End of Co-op Presentation

By: Jessica Nicholson

Agenda

- About me
- Project overviews
  - Water transfer tube weld cracking investigation
  - Fuel pump idler gear slipped ring + thrust bearing commonization
  - Odin breather hose installation / fit solution
  - ISX12N drain tube failed check valve and material replacement
  - UIG idler shaft expansion and dual-cam service change
- Lessons learned
- Next steps

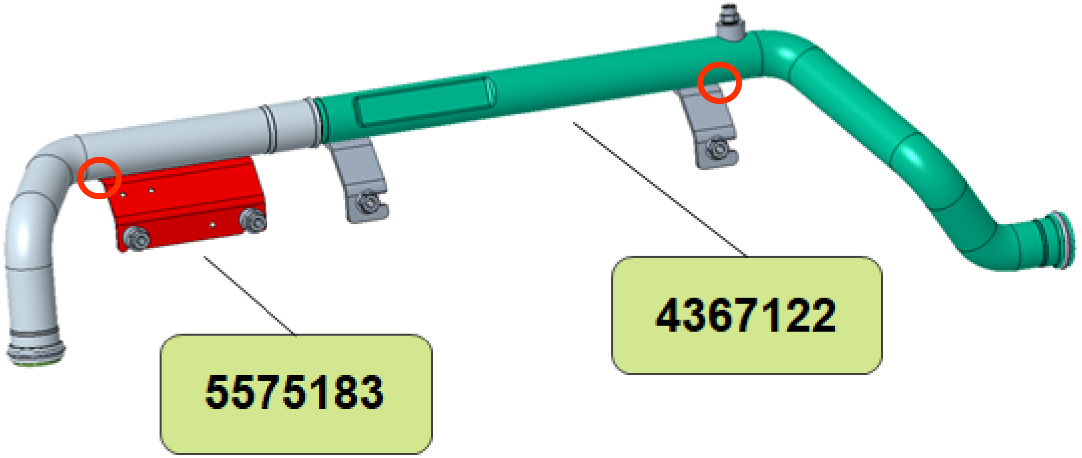
[redacted] = Censored for confidentiality

About Me

- Hometown: [redacted]
- Family: [redacted]
- University: University of Illinois, Urbana-Champaign
  - Mechanical Engineering major, Business minor
  - Involvement: Illinois Student Government, Society of Women Engineers, Illini Rotaract
- High school: Bartlett High School STEM Academy
- Hobbies / activities: Creating music, writing, wood burning
- First jobs: Math and reading Tutor at Kumon, Event Specialist at Advantage Solutions, Comcast Intern

Water Transfer Tube Weld Cracking Investigation

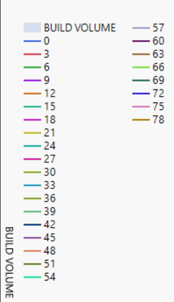
Failure Description



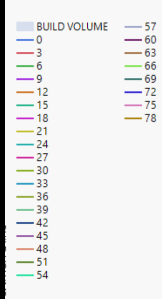
- Failure along circled weld joints
- Both tubes crack along weld joint connecting bracket to tube
- Crack can get worse and lead to coolant leak if not identified and fixed
- Many claims of cracked welds with leaking coolant

Worldwide vs Australia Claims Research

[redacted]



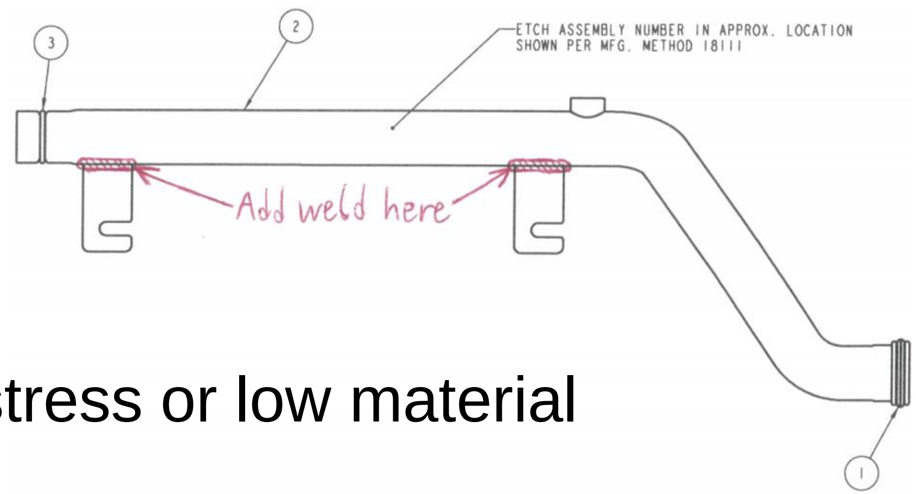
Build volume: 552,078
Claims: [redacted]
Avg RPH = [redacted]
Avg CPE = $0.42



Build volume: 17,934
Claims: [redacted]
Avg RPH = [redacted]
Avg CPE = $12.28

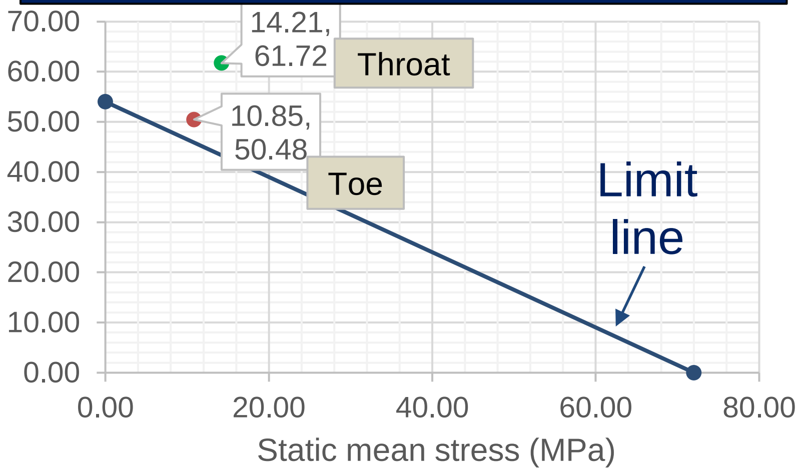
- Rottweiler engines experience a significantly higher failure rate in Australia than all other regions
- Australia experiences a higher failure rate among
  - All engine configurations, primarily due to greater Rottweiler purchases
  - Rottweiler engines, indicating a location-related cause of failure
- Failure modes are both a fault of the water transfer tubes in Rottweiler and location-related

Step 3 Solution

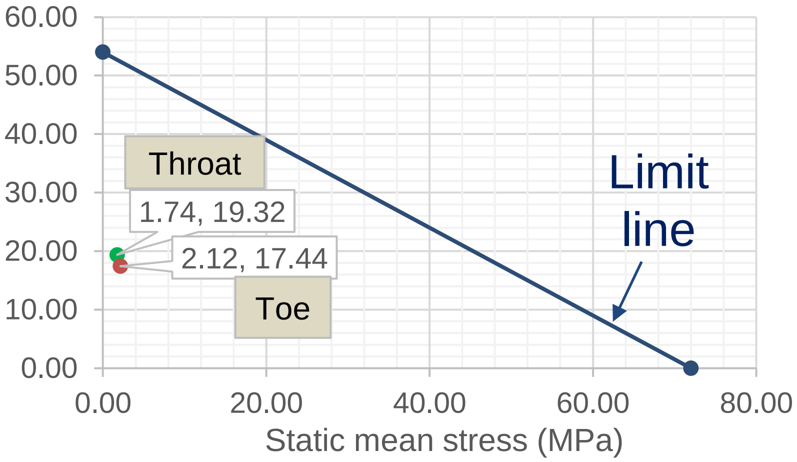


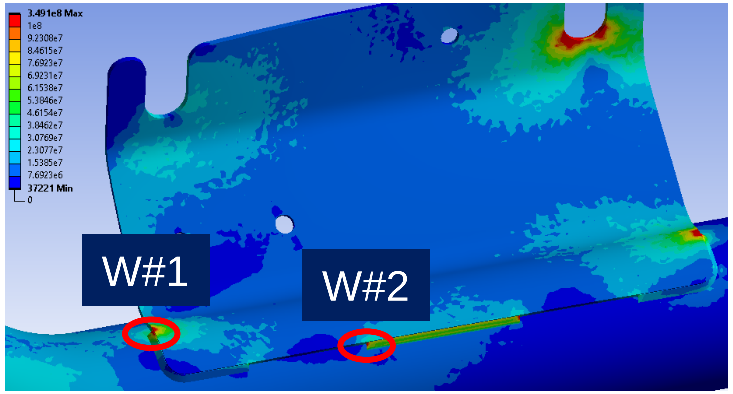
- Failure mode is most likely a form of excessive stress or low material strength due to quality issues
- Create additional welds behind each long tube bracket to secure to the pipe
- Create two 5/16” slot welds in the short tube bracket and add weld
- These temporary solutions will create a stronger joint to resist failure due to stress or low strength while I investigate the cause of the weld cracking

Short Tube Weld Stress, Current Design



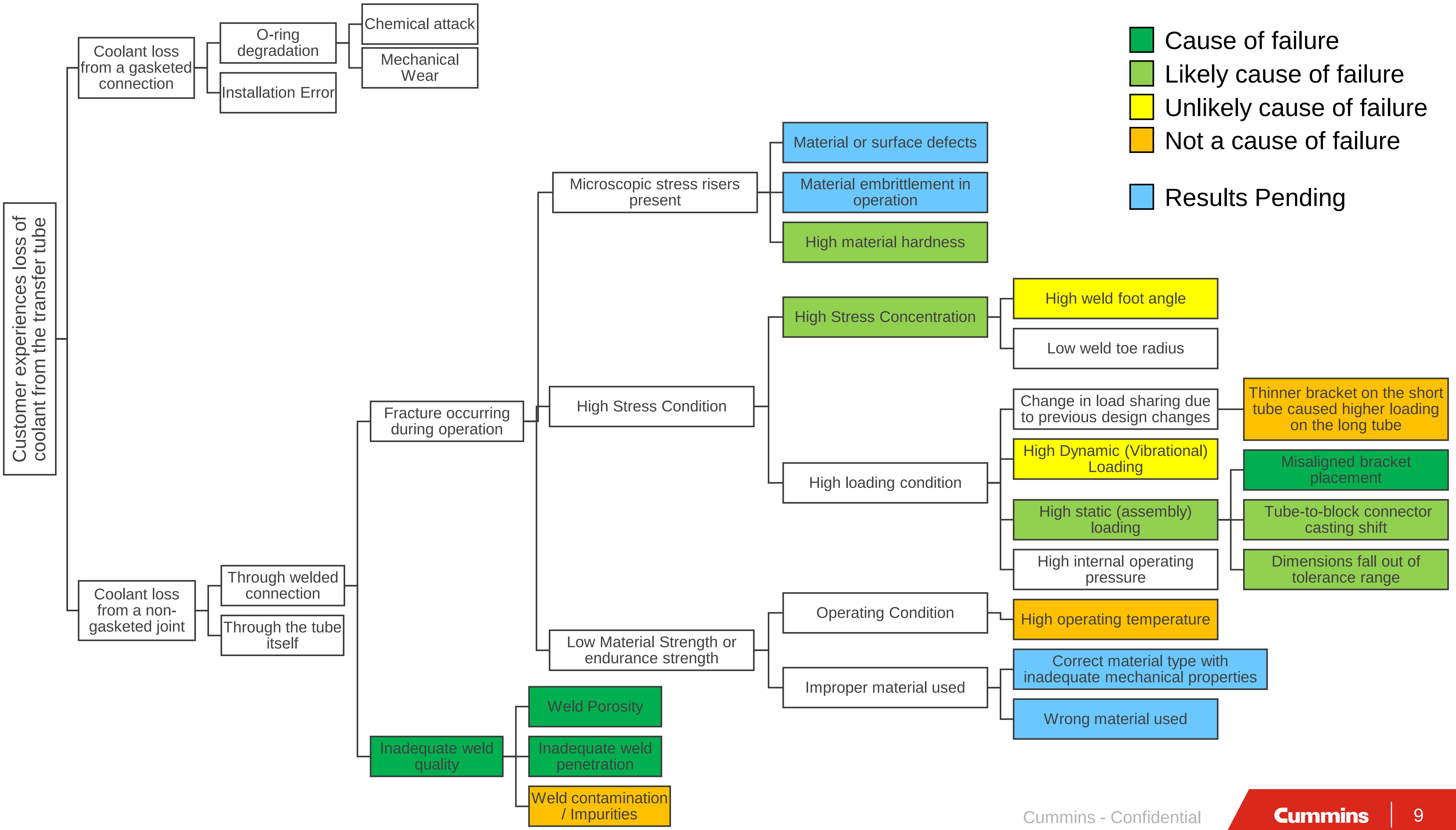
Short Tube Weld Stress, With Added Welds





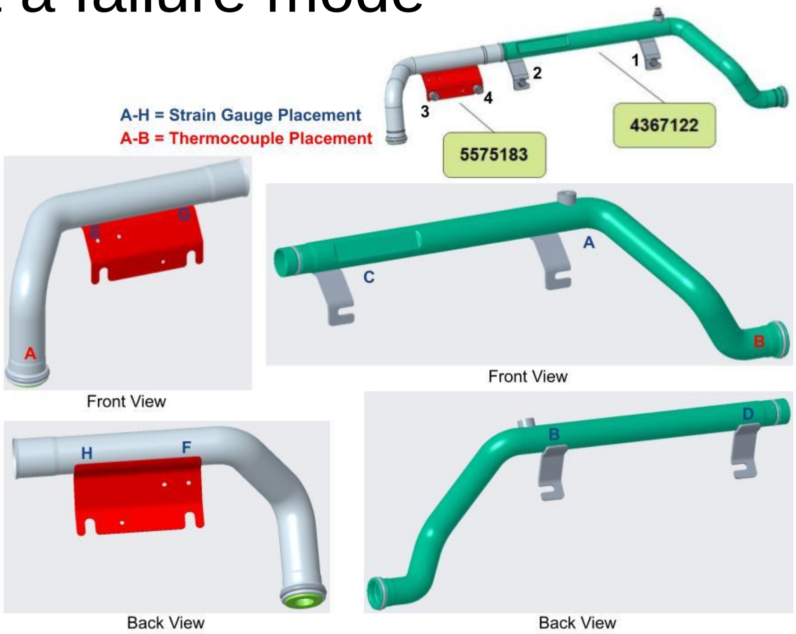
Cost Justification – All Regions

- 4367122 savings
  - Additional cost from supplier quote = in progress
  - Projected annual quantity: 15,364, Annual additional cost = in progress
  - Part-specific avg RPH = [redacted], avg cost per repair = $543.56, CPE = $26.38
  - Solution effective rate = 21.1%   Annual savings = $85,518.79
- 5575183 savings
  - Additional cost from supplier quote = $1.35 / piece
  - Projected annual quantity: 15,364, Annual additional cost = $20,741.40
  - Part-specific avg RPH = [redacted], avg cost per repair = $537.64, CPE = $4.32
  - Solution effective rate = 38.9%   Annual savings = $25,818.89
- Implement to reduce failures, evaluate other solutions from investigation
- Approximate savings, assumes direct correlation between stress and failure

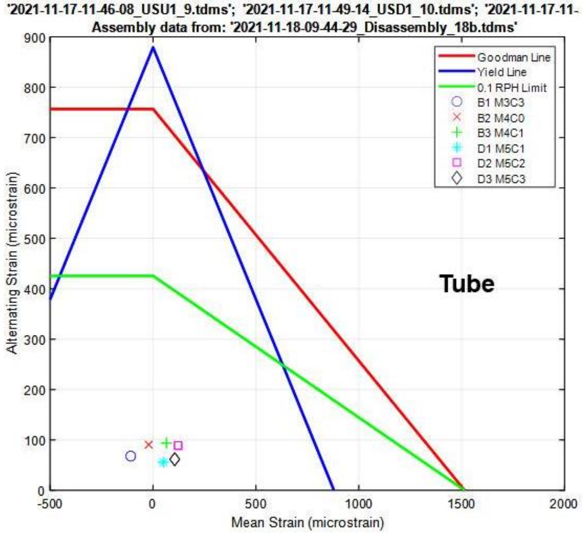

Customer experiences loss of coolant from the transfer tube
Coolant loss from a gasketed connection
O-ring degradation
Chemical attack
Mechanical Wear
Installation Error
Coolant loss from a non-gasketed joint
Through welded connection
Through the tube itself
Fracture occurring during operation
Microscopic stress risers present
Material or surface defects
Material embrittlement in operation
High material hardness
High Stress Condition
High Stress Concentration
High weld foot angle
Low weld toe radius
High loading condition
Change in load sharing due to previous design changes
Thinner bracket on the short tube caused higher loading on the long tube
High Dynamic (Vibrational) Loading
High static (assembly) loading
Misaligned bracket placement
Tube-to-block connector casting shift
Dimensions fall out of tolerance range
High internal operating pressure
Low Material Strength or endurance strength
Operating Condition
High operating temperature
Improper material used
Correct material type with inadequate mechanical properties
Wrong material used
Inadequate weld quality
Weld Porosity
Inadequate weld penetration
Weld contamination / Impurities
Cause of failure
Likely cause of failure
Unlikely cause of failure
Not a cause of failure
Results Pending

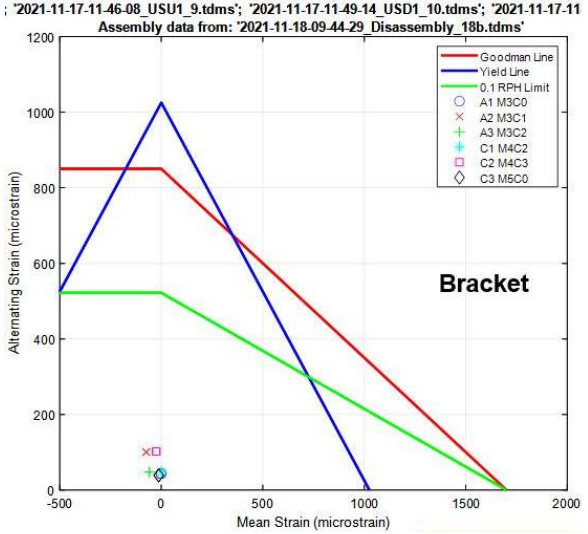
High Stress Condition – Strain Gauge Test

- Strain gauge engine operation
- Measured set of transfer tubes
- Static, mean, alternating stress
- Low + high idle, loaded, unloaded
- Not a failure mode

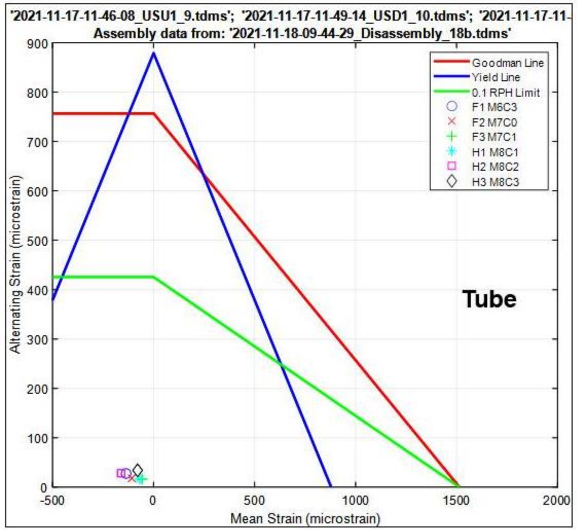


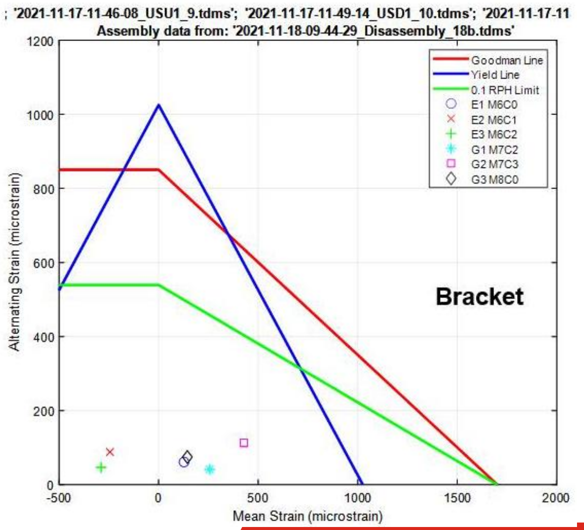
4367122



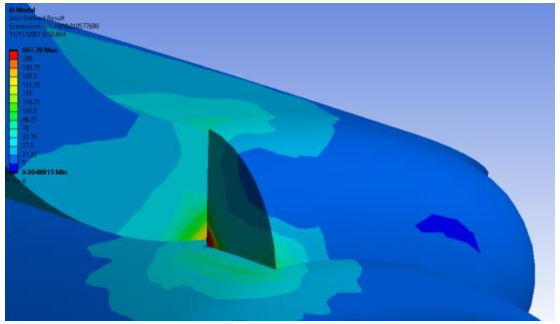


5575183

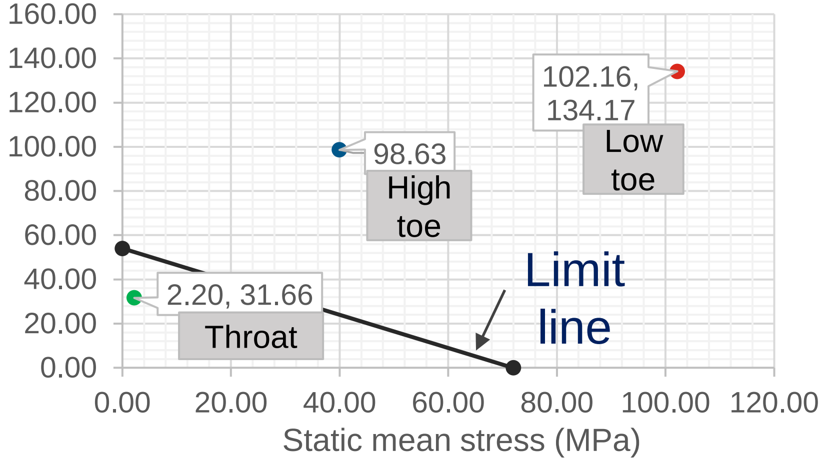


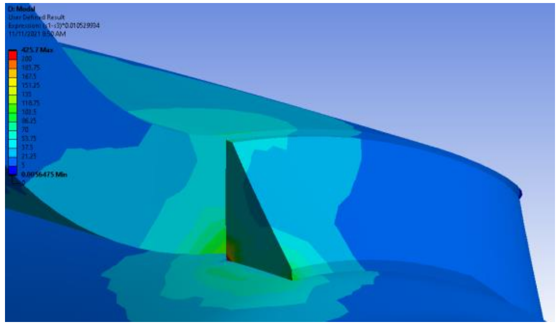


High Stress Concentration – Weld Geometry ANSYS

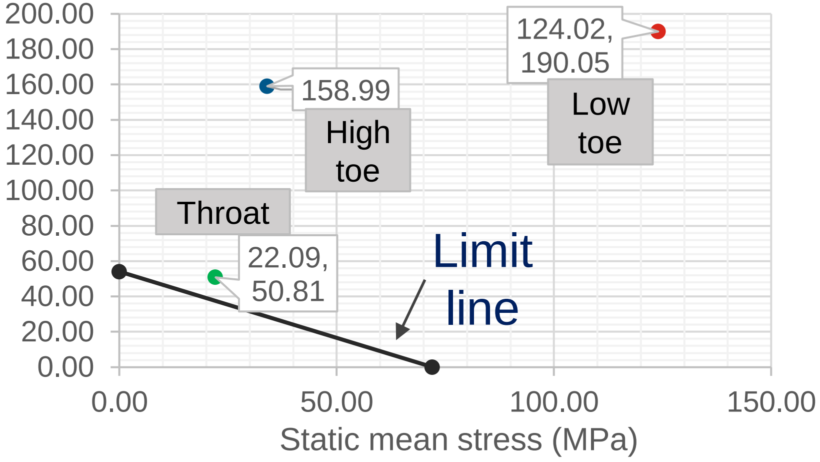


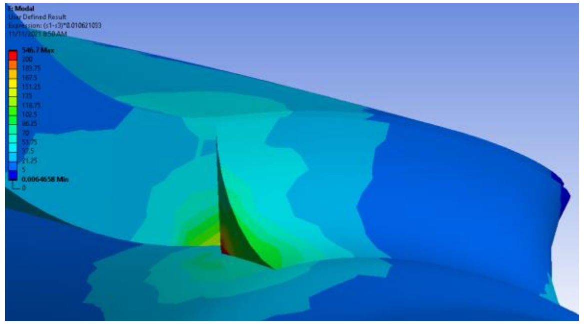
Long Tube, Weld 1, Convex



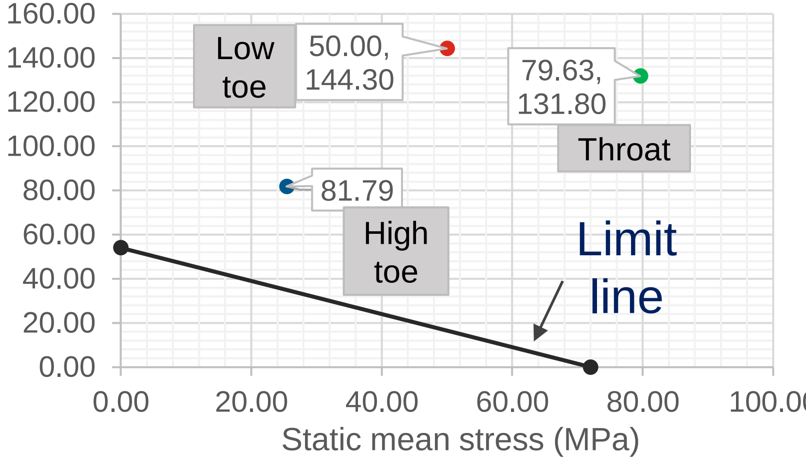


Long Tube, Weld 1, Flat



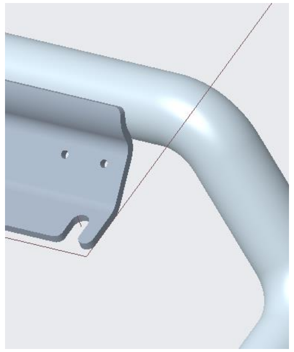


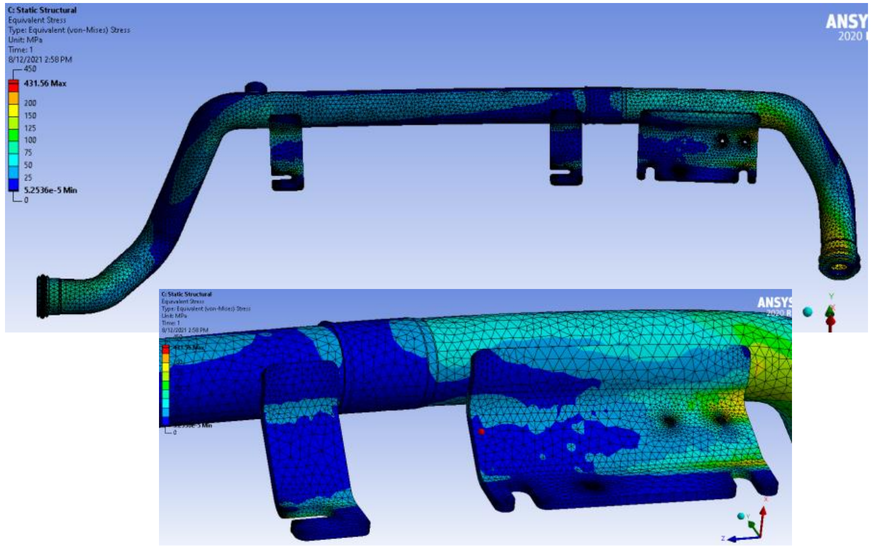
Long Tube, Weld 1, Concave



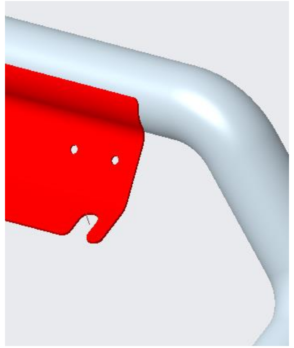
Past Design Changes – Thinner Bracket

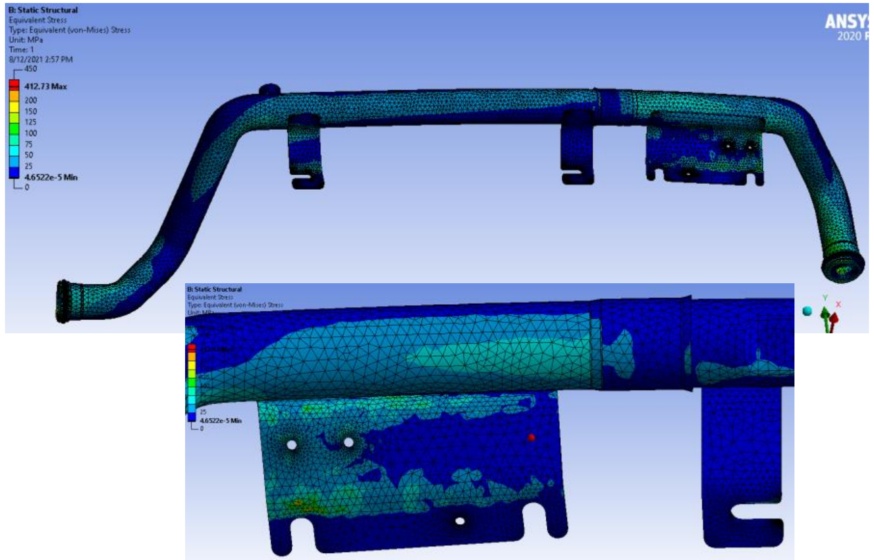
Old Bracket: 4.8 mm





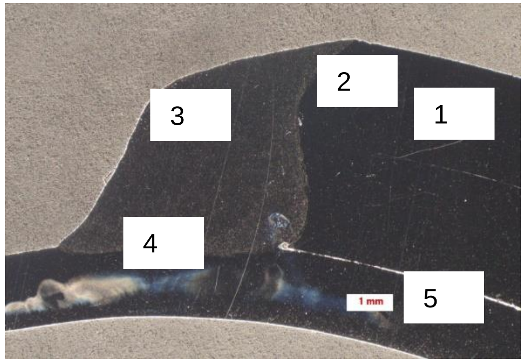
New Bracket: 1.5 mm

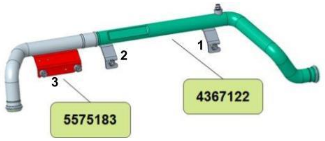


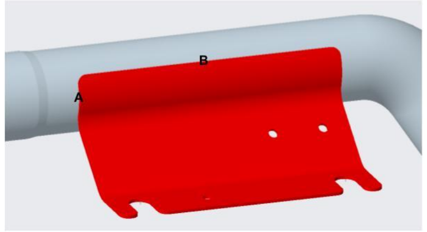


- Two ANSYS simulations performed for 1.5 and 4.8 mm bracket thicknesses
  - Given displacement perpendicular to each bracket face = 1 mm
  - Same fixed support points at both ends of transfer tubes
  - Both full-system simulations measure equivalent stress
- Locations of interest – each welded bracket-to-pipe joint
  - Stress measured at these locations; stress distribution shown
  - Stress distribution compared between simulations

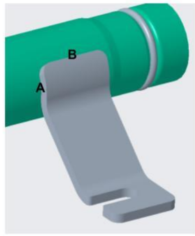
High Material Hardness







Back of Bracket View (3)

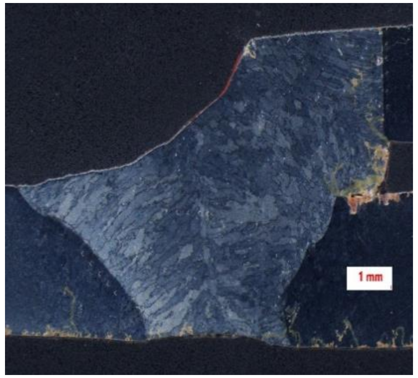


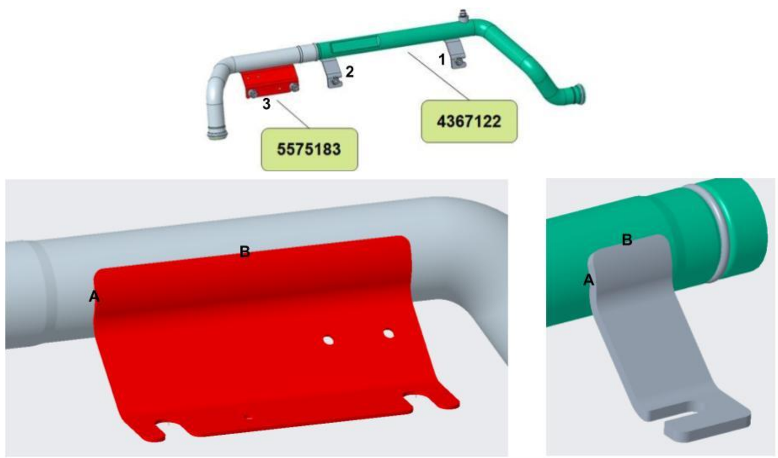
Back of Bracket View (1 and 2)

- High hardness leads to low ductility and low fracture toughness
- Hardness measured in multiple locations to ensure no welds exceed 350 HV
- No hardness measurements exceed this specified maximum
- Hardness expectedly increases at HAZ and weld, joints more susceptible
- Not a primary mode of failure, only a concern with high stress conditions

Area	Description	Spec (HV)	Bracket 1						Bracket 2						Bracket 3			
			Long Failed Tube 1		Long Failed Tube 2		New Long Tube		Long Failed Tube 1		Long Failed Tube 2		New Long Tube		Short Failed Tube 1		New Short Tube	
			A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
1	Bracket	N/A	143	166	157	151	162	156	144	151	148	154	149	156	129	134	141	148
2	HAZ Bracket	<350	142	165	170	157	162	148	152	153	168	161	185	140	141	147	153	169
3	Weld	<350	224	190	249	188	236	186	229	191	229	193	242	183	212	216	201	205
4	HAZ Pipe	<350	168	150	154	145	164	144	155	150	127	149	164	142	152	146	140	147
5	Pipe	>85	123	131	122	135	116	118	131	136	120	127	129	119	115	111	121	106

Improper Weld Qualities



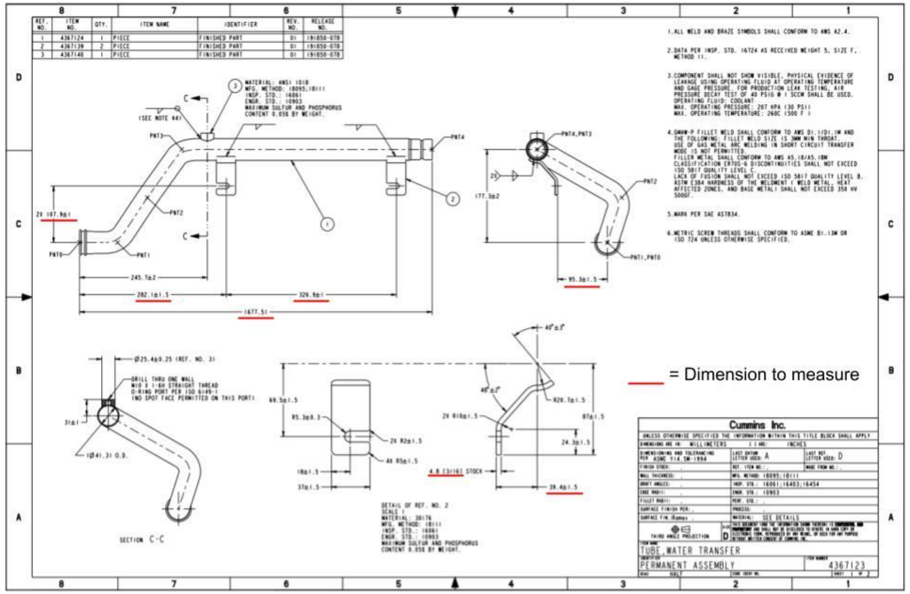


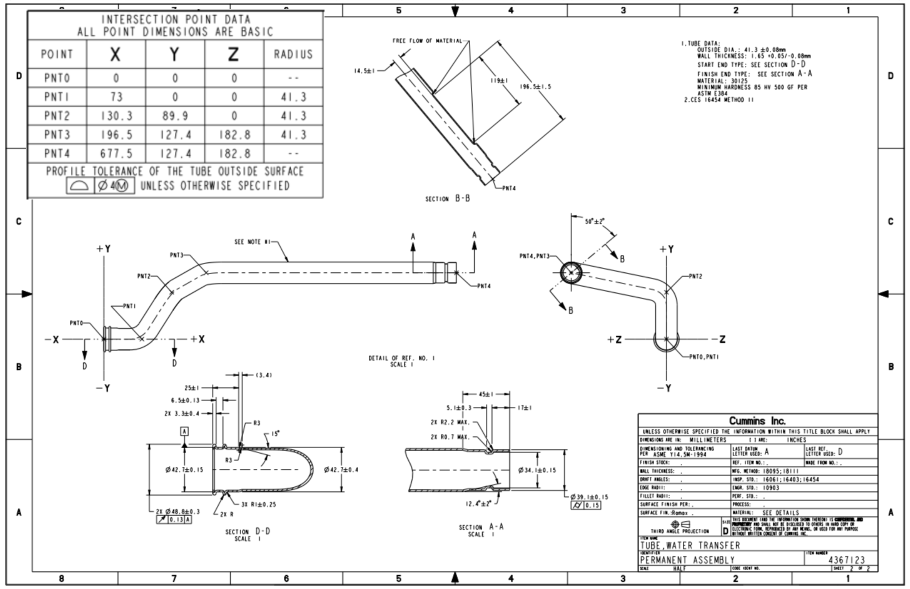
Back of Bracket View (3)

Back of Bracket View (1 and 2)

- Inspected weld properties in failed and new samples
  - ISO 5817 standards used to evaluate weld properties
  - Cross-section cut to evaluate each weld
- Evaluated properties: Throat size, porosity, penetration, root fusion
- Multiple imperfections found, **likely modes of failure**
  - Most welds on bracket 1 and 2 do not meet the minimum throat size, 3 mm
  - Root gaps of several welds exceed max limit of 0.6 mm for a 3 mm throat size
    - The larger the root gap, the smaller the area of fusion and the weaker the joint
  - Most welds exceed allowable porosity, 1.5%
  - Some welds have minimal weld penetration, only one critically inadequate weld
  - Clearance between bracket and pipe near weld that can weaken the joint
- Supplier quality issue; contacted suppliers to make changes in weld process

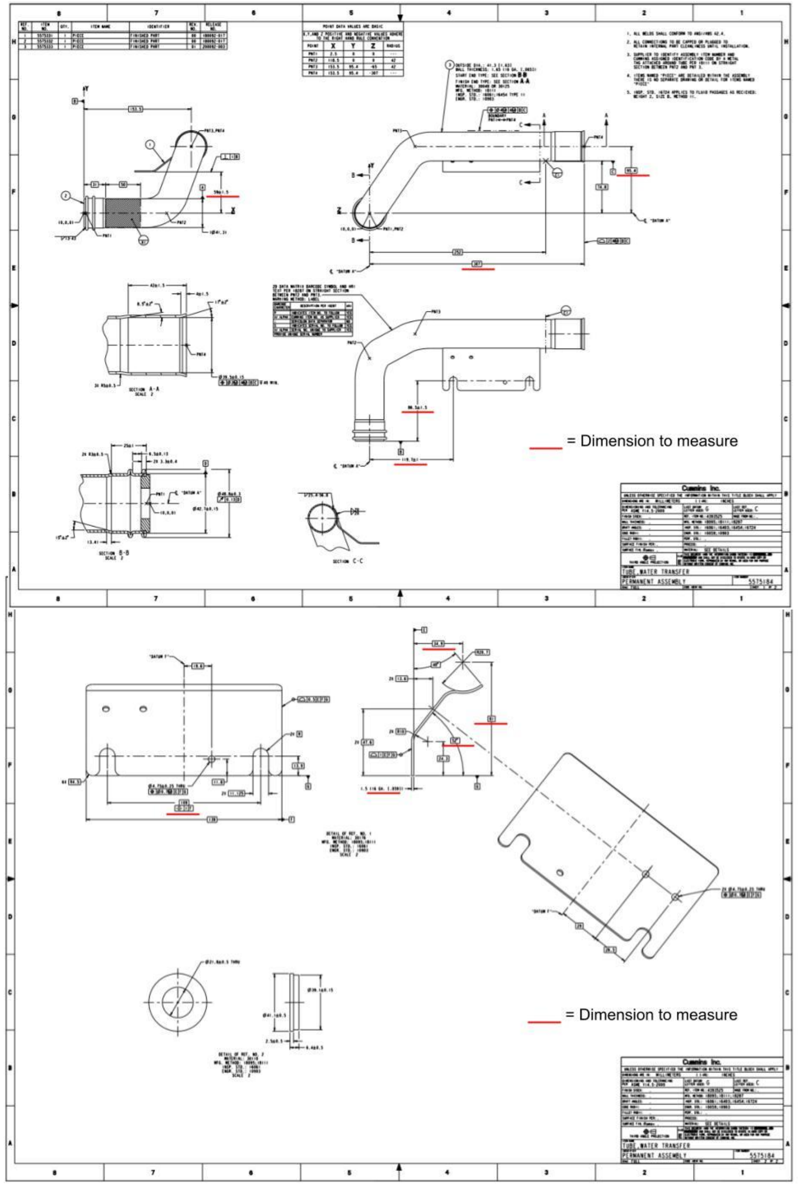
Dimensional Quality – Transfer Tube 4367122





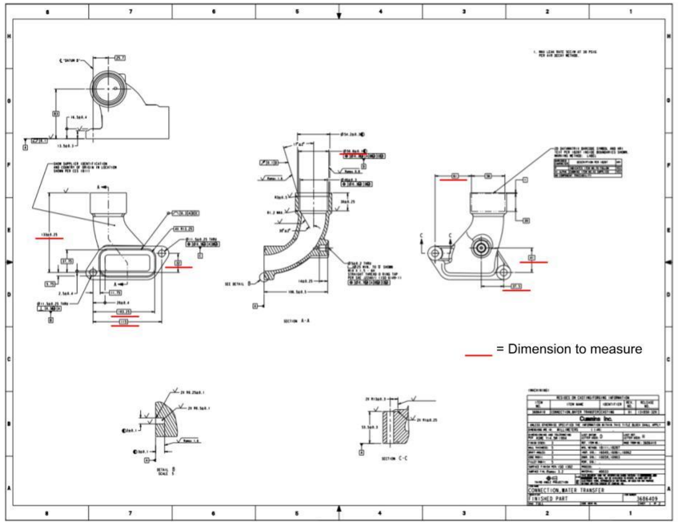
- No dimensional variation greater than 0.5 mm between samples
- Some dimensions consistently deviate from print tolerance
- Failure due to excess static stress from:
  - Deviation at point 2 is **not** a failure mode
    - Point 2 position is not relevant to fit to the block
  - Deviation at point 4 x-direction is a **low-likelihood** failure mode
    - Point 4 position has a minimal effect on tube fit to the block
  - Variation between parts is **not** a failure mode

Dimensional Quality – Transfer Tube 5575183



- More dimensional variation than long tube, no greater than 3 mm
- Failed dimensions are not consistent, but many dimensions are out of tolerance
- Failure due to excess static stress from:
  - Deviation at point 4 is **low-likelihood**
    - Point 4 position has minor effect on fit
  - Incorrect bracket dimensions is a **low-likelihood** failure mode
  - Incorrect bracket position in any direction **is** a likely failure mode
  - Variation between parts is **low-likelihood**

Dimensional Quality – Connector

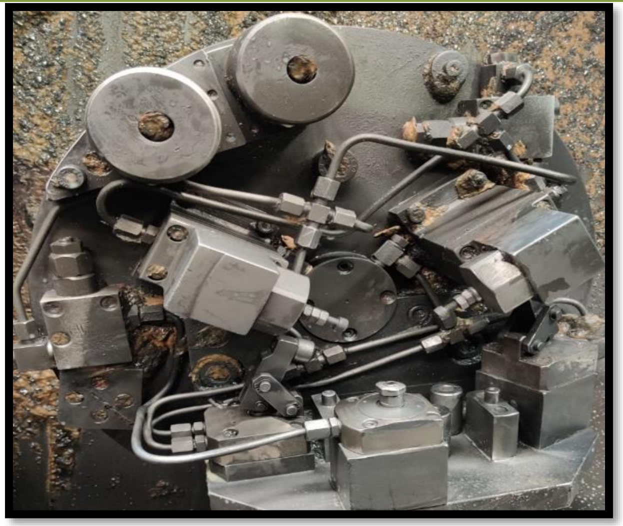


- Dimensional issues commonly found in these connectors
- 4 samples chosen with visually-evident machining errors
- Measurement failures with an impact on pipe fit and internal stress
  - Datum C position deviates in the x direction
  - Distance from datum B to end of tube fit variation
    - These dimensions have an impact on the x-direction fit of the connector
  - Datum D position variation
    - This position affects the horizontal x-y location that the pipe fits to the connector
  - Distance from datum B to top of connector
    - If this dimension is too long, it may push pipe upward (z-direction)
- These dimensional errors found in the 3686409 connector **are** a likely cause of failure due to internal stress caused by imperfect fit

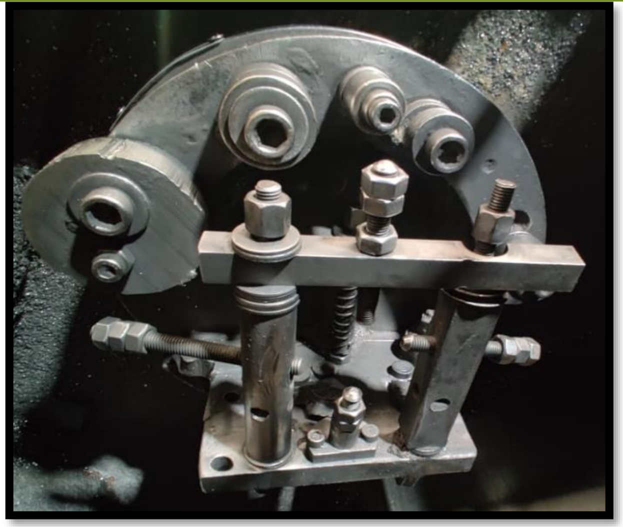
SQIE Solution – Connector Dimensional Quality

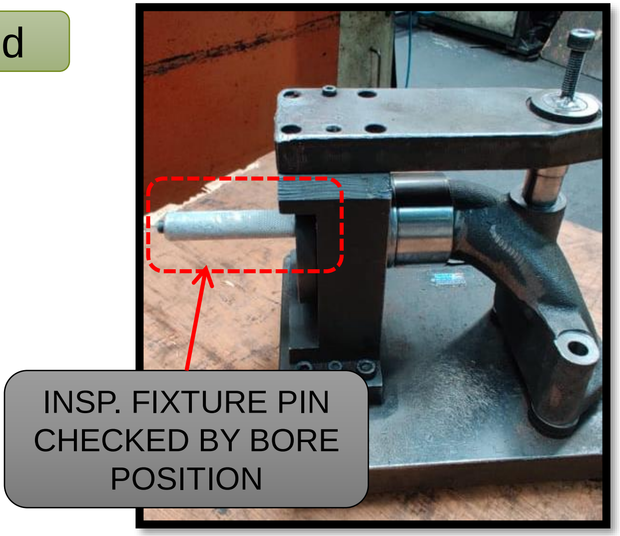
- Connector supplier: Mahabal Metals
- 3 hydraulic and 1 manual clamping mechanism in each CNC machine
- Manual clamping mechanism identified as cause of dimensional issues
- Replaced manual clamping mechanism with hydraulic clamping mechanism
- Should significantly reduce error in positioning from manual mechanism

Hydraulic Clamping Production Method



Manual Clamping Production Method





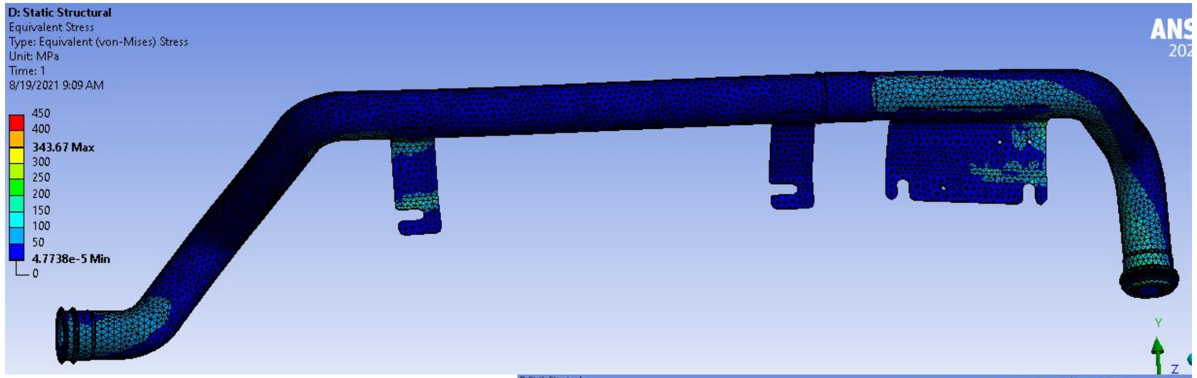
SQIE Solution – 5575184 Weld Quality

As-run

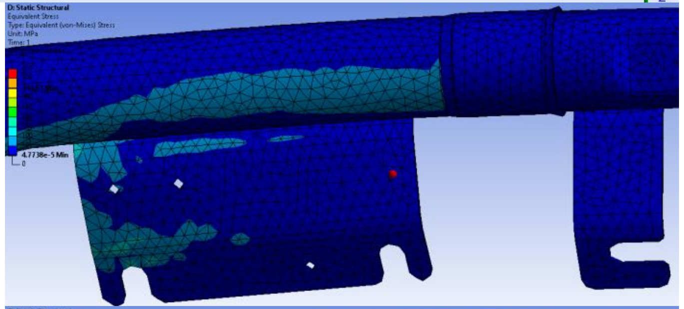
After changes

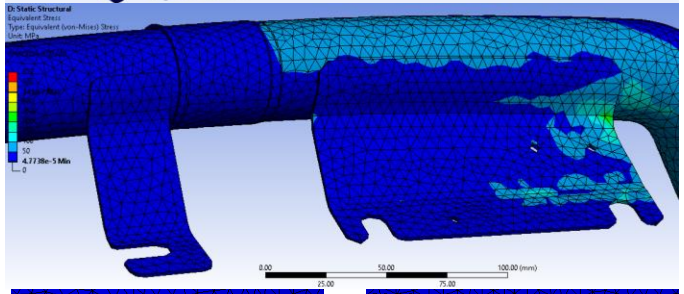
- 5575184 supplier: MEC
- Weld heat increased to ensure adequate weld penetration
  - Still within specified WPS parameters
  - Hardness testing to confirm no significant change
  - Did a “runout” to ensure no burnthrough or undercut
- Torch position and work angles changed in welding robot program to eliminate ropey appearance – perpendicular weld application
- Dye penetrant testing conducted to ensure no undercut or cracking issues
- Evaluated fixturing between bracket and pipe to ensure correct placement
- Reviews conducted for use of anti-spatter and cleaning operations
- Communicated changes and expectations with welders/operators, quality control, engineering, and production management

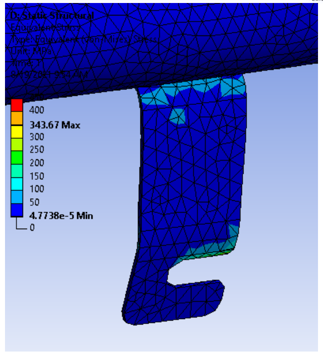
Design Solution – Stress

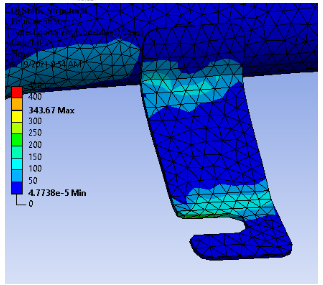


- Excess static / dynamic stress is not a failure mode
- However, implement a cost-justifiable solution to reduce overall loading on the system
  - Will take additional loading off joints to offset any remaining quality issues weakening the joint
    - Weld quality issues that lower allowable joint stress
    - Dimensional quality issues that increase assembly stress
- **Change long tube bracket thickness to 1.5 mm**
  - Short tube brackets have already been reduced to this
  - Thinner brackets reduce stress due to deflection
- ANSYS analysis completed to measure stress distribution change



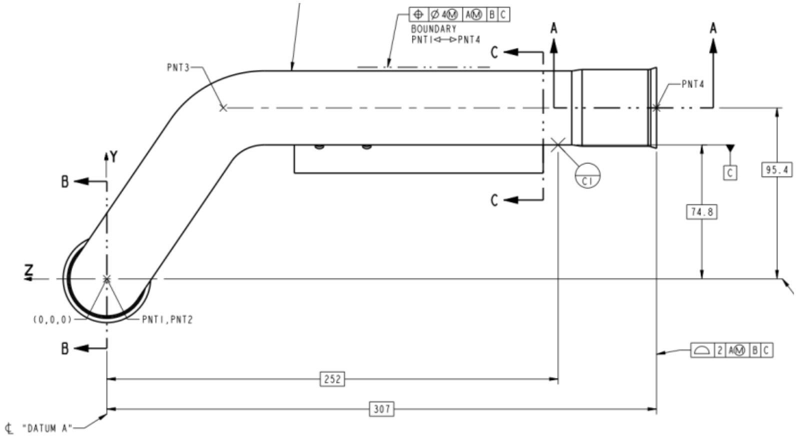




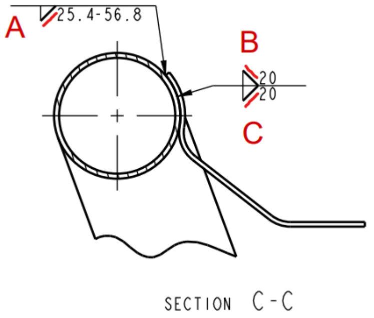


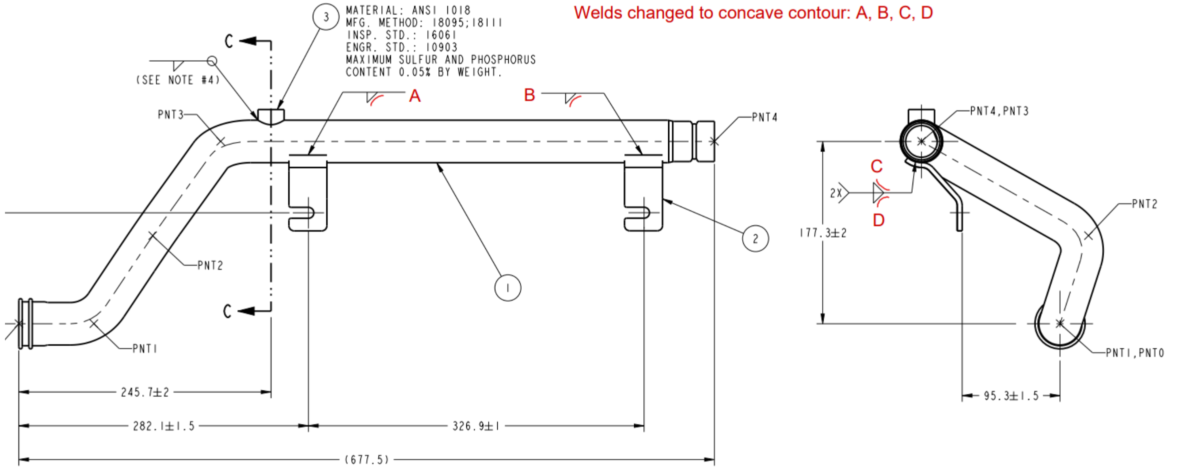
Design Solution – High Stress Concentration

- Weld geometry is currently unspecified on both transfer tube prints
  - Reflected by inconsistent welds seen on new and failed samples
- Add weld geometry specifications to each tube based on ANSYS analysis
  - Determined the best weld geometry to reduce stress concentration (slide #_)
  - Short tube weld contour: Flat weld
  - Long tube weld contour: Concave weld



Welds changed to flat contour: A,B,C





Next Steps and Recommendations

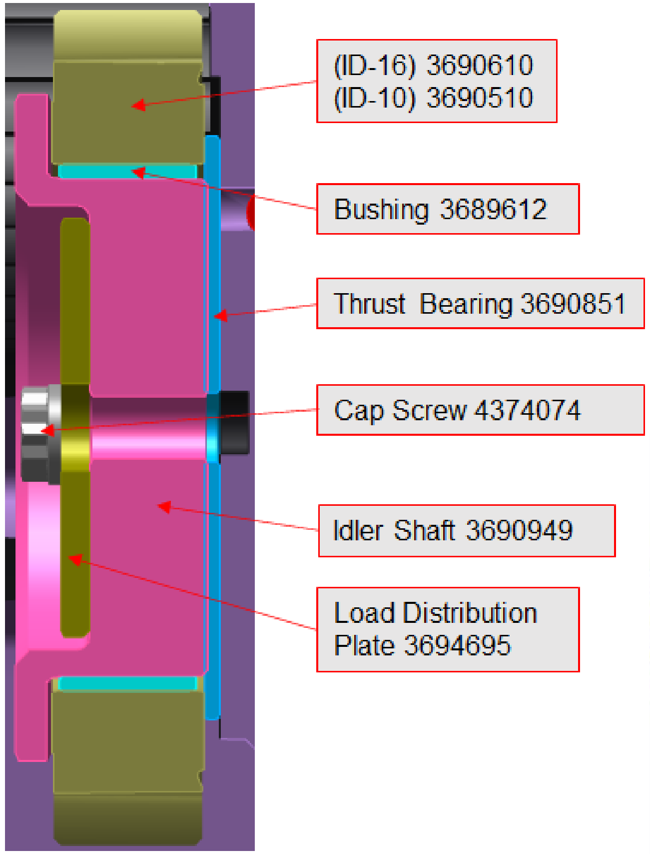
- Dimensional quality issues
  - 5575183 bracket out-of-tolerance dimensions, inform supplier
  - If fit is an issue once dimensions are fixed, evaluate print dimensions
- Weld quality issues
  - 4367122 supplier has not yet responded to SQIE, follow up
  - Have supplier evaluate weld application process that may cause quality issues
- Static/dynamic stress
  - Validate durability of 1.5 mm long tube bracket thickness change
    - Cost justify with quote from supplier, create CTR -> CR to make this change

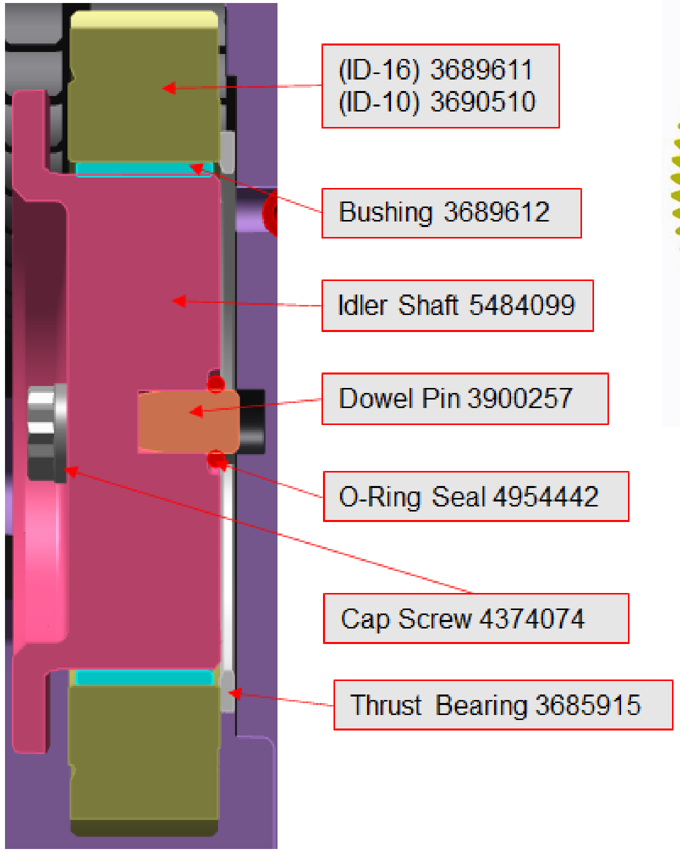
Next Steps and Recommendations

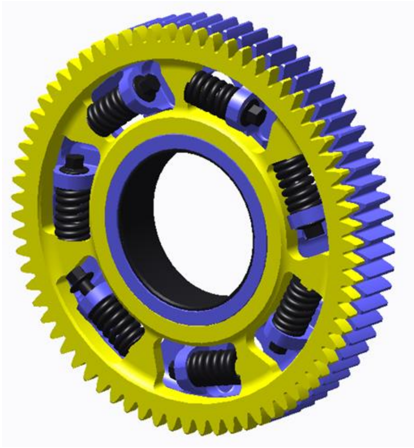
- Stress concentration
  - Repeat ANSYS analysis for thinner 4367122 bracket to confirm best weld geometry
  - Create CTR -> CR for weld geometry change with analysis results, cost justify
- If significant failure still occurs after current solutions are implemented, implement step 3 additional welds as permanent solutions
  - This case would make the solution likely to be cost justified
    - Continued failure leads to projected failure reduction from this solution to offset cost
    - Follow up with long tube supplier for added weld quote, finish cost justification

Fuel Pump Idler Gear Thrust Bearing Commonization

Adjustable vs Fixed Fuel Pump Idler Gear







Adjustable: Plate thrust bearing

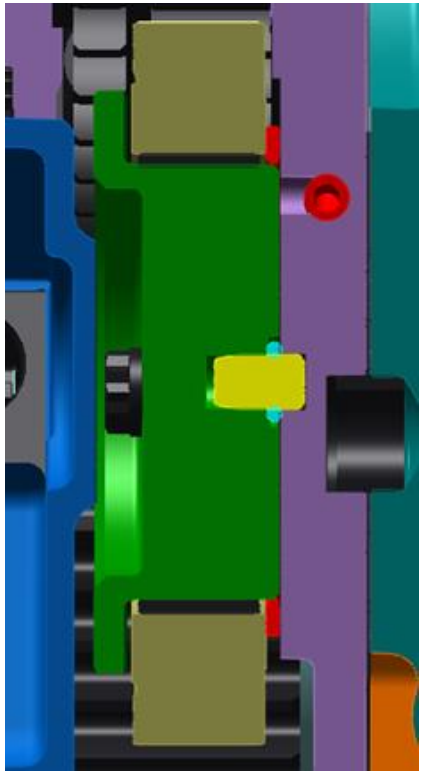
- Solid driving gear
- Distribution plate to adjust

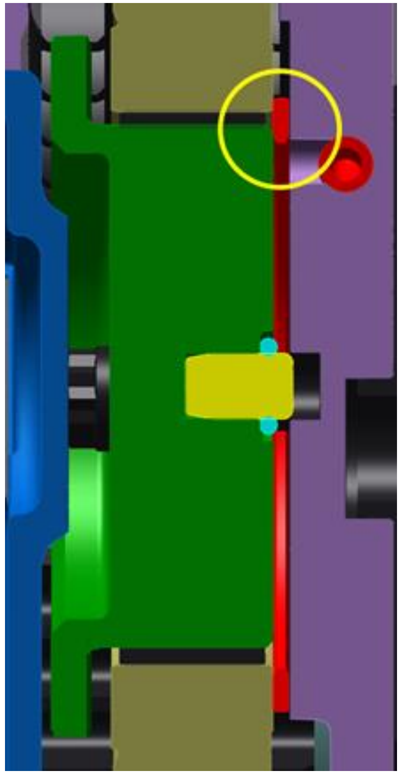
Fixed: Ring thrust bearing

- Scissor driving gear
- No adjustment for lash

Failing Component and Effects

Correct Design
(section view)





Failed Design
(section view)

- **Fixed system fails**
- Thrust bearing slips behind FPIG idler shaft
- Slip occurs during installation

- Issue often goes unnoticed
  - Ring is thin
  - Changes in idler shaft position are difficult to visually identify
  - Ring is obscured by gear and idler shaft once installed
- Effects of failure
  - Idler shaft not fully connected by dowel pin, can misalign or detach
  - Efficiency loss and excess radial load occur
  - Gears fall out of mesh, short-term engine failure

ISX3 – RPH and CPE Plots

[redacted]

- Build volume: 113,866, Claims: [redacted], Average RPH = [redacted]
  - BKTB: [redacted], BKIF: [redacted], BKIS: [redacted]
- Average replacement cost = $2,864.12, Average CPE = $7.67

ISX1 Comparison – RPH and CPE Plots

[redacted]

- Build volume: 468,391, Claims: [redacted], Average RPH = [redacted]
  - BKTB: [redacted], BKIF: [redacted], BKIS: [redacted]
- Average replacement cost = $10,240.43, Average CPE = $4.37

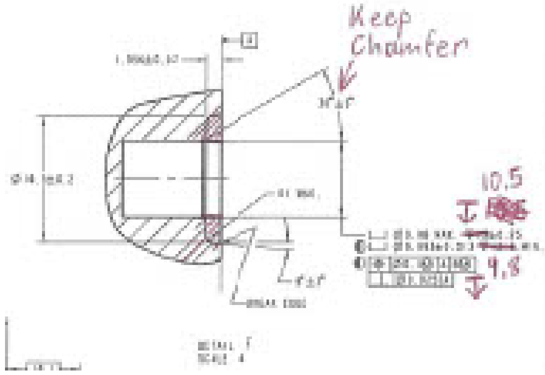
Claims Research Results

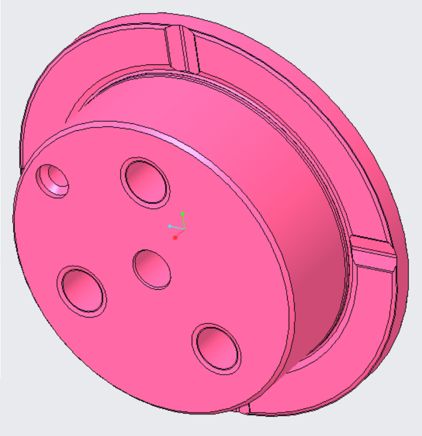
- According to claims research, solving this slipped ring issue can:
  - Reduce average RPH for recorded idler gear, idler shaft, and thrust bearing failures of ISX3 systems by up to 0.268
  - Eliminate an average of $2,864.12 in cost per prevented claim
  - Reduce average CPE from these ISX3 failures by up to $7.67
- ISX1 system has a lower rate and cost of failure than ISX3
  - CPE difference between ISX3 and ISX1 = 7.67 - 4.37 = **$3.30**
  - All ISX3 thrust rings will be replaced by the plate thrust bearing
    - ISX3 systems will experience a similar failure rate to ISX1 systems
  - This causes an expected $3.30 CPE reduction in ISX3 engines
- Compare predicted CPE reduction to cost of implementation

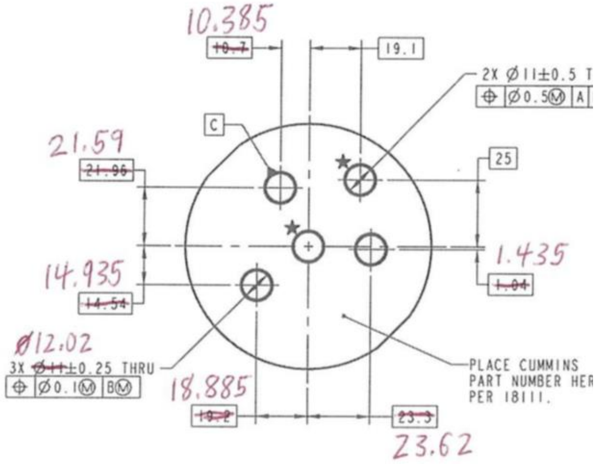
Solution and Justification

- Prevent the ring thrust bearing from slipping during installation
- **Design a plate thrust bearing for the fixed system** to replace the ring
- Opportunity to commonize the thrust bearing for both the fixed and adjustable systems; new plate design that works for both systems
- Must redesign components of fixed system to accommodate plate
  - Both systems perform the same function, are similarly assembled
  - Adjustable system does not experience the same failure mode
    - Plate thrust bearing bolted into the housing with the idler shaft
    - Cannot slip out of place or misalign the idler shaft
  - Plate thrust bearing advantages
    - Cheaper by part than the ring thrust bearing (long-term savings)
    - Proven reliable, lower failure rate shown by ISX1 claims data

Redesign Process







- Fixed idler shaft redesign
  - Reduced the width of the thinner section of the idler shaft by 2.5 mm to accommodate a plate thrust bearing rather than the ring
  - Reduced the depth of the dowel pin hole by 2.5 mm to account for the additional space taken by the plate thrust bearing
  - Removed the obsolete O-ring seal and filled in the space within the idler shaft
- Commonized plate thrust bearing redesign
  - Eliminated the ring design
  - Modified the original plate thrust bearing 3690851 to fit to fixed design as well
  - Moved cap screw hole positions to accommodate hole position difference
  - Changed hole diameters to 12.02 mm
    - Each hole fits the corresponding screw position of both idler shafts

Design Validation Plan

#	DFMEA	Identified Risk	Validation Tasks	Pass/Fail Criteria	
1	Commonized Plate Thrust Bearing	Experiences a slight position shift and becomes misaligned. This can happen in both designs during installation or in the adjustable design during adjustment, and it leads to uneven / excessive wear due to friction on the edges of the plate.	Complete DVA, calculate worst-case friction wear, fit test the commonized thrust bearing on both designs, and perform an engine endurance test if misalignments are unavoidable	Thrust bearing must be axially aligned with the idler shaft at the oil input and center holes within 2.5 mm	* = risk too low to justify testing and/or similar designs dont fail
2		Experiences a major position shift and loses some contact with the idler gear. This can happen in both designs if the cap screws lose enough preload to angularly shift, and it leads to excess friction wear with a loss of hydrodynamic oil film.	Use DVA results to ensure idler gear will not lose contact with thrust bearing, perform engine audit test for oil leak	No oil leak detected	
3		Components slip due to excessive radial load and/or inadequate preload during installation or due to corrosion	None	Thrust bearing, idler shaft, and gear housing must be axially aligned at the oil input and center holes within 2.5 mm	*
4	Non-Adjustable Idler Shaft	Slips out of place due to excessive radial load and/or inadequate preload	None	Thrust bearing, idler shaft, and gear housing must be axially aligned at the oil input and center holes within 2.5 mm	*
5		Oil does not cover full width of bushing during operation, leading to high levels of friction, loss of efficiency, and failure	Compare oil output and bushing coverage of current system and revised system	Comparable amounts of oil coverage and oil loss	*

Tolerance Analysis – Thrust Bearing Alignment

- DVA measurements of each system at critical edge:
  - Each aligned system: 6.07 mm +- 0.1901
  - Narrow edge, fixed system misaligned: 5.07 mm +- 0.4404
  - Narrow edge, adjustable system misaligned: 2.94 mm +- 0.4185
  - Narrow edge, original adjustable system: 3.96 mm +- 0.4185
- Magnitude of misalignment must not allow thrust bearing to lose contact with the idler gear as the bushing would be exposed and leak oil
- Additional calculation using DVA of worst-case thrust bearing misalignment
  - Added clearance between gear + bushing inner diameter and idler shaft
  - Max distance from thrust bearing edge to inner gear edge: **0.958 mm +- 0.4241**
- **No significant risk** of failure due to leakage from misalignment

Production Cost Savings

- Current component costs
  - Plate thrust bearing
    - Current annual volume = 4,056
    - Cost per thrust bearing = [redacted]
    - Total annual cost = [redacted]
  - Ring thrust bearing
    - Current annual volume = 13,592
    - Cost per thrust bearing = [redacted]
    - Total annual cost = [redacted]
  - Non-adjustable idler shaft
    - Current annual volume = 13,592
    - Cost per idler shaft = [redacted]
    - Total annual cost = [redacted]
- New plate thrust bearing volume
  - New production volume is the demand for both thrust bearings = **17,648**
- New component costs
  - Plate thrust bearing
    - Projected annual volume = 17,648
    - Quoted cost per thrust bearing = [redacted]
    - Total annual cost = [redacted]
  - Non-adjustable idler shaft
    - Projected annual volume = 13,592
    - Quoted cost per idler shaft = [redacted]
    - Total annual cost = [redacted]
- Annual cost savings = **$10,692.85**

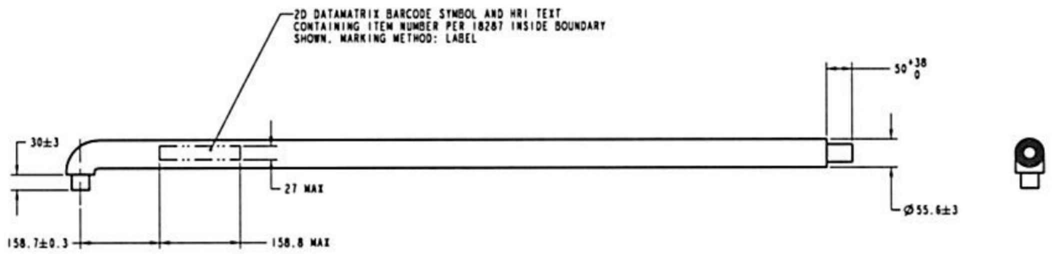
Fixed Costs, Claims Savings, and Break-Even Point

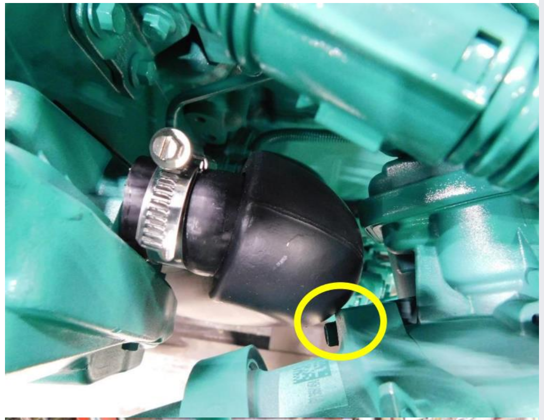
- Fixed costs and production lead-time
  - Thrust bearing tooling modification cost = [redacted], lead time 10 weeks
  - Idler shaft tooling modification cost = [redacted], lead time 8 weeks
- Break-even point = 1.68 years
  - Considering annual production savings vs all fixed costs
- Claims cost comparison
  - Predicted reduction in ISX3 CPE with design change = $3.30
  - ISX3 average annual production volume = 8,133
  - Projected annual claims cost reduction / savings = $26,838.90
- This redesign creates a cost reduction in both production and warranty costs
- Total annual savings = **$37,531.75**, Actual break-even point = 0.48 years
  - Considering annual production savings **and projected claims reduction**

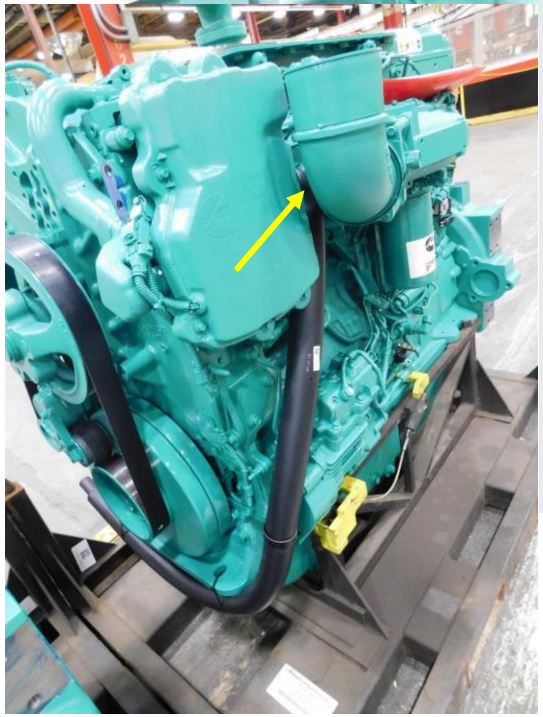
Odin Breather Hose Interference and Insulation Removal

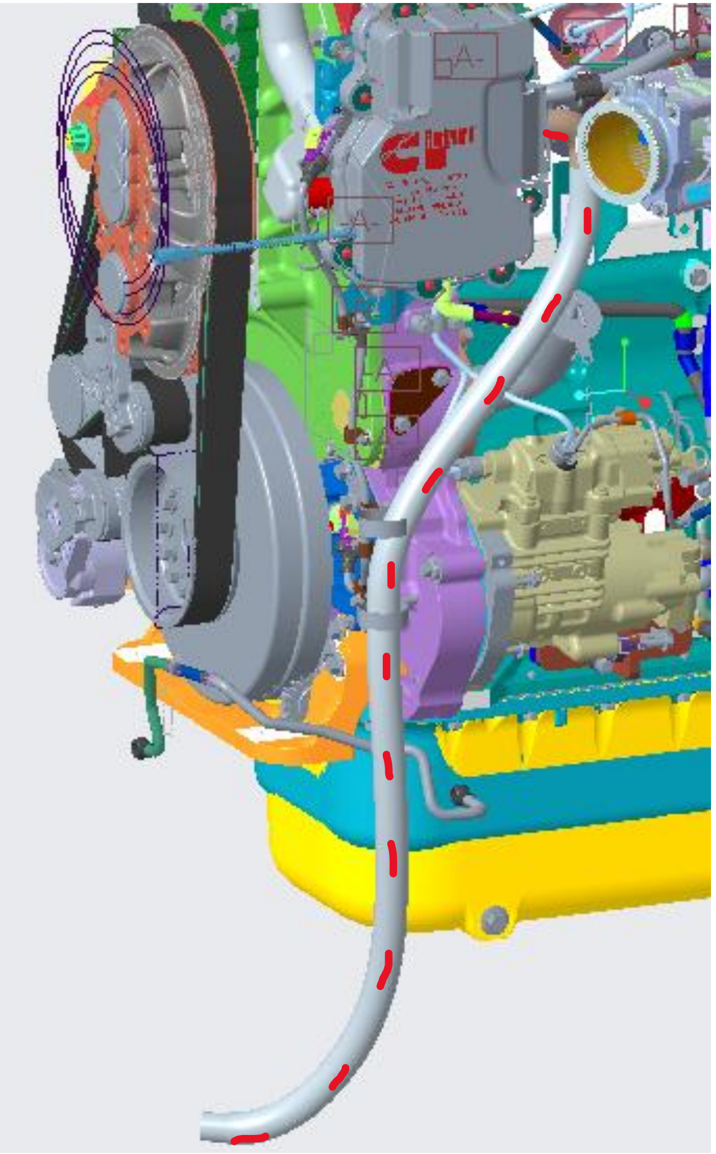
Failing Component and Effects

- Odin breather hose (5348266) is difficult to install due to size and fit
- Insulation covers full length of hose and makes contact with the shutoff valve
- May create long-term wear on insulation and shutoff valve during operation





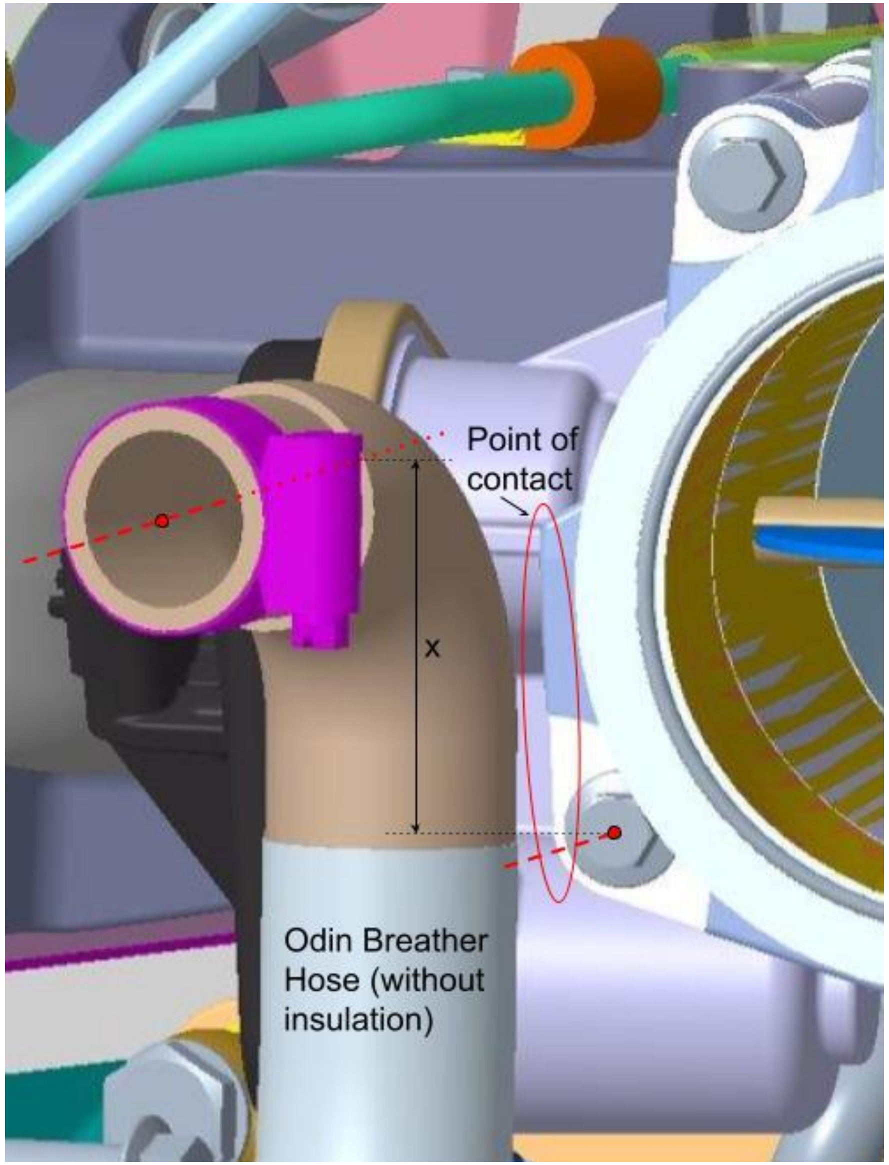




Solution and Justification

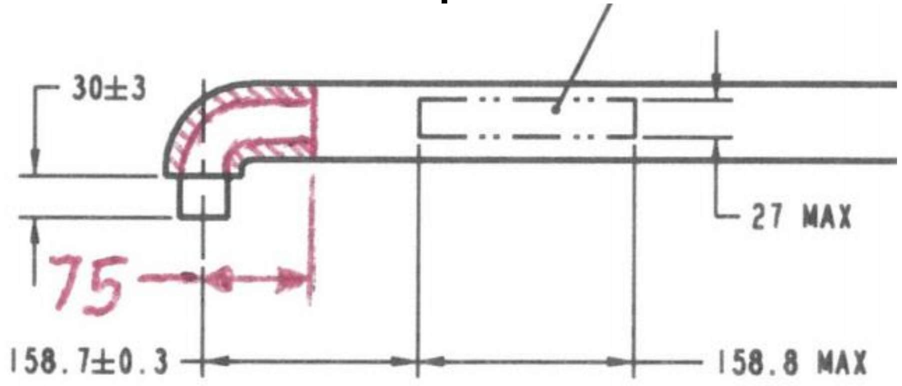
- Remove a section of insulation from top part of breather hose
  - Identify point of interference / contact and determine insulation removal needed
- Engine heat is sufficient to prevent top section of breather hose from freezing without insulation
  - Insulation at the top of the breather hose is due to supplier design
  - Automotive hose (3688597) previously used in the Odin engine has a significant top portion with no insulation and no freezing issue claims
- Current insulated Odin hose (5348266) was implemented due to better fit as a longer hose than the automotive model
  - We cannot re-implement the automotive model, must modify existing hose
- The fit of insulation to the tube has no clearance, will remain secure if any insulation is removed

Point of Contact and Insulation Removal



(Obstructive parts hidden)

- Distance from top of hose centerline to lowest possible insulation contact point x = 47.7 mm
- Insulating tube from the top of the hose must be removed so that there is no insulation above 47.7 mm from this centerline
- To allow for plenty of clearance, remove at least **75 mm** of insulation from top hose centerline



Claims Research

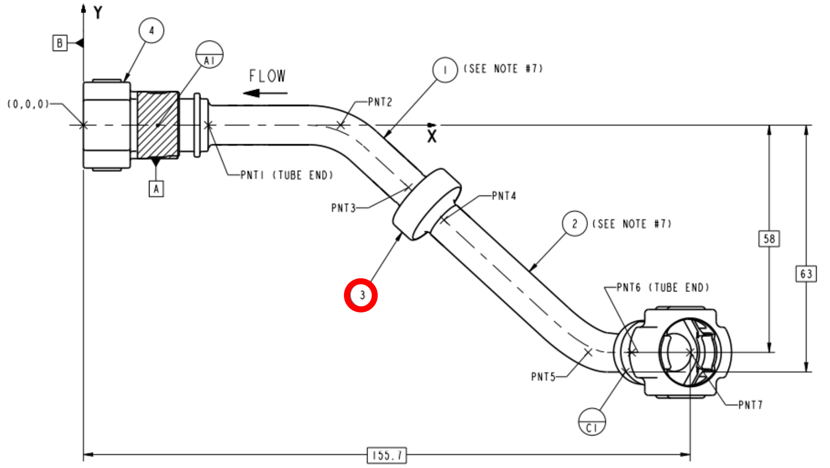
If a large portion of engines with the automotive breather hose are sent to our coldest regions and have an insignificant rate of failure due to freezing, we can conclude that the Odin breather hose (5348266) without top insulation will also not experience this failure at a significant level.

- Claims data summary →
  - Build range: 01/2018 - 07/2021
  - Regions: North America, Europe, Korea
- Sample size sufficient for accurate data
- Insignificant level of failure among engine configurations with the 3688597 breather hose
- We can reasonably assume that **the modified Odin hose will not freeze or experience failure**

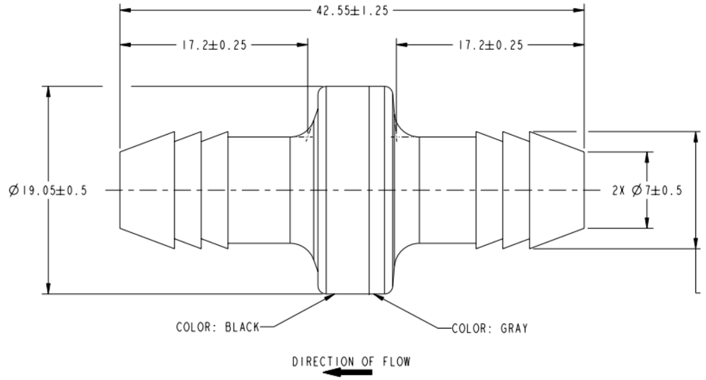
Engine Configuration	Number of Engines Sent	Number of Failures	RPH
D103021GX03	[redacted]	[redacted]	0
D103020CX03	[redacted]	[redacted]	0
D103015CX03	[redacted]	[redacted]	0.0125
D103014GX03	[redacted]	[redacted]	0
D103012BX03	[redacted]	[redacted]	0
D103010CX03	[redacted]	[redacted]	0
D103009BX03	[redacted]	[redacted]	0
Total:	[redacted]	[redacted]	0.0103
	[redacted]	[redacted]	
Total (all configs):	[redacted]	[redacted]	0.029

ISX12 Drain Tube Failed Check Valve and Material Replacement

System Background and Failure



- ISX12N system oil drain tube (4399782)
- Check valve (5272318) controls direction of flow with internal membrane
  - Controls direction of oil flow
  - Membrane inside valve opens for flow
- Oil flows from the oil pan into the turbocharger and CAC
  - Check valve fails to control oil flow in correct direction
  - Some drain tubes are assembled in the wrong direction, causing a backwards check valve
  - This failure can damage both the turbocharger and CAC
  - Oil pressure loss and oil found on other components
- Drain tube is brittle, breaks, and experiences leaks
  - Internal system temperature exceeds material specs



Claims Research – RPH and CPE Plots

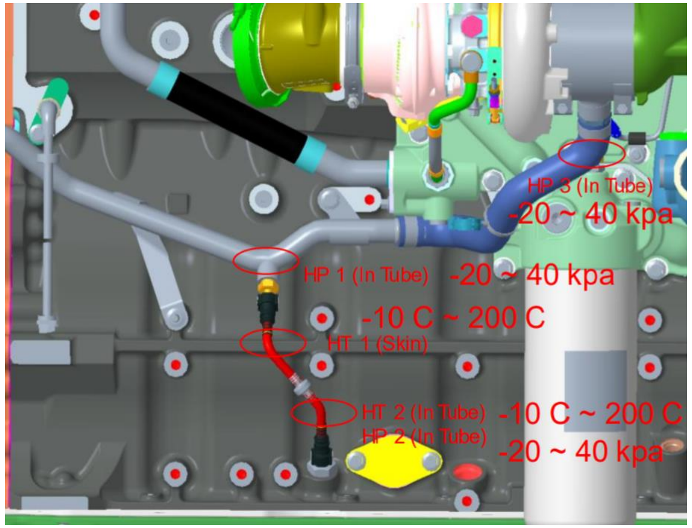
[redacted]

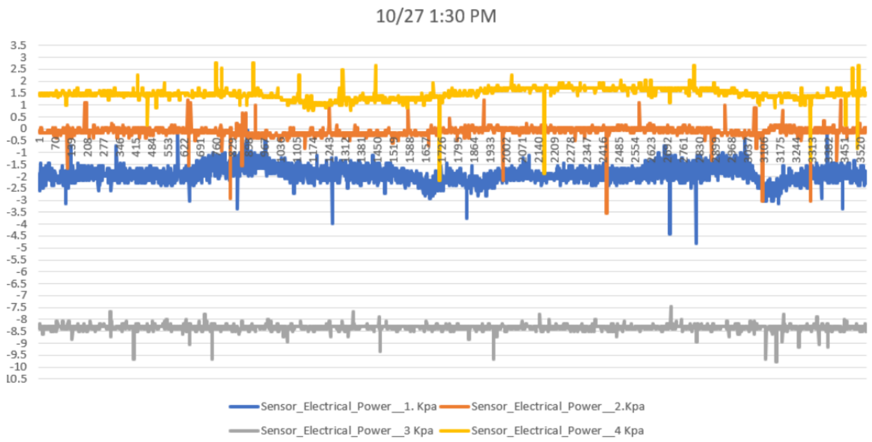
- Build volume: 12,302, Claims: [redacted], Average RPH = [redacted]
    - Failed check valve: [redacted], leaking drain tube: [redacted], installed backwards: [redacted]
- Average replacement cost = $2,163.26, Average CPE = $7.67

Proposed Solutions

- Mark drain tube connectors to indicate direction of flow
  - This will prevent drain tubes from being installed backwards
- Choose new check valve
  - Must handle negative pressure values due to turbocharger
  - Must withstand internal pressure range, check valves found broken
  - Must open to allow flow at the correct amount of pressure and close when pressure drops below this value
  - Compare specs of potential check valves with drain tube operating conditions
- Choose new tube material that can withstand greater temperature range
  - Use more heat-resistant material, must retain its ductility over time
  - Must not be prone to phase changes from long-term temperature fluctuations
  - Consider systems run at higher temperature than specs with high duty cycle

Solution Validation – Field Test





- Internal operating pressure and temperature gauges
- Peak measured operating temperature = 130 C
- Measured internal operating pressure at each gauge location
- Pressure measurement observations
  - Pressure in drain tube above check valve is negative
  - Pressure in below check valve averages 0 kPa
  - Check valve opens:
    - To allow drainage when internal pressure above check valve rises or below drops
    - When pressure within the tube above the check valve exceeds the pressure below

New Check Valve Evaluation Results

- Replacement check valve test criteria, based on drain tube conditions
  - Must open / close at same pressure difference as the current check valve
  - Must withstand a pressure range of -5 to 1.5 kPa by a reasonable margin
    - Pressure range determined by measured internal pressure during operation
  - Must not allow reverse flow, pass reverse flow test

https://cummins365-my.sharepoint.com/:w:/g/personal/ss378_cummins_com/ESrmfl7stqVCureIX9nykTABcPYmfusn3SgHqatR1KYV9w?e=YO1gno

Check Valve (Part #)	System assembly #	Assembly description	Specifications								Measured Quanitites			
			Valve diameter [mm]	Tube diameter [mm]	Length [mm]	Allowable leakage [L/min]	Min. allowable temperature [F]	Max. allowable temperature [F]	Oil compatible?	Fluid transfer rate standard	Opening pressure (delta p) [kPa]	Closing pressure (delta p) [kPa]	Flow rate at pressure of 3 kPa [L/M]	Burst pressure (delta p) [psi]
5272318	4399782	Water drain tube	19.05	10.75	42.55	0.2	-40	250		> 35 mL/min	>0	< -0.1433	17.67	49.67
5442592	5402614	Air transfer tube	30	7.4	57.7	0.2	-40	250	Yes	0.5 to 1 CFM	>0	< -0.2367	1.36	>90
5588864	5567571	Oil drain tube	20.2	9.6	44									
5589463	5588854	Fuel drain tube	11	5	36.2	0.045		266	Yes	> 33 L/h				
5592641	5590792	Oil drain tube	19.1	9.75	42.5	0.2	-40	250	Yes	> 35 mL/min	>0	< -13.8		
4998310	4325210	Oil drain tube	19.05	8.8	43.4		-40	257			>0	< -0.1867	53.5	51.33

- None of these check vales meet criteria, repeat test for other check valves

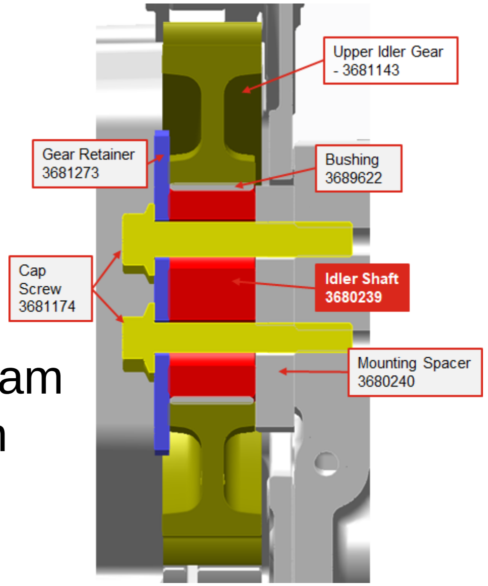
New Tube Material Evaluation Results

Material	Applicable Standards	Ductile material?	Formable hose?	Max. operating temperature [C]	Min. operating temperature [C]	Oil and water compatible?	Piece price with material
ASTM PA12	ASTM F2785, MPAPS F-7134	Yes	Yes	82	-40	Yes	[redacted]
ASTM PA1010	2017-01-0490, MPAPS F-7134	Yes	Yes	N/A	N/A	Yes	
Flexible reinforced silicone hose	25043	Yes	Yes	219	-54	Yes, ASTM D6210	High cost, premium material
Fluorosilicone liner		Yes		219	-54	Yes, ASTM D6210	High cost, premium material
HNBR - GH100	From Eaton, N/A	Yes	No	150	-40	Yes	
HNBR - GH101	From Eaton, N/A	Yes	No	150	-40	Yes	
Silicone	Not available	Yes	Yes				
Peroxide-cured FKM liner		Yes					
Reinforced silicone	25055	Yes	Yes	150		Yes	High cost, premium material
Compounded fluorocarbon liner		Yes		150		No	High cost, premium material
Silicone rubber (VMQ)	Silicone rubber: 23274, Fluorosilicone liner: 25043	Yes	Yes	180	-54	Yes	
Fluorosilicone liner		Yes		219	-54	Yes, ASTM D6210	

- Evaluated properties of alternate materials to find a replacement material
  - Current material: ASTM PA12
- Must have a maximum operating temperature above 130 C
- Chosen material: **Silicone rubber (VMQ) with fluorosilicone liner**

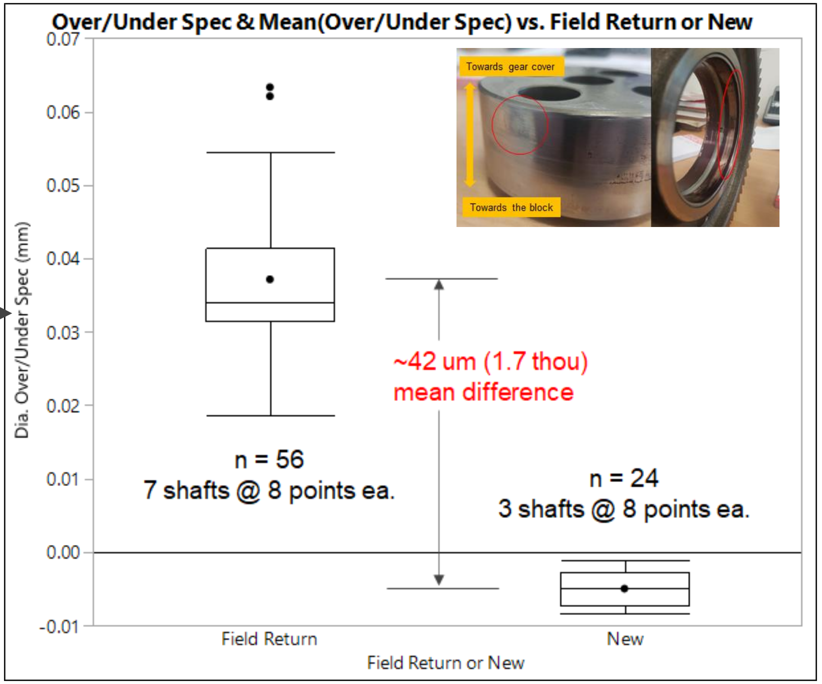
UIG Idler Shaft Expansion and Dual-Cam Service

Dual-Cam UIG Idler Shaft Failure



Dual-Cam System

- Retained austenite within dual-cam UIG idler shaft transitions into martensite over time with preload applied to bolts
  - Martensite formation causes diameter increase
  - Clearance between idler shaft and bushing narrows
  - Hydrostatic oil film narrows and eventually closes
  - Lack of oil film causes system failure
- Field return vs new part diameter measurements
- Heat treatment procedure altered for single-cam system to decrease retained austenite
  - Original = 350 F – 90 mins, New = 400 F – 90 mins
  - CTR 4120261 – 19% radial growth decrease



RPH and CPE Plots – Dual Cam System

[redacted]

- Build volume: 377,503, Claims: [redacted], Average RPH = [redacted]
  - BKIS: [redacted], BKIG: [redacted], BKIP: [redacted], BKIB: [redacted], BKIQ: [redacted], BKMI: [redacted]
- Average replacement cost = $8,275.55, Average CPE = $1.95

RPH and CPE Plots – Single Cam System

[redacted]

- Build volume: 764,311, Claims: [redacted], Average RPH = [redacted]
  - BKIS: [redacted], BKIG: [redacted], BKIP: [redacted], BKIB: [redacted], BKIQ: [redacted], BKMI: [redacted]
- Average replacement cost = $13,675.51, Average CPE = $2.49

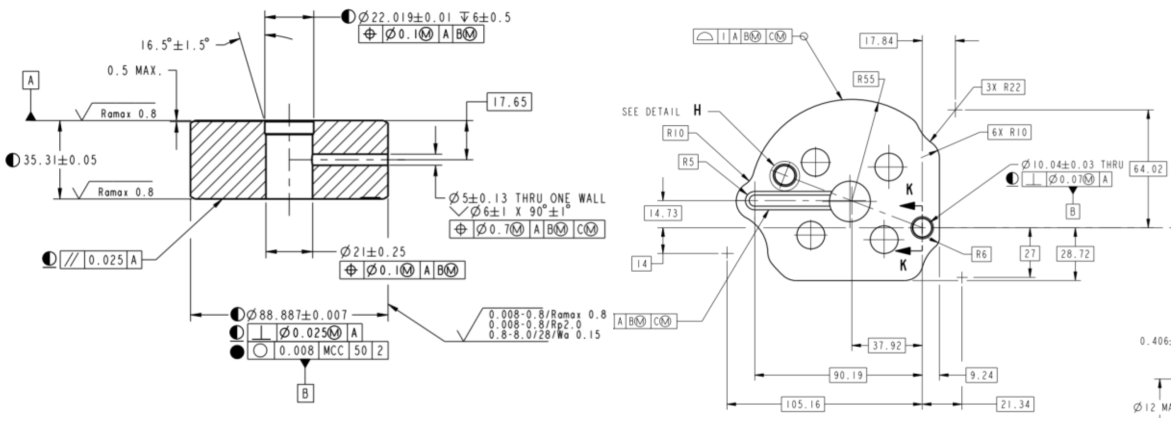
Dual-Cam System Claims Research Results

- Based on single-cam results, a dual-cam solution would likely yield a lower RPH in dual-cam systems, however:
  - The single-cam idler shaft heat treatment change was implemented 4/12/2021
  - There is not enough claims data past this date to evaluate fix effectiveness and compare to current dual-cam claims
- We cannot yet see the effect the single-cam solution has on its failure rate
  - However, we can reasonably assume that the reduction in retained austenite from heat treatment changes reduces martensite growth and reduces diameter growth and failure due to operation
- A similar solution in the dual-cam system for service should reduce failure
- Any dual-cam system changes would be for service only, production ended

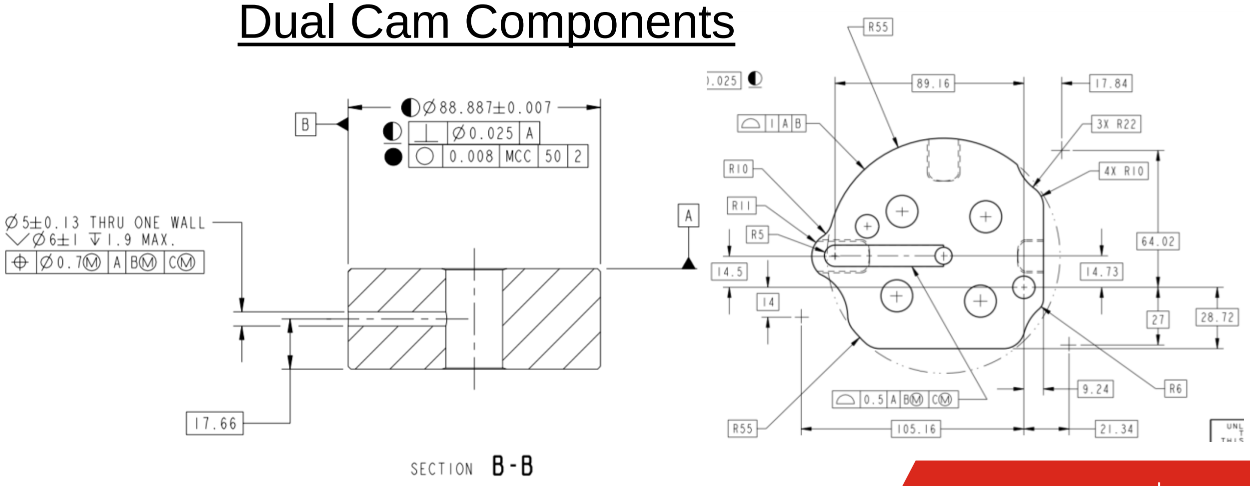
Proposed Solution and Justification

- **Replace dual-cam idler shaft** with the single-cam system UIG idler shaft
- Will prevent the dual-cam UIG idler shaft from generating martensite
- Apply change as a service part replacement, not production
  - Commonized UIG idler shaft for both systems in service
- Geometry and compatibility of both idler shafts
  - No need to redesign single-cam idler shaft or dual-cam system components

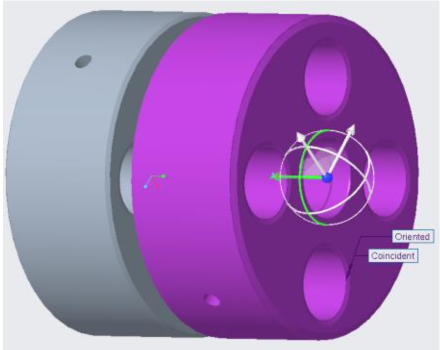
Single Cam Components

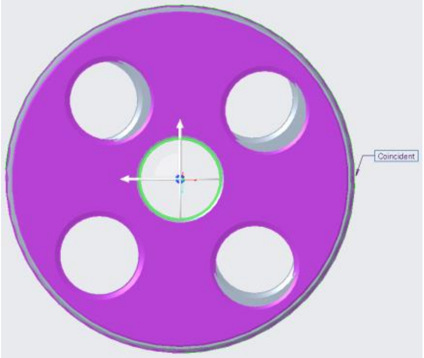


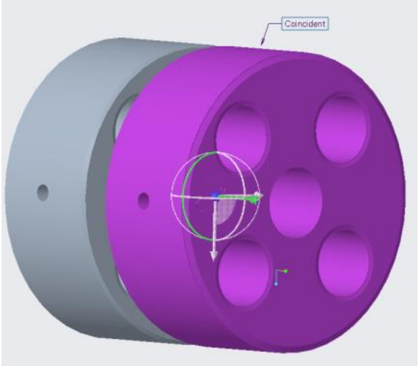
Dual Cam Components

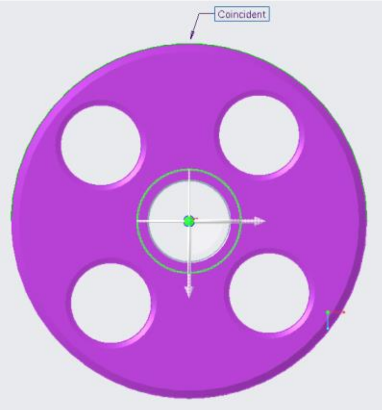


Dual-Cam System Adaptation

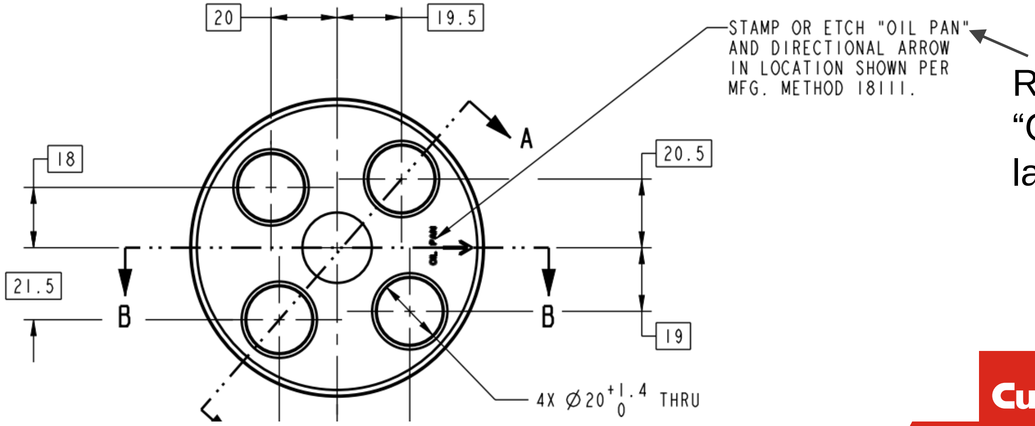








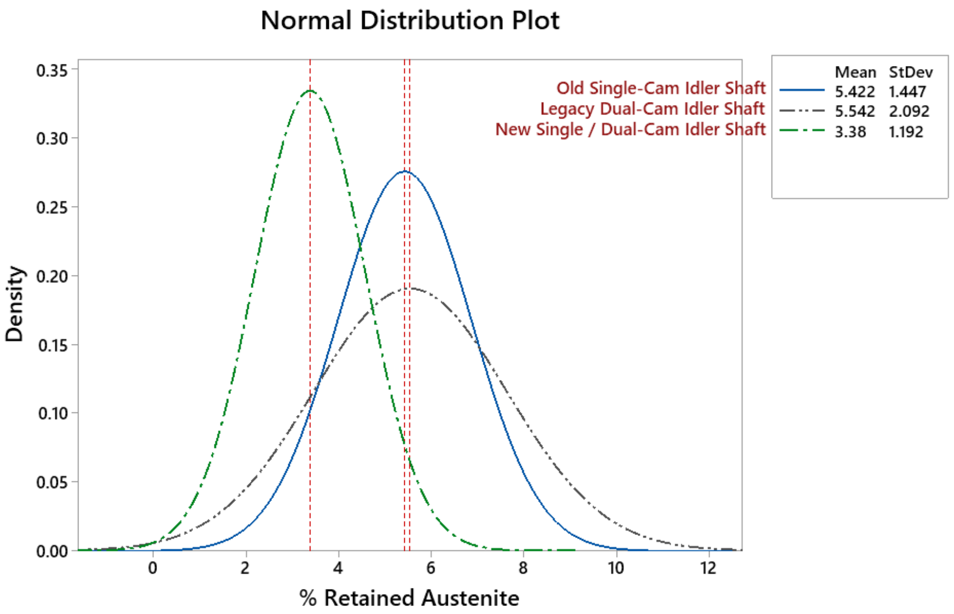
- Single-cam idler shaft can replace dual-cam shaft
  - No idler shaft design differences significant to fit
  - Compatible with dual-cam mounting spacer
  - Rotate single-cam idler shaft 90 degrees clockwise
- System differences and supply considerations
  - Dual-cam idler shaft is not rigidly connected
  - Ensure single-cam idler shaft is not fitted with a dowel ring for this application
  - Have idler shaft supplied for service as individual part rather than assembly
- Service changes
  - Labeling change for compatibility
  - Make idler shaft saleable
  - TSB submission, manual edits



Remove "OIL PAN" label

Solution Effective Rate

- Old single-cam idler shaft avg RA = 5.422% Std dev = 1.447
- Legacy dual-cam idler shaft avg RA = 5.542% Std dev = 2.092
- **New single-cam idler shaft** avg RA = 3.380% Std dev = 1.192
  - 37.7% reduction in retained austenite from old single-cam idler shaft
  - 39.0% reduction in retained austenite from current dual-cam idler shaft



- Since these reduction rates are approximately the same, we can assume about the same **19% reduction in radial growth** from the dual-cam idler shaft
- Radial growth reduction should correlate to reduction in failures

Claims Cost Savings

- Approximated current single-cam savings:
  - RPH = [redacted], CPE = [redacted]
  - Approximate annual volume = 54,594
  - Annual cost due to claims = $135,939.06
  - With a 19% reduction in failure, annual savings = $25,828.42
- Projected dual-cam savings after change:
  - RPH = [redacted], CPE = $1.95, average service cost = $8,275.55
  - Production will cease, so savings will be from prevented repeat services
  - Total repeat services = 46, annual repeat services = 3.3
  - Approximate annual cost of repeat services = $27,309.31
  - With a 19% reduction in failure, annual savings = **$5,188.77**
- Total annual savings = **$31,017.19**

Lessons Learned and Next Steps

Lessons Learned

- Engineering investigations require a lot of time, thought, and validation work
  - The proportion of time spent on the design (or redesign) phase is small
- Project support systems are incredibly valuable as they delegate to specialists, divide workload, and ultimately make projects more efficient
- Many layers of approval needed to implement a change in production
- College can't teach you everything as industry always evolves
  - Learning new technical skills is often valuable to completing projects, ability to learn is critical to an engineering role
- Critical skills developed:
  - Concision in presenting info or results
  - Clear communication
  - Time management, project management and organization

My Next Steps

- Two semesters remaining in college
- Undergraduate research in hydroelectric power turbine blade testing
  - Flow simulation
  - Stress and motion analysis
  - Manufacturing methods and feasibility
- Leadership certificate
- Summer internship
  - Potentially with Cummins, working with natural gas / B-platform engines in Columbus
- Long-term position after graduation
  - Sustainable energy field
  - Hydrogen fuel cells at Cummins


Cummins