TORSIONAL CRITICAL SPEED ANALYSIS

of

SIEMENS 5810S CGZ MOTOR
REXNORD/THOMAS AMR-500 COUPLING
ARIEL MODEL JGE/2 COMPRESSOR

TRIAD
01003

SUBMITTED TO:

SCFM COMPRESSION SERVICES CO.
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DESCRIPTION

Motor: Manufacturer: Siemens, 400 HP, 894 RPM, Frame 5810S, Type CGZ, 4160V/3Ph/60Hz, with 3.875” Dia. Keyed Drive Shaft

Coupling: Manufacturer: Rexnord/Thomas, Dry Flexible Disc, Size 500, Type AMR, with Carbon Steel Hubs and SS Shims, without a Hub for the Compressor Shaft

Flywheel: 20.5” diameter by 3.5” thick with taper-lock bushing

Compressor: Ariel, Model JGE/2, with 5.000” Dia. Drive Shaft

RESULTS

The original coupling for the unit was an AMR 500 without a flywheel. With it, the first torsional critical coincided with the 7th order harmonic of compressor speed and was unacceptable. Since excess service factor was available, an AMR 450 coupling was tried. It also resulted in the first torsional critical coinciding with the 7th order harmonic. At this point it was decided to size a flywheel adapted for direct bolting of the coupling shim pack and eliminate the compressor hub. Thus, for the system with a 20.5” diameter flywheel, 3.5” thick and an AMR 500 coupling (without the compressor hub), the train torsional system natural frequencies and API 618 Code checks are as follows:

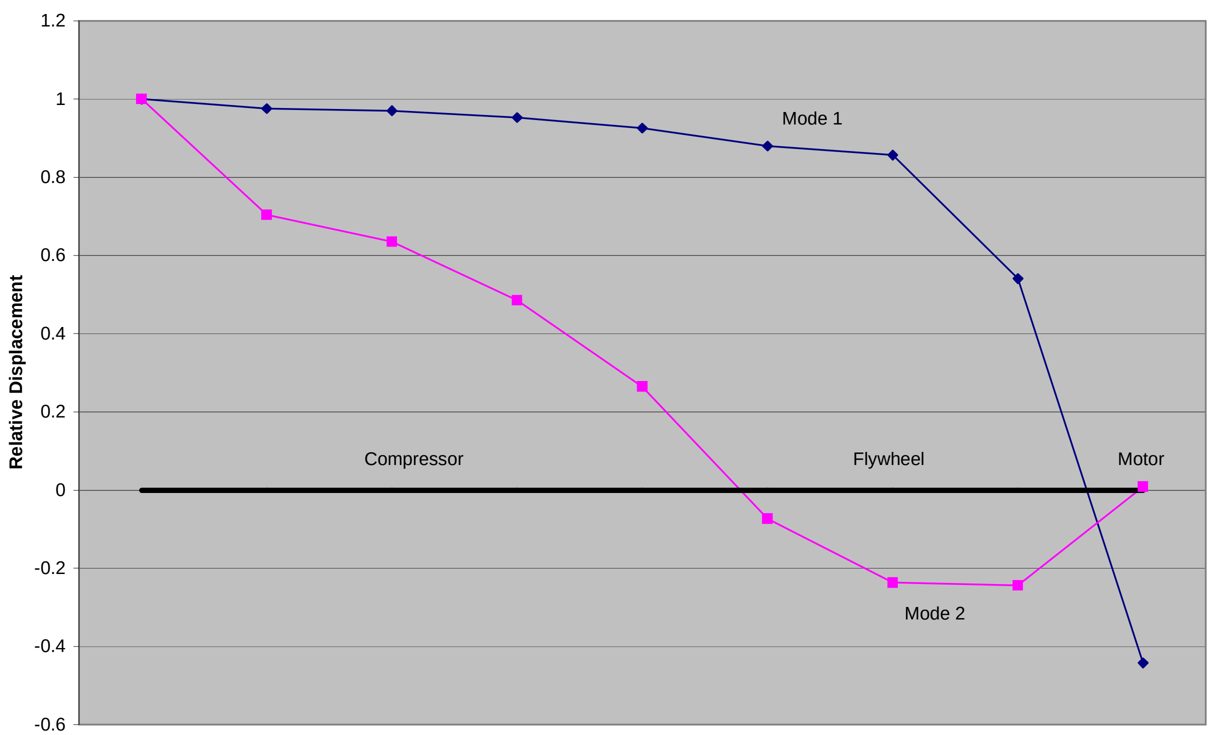
Mode	Frequency	Check condition	Margin	Acceptance Criteria
1	81 Hz	1x compressor speed or 14.9 Hz	443% above	±10%
		2x compressor speed or 29.8 Hz	172% above	±5%
		3x compressor speed or 44.7 Hz	81% above	±5%
		4x compressor speed or 59.6 Hz	36% above	±5%
		5x compressor speed or 74.5 Hz	9% above	±5%
		6x compressor speed or 89.4 Hz	9% below	±5%
		7x compressor speed or 104.3 Hz	22% below	±5%
		8x compressor speed or 119.2 Hz	32% below	±5%
		9x compressor speed or 134.1 Hz	40% below	±5%
		10x compressor speed or 149.0 Hz	46% below	±5%
		1x electric power freq. or 60 Hz	35% above	±10%
		2x electric power freq. or 120 Hz	33% below	±5%
2	287 Hz	10x compressor speed or 149.0 Hz	93% above	±5%

Per API 618 Reciprocating Compressor Code, torsional natural frequencies of the compressor-driver system shall not be within 10 percent of any shaft speed in the rotating system, nor within 5 percent of any other multiple of operating shaft speed in the rotating system up to and including the tenth multiple. Additionally, for motor-driven compressors, torsional natural frequencies shall be separated from the first and second multiples of the electrical power frequency by the same ranges.

The 1st critical is positioned midway between the 5th and 6th harmonic and the 2nd critical is considerably above the 10th order. Hence, the above critical speeds are sufficiently removed from any significant torsional excitation source and meet API 618. From a torsional standpoint, the above critical speed margins are sufficient to prevent any torsional resonant conditions and will give low torsional alternating stresses in train components.

The torsional mode shapes are shown below.

Thomas AMR-500 Coupling w/Flywheel - Modal Response



DISCUSSION

The standard torsional analysis of turbo-machinery is the calculation of undamped torsional natural frequencies and mode shapes for the system composed of all coupled rotating elements. Since most linear component equipment elements are considered to have a low level of system torsional damping, the undamped natural frequency calculation accurately predicts the actual train natural frequencies without the added complexities and uncertainties of estimating levels of torsional damping.

The undamped natural frequencies of torsional oscillation were calculated using the finite element method. In this approach the components of the torsional system are modeled as discrete structural elements with inertial masses. By performing an eigenvalue analysis of the mass and stiffness matrix, characteristic values and vectors are produced which represent the natural frequencies and mode shapes of free vibration.

Each major machinery element is considered a separate oscillating disk in the system. Stiffness of shaft portions between disks are calculated taking into account penetration of the shaft into an adjacent section of larger diameter when applicable.

The torsional idealized system is as follows:

DISK	INERTIA WR^2 ($lb\text{-}in^2$)	STIFFNESS K ($lb\text{-}in/rad \times 10^6$)
Aux Drive End	550.8	
Compressor Shaft		15.7
Main #2 CL	1046.3	
Compressor Shaft		156.6
Throw #2 CL	2,498.5	
Compressor Shaft		161.0
Throw #1 CL	2,498.5	
Compressor Shaft		156.6
Main #1 CL	1,103.9	
Compressor Shaft		109.0
Shoulder Stop	608.3	
Shaft Extension		222.8
Flywheel + ½ Coupling	18,138.0	
AMR 500 Coupling less 1 hub		49.7
½ Coupling	2,255	
Motor Shaft		16.8
Motor Rotor	55,872	
Motor Shaft Extension		nil

All compressor train components inertia and stiffness were provided Ariel, the coupling by Rexnord Engineering, and the motor by Siemens. The compressor stiffnesses and inertia were adjusted for shaft penetration factors and the flywheel inertia was calculated. Also, the coupling stiffness was adjusted for direct bolting to the flywheel.

<u>RECOMMENDATIONS</u>

Use the Thomas AMR-500 coupling with a 20.5” diameter by 3.5” thick flywheel on this application. With this coupling and flywheel, no problems are anticipated with torsional resonant conditions affecting the coupling or shafts.

Thank you for allowing us to assist in this project. If any additional information is required, please contact us.

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