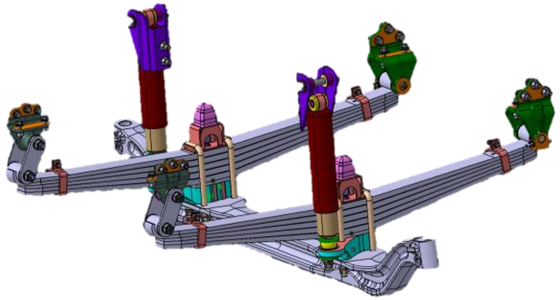
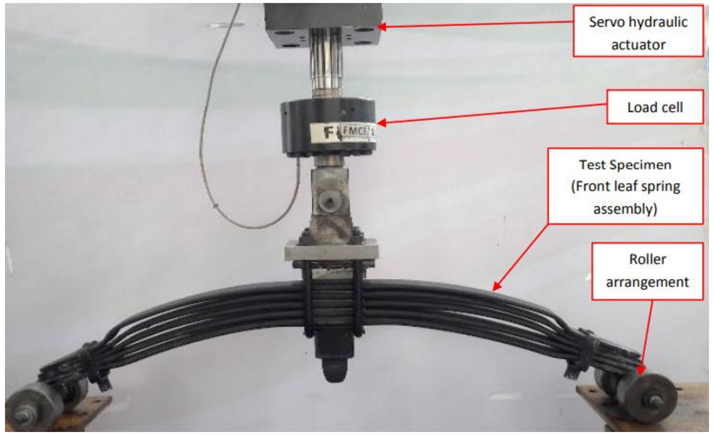
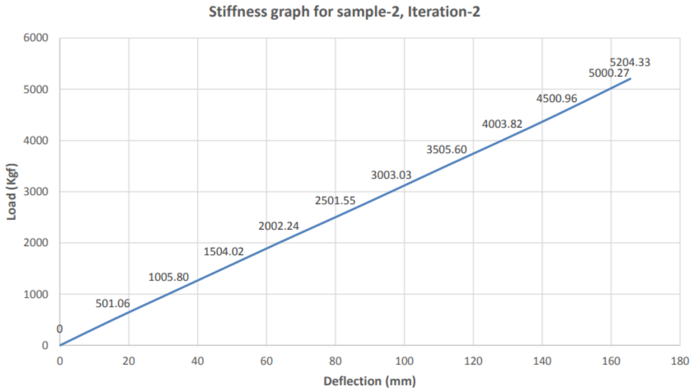
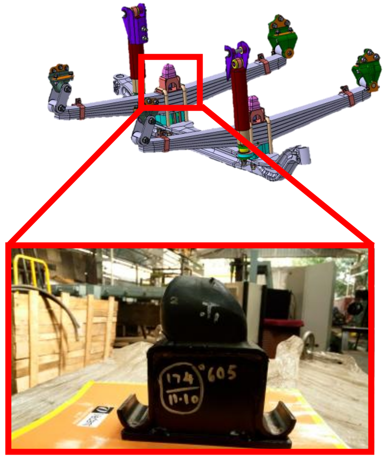
SID’s TECHNICAL PORTFOLIO

Validation based design improvement

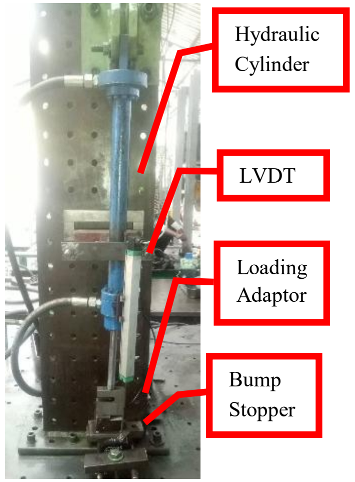








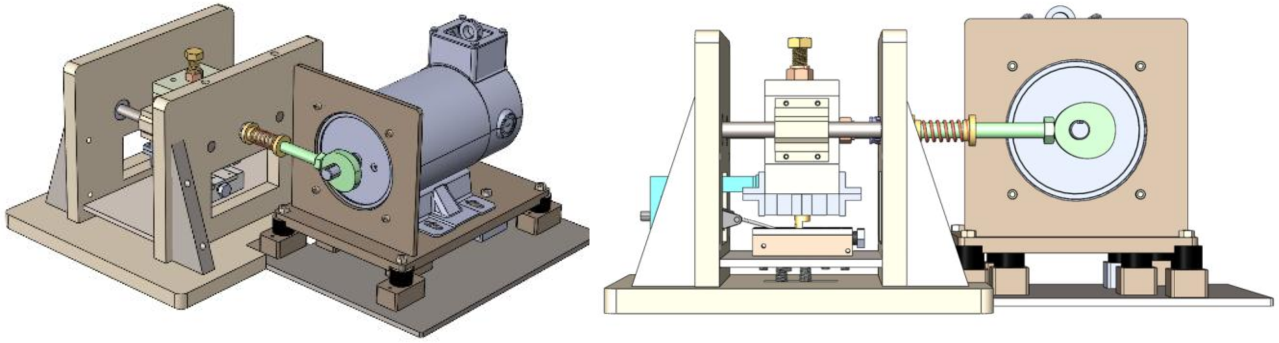
Suspension system design & prototype made based on theoretical vehicle dynamic calculations



SAE based Validation test set-up design for individual component

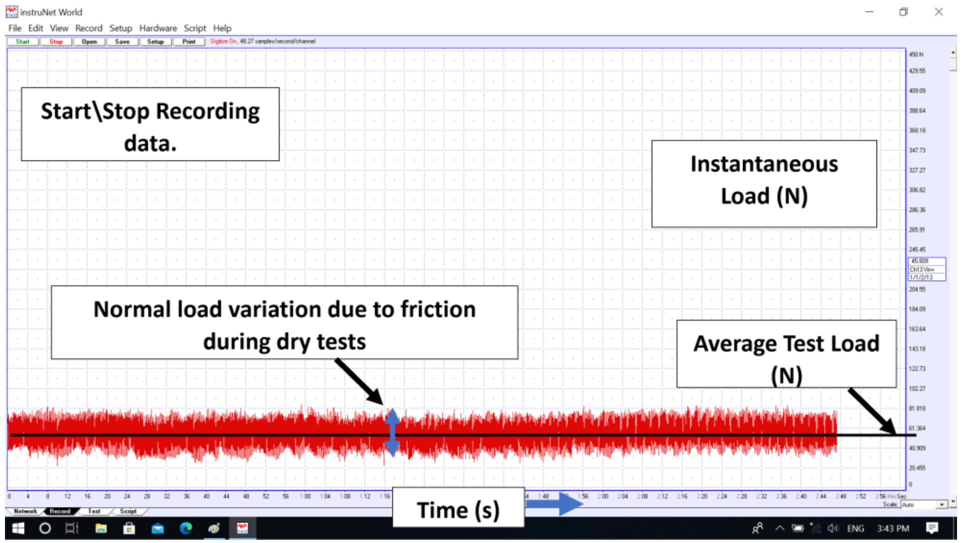
Validation tests results utilized to fine tune ride and handling benchmarks

Design and Development of Tribometer

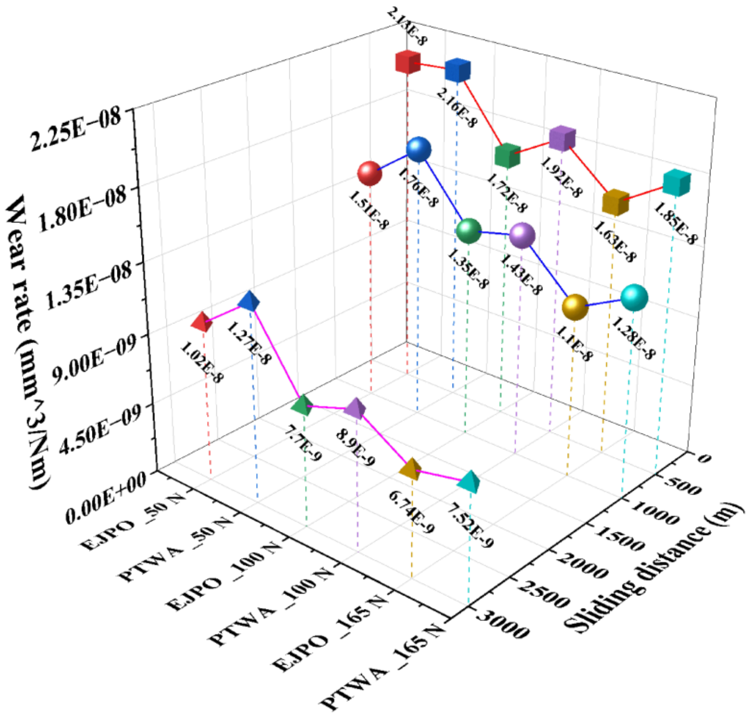


Design of the wear testing device on SolidWorks

Functional Prototype with dual sensor data acquisition



Load cell transducer reading



Results of durability test using my prototype device

Design + Fab of hyd. inching drive to resolve material failure

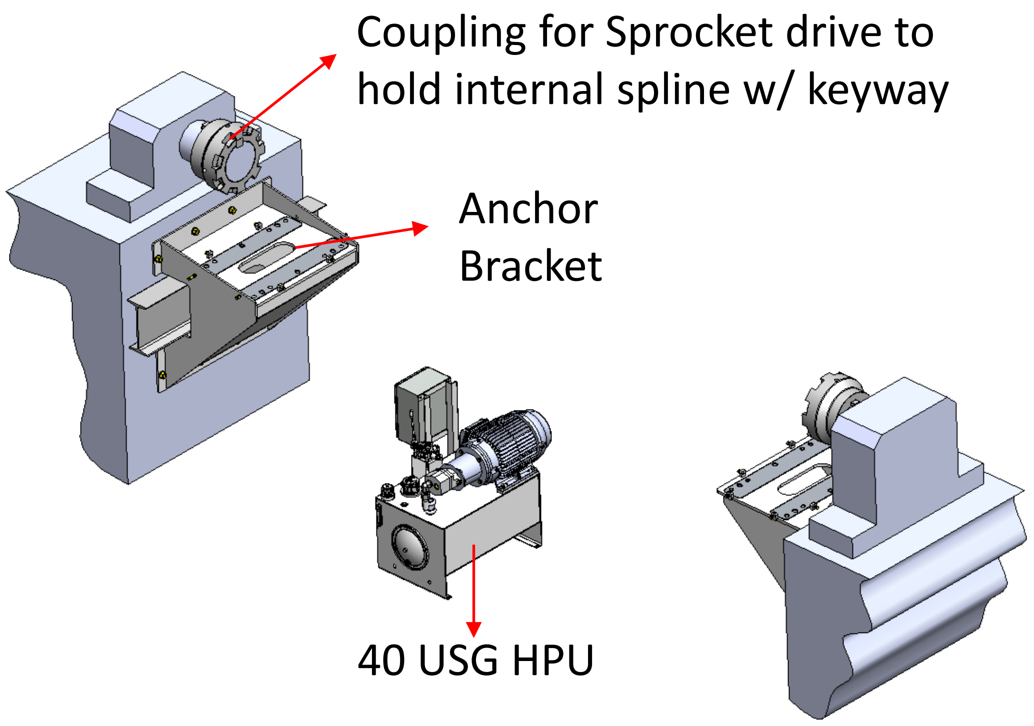


Figure: 1; Inching drive mechanism for controlled circumferential welding

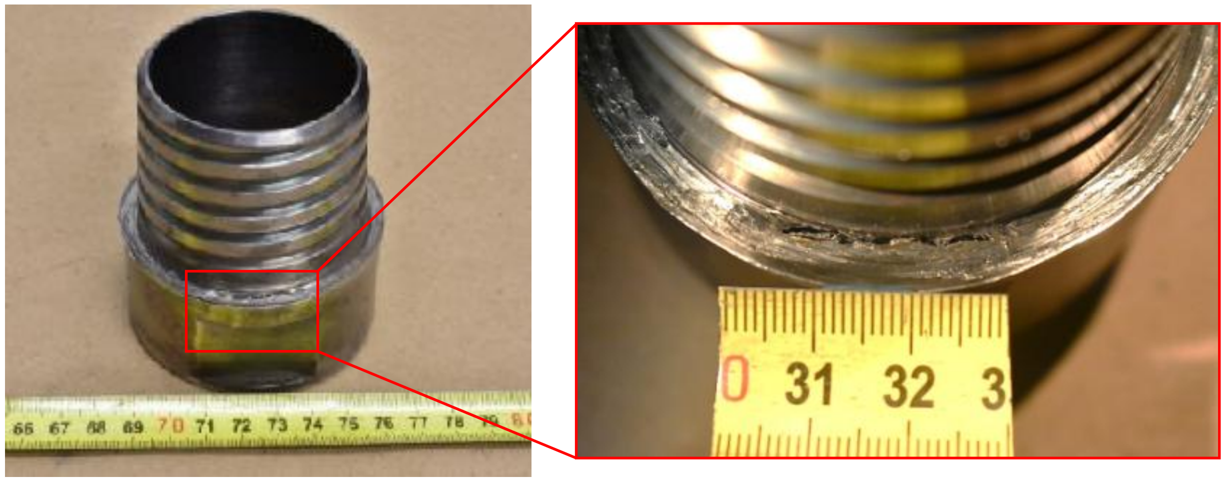


Figure: 2 ; fractured drill rod pin ends during torsional load.

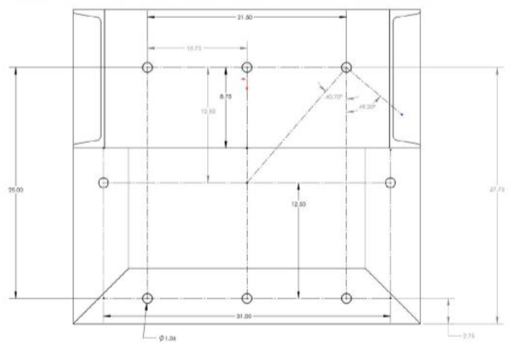
- **Objective:** Make modular welding unit under the guidance of Engineering & Manufacturing lead to resolve the weld failure issue observed on torsional load test post pin weld on drill rod.

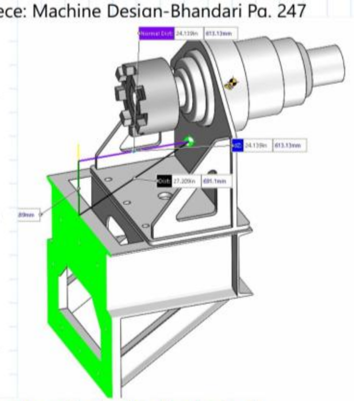
Challenge: Design an in-house drive system that can be controlled by the welder without affecting the weld pass precision and accommodating multiple rod lengths.

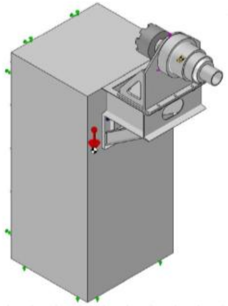
Bolt Stress Calc & Design of Inching Drive for M/C Shop

WO# 50281-Bolted bracket calculation JUL-27-2025

Referece: Machine Desian-Bhandari Pa. 247







The bolts can be designed on the basis of principal stress theory or principal shear stress theory.

The principal stress σ_1 is given by,

$$\sigma_1 = \frac{\sigma_t}{2} + \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2} \qquad (7.13)$$

The principal shear stress is given by,

$$\tau_{max.} = \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2} \qquad (7.14)$$

INPUTS

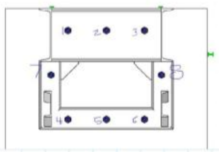
Note: bolts 7 & 8 not considered

$V := 2800 \cdot lb$ Weight including base, reducer, reducer bracket and male coupling

$P := V$

$T := 1052688 \cdot in \cdot lb$ Maximum output torque

$B_{TSA} := .472\ in^2$ Bolt tensile stress area



$N_B := 6$ Number of bolts

$Y_s := 60000 \cdot \frac{lb}{in^2}$ Anchor bolt Yield Strength, considering A354-Grade BD

CALCULATION

Primary Shear Forces, P'=P'1, P'2, P'3, P'4, P'5, P'6,

$P' := \frac{V}{N_B} = 466.67\ lb$

$P'_3 := P'$

Obtaining distances from center of bolt pattern to center of bolts r1=r4=r3=r6 and r2=r5

$r_1 := \sqrt[2]{(10.75 \cdot in)^2 + (12.5 \cdot in)^2} = 16.49\ in$

$r_2 := 12.5 \cdot in$ $\quad r_7 := 15.5 \cdot in$

From fig.1 we can deduct that bolts 3, 6, 1 and 4 would be under max. shear forces

$P''_3 := \frac{T \cdot r_1}{4 \cdot (r_1{}^2) + 2 \cdot (r_2{}^2) + 2 \cdot (r_7{}^2)} = 9230.363\ lb$

Resultant shear force on bolt 3 (Ps)

$\Sigma Fy := P'_3 + ((P''_3) \cdot \cos(49 \cdot deg)) = 6522.33\ lb$

$\Sigma Fx := P''_3 \cdot \sin(49 \cdot deg) = 6966.24\ lb$

$P_s := \sqrt[2]{(\Sigma Fy)^2 + (\Sigma Fx)^2} = 9543.03\ lb$

Hydraulic motor w/ CB valve for metered rotation

STRESS DUE TO TENSION CALCULATION

The moment V x e tends to tilt the bracket about the lower edge, this way the top bolt row is taking maximum forces

$e := 24.25 \cdot in$ base, reducer, reducer bracket and male coupling COG to mounting face

$l_1 := 27.75 \cdot in$ Lower edge to upper bolt row distance

$l_2 := 15.25 \cdot in$ Lower edge to lower bolt row distance

$l_3 := 2.75 \cdot in$

$P_3 := \frac{P \cdot e \cdot l_3}{3 \cdot (l_1{}^2 + l_3{}^2) + 2 \cdot (l_2)^2} = 66.74\ lb$ Resisting forces in bolts 1 to 3

$\sigma_t := \frac{P_3}{B_{TSA}} = 141.39\ \frac{lb}{in^2}$ Stress in bolt due to tension

$\tau := \frac{P_s}{B_{TSA}} = 20218.27\ \frac{lb}{in^2}$ Stress in bolt due to shear

$\tau_{max} := \sqrt[2]{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2} = 20218.4\ \frac{lb}{in^2}$ Maximum shear stress in bolts

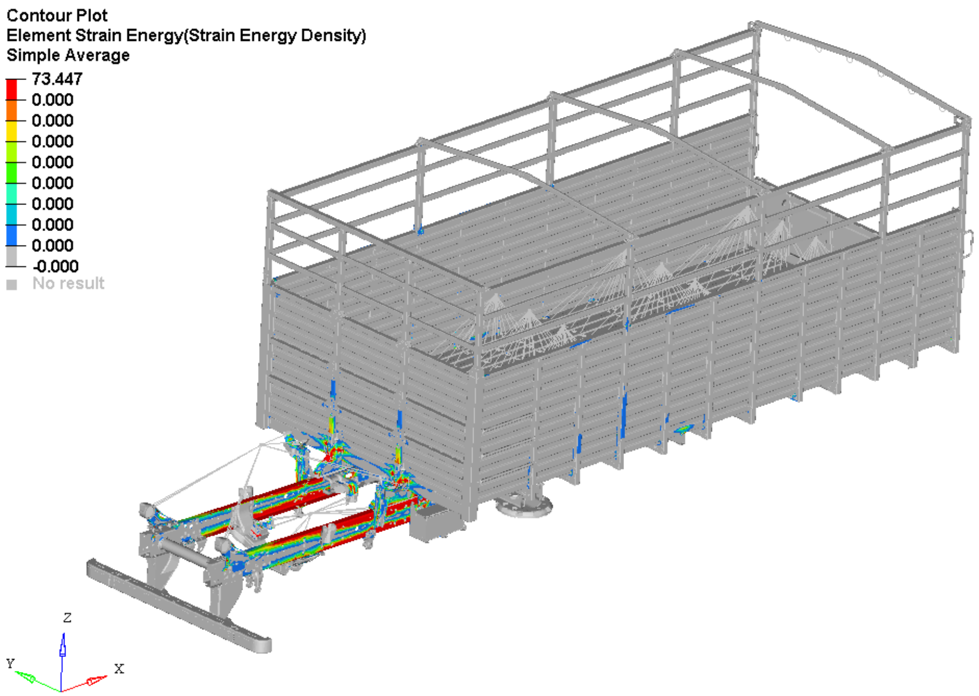
Load factor top row bolts

$L_f := \frac{Y_s}{\tau_{max}} = 2.97$

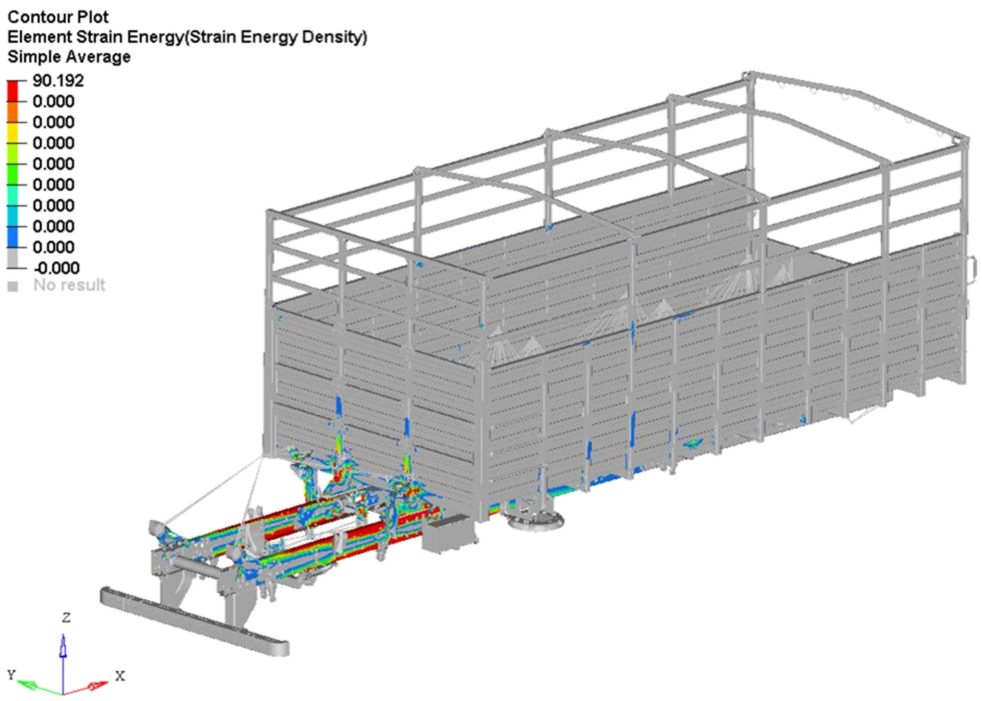
My contribution: Designing the structural components, outsource pump for hydraulic control based on Q-H curves, design hose paths and analyse shear stress.

Next Steps: Prototype validation

Stress-strain energy plot



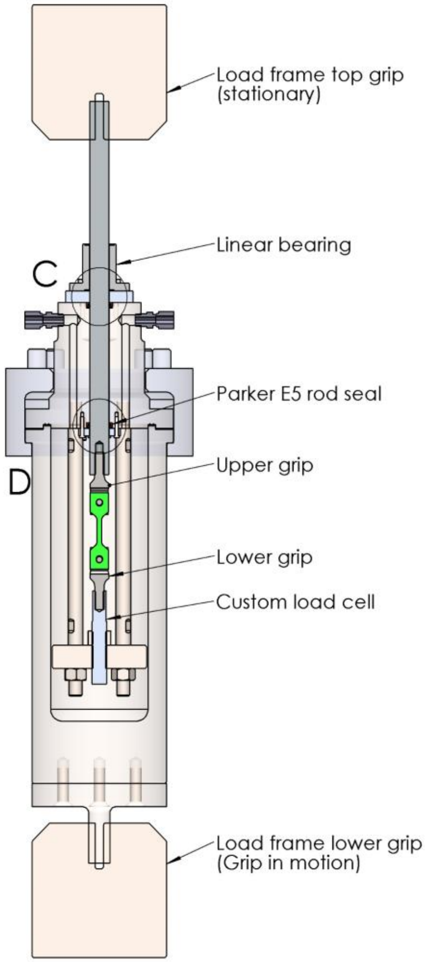
Cabin rear mount bounce: 5.6 Hz



Cabin rear mount bounce: 5.7 Hz

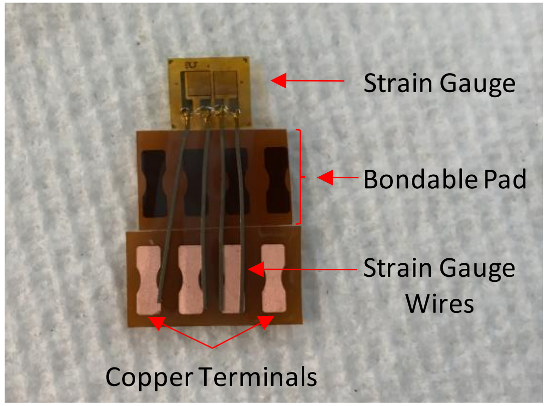
- Cabin rear suspension natural frequency plot based on ride and handling design validation trials for one of the new vehicle development projects.
- The stiffness of the rear cabin suspension was decided based on the study.

Instrumentation of HP autoclave



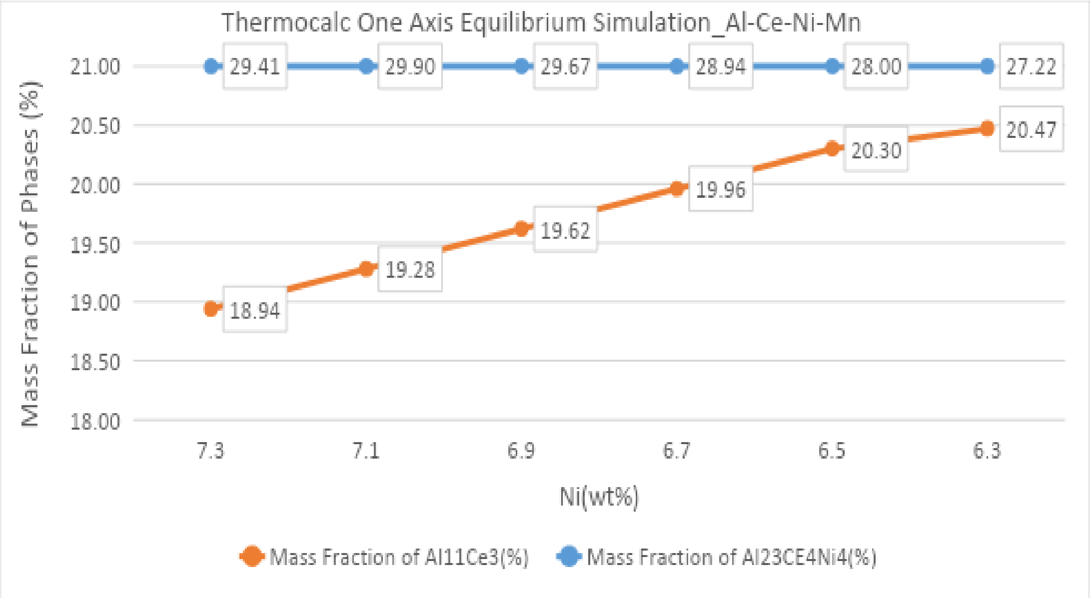
Sealed Autoclave design for HP Hydrogen embrittlement study

Autoclave installed on the MTS UTM

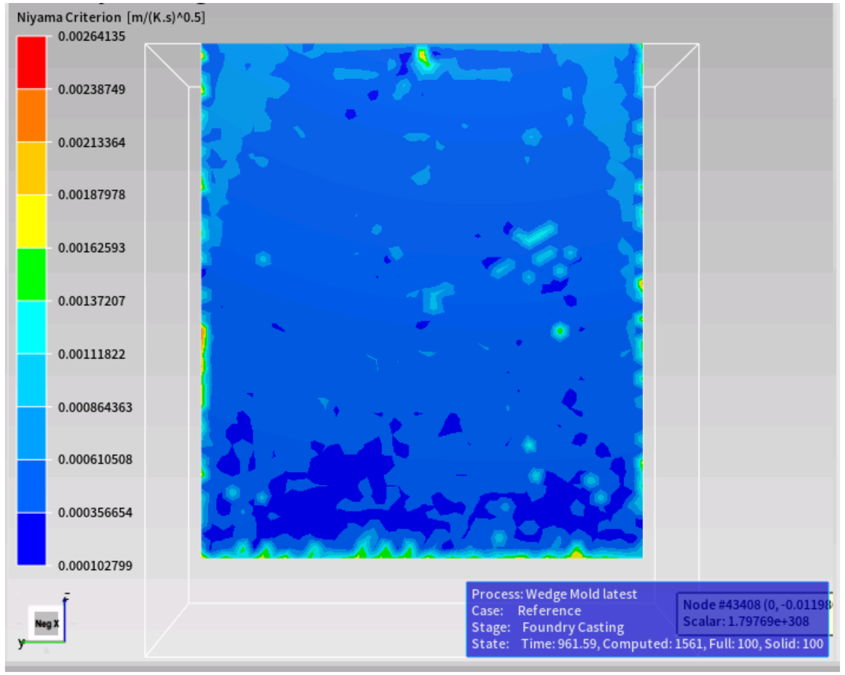


Strain gauge assemblies to detect deformation of the during tensile test specimen

Improving casting parameters using CFD



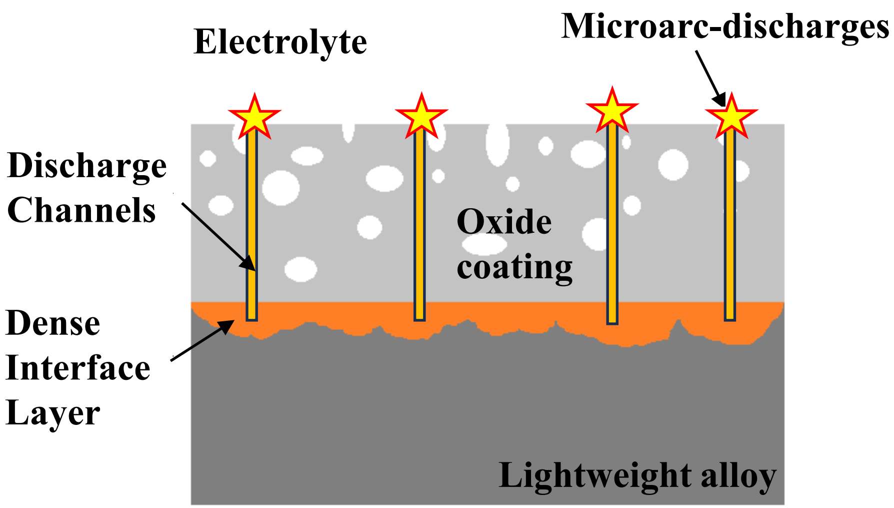
Phases post equilibrium solidification



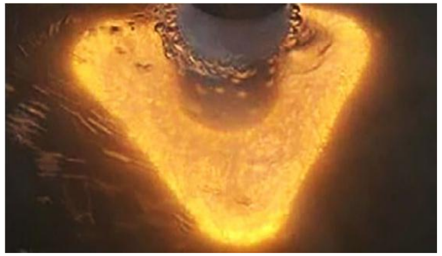
Niyama shrinkage criterion

- $Al_{23}Ce_4Ni_4$ => Lower TCE
- Casting simulation showed low shrinkage in a V mold.
- Casting done with the composition of Ni = 7.3 (wt%) in Al-Ce alloys.
- Mechanical property test showed promising results.

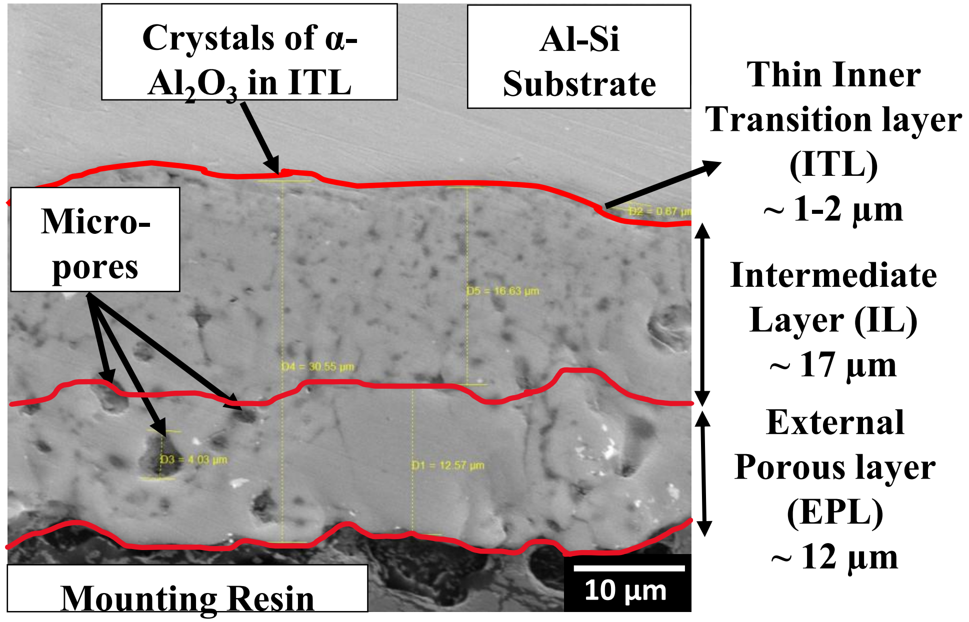
Fine-tuning of the EJPO coating process

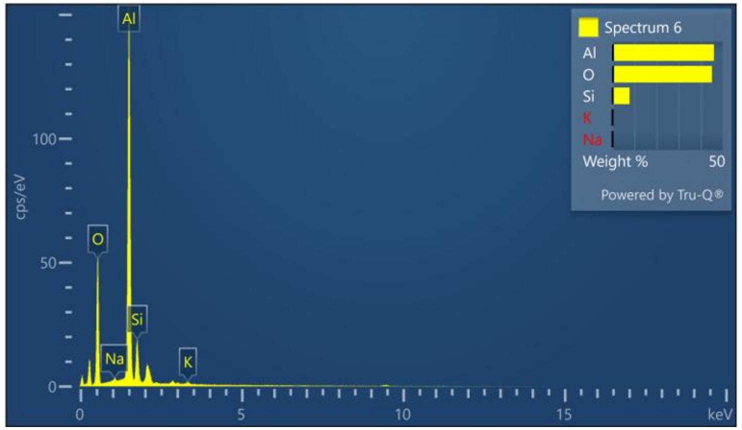


Schematic diagram of the coating process



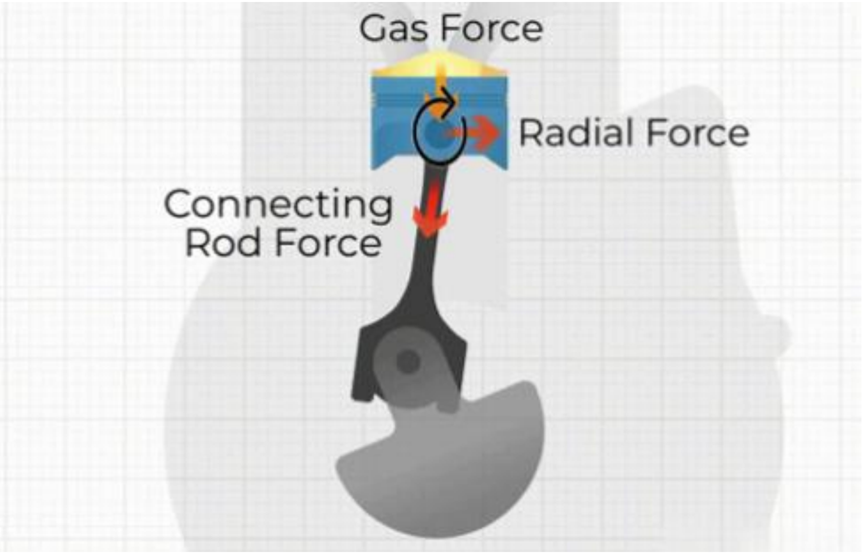
Plasma discharge around component immersed in chemically benign low-concentration electrolyte



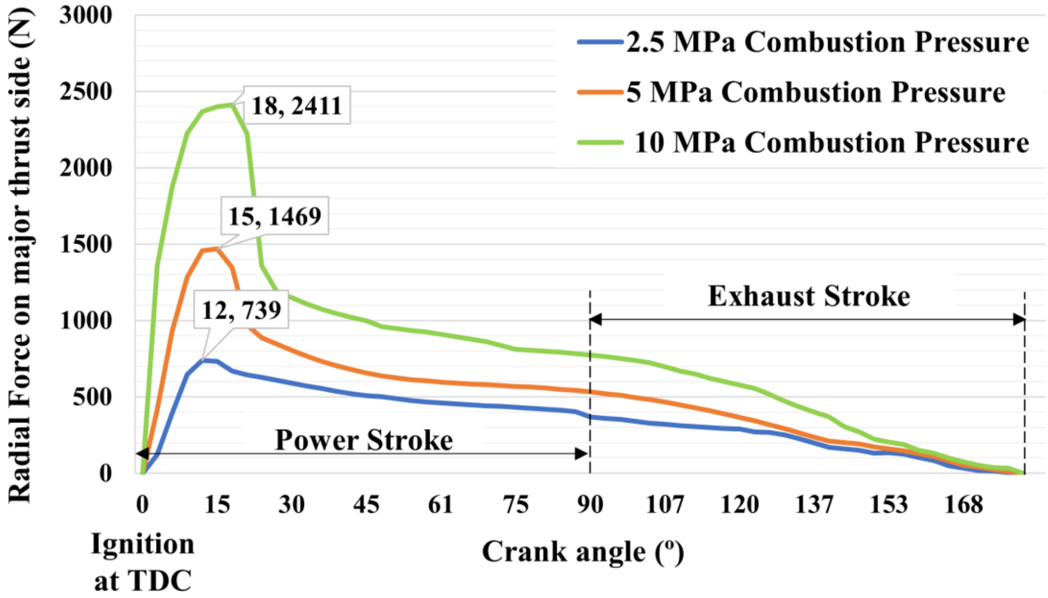


Side view of EJPO coating microstructure & elemental composition

Radial forces experienced by cylinder liner under varying combustion pressures (MATLAB coding)

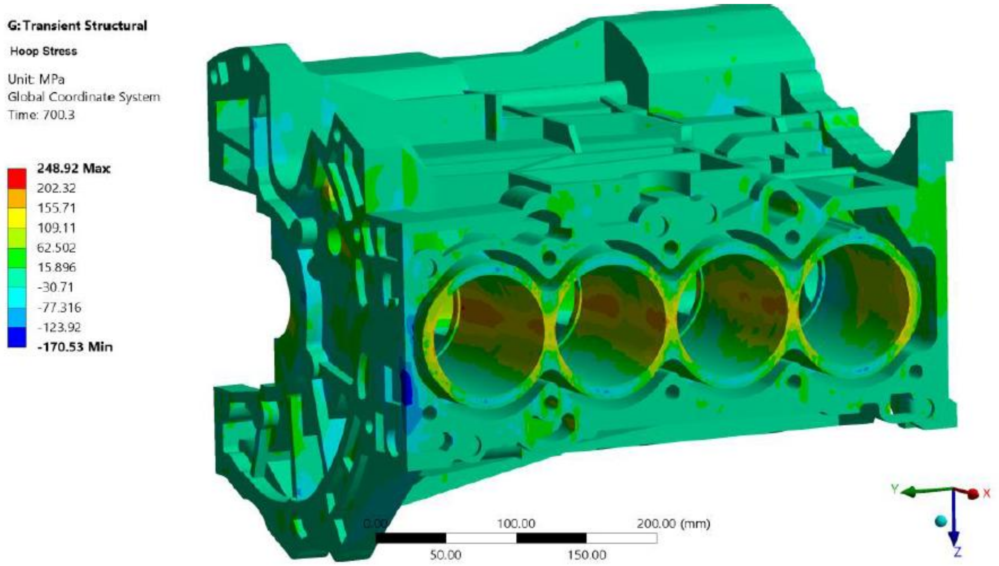


Kinematics of radial force

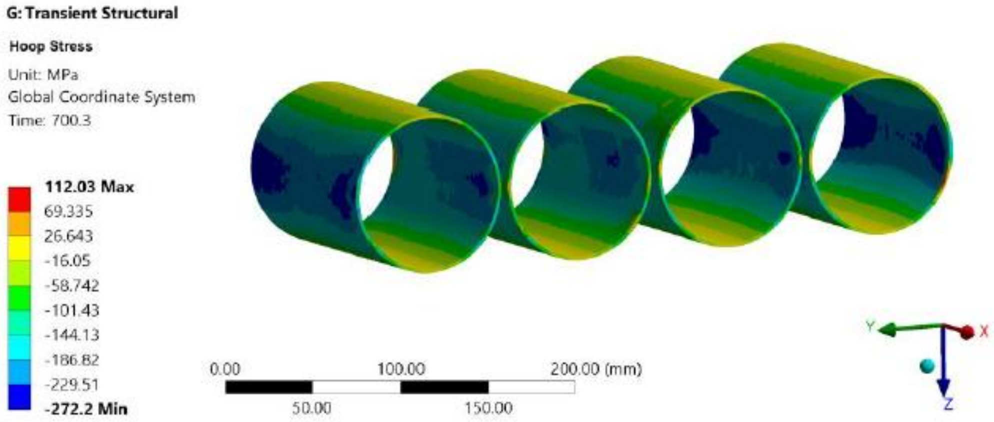


Resultant plot showing radial forces at varying crank rotation for the specified engine geometry

Cylinder Liner Stress Study



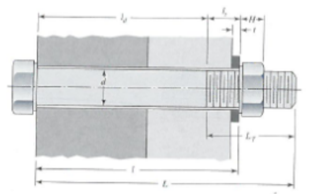
Tensile residual stress in Inline Engine Block



Compressive residual stress in the CI liner of same block

- ANSYS thermal stress simulation on an as-cast Al-Si engine block.
- Maximum hoop stress observed along the adjoining bridge surface between cylinders due to compressive residual stress in CI liner of the same block .
- Residual stress develops in an aluminum engine block equipped with a cast iron liner due to the differential thermal expansion/contraction coefficients of the materials.

Calculator for static + fatigue load cases



Bolt Calculator - Tensile loading .XLSX

Tensile Load Calculator -Static Loading Only

(All dimensions in inches unless noted)

*Valid only for the type of connection shown in the figure ->

INPUTS	
INPUT PARAMETERS	**ENTER VALUES**
Bolt Diameter (d)	0.375
Number of Bolts (N)	8
Total Load (P_total) [lbs]	30,000
Material 1 Youngs Modulus (E) [psi]	3.00E+07
Material 1 thickness (t1)	0.625
Material 2 Youngs Modulus (E) [psi]	3.00E+07
Material 2 thickness (t2)	0.375
Thread type	UNC
Fastener length (L)	1.5
Torque Condition	Plain (Dry)
Bolt Grade	8
Type of Joint	nonpermanent, reused fastener:

* Only Enter values that are highlighted

Young's Modulus

	E [Mpsi]
Steel	3.00E+07
Aluminum	1.03E+07
Cast Iron	1.45E+07

*MUST BE L > l + H = 1.411

('L' may need to be bigger for 1.5" bolts)

$$F_i = \begin{cases} 0.75F_p & \text{for nonpermanent connections, reused fasteners} \\ 0.90F_p & \text{for permanent connections} \end{cases}$$

F_p is the proof load, obtained from the equation

$$F_p = A_t S_p$$

OUTPUTS	
OUTPUT PARAMETERS	**OUTPUT VALUES**
Bolt Preload Force (Fi) [lbs]	6,975
Bolt Torque (T) [in.lbs]	523
Bolt Torque (T) [ft.lbs]	44
External Load per bolt (P) [lbs]	3,750
Stiffness Constant of the Joint (C)	0.189
Portion of External Load Taken by Bolt (Pb) [lbs]	709
Bolt Stress (σ_b) [psi]	99,151
Proof Strength (Sp) [psi]	120,000
Safety Factor (np) - Yielding	**1.21**
Load Factor (nL) - overloading	3.28
Safety Factor (n0) - Joint Separation	**2.29**

$$P_b = \frac{k_b P}{k_b + k_m} = C P$$

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t}$$

$$n_p = \frac{S_p}{\sigma_b} = \frac{S_p}{(CP + F_i)/A_t}$$

$$n_L = \frac{S_p A_t - F_i}{CP}$$

$$n_0 = \frac{F_i}{P(1 - C)}$$

CALCULATION DETAILS - DO NOT EDIT

Torque Coefficient	0.2
Nut Size (H)	0.328

Bolt with nut - Static Load | Bolt with nut - Fatigue Load | Bolt without nut - Static Load | Bolt without nut - Fatigue Load

- **Objective:** Make it easier for design engineers to draft assemblies with a accurate bolt pre-load and torque values for various hardware configurations.