

SID's TECHNICAL PORTFOLIO

Page 2: Product durability test design (NDT/DT)

Page 3: Design and fabrication of Electro-mechanical wear testing device

Page 4: Design & analysis of Inching drive system to resolve material failure issue

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Page 8: Improving casting using CFD

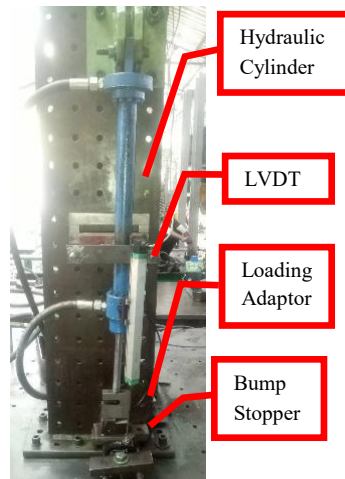
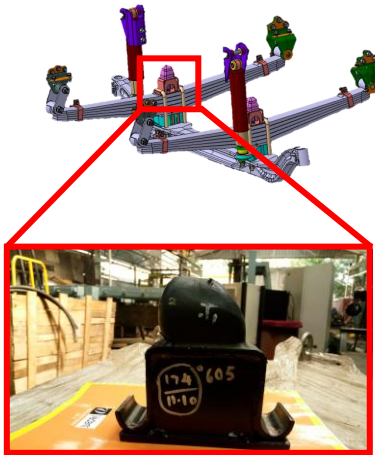
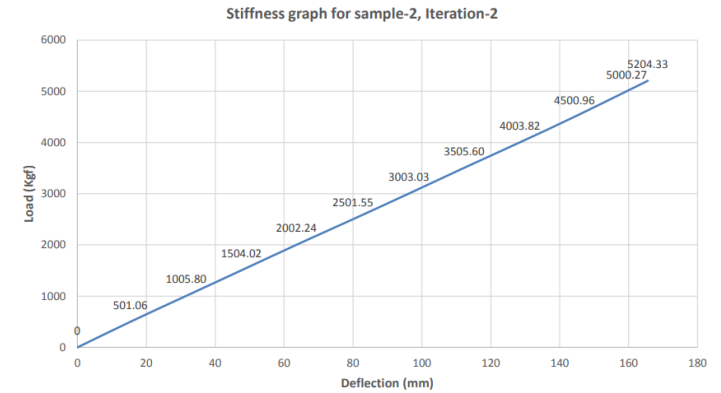
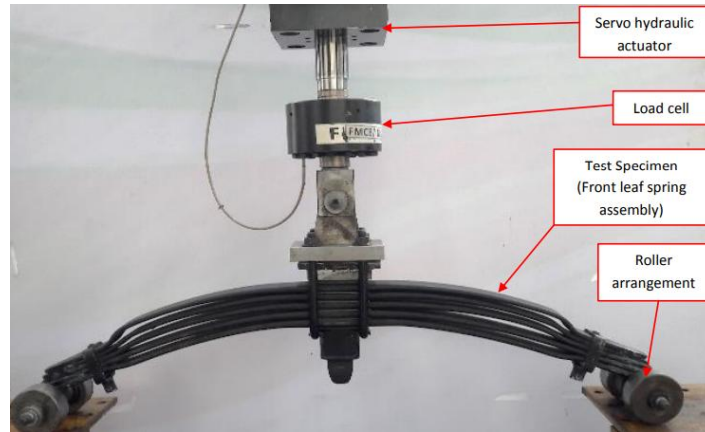
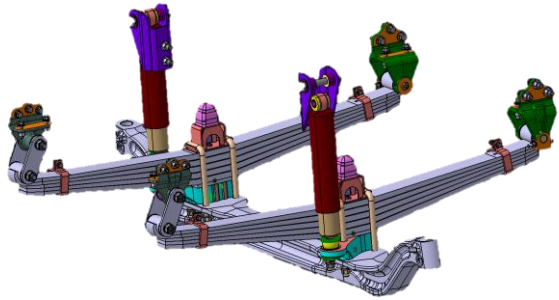
Page 9: Microstructure characterization of new plasma coating on Al-Si alloy

Page 10: Piston motion modeling using MATLAB/SIMULINK

Page 11: Cylinder liner thermal stress analysis

Page 12: Tensile static and fatigue load calculator

Validation based design improvement

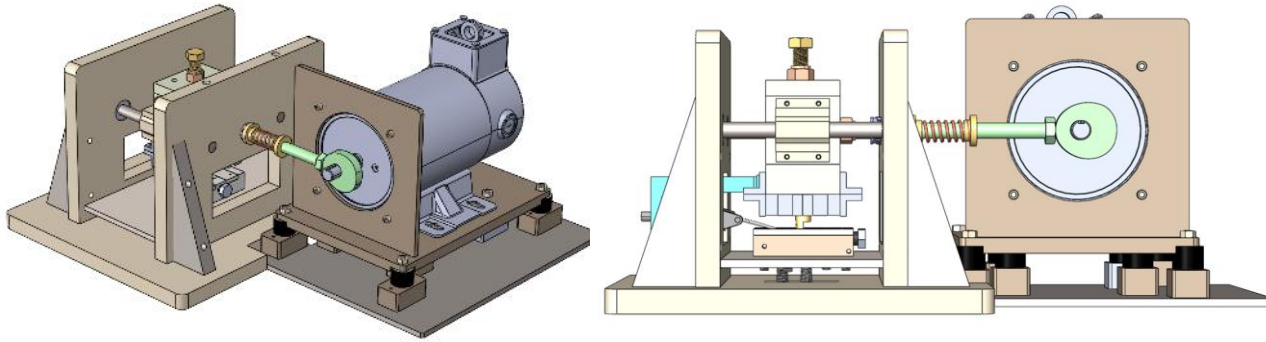


Suspension system design & prototype made based on theoretical vehicle dynamic calculations

SAE based Validation test set-up design for individual component

Validation tests results utilized to fine tune ride and handling benchmarks

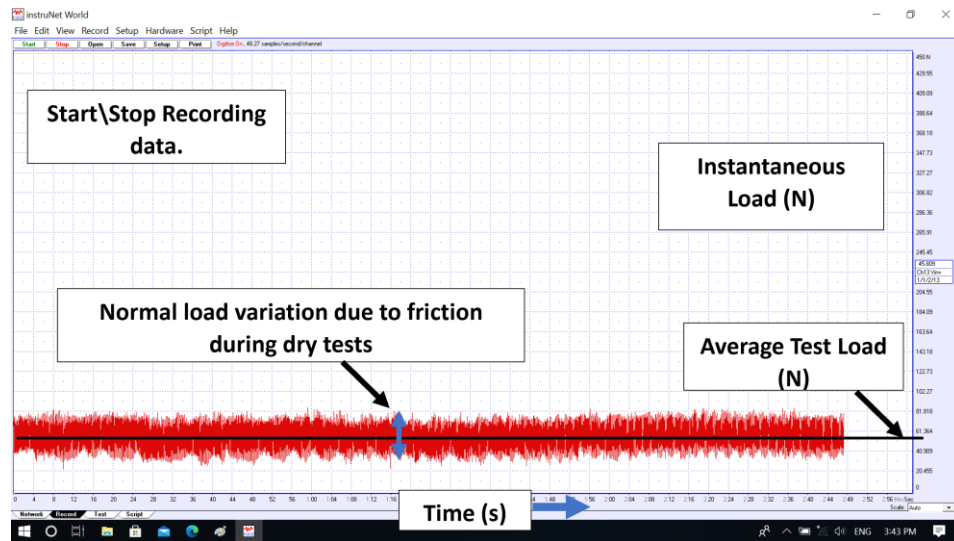
Design and Development of Tribometer



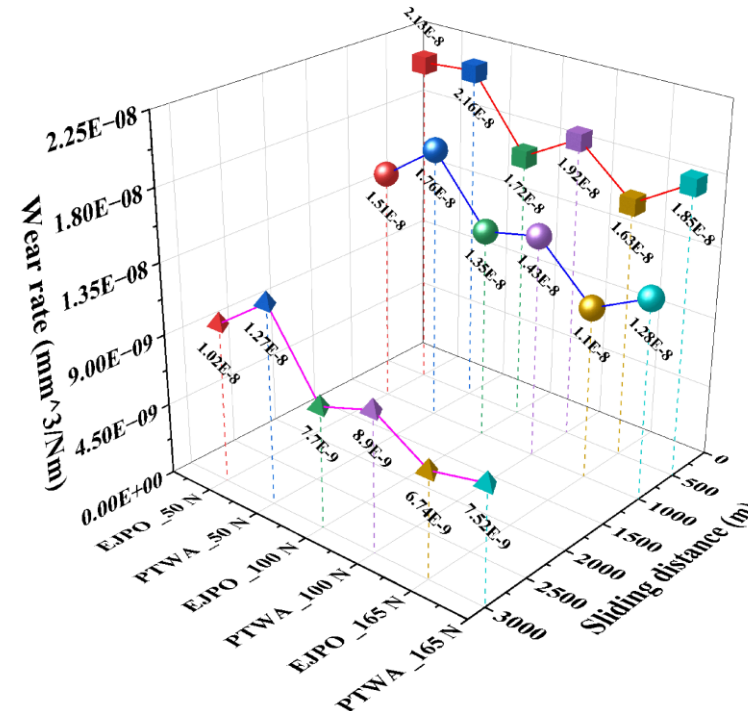
Design of the wear testing device on SolidWorks



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

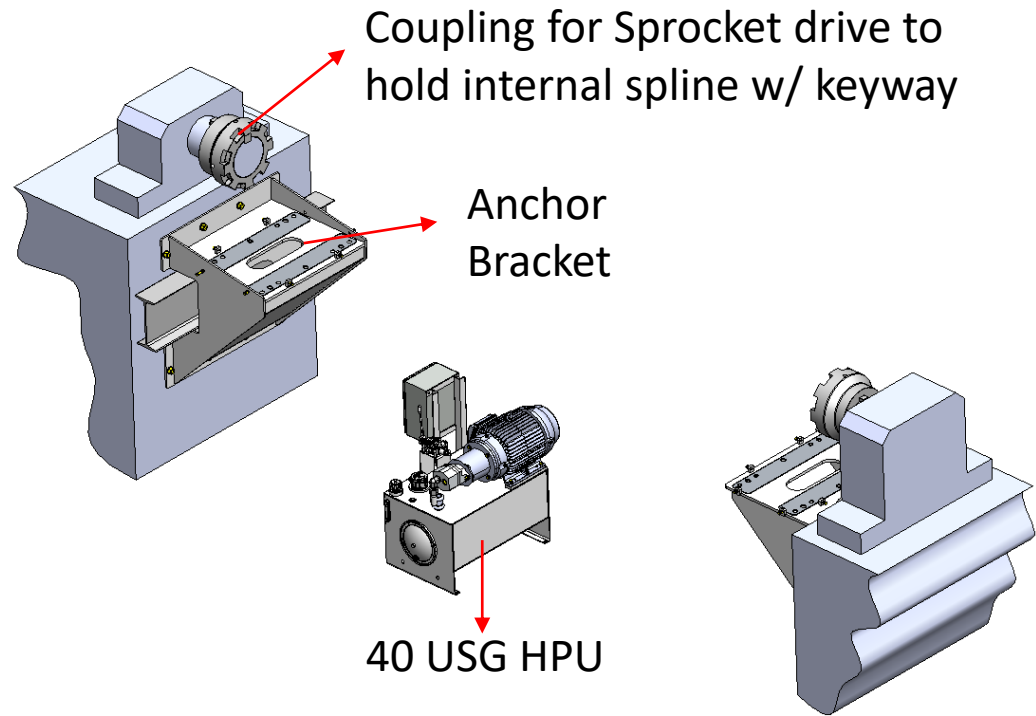


Figure: 1; Inching drive mechanism for controlled circumferential welding

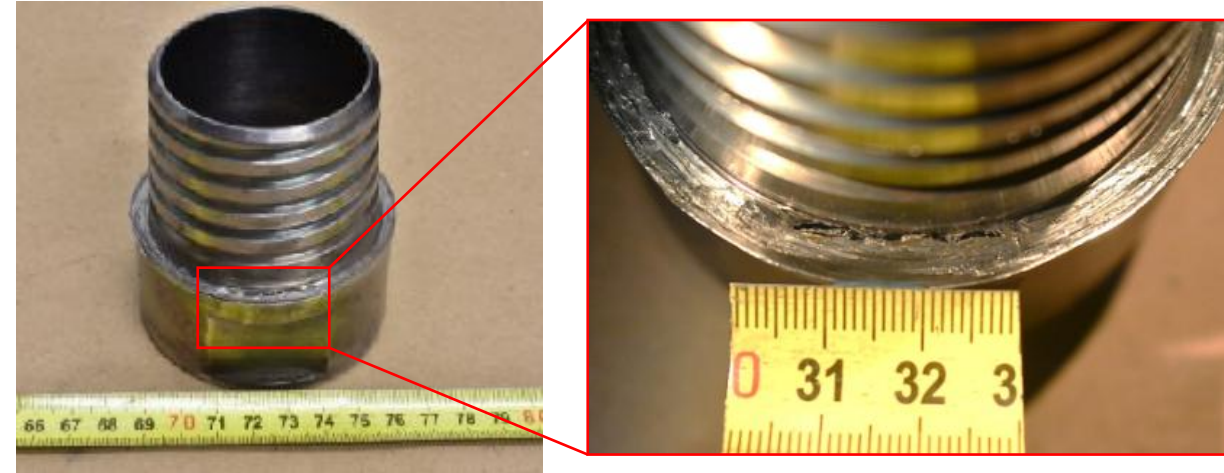


Figure: 2 ; fractured drill rod pin ends during torsional load.

- **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.

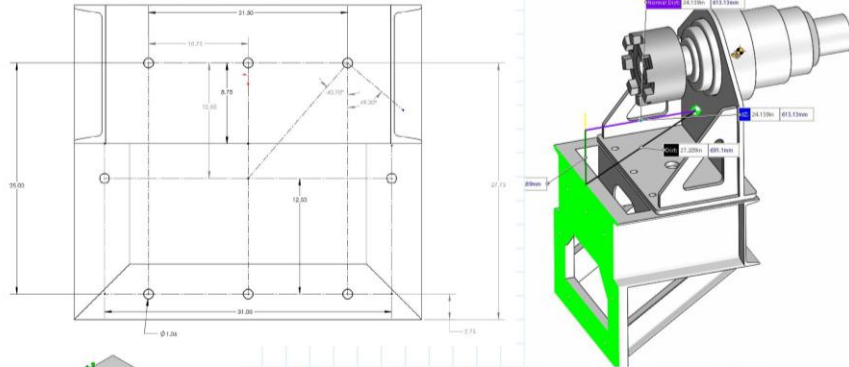
Challenge: Design an in-house drive system that can be controlled by the welder without affecting the weld pass precision and accommodating multiple rod lengths.

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

WO# 50281-Bolted bracket calculation

JUL-27-2025

Reference: Machine Design-Bhandari Pg. 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,

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

The principal shear stress is given by,

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

Note: bolts 7 & 8 not considered

INPUTS

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

$P := V$

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

$B_{TSA} := .472 \cdot in^2$ 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, $P' = P_1, P_2, P_3, P_4, P_5, P_6$,

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

$$P'_3 := P'$$

Obtaining distances from center of bolt pattern to center of bolts $r_1 = r_4 = r_3 = r_6$ and $r_2 = r_5$

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

$$r_2 := 12.5 \cdot in$$

$$r_7 := 15.5 \cdot in$$

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

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

Resultant shear force on bolt 3 (Ps)

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

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

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

STRESS DUE TO TENSION CALCULATION

The moment $V \times e$ 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

$l_3 := 2.75 \cdot in$

$$P_3 := \frac{P \cdot e \cdot l_3}{3 \cdot (l_1^2 + l_3^2) + 2 \cdot (l_2^2)} = 66.74 \cdot lb$$

Resisting forces in bolts 1 to 3

$$\sigma_t := \frac{P_3}{B_{TSA}} = 141.39 \cdot \frac{lb}{in^2}$$

Stress in bolt due to tension

$$\tau := \frac{P_s}{B_{TSA}} = 20218.27 \cdot \frac{lb}{in^2}$$

Stress in bolt due to shear

$$\tau_{\max} := \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2} = 20218.4 \cdot \frac{lb}{in^2}$$

Maximum shear stress in bolts

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

Load factor top row bolts

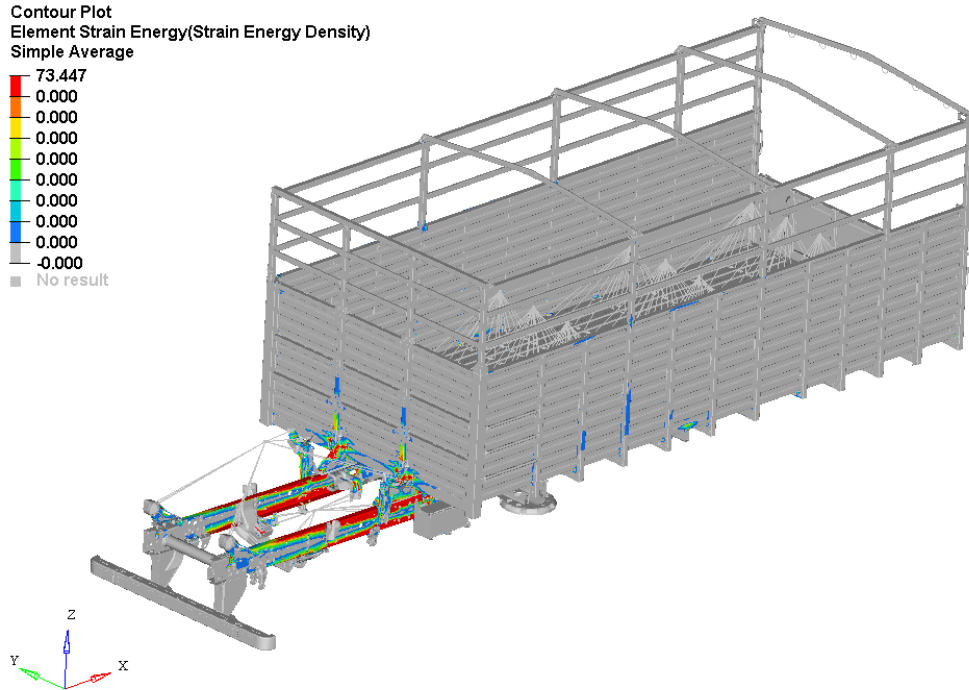


Hydraulic motor w/ CB valve for metered rotation

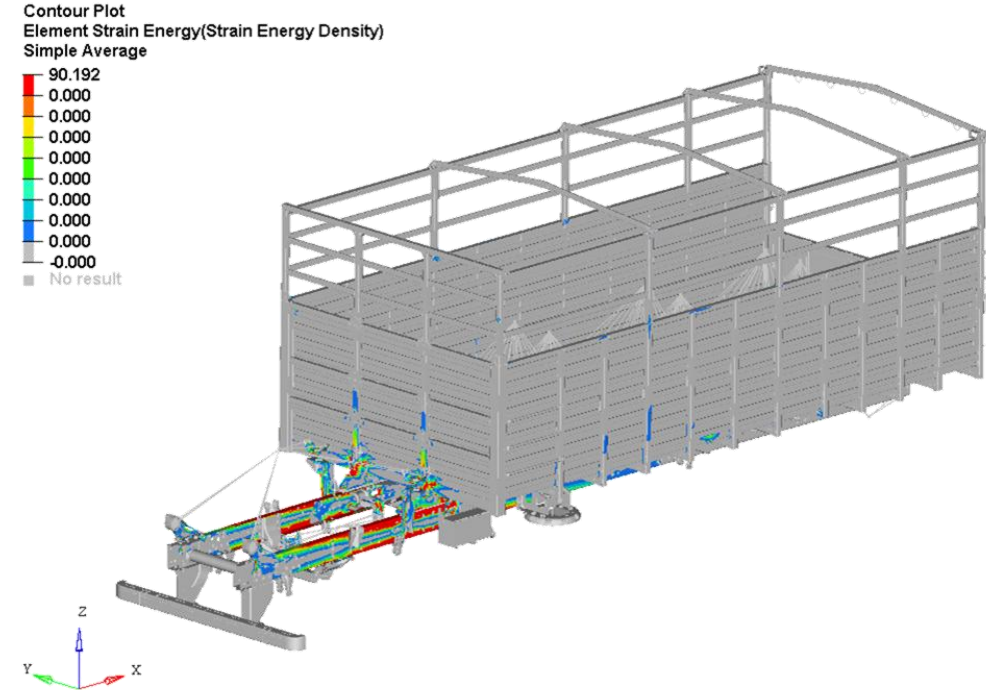
My contribution: Designing the structural components, outsource pump for hydraulic control based on Q-H curves, design hose paths and analyse shear stress.

Next Steps: Prototype validation

Stress-strain energy plot



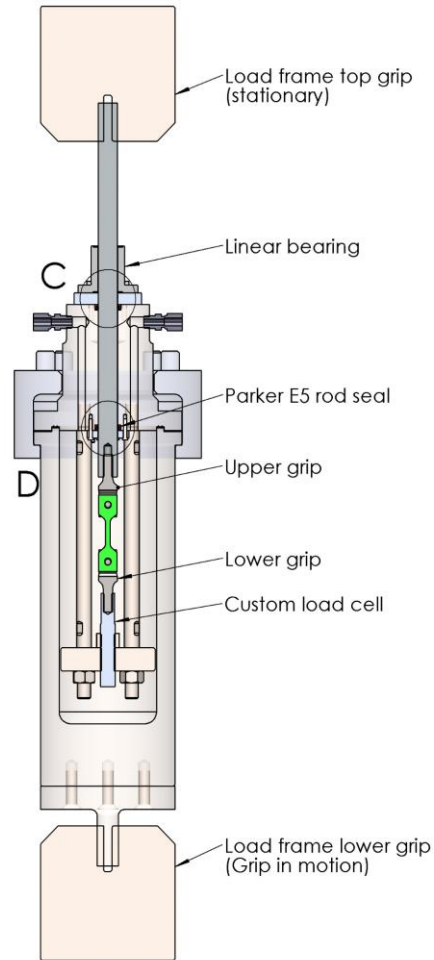
Cabin rear mount bounce: 5.6 Hz



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.

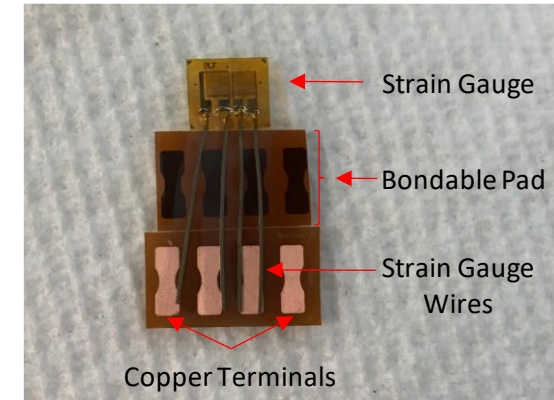
Instrumentation of HP autoclave



*Sealed Autoclave design for HP
Hydrogen embrittlement study*

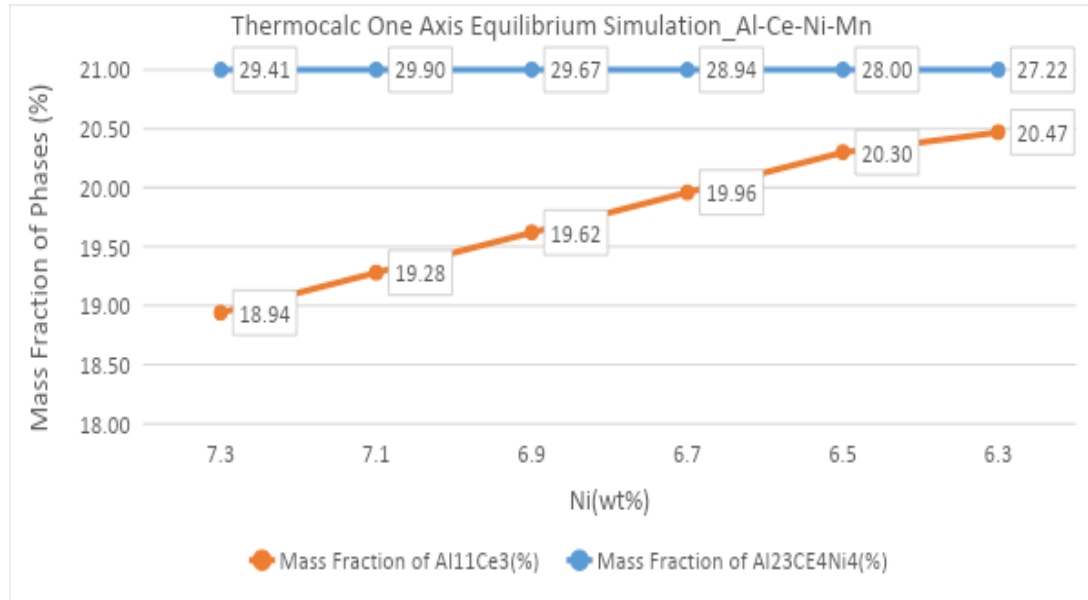


*Autoclave installed on the
MTS UTM*

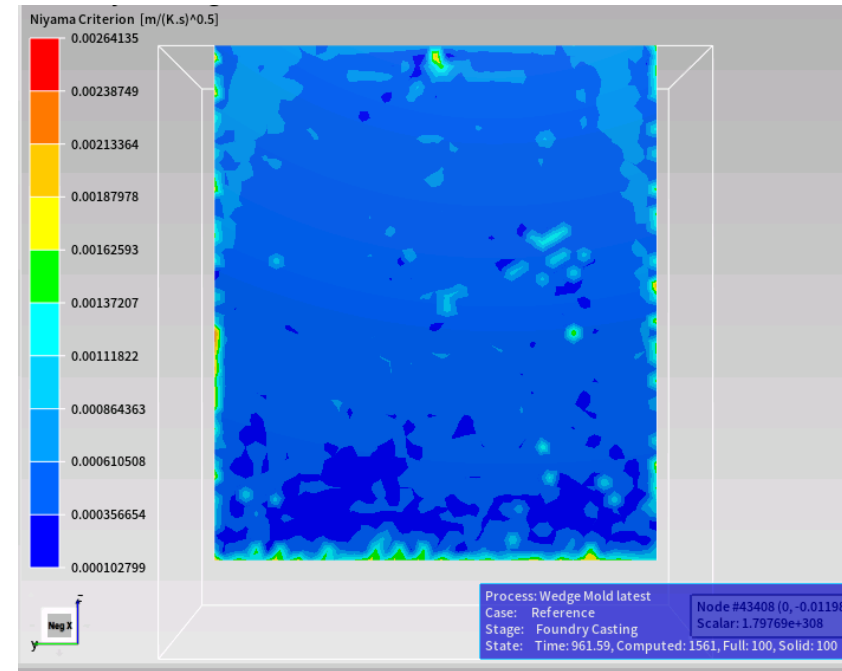


*Strain gauge assemblies to
detect deformation of the
during tensile test specimen*

Improving casting parameters using CFD



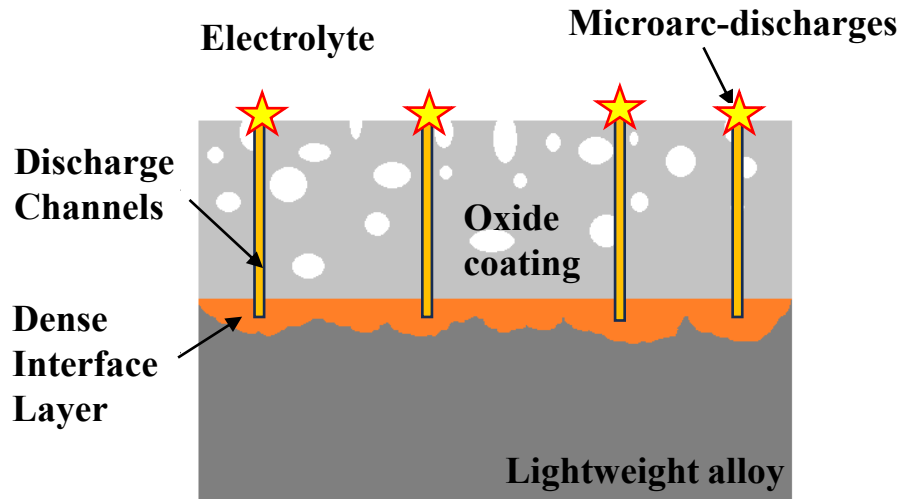
Phases post equilibrium solidification



Niyama shrinkage criterion

- Al₂₃Ce₄Ni₄ => 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.

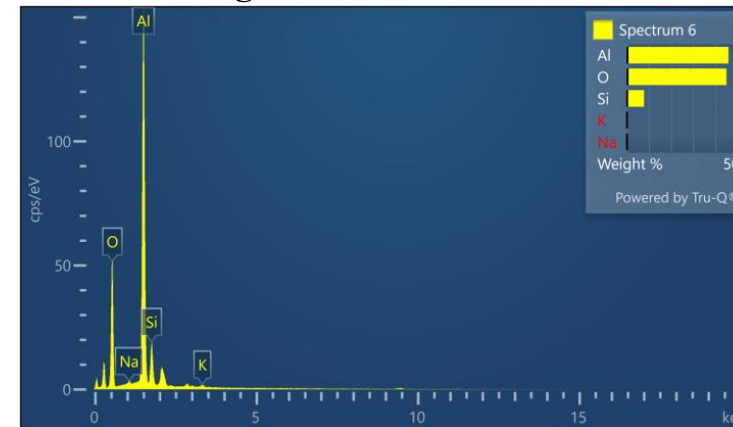
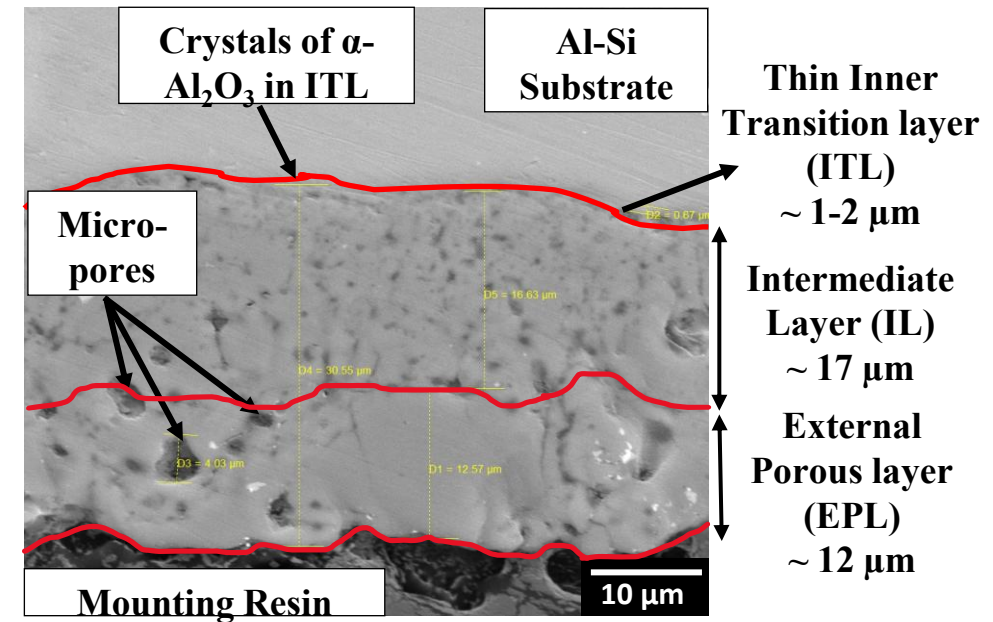
Fine-tuning of the EJPO coating process



Schematic diagram of the coating process

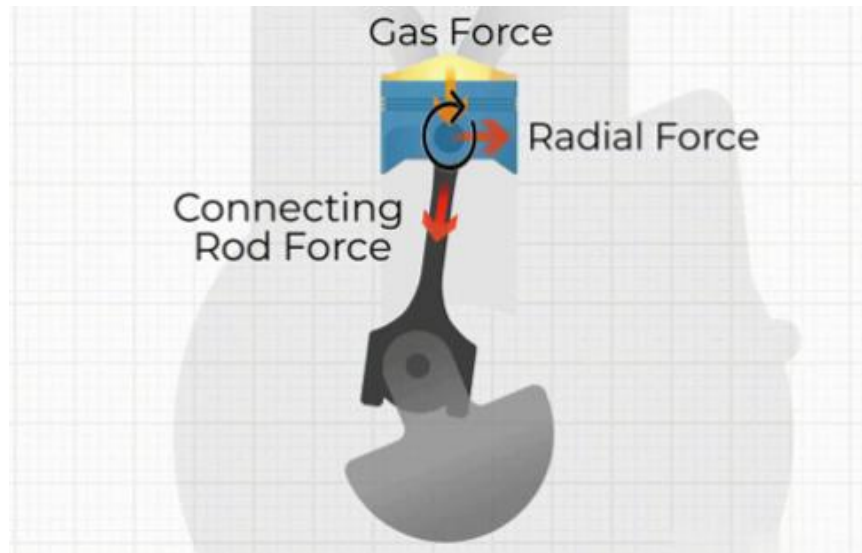


Plasma discharge around component immersed in chemically benign low-concentration electrolyte

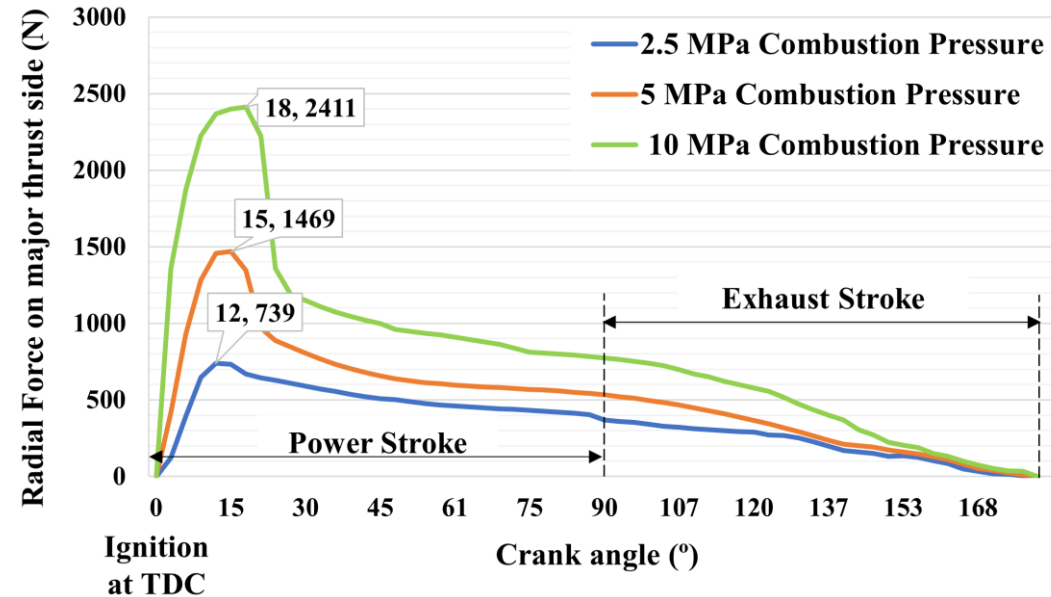


Side view of EJPO coating microstructure & elemental composition

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



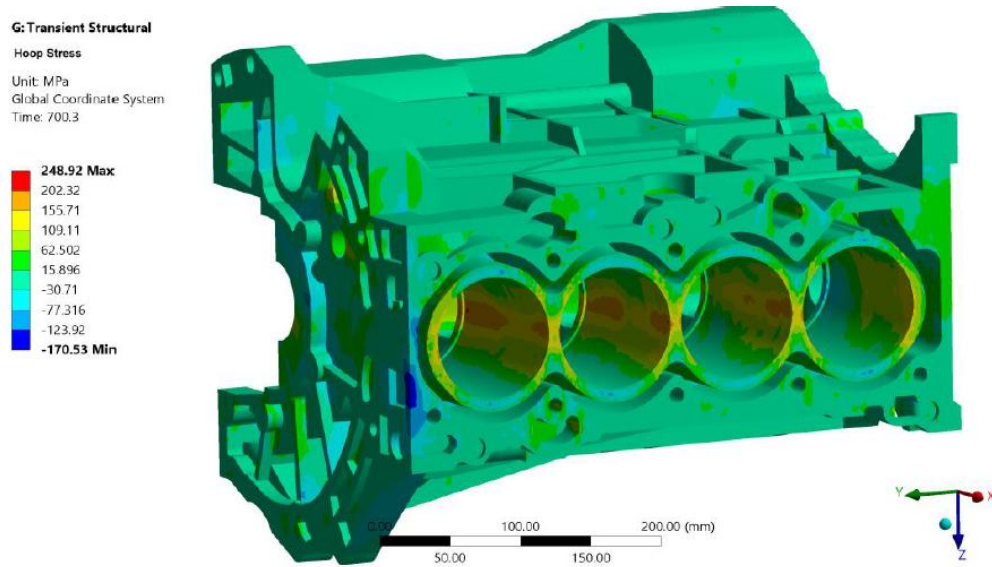
Kinematics of radial force



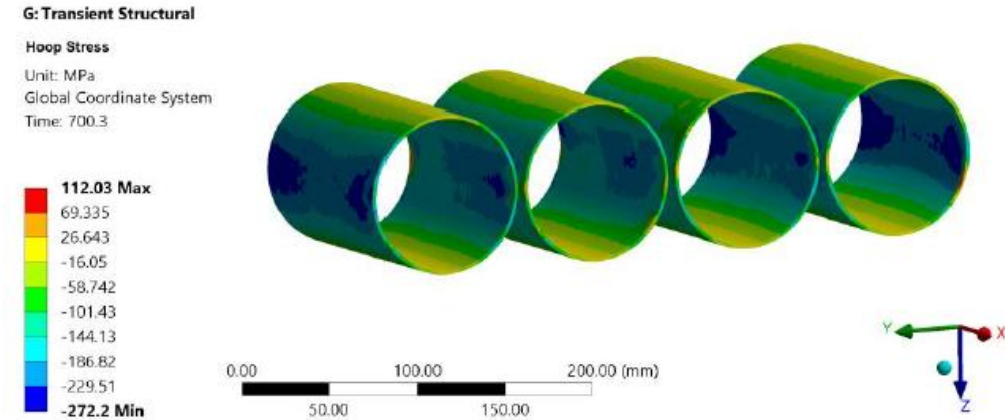
Resultant plot showing radial forces at varying crank rotation for the specified engine geometry

[MATLAB SCRIPT IN THESIS \(Page 108\) Click here](#)

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

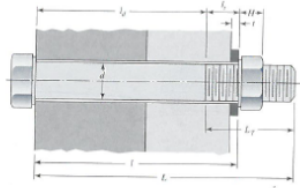
Bolt Calculator - Tensile loading .XLSX

File Edit View Insert Format Data Tools Help

100% 123 Default... 11 + B I A

H31

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)

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

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

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

$$\sigma_b = \frac{F_b}{A_t} = \frac{C P + F_i}{A_t}$$

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

$$n_L = \frac{S_p A_t - F_i}{C P}$$

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

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