rigs and welded steel jacking stands had been developed and utilized to replace the failed axle in January of 2012. Additional components of that same jacking system were fabricated and installed in the appropriate locations for the 15 Interim wheel change out.

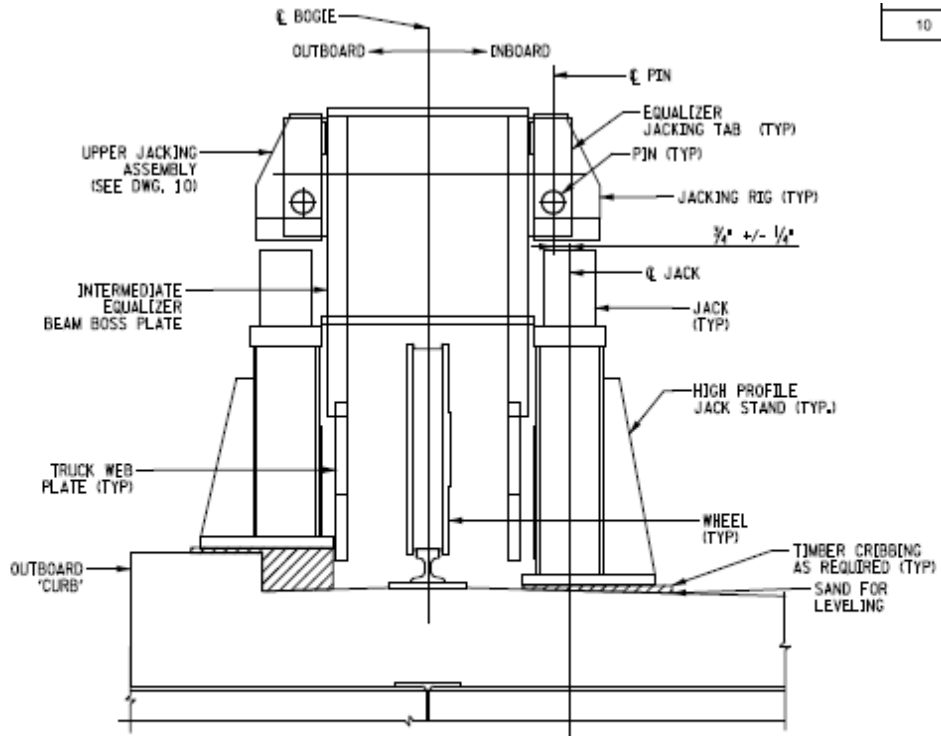
The following is a partial schematic plan of the roof layout and the location of the 15 wheel that required replacement.



Multiple jacking configurations were required depending on where in the system the wheel assembly to be replaced was located. A typical jacking configuration is shown below.



BOGIE ELEVATION - JACKING CONFIGURATION #4 (WHEELS 4/5)



VIEW A-A - HIGH PROFILE JACKING CONFIGURATION

NOTE: EXISTING JACKING CONFIGURATION SHOWN; NEW EQUIPMENT SIMILAR

In January 2013, the Interim 15 wheel change out was implemented.



Interim Wheel Assemblies Ready for Installation



Interim Wheel Assembly Installation

CHAPTER 4 – LONG TERM BOGIE SYSTEM UPGRADE

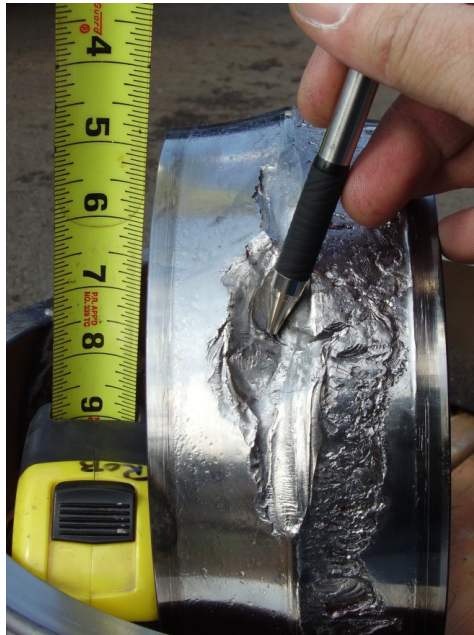
Axle and Wheel Bearing Design Improvements

A long term corrective action plan was developed to replace all wheel axles as well as to consider modifications related to the axle bearings which had a history of periodic failures. The scope of our evaluation was specific to the wheel assemblies including the wheels, axles and bearings and their associated mounting configuration. The drive machinery, truck structure and equalizer pins were not evaluated as part of this study.

Since being put into operation, the retractable roof system experienced multiple wheel bearing failures. Safeco's maintenance forces were spending considerable effort on wheel bearing inspection efforts as well as general maintenance and replacement of failed bearings.



Annual Bogie Wheel Bearing Inspection



Failed Rollers and Inner Bearing Race

Long Term Design Improvement Options

Three options were developed as follows:

- **Option 1** – Replace all remaining wheel assemblies (112 units) with the revised axle design currently being utilized for 15 units.
- **Option 2** – Replace all 128 wheel assemblies with an upgraded axle and wheel bearing design and modify the existing (64) bogie trucks to accept larger axle to provide extended life.
- **Option 3** – Replace all 64 bogie trucks with upgraded axle and wheel bearing design to provide “infinite life”.

Relative costs and benefits were discussed for the three options with the Mariners. Option 1 provides axles which have an expected life exceeding the practical life of the stadium. For this option, bearing size and mounting remains unchanged and bearing failures can be expected to continue in the future. Bearing failures were to be considered more than a nuisance or basic maintenance item if they prevent proper operation of the roof or cause damage to other components. It is difficult to predict the likelihood or to place a cost on these impacts.

Option 2 replaces the wheel assemblies while recycling and reusing components from the existing bogie trucks where practical. This option includes a larger, more robust bearing which should operate without failure for the remaining life of the facility.

Option 3 provides for newly designed trucks and would give full latitude to incorporate components which will have theoretically “infinite fatigue life”. Since the stadium is designed to have finite life and use of the retractable roof is limited, it may not be necessary to provide components with theoretically “infinite fatigue life” for this project.

Option 2 Selected

One of the three options would be selected by the Mariners as the long term program for the bogies. Option 1 was the least costly but does not address bearing failures which could be problematic and affect the reliability of the retractable roof. Option 2 would seem to be the best option if it is decided to address the bearing failures as part of the long term solution. Option 2 would provide reasonable life expectancy while minimizing cost as compared to Option 3.

It was important to the Mariners to minimize future wheel bearing maintenance, while working within their capitol improvement restrictions. As such, Option 2 was selected for the long term system repair option.

Work Sequence

Option 2 would be slightly augmented to include fabrication of 8 New Bogie Truck Assemblies (for the initial phase) and Re-machine/Recycle the existing remaining bogie trucks. A summary of the work sequence for the selected option would be as follows:

- Fabricate 8 new bogie trucks (complete with upgraded axles and axle bearing design) and ship to site
- Jack roof panel and install the 8 new bogie truck assemblies (2 groups of 4 new bogies)
- Strip original wheels and bearings from the bogie truck frames that were removed and

ship them to a machine shop for re-machining and fitting of new wheel assemblies

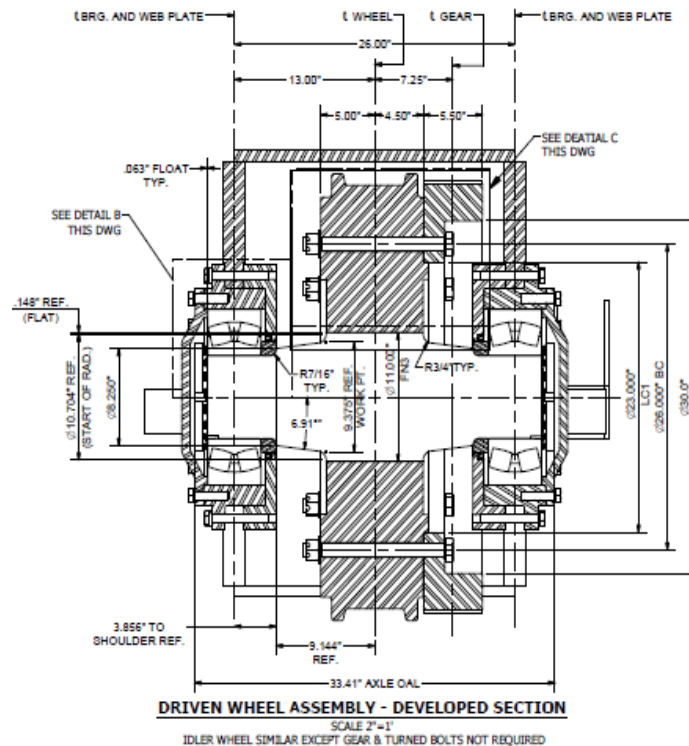
- Install the modified bogie trucks at the next location the following year (off season)
- Repeat the process for 8 years until all bogie trucks have been renewed

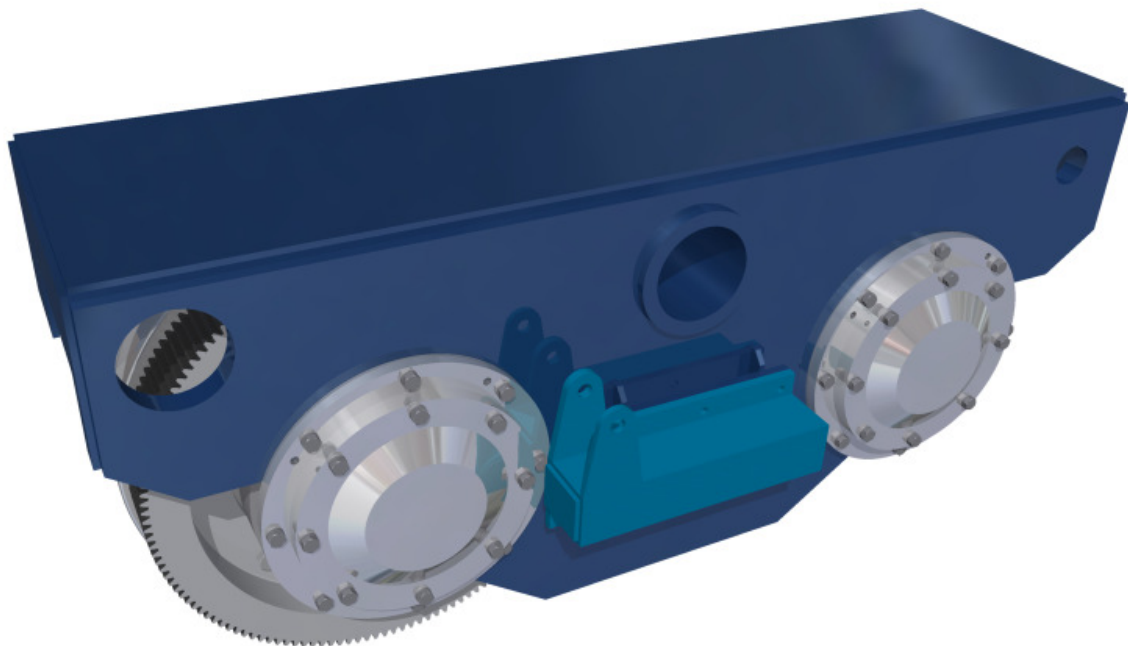
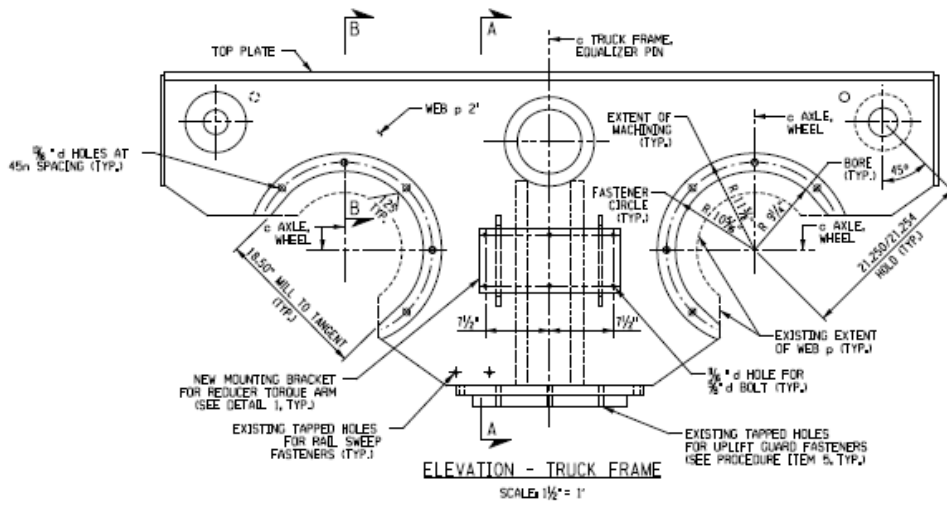
New Wheel Bearing Selection

Preliminary design for Option 2 (modified truck) revealed the 23234 bearing to be the largest that could be practically fit into modified trucks while maintaining the bearing centerline in line with the truck web plate centerline. When selecting a bearing size, the proximity of the bearing cartridge to the existing pinion shaft bearing was also a consideration. When using standard catalog formulae for L_{10} life, the 23234 (Option 2) bearing was calculated to have 12,000 hours L_{10} life as compared to the existing 22232 (Option 1) bearing which was calculated to have 2,500 hours L_{10} life.

A more rigorous analysis was done by Timken Company for the three bearing sizes used in Options 1, 2 and 3. Option 1 results were discussed above. For bearing 23234 (Option 2), a life of 5,213 hours was calculated for the most heavily loaded row of rollers. The expected bearing life based on average roof moves is 59 years for Options 2 compared to 3.5 years for Option 1 (307 hours L_{10} life). It should be noted that the life equations assume the axial load is always in the same direction and is always 15% of the radial load which is very conservative. In actual practice the axial load is most likely less than 15% of radial load most of the time and most likely changes direction depending on runway conditions and direction of travel. When the axial load direction changes it is likely the load shifts from one axle bearing to the other. These considerations likely extend actual bearing life beyond the calculated values above.

The detail of the upgraded wheel assembly with larger 23234 series wheel bearing is shown below. Other design improvement include the “Heavy Force Fit” axle to wheel interference as well as a bolted on ring gear.





Truck Re-machine Detail (above), 3D Model of Improved Bogie (below)

Implementation and Construction Progress

In the fall of 2013, the Mariners awarded a construction contract to implement the long term movable roof bogie system improvement plan. The plan utilizes a phased fabrication and installation approach. At the three year mark, the construction progress is as follows:

Phase 0 (2014) – Fabricated the 8 completely new bogie truck assemblies. All remaining jacking tabs were welded on throughout the system.

Phase 1 (February 2014) – The 8 new bogie truck assemblies were installed at roof panel P1 North Bogies 1 and 2. The 8 original bogie trucks that were removed from this location were sent for re-machining and fitting of the new wheel assemblies.

Phase 2 (February 2015) – The 8 re-machined and re-fitted bogie trucks from the previous year were installed at roof panel P1 South Bogies B1 and B2. The 8 original bogie trucks that were removed from this location were sent for re-machining and fitting of the new wheel assemblies.

Phase 3 (February 2016) - The 8 re-machined and re-fitted bogie trucks from the previous year were installed at roof panel P2 North Bogies B1 and B2. The 8 original bogie trucks that were removed from this location were sent for re-machining and fitting of the new wheel assemblies.



2016 Installation

Panel 2N Bogie 1

There are 5 more cycles to go.

The project will be completed in 2021!

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Forensic Testing

Northwest Laboratories, Seattle WA

Ultrasonic Testing

Wiss, Janney, Elstner, Inc.

Roof Wheel Load Analysis

Thornton Tomasetti

Fitness For Service Evaluation

Dr. Robert Connor

Bearing Analysis

Timken Bearing Company

Contractors

PCL Constructors (Emergency Wheel Change Out)

Roglins Inc. (Emergency Wheel Change Out)

Skanska Inc. (Interim and Long Term Installations)

Major Fabricators

Morgan Engineering - Alliance, Ohio

Steward Machine Company – Birmingham, Alabama

Seattle Gear Works – Seattle – Seattle, Washington