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Rotordynamics Analyses of a Modified Hydraulic Power Recovery Turbine

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Presenter/Author Bios

Cheong Kai Chiat is the design engineer for Flowserve's Aftermarket Project Engineering Team. He obtained his Bachelor degree in Mechanical Engineering from Monash University. He had 6 years of experience in the pump engineering.

Edmund Lek is the engineering manager for Flowserve's Aftermarket Service & Solution Engineering team. He graduated with a Diploma in Mechanical Engineering from Singapore Polytechnic and worked in Flowserve for 23 years. He established the Central Engineering team in Asia Pacific region to provide aftermarket engineering solutions for pump upgrade and rerate.

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The Outline

- The Problem Statement
- The Machine Information
- The Torsional Model & Assumption
- The Torsional Response & Results
- The Tuning Methods
- The Field Validation
- The Conclusion

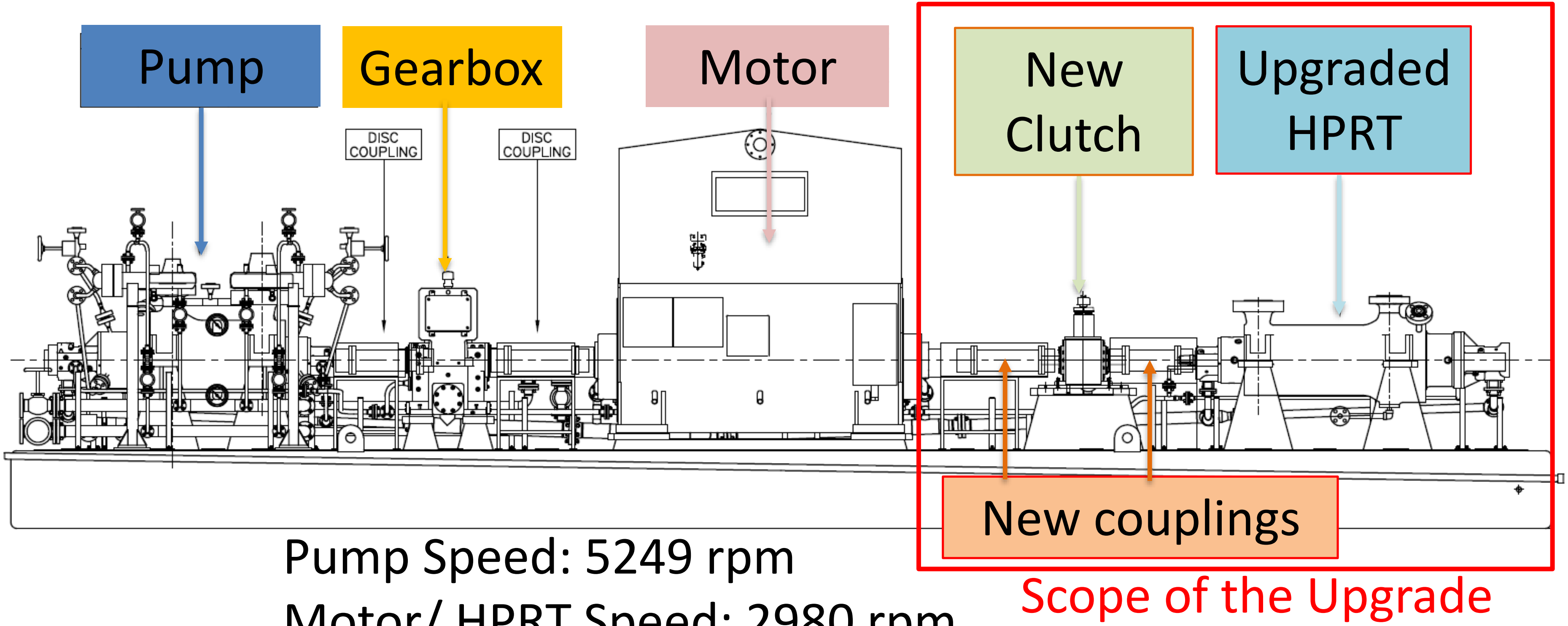
The Problem Statement

- A multistage HPRT was upgraded to meet the new operating requirement of the refinery plant in China.

- The rotordynamics analyses revealed the possible presence of 1st torsional resonance mode for the upgraded design, which could lead to the premature failure of the machine train.

- To resolve this, various methods were considered to tune the 1st torsional mode away from its design operating speed range.

The Machine Train



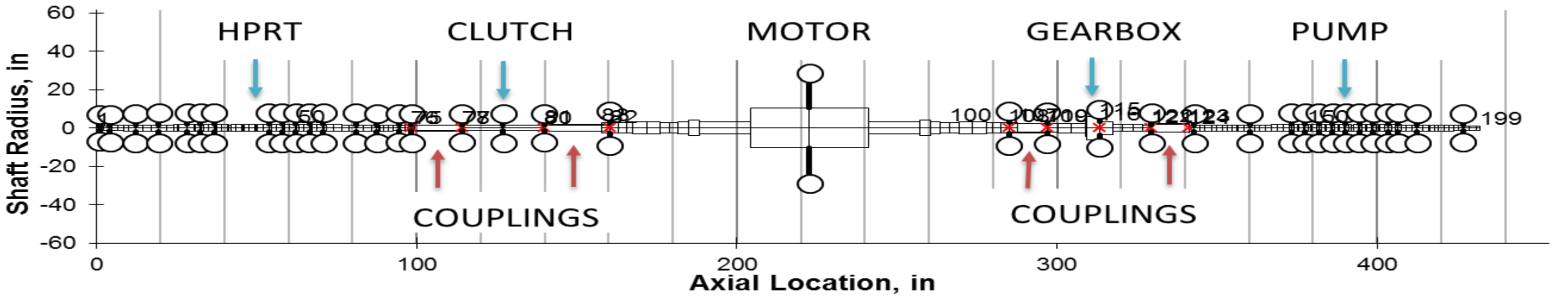
Pump Speed: 5249 rpm

Motor/ HPRT Speed: 2980 rpm

Gearbox ratio: 1: 1.7561

Power: 1500kW

The Torsional Model



- Model with the 35 stations of lumped mass & elastic beam.
- Iterative Holzer method to solve the differential equations.
- Component looseness effect was investigated in flexible model.
- Two scenarios were analyzed:
  a) With clutch disengaged (HPRT is not running)
  b) With clutch engaged

The Model Assumption

- Rotor: forcing torque was constant and steady.
- Shaft: Small deformation theory applied.
- Gear: wheels & teeth were infinitely stiff.
- Gearbox: modelled as a reduced single shaft.
- Torsional mass moment of inertia values were provided by individual component vendors.

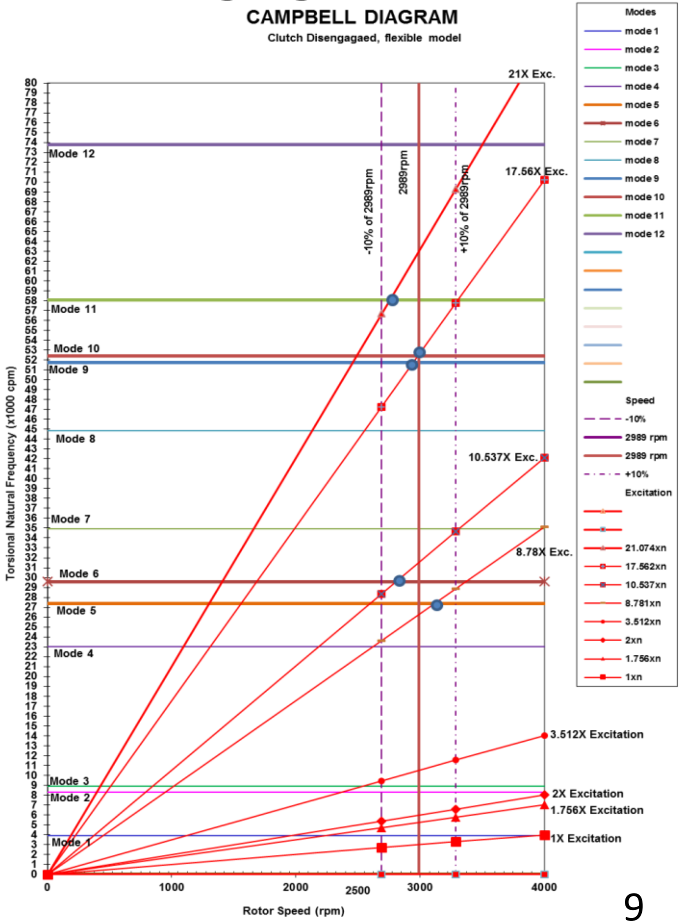
The Stiff vs Flexible Model

Flexible model	Stiff model
Including fluid in impellers	No fluid in impellers
Shrink-fit of impellers and sleeves does not added to the shaft torsional stiffness	40% of diameter of impellers' hub and sleeves are added to shaft torsional stiffness
2/3 portion of shaft overlapped at coupling hubs assumed to twist freely	1/3 portion of shaft overlapped at couplings hubs assumed to twist freely

Torsional Analysis Result – Clutch Disengaged

- Torsional response was calculated for critical speeds lie within the 10% separation margin.
- Analysis indicated that the component stresses were below endurance limit.

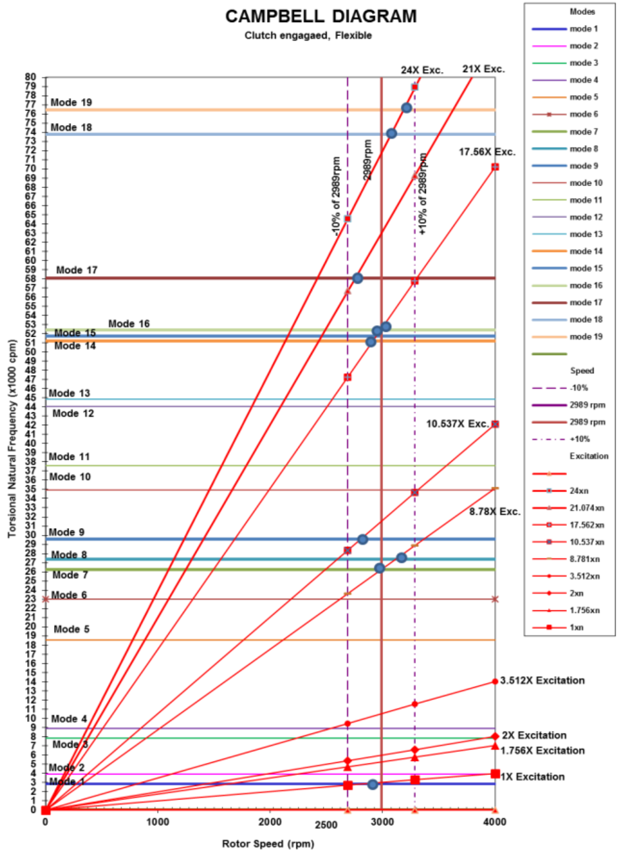
Mode shape	Natural frequencies, CLUTCH DISENGAGED			
	Flexible model		Stiff model	
	Critical Speed	Separation margin	Crit. Speed	Separation marg.
1	3875	>10 %	4140	>10 %
2	8307	>10 %	8331	>10 %
3	8912	>10 %	9419	>10 %
4	23042	>10 %	25006	-4.73 % at 8.781 x RPM
5	27349	4.2 % at 8.781 x RPM	27461	4.62 % at 8.781 x RPM
6	29593	-6.04 % at 10.537 x RPM	32081	1.86 % at 10.537 x RPM
7	34958	>10 %	37233	>10 %
8	44839	>10 %	50266	-4.25 % at 17.562 x RPM
9	51719	-1.48 % at 17.562 x RPM	53823	2.53 % at 17.562 x RPM
10	52417	-0.15 % at 17.562 x RPM	55108	4.98 % at 17.562 x RPM



Torsional Analysis Result – Clutch Engaged

- The predicted 1st critical torsional speed was very close to the motor running speed. Further review is required.

Mode shape	Natural frequencies, CLUTCH ENGAGED			
	Flexible model		Stiff model	
	Critical Speed	Separation margin	Crit. Speed	Separation marg.
1	2871	-3.95 % at 1 x RPM	2989	0 % at 1 x RPM
2	3880	>10 %	4144	>10 %
3	7864	>10 %	7999	>10 %
4	8910	>10 %	9414	>10 %
5	18524	>10 %	20236	>10 %
6	23042	>10 %	25006	-4.73 % at 8.781 x RPM
7	26234	-0.05 % at 8.781 x RPM	26650	1.53 % at 8.781 x RPM
8	27351	4.2 % at 8.781 x RPM	28372	-9.92 % at 10.537 x RPM
9	29593	-6.04 % at 10.537 x RPM	32083	1.86 % at 10.537 x RPM
10	34958	>10 %	37233	>10 %

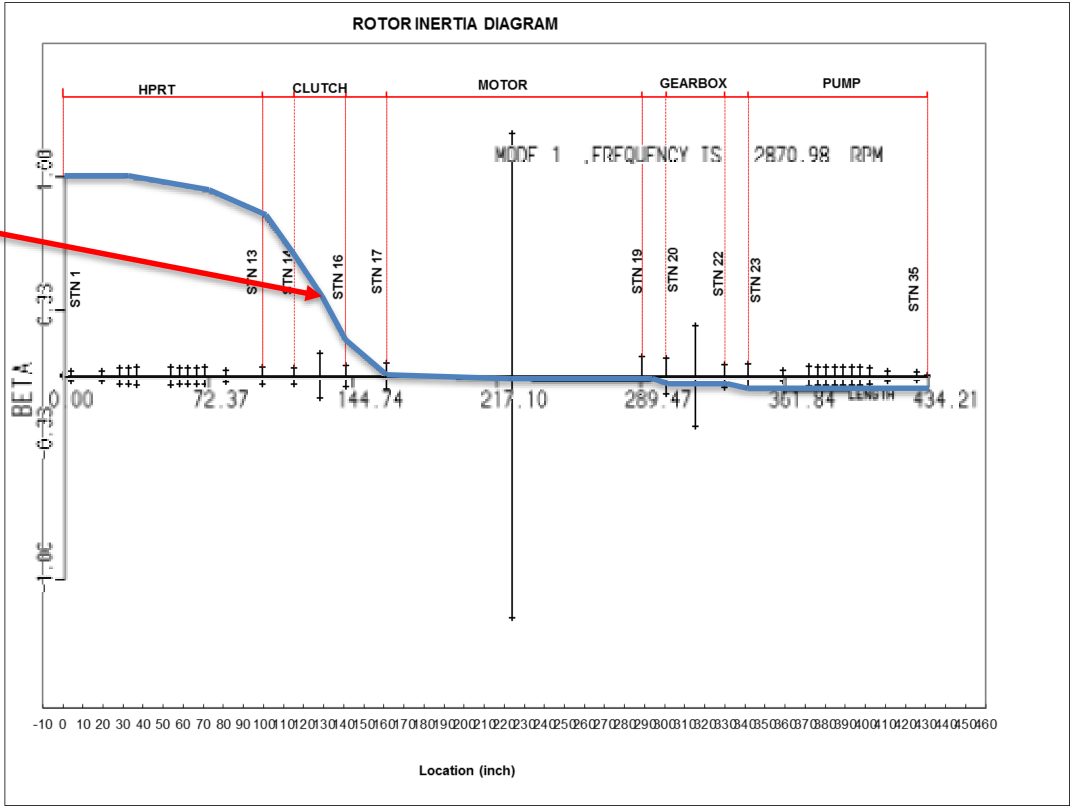


Torsional Analysis Result – Clutch Engaged

1st mode (clutch engaged), Natural frequency = 2871 rpm

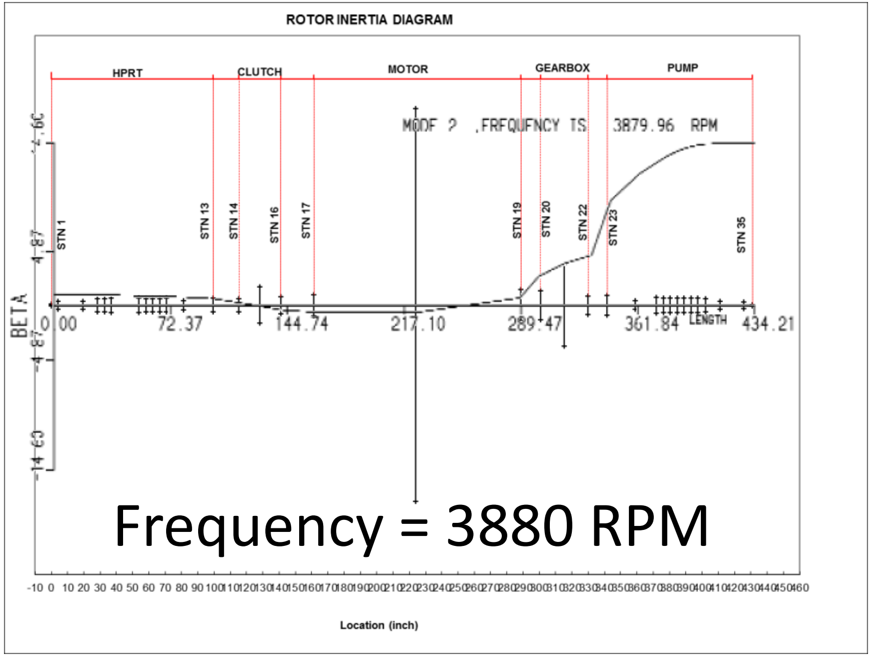
The highest change of angles

Clutch is the weakest link

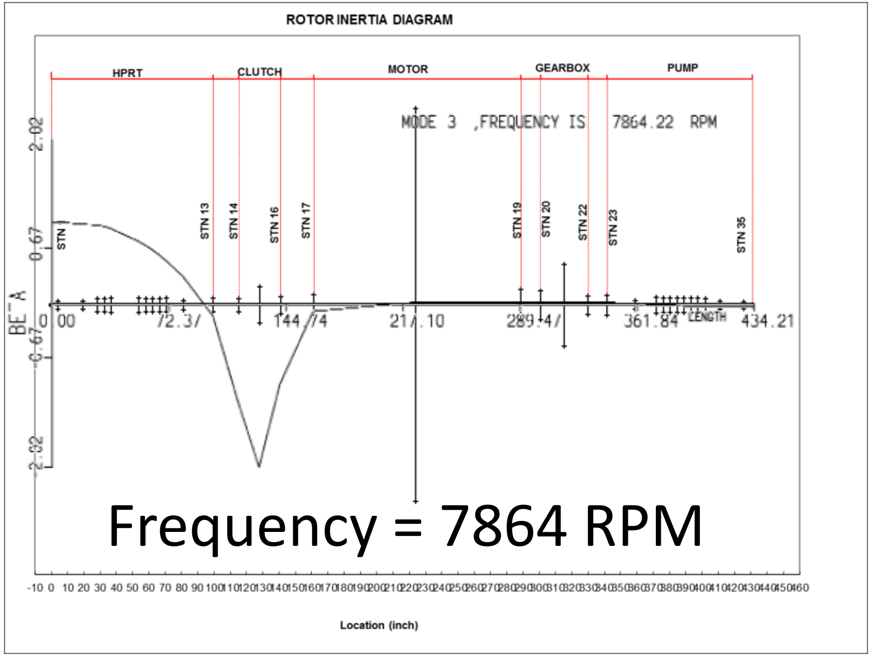


Torsional Analysis Result – Clutch Engaged

2nd mode



3rd mode



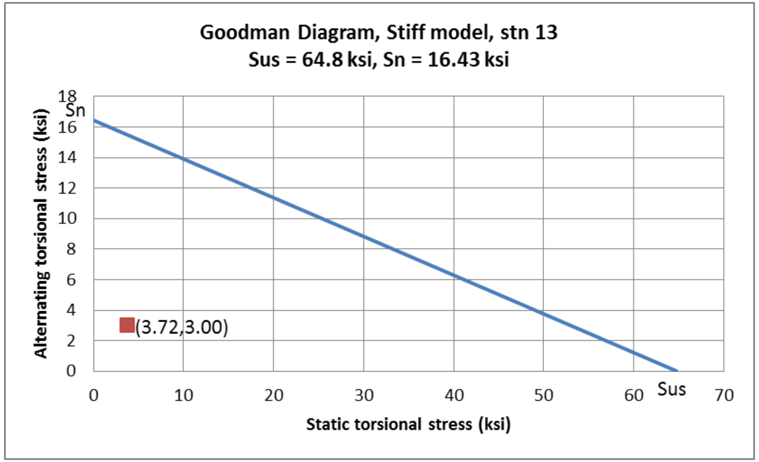
Torsional Response Assumption – Clutch Engaged

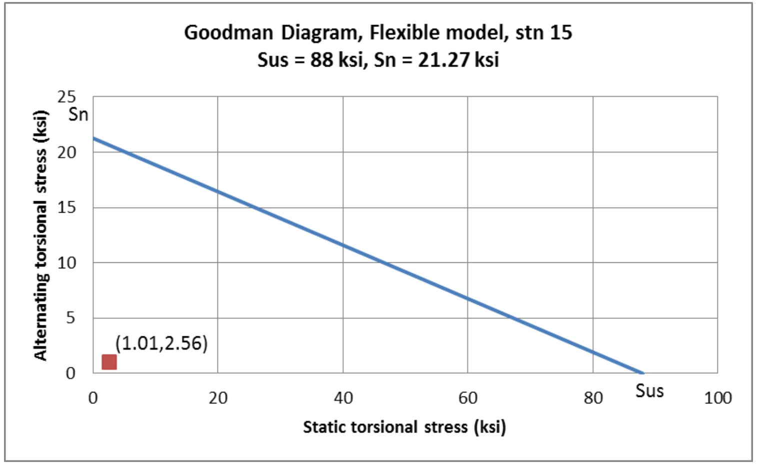
- Damping factor: 0.5% of critical damping.
- Dynamic torque:
  a) Impeller: 4% of static torque @ vane pass frequencies.
  b) Gearbox: 1% of static transmitted torque at 1x LS shaft speed and 0.5% of static transmitted torque at 2x LS shaft speed.
  c) Couplings: 1% of static transmitted torque at 1x and 2x running speed.
- Allowable stresses included fatigue stress concentration factor.

Model	Station	Station name	Allowable Stress	
			Sn	*Sus*
			ksi	ksi
Flexible	15	COMPLETE CLUTCH	21.27	88
Stiff	13	COUPLING PRT OUT	16.43	64.8

Torsional Response Result – Clutch Engaged

- Calculated stresses were within fatigue limit.
- However, a better design is needed to **avoid resonant condition** in steady-state operation for high cycles load.





flexible model:

safety factor = 13.0

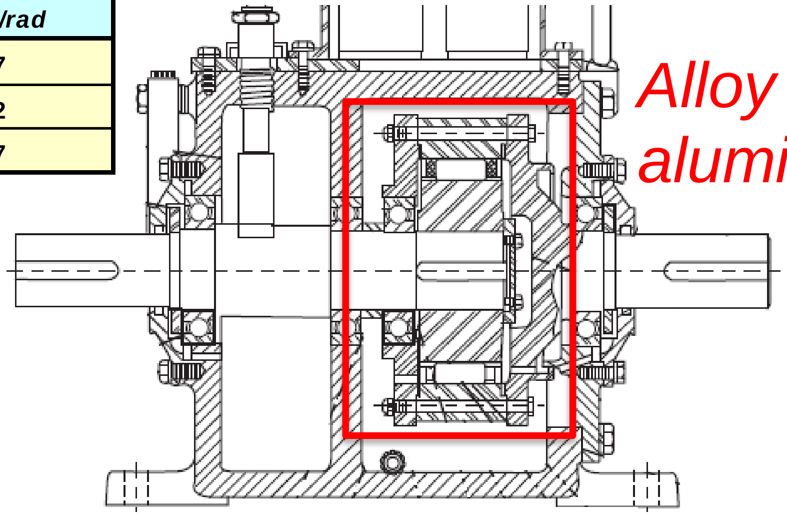
stiff model:

safety factor = 4.1

Tuning Torsional Natural Frequencies – Method 1

- Increase clutch torsional stiffness will shift the frequency up.
- Changing the clutch stiffness is costly, as it does not have any rubber elements.
- Upsizing the clutch has a long lead time constraint.

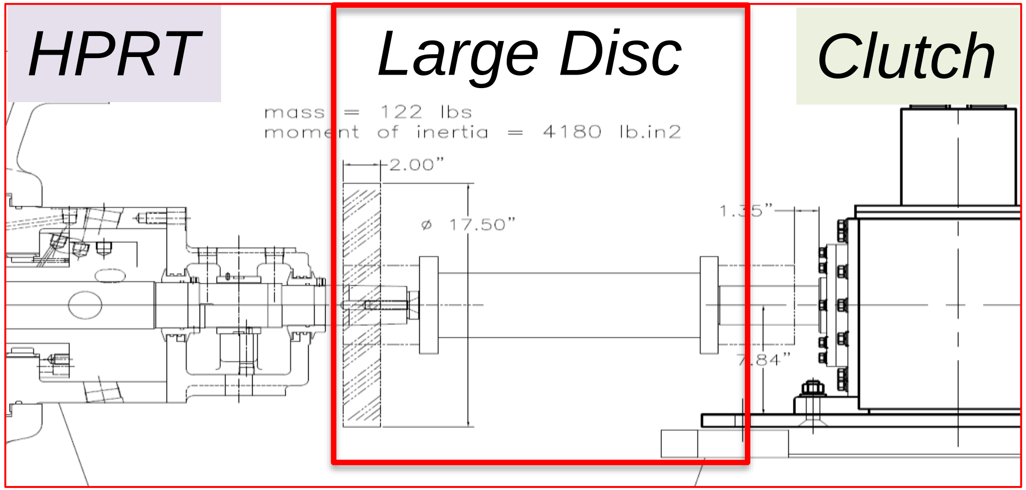
Component	*Moment of inertia* $lb.in^2$	*K torsion* *M lb.in/rad*
Coupling clutch to motor	888.5	4.27
Coupling clutch to HPRT	410.0	2.72
Clutch	1604.0	1.57



Tuning Torsional Natural Frequencies – Method 2

- Increase moment of inertia near both sides of clutch.
- Attach a large disc (large diameter in comparison to length).
- This would reduce 1st mode frequency to 2666 rpm.
- Challenge: Very costly to accommodate the larger disc.
- Heavy overhanging mass could also affect rotor lateral stability.

Component	*Moment of inertia lb.in^2*	*K torsion M lb.in/rad*
Coupling clutch to motor	888.5	4.27
Coupling clutch to HPRT	410.0	2.72
Clutch	1604.0	1.57



Tuning Torsional Natural Frequencies – Method 3

- Reduce torsional stiffness of couplings at both clutch ends.
- Iterative process to find a suitable stiffness value of couplings.
- 1st torsional frequency now has > 10% separation margin.
- This solution was adopted by customer.

Component	Moment of inertia lb.in ^{2}		K torsion M lb.in/rad	
	Before	After	Before	After
Coupling clutch to motor	888.5	947.6	4.27	2.41
Coupling clutch to HPRT	410.0	323.0	2.72	1.77
Clutch	1604.0	1604.0	1.57	1.57

Mode shape	Natural frequencies, CLUTCH ENGAGED			
	Flexible model		Stiff model	
	Critical Speed	Separation margin	Crit. Speed	Separation marg.
1	2572	>10 %	2675	>10 %
2	3871	>10 %	4143	>10 %
3	6987	>10 %	7100	>10 %
4	8895	>10 %	9414	>10 %
5	18635	>10 %	20535	>10 %

Field Validation

- User feedback: The modified HPRT was running well at desired duty since August 2017.
- Challenge: Pure torsional vibration could not be detected by the accelerometer and proximity probe.
- Field test using strain gauge telemetry system to measure the dynamic torque at the coupling spacer was planned.
- Site validation data would be attached when available later.

Lesson learned

- Essential to perform the torsional analyses whenever there is any change in the component of machine train.
- Changing the coupling torsional stiffness is a more effective and commercial viable solution.
- Effect of component looseness could have significant impact on the calculated torsional frequencies and must be included in the analyses.

Questions?