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Overcoming Failure of Synchronous Motor Driven Compressor Train by Application of Controlled Slip Clutch

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Since 2009, he has been involved in rotating machinery business of oil & gas such as petrochemical and gas industry for Asian, European and Middle Eastern markets.

Abstract

This case study describes failure incident of a centrifugal compressor, its root cause investigation, and countermeasure implementation.

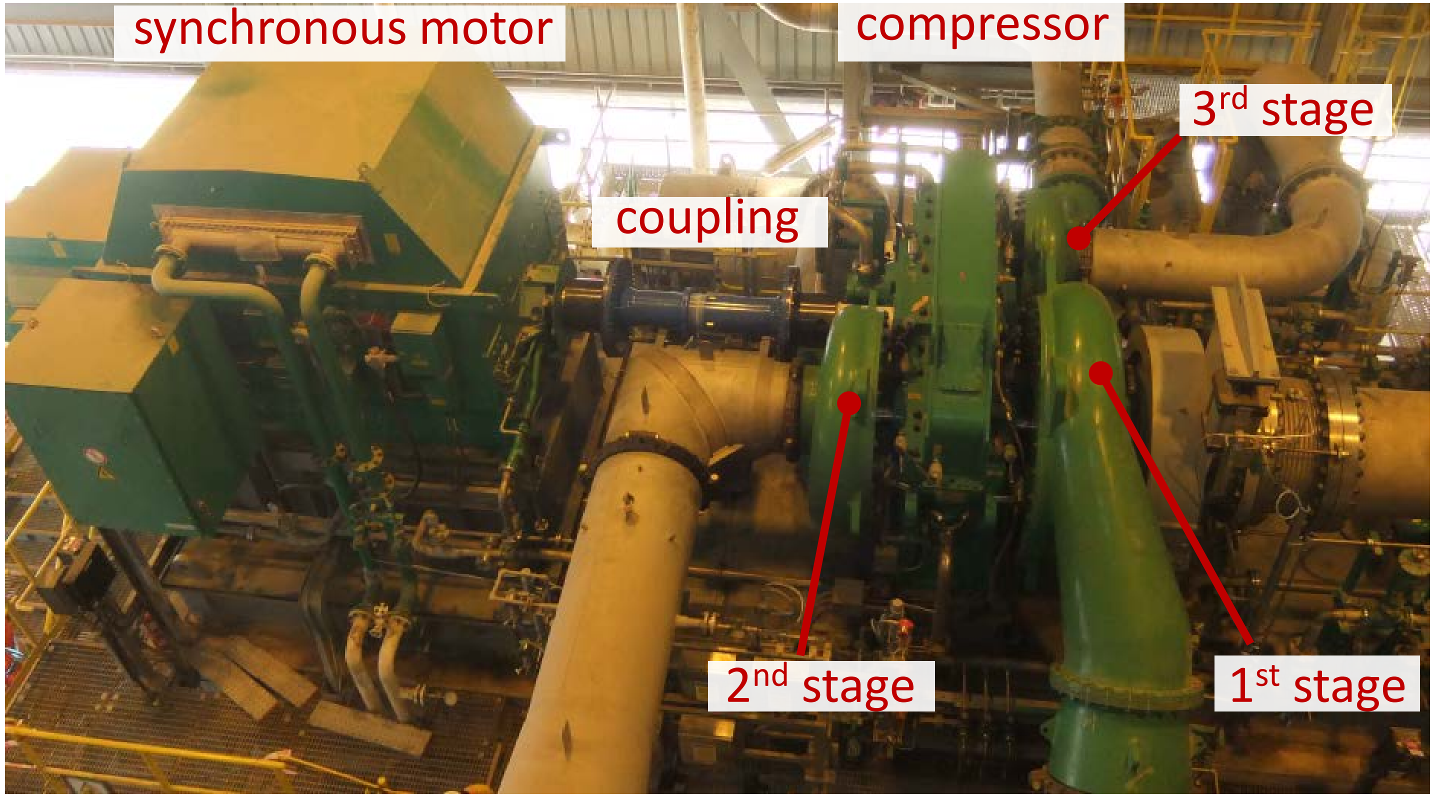
An integrally geared centrifugal air compressor driven by a synchronous motor failed during its initial startup in a newly constructed petrochemical plant. Failure mechanism was identified as torsional resonance during synchronous motor's startup with stronger torque amplification than the design intention.

As a quick yet very effective solution to this problem, a clutch with controlled slip mechanism was applied. Torsional analysis reflecting the clutch mechanism as well as on-site strain gauge measurement verified its effectiveness. After implementation of this clutch, no malfunction or abnormality has been observed in the compressor train.

Introduction
&
Problem Statement

Train overview

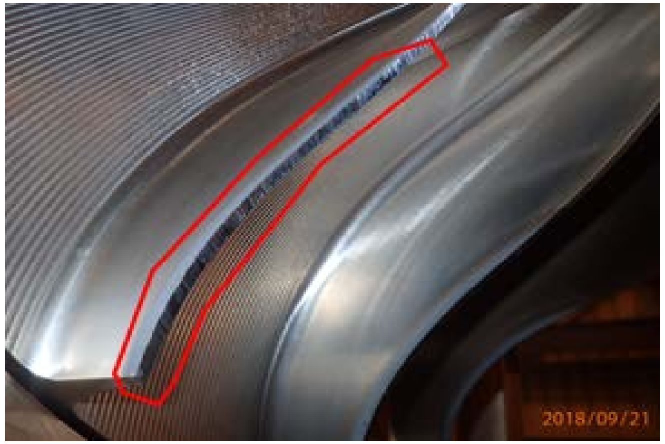
[Compressor]	Integrally geared centrifugal air compressor (three stages compression)
[Driver]	Salient pole synchronous motor (4 poles), 7,500 kW (10,000 hp) rated output, direct on line starting, 50 Hz line frequency
[Coupling]	Diaphragm coupling with spacer



Problem statement

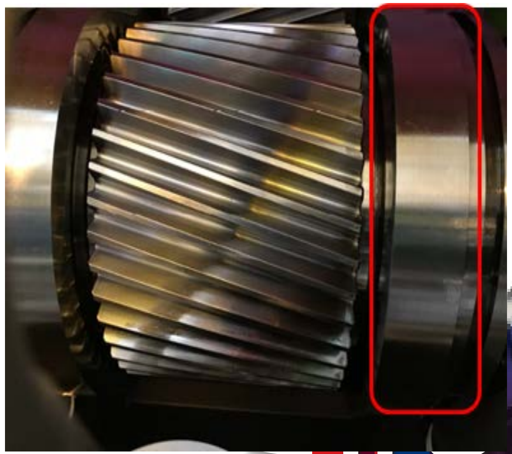
- An integrally geared centrifugal air compressor driven by a synchronous motor was damaged during its initial startup (i.e. during its very first startup attempt) after installation in a newly constructed petrochemical plant
- During the incident, compressor's 2nd stage impeller severely touched the inlet casing (i.e. hard rubbing), damaging both the impeller and the casing, and the compressor tripped due to HH radial vibration of the pinion shaft
- Hard rubbing occurred due to the slipped rider ring of the pinion rotor, which otherwise should have sustained the pinion rotor's axial position relative to the bull gear and the compressor casing

damaged 2nd stage impeller



damaged 2nd stage casing

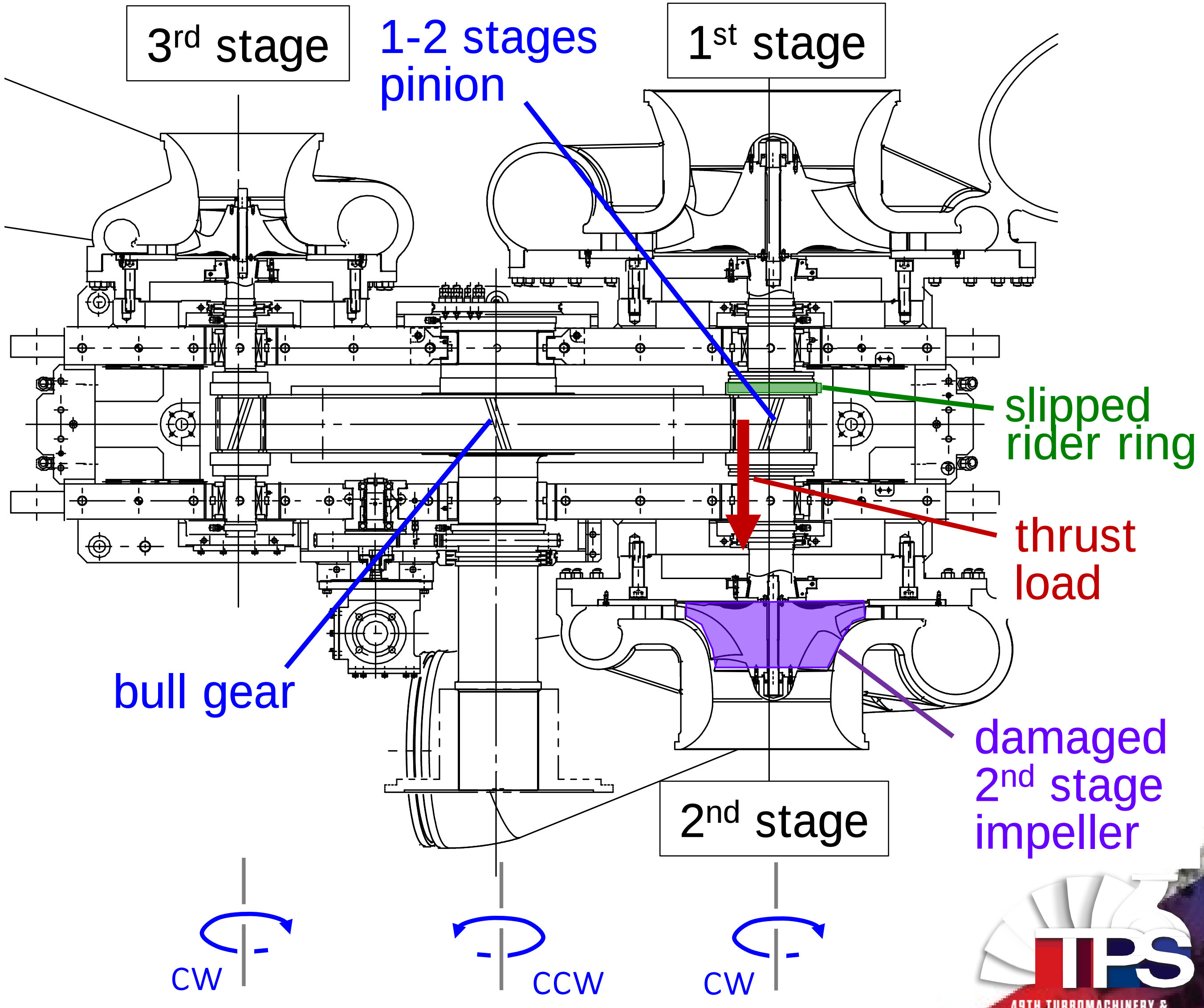
slipped rider ring



Compressor construction and thrust load

- Due to single helical meshing between bull gear and pinion, transmitted torque is converted to thrust load
- In case of excessive torque transmission, thrust load can exceed the design limitation and destroy the rider ring, resulting in unrestricted axial movement of the pinion rotor

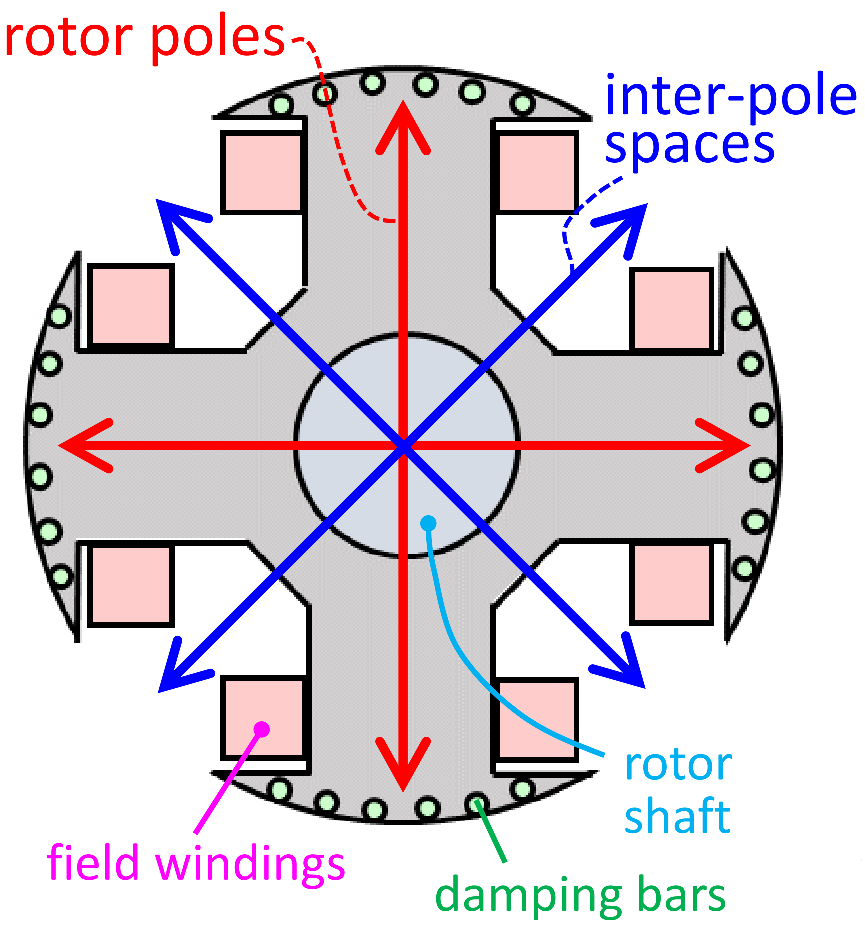
→ What could have caused such an excessive torque transmission?



Torsional Resonance in Synchronous Motor Train

Startup torque oscillation by synchronous motor

- Many synchronous motors are self-starting (i.e. they are started as induction motors by damping bars)
- Magnitude of the instantaneous torque during startup varies depending on the rotor position relative to the rotating magnetic field, because;
  - Unlike a squirrel cage induction motor, rotor of a synchronous motor has poles and inter-pole spaces
  - Instantaneous starting torque is greater when the magnetic flux by the stator passes rotor poles
  - Less torque is developed when passing inter-pole spaces
- In this way, generation of startup torque oscillation is unavoidable in a synchronous motor (unless soft started w/ VSD)



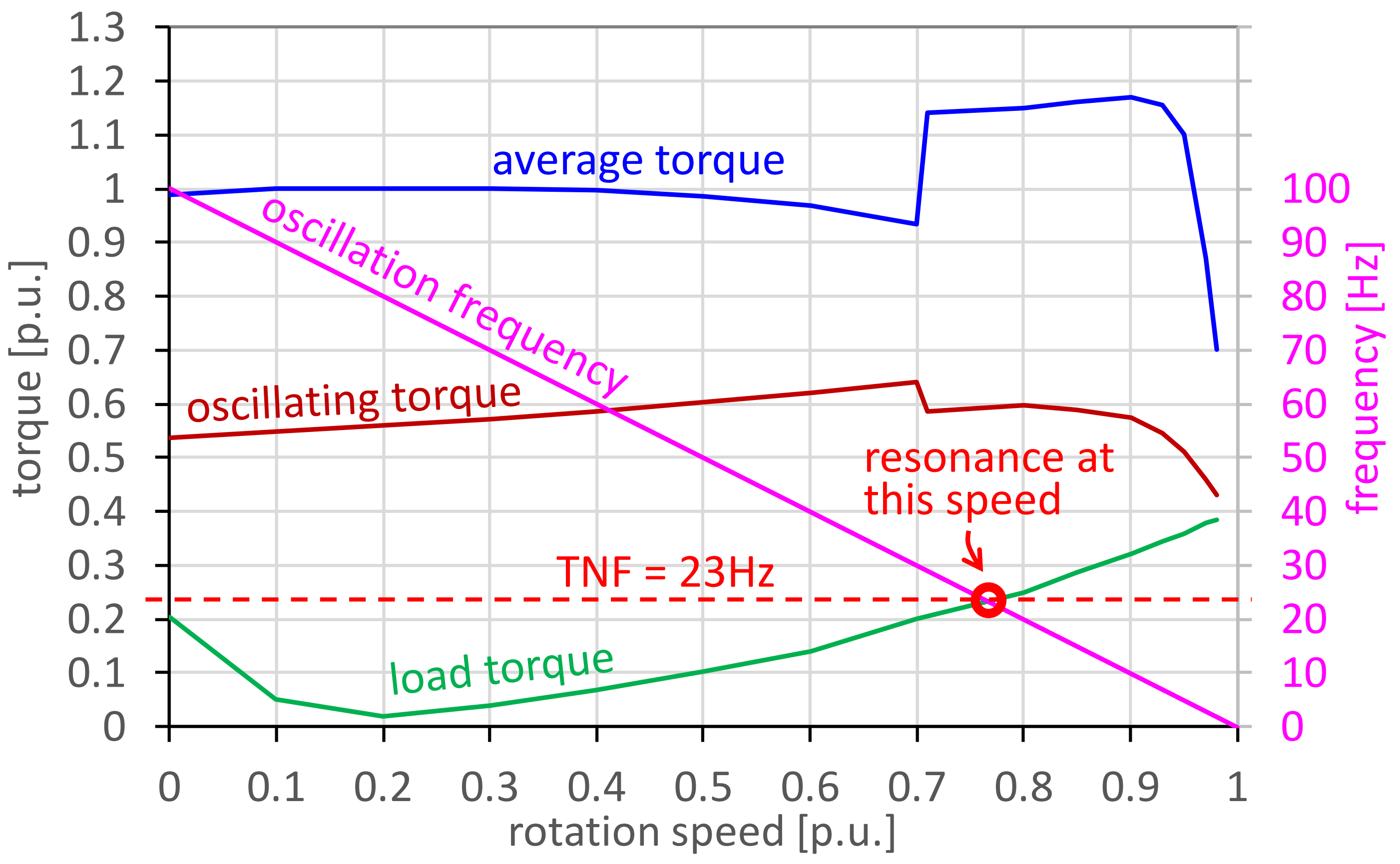
Rotor of salient pole synchronous motor

Torsional resonance in synchronous motor train

- Generated torque oscillation frequency = 2 x slip x line frequency = 2 x slip frequency

  → Oscillating torque frequency varies from 100 Hz (at standstill) to 0 Hz (at synchronous speed) during startup

- At some point during motor acceleration, oscillating torque frequency coincides with the torsional natural frequency (TNF) of the train (= 23 Hz in this case) → Torsional resonance is inevitable during startup (at rotation speed of 0.77 p.u. = 1155 rpm in this train)



Motor startup torque characteristics

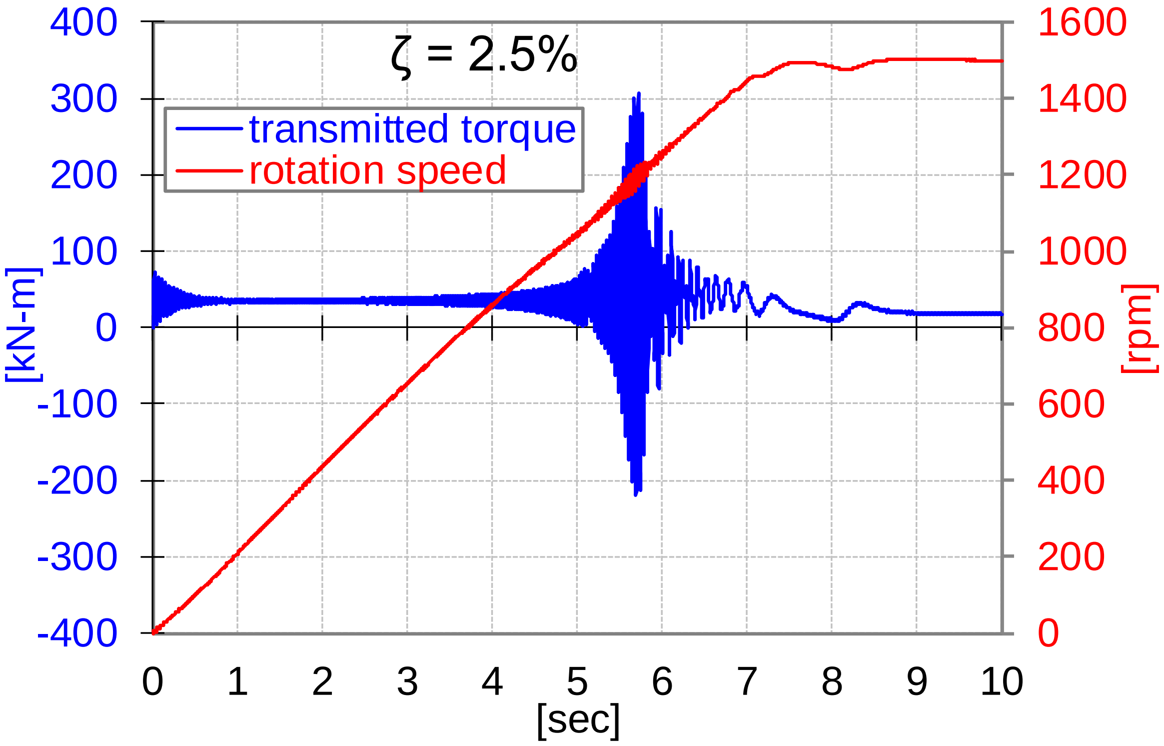
Note: In this application, 1 p.u. (torque) = 47.75 kN-m, 1 p.u. (speed) = 1500 rpm

Failure Mechanism

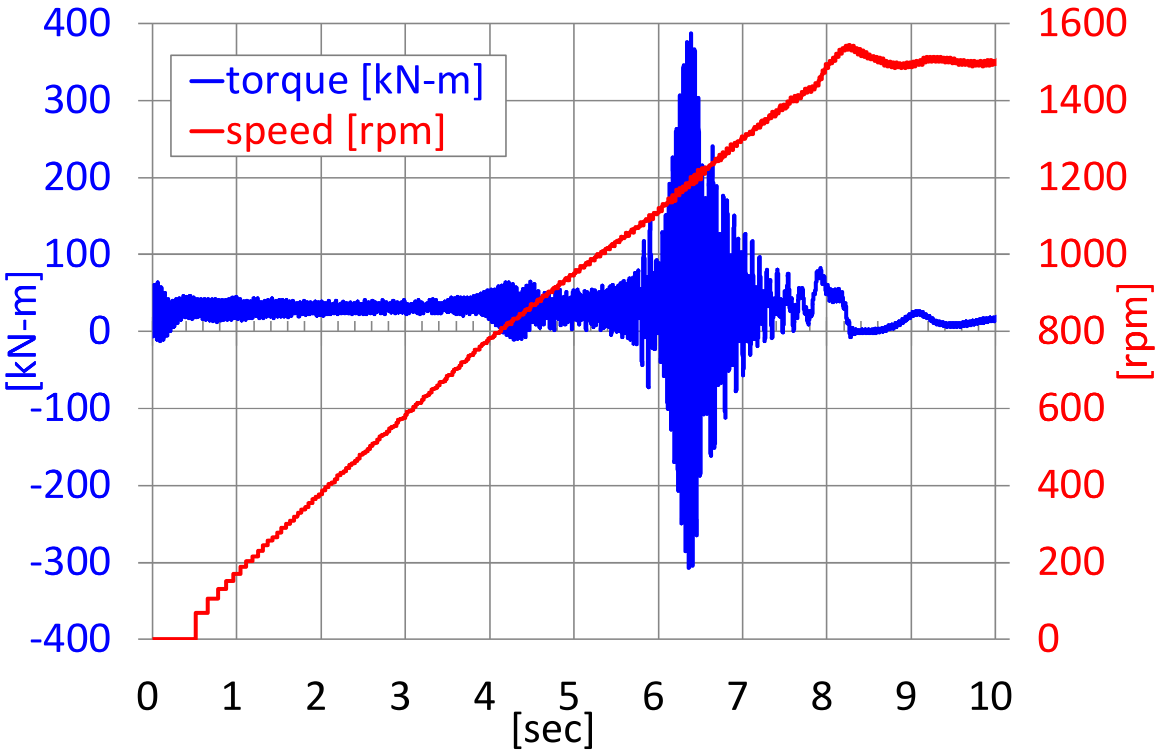
Torque measurement by strain gauge

- Measured peak torque by strain gauge at the coupling spacer (388 kN-m = 8.1 p.u.) was significantly larger than the prediction by torsional analysis (300 kN-m = 6.3 p.u.)

→ What has caused the discrepancy?



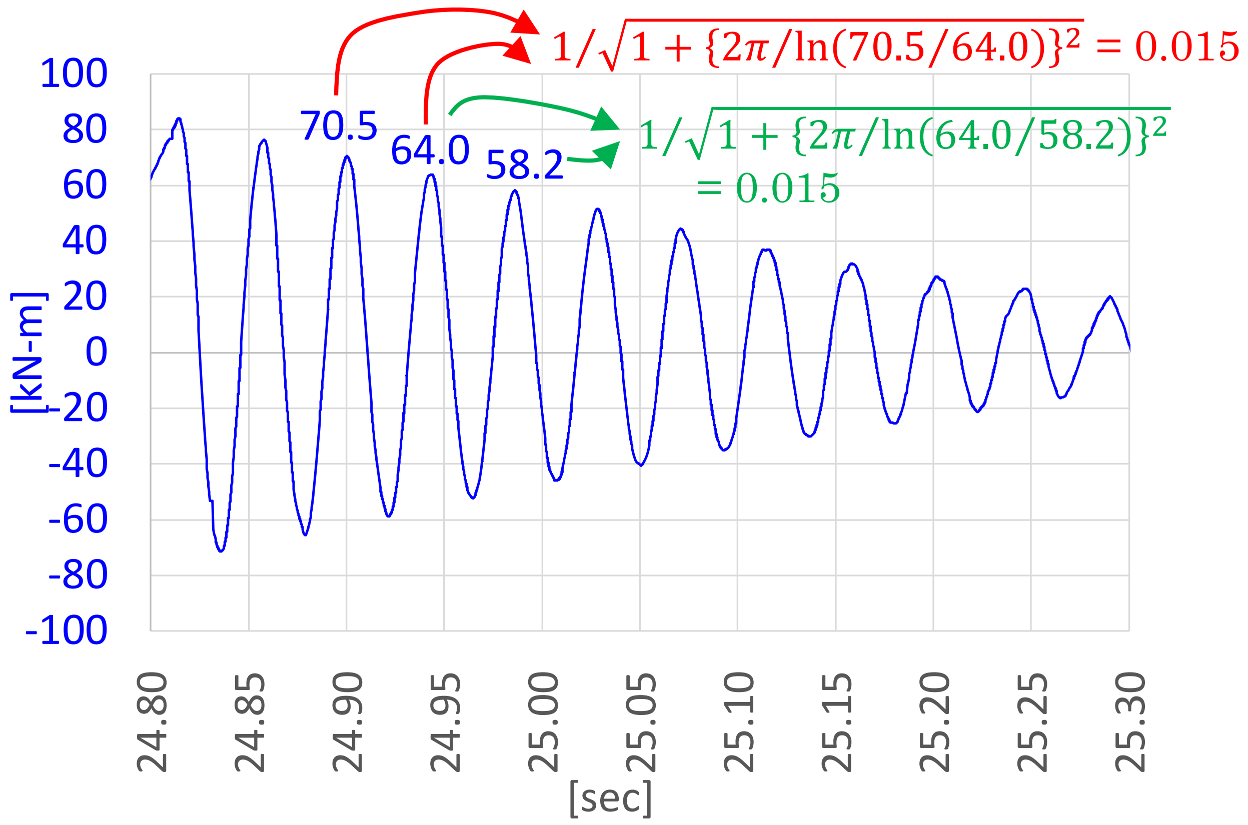
Original analysis with ζ = 2.5%



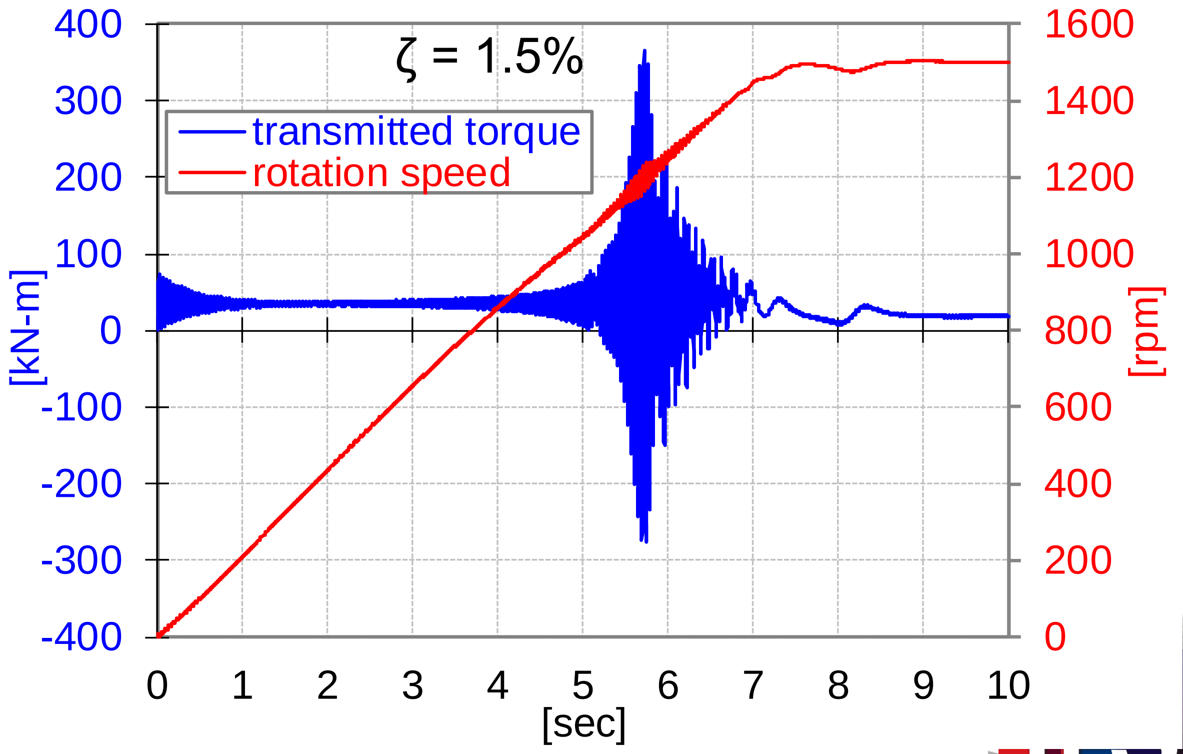
Field measurement

Source of discrepancy

- Original torsional analysis was based on the assumed damping ratio (ζ) of 2.5%
- Actual damping ratio was found to be 1.5% by field measurement during shutdown
- Damping ratio of 2.5% was too optimistic (With damping ratio of 1.5%, torsional analysis better matched the field measurement)



Field measurement during shutdown



Modified analysis with ζ =1.5%

Failure Mechanism

Failure occurred due to:

- Generation of torque oscillation during the synchronous motor's startup
- Amplification of the torque oscillation by torsional resonance at the rotor system's first torsional natural frequency
- Stronger torque amplification due to damping assumption discrepancy (actual damping was weaker than the assumption), resulting in larger torque pulsation than design intention
- Conversion of pulsating torque to pulsating thrust load at the helical meshing between the bull gear and the pinion
- Rider ring slippage due to large thrust load beyond its capacity, causing loss of axial constraint of the pinion rotor relative to the bull gear and the compressor casing
- Hard rubbing between compressor impeller and stator due to unrestricted axial movement

Countermeasure

Necessity of countermeasure

- With the measured peak torque magnitude, in addition to the concern of rider ring slippage, expected life of the pinion teeth would be reduced to less than 20 years
- To satisfy design requirement, suppression of peak torque during startup was mandatory

Countermeasure evaluation

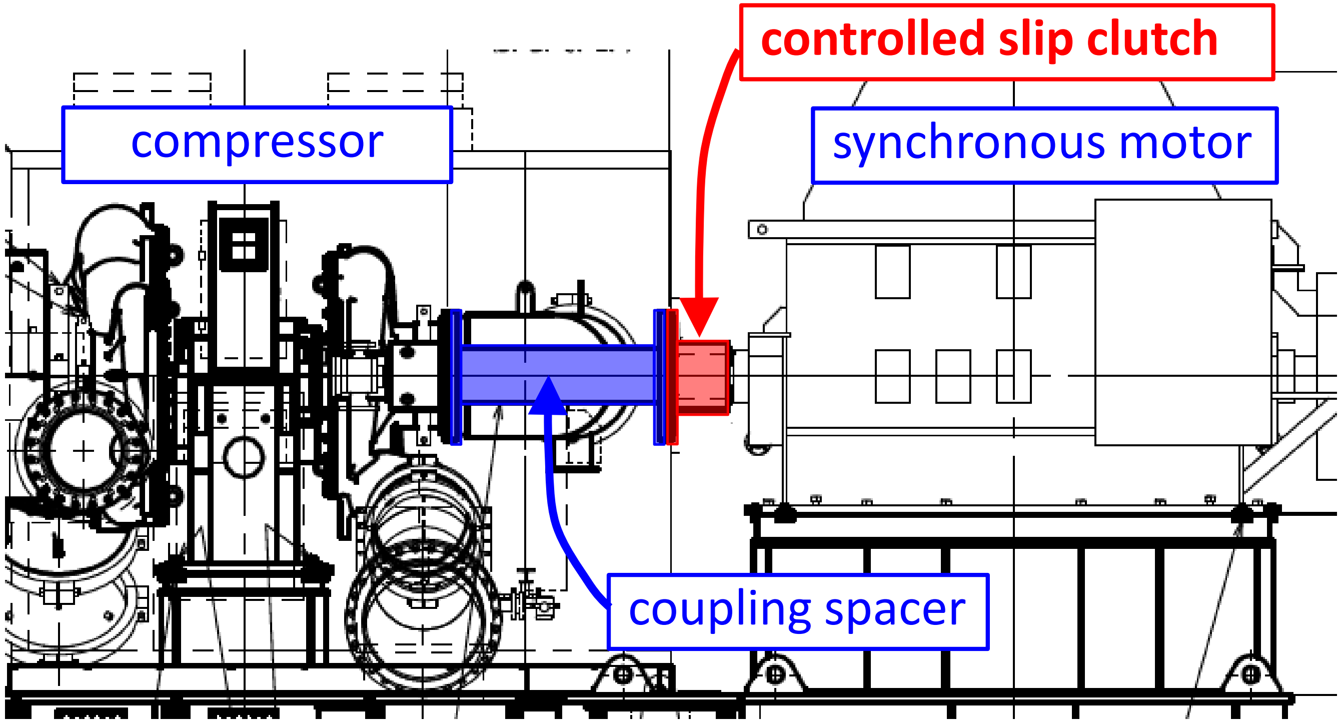
- Countermeasure candidates were investigated
- Among the candidates, controlled slip clutch was the most effective option which would certainly suppress the peak torque to the desired level

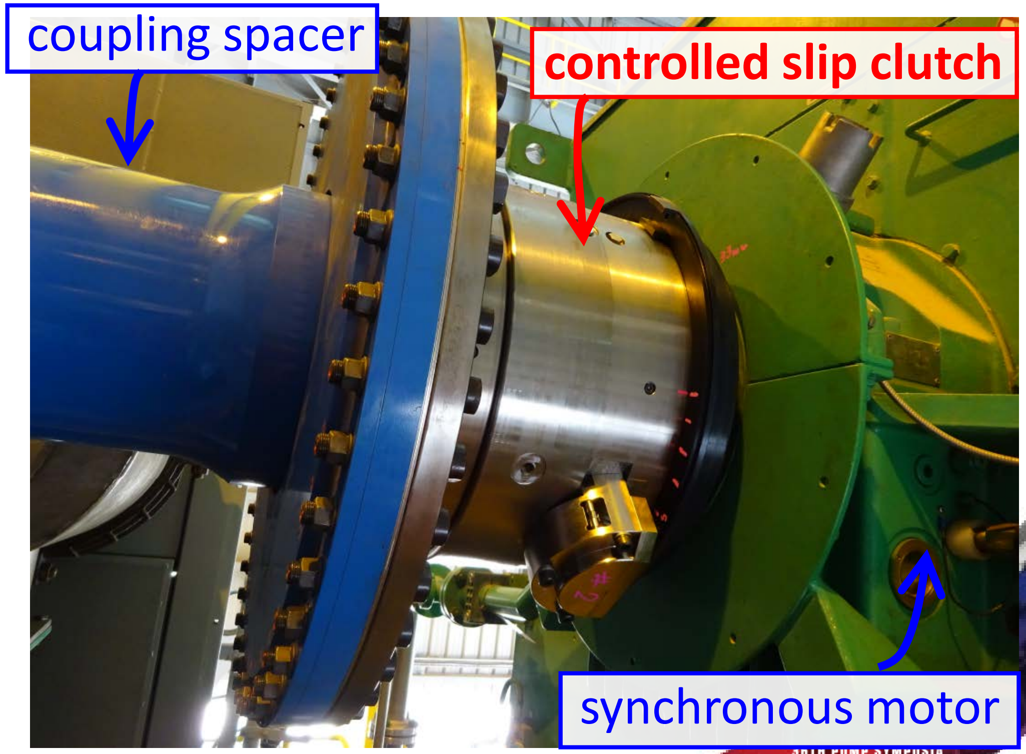
	Expected peak torque	Design impact	Conclusion
Rubber block coupling (rubber-in-compression coupling)	Around 5.3 p.u.	Manageable (Only coupling replacement is required)	Peak torque reduction is marginal and insufficient to surely mitigate the problem
Reduced voltage soft starter (step down transformer)	Around 6.3 p.u.	Significant (Adversely impacts the already completed electrical room arrangement)	Peak torque reduction is marginal and insufficient to surely mitigate the problem
Controlled slip clutch	Less than 3 p.u.	Minimum (Only motor coupling hub replacement is required)	To be implemented

Controlled Slip Clutch

Controlled slip clutch – location and weight

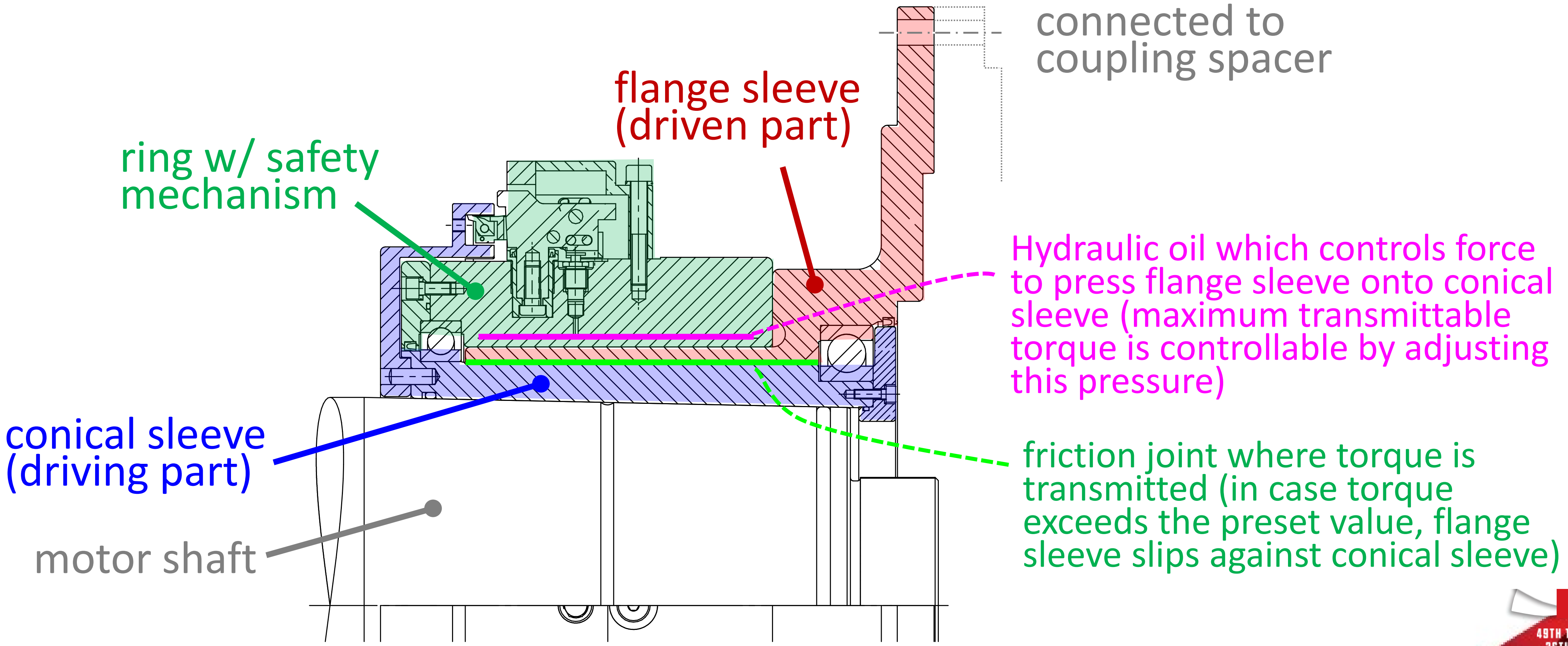
- Controlled slip clutch is integrated in the driver side coupling hub
- As such, it suffices to merely replace the existing motor coupling hub with the clutch, simplifying the field modification processes
- As the total weight of the controlled slip clutch (310 kg) is comparable to the original coupling hub (265 kg), there is no adverse effect (e.g. lateral resonance) due to this modification





Controlled slip clutch - construction

- Controlled slip clutch consists of three main parts – conical sleeve (driving part), flange sleeve (driven part), and ring w/ safety mechanism (protection mechanism part)
- Torque is transmitted through a friction joint where the torque capacity is precisely controlled by hydraulic pressure



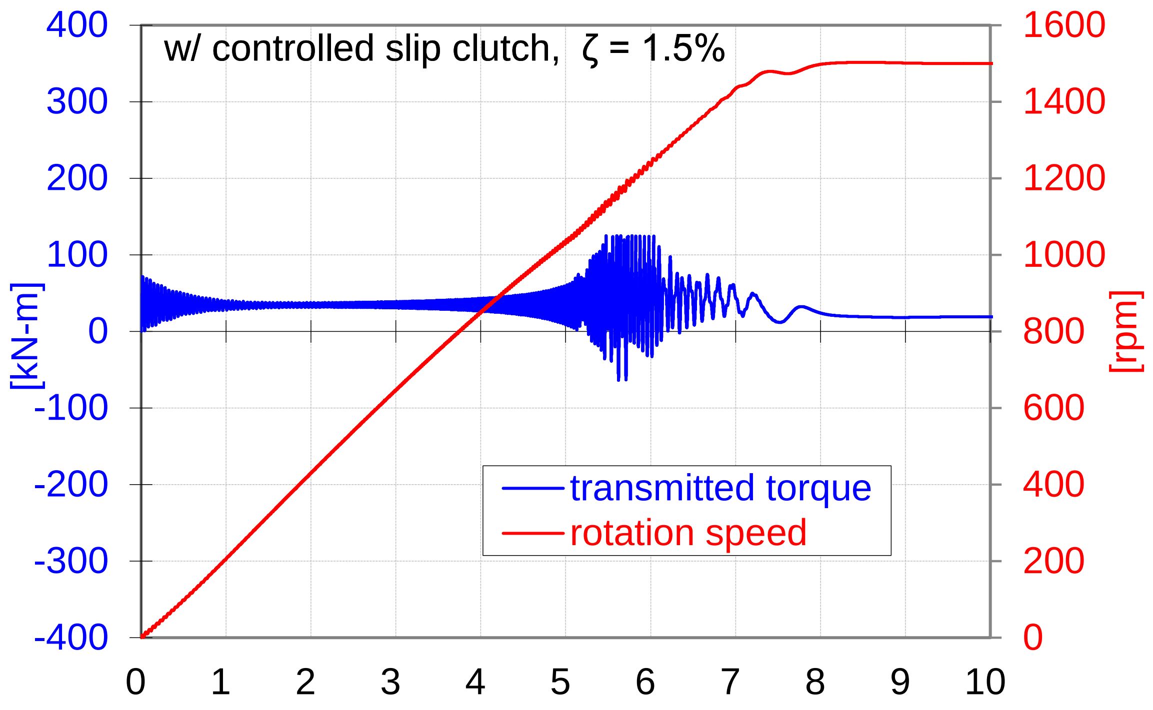
Controlled slip clutch - mechanism

- As long as torque is within the predetermined value (= the torque value that is sufficiently within the allowable limit to prevent rider ring slippage and pinion fatigue failure), it is fully transmitted to the driven part through static friction between two concentric sleeves (i.e. conical sleeve and flange sleeve) within the clutch
- On the other hand, when torque exceeds the preset value, flange sleeve (which is the driven part) slips against conical sleeve (which is the driving part) such that only the predetermined amount of torque is transmitted to the driven part through dynamic friction
- Since the maximum transmitted torque is precisely controlled by preadjusting the hydraulic oil pressure (which presses flange sleeve onto conical sleeve) that dictates clutch engagement, transmitted torque during any transients including synchronous motor’s startup can be maintained within the desired level while unaffecting normal operation of the compressor train
- [Note] Since overhaul is required in every five years of operation, keeping a spare clutch is essential

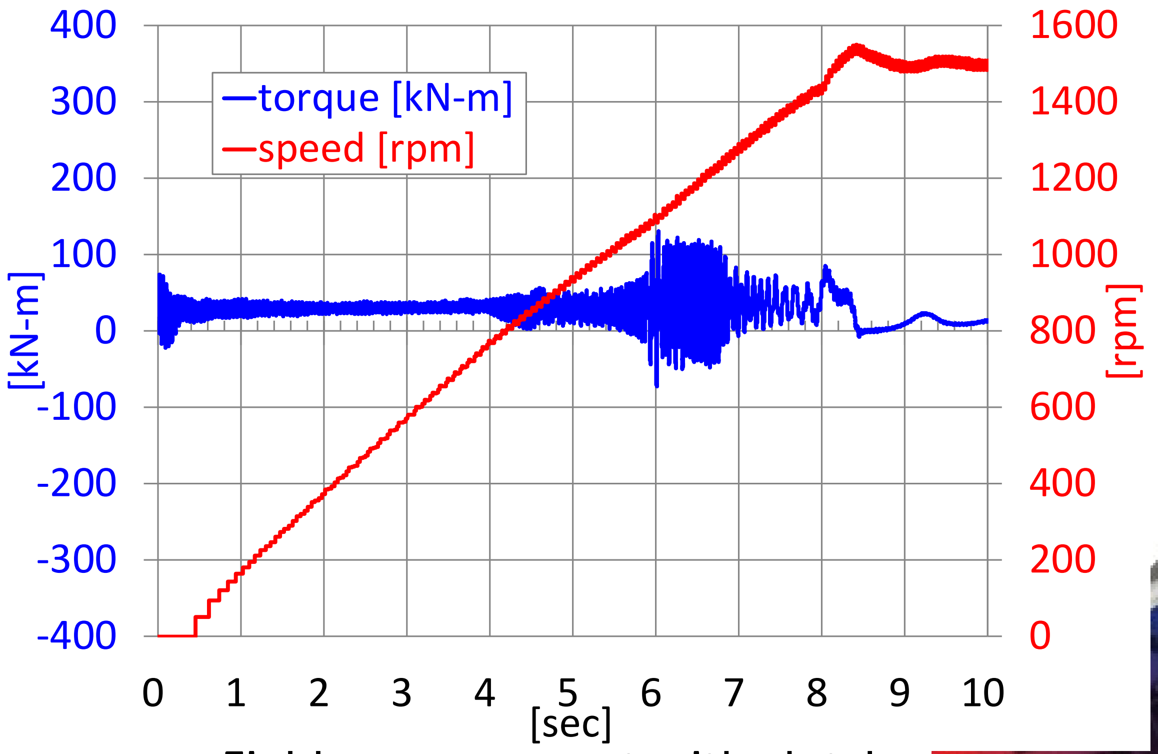
Confirmation of clutch effectiveness

- Prediction by a detailed torsional analysis (125 kN-m = 2.6 p.u.) was promising
- Field strain gauge measurements after the clutch installation (131 kN-m = 2.7 p.u.) verified its effectiveness (peak torque with less than 10% error from the target figure was achieved)

→ After these confirmations, controlled slip clutch was adopted as the permanent countermeasure



Analysis reflecting clutch



Field measurement with clutch

Conclusion

Conclusion

- After installation of this clutch, no malfunction or abnormality has been observed in the compressor train
- The compressor train has been stably and reliably operating while having experienced a number of startups
- Controlled slip clutch is confirmed as a reliable and viable option to counteract the problems associated with excessive peak torque during synchronous motor’s startup
- Controlled slip clutch is a very attractive countermeasure option especially for field troubleshooting purposes, because only the driver side coupling hub needs to be modified without affecting anything else in the train while providing sufficient and reliable peak torque suppression effects

References

[1] Yeiser, C.W., Hutten, V., Ayoub, A. and Rheinboldt, R., 2006, “Revamping a Gas Compressor Drive Train from 7000 to 8000 HP with a New Synchronous Motor Driver and a Controlled Slip Clutch Mechanism”, Proceedings of the 35th Turbomachinery Symposium, Texas A&M University.

[2] Smith, P.J., 2017, “Strategies to Prevent Sudden Catastrophic Compressor Failures during Transient Operating Conditions”, Proceedings of the 46th Turbomachinery Symposium, Texas A&M University.

[3] Maier, M.D. and Studley, G., 2017, “Optimizing Component Selection in Synchronous Motor Compressor Trains Based on Technical and Financial Considerations”, Proceedings of the 46th Turbomachinery Symposium, Texas A&M University.

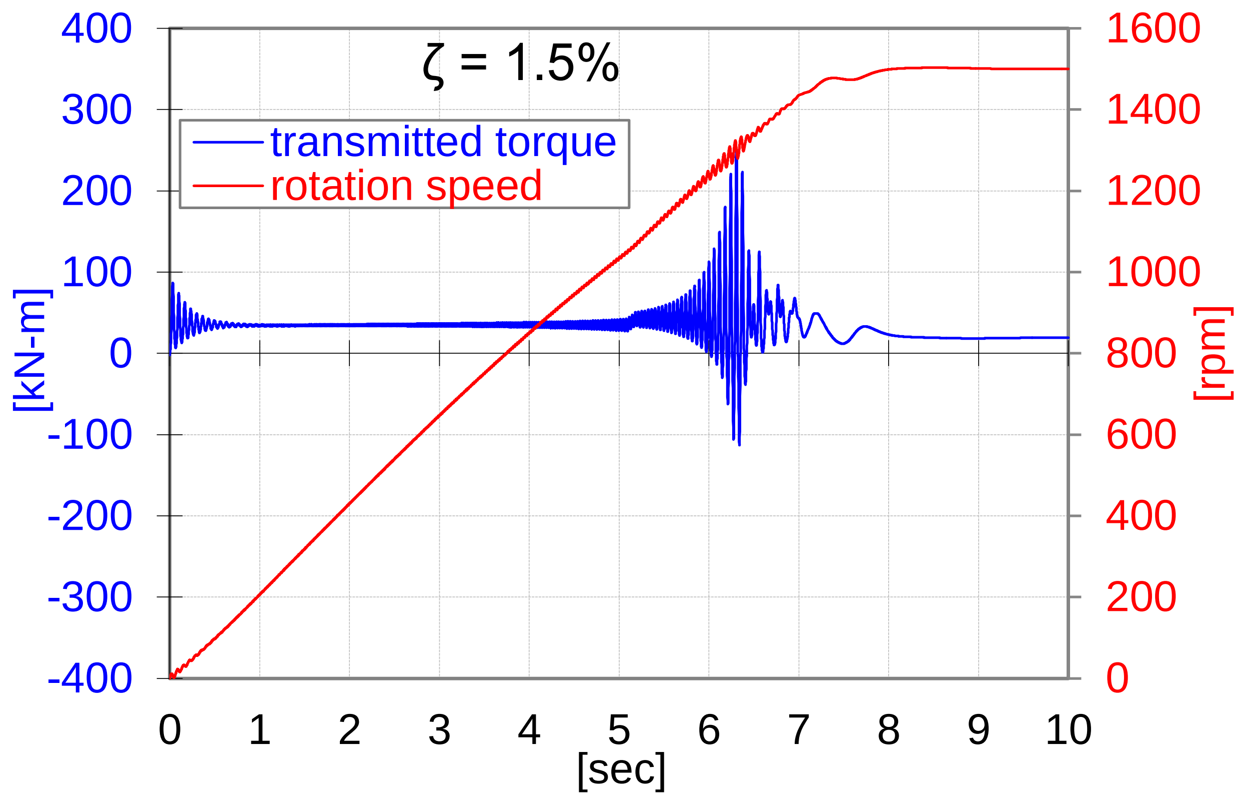
[4] Oscarson, G., Imbertson, J., Imbertson, B., and Moll, S., 1954, “The ABC’s of Synchronous Motors”, Electric Machinery Company

Addendum

Addendum: rubber block coupling

Supplementary to slide #17

- Rubber block coupling (rubber-in-compression coupling) reduces the peak torque due to variable stiffness and additional damping by rubber blocks
- Expected peak torque = 251 kN-m (5.3 p.u.)



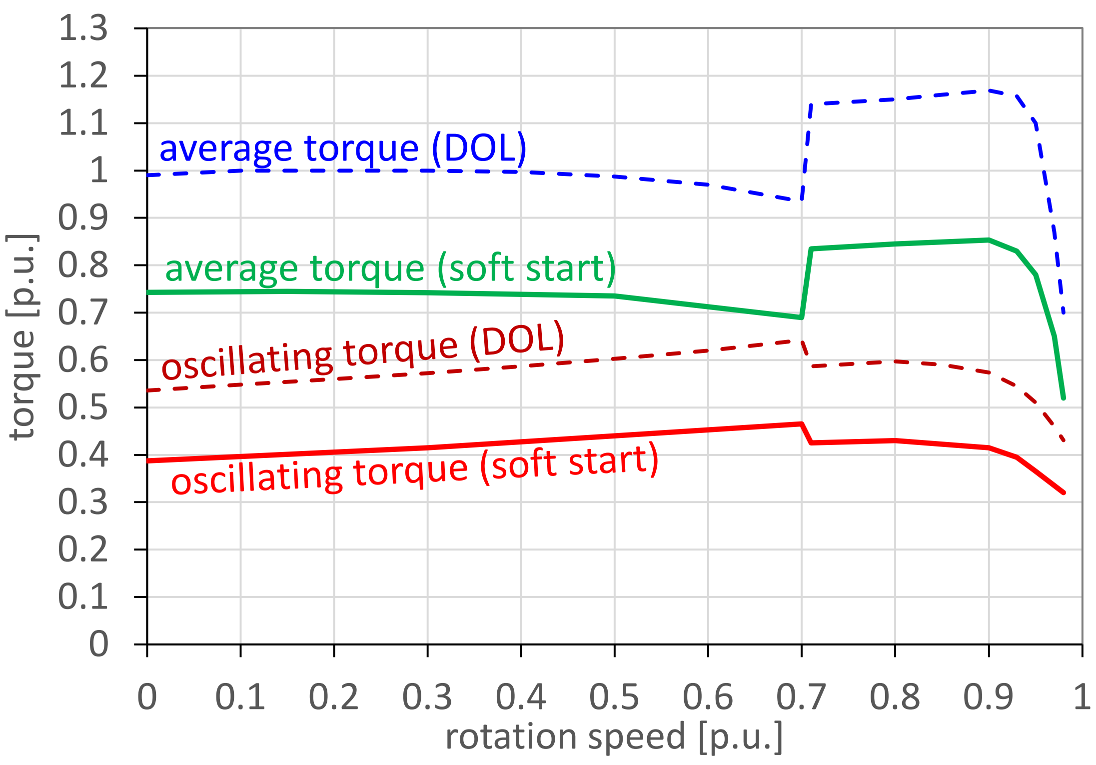
Analysis reflecting rubber block coupling

Note: ζ = 1.5% if not accounting for the additional damping and variable stiffness due to rubber blocks (This transient analysis actually reflects the additional damping (and also variable stiffness) by rubber blocks, whose stiffness and damping values are nonlinear based on angular deflection. As such, in this transient analysis, ζ is actually variable and bigger than 1.5%)

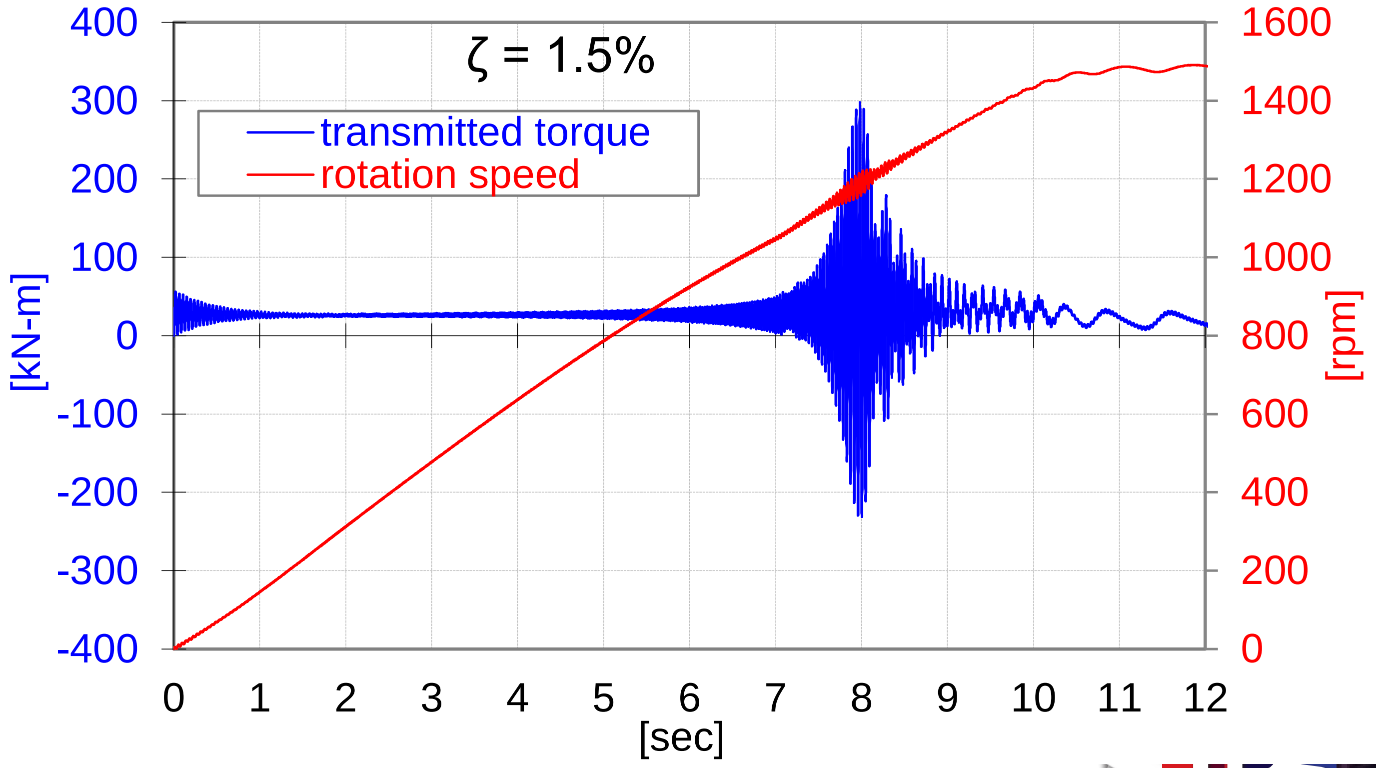
Addendum: step down transformer

Supplementary to slide #17

- Reduced voltage soft starter (step down transformer) (Note: this is different from VSD) lowers the oscillating torque magnitude, hence reduces the peak torque
- Expected peak torque = 300 kN-m (6.3 p.u.)
- Acceleration is slower because of lower average torque



Motor output torque curve (DOL and soft start)

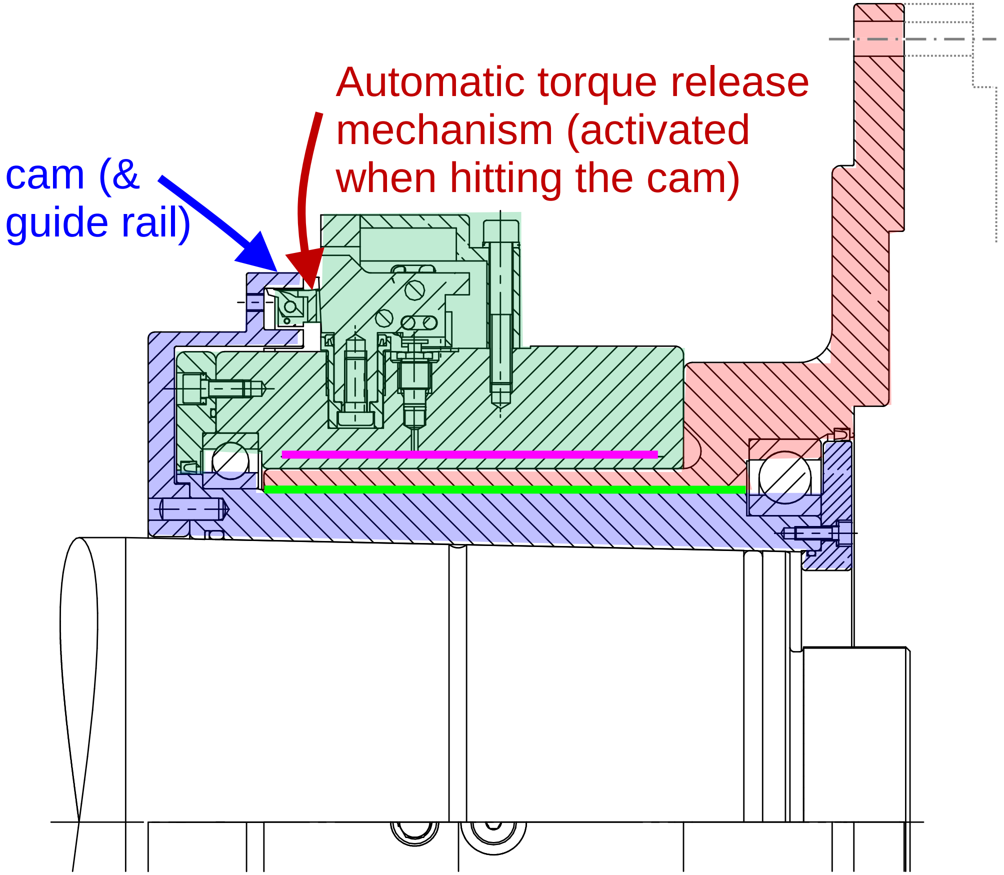


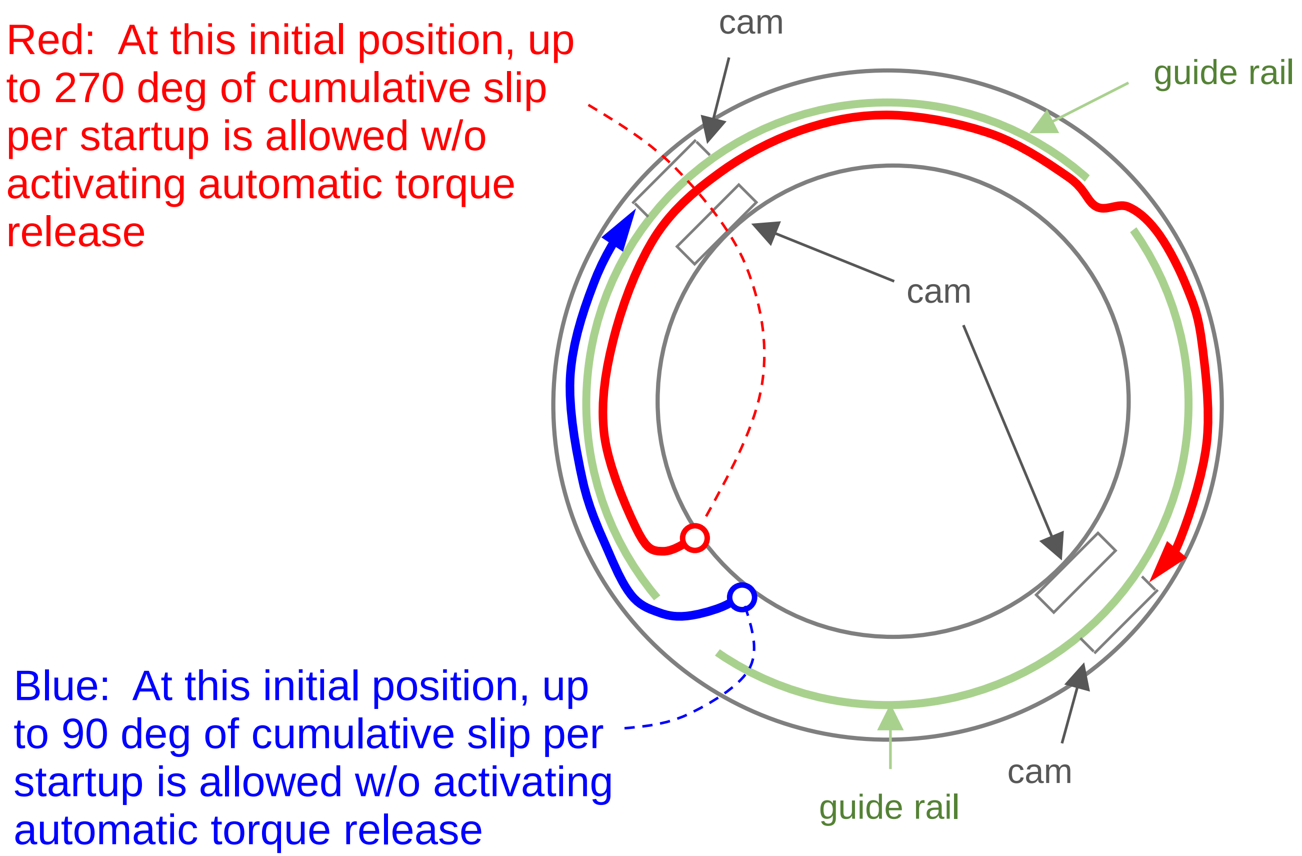
Analysis reflecting soft starter

Addendum: automatic torque release

Supplementary to slide #20

- Controlled slip clutch has a safety mechanism to automatically release the torque transmission in case of abnormally sustained slip
- In order not to activate the safety mechanism, cumulative slip per each startup needs to be less than 90 deg

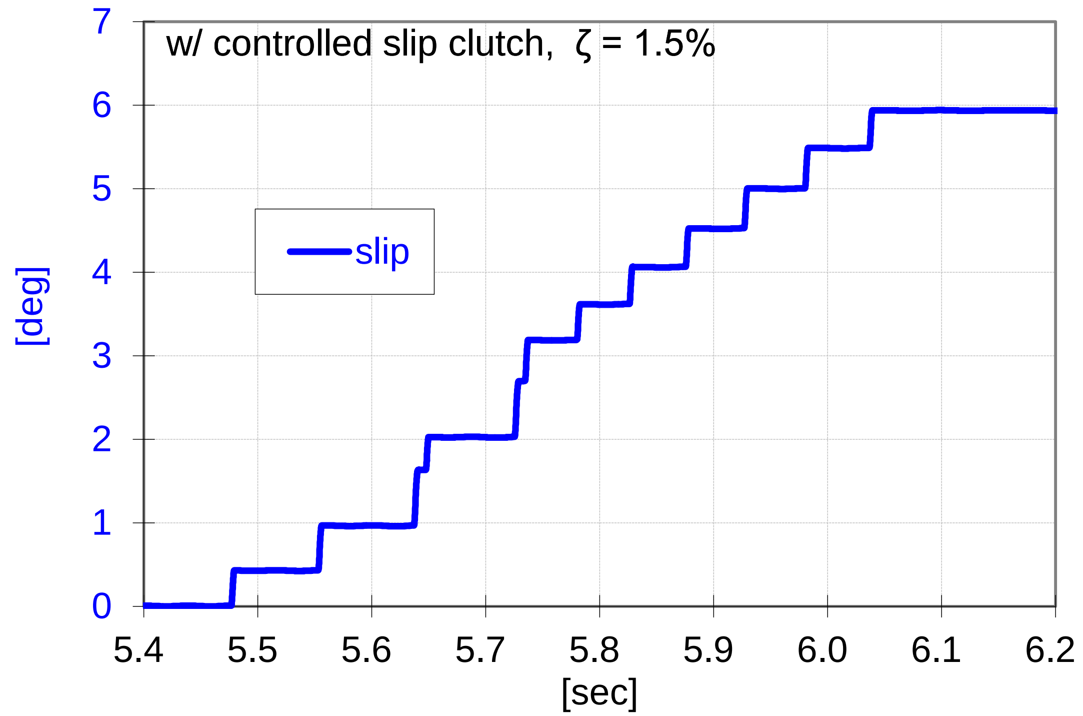




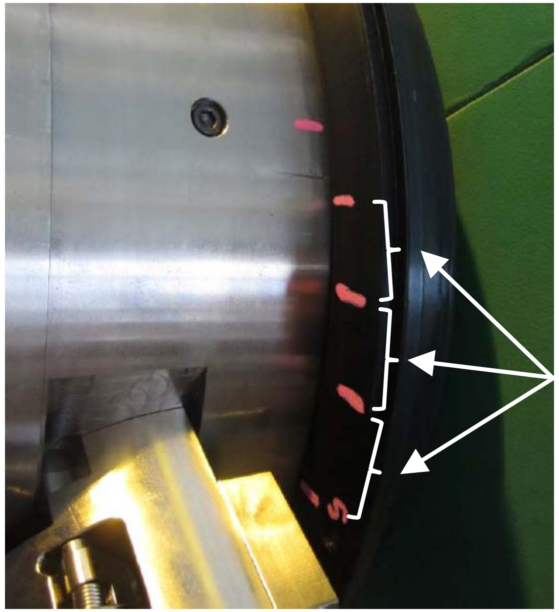
Addendum: slip amount confirmation

Supplementary to slide #22

- Expected cumulative slip per each startup was around 6 deg according to torsional analysis
- Actual cumulative slip by field observation was confirmed to be in line with the prediction
- Since the clutch can accommodate at least 90 deg of cumulative slip per each startup without activating automatic torque release, observed level of cumulative slip is sufficiently below threshold



Slip prediction by torsional analysis

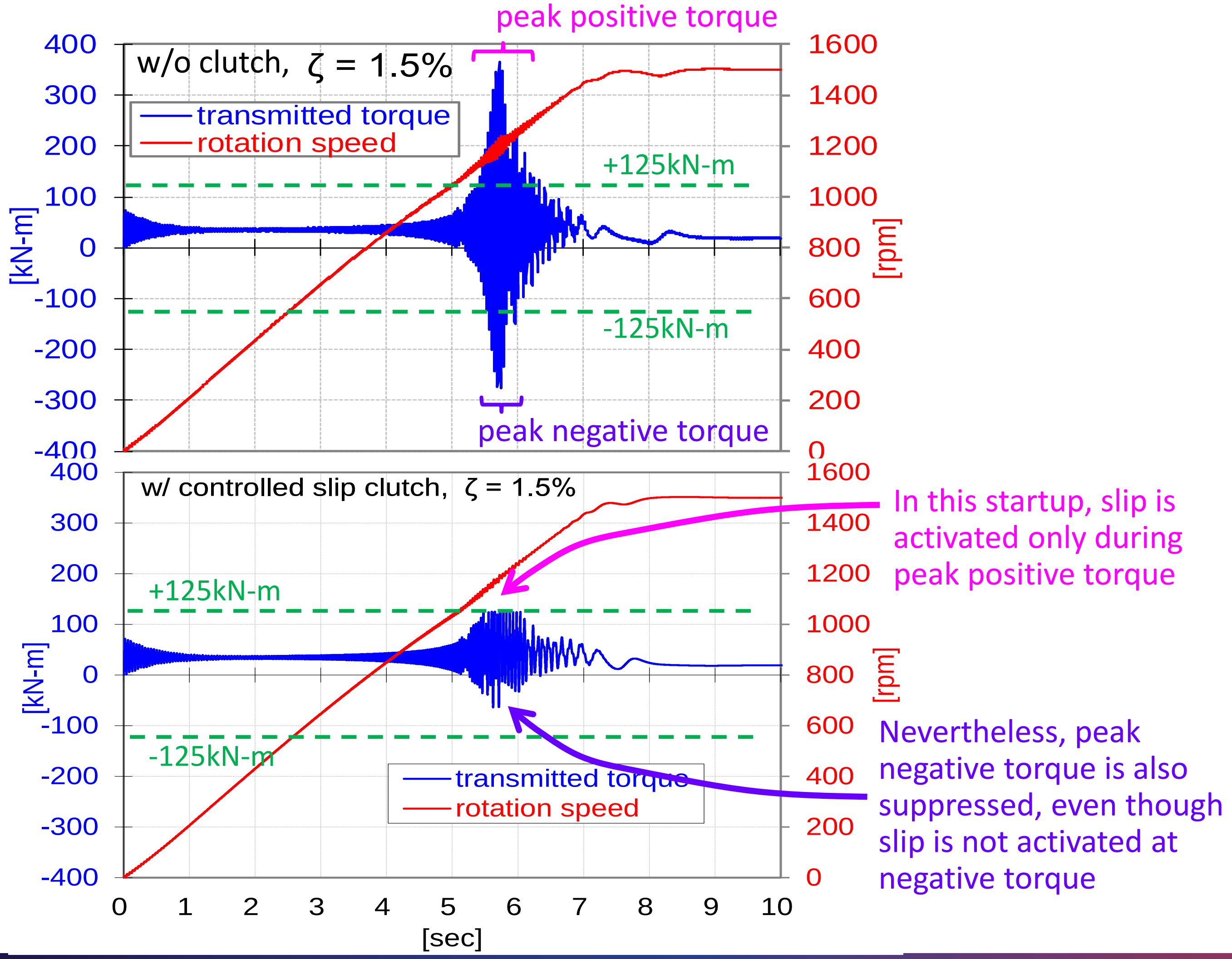


Field verification of slip

Addendum: suppression mechanism

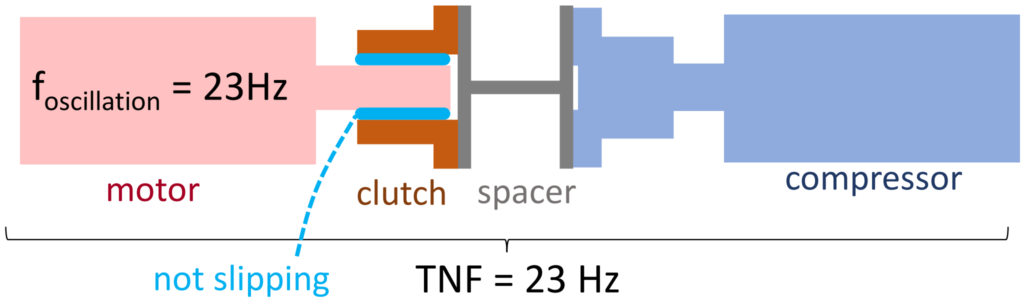
Supplementary to slide #22

- Due to slip during peak positive (+) torque, peak negative (-) torque is also suppressed
- This is because resonance condition is detuned during slippage



While clutch is ***not slipping***, TNF = 23 Hz

→ Resonance with $f_{oscillation}$ = 23Hz (@ 1155 rpm) occurs



At the moment when clutch is ***slipping***, TNF ≠ 23 Hz

→ Resonance with $f_{oscillation}$ = 23Hz (@ 1155 rpm) is detuned