A Case Study of an ISG Failure Using a Time Domain TVA

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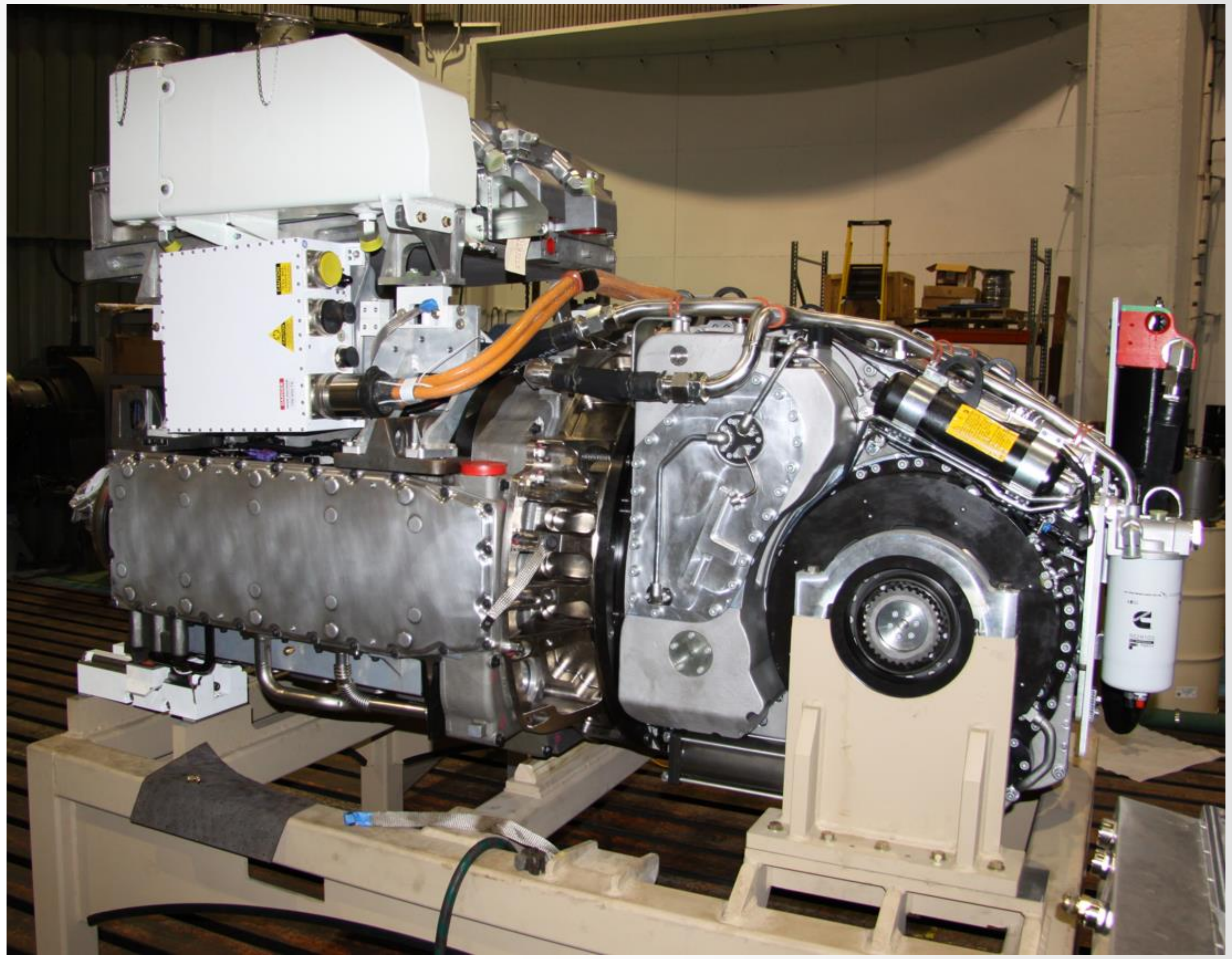
Application Description

- **Advanced Powertrain Demonstrator**
  - Engine
    - Opposed Piston Two Stoke Engine (OP2S)
      - Rated for 1000HP
      - Four Cylinders
      - Two crankshafts with combiner gearbox
  - Transmission
    - 32 Speed Binary Shift Steering Transmission
  - Integrated Starter Generator
    - 160kW Motor/Generator Unit
    - Sandwiched between engine and transmission
    - Doubles as the transmission bell housing
    - Used as the primary engine starter
    - Used to power vehicle's HV electrical bus



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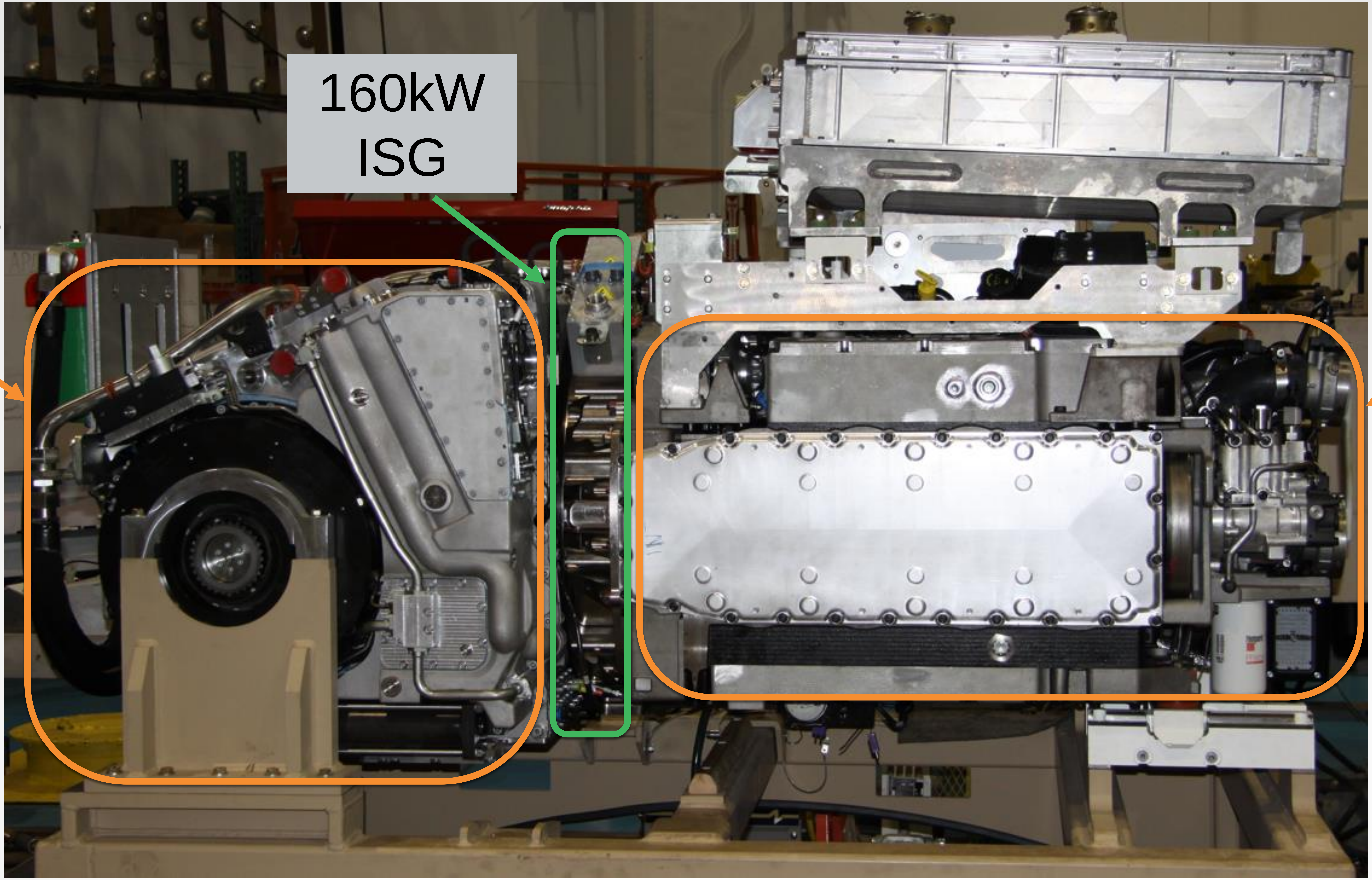


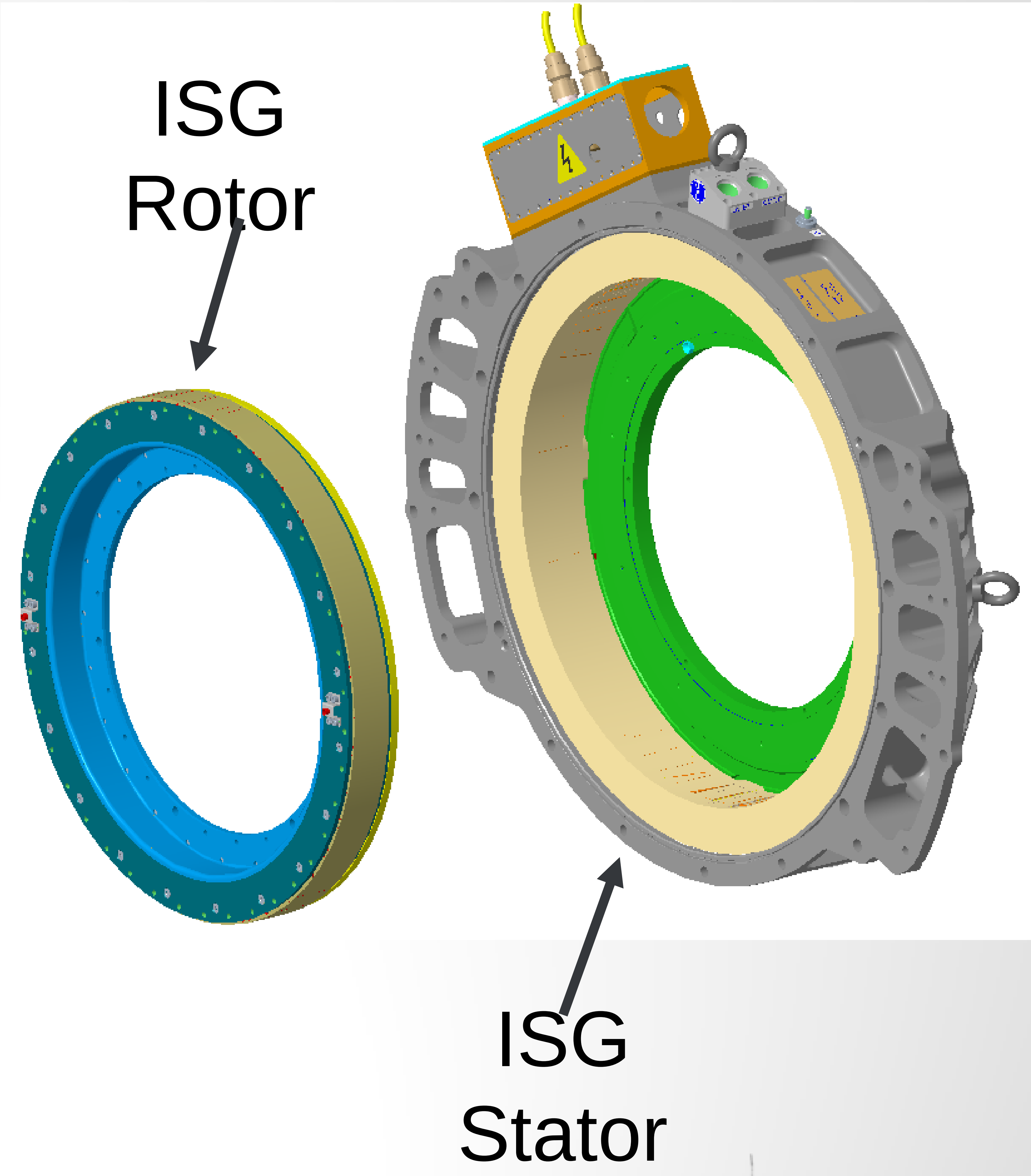
32 Speed Binary Shift Transmission

160kW ISG

OP2S Engine

Application Description

- **Integrated Starter Generator (ISG)**
  - The ISG has no bearings
  - The stator is fixed to the transmission
    - Structurally ties engine to transmission
  - The rotor is supported by the transmission input shaft bearings
- **Torsional Coupling**
  - A Geislinger torsional coupling is packaged inside the rotor space claim
  - The Geislinger couples the engine output shaft to the transmission input shaft
  - The ISG rotor is downstream of the Geislinger
    - The ISG rotor is not directly tied to the engine shaft



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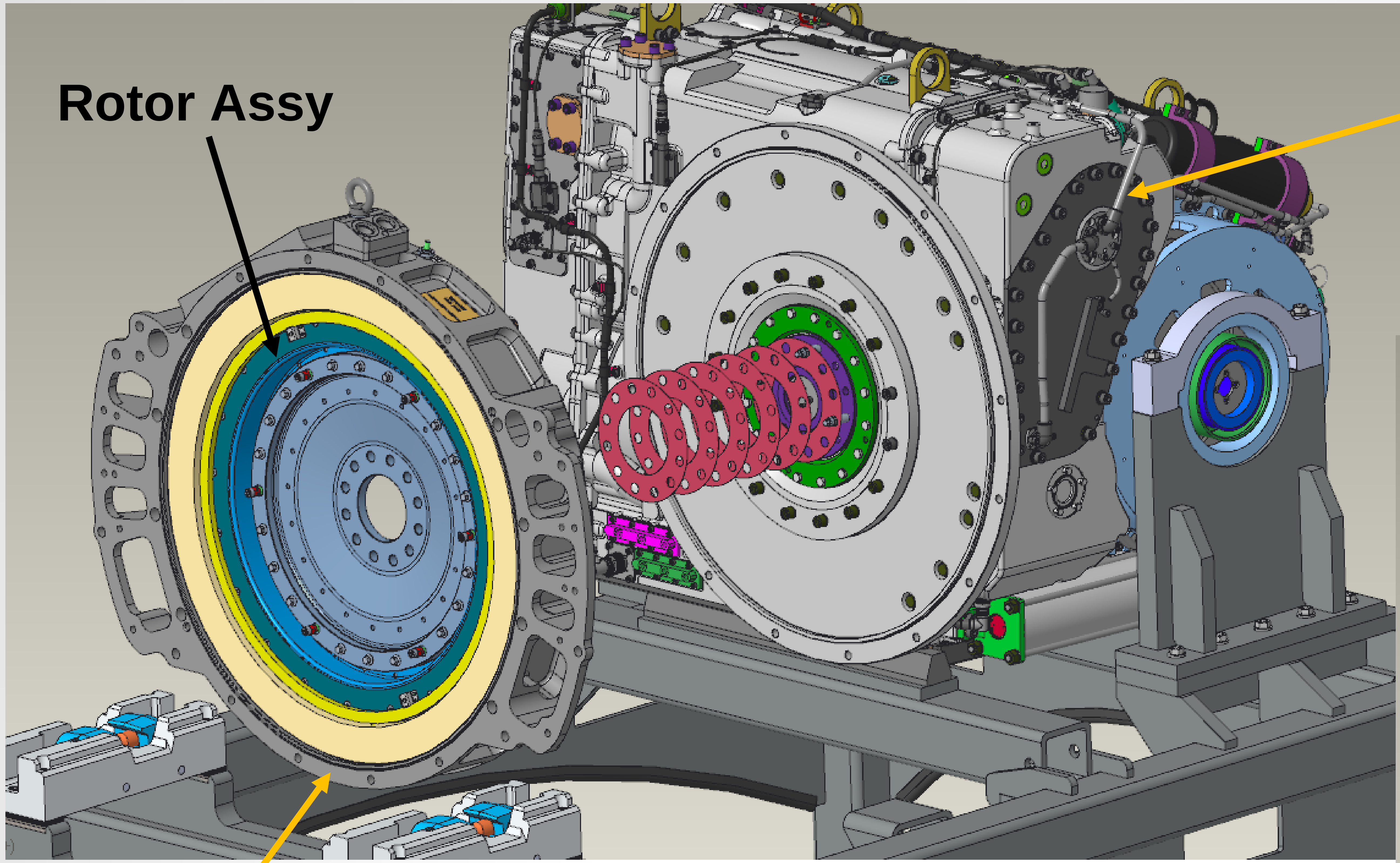
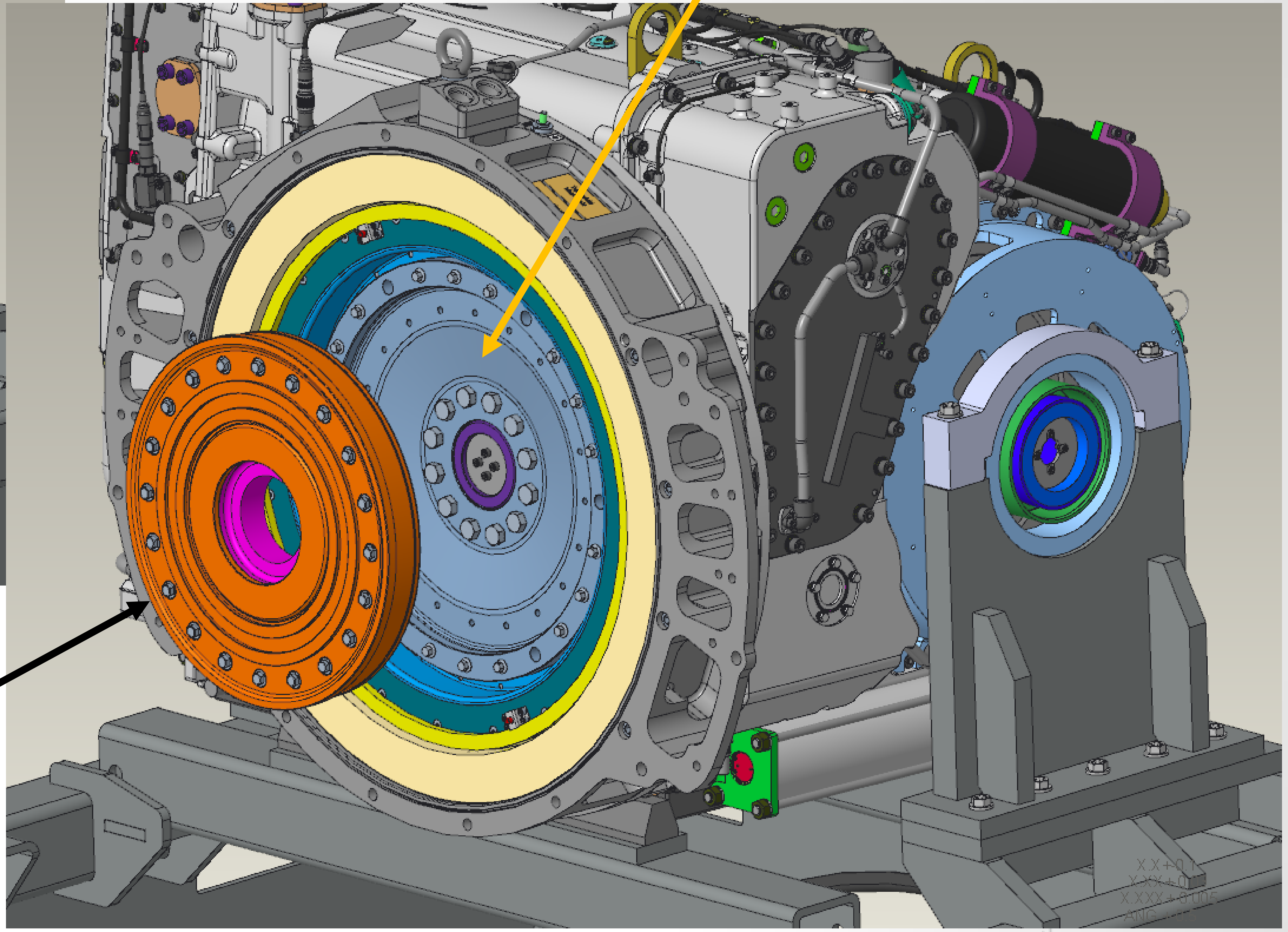


Rotor Assy

Stator and Housing Assy

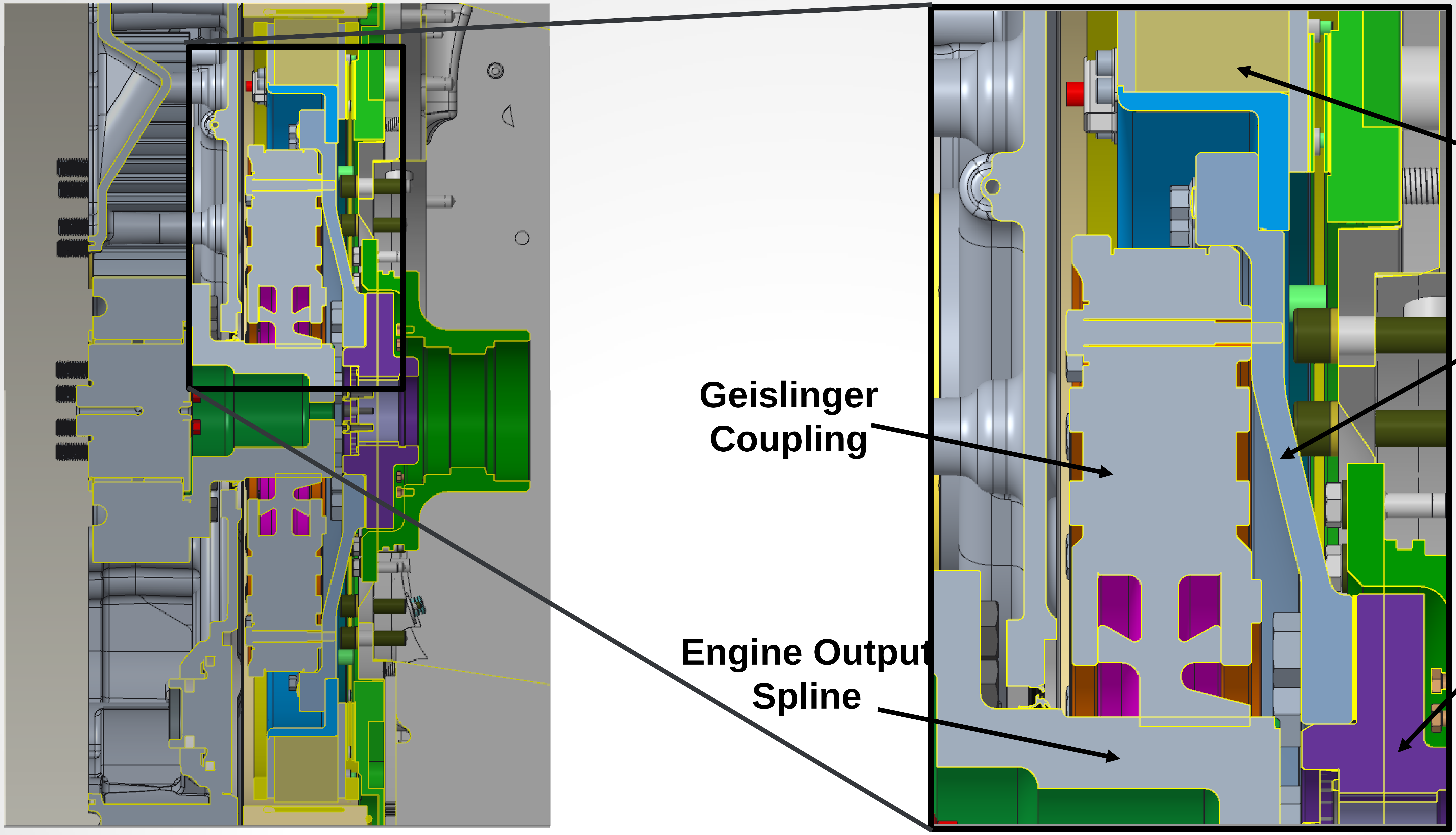


Transmission

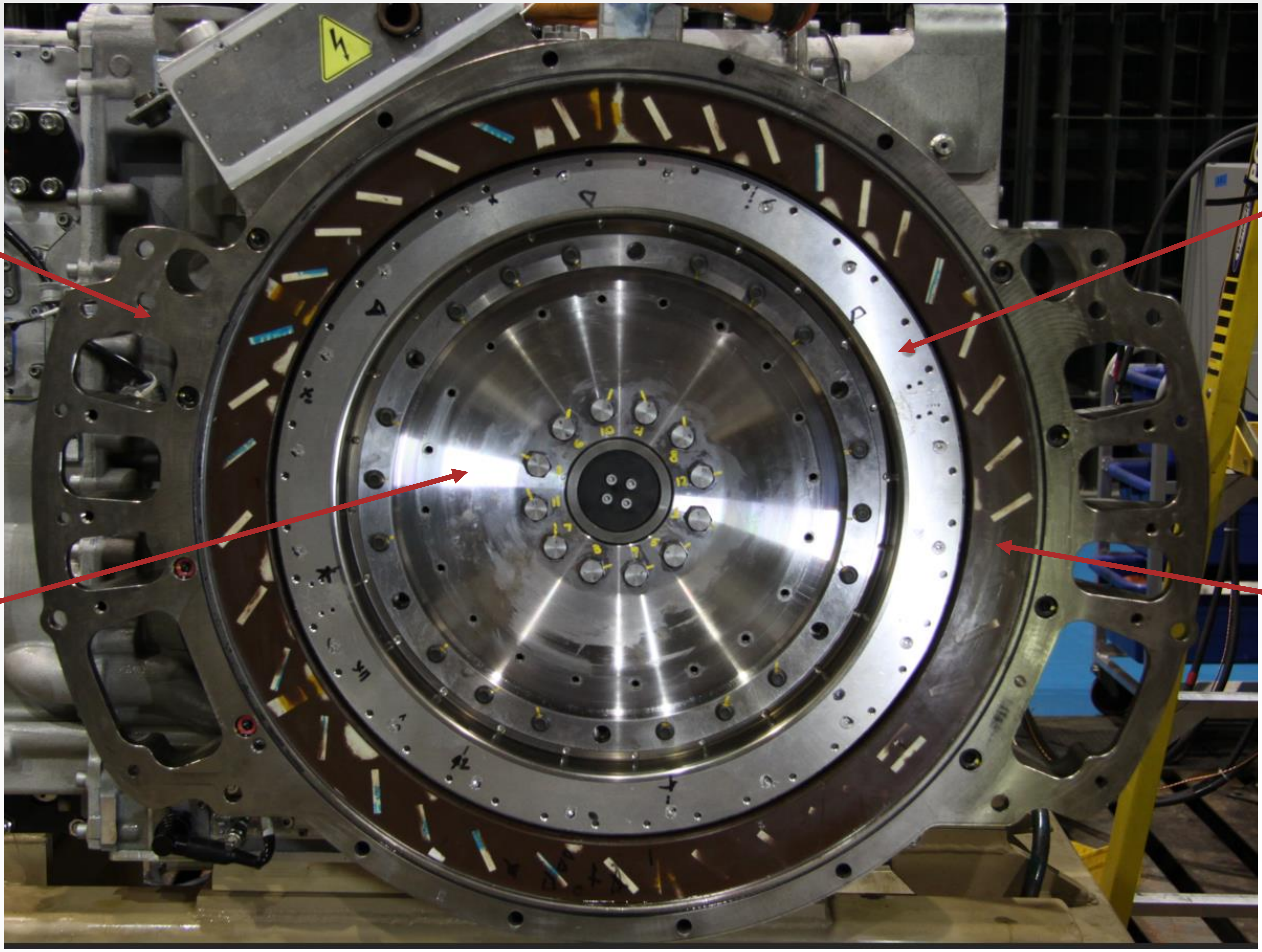
Rotor Carrier Coupling

Geislinger Coupling

Application Description



Application Description



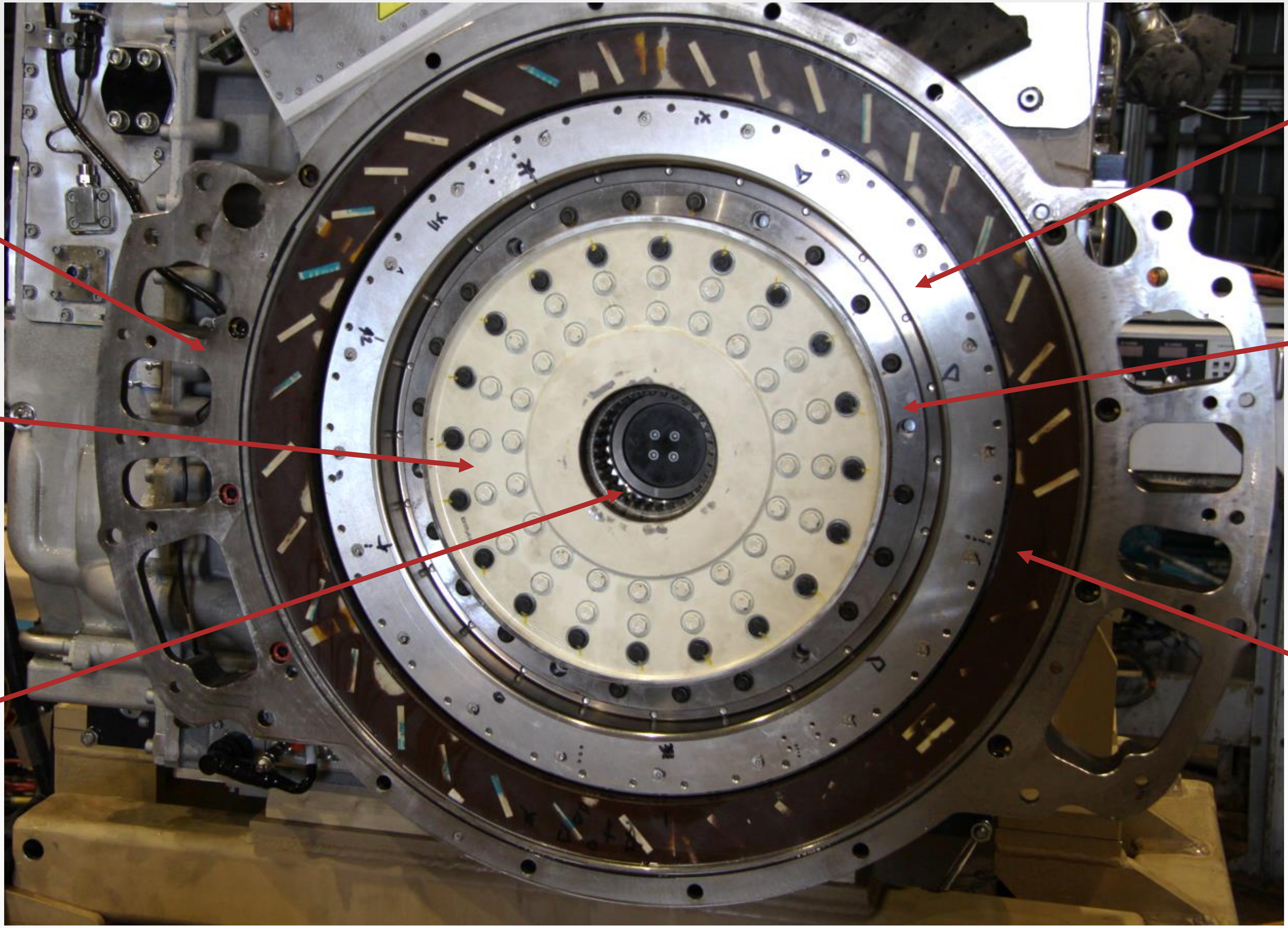
Stator Housing

ISG Rotor

Rotor Carrier Coupling

Stator Windings

Application Description



Stator Housing

Geislinger Coupling

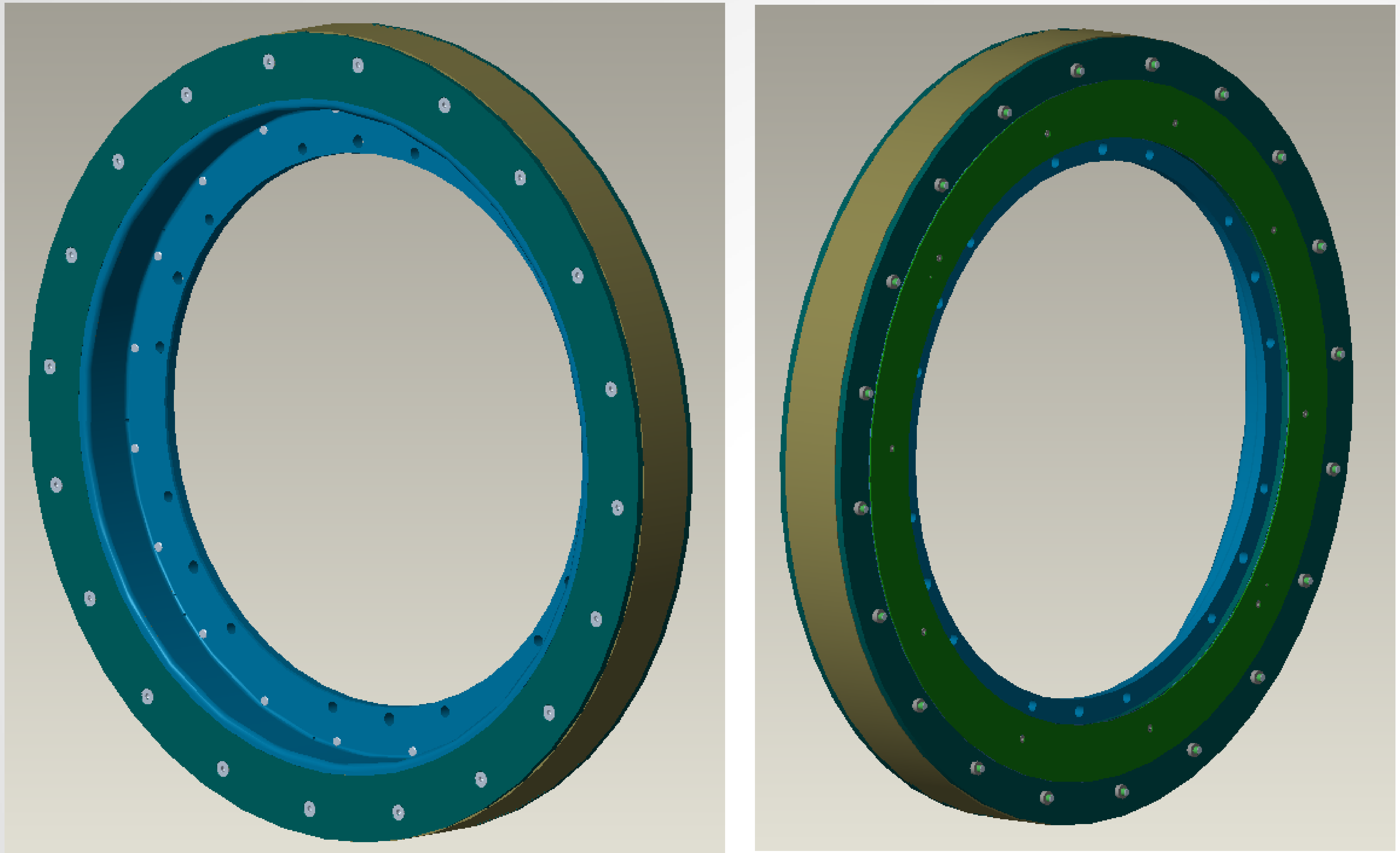
Spline Connection to Engine

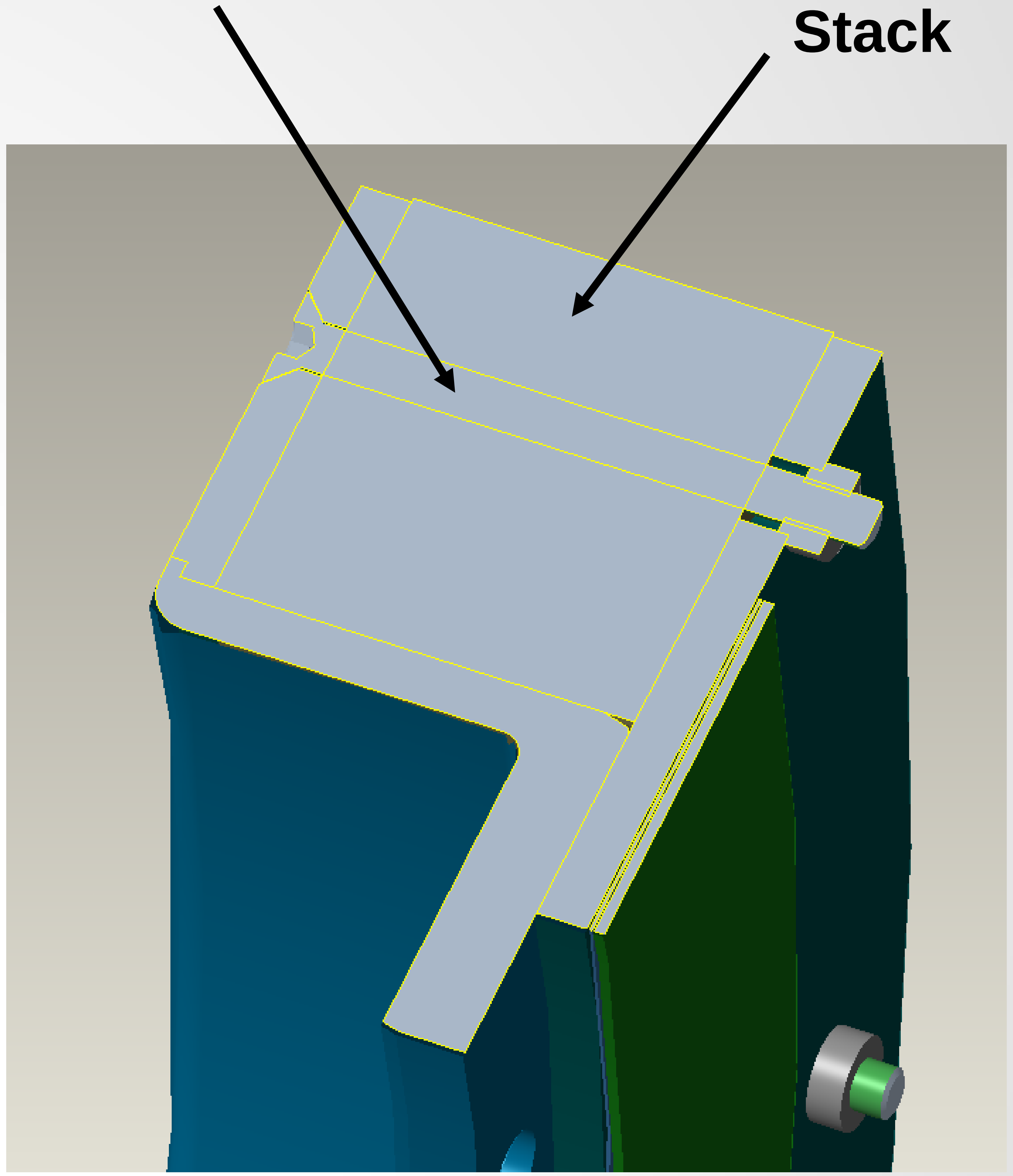
ISG Rotor

Rotor Carrier Coupling

Stator Windings

ISG Rotor Construction



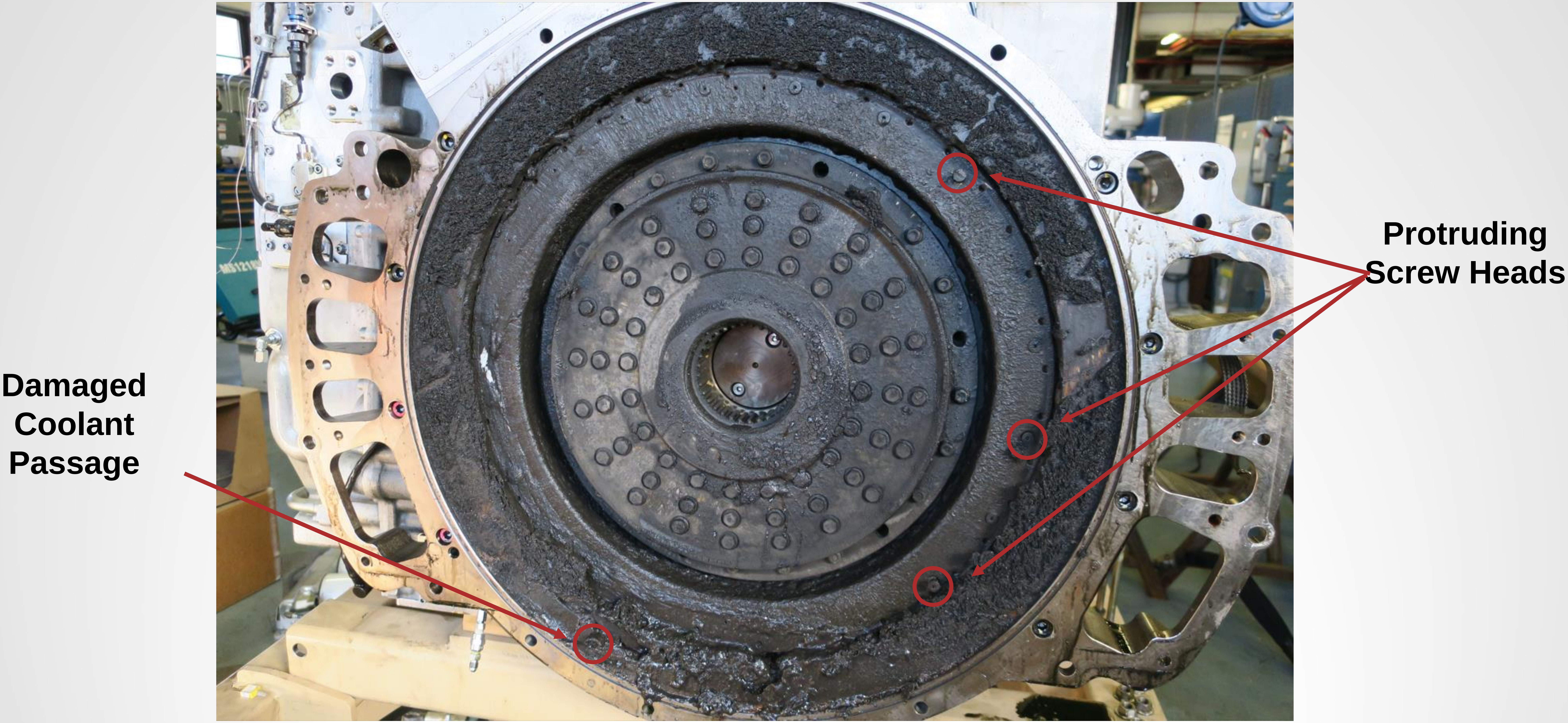


Failure Description

- **Loss of Resolver Signal**
  - Resulted in loss of inverter functionality
  - The ISG stayed in the powerpack as a passive device
- **Elevated Stator RTD Readings**
  - Several RTD's began indicating excessive winding temperatures even though the inverter was offline
- **Coolant Leak**
  - The air space surrounding the rotor and Geislinger coupling is sealed for powerpack submersion
  - A large volume of coolant was drained from this sealed air space

- **The powerpack was disassembled for ISG inspection**

Failure Description



Damaged Coolant Passage

Protruding Screw Heads

Failure Description

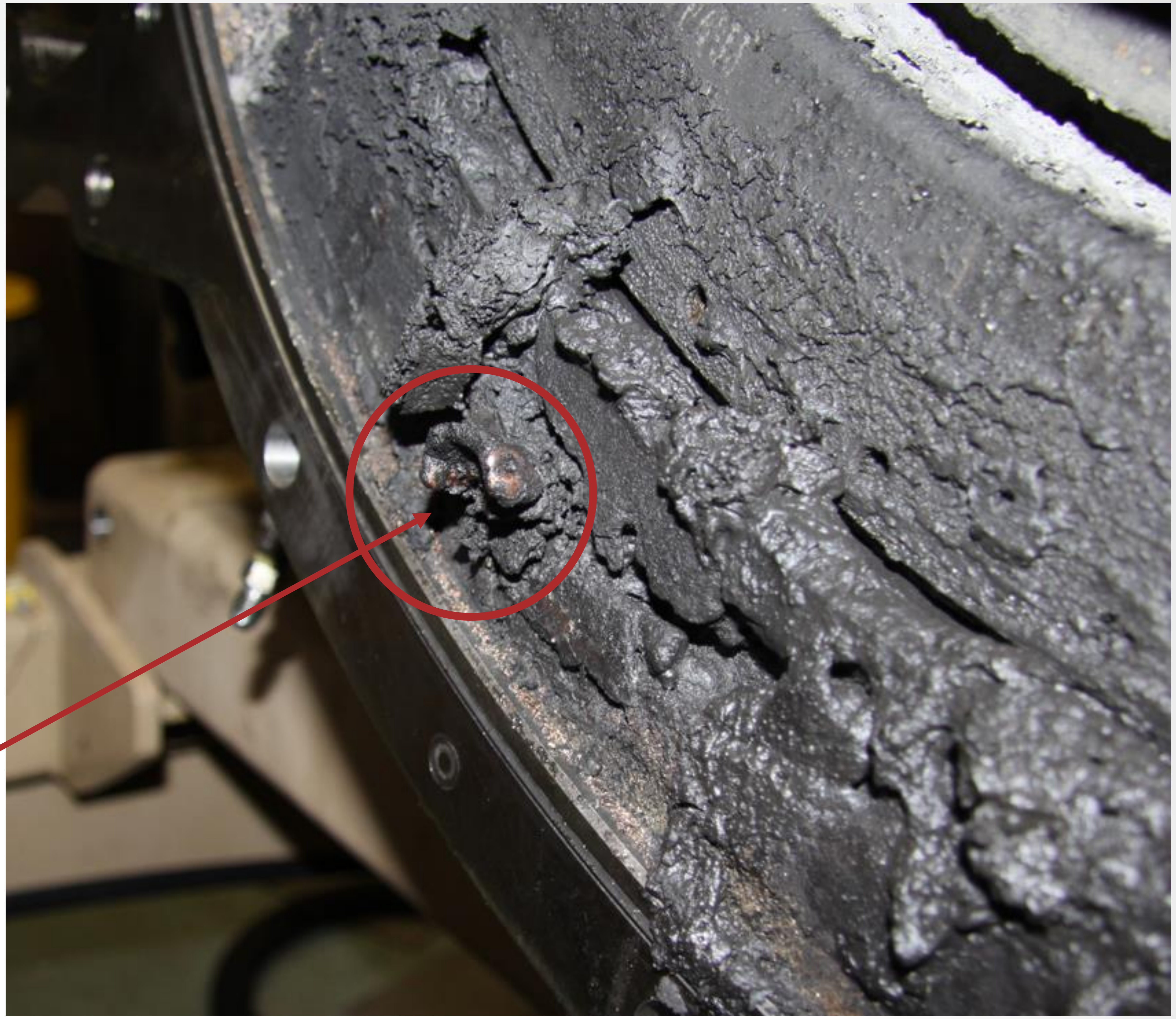
Protruding Bolt Heads

Nuts and Broken Bolts found in Debris

19 of 20 Rotor thru Bolts were found broken

Remaining thru Bolt was Loose

Failure Description



Damaged Coolant Passage

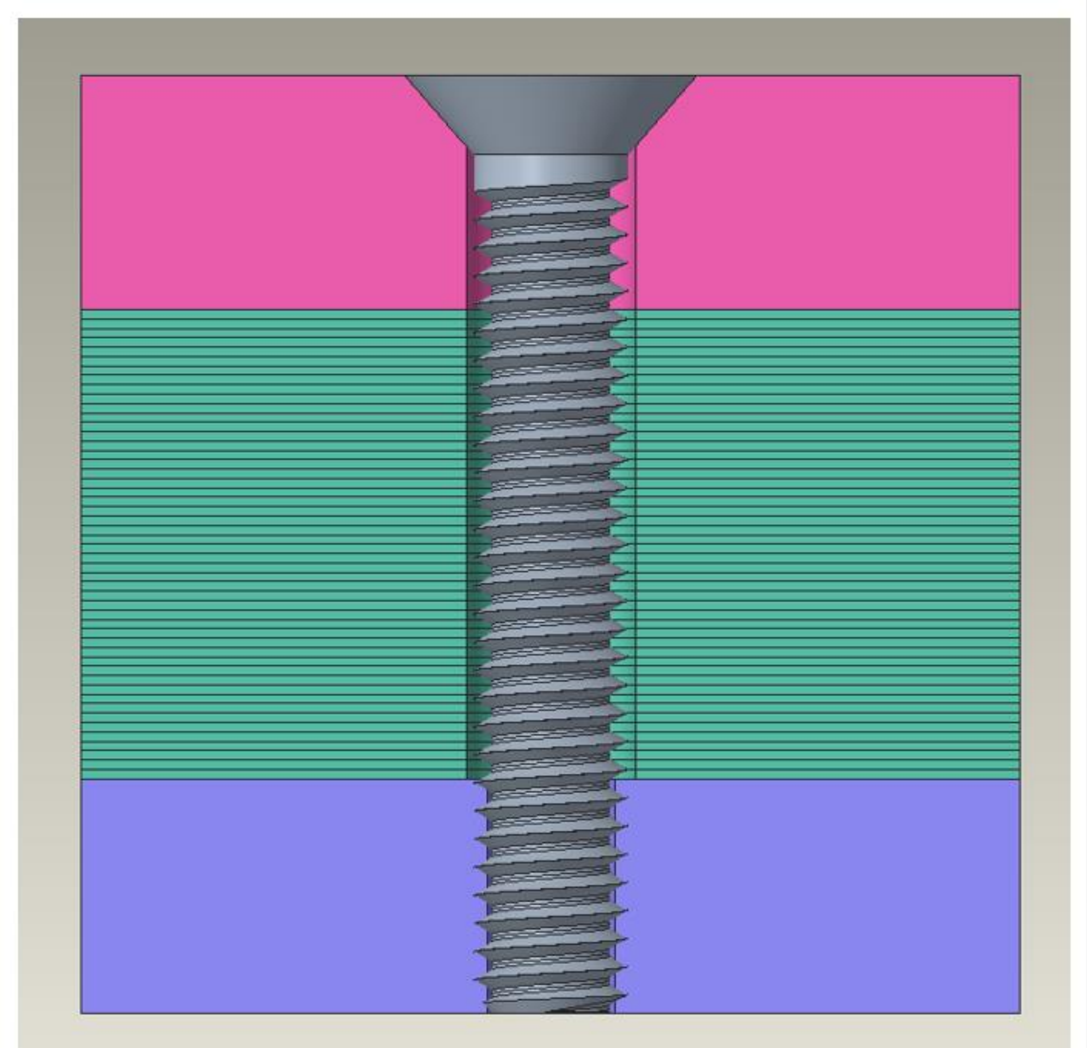
Failure Description

- **The bolted joint holding the rotor laminations together failed**

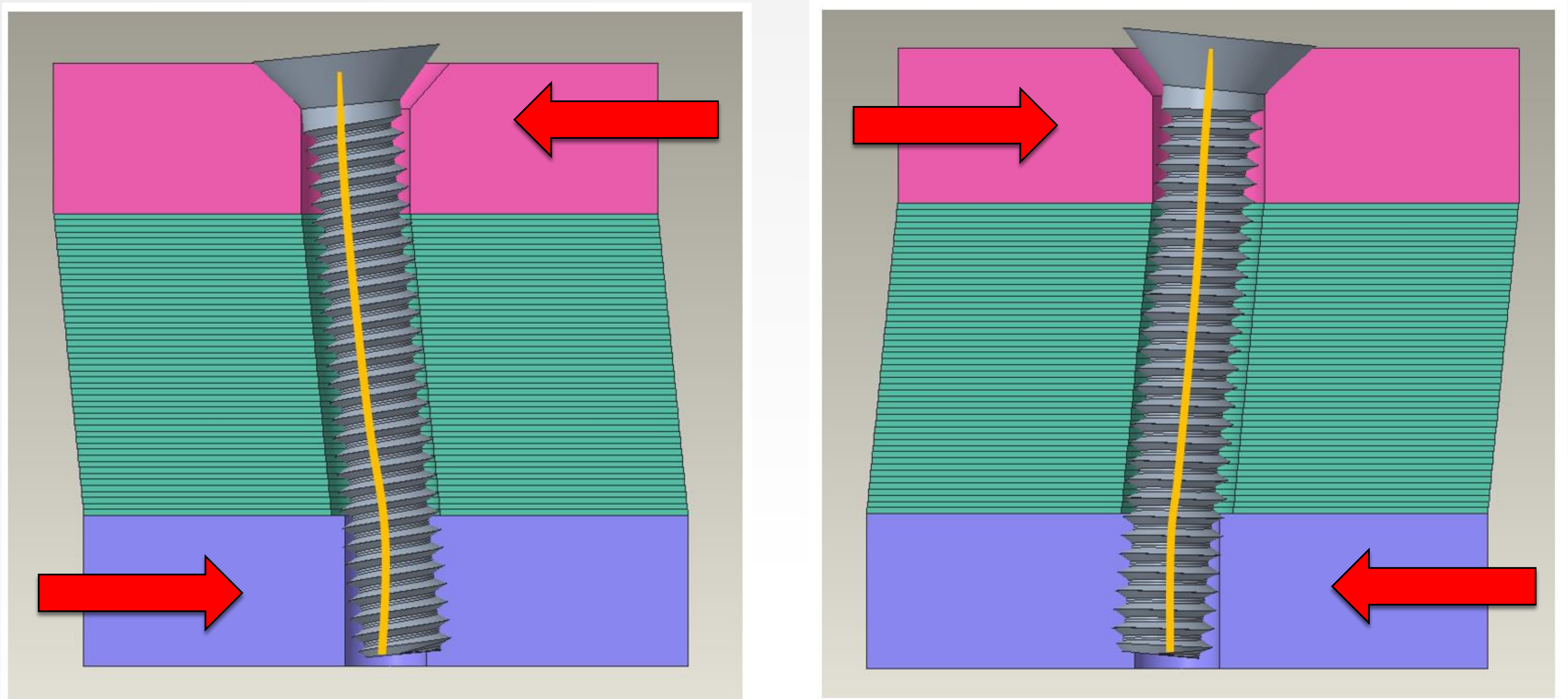
- **The loose screws and nuts became trapped between the rotor and the stator**
  - This led to broken coolant passages and electrical damage to the resolver and stator windings
  - Shorted stator windings led to a winding over temperature condition
  - Coolant leaking into the ISG air space was churned by the spinning rotor creating more heat
  - Damage to the coolant passages led to a loss of ISG cooling

- **Torsional vibrations became prime suspect since the ISG was tested extensively at the supplier under full load without any issues**

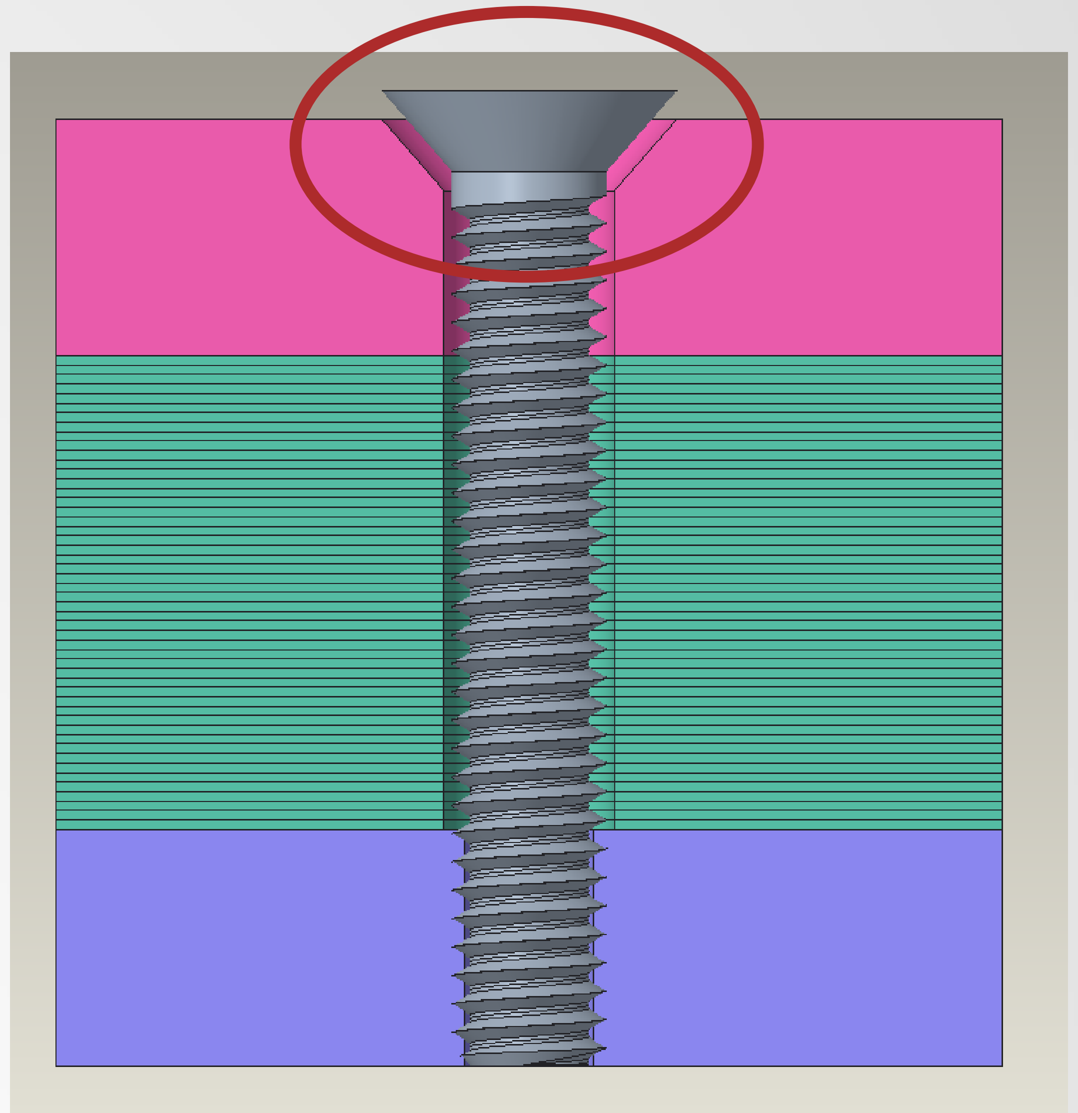
Bolted Joint Failure Mechanism



Initial Unstressed Joint



Joint Exposed to Alternating Loads



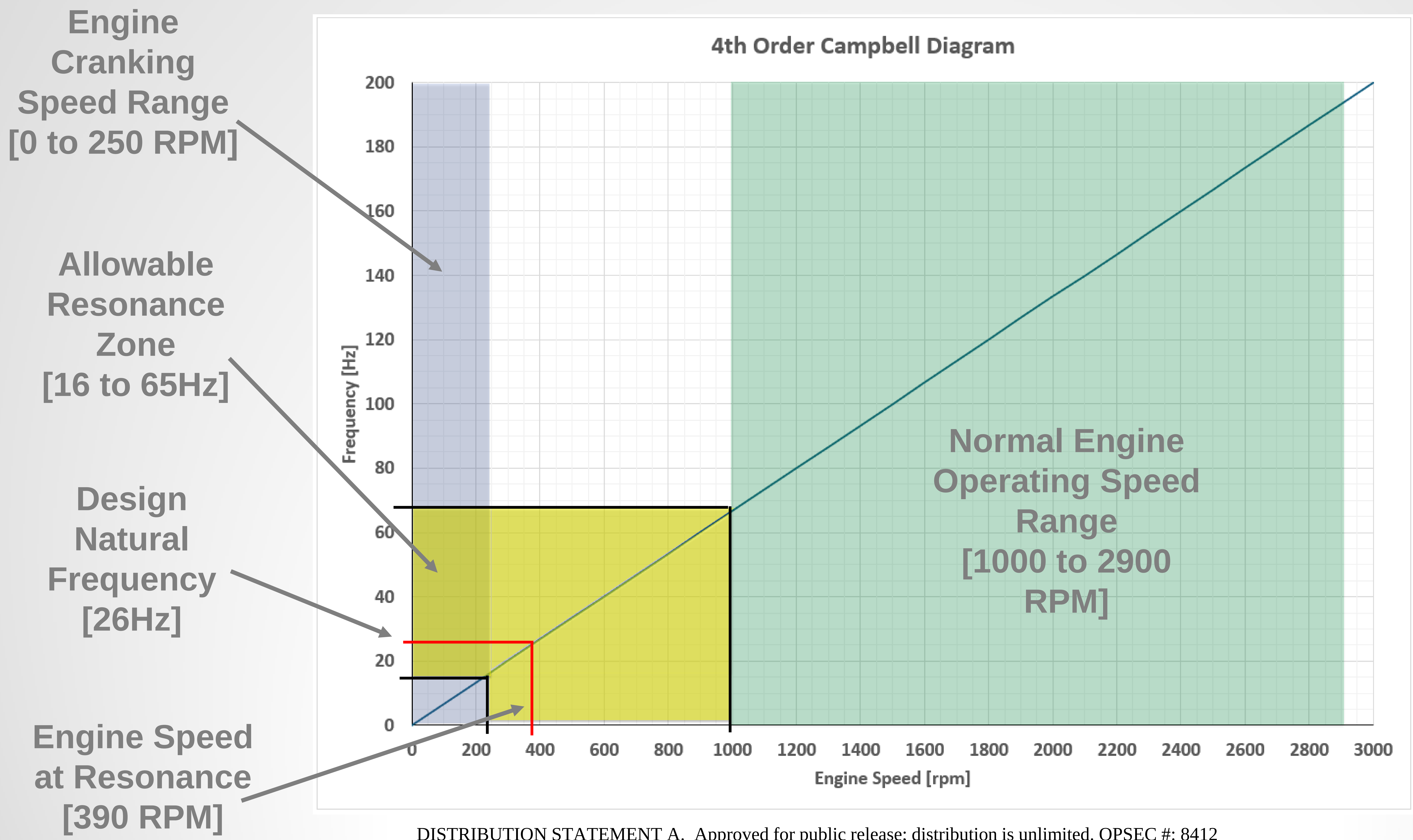
Stretched Bolt

- Alternating loads exceed bolted joint friction allowing for joint slippage
- Joint slippage causes bolt deformation which results in a total loss of joint clamp force
- The loose bolt is continuously hammered by the alternating forces until it breaks

Pre-Failure TVA

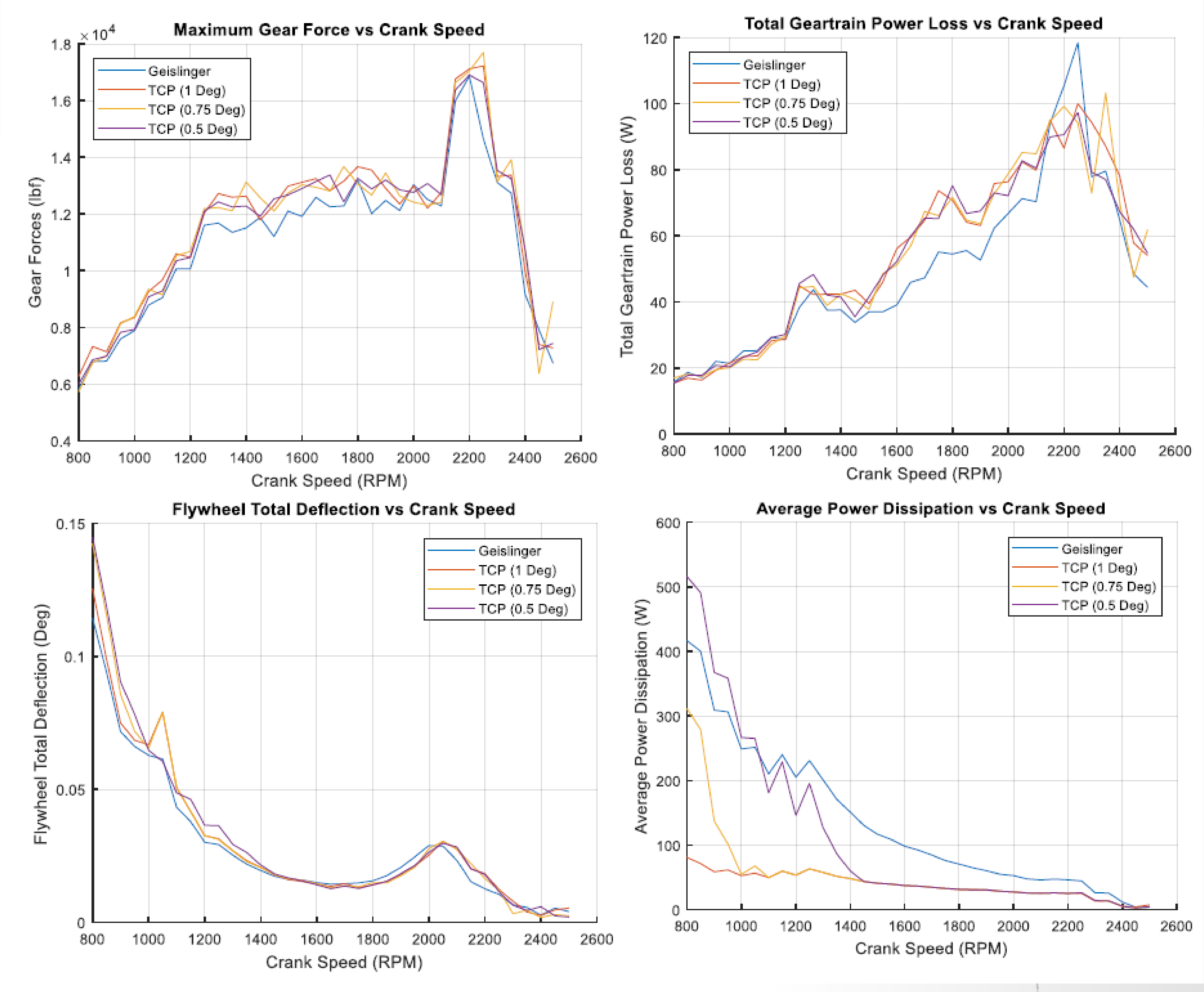
- **Selection of Natural Frequency**
  - The spring rate of the Geislinger Coupling was selected to ensure that 4^{th} order resonance was outside of the normal engine operating range
- **Crankshaft Integrity**
  - A frequency domain TVA was conducted to ensure that the crankshaft was safe from a fatigue standpoint
  - This frequency domain TVA was focused on normal engine operating speeds
- **Traditional design philosophy was followed**
  - Place the natural frequency of the driveline between engine cranking and engine idle speeds
  - The time spent accelerating from cranking speed to idle speed is too short to cause any real damage

Pre-Failure TVA



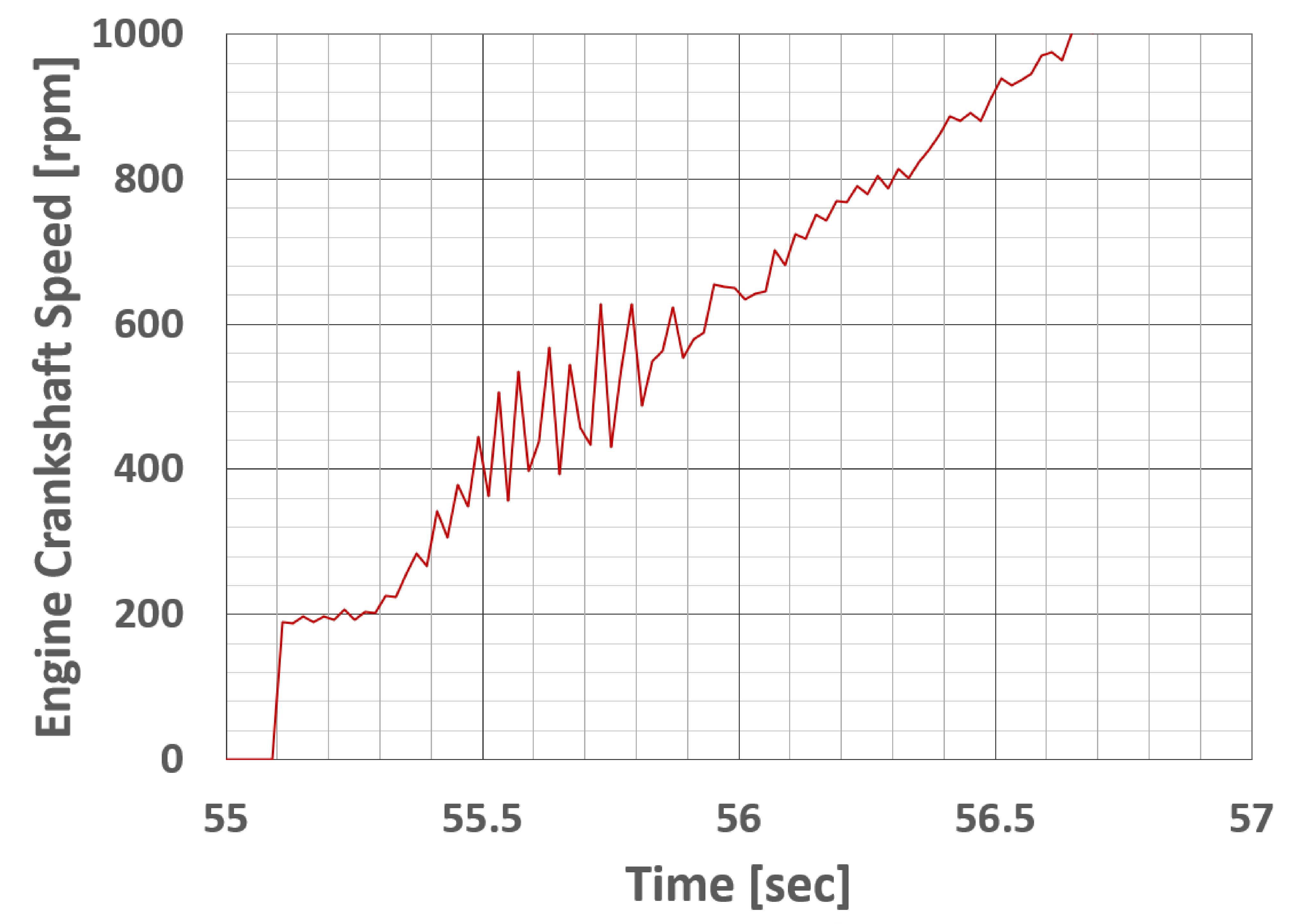
Pre-Failure TVA

- **Initial frequency domain torsional vibration analysis focused on the normal engine operating range**
  - 800rpm to 2500rpm Crank
  - 900rpm to 2900rpm ISG
- **The TVA was focused on engine gear train, crankshaft, and torsional coupling health**
  - All design limits were met with margin
- **Peak ISG deflection of 0.11 deg**
  - This equates to ~20 rad/s^2 at 26Hz
- **There was no indication that the ISG would have issues**

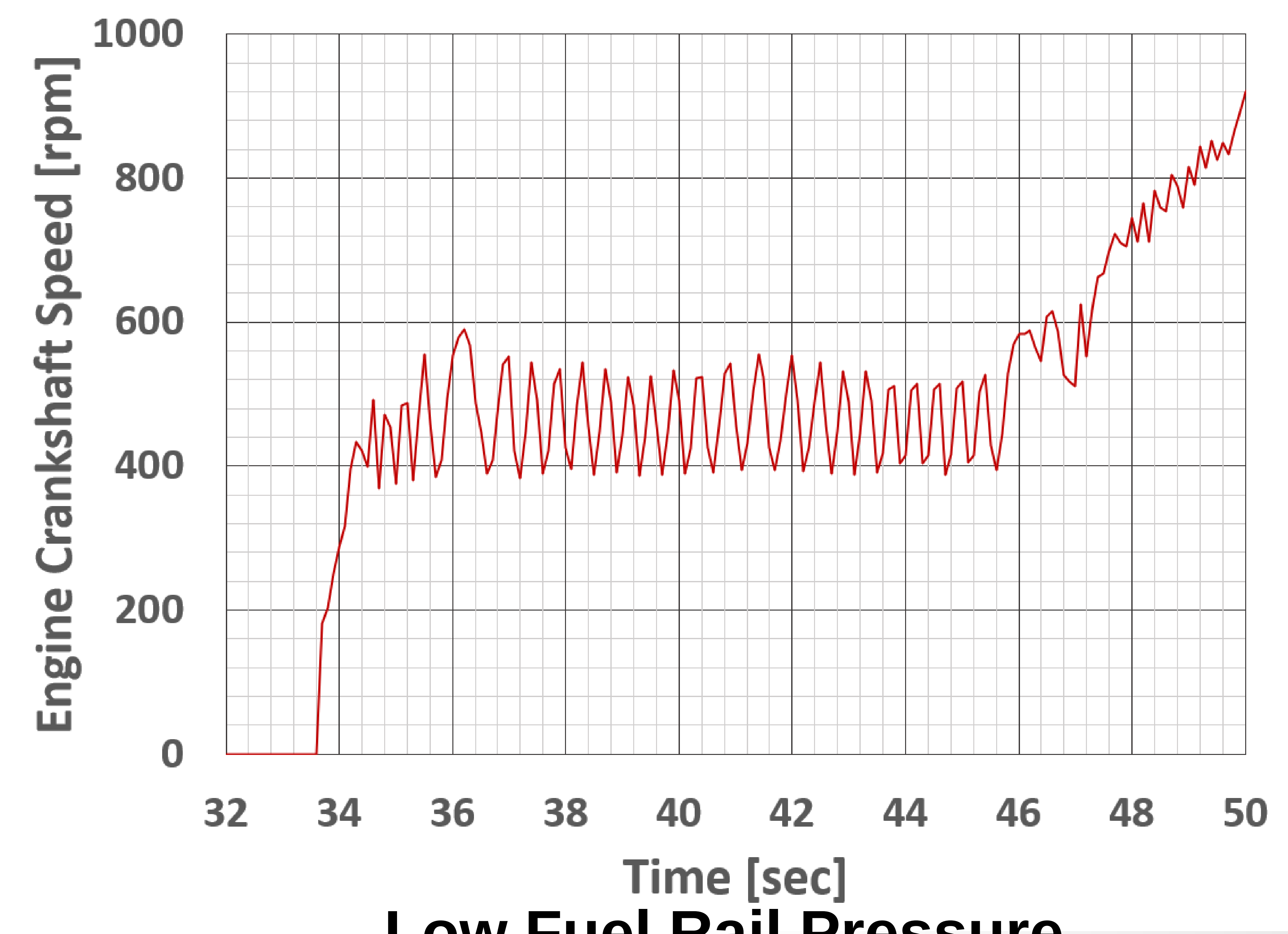


Post Failure TVA

- Engine Speed Traces Recorded from CAN Bus at 50Hz Sample Rate



Normal Engine Start



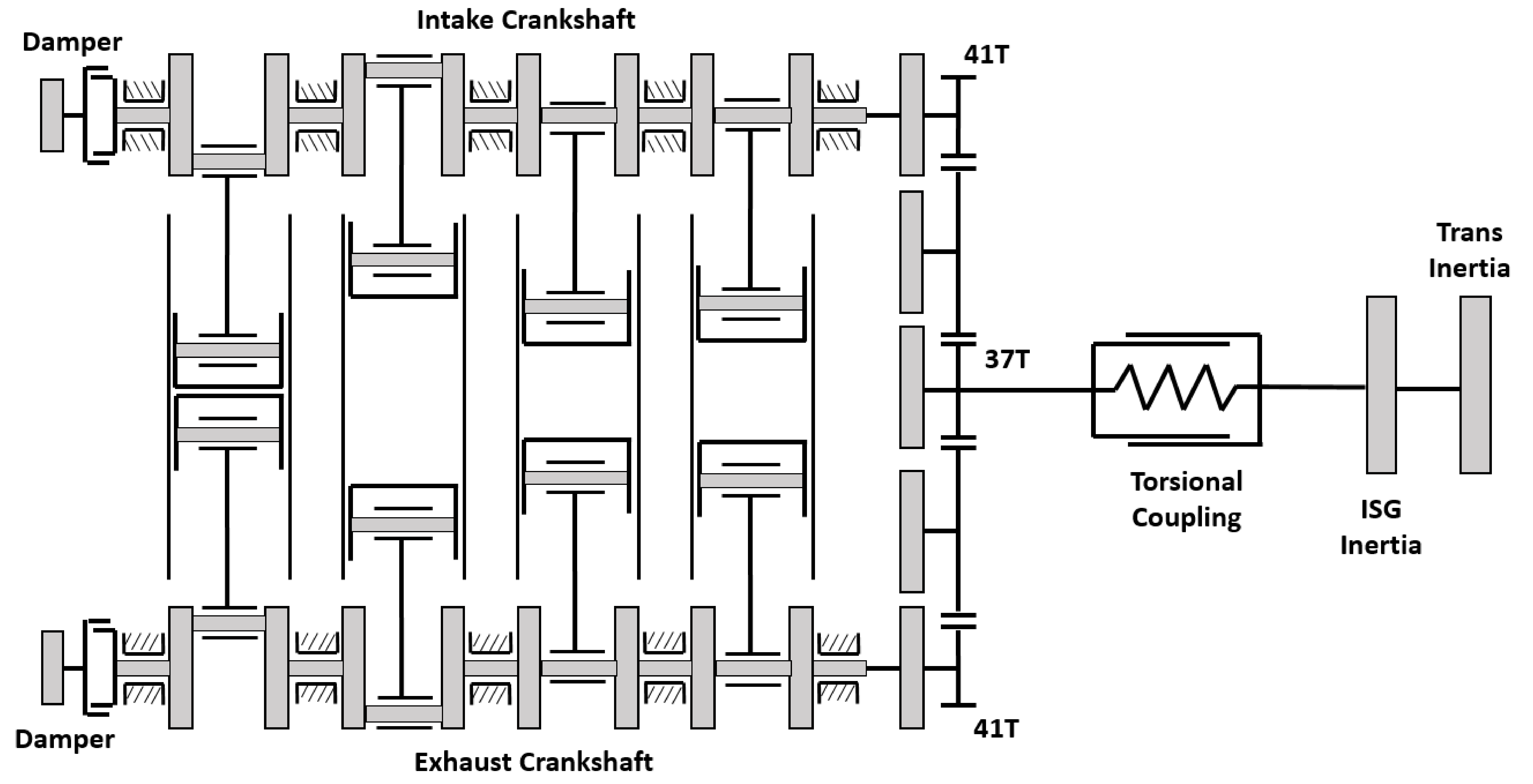
Low Fuel Rail Pressure Start

Post-Failure TVA

- **The amplitude of the vibrations and the number of cycles experienced during engine starting was surprising**
- **The CAN bus speed data shows limited fidelity due to the low sample rate**
  - Actual torsional behavior is likely more severe
- **This prompted the creation of a time domain torsional vibration model to obtain a better understanding of the torsional behavior**

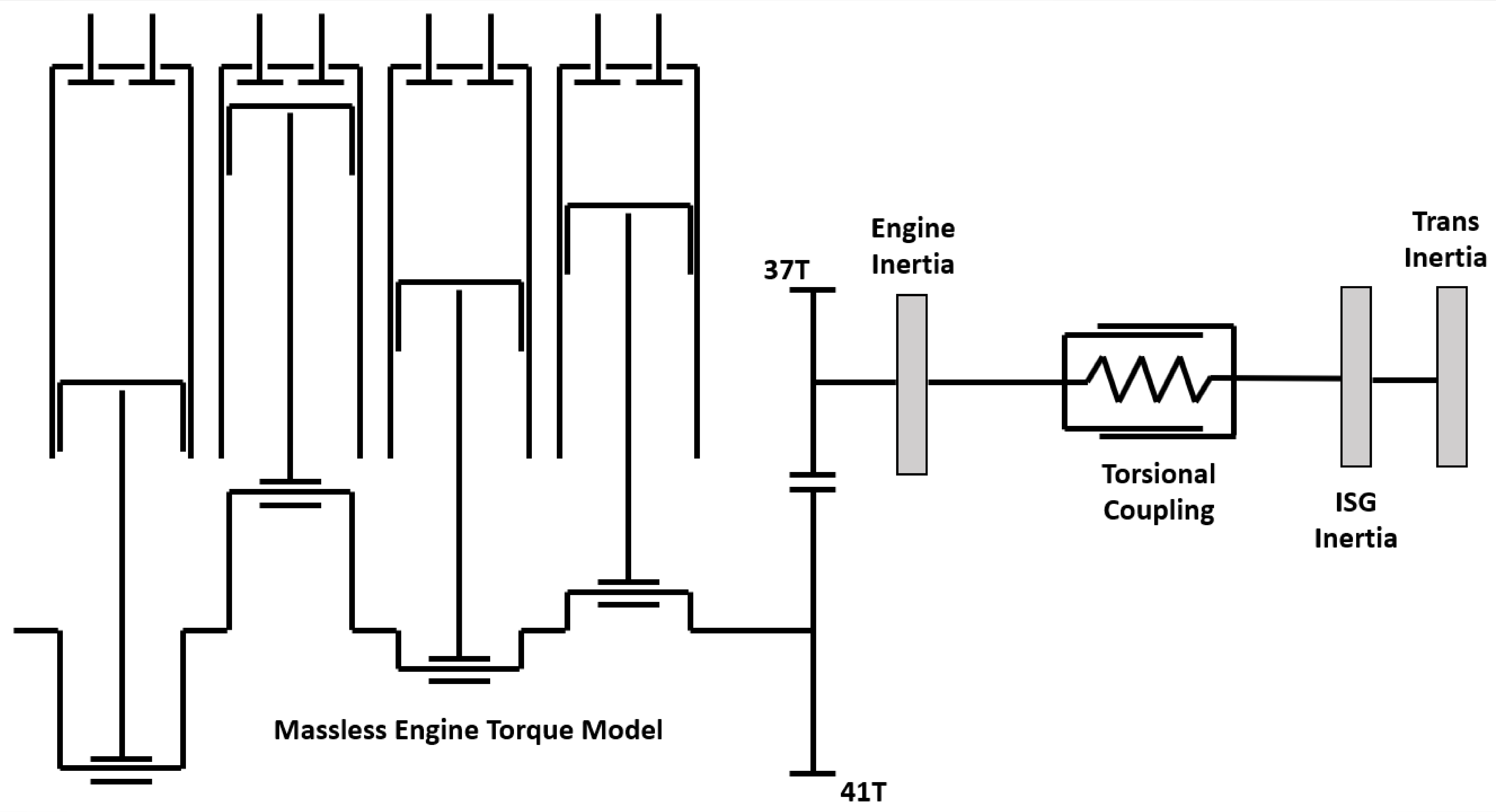
Post Failure TVA

- Mechanical Schematic of Engine / ISG System



Post Failure TVA

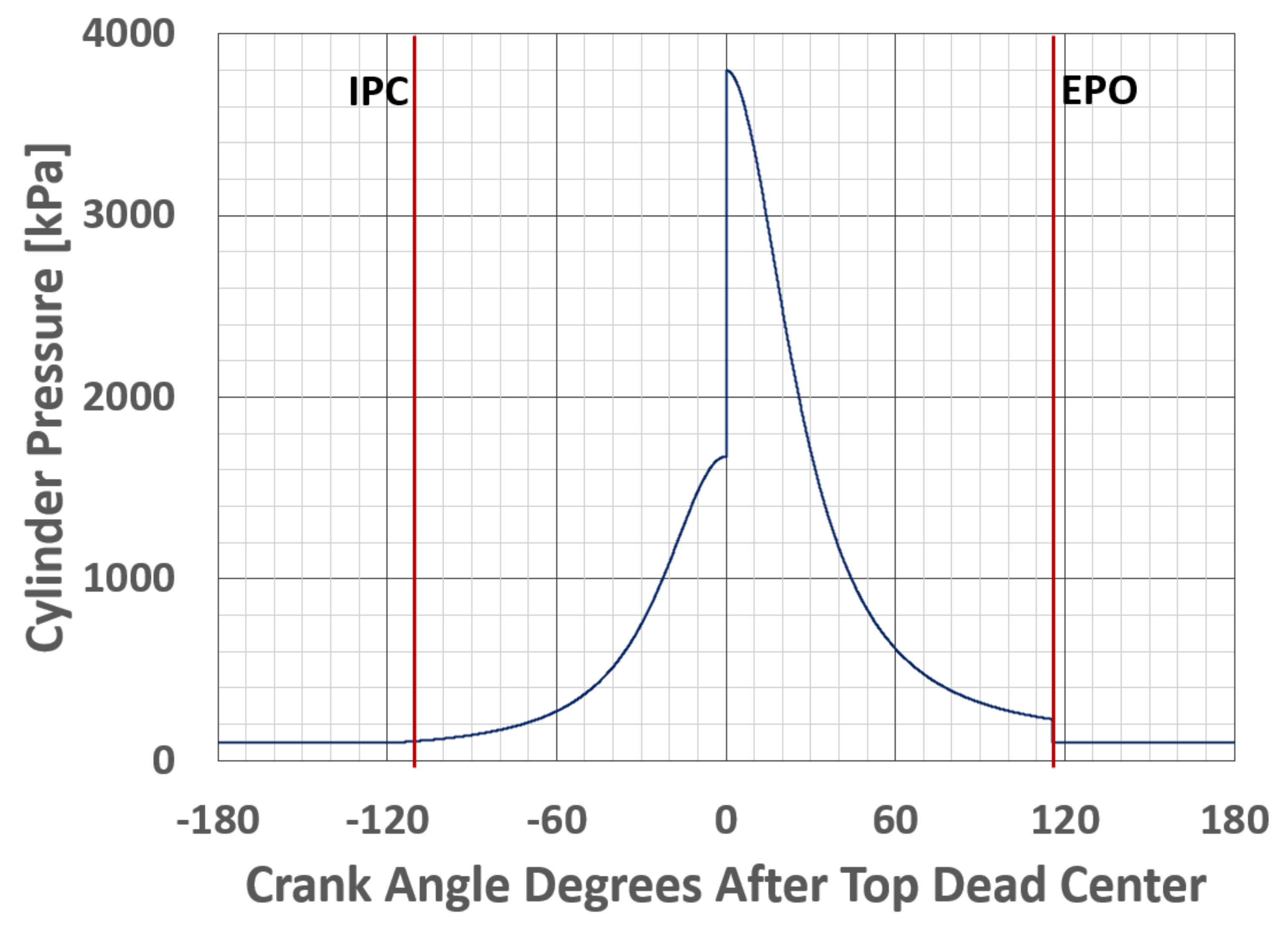
- Schematic of Simplified Engine / ISG System for TVA Model



Post Failure TVA

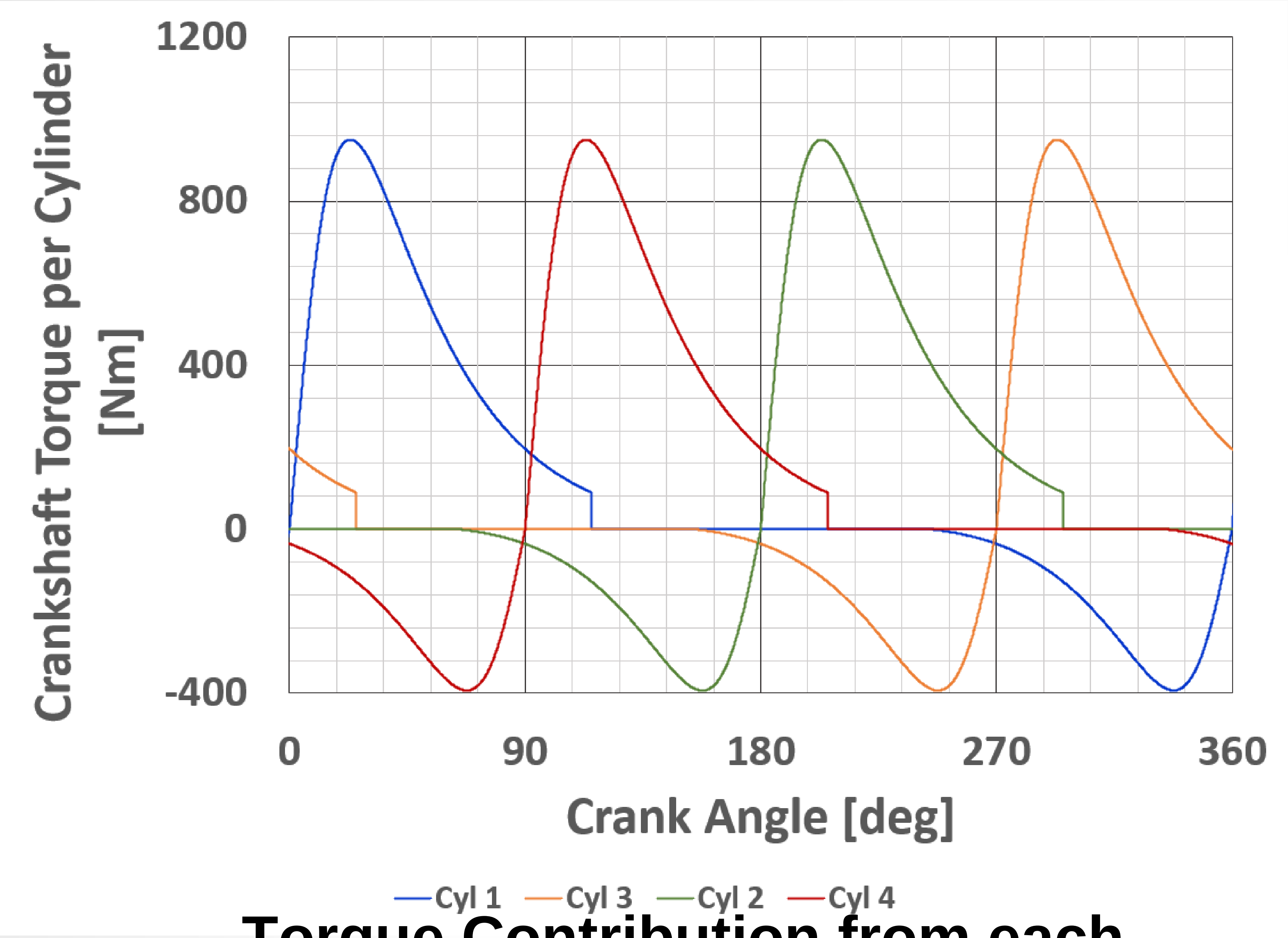
- **Multi-cylinder time domain model**
  - The engine model is built on a simplified crankangle based cylinder model
  - The cylinder model is based on the polytropic process with an instantaneous heat release at TDC
  - The heat release was tuned to match the measured engine acceleration during normal starting events

Bore	130	mm
Stroke	135	mm
Conrod Length	226	mm
Clearance Volume	0.18	liters
Inlet Port Closure	115.4	deg BTDC
Exhaust Port Opening	115.4	deg ATDC
Polytropic Exponent	1.3	-
Peak Cylinder Pressure	38	bar

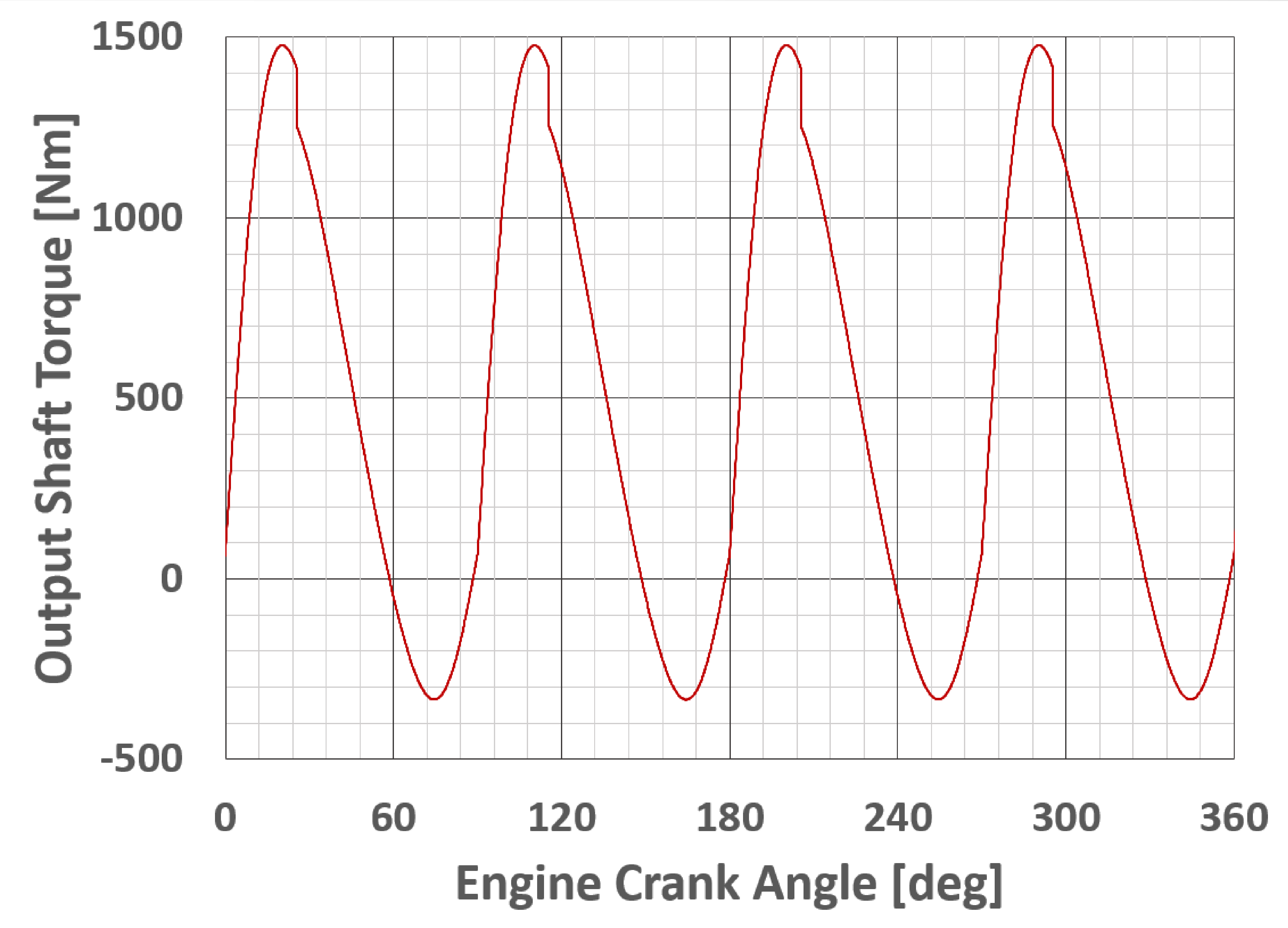


Post Failure TVA

- **Multi-cylinder Crank Angle Engine Model**



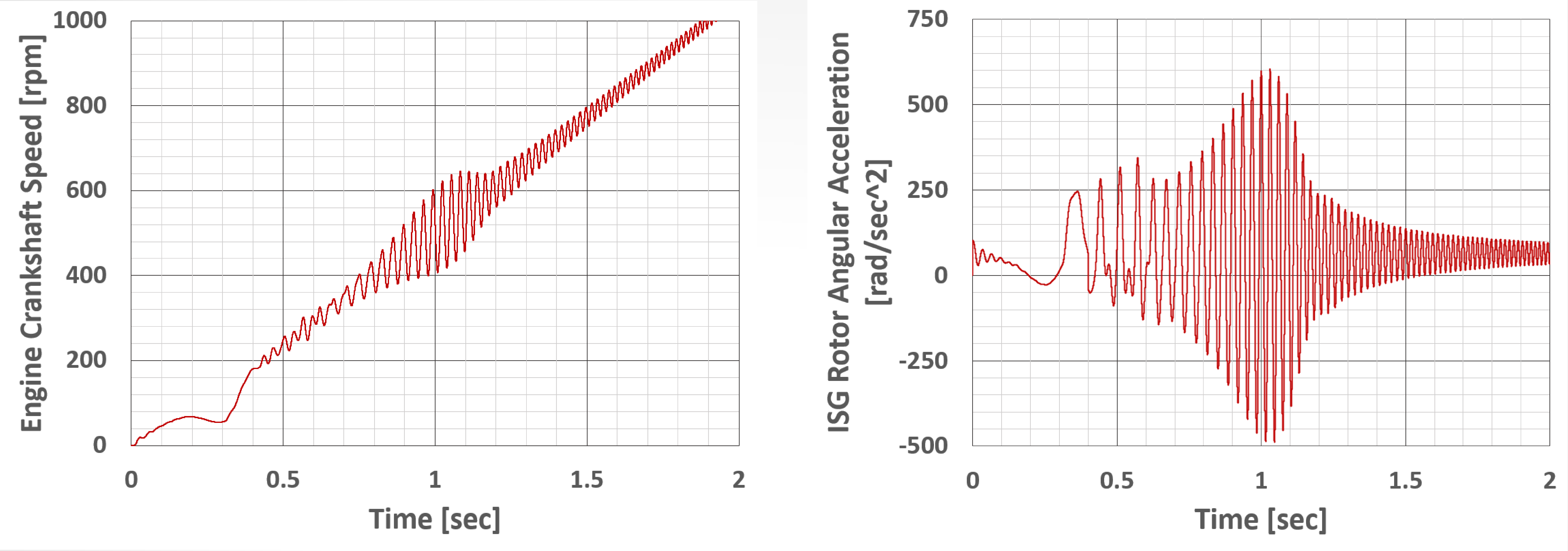
Torque Contribution from each Cylinder



Four Cylinder Engine Torque Trace

Post Failure TVA

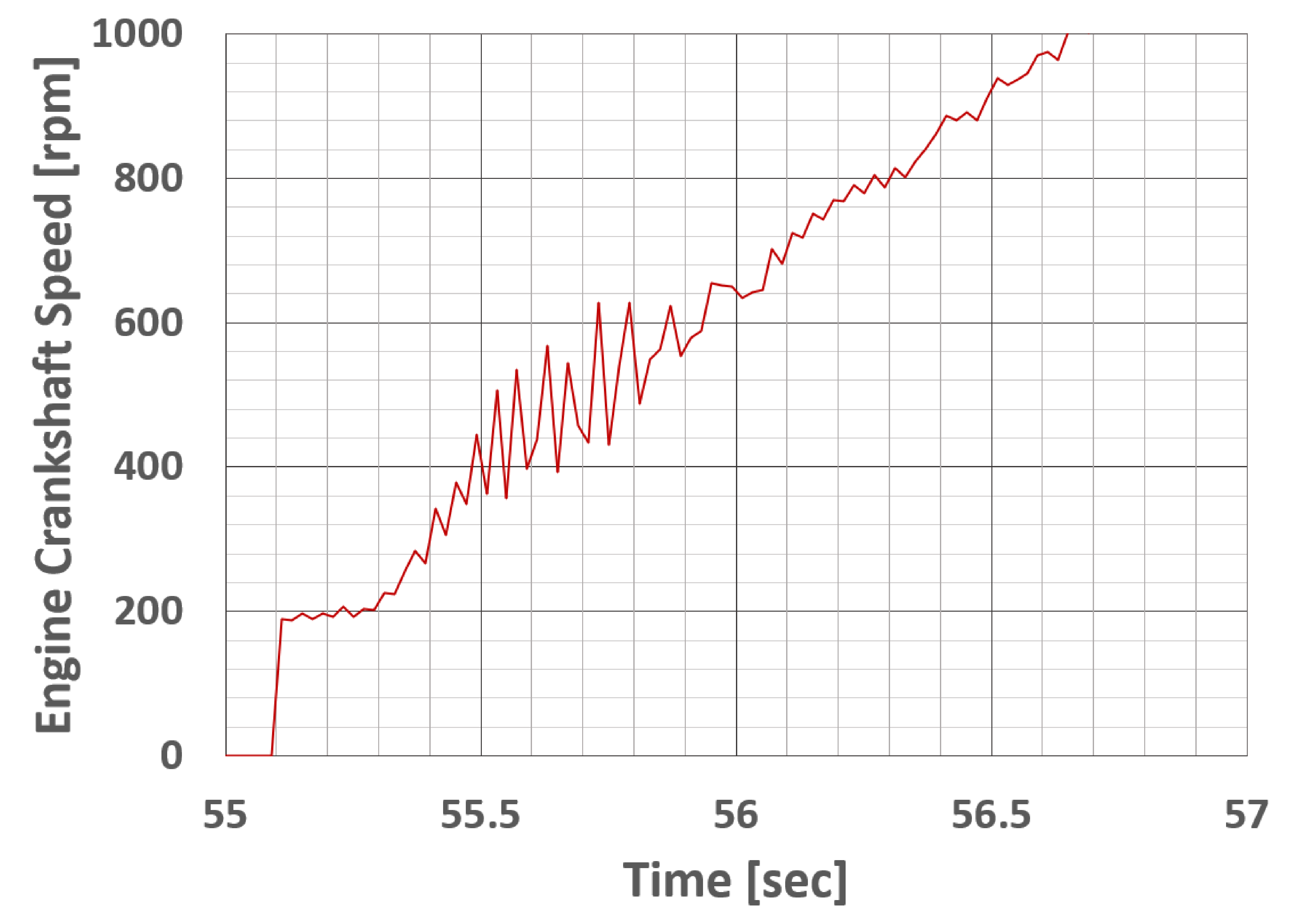
- Time Domain Model Results – Normal Engine Start



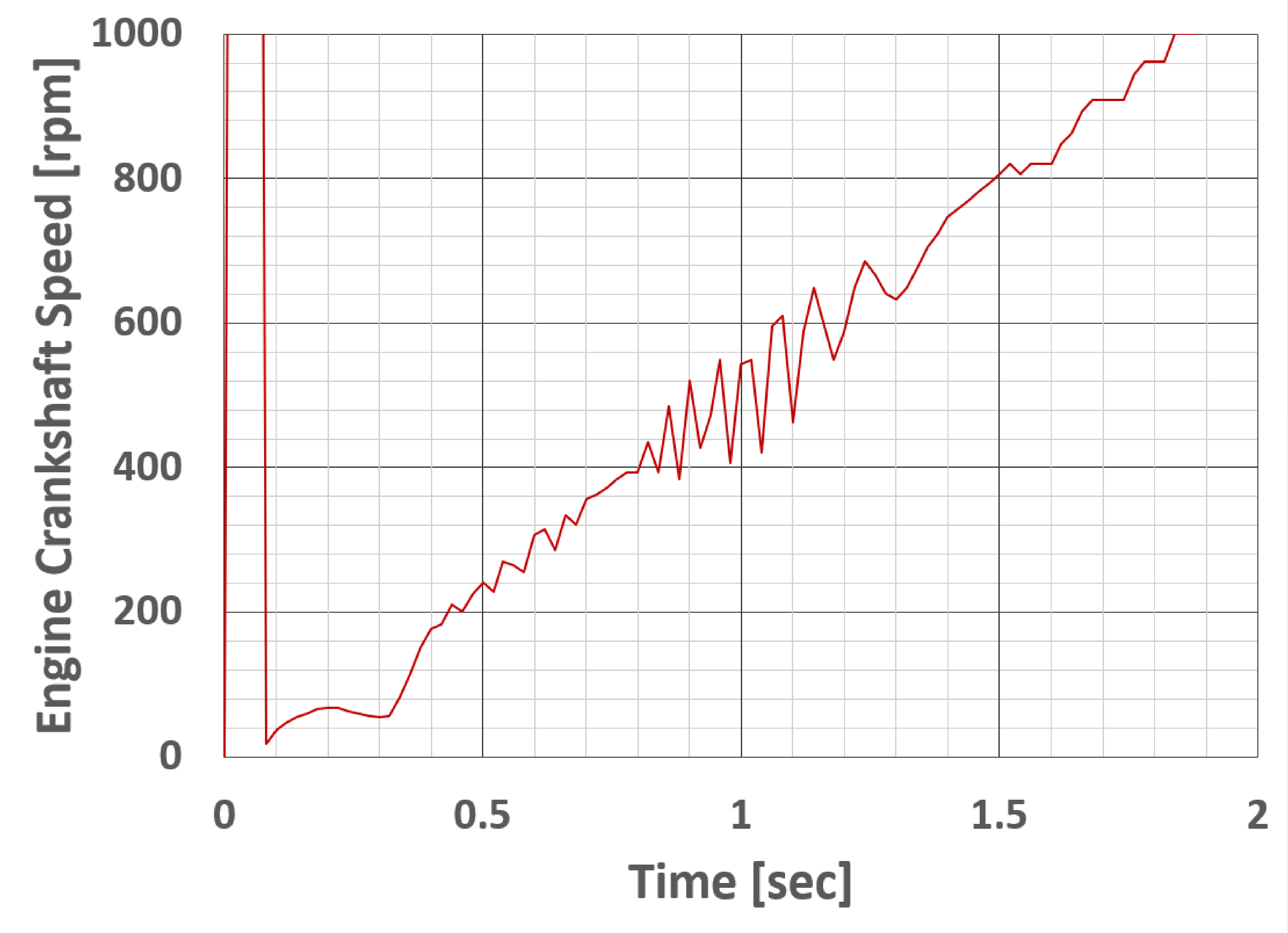
- Model predicts peak ISG rotor acceleration of 600 rad/s^2
  - Results in 2000Nm reaction torque when applied to a rotor inertia of 3.3 kg-m^2
- 21 cycles in excess of 250 rad/s^2

Post Failure TVA

- **Time Domain Model Results – ECU Data Comparison**



Actual ECU CAN Data



Simulated ECU CAN Data

– TVA model results were discretized at 50Hz sample rate to simulate measured ECU CAN data with good correlation

Solution

- **The time domain TVA model revealed the true torsional environment of the ISG rotor**
- **Based on these results, the ISG rotor was redesigned to meet an angular acceleration requirement of 1500 rad/s^2**
- **The redesigned rotor has been exposed to over 500 miles of in-vehicle operation without issue**
  - More than 300 engine starts

Lessons Learned

- **You must perform a TVA to predict the maximum angular acceleration at resonance**
  - The results are sensitive to the torsional damping of the system
  - Make sure you apply a healthy margin to the TVA predictions to protect against uncertainty in damping values
  - Applies to both ISG’s and PTO driven machines
    - High Speed PTO driven machines can be worse due to amplitude amplification thru the gear ratio

- **Every new electric machine must have a peak angular acceleration requirement**

- **New integrations of existing machines must be designed to stay within the electric machine angular acceleration limits**

Lessons Learned

- You cannot assume that the driveline will pass through resonance fast enough to avoid damage

- You cannot assume that the presence of a torsional damper between the engine and the electric machine will protect the machine from damage

- All bolted joints on rotating components must be designed against slippage at the peak angular acceleration seen in resonance

Lessons Learned

- **You should design for an infinite life against fatigue in resonance**
- **You should perform a time domain TVA and assess total cycle accumulation when**
  - The materials used in rotating components do not have an endurance limit
  - You are unable to achieve infinite life due to other design constraints (weight, volume, inertia, etc.)

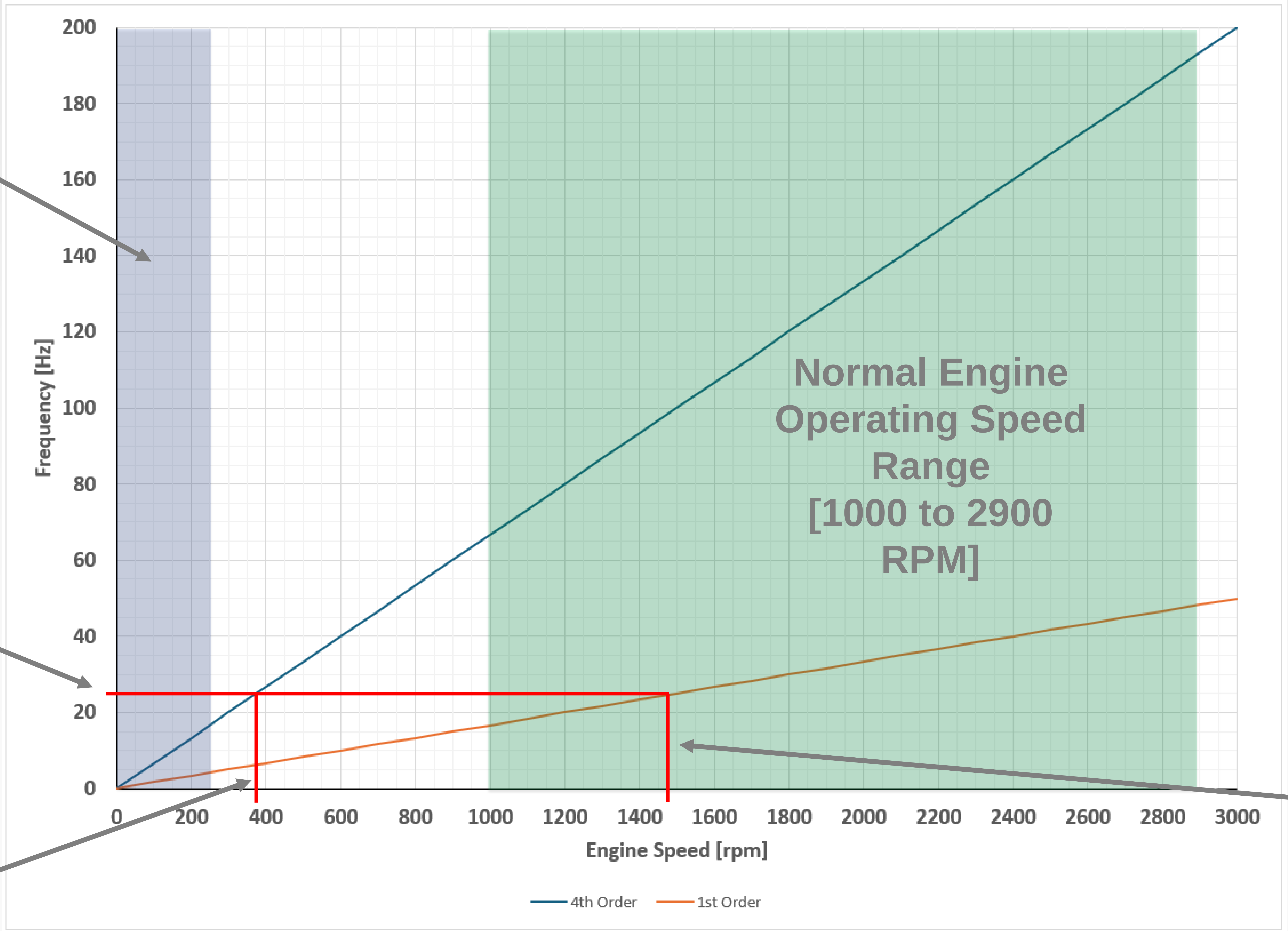
Lessons Learned

- **Assess cycle accumulation using Miner's Rule when infinite life is not possible**
  - Create a histogram of alternating stress conditions when passing thru resonance on each start and shut down
  - Examine vehicle usage profile to determine number of start/stop events per unit time
    - Anti-Idle capabilities significantly increase the number of start/stop events per day
  - Determine the cumulative effective cycles using Miner's Rule to determine time to fatigue failure

Don’t Forget About Engine Misfire

Engine Cranking Speed Range [0 to 250 RPM]

Design Natural Frequency [26Hz]



4th Order Resonance [390 RPM]

1st Order Resonance [1460 RPM]