Summer Interns 2018

End Review Presentation

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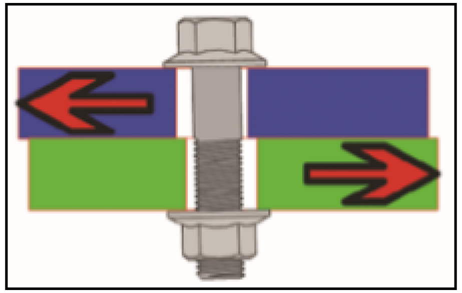
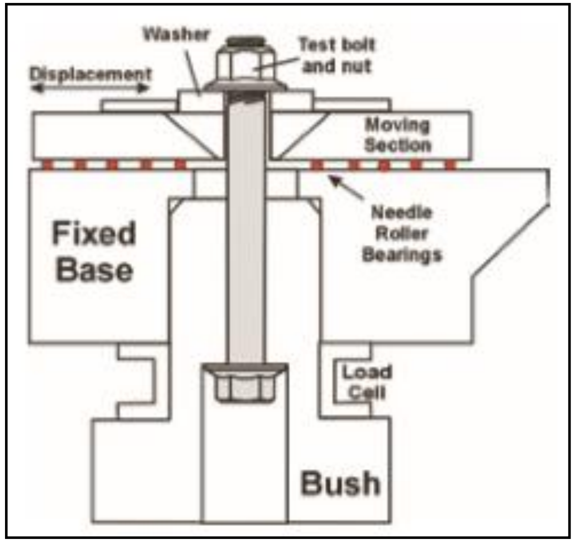
Design, Manufacture and Build in-House Junker Test Setup to simulate loosening of bolted joints in transverse vibration



Principle

- Bolt loosening is more severe for transverse dynamic loads than dynamic axial loads
- Preloaded fasteners self - loosen when there exists relative movement between mating threads and fastener bearing surface

Theory of Self-Loosening of fasteners



- In 1969, Gerhard Junker published a paper on his theory of self loosening
- Radial movement is significantly smaller under dynamic axial loads than that which is sustained under dynamic transverse loading
- Relative motion between mating threads cancels friction grip and induces an off torque proportional to thread pitch and preload
- Differential thermal effects and pressure changes at the joint interface

Deliverables and Motivation

Interim Deliverables

- Development of basic design concept
- 3D Modelling of Test setup
- Multi Body simulation, Design calculations and FEA analysis of test machine parts
- Design release for manufacturing
- Instrumentation for test monitoring

Final Deliverables

- Conduct test on different combination
- Identify issues and suggest further development in the Junker test setup developed in-House
- Effect of variable frequency, cross movement and Preload conditions on test results & conclusions
- Assess possibility of testing multiple specimen simultaneously to reduce overall run time

Motivation

Conventional Junker setup operates at low frequency (12 Hz) and less total cycles (typically 1000). In contrast Engine components are observed to loosen at high frequency and low load conditions. Although total cycles can be reduced by increase load, it might lead to failure of bolts rather than loosening. Thus there is need for a robust test setup capable of high frequency and adaptable to different bolt combinations and test loads while producing repeatable and conclusive results.

Development of basic design concept

Transverse Vibratory Motion

❖Piezo electric transducers, Cam Shaft, Eccentric Shaft and Slider Crank mechanism

❖Eccentric shaft was chosen keeping in mind the existing resources and limited time

Actuator for transverse motion

- Induction motor for maintaining constant rotational speed at different load torque
- Available motor with Rated torque of 35Nm
- Max speed of 3000 RPM/50Hz (can be stepped upto 9000 RPM/150 Hz using Gearbox)

Fig. Old Load washers – Axial strain

Load Washer Design

- Old designs were based on axial strain of cylindrical load washer which wasn't repeatable due to surface asperities
- Strain induced in bending beam produced repeatable results independent of initial orientations

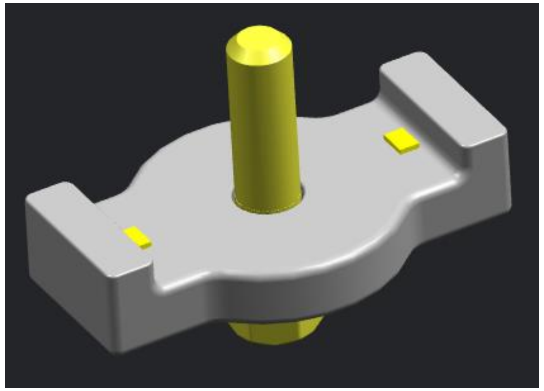
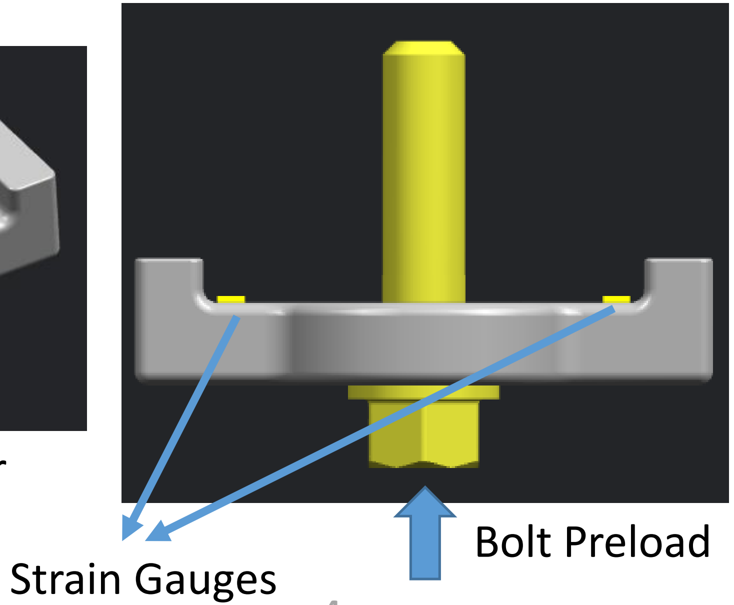
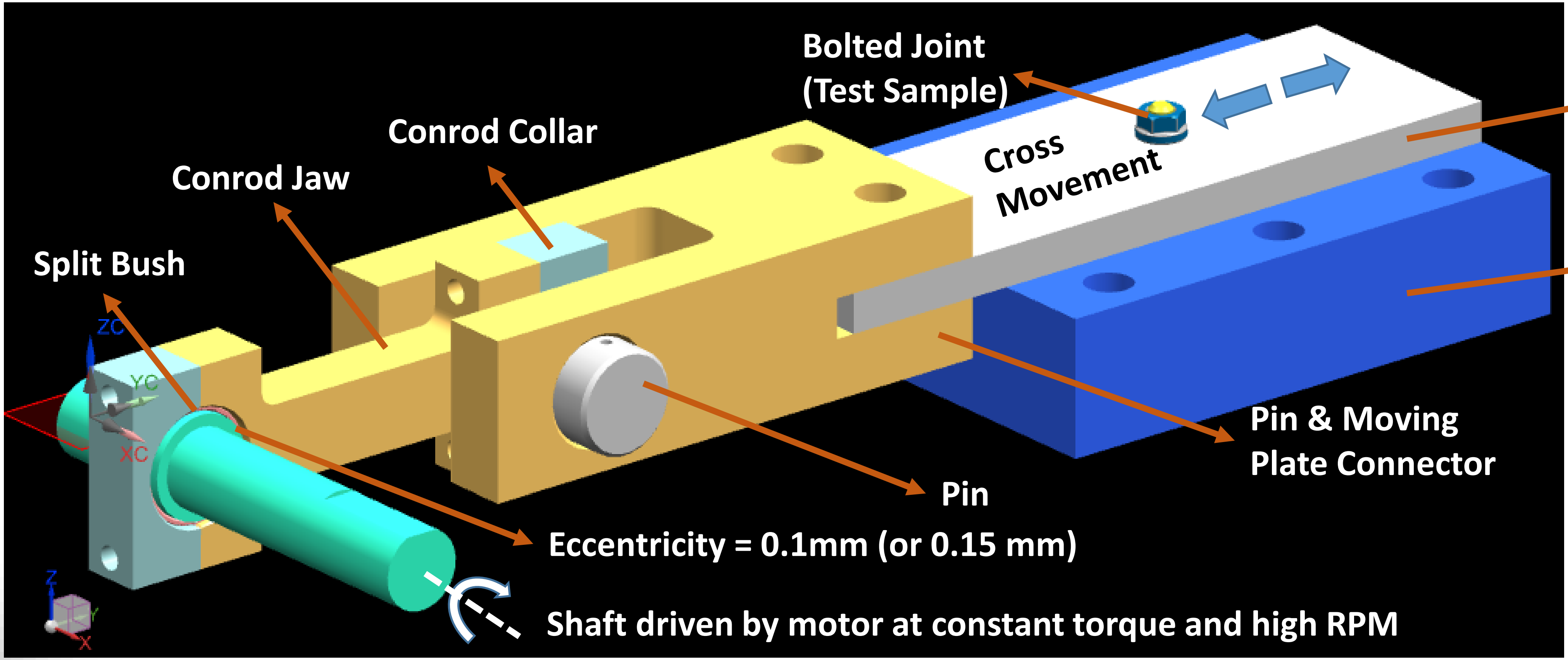


Fig. New Load washer



3D Modelling – Mechanism



Key points

- Shaft with very small eccentricity (<0.2 mm) in the central portion converts rotary motion from motor to transverse vibratory motion of a plate with a Cross movement = 2*Eccentricity

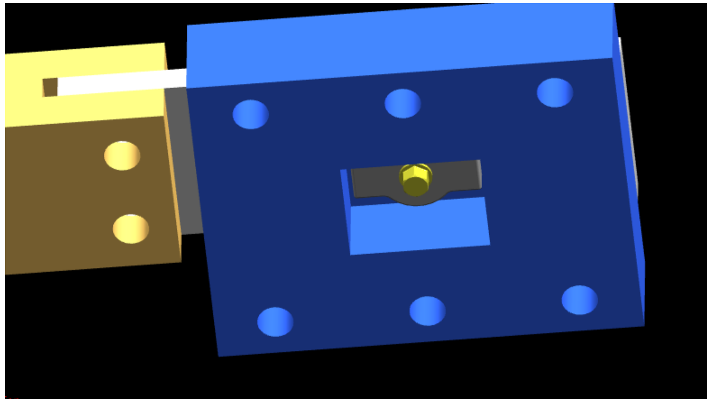
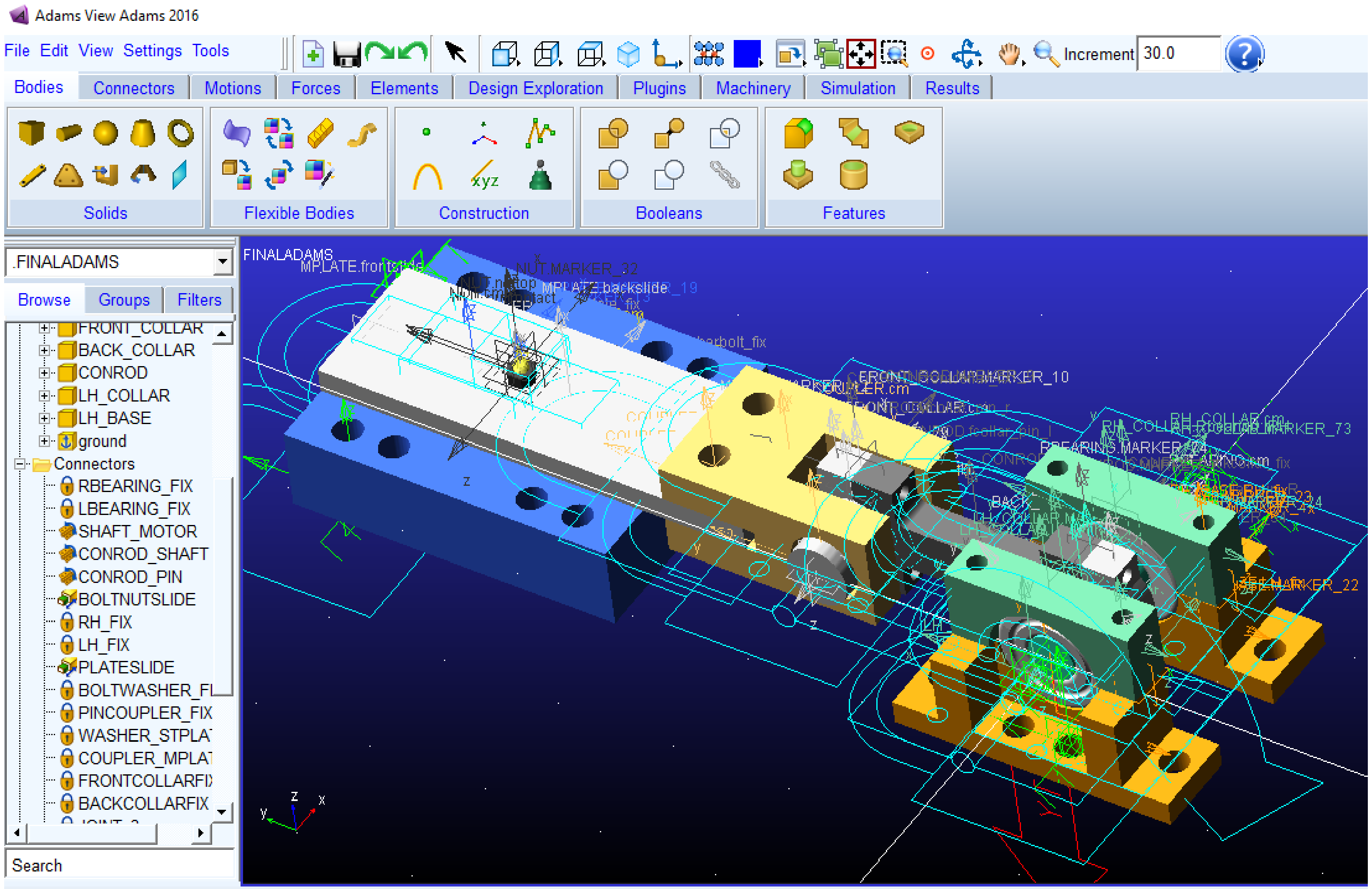


Fig. Pocket in Stationary Plate for Load Washer and test sample

Important Joints

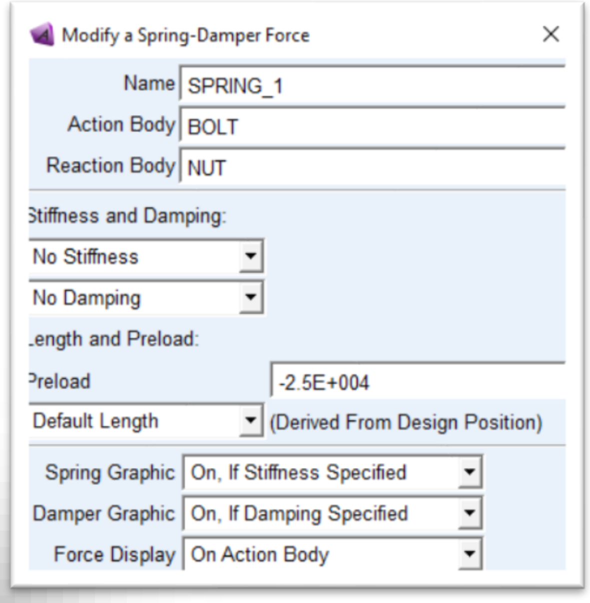
- Revolute
  1. Shaft & Motor (gnd)
  2. Shaft & Conrod
  3. Pin & Conrod
- Fixed
  1. Housings & gnd
  2. Stationary Plate & gnd
  3. Washer & St. Plate
  4. Washer & Bolt head
  5. Connector & Moving Plate
  6. Conrod Jaw and Collars
- Translational
  1. Moving & Stationary Plates



MBS – Contact & Spring Force Modelling

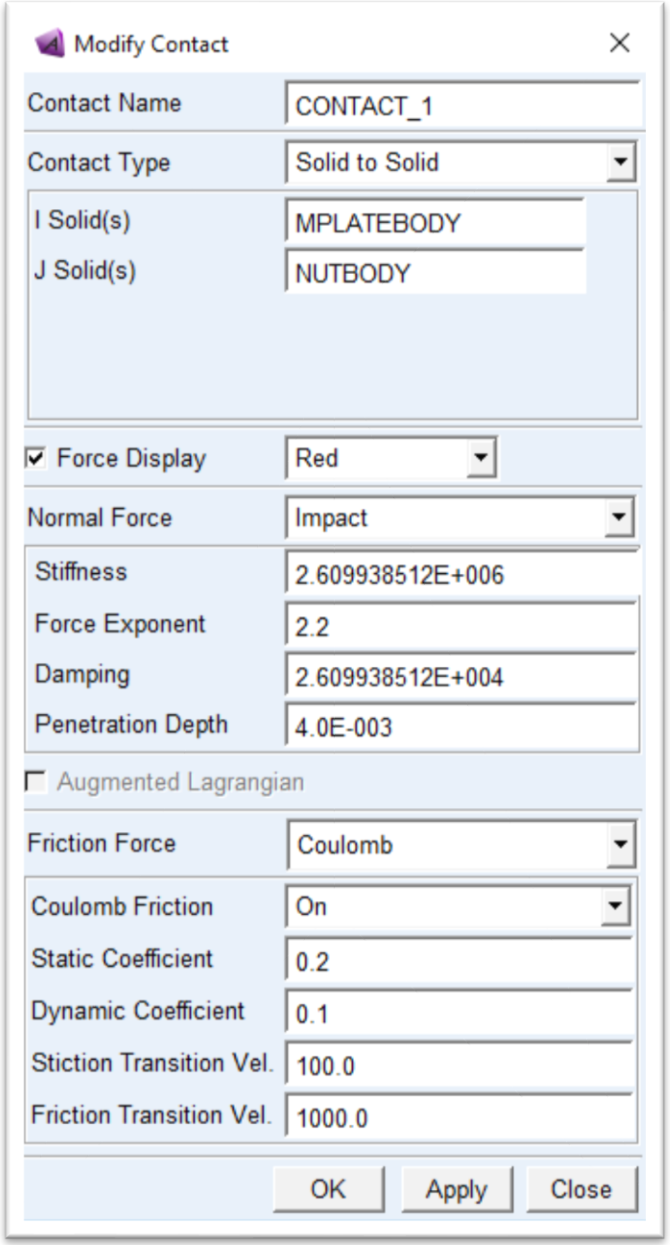
Spring Force

- Bolt and Nut interaction is modelled as a spring force with Preload equal to preload of the bolted joint

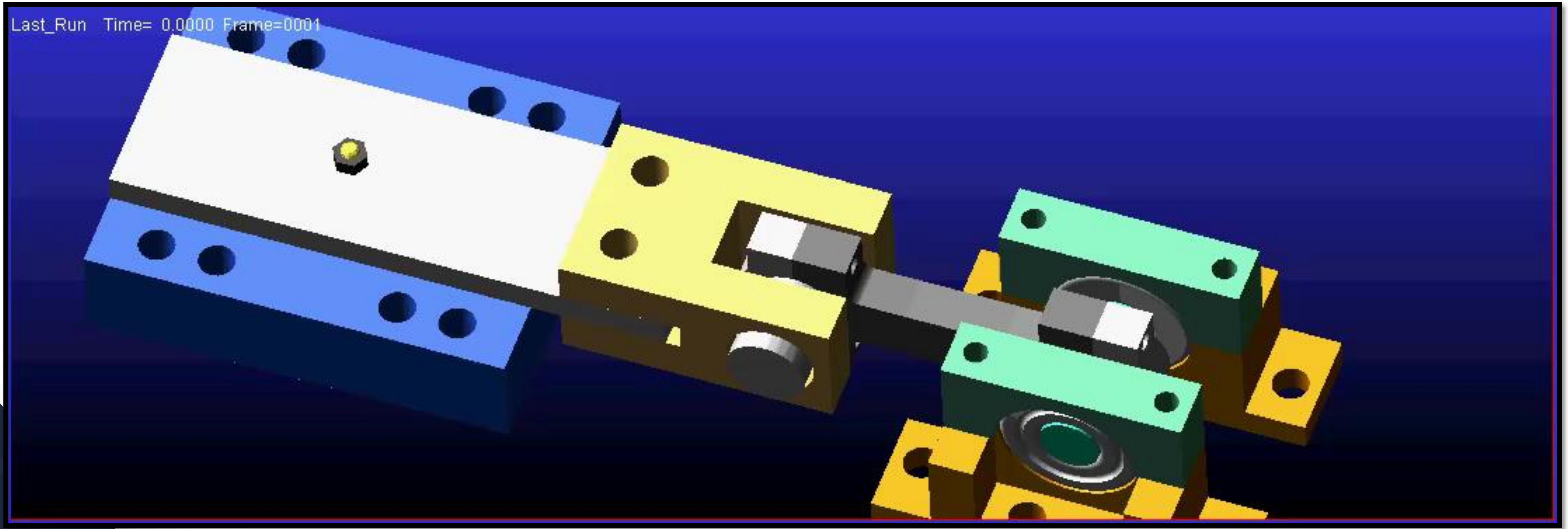


Frictional Contact

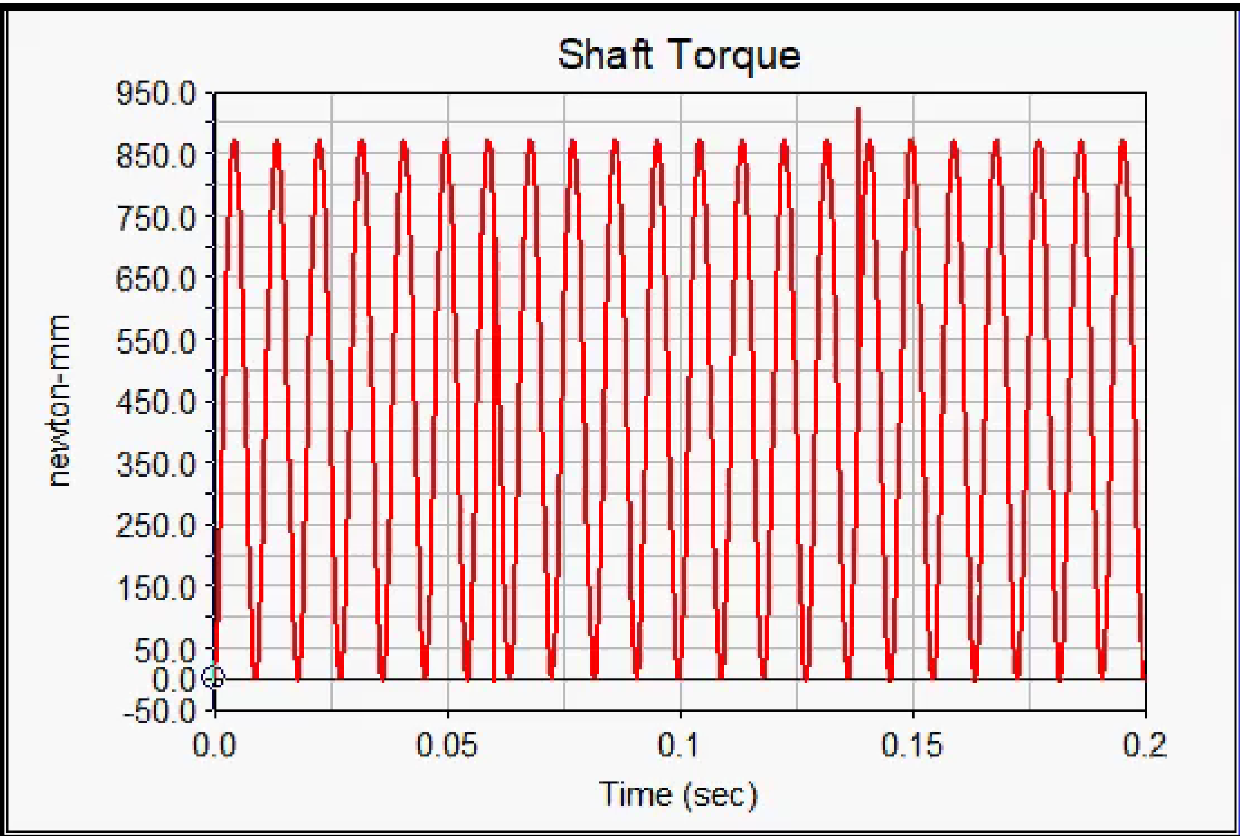
- Surface interaction between Nut and Moving Plate is modelled as a contact force with Impact normal force and Coulomb friction
- Impact functions parameters
  - Stiffness (K) $=EA/L$ (Nut Body)
  - Damping ~ 1% of Stiffness Coeff
  - Force exponent (e) > 2.1 (metals)
  - Penetration depth (δ) ~ Preload/ K
- Coulomb Friction parameters
  - μ (static) = 0.2
  - μ (dynamic) = 0.1
- Results were observed to be very sensitive to contact parameters
- Improper modelling generated noise in joint forces and input load torque
- Accurate values produced sinusoidal results



MBS – Results (Animation)



MBS – Results (Animation)



MBS – Results

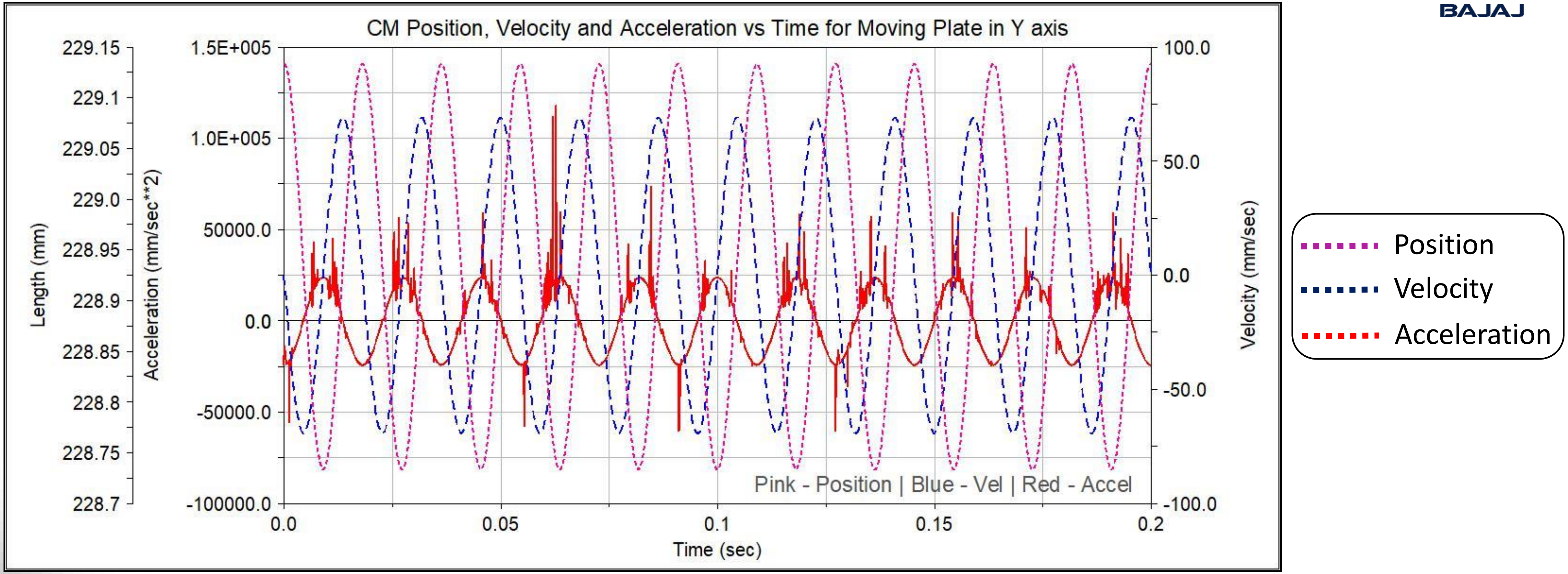
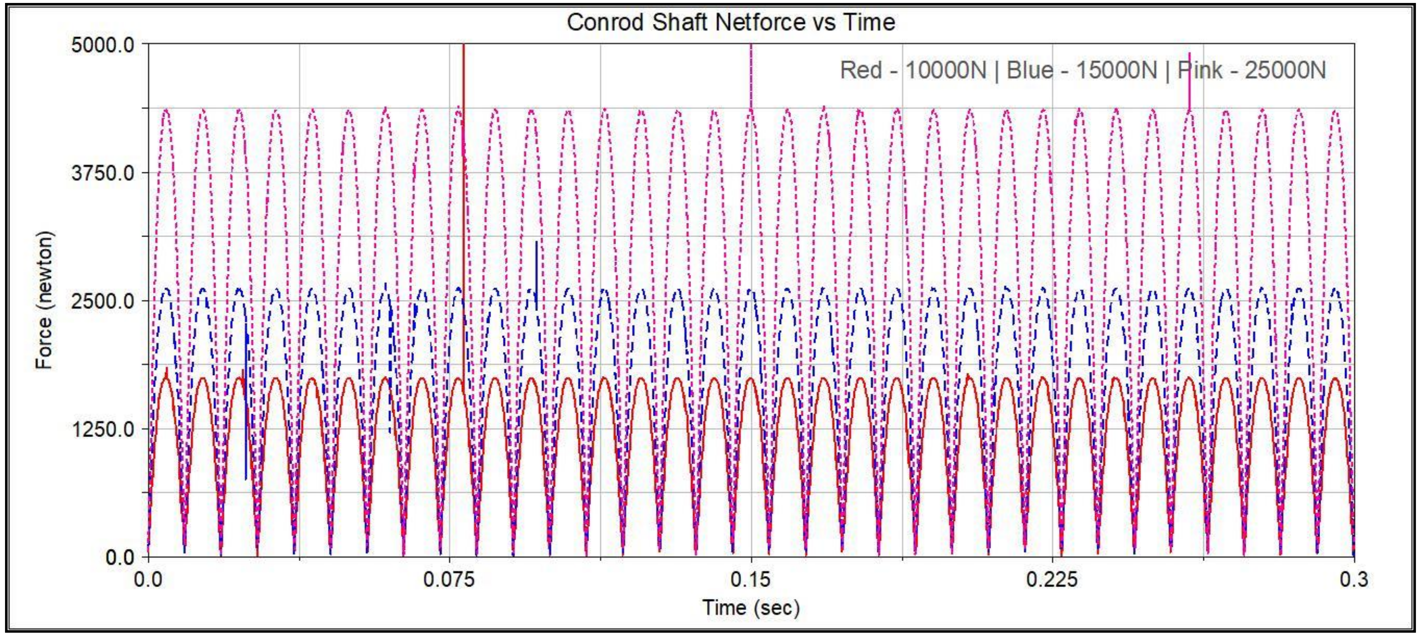


Fig. Plots for Position, Velocity and acceleration of Moving Plate Centre of Mass versus Time

MBS – Results



25000N
15000N
10000N

Fig. Plots for Conrod Shaft Joint Net force shows Joint Forces increase with Bolt preloads

Key points

- Joint Forces and Moments were found to be maximum at **25000N** Bolt Preload
- Results for max preload were used for Shaft and Pin design calculations

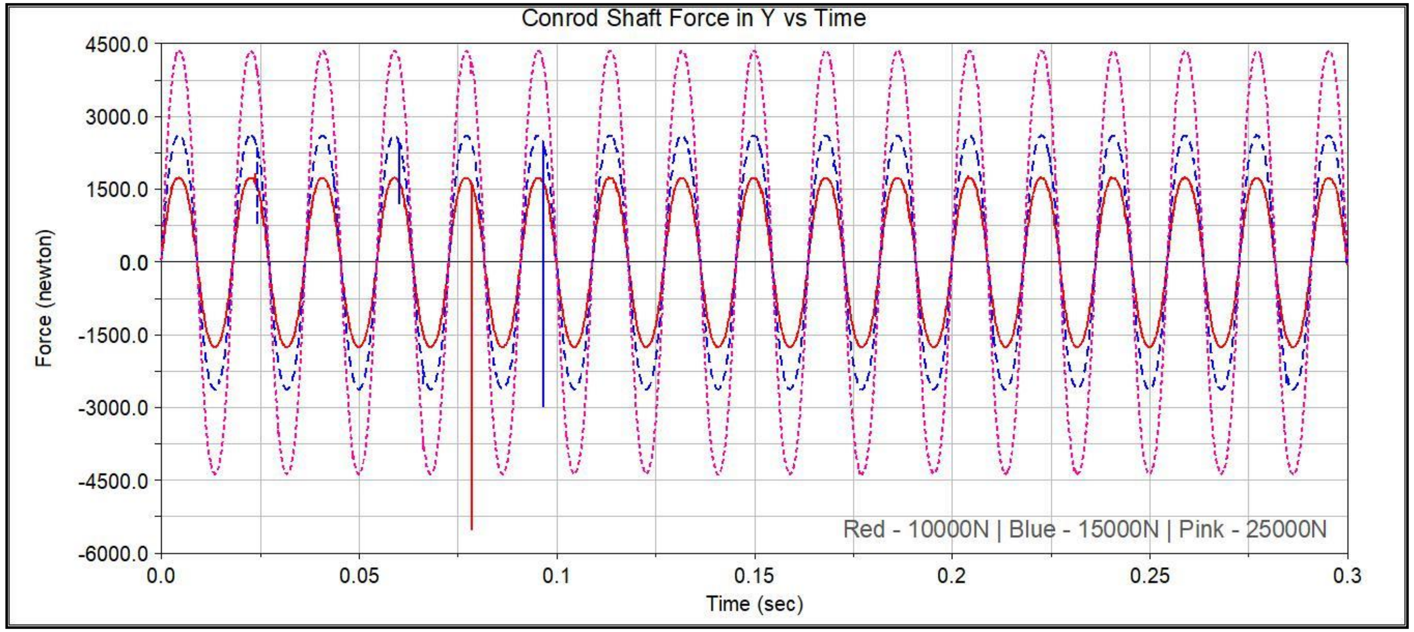


Fig. Conrod Shaft Joint force along Y axis for 3 different Preloads

25000N
15000N
10000N

MBS – Results

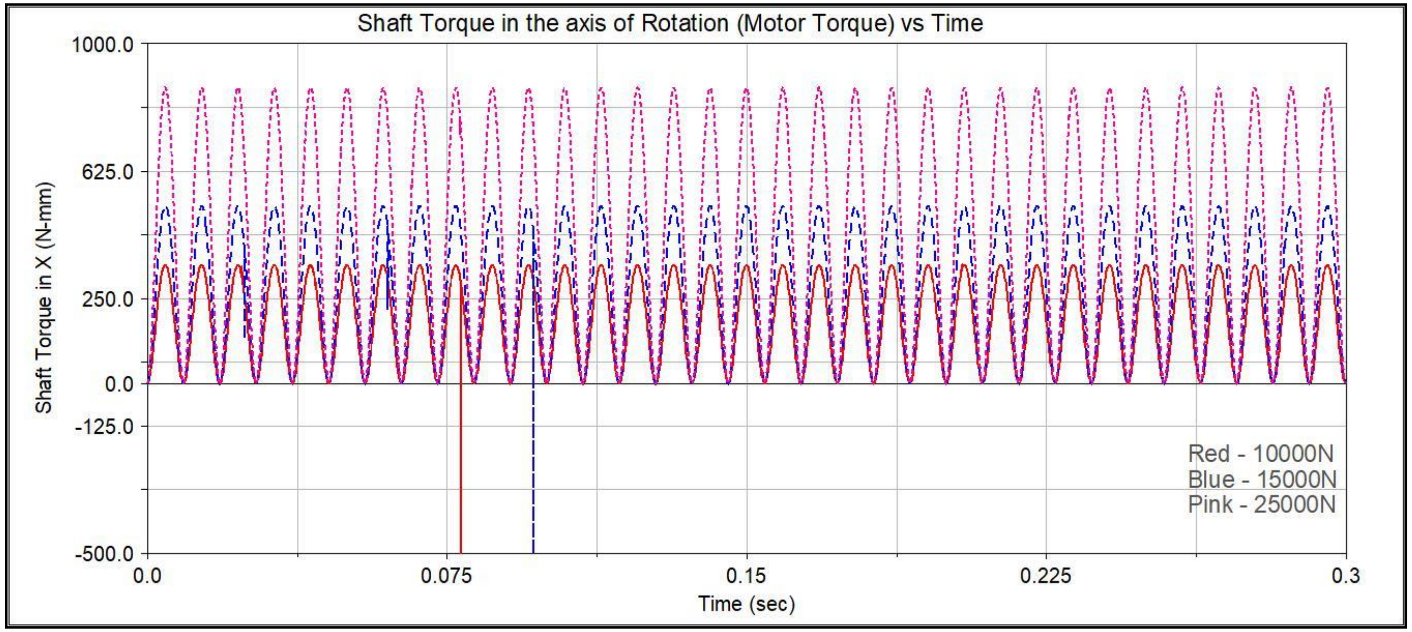


Fig. Shaft input torque for different Bolt preload values

- 25000N
- 15000N
- 10000N

Key points

- Bolted Joint between Conrod Jaw and collars undergo dynamic loading
- Shaft couplings were designed using the input torque value at **25000N** Bolt Preload

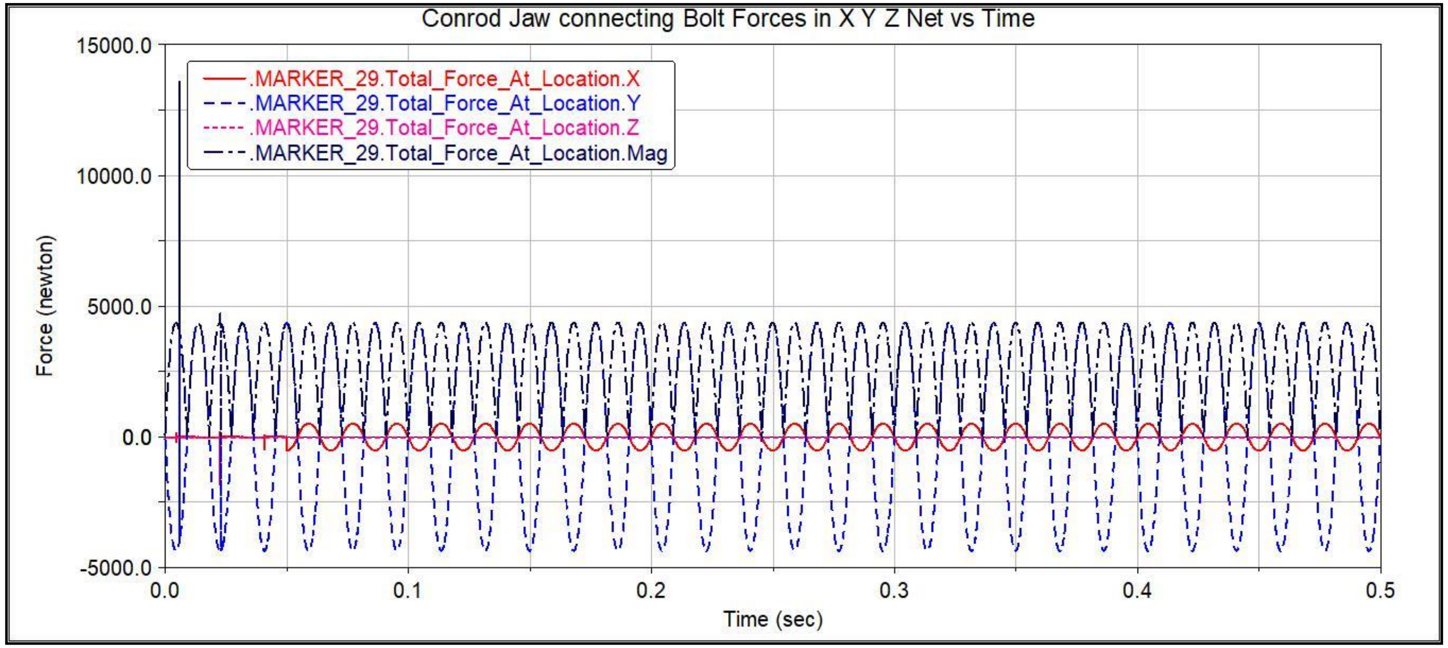
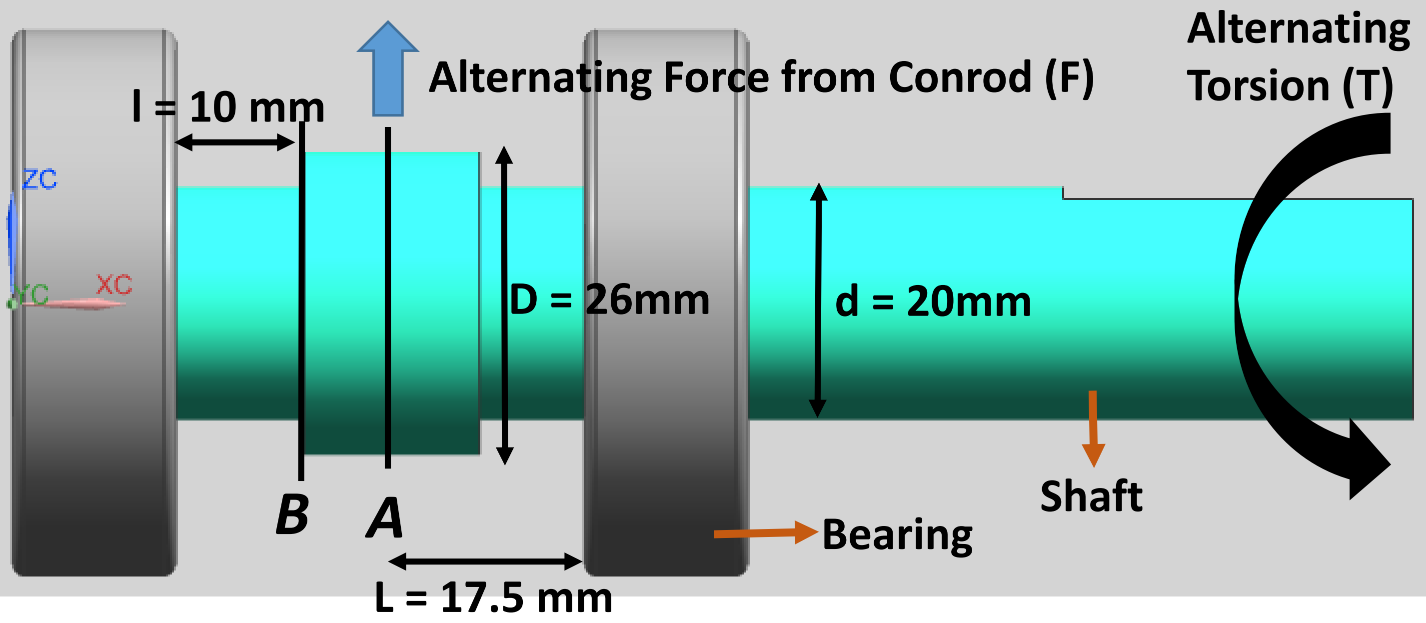


Fig. Bolted Joint Forces between Conrod Jaw and collars for different Preload

- 25000N
- 15000N
- 10000N

Design Calculations (Shaft Design)



- Shaft experiences alternating bending moment due to alternating force from revolute joint between Conrod and Shaft
- Sections ***A*** & ***B*** are critical points on shaft

Considering SCM 435H as material :

S_{ut} = 930 MPa $S_{e'} = 0.5 * S_{ut}$ = 460MPa

C_{load} = 1 (Bending) C_{reliab} = 1 C_{temp} = 1

$C_{surf} = a\,(S_{ut})^b$ =0.74 $C_{size} = 0.897(d)^{-0.107}$ = 0.65

$S_e = C_{load}\, C_{surf}\, C_{reliab}\, C_{temp}\, C_{size}\, S_{e'}$ = 222 MPa

From MBS results for Conrod Shaft Joint force :

T_{max} = 0.87 Nm T_{min} = 0 T_a = 0.435 Nm T_m = 0.435 Nm (At Both sections)

F_{max} = 4360 N F_{min} = -4360 Nm F_a = 4360 Nm F_m = 0 (At Both sections)

As L = 17.5 mm, M_a = 76.3 Nm M_m = 0 Nm (At Section ***A***)

As l = 10 mm, M_a = 43.6 Nm M_m = 0 Nm (At Section ***B***)

Considering 3 mm radius of fillet (r) at the step portion on the shaft :

(r/d) = 0.05 (D/d) = 1.3 ⇒ K_t = 1.5

$K_f = 1 + q(K_t - 1)$ q = 0.8 ⇒ K_f = 1.4

$K_f = 1 + q(K_t - 1)$ q = 0.85 ⇒ K_{fs} = 1.425

Using **ASME** code for Shaft Design, the equation for Factor of Safety (FoS) is :

$$\frac{1}{n} = \frac{16}{\pi d^3}\left\{\frac{1}{S_e}\left[4(K_f M_a)^2 + 3(K_{fs} T_a)^2\right]^{1/2} + \frac{1}{S_{ut}}\left[4(K_f M_m)^2 + 3(K_{fs} T_m)^2\right]^{1/2}\right\}$$

At Section ***A***, Shaft OD = 26mm
$(FoS)_A$ = 3.586

At Section ***B***, Shaft OD = 20mm
$(FoS)_B$ = 2.856

Bolt Preload and Theoretical Strain (Load Washer) Calculations

Bolt Preload Calculations

- Coefficient of friction between nut and clamping material is 0.15 on an average
- Tightening Torque (**T**) = KF_id, where
  - d is nominal diameter of the bolt
  - Torque coeff (K) = 0.2 (constant)
  - F_i is the Bolt Preload

Strength Grade	Size	Tightening Torque -T(Nm)	d (mm)	Torque Coefficient (K)	Bolt Preload - Fi (kN)
8.8	M6	10.5	6	0.2	8.75
8.8	M8	25.3	8	0.2	15.8125
8.8	M10	50.9	10	0.2	25.45
10.9	M6	14.7	6	0.2	12.25
10.9	M8	35.5	8	0.2	22.1875
10.9	M10	71.5	10	0.2	35.75
12.9	M6	17.7	6	0.2	14.75
12.9	M8	42.7	8	0.2	26.6875
12.9	M10	86.8	10	0.2	43.4

Minimum Strain Level Calculations for Load Washer

- For best results the minimum level of strain in load washer strain gauges greater than 150 με
- New Load Washer (SCM 435H) can be approximated to a rectangular bending beam with hinged support on both ends

- Flexure's formula : $\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{\rho}$
- Strain in bending beams : $\varepsilon = -\frac{y}{\rho} \Rightarrow \varepsilon = -\frac{M*y}{E*I}$

Here,

M **=** Bending moment
y **=** Height from Neutral axis (NA)
E **=** Modulus of Elasticity
I **=** Area moment of Inertia
P = Bolt Preload
x = Position of strain gauge from free end

Bolted Joint in Conrod Assembly

- **Material Properties** :

  Proof Strength (S_p) = 970 MPa

  Ultimate Tensile Strength (S_{ut}) = 1220 MPa

  Endurance Strength (S_e)= 570 MPa

  Proof Load (F_p) = $S_p * A_t$

  Preload (F_i) = 0.75*F_p ⇒ F_i = 16995N

  Preload Stress (σ_i)= $\frac{F_i}{A_t}$

- **Geometry Parameters :**

  Tensile Stress Area (A_t) = 34.86 mm^2

  Unthreaded shank area (A_d) = 37.41 mm^2

  Thread length (L_t) = 18 mm

  Unthreaded Length (L_d) = 28mm

  $\mathbf{K_m = AEd(e)^{-Bd/l}}$ ⇒ 900579269 N/m

  $\frac{1}{K_b} = \frac{1}{E}\left(\frac{L_t}{A_t} + \frac{L_d}{A_d}\right)$ ⇒ K_b = 162863886 N/m

  Stiffness Coeff of bolted Joint (**C**) = $\frac{K_b}{K_m+K_b}$

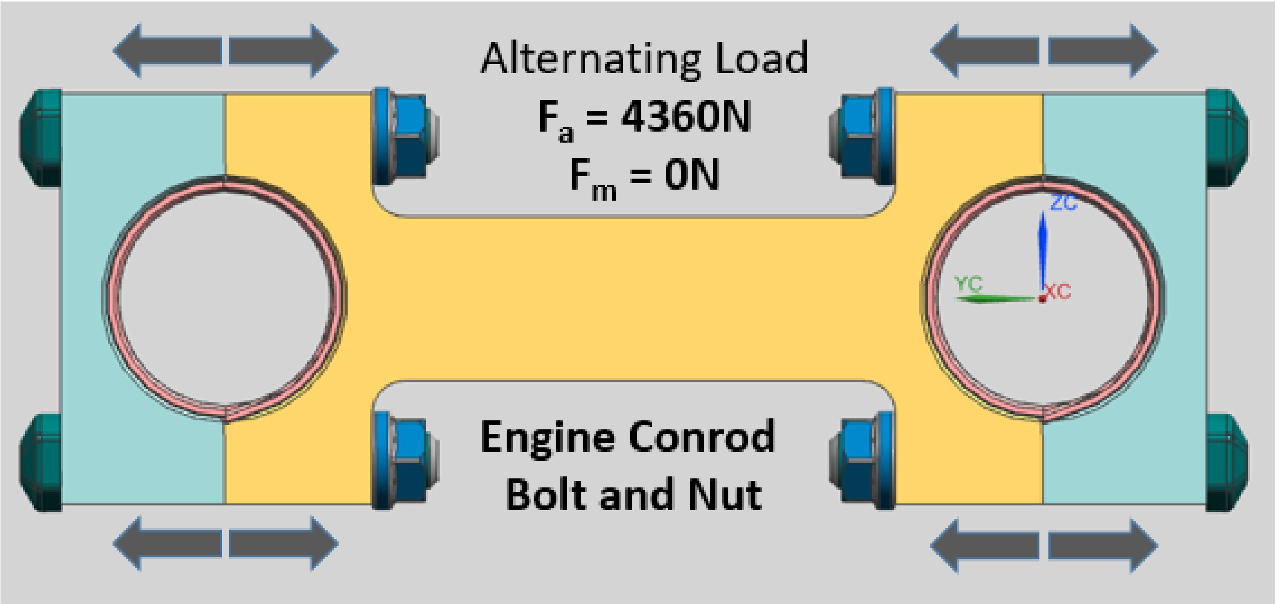
  ⇒ C = 0.1358

Alternating Stress due to dynamic Loading (σ_a) = $\frac{F_a}{A_t}$

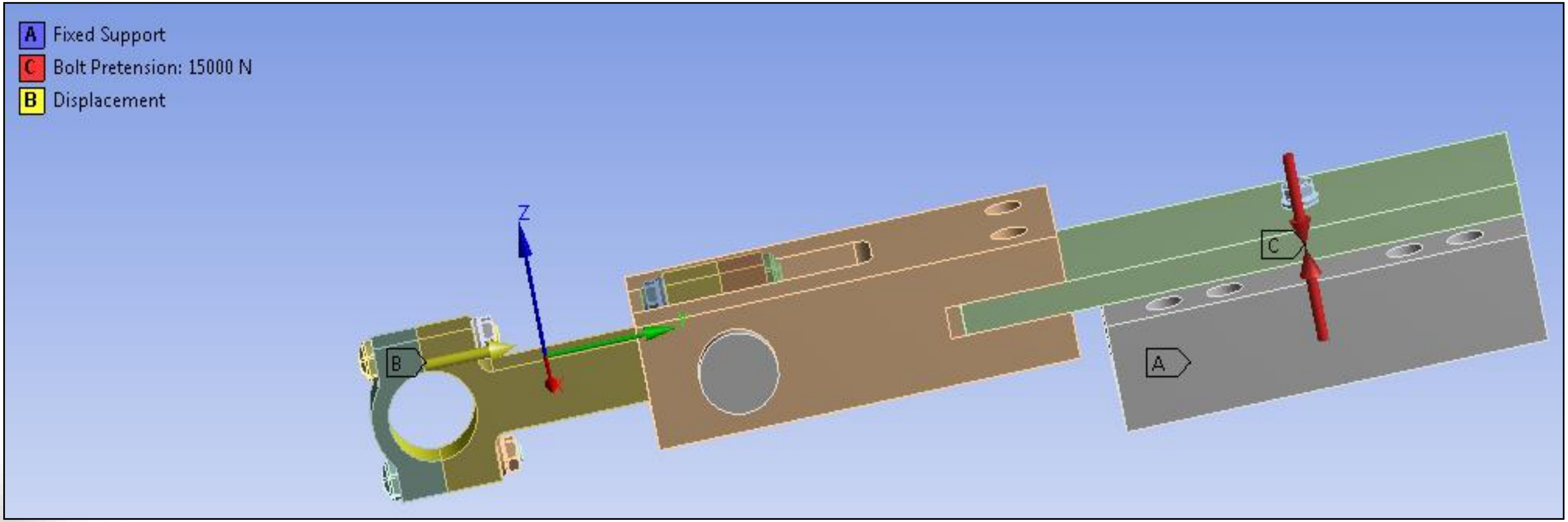
Factor of Safety against Joint Separation (**N_o**) = $\frac{F_i}{P(1-C)}$

Factor of Safety against Fatigue Failure (N_f) = $\frac{S_{ut}(S_{ut}-\sigma_i)}{\sigma_a(S_{ut}+S_e)}$

N_o = 4.51 N_f = 2.415



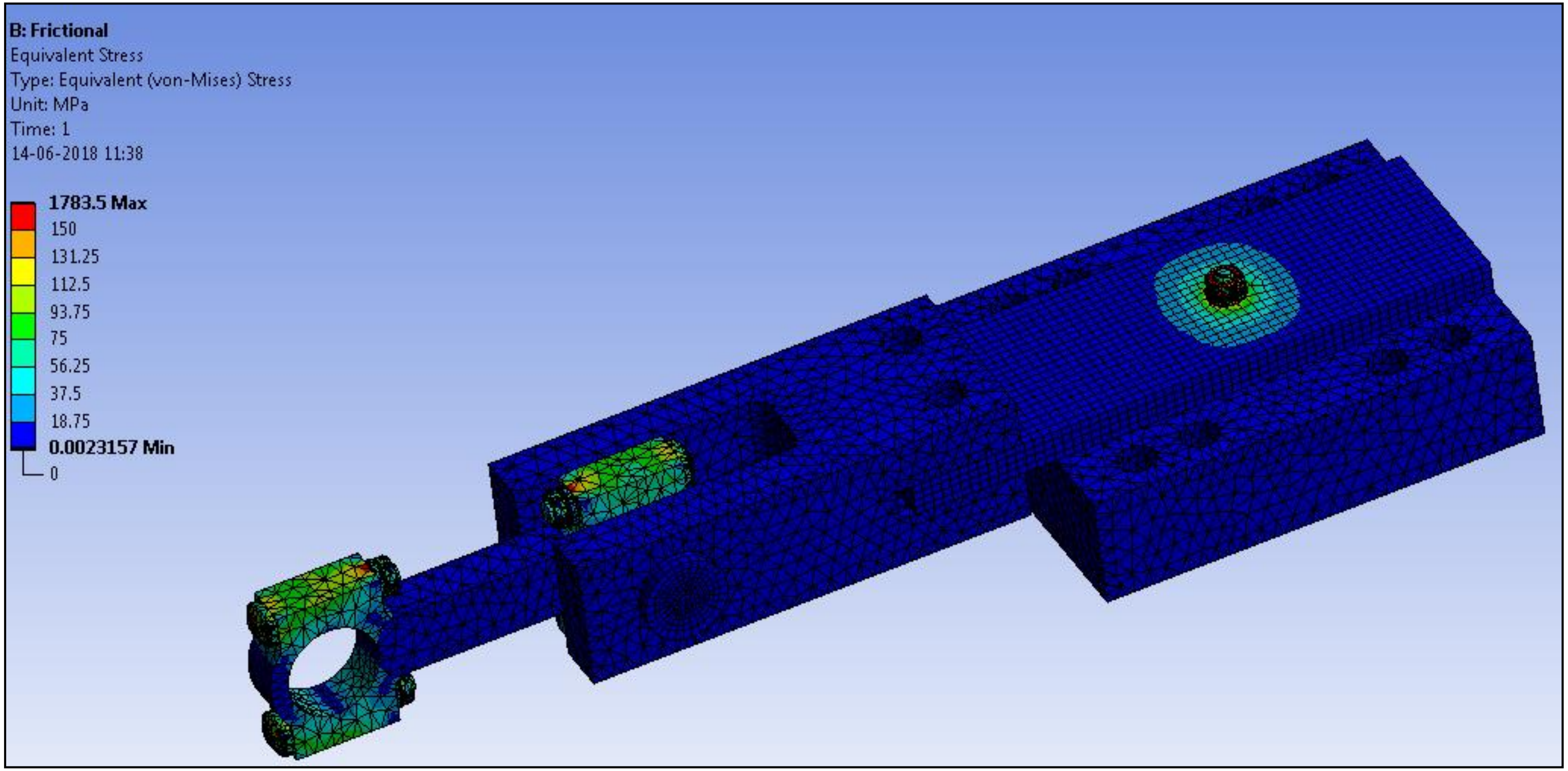
FEA – Loading



Loading

- ❖ Stationary Plate rigidly fixed to ground
- ❖ 15000N Bolt Preload and a frictional contact applied at bolted joint to be tested
- ❖ Frictional contact between Pin and Connecting Rod
- ❖ Displacement of 0.2mm applied to Connecting Rod

FEA – Stress Analysis



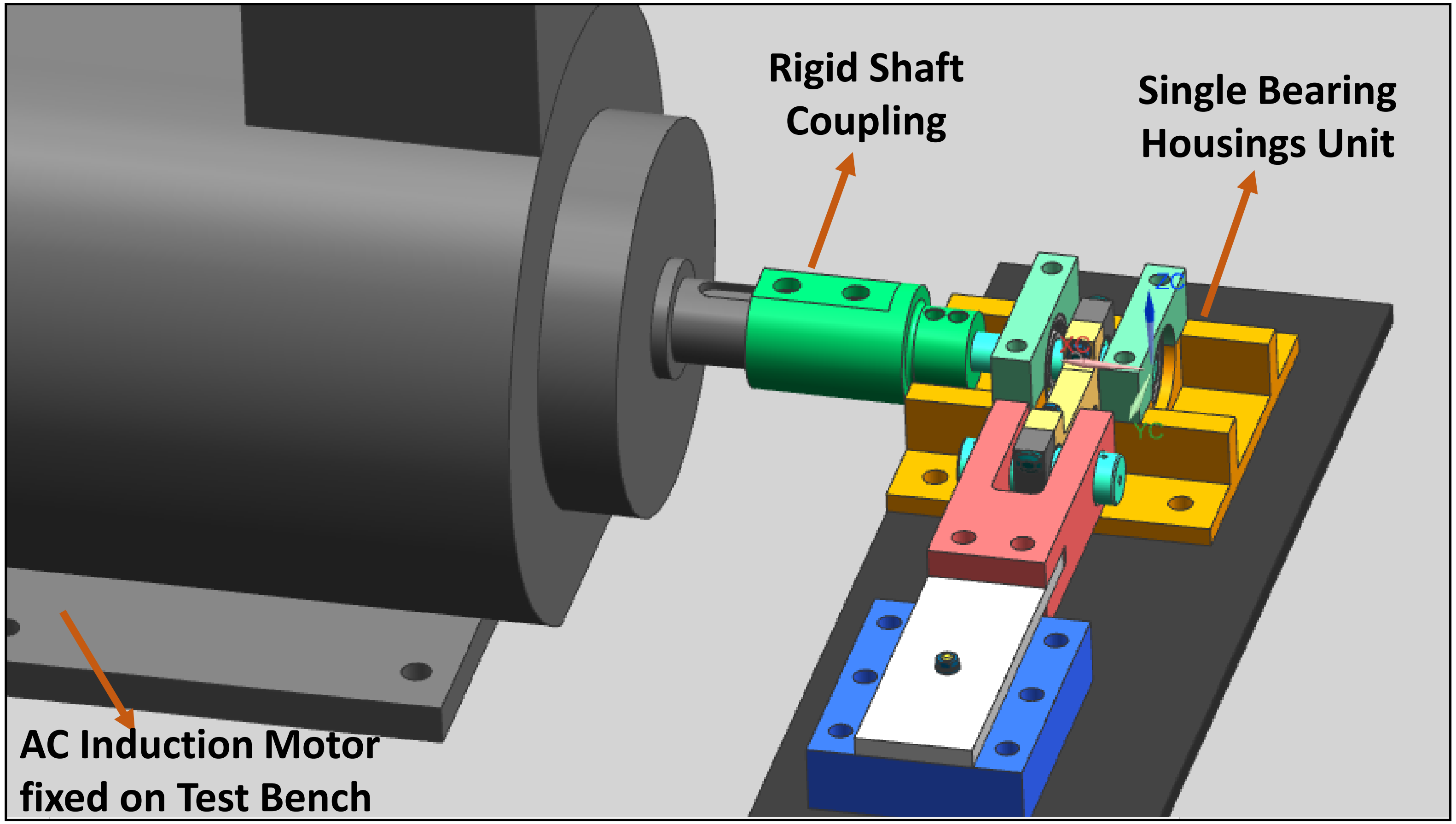
Stress Analysis

- Mild steel (UTS ~ 500 MPa) was chosen as material for Plates and Housings as they undergo minimal stress
- Higher Strength Material (SCM 435H) for critical components like Conrod Assembly, Eccentric Shaft and Pin

Assembly and Manufacturing

Key points

- Rigid Shaft coupling was manufactured for the initial prototype due to limited time
- Bearings Housings on either side of shaft was manufactured as single units for perfect alignment of bores
- Close tolerances assigned for all manufactured components
- Bearings are press fit on the shaft
- Interference fit for pin and connector



Strain Gauging and Calibration Setup

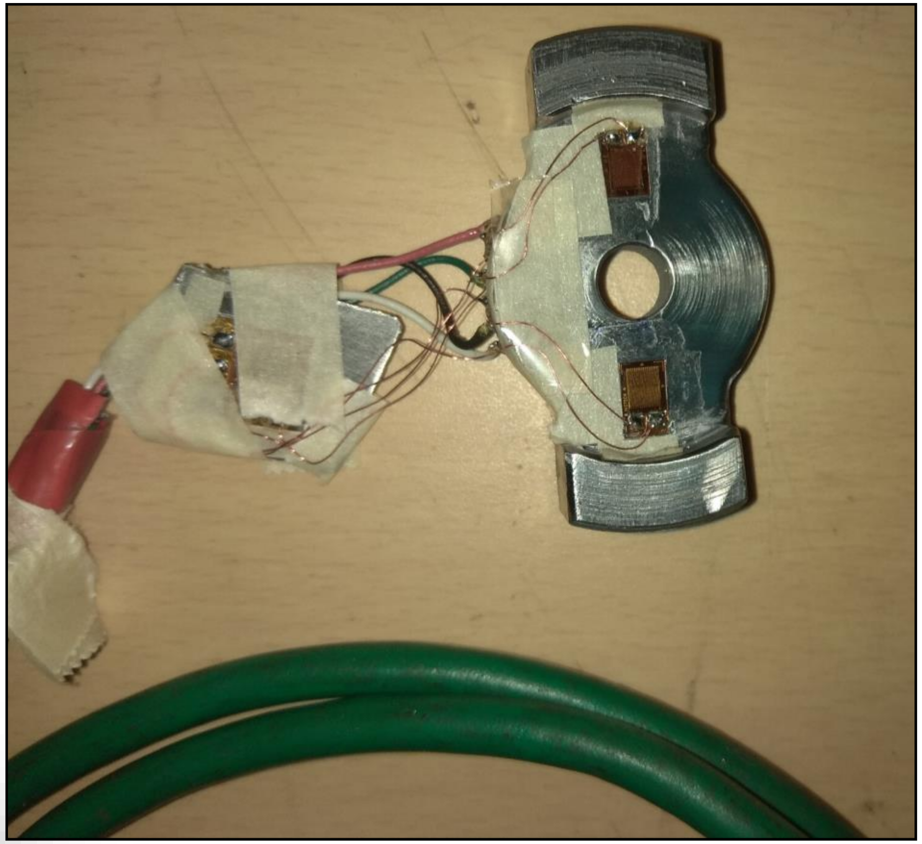


Fig. Bending beam arrangement of Strain Gauges on the Load Washer with 2 active resistors and 2 dummy resistors

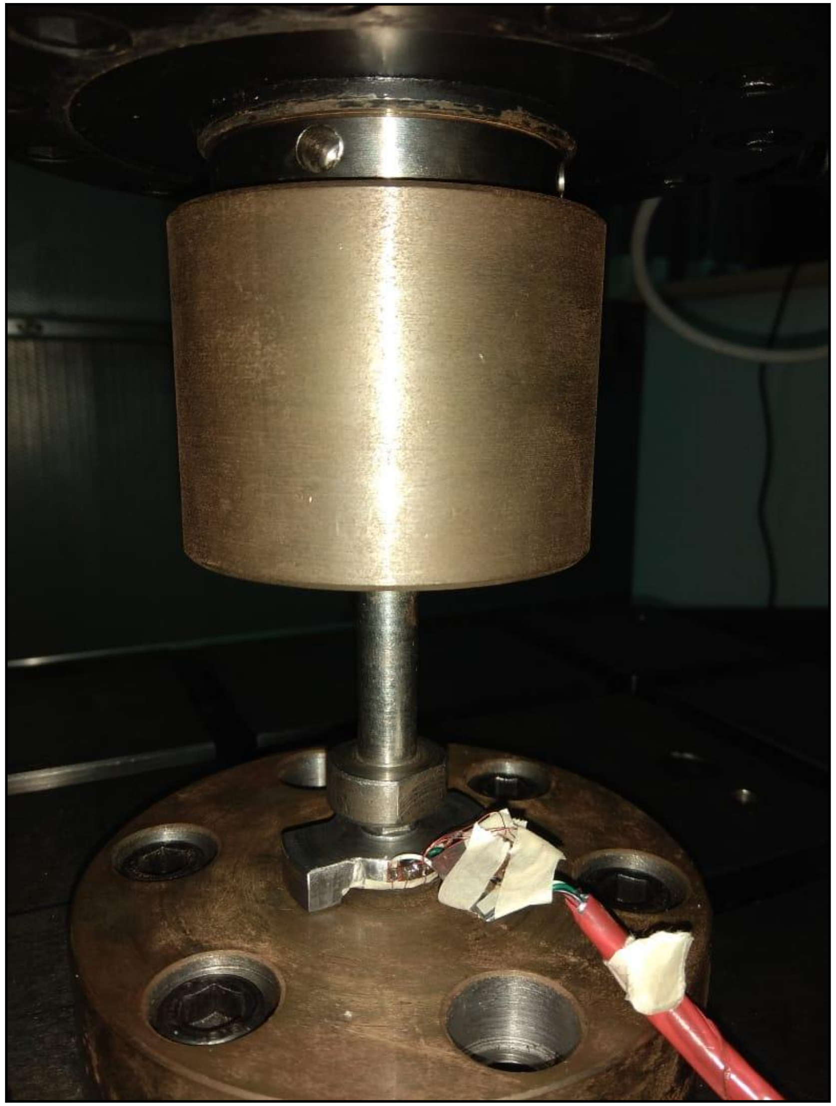


Fig. UTM load cell in contact with head of Flange M6 bolt for applying preload on the Load Washer

Strain Gauge Data & Filtering – M6

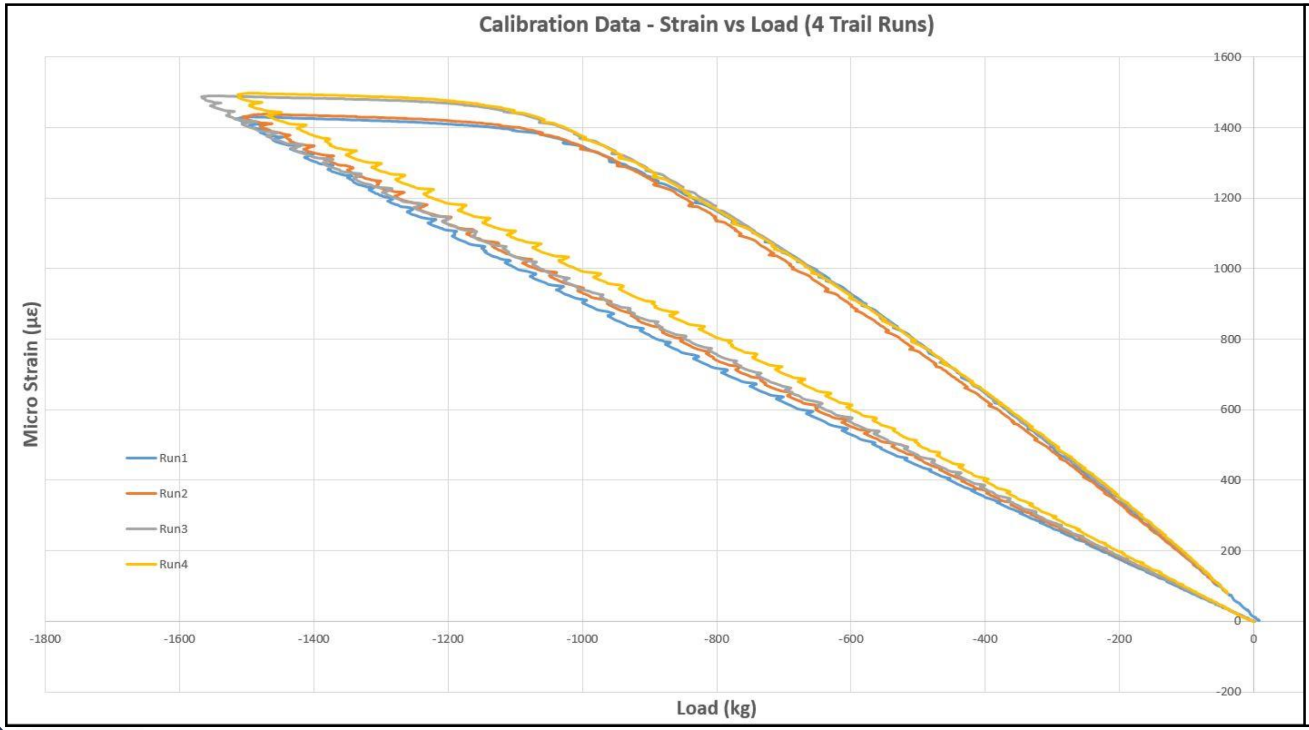


Fig. Complete cycle of loading and unloading of Load Washer with flange M6 bolt under UTM for a maximum load of 1500 kg

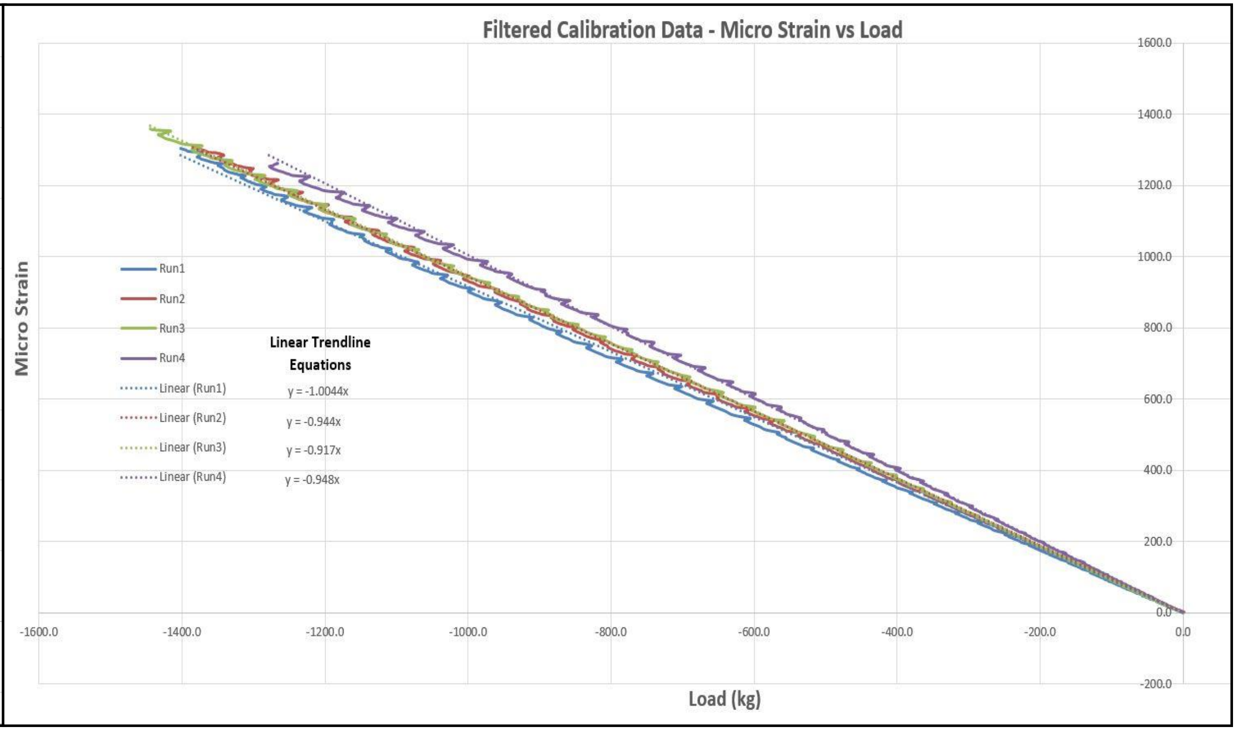


Fig. Filtered data for 4 runs and their linear fits

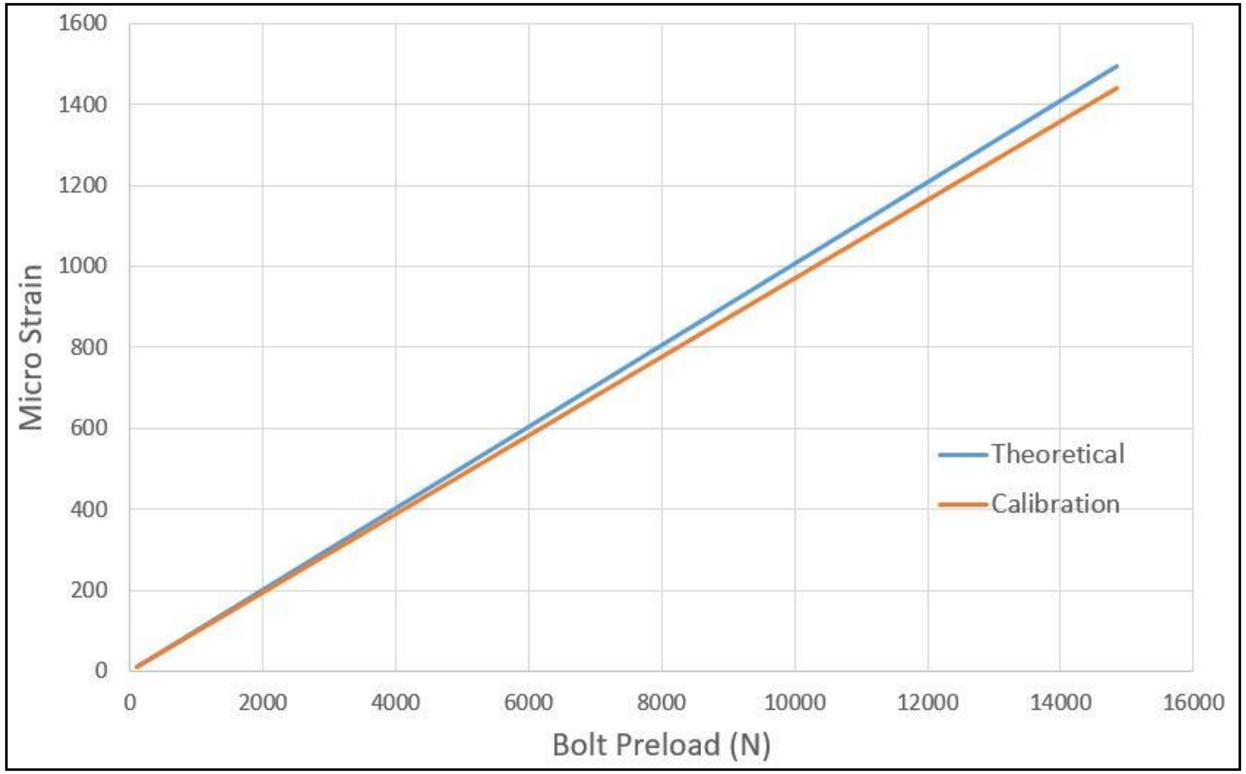
- Loading above 1400 kg and Unloading cycle data was cut off for accurate linear fit
- **Trendline** feature of Excel charts was used to estimate a linear for four trials with **zero intercept**

Validation of Theoretical Strain Calculations (Flange M6)

- Area moment of Inertia and height from NA were calculated with suitable approximations
- Since the load washer can be considered as beam with both hinged supports,

  Bending Moment $(M) = \left(\frac{P}{2}\right) * x$
- Final relation between strain and Bolt Preload is Linear with an equations as follows :

$$P = -\frac{2*E*I}{x*y} * \varepsilon \Rightarrow P(N) = -9.93882 * \mu\varepsilon$$



Theoretical Relation

$$Bolt\ Preload\ (N) = -9.93882 * MicroStrain$$

Average Linear fit from Calibration

$$Bolt\ Preload\ (N) = -10.29542 * MicroStrain$$

Strain Gauge Data & Filtering – M10

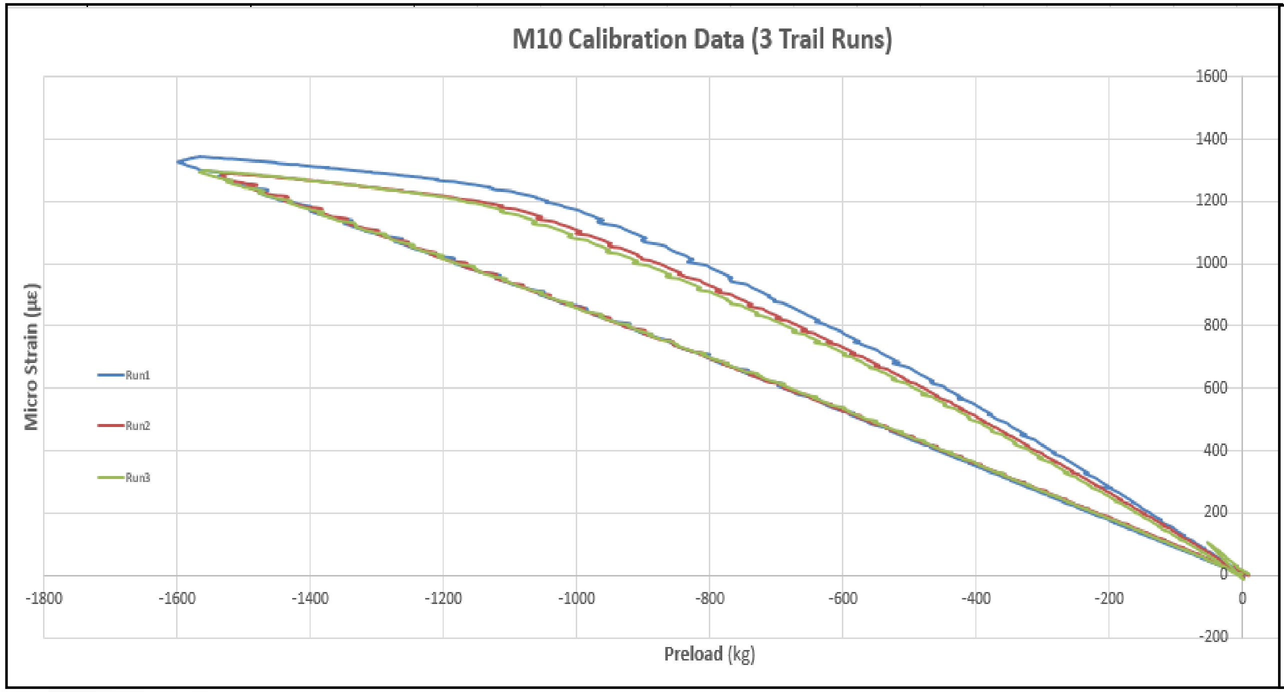


Fig. Complete cycle of loading and unloading of Load Washer with flange M10 bolt under UTM for a maximum load of 1500 kg

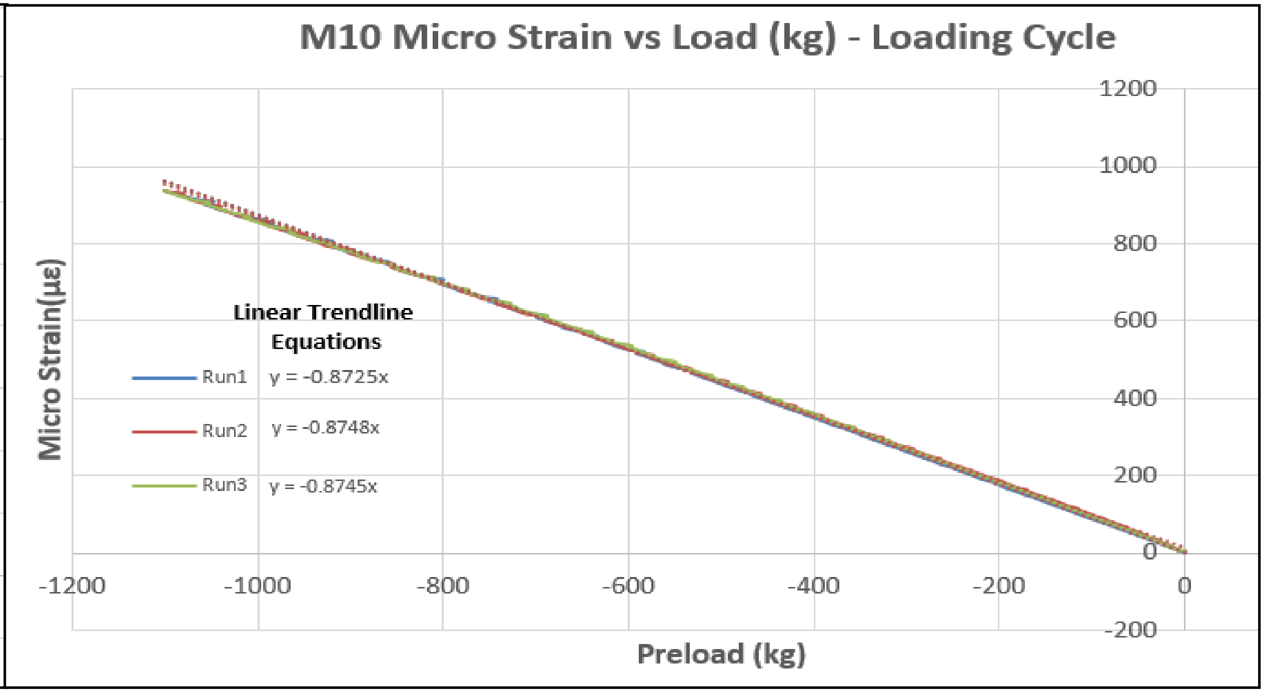


Fig. Filtered data for 3 runs and their linear fits

- Loading above 1100 kg and Unloading cycle data was cut off for accurate linear fit
- **Trendline** feature of Excel charts was used to estimate a linear for four trials with **zero intercept**

Averaging of the Calibration Data – M6 & M10

- Linear fits of all four test runs were developed with **zero intercept**
- Angle of inclinations for respective linear fits were found from slopes
- Slope of best fit line = tan(average of angle of inclinations)

M6

Trendline	1	2	3	4	Average
Slope	-1.0044	-0.944	-0.917	-0.948	-0.95285
Theta	-0.78759	-0.7566	-0.74213	-0.75871	-0.76126

Best Fit

Micro Strain = -0.95285*Load(kg)

$$\Rightarrow Bolt\ Preload\ (kg) = -1.04948 * MicroStrain$$

$$\Rightarrow Bolt\ Preload\ (N) = -10.29542 * MicroStrain$$

M10

Trendline No.	1	2	3	Avg.
Slope	-0.8725	-0.8748	-0.8745	-0.87393
Theta	-0.71741	-0.71872	-0.71855	-0.71823

Best Fit

Micro Strain = -0.87393*Load(kg)

$$\Rightarrow Bolt\ Preload\ (kg) = -1.14425 * MicroStrain$$

$$\Rightarrow Bolt\ Preload\ (N) = -11.22515 * MicroStrain$$

Full Assembly – 3D model

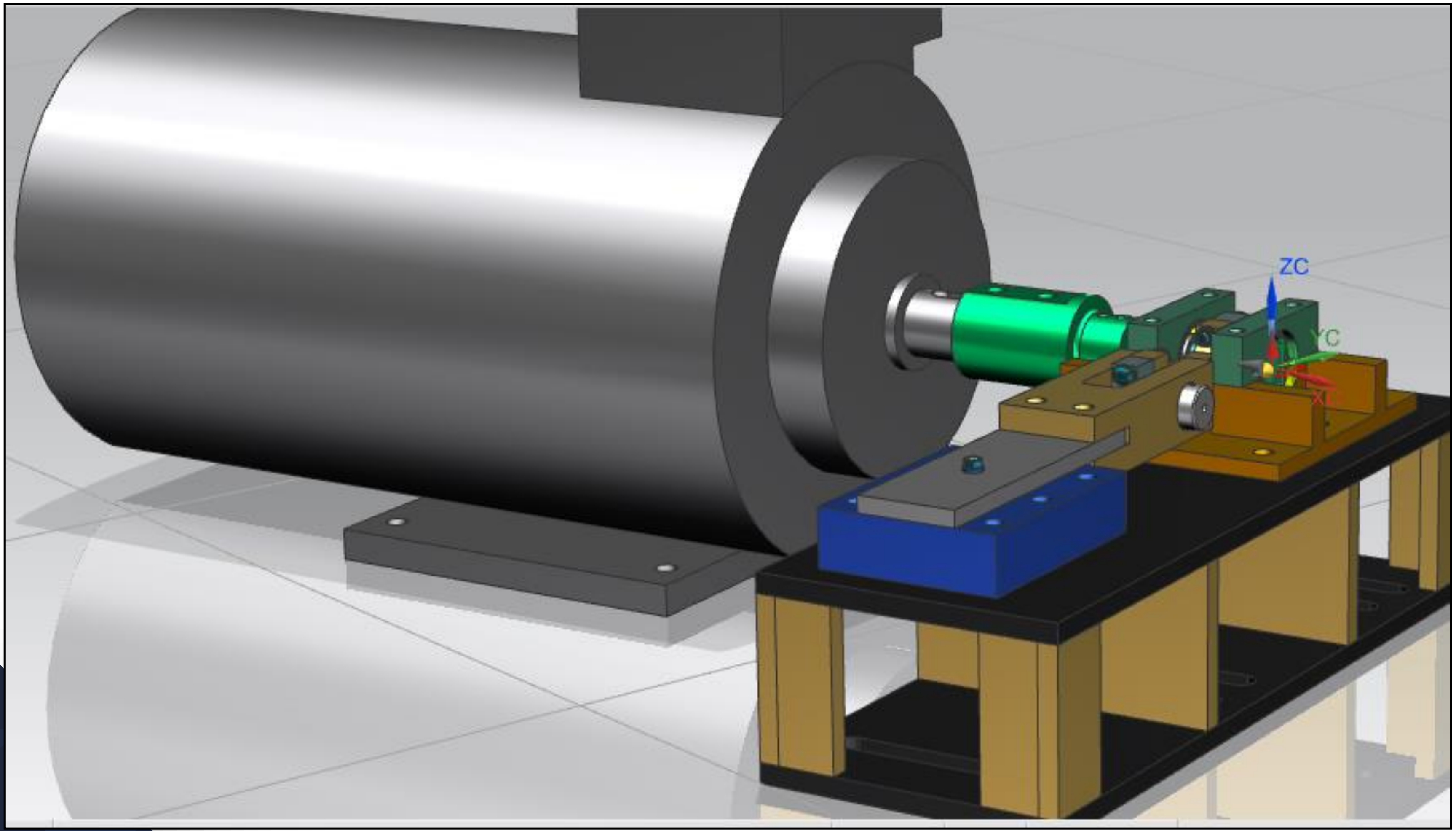
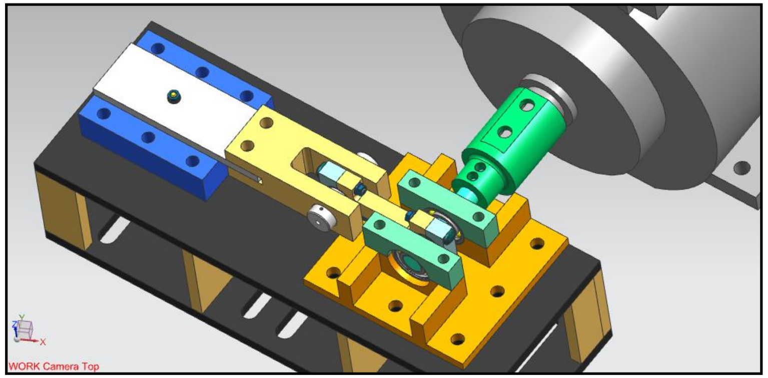


Fig. Isometric View of the complete assembly



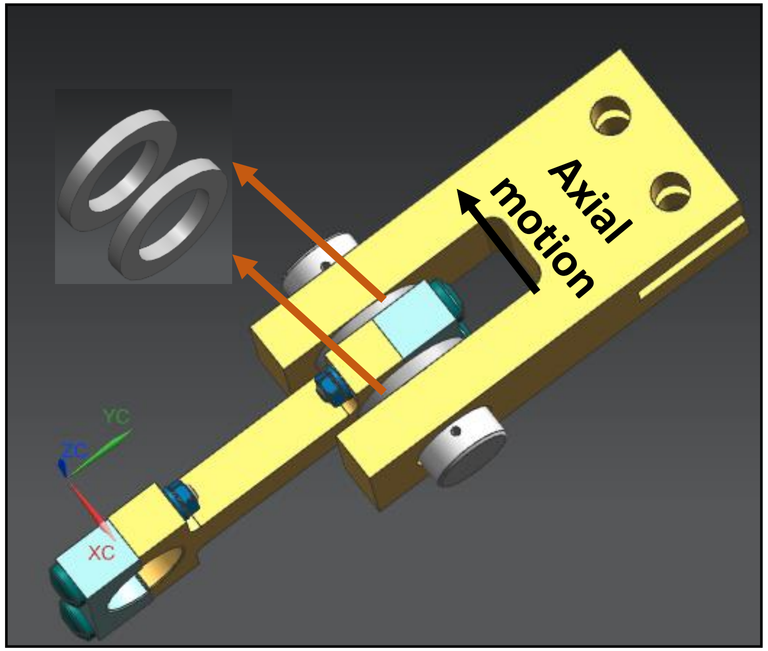
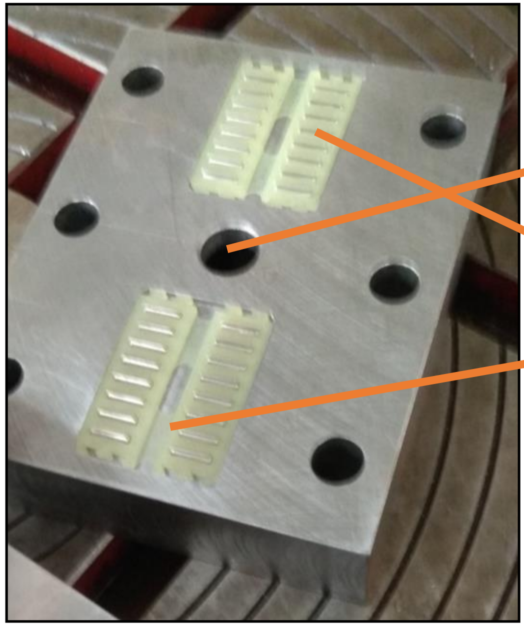


Fig. Custom Washers to lock axial motion of Conrod

Components and Manufactured parts

Fig. Fixture acts as a base plate and height riser for the setup



Bore for test specimen

Flat cage needle rollers

Fig. Stationary Plate with 2mm pockets for needle roller bearings

Fig. Needle roller and Split Bush with ID =26mm

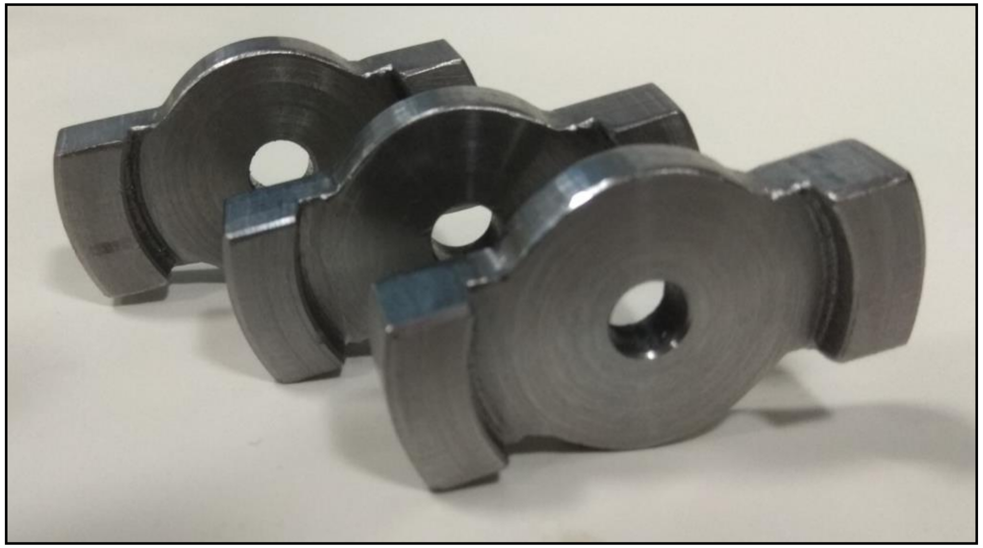


Fig. Load Washers with M6 bore

Full Assembly – Physical Test setup

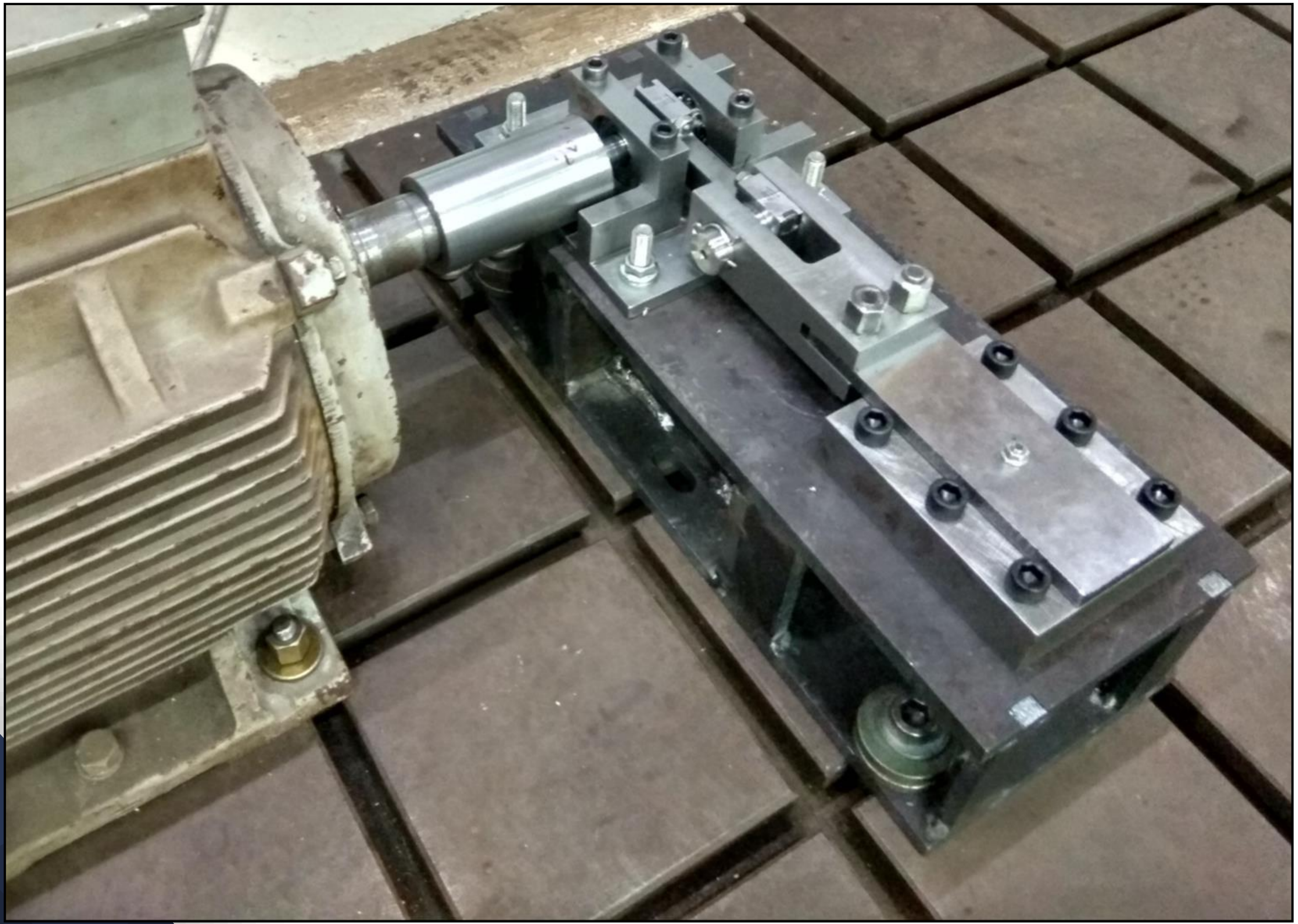


Fig. Complete test setup fixed to test bed (ground)

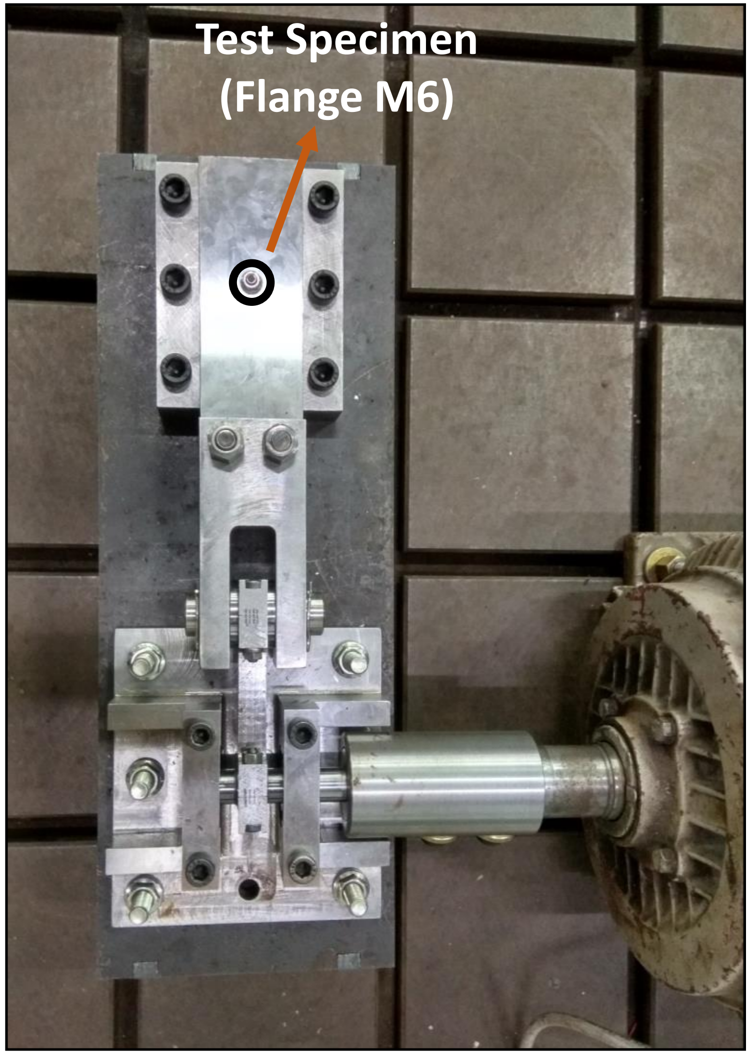
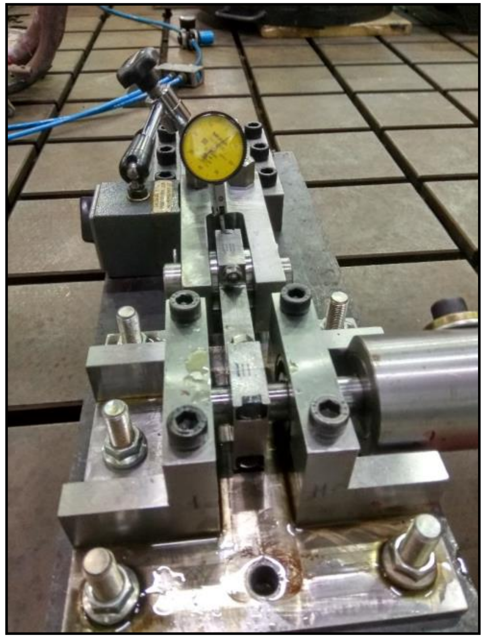
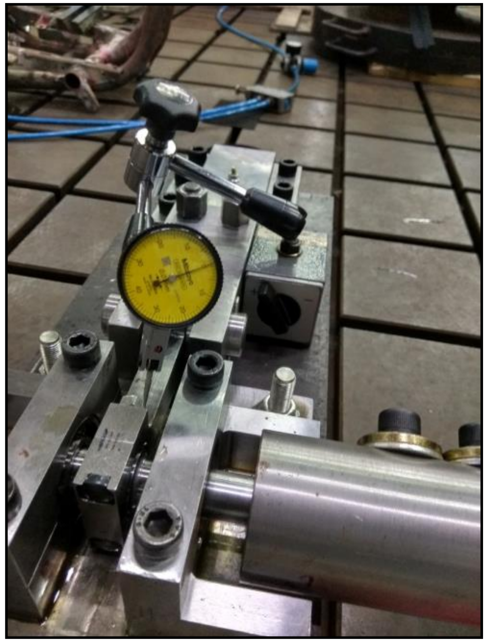


Fig. Top View

Losses in cross movement





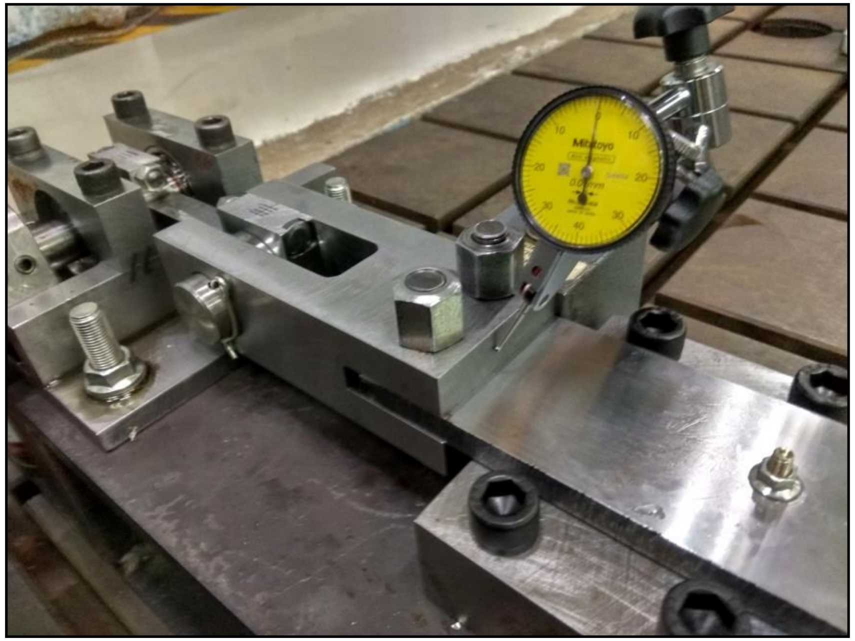


Fig. Checking cross movement at various location using dial gauge

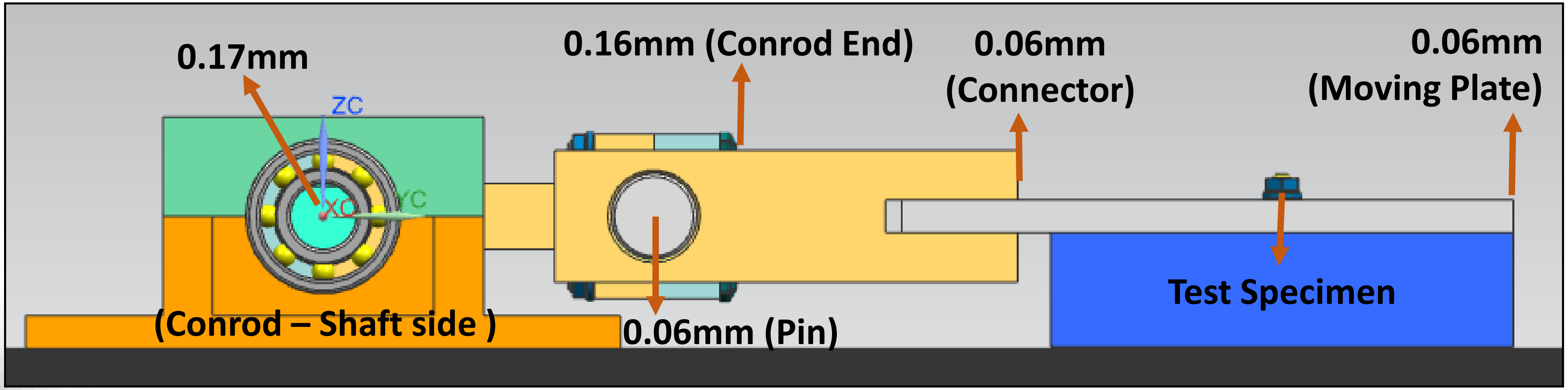


Fig. Measurements of cross movement at various locations in the assembly

Further Modifications

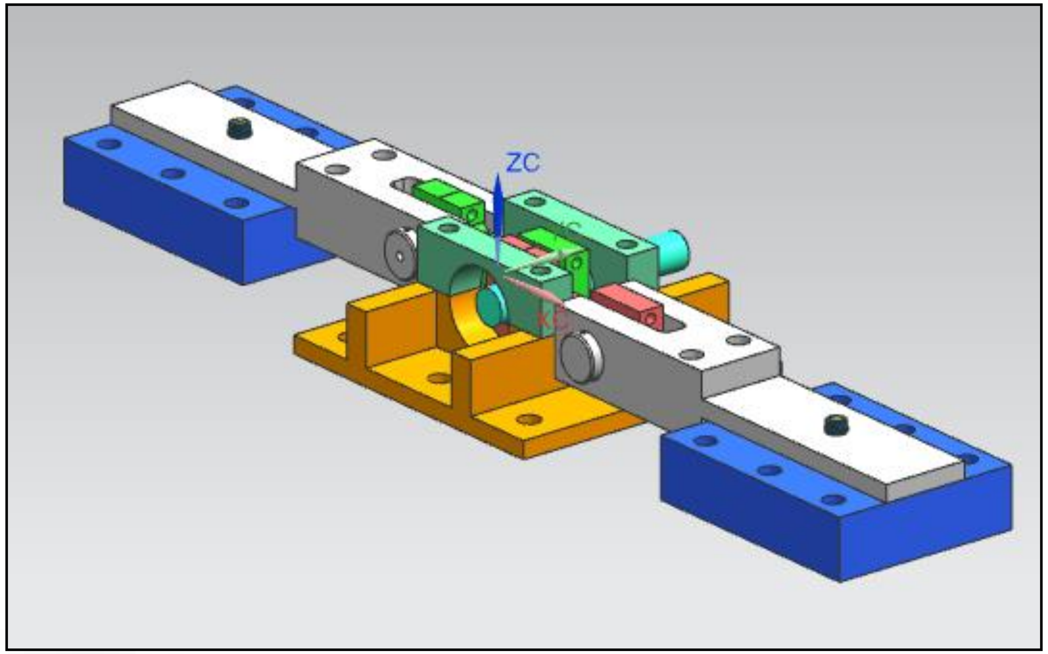
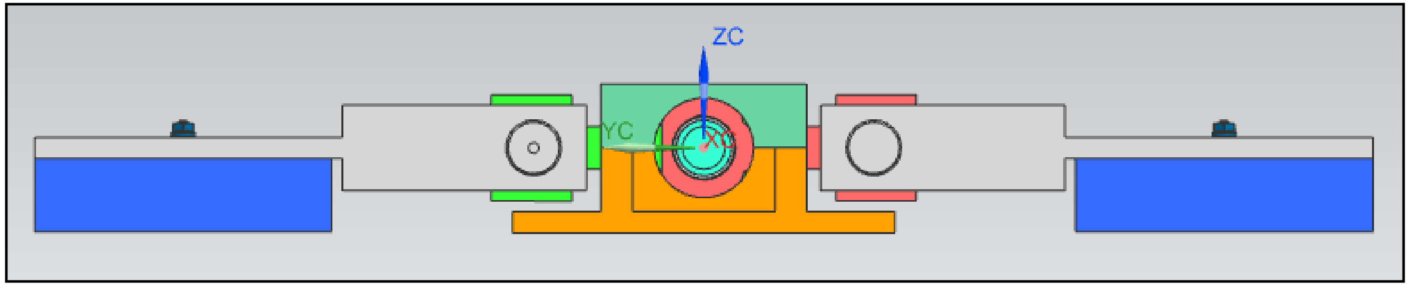


Fig. Isometric View of the assembly for testing 2 specimen simultaneously



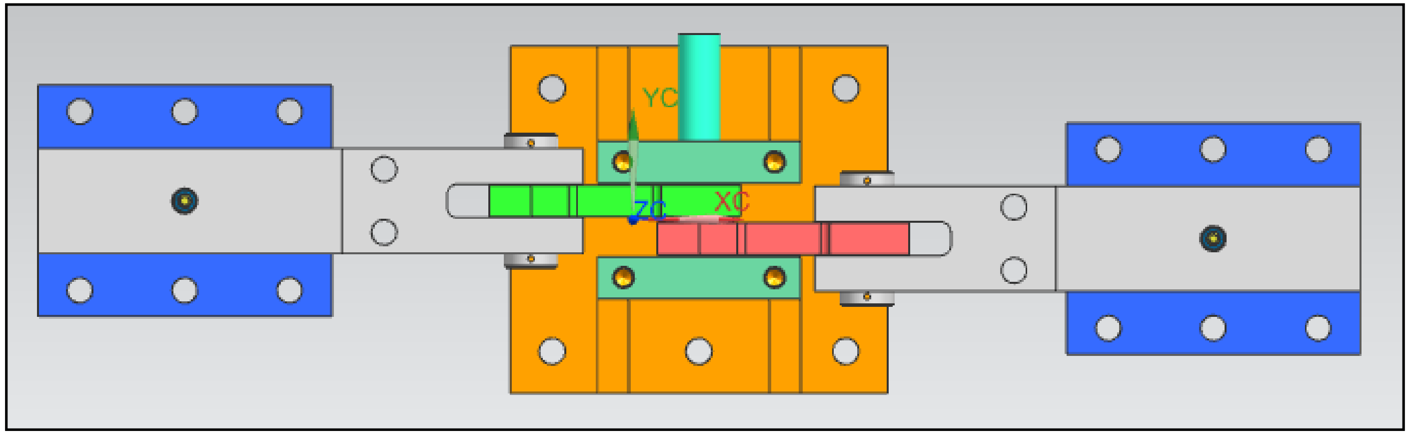


Fig. Side View and Top View of the complete assembly

- ❖ Minimized spaces between eccentric portions of the shaft and bearings supports to reduce the bending moment on the shaft
- ❖ Eliminated axial motion of Conrod without any additional washers
- ❖ Additional Constraining of the moving plate on both sides and top

Further Modifications

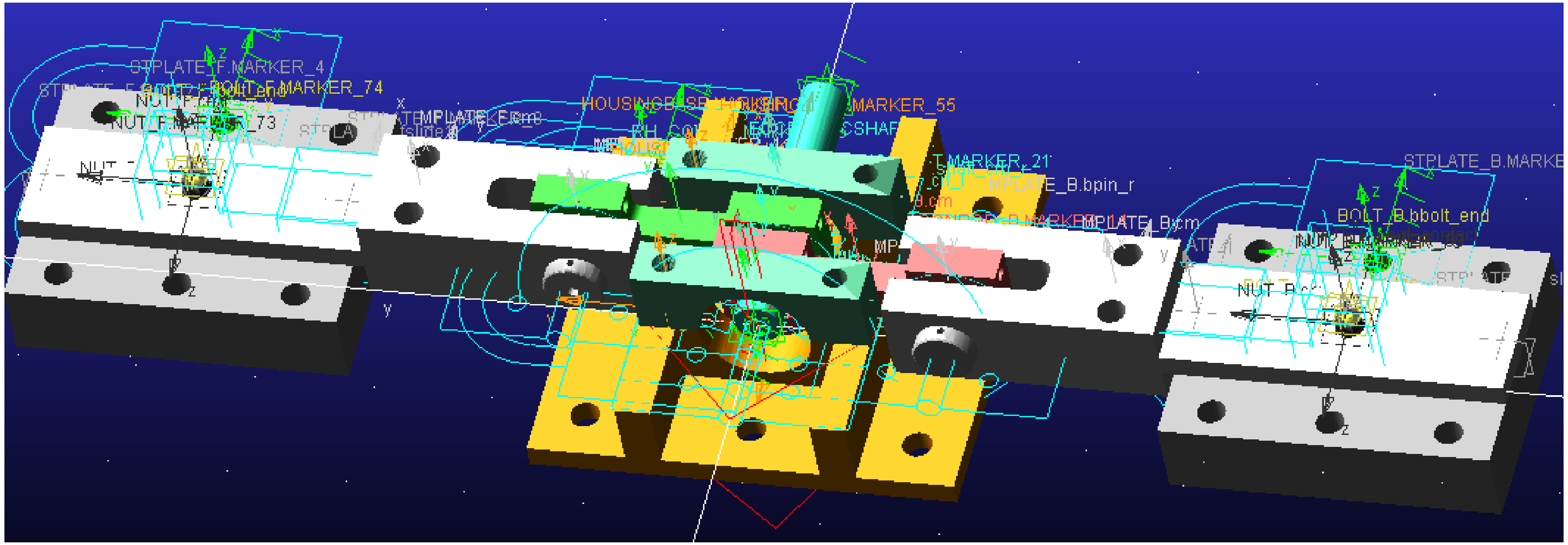
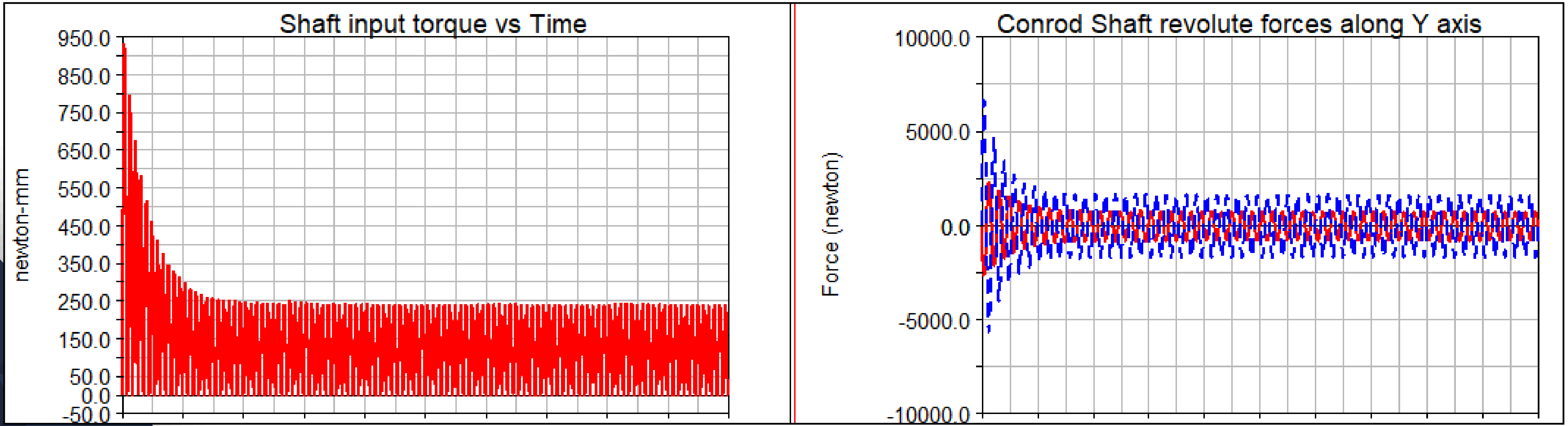


Fig. Modelling of Assembly for double testing in Adams

Fig. Input torque and Conrod Forces versus time



Key Learnings

- Theory of fastener Self loosening, Junker Test
- Different types of fasteners, locking techniques and their applications
- Review of all my DOME concepts & design calculations (shaft, Bolted joints, bearings)
- CAE software – Adams (Multi Body Simulation)
- CAE software – Unigraphics (NX) – 3D modelling
- Strain Gauging, Data Acquisition and Calibration techniques
- Basics of AC induction motor & their drives
- Manufacturing techniques and DFM – Dowelling, Laser Cutting, grinding

Thank You