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AN ENGINE DRIVEN, HIGH PRESSURE RECIPROCATING TYPE INJECTION GAS COMPRESSOR DESIGN

by
Gary A. Ehlers
Senior Engineering Manager
WorleyParsons
Monrovia, California
and
Kurt Haaland
Operations & Maintenance Manager
BP
Houston, Texas

Gary A. Ehlers is Senior Manager, Mechanical Engineering Department/Chief Engineer—Rotating Equipment Engineering Group at WorleyParsons, in Monrovia, California. He supervises activities related to the rotating equipment engineering on projects. The types of machinery responsibilities include centrifugal and reciprocating compressors, axial flow compressors, steam and gas turbines, centrifugal and reciprocating pumps and gas expanders. He manages the Mechanical Engineering Department for WorleyParsons West Coast, which includes rotating machinery engineering, heat transfer, pressure vessel engineering, materials selection, mechanical utilities, materials handling, and power engineering groups.

Mr. Ehlers has a B.S. degree (Industrial Management, 1968) from Gannon University and a B.S. degree (Mechanical Engineering, 1980) from California State University, Los Angeles. He is a member of ASME, and has been a member of the Mechanical Equipment subcommittee of API for the last 12 years representing WorleyParsons. In addition, Mr. Ehlers is a registered Professional Engineer in the States of California, Texas, and Pennsylvania.

ABSTRACT

A 10,000 psi high pressure natural gas injection compressor train was procured, manufactured, tested and installed in the Boquerón production facility in Eastern Venezuela. The injection machinery consisted of three (3) parallel slow speed reciprocating gas compressors each driven by a natural gas driven internal combustion engine through a speed reducing gearbox. The equipment had been purchased in an early phase of the project by the original oil company, so it could not be changed. Due to the critical timing of compression required to maintain reservoir pressure and incremental oil recovery, the equipment could not be changed without large financial consequences. After preliminary review of the purchased equipment, it was recognized early by the engineer, procure, and construct (EPC) design engineering team that the compressor type and selection including the design were not the preferred arrangement. However, due to schedule and contract constraints the selection and design could not be significantly changed.

Although the compressor itself was considered properly selected there was considerable attention given to the drive train. This drive train consisted of the internal combustion engine, a gearbox, a steel torque shaft that connected the gearbox to the compressor and an intermediate bearing/bearing support. As the project progressed there were significant design improvements made to improve the safety, operation, and life of the compressor train. The initial design of the torque shaft indicated that it would fail every time the machinery experienced a shutdown. In order to overcome such a failure the drive train included a special coupling with shear bolts that would shear every time the compressor train had a shutdown, the function of which was to prevent the torque shaft from fatigue failure. Shearing bolts during every shutdown was not considered safe or acceptable operating practice.

During the course of the project, there were significant improvements made, first to the torque shaft, and then to the coupling connecting the gearbox to the torque shaft, then to the coupling between the engine and high speed gear pinion. The torsional characteristics were improved significantly with the upgrades and design improvements.

After installation, extensive field testing was conducted on each injection gas compressor to ascertain the torsional characteristics of the drive train, including stresses in critical components during the shutdown events. The data confirmed that the torque shaft was in fact designed for infinite life and would not fail from material fatigue as was feared initially.

During the design phases engineering consultants were contracted to perform torsional studies to confirm the torsional critical speeds initially predicted by the compressor manufacturer. After installation the same consultant used strain gauges to measure the torsional shear stresses and the torsional critical speeds.

There was an enormous amount of valuable engineering data generated during the design verification process and field testing. The authors feel that this information should be shared with others in the oil and gas industry when considering this type of machinery and machinery train for their high pressure injection gas compressor services or projects.

INTRODUCTION

This paper presents some of the significant engineering aspects and challenges of the high pressure gas injection compressors which were part of the Boquerón Field development project in Eastern Venezuela. The purpose of the project was to increase oil production from an existing oil field. The declining oil production required high pressure gas injection to increase reservoir pressure, improve sweep efficiency and produce incremental crude oil. High pressure, 10,000 psi gas injection compressors were selected for this function. Figure 1 shows the facility in an advanced state of construction. Initial estimates of oil production were around 25,000 to 30,000 bopd.

Figure 1. Construction of Boquerón Oil Field Facilities.

Figure 2 shows the schematic arrangement of Phase 1 for the gas injection system. The gas handling compression system included two intermediate pressure (IP) gas compressors and one high pressure (HP) gas injection compressor. Gas separation, gas cooling, and the liquid knockout facilities were also part of Phase 1.

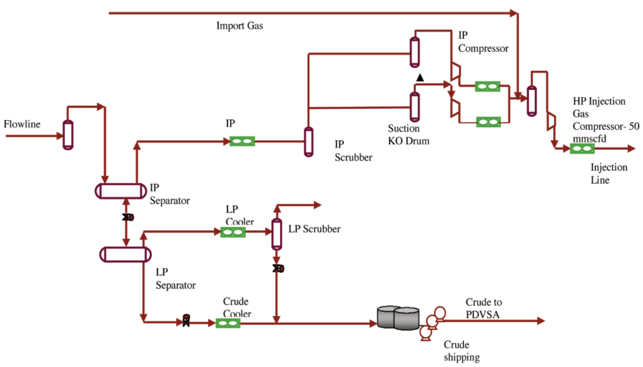

Figure 2. Process Arrangement of the Gas Injection System—Phase 1.

After the original operating oil company was sold, the new operating oil company contracted the engineering company for detail engineering and construction services. All major compression equipment had already been purchased and selected and was in manufacturing at that time. The engine driven high pressure injection gas compressors had engine drivers that had actually been delivered to the compressor packager's shop for both the IP and the HP injection gas compressors. After a detailed equipment review, there was much concern about using the HP injection gas compression equipment. Those involved in the project wanted to investigate gas turbine driven centrifugal gas compressors even though it was known that the experience level with low flow impellers was limited. Discussions centered on the risks associated with using inexperienced low flow impellers versus the existing engine driven reciprocating compressors with the associated intermediate gearbox and low speed torque shaft. Regardless, the previous team had selected and purchased the entire engine driven compression equipment and schedule constraints dictated that the EPC team was committed to construct the facility with this selected machinery.

Figure 3 shows the schematic arrangement for Phase 2 of the high pressure gas injection system. For this phase there are five high speed engine driven reciprocating gas compressors designated as the intermediate pressure compressors, and three slow speed engine driven reciprocating gas compressors designated as the high pressure compressors.

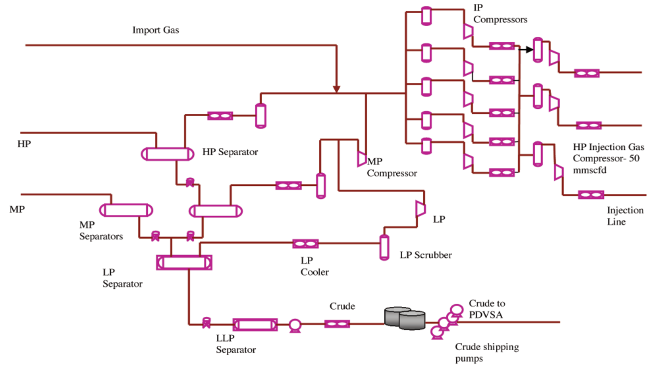

Figure 3. Process Arrangement of the Gas Injection System—Phase 2

The Uniqueness of the Project

The EPC engineering team recognized the challenges of the project as it involved compression of the separated natural gas and compression up to 10,000 psig and the consequent injection into the designated gas injection wells. All initial concerns were focused on the compressors. Most natural gas compressors are designed to handle gas at much lower pressures. Injection gas compressors designed to inject the gas at 10,000 psig are unique. The significant engineering concerns centered around containment of the gas (cylinders, packing, seals, valves, and check valves). Other concerns included prediction of the thermodynamic properties of the gas mixture as well as the limitations of the machinery when handling the gas with predicted properties.

Once the engineering team became familiar with the project there were numerous engineering issues identified that needed to be addressed. These complicated engineering design issues required approximately two years to grasp and correct. It should be noted that the engineering issues do not really reflect on the capabilities of the manufacturers or consultants initially hired. The initial design was not advanced to where the new engineering team felt comfortable because of the initial tight schedule. The engineering design of the torque shaft, gears, etc., was released in time to meet the critical project field schedule for injecting the high pressure gas back into the reservoir.

After some initial studies it was decided to proceed with modifications to only one HP gas compression train. After proving changes in design and operational testing of the first HP compression train, modifications could be made to the other two trains of HP gas compression equipment. The reasoning was any design or operational problems experienced on the first HP compression train would not be repeated for the other two HP compression trains. After field testing if the first train required modifications, it would be accomplished after installation of the second and third HP compression trains.

Phase 1 consisted of two IP booster compressors and one HP injection gas compressor drive train. There was a gap of approximately one year between Phase 1 and Phase 2 installation and commissioning due to the studies and changes made to the Phase 2 HP units. Phase 2 of the project included installation of the remaining three IP booster compressor trains and the two HP compressors. All this equipment included the associated pulsation dampeners, suction scrubbers/knockout drums and associated piping, foundations, etc.

Technical Resources

Due to the safety concerns around the handling of the high discharge/injection gas pressure nature of the project, a significant amount of attention was given to selection of specialists who actually had experience in designing facilities with such high pressures. Major concerns included the drive train itself and the pulsation analysis of the machinery and piping systems. API 618 Standard limits its guidance to approximately 5,000 psi discharge pressure. The project's approach involved utilizing experienced design professionals augmented by audits that were conducted by similar experienced firms. Due to the limited number of installations where reciprocating gas compressors were injecting gas at 10,000 psig discharge pressure, the experience was difficult to obtain. Therefore, the owner and engineering teams utilized the best engineering available recognizing that there was a certain risk. This was the only approach available due to the schedule driven nature of the project and the limited gas injection technology available to the team at that time.

Other high pressure compressor installations were designed using gas turbine driven centrifugal compression. The gas flow rates were much higher than the flow rate of the Boquerón injection compressor units.

In addition, the project needed to place the injection gas compression trains into operation as soon as possible as the owner was already producing crude oil the rate of which depended on maintaining reservoir pressure, and that could only be accomplished by injecting the high pressure gas.

The engineering/project team made extensive use of third party/outside engineering consultants. It was determined that the project would be beneficial if the designs were reviewed (audited) by engineering experts not connected with the project or owner. Valuable information was developed and their reports provided another level of assurance to the owners and operators. It was justified in consideration of the extreme pressures of the HP injection gas compression systems and the safety aspects that were not compromised.

Pulsation Designs Not Included

This paper does not address or discuss the pulsation/acoustic designs of the pulsation suppression system. There was considerable attention and effort paid to acoustic designs but is outside the scope of this paper.

DESCRIPTION OF THE HIGH PRESSURE INJECTION GAS COMPRESSOR TRAIN

The low pressure (IP) compression machinery consisted of a “high speed” balanced-opposed reciprocating gas compressor each driven by a medium speed, 1000 rpm internal combustion engine. The details of the selection and performance are shown in Table 1.

Table 1. Intermediate Pressure Compressor Details.

	1st Stage	2nd Stage	3rd Stage
Suction pressure, psia	361.00 psia	865 psia	1705 psia
Suction temperature, F	105 F	105 F	105 F
Suction compressibility, Z	0.9318	0.8308	0.6922
Discharge pressure, psia	935 psia	1757 psia	3236 psia
Discharge temperature, F	222 F	200 F	186 F
Discharge compressibility, Z2	0.9197	0.8413	0.8089
Gas flowrate, mmscfd	30.90 mmscfd	30.90 mmscfd	30.90 mmscfd
Compressor speed, rpm	1000 rpm	10000 rpm	1000 rpm
Speed range, rpm	800 – 10000 rpm	800 - 1000 rpm	800 – 1000 rpm
Compressor stroke, inches	6.00 inches	6.00 inches	6.00 inches
Cylinder diameter, inches	9.50 inches x 3	700 inches x 2	6.00 inches x 1
Stage hp, bhp	1693 bhp	1206 bhp	993 bhp
Rod Load, Compression, Lbs (gas+inertia)	-31,256 lbs	-43,953 lbs	-53,953 lbs
Rod Load, Tension, Lbs (gas+inertia)	28,771 lbs	30,958 lbs	46,960 lbs
Piston Rod diameter, inches	2.500 inches	2.500 inches	2.500 inches

The high pressure injection compression machinery consisted of a slow speed balanced-opposed reciprocating gas compressor (API 618 design) driven by a medium speed internal combustion engine. The natural gas fired engine utilized a speed reducing gearbox (API 613 design) with a specially engineered torque shaft connecting the gearbox with the injection compressor. The function of the torque shaft was to flex with the torque fluctuations and to maintain the gears in mesh.

Pulsation dampeners for both the fourth and fifth stages of the HP injection gas compressor were all specially designed spherical. The spherical design was used due to the high pressure gas handled.

The project included three HP injection compressors provided for the project, each one per the details shown in Table 2. These compressors are slow speed, balanced opposed reciprocating type and purchased per API 618 Standards.

Table 2. High Pressure Injection Compressor Details.

	4th Stage	5th Stage
Suction pressure, psia	3069 psia	5800 psia
Suction temperature, F	120 F	120 F
Suction compressibility, Z	0.796	1.0468
Discharge pressure, psia	5971 psia	10,100 psia
Discharge temperature, F	221 F	186 F
Discharge compressibility, Z2	1.0699	1.4402
Gas flowrate, mmscfd	51.44 mmscfd	51.45 mmscfd
Compressor speed, rpm	360 rpm	360 rpm
Speed range, rpm	324 – 360 rpm	324 – 360 rpm
Compressor stroke, inches	14 inches	14 inches
Cylinder diameter, inches	8.000 inches	7.125 inches
Stage hp, bhp	1912 bhp	2097 bhp
Piston speed, fpm	840 fpm	840 fpm
Maximum allowable frame load	180,000 Lbs	180,000 Lbs
Piston Rod diameter, inches	5.000	5.000

Figure 4 shows a simplified proportional-integral-derivative (P&ID) of one HP injection gas compressor. All three HP compressor trains are exactly identical as far as the compressor and driver train goes. There were changes, however, in the pulsation dampeners and suction knockout drums between Phase 1 and Phase 2. The details are described elsewhere in this paper. Figure 5 shows a diagram of the HP (high pressure) gas injection compressor trains.

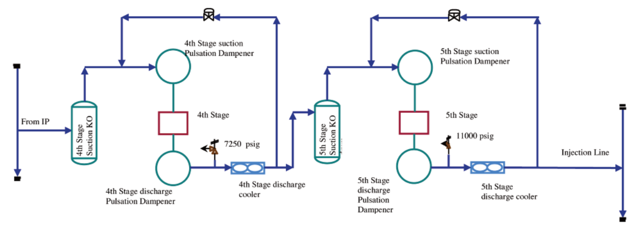

Figure 4. Simplified P&ID of the HP Injection Compressor Train.

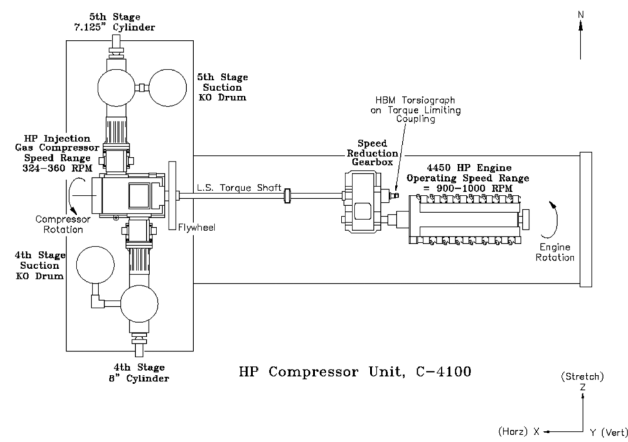

Figure 5. Schematic of HP Injection Gas Compressor. (Courtesy Engineering Dynamics Inc.)

THE INITIAL DESIGN

Each of the three HP injection gas compressors consisted of the following components, shown in Table 3. The gas was essentially sweet natural gas so materials selection was not a significant issue as it is for machinery handling sour (high H_2S) gas compositions.

Table 4 shows the major materials selected for the HP injection gas compressor project.

Table 3. HP Injection Gas Compressor Train.

(1)	Gas engine designed to produce 4450 hp at 1000 rpm
(2)	Speed reducing gearbox, double helical type
(3)	Slow speed reciprocating gas compressor, rated at 50 mmscfd flowrate
(4)	Torsional resilient high speed coupling between engine and gearbox
(5)	26 foot torque shaft between gearbox and HP injection gas compressor
(6)	Torque limiting coupling on outboard end of torque shaft.
(7)	Intermediate shaft bearing on torque shaft.

Table 4. Major Material Selections.

Part or System	Materials
Compressor	
Cylinders	Forged steel (4th & 5th stages)
Valve Plates	420 ss / 17-4 PH SS
Crankshaft	Forged steel
Pulsation dampeners	Forged carbon steel
piping	Carbon steel, seamless
Torque shaft	Carbon steel / AISI 4340H

The speed change between the engine driver and compressor was handled by the speed reducing gear, this was a single stage double helical design selected to reduce the speed from a nominal 1000 rpm to 360 rpm and was connected to the compressor through a 26 foot long torque shaft. Details are shown in Table 5.

Table 5. Speed Reducing Gear and Low Speed Torque Shaft.

(1)	Speed reducer
(2)	Single stage / double helical design (1000 rpm / 360 rpm)
(3)	Speed ratio 2.784 : 1
(4)	Torsional resilient high speed coupling connecting engine driver to high speed gear (pinion)
(5)	Torque limiting low speed coupling connecting gear quill shaft to torque shaft
(6)	Torque shaft (26 feet long) connecting gearbox to compressor

The main engine drivers for the IP booster compressors and HP injection gas compressors were all identical gas engines. The details of are shown in Table 6.

Table 6. Internal Combustion Engine Drivers for All Compressors.

	Main Gas Engine Driver
(1)	Gas engine, rated at 4450 hp @ 1000 rpm, derated due to fuel composition.
(2)	V-16 cylinders, arranged 8 per side
(3)	Rated speed = 1000 rpm
(4)	Turbocharged design / 1 turbo per bank of cylinders
(5)	Compression ratio = 9.2:1
(6)	Rated hp = 4450 hp (basis fuel used)
(7)	Engine bore = 300 mm (11.8 inches)
(8)	Engine stroke = 300 mm (11.8 inches)
(9)	Displacement = 339.3 liters
(10)	Electronically controlled ignition system.
(11)	2 intake / 2 exhaust valves per cylinder
(12)	Digital microprocessor based monitoring and engine control system.

The equipment was assembled in the compressor packager's shop. The compression equipment combined with the discharge gas cooler (an air cooler) was overwhelming due its sheer size. The length of just the compressor, gearbox and engine driver skid was approximately 45 feet. With the air cooled discharge gas cooler it was approximately 70 feet long. Most of the modifications were made to the drive train and pulsation system, many of which are discussed elsewhere in this paper.

The HP Train Torsional Analysis

The torsional analysis for the HP injection gas compressor was developed initially by the compressor manufacturer. As the final design progressed another torsional study was commissioned by a third party engineering consultant. The mass elastic system was modeled with data supplied by the suppliers of the components in the train. The engine damper flywheel was modeled as a separate inertia. The damper housing is modeled as a separate inertia as are the eight throws of the engine. The torsionally resilient coupling is mounted to the engine flywheel and also modeled as a separate inertia although mounted to the engine flywheel (Figure 6). The hub is on the pinion shaft and modeled with that inertia.

Figure 6. Torque Limiting Coupling/Low Speed Gears and Torque Shaft.

As noted in this paper, the torque shaft is 26 feet long and is different for the Phase 1 and Phase 2 unit trains. Of the three units manufactured, the first unit (Phase 1 unit) has a separate hub for the torque limiting coupling, while the second and third units (Phase 2 units) have a design that incorporates an integral hub, that was intended to reduce any possibility of failure due to stress concentration factors. It was found however that the mass-elastic properties of the new integral quill shaft were almost identical to those for the separate hub design of Phase 1. In referring to Figure 5 it can be seen that the "torque shaft" is arranged such that it goes through the low speed gear to the flywheel of the HP injection gas compressor.

The torsionally resilient high speed coupling between the engine and high speed pinion of the gear consists of rubber blocks in compression. The torsional stiffness of the coupling is nonlinear and increases with the transmitted torque. As a result of this varying stiffness, some of the torsional natural frequencies (depending on the mode shape) are represented by a range of frequencies rather than a single frequency. In addition, the torsional stiffness of the torsionally resilient high speed coupling is sensitive to the temperature of the rubber blocks. Special blocks were used that were rated up to a temperature of 130°C. The catalog stiffness values are based on a temperature of 30°C, but could vary by ±15 percent from the actual coupling stiffness.

In order to bracket the range of torsional natural frequencies, the full load and no-load cases were analyzed. The first eight such torsional natural frequencies of the system are listed in Table 7 and 8.

Table 7. Torsional Natural Frequencies—Full Load.

Torsional Natural Frequencies at Full Load Operation						
Mode	30 C		100 C		130 C	
Number	Hz	CPM	Hz	CPM	Hz	CPM
1	4.3	259	4.2	250	4.0	242
2	20.4	1,233	16.7	1,002	14.8	887
3	30.7	1,844	30.7	1,843	30.7	1,843
4	51.2	3,070	50.8	3,049	50.7	3,040
5	118	7,104	118	7,104	118	7,104
6	133	8,003	133	8,002	133	8,001
7	201	12,079	201	12,076	201	12,074
8	216	12,985	216	12,985	216	12,984

Table 8. Torsional Natural Frequencies—No-Load Operation.

Torsional Natural Frequencies based on No Load Operation						
Mode	30 C		50 C		130 C	
Number	Hz	CPM	Hz	CPM	Hz	CPM
1	4.2	252	4.2	250	3.8	229
2	17.4	1,044	16.6	997	12.9	774
3	30.7	1,844	30.7	1,843	30.7	1,843
4	50.9	3,052	50.8	3,048	50.5	3,033
5	118	7,104	118	7,104	118	7,103
6	133	8,002	133	8,002	133	8,001
7	201	12,076	201	12,076	201	12,073
8	216	12,985	216	12,985	216	12,984

It was noted that the long torque shaft and large compressor flywheel were instrumental in keeping the first torsional natural frequency below the compressor operating speed range (324 to 360 rpm). The varying stiffness of the torsionally resilient high speed coupling affects the first torsional natural frequency only slightly with a range from 229 cpm to 259 cpm.

As determined by the torsional studies, the second torsional natural frequency of the HP compressor drive system is primarily determined by the torsionally resilient high speed coupling's stiffness, which is determined by the engine load and the rubber block temperature (Figure 7). Depending on the load and block temperature, the second torsional natural frequency could range from 774 cpm to 1223 cpm, which is approximately ±20 percent of the engine speed. This means that for some operating conditions, the engine speed could be coincident with the HP drive system's second torsional natural frequency. There is a temperature between 30°C and 130°C for each load condition where the high speed rubber block stiffness is approximately 6.7×10^6 in-lb/rad and second natural frequency is near 1,000 cpm. For example, a resonant condition would be predicted at full load with a temperature of 100°C and at idle load with a temperature of 50°C. Although the predicted stress levels are below the endurance limits, the calculated heat dissipation in the DCB coupling could exceed the rated wattage.

DCB 847.5 Coupling with 70 Durometer EPDM Blocks

Temp. deg C	Correction Factor (St)	Dynamic Magnifier	Torsional Stiffness (million in-lb/rad)				
			Idle & 20%	40% Load	60% Load	80% Load	100% Load
30	1.00	6.0	7.45	7.61	8.27	9.39	11.01
40	0.94	6.3	7.03	7.18	7.81	8.86	10.39
50	0.89	6.7	6.62	6.76	7.34	8.33	9.78
60	0.83	7.2	6.20	6.33	6.88	7.81	9.16
70	0.78	7.7	5.78	5.90	6.42	7.28	8.54
80	0.72	8.3	5.37	5.48	5.95	6.76	7.93
90	0.66	9.0	4.95	5.05	5.49	6.23	7.31
100	0.61	9.8	4.53	4.63	5.03	5.71	6.69
110	0.55	10.8	4.11	4.20	4.57	5.18	6.08
120	0.50	12.1	3.70	3.77	4.10	4.66	5.46
130	0.44	13.6	3.28	3.35	3.64	4.13	4.85

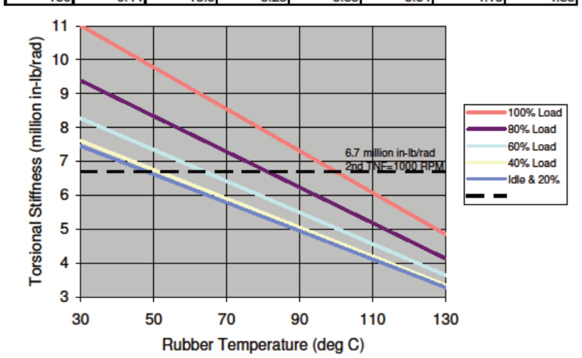

Figure 7. Torsional Resilient Coupling—DCB with 70 Durometer EPDM Blocks.

The third and higher HP drive train system torsional natural frequencies are not greatly affected by the high speed coupling and would not vary significantly with load or rubber block temperature. The third HP drive train torsional mode involves the engine damper flywheel inside the damper housing and is well damped. The fourth mode is the engine crankshaft mode and fifth mode is the compressor crankshaft mode.

The interference diagram (Campbell diagram) plotted in Figure 8 compares the HP injection compressor drive train torsional natural frequencies with the excitation frequencies provided by the IC engine driver. A torsional resilient high speed coupling stiffness of 6.7×10^6 in-lb/rad was used to construct this interference diagram. Half orders are produced by the four-stroke cycle engine. Harmonics up 12× engine speed were considered in the analysis based to the torque effort data supplied by the engine manufacturer. For this engine driver engine operating normally, the most significant harmonics are 0.5×, 2.5×, 3.5×, 4×, and 4.5× engine operating speed.

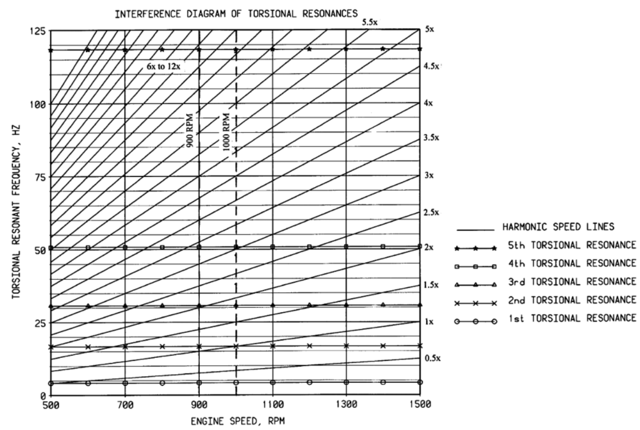

Figure 8. Interference Diagram with Engine Driver Excitation. (Courtesy Engineering Dynamics Inc.)

A torsional resonance was predicted where the 1x engine speed coincides with the first torsional natural frequency of the HP injection compressor drive train. As the coupling stiffness changes with engine load and rubber block temperature, this resonance could be located near the maximum operating speed of the engine. The 3.5× intersects the fourth mode (engine crankshaft mode) near 870 rpm which is 3 percent below the minimum operating speed of the engine. It is good engineering practice to have a separation margin of at least 10 percent be maintained between the operating speed and major torsional resonances or interference points (also API standard recommendation) in order to prevent damage to the shaft.

The interference diagram plotted in Figure 9 compares the torsional natural frequencies with the excitation frequencies provided from the load compressor. The first 12 compressor harmonics were considered in the analysis based on torque harmonic data provided by the compressor manufacturer. The first torsional natural frequency was calculated to be at least 20 percent below the minimum compressor speed of 324 rpm. The second compressor order is located between the first and second torsional natural frequencies.

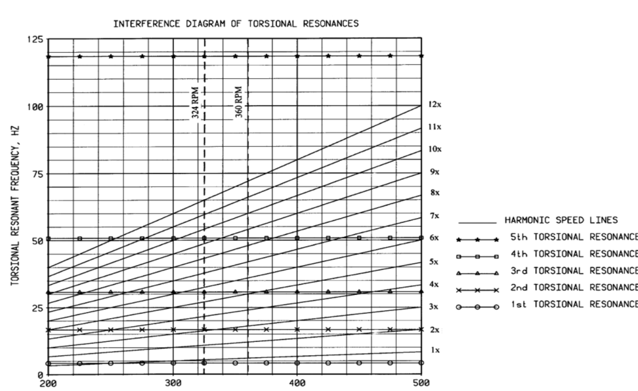

Figure 9. Interference Diagram with Compressor Excitations. (Courtesy Engineering Dynamics Inc.)

DESIGN ISSUES AND WHAT WAS DONE TO IMPROVE THE INJECTION GAS COMPRESSOR DESIGN

Due to the extreme high pressures involved in the reinjection of the produced gas and the machinery required to accomplish those project objectives, the design engineering team spent a considerable amount of time at the start to evaluate the design(s) that had been specified and purchased. The design team thoroughly investigated the design as it related to:

- Safety,
- Operability,
- Maintainability, and
- Suitability for the service involved.

The initial studies showed the following (major) design issues that needed to be addressed. Some of these design features were selected in the interest of schedule, as it was discovered that the initial teams proceeded rather quickly in their effort to provide the reinjection of gas within one year.

- The original torque shaft design was 18 feet long and was connected to the gearbox through a shear bolt coupling. The design was based on the fact that the shear bolts would shear off every time the compressor went through a shutdown from a discharge pressure of 8000 psig or higher, planned or otherwise. This arrangement was not considered acceptable as replacement of the shear bolts would be necessary each and every time the compressor was shutdown.
- The initial torque shaft design was 18 feet long and had a limited life, which was the reason for the torque shaft redesign. A fatigue analysis showed that it would fail in approximately 111 cycles if allowed to cycle during compressor shutdown.
- The quill shaft connecting the low speed gear to the torque shaft had a separate hub that was attached via a shrink fit.
- The baseplate for the compressor, gearbox and engine was undersized.
- The fifth stage gas discharge cooler did not have sufficient surface area to handle the duty required for all operating cases.
- The gas blowdown valves were not sufficient to handle the high pressures developed by the compressor.
- The adequacy of the design of the engine's torsional vibration dampener was questioned.
- The compressor pedestal was not suitable because of the high vibration measured during field testing.
- The compressor's unbalanced forces were excessive.
- Numerous process issues, these included sizing of the recycle gas valves and the overall capacity control system. Depressurization of the HP compressor system was a significant design challenge. The system needed to be designed such that it could be depressurized without explosive decompression.
- Relief valves needed to be reselected. The initial selection indicated that they were too small.
- The compressor's pulsation dampeners needed to be checked during the acoustic study. The initial project was proceeding so fast that the pulsation equipment was essentially released to be manufactured prior to conducting the acoustic study.
- A torsional vibration analysis needed to be developed based on the new 26 foot long torque shaft.
- The equipment was not designed for an engine misfire condition.
- Due to the longer torque shaft an intermediate bearing was added for support.

The most significant change to the initial design was to increase the diameter of the torque shaft thereby reducing the shear stresses. That change increased the life of the shaft significantly and field tests confirmed that point. And at the end of the project it was determined that the stresses were lower in the shaft resulting in infinite life.

The longer torque shaft was configured longer for the same torsional stiffness. The changes also resulted in a larger compressor flywheel. That flywheel is there to reduce the dynamic torque generated by the reciprocating motion of the running gear/piston.

One of the changes that was made to the initial design was to add torque limiting coupling to the low speed torque shaft that connected the low speed gear to the torque shaft that runs to the HP injection compressor. Figure 10 shows this safety item intended to limit the torque that could be transmitted to the 26 foot long torque shaft.

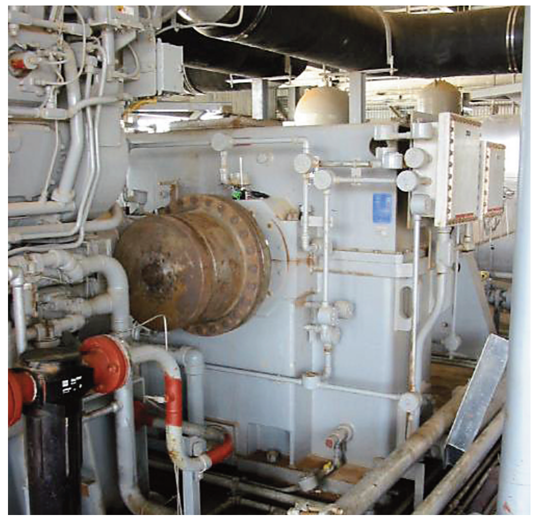

Figure 10. Torque Limiting Coupling Connects Low Speed Gear to Torque Shaft.

Figure 11 shows the fourth stage cylinder with suction and discharge spheres. This clearly shows the spherical design used for construction of the pulsation dampeners. This design was chosen specifically because of the high pressures of the system.

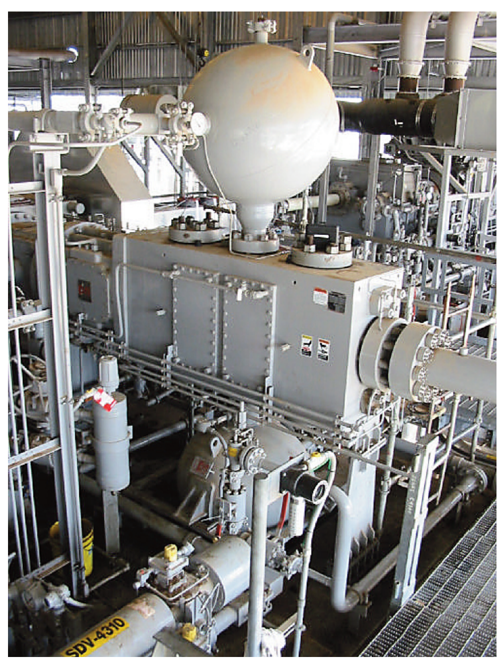

Figure 11. Fourth Stage Cylinder.

Initial HP Field Tests

Field commissioning of the first HP injection gas compressor proceeded cautiously. The compressor was started and run under no load conditions. Loading was increased gradually up to 3500 psig discharge pressure, but while on recycle. Field strain gauge readings were taken on the nozzles of the suction scrubbers. Vibration readings were taken on the compressor support pedestal.

After studying the data the HP injection gas compressor was operated up to 3600 psig discharge pressure. Field readings were taken again on the connected vessels and compressor support pedestal. It was found that the vibration on the compressor pedestal was excessive. That excessive vibration was caused by unbalanced forces generated by the compressor.

The design change for this condition was to add additional counterweights to the compressor crankshaft and increase the size of existing counterweights to the maximum size permitted by the manufacturer. After this was completed additional counterweights were added and existing counterweights increased. The first HP injection gas compressor was run again, up to a final discharge pressure of approximately 8,000 psig.

During the run at 8,000 psig extensive data were collected on the torque shaft, engine flywheel, gears, compressor, compressor pedestal, baseplate, piping, pulsation dampeners, suction scrubbers and intermediate support bearing under the 26 foot torque shaft. At that point it was decided to review the data and study the results before conducting additional runs. The next step would be to fully load the compressor and operate it up to the 10,000 psig discharge pressure.

Possible resonant conditions exist if the mechanical natural frequencies of the drive train components fall within the compressor and engine excitation frequencies for the operating speed range shown in Table 9. During the field testing, the mechanical natural frequencies of all drive train components were measured and compared to the Table 5 compressor and engine excitation frequencies to determine if resonant conditions were present. Many of the component mechanical natural frequencies (MNFs) were determined from mechanical impact tests of the components.

Table 9. Compressor and Engine Excitation Frequencies.

Order	Frequency (Hz)	
	Compressor	Engine
1x	5.4 – 6.0	15.0 – 16.7
2x	10.8 – 12.0	30.0 – 33.4
3x	16.2 – 18.0	45.0 – 50.1
4x	21.6 – 24.0	60.0 – 66.8
5x	27.0 – 30.0	75.0 – 83.5
6x	32.4 – 36.0	90.0 – 100.0
7x	37.8 – 42.0	105.0 – 116.7
8x	43.2 – 48.0	120.0 – 133.4

DESIGN AUDITS

The Phase 1 HP Unit

As previously noted above, the project was divided into a Phase 1 and Phase 2. Phase 1 proceeded with one HP injection gas compressor on an accelerated schedule in order to begin high pressure gas injection gas as soon as possible to improve reservoir performance and increase oil production. This first unit required numerous design improvements, which were discovered and needed to be implemented during the manufacturing and assembly of the first train of machinery.

Many issues and design improvements were discussed and resolved during the numerous engineering meetings. This paper will only be presenting the issues associated with the torque shaft and drive train. The other engineering issues are mentioned just to show the enormous engineering effort required to provide safe and operable equipment.

It was recognized during the initial design evaluation that the HP injection compressors were somewhat state-of-the-art in that there just was not sufficient and existing design and operational data available to use as reference. Therefore the engineering team and owner devised a plan for a phased startup of the machinery. The Phase 1 HP injection compressor was started and run up to approximately 7,000 to 7,500 psig and extensive data taken. The data were reviewed by consultants, the owner and the EPC engineering consultants and then new field testing plans were drawn up to test the compressor and system up to 10,000 psig discharge pressure. After the initial 7,000 psig field testing, an outside consultant was engaged to perform a third party review of the design and test data to see if anything had been overlooked before running the HP compressor up to the full rated discharge pressure of 10,000 psig.

Torque Shaft Stress Limits

One of the major concerns as noted throughout this paper was design of the low speed torque shaft connecting the low speed gear and the HP injection gas compressor. Data were collected and analyzed thoroughly before the compressor was ever operated at the full 10,000 psig discharge pressure. The data in Figure 11 relate these torque shaft stresses to the design limit. Engine misfire was one such condition that could have had a dramatic effect on the stresses and material life.

Figure 12 summarizes the results for the low speed torque shaft. It plots torque against engine load. The lowest line is the predicted torque. The higher lines of Figure 12 show the two sections of line for the measured overall vibration at 1000 rpm. The discontinuity at around 70 percent load is apparent, and reflects the change between field data, introduced by the spike excitation of the first torsional natural frequency, in whose mode shape the torque shaft flexibility dominates. As with previous summary plots, Figure 12 also shows two individual data points for the maximum misfire torque at 8500 and 9500 psig discharge (84 percent and 94 percent load). The 9500-misfire point is higher than the maximum normal operation value, but at 8500, the misfire point actually falls below the normal operation line.

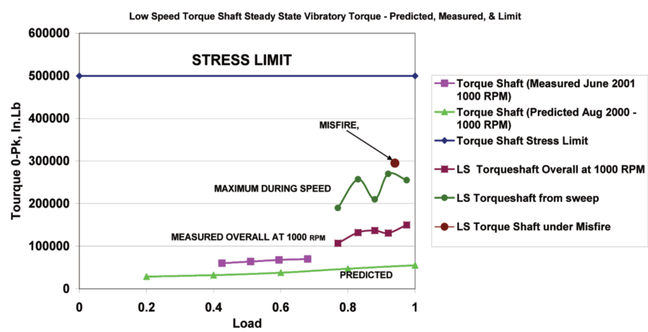

Figure 12. Torque Shaft Stresses.

Gear Mesh Torque

The closest physical torque measurement to the gearbox is on the pinion shaft. For normal operation, the ratio of gear mesh torque to pinion torque varies only a small amount with load (e.g., 0.86 at 100 percent load and 0.84 at 80 percent load), and so inference of gear mesh torque via these ratios seems justifiable for normal operation, and has been done in this way. However, under misfire conditions, the model gives a ratio that varies much more strongly with load (1.12 at 100 percent and 0.77 at 80 percent load). This reduces the credibility of this approach for misfire conditions and, for this reason, the misfire gear mesh torques at 94 percent and 84 percent load (for 9500 and 8550 discharge pressure) were simply set equal to the pinion torque. It is believed this is conservative.

Figure 13 summarizes the results. It shows, at the lowest level on the chart, the predicted torques for pinion shaft and gear mesh. Above these are the measured pinion shaft torque (red squares) and inferred gear mesh torque for normal conditions of operation (blue circles), based on the overall vibration at 1000 rpm. These are well below the vibratory torque limit for the gearbox (40 percent nominal static torque). Above this in the range from about 75 percent to 100 percent load is the gear mesh torque inferred from the maximum pinion shaft vibration over the speed sweep. This also remains below the gearbox limit.

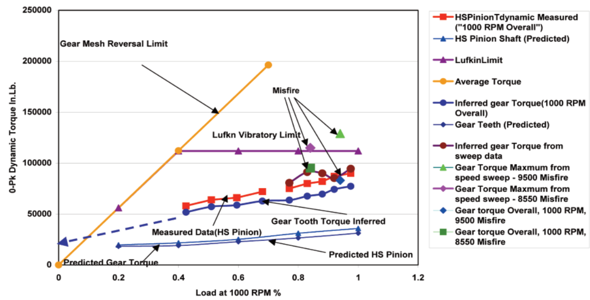

Figure 13. Gear Mesh Dynamic Torque Versus Percent Load with Limits.

For misfire conditions, four points have been included on the chart: two at 84 percent load (8550 psig discharge), and two at 94 percent load (9500 psig discharge). The highest at each load is the maximum from the speed sweep; as discussed, rather than ratio the misfire data, this is set equal to the maximum pinion shaft torque from the speed sweep data. The lower value at each load is equal to the overall dynamic pinion torque at 1000 rpm.

It may be seen that the 84 percent load maximum misfire point is just above the gearbox limit line and at 94 percent, the maximum misfire torque is distinctly above the limit. The reason for including the lower values at each load in this chart is that comparison of the maximum value out of an 800-second record with the limit is not quite the same as comparing a sustained level of vibration with the limit. The overall is more representative of sustained operation, and this is below the gear manufacturer's limit. The maximum from the speed sweep is more like an occasional or transient situation. In fact, only about eight points in 800 seconds exceed 112,000 in-lb. under misfire conditions, so the frequency of exceedance is once in one hundred seconds—very low rate.

Engine Damper Vibration Data

Figure 14 presents main engine driver damper vibration data as a function of engine load (which, of course, increases with increasing discharge pressure). There are a number of lines and data points on this graph, as discussed below.

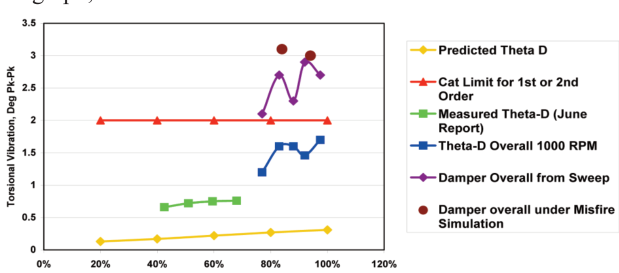

Figure 14. Engine Damper Vibration Predictions, Measurements, and Limit.

The lowest line on the chart (in yellow) represents the predicted damper vibration at 1000 rpm, based on field data. This is obtained by scalar addition of the predicted vibration for compressor excitation and engine excitation, for each load percentage (20 percent, 40 percent, 60 percent, 80 percent, and 100 percent); however, data were only available for excitation by the engine, so the predicted vibration is understated by an unknown amount (probably by less than 10 percent, based on other data).

The next highest level line (green) is the measured damper vibration at 1000 rpm from the May test series, which only went to 7920 psig. To the right of this (blue) is the measured overall damper vibration from the July test series at 1000 rpm, and reflects the discontinuity between the two data sets for damper vibration—supposedly as a result of the repeated excitation of the 1TNF.

The next higher line is the engine manufacturer's limit for first and second order vibrations. The engine manufacturer has no published overall limit, and no published limit for nonorder related vibrations. The engine manufacturer's first and second order limits are 2 degrees each, so the overall damper vibration at 1000 rpm remains below the first order limit. However, the maximum vibration excursion from the speed sweep data, plotted in purple, goes slightly over the first order limit, and the misfire maximum excursion from the speed sweep is even higher reaching 3.1 degrees at 8550 psig.

Since the engine moves torsionally as a rigid body at these low frequencies, torsional vibration should not impose any significant stress on the engine crankshaft. The primary issue would be whether any excess torque was transmitted across the bolts, which hold on the damper and the flywheel.

Torsional Resilient High Speed Coupling Dynamic Torque

No torque measurement is located on the torsionally resilient high speed coupling. The closest torque measurement is on the high-speed pinion shaft. Some method of inferring the coupling torque from the high-speed pinion data is needed. One option is to assume the coupling torque equals the pinion torque. The alternative is to use predictions of the third party engineering model to relate the coupling torque to the pinion torque. Figure 15 presents the results of the latter approach. It plots torque as a function of engine load. The lowest two lines are the predictions of torque for coupling and pinion from the field report, obtained by adding the responses to compressor and engine excitation. The two are close, but the predicted coupling torque is higher than the pinion torque by a factor, which varies slightly with load—typically about 1.1 or lower. Assuming the ratio of coupling torque to measured pinion torque preserves the same functional relationship with load as for the model, the measured pinion torque can be used to infer the coupling torque. The red line shows the measured overall pinion torque at 1000 rpm, and the olive line shows the overall coupling torque inferred in this way as a function of load. The dark blue line with diamond markers shows the coupling torque inferred from the maximum speed sweep pinion data using the same ratios as for the overall vibratory torque data.

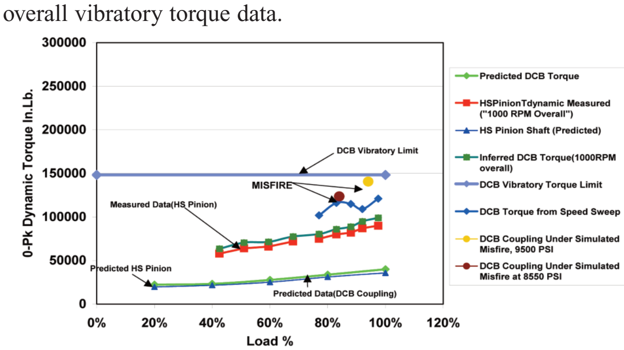

Figure 15. Dynamic Torque at Torsional Resilient High Speed Coupling.

The chart includes two individual data points that correspond to the simulated misfire condition at 8550, and at 9500 psig. An attempt was made to develop ratios that relate coupling to pinion torque under misfire conditions based on the model predictions for misfire conditions, but this effort encountered some difficulties. Whereas the ratios under normal conditions changed by only a small amount with load, the ratio (coupling/pinion torque) with misfire was 0.81 at 100 percent load, and 1.14 at 80 percent load. Rather than use these widely varying numbers, it was decided to base the ratio on the normal conditions data (1.1 percent at 100 percent load and 1.07 percent at 80 percent load, and interpolate for intermediate loads).

The two data values for misfire correspond to the maximum vibration over the speed sweep for the condensed data plot. These and all other vibration levels are seen to lie below the allowable vibration for stress (which accounts for stress concentration, mean stress, size effects, shear, and surface effects, and has a 2:1 factor of safety).

The high speed coupling stiffness is a dominant factor in the second torsional natural frequency. The analyses indicated that depending on this torsional stiffness, the second torsional natural frequency (2TNF) could lie above or below the 1000 rpm engine running speed (16.7 Hz). As already discussed, the primary evidence of a second torsional natural frequency shows up in the pinion shaft raster plots, which suggest it is at or just above 25 Hz. For this to occur, the rubber block coupling stiffness would have to be 17.3×10^6 in-lbs/rad. The EPDM data interpolated for torque and temperature gives a stiffness of 11.5×10^6 in-lbs/rad. Adjusting up 10 percent for the frequency (which tends to stiffen elastomers) suggests one could have 12.7×10^6 in-lbs/rad. The 2TNF with 12.7×10^6 in-lbs/rad would be about 21.4 Hz, about 14 percent below what the data indicates.

One of the weaknesses of the torsional resilient coupling was the material used. The principle of this design was such that as the torque in the system reversed the rubber blocks would absorb the angular oscillations. During the operation this caused the temperature of these blocks to increase. That temperature increased with the torque that was transmitted and also varied with the type of block material selected (Figure 16).

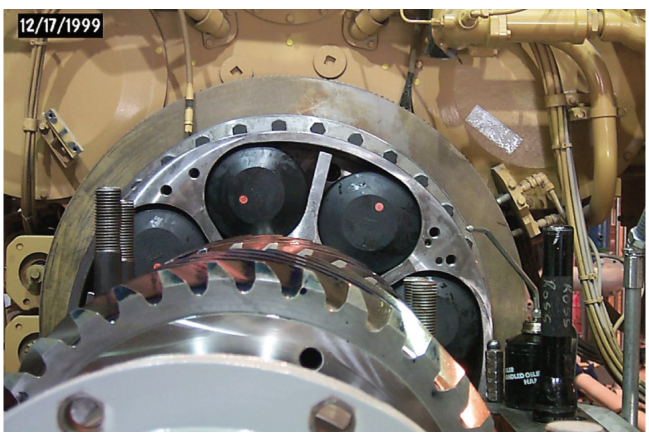

Figure 16. High Speed Torsional Coupling.

FIELD VERIFICATION/DATA OF FINAL DESIGN UNIT

Torsional Vibration Measurements at 10,000 PSIG

The initial field torsional vibration testing of the HP compressor was on Unit C-4100 at the BP Boquerón Plant in Venezuela. The measurements were taken with an inlet pressure of approximately 2,900 psi and a final discharge pressure ranging from 8,000 to 10,100 psi.

The purpose of the torsional tests was to check the acceptability of the unit under normal operation and engine misfire conditions. Transient events such as startups and emergency shutdowns (ESDs) were also evaluated. Torsional vibration data were acquired to address the following areas of concern for this system:

- Sensitivity of the compressor system to operating and upset conditions
- Integrity of the rubber blocks in the torsional resilient coupling between the engine and gearbox
- Dynamic torque at the gear mesh
- Alternating shear stress in the low speed torque shaft
- Integrity of the engine torsional vibration dampener

The unit typically operated at an engine speed of 975 rpm with an approximate compressor discharge pressure of 8,300 psig. After the tests were completed, the injection well was changed from #10 to #14, which has a higher pressure. The engine speed was increased to 980 to 990 rpm with a compressor discharge pressure of 9,100 psi. These operating conditions vary depending on the field requirements and pressure at the injection wells.

As noted previously, the HP compressor is driven by a 6,500 hp gas engine through a speed reducer and 26 ft long torque shaft. The operating speed range is 900 to 1,000 rpm for the engine and 324 to 360 rpm for the compressor. The engine has a viscous damper on the nondrive end to help limit shear stresses in the engine crankshaft. A torsional resilient coupling is mounted to the engine flywheel and connects to the HS pinion. This coupling contains 70 durometer, EPDM rubber blocks that are rated for higher temperature. Due to the relatively low torsional stiffness, this coupling provides some torsional isolation between the engine and the gearbox. Also, the rubber blocks provide damping to help limit the torsional vibrations.

The gearbox utilizes a quill shaft arrangement where the LS torque shaft passes through the hollow gear shaft. A torque limiting coupling connects the gear quill shaft to the torque shaft that drives the compressor. The torque limiting coupling is a device to help protect the LS torque shaft from excessive shear stresses. The hub where the torque limiting coupling mounts to the quill shaft is shrunk-on for the first unit, C-4100, but is integral to the shaft for units C-4200 and C-4300. Torsionally, these two quill shaft designs are identical.

The first stage of HP compression (fourth overall stage) is provided by an 8.00 inch bore cylinder. The specified fourth stage pressures are 3,069 psig for the suction and 5,571 psig for the discharge (ΔP = 2,502 psi). The second stage of HP compression (fifth overall stage) is provided by a 7.125 inch bore cylinder. The specified fifth stage pressures are 5,414 psig for the suction and 10,101 psig for the discharge (ΔP = 4,687 psi).

Phase 2 of the project includes an additional three IP units and two HP units that were installed in the same compressor building. The rated gas flow is 30 mmscf per day for each IP unit and 50 mmscf per day for each HP unit. When all eight of the compressors are operational, the rated flow for the entire station will be 150 mmscf per day. From a torsional standpoint, the HP units from Phase 1 and 2 are identical in design.

During the design phase of the compressor system, a number of torsional analyses were performed. The gearbox manufacturer also performed calculations regarding the predicted fatigue life of the LS torque shaft. A comprehensive analysis was published that predicted the first torsional natural frequency (TNF) at 4.2 Hz (252 cpm), which was 22 percent below the minimum compressor speed. The first TNF is primarily influenced by the low torsional stiffness of the torque shaft and the large polar moment of inertia of the compressor flywheel. The second TNF is primarily controlled by the stiffness of the torsional resilient high speed coupling. This coupling consists of rubber blocks in compression. The torsional stiffness of the coupling is nonlinear, increases with the transmitted torque, and decreases at higher block temperatures. Therefore, the second TNF, which was predicted in close proximity to the 1× engine speed, will vary depending on the operating conditions.

Assuming the worst case of the engine operating on the second TNF, the torsional analysis predicted an acceptable system under normal operating conditions. However, continuous engine misfire could create higher 1× engine excitation that could be amplified if running on resonance. With engine misfire at loads above 68 percent, there was a potential for the torsional response to exceed the allowable levels for the torsional resilient high speed coupling and gearbox. Therefore, field torsional measurements were warranted to verify the torsional characteristics of the system.

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Torsional oscillation was measured on the damper end of the main gas engine crankshaft using a torsiograph. Recall that a four-cycle engine has a combustion cycle every other crank revolution. Therefore, dynamic torque occurs at multiples of half engine speed.

Main Gas Engine Driver
Allowable Torsional Oscillation

The engine manufacturer gives allowable torsional oscillation at the damper end of the crankshaft in degrees peak to-peak (p-p) for selected engine orders as shown in Table 10. These allowables are for steady-state operation and do not apply to transient events such as starts and shutdowns. Allowable levels were not provided for overall, or amplitudes at frequencies not corresponding to engine orders or half-orders.

Table 10. Allowable Torsional Vibration at Damper End of Compressor Gas Engine.

Engine Order	Allowable Oscillation
0.5 x	2 deg p-p
1.0 x	2 deg p-p
1.5x	0.5 deg p-p
2 x and above	0.3 deg p-p

Torsional Resilient High Speed Coupling

The coupling consists of rubber blocks in compression. The blocks provide damping and limit torsional vibration. The vibratory shaft action on the blocks is converted to heat energy that causes the temperature of the blocks to increase. If the torque or temperature of the blocks is excessive, the blocks can fail by cracking or even melting from the inside out. The damaged blocks would not provide the same protection to the system as when new. Therefore, the coupling manufacturer provides allowable limits as given in Table 11. According to the torsional resilient coupling manufacturer, the key indicator of coupling durability is the heat load.

Table 11. Allowable Levels for High Speed Torsional Resilient Coupling.

Description	Allowable Level
Dynamic Torque on Continuous Basis	148,471 in–lb 0–p
Peak Torque During Transient Event	1,187,770 in–lb
Maximum Block Temperature	266° F (130° C)
Maximum Heat Dissipation	1,010 Watts

High Speed Gearbox/Speed Reducer

The dynamic torque limit for the HS pinion was specified in two ways: based on the gear mesh torque and the alternating shear stress in the shaft. The allowable dynamic torque at the gear mesh will be reached before exceeding the allowable shaft stress and is therefore the limiting factor for this gearbox.

Initially the gearbox manufacturer specified that the allowable dynamic torque in the gear mesh was 30 percent of the rated torque based on a specification from the Marine Institute. This limit was subsequently extended to 40 percent, provided the gears remain in mesh. The rated torque in the HS pinion shaft is 280,461 in-lb based on 4,450 hp at 1,000 rpm. Therefore, the allowable dynamic torque at the gear mesh is 112,184 in-lb 0-p.

For low loads, the dynamic torque at the gear mesh should be less than the transmitted torque to prevent torque reversal and gear backlash. The gear manufacturer's primary concern is that continuous operation with tooth separation would damage the gear teeth. Due to the hardness of the carburized teeth, a visual inspection may not detect a problem with the surface before actually breaking a tooth.

The dynamic torque in the pinion shaft at which the alternating shear stress at the keyway would reach the allowable limit of 6,061 psi 0–p was calculated to be 205,000 in-lb 0-p (73 percent of rated torque). This was based on an outside diameter of 8.25 inch and a stress concentration factor (SCF) of 3.26. This SCF is based on a fillet radius of 1/8 inch at the base of the keyway.

The HS pinion and LS gear both have proximity probes in the horizontal and vertical directions to monitor shaft vibration. The probes are located on one end of the pinion and gear shafts for a total of four. The shaft vibration limits are given in Table 12 along with the bearing clearances. Note that when the unit starts, there is a one-minute time delay on the shutdown levels to prevent the unit from tripping on high vibration when the gears are unloaded. The logic and programming are incorporated into the plant distributed control systems (DCS), and not in the local engine/compressor local control panel.

Table 12. Allowable Shaft Lateral Vibration and Bearing Clearances for the Speed Reducing Gearbox.

Shaft	Alarm Level (mils p-p)	Shutdown Level (mils p-p)	Bearing Dia. Clr. (mils)
HS Pinion	2.5	5.00	10 12
LS Gear	3.5	6.00	14 - 16

Once the compressor operating conditions stabilized, the engine speed was slowly varied from 900 rpm to 1,000 rpm (compressor speed of 324 to 360 rpm). To simulate misfire, the engine was operated with a closed spark plug gap in cylinder 2. Although ignition was prevented in engine cylinder 2, the gas compression effects were still present. Speed sweeps were performed with the compressor final discharge pressure at 8,550 psi and 9,500 psi. Waterfall plots were created to show the torsional and gearbox vibrations.

During steady-state operation, the gearbox vibration from the proximity probes remained below alarm levels. Speed rasters were prepared.

Table 13 summarizes the measured torsional response at several locations versus final discharge pressure for HP injection compressor unit C-4300. The amplitudes at the various engine orders were taken from the slice plots. The maximum overall values were determined by analyzing the time-domain data acquired during the speed sweeps (compressed data plots). The worst case values presented in Table 13 are higher than the average values and do not occur every cycle.

Table 13. Maximum Torsional Vibration and Dynamic Torque Levels During Speed Sweep Tests.

				5th Stage Discharge Pressure (PSIG) - Unit C-4300 HP Compressor						
Location	Order	Units	Allowable	8,050	8,500	9,025	9,582	10,121	8,550	9,500
									Misfire	Misfire
Damper End	Overall	Deg p-p		2.1	2.7	2.3	2.9	2.7	3.1	3.0
of Gas Engine	0.5x	Deg p	2.00	0.22	0.17	0.18	0.23	0.18	0.25	0.30
	1x	Deg p	2.00	0.13	0.10	0.10	0.11	0.09	0.16	0.17
	1.5x	Deg p	0.5	0.17	0.13	0.13	0.17	0.14	0.22	0.23
	2x	Deg p	0.3	0.02	0.02	0.01	0.02	0.01	0.02	0.02
	2.5x	Deg p	0.3	0.05	0.04	0.04	0.05	0.04	0.07	0.07
LS Coupl	Overall	Deg p		0.8	1.1	0.9	1.2	1.1	1.2	1.1
HS Pinion	Overall	In-lb 0-p	205,000	95,000						
Torque Shaft	Overall	In-lb 0-p	500,000	190,000	257,000	210,000	270,000	255,000	223,000	295,000
HS Coupling	Overall	In-lb 0-p	148,471	108,300	123,120	120,840	114,000	125,400	131,100	147,060
Gear Mesh	Overall	In-lb 0-p	112,184	85,500	97,200	95,400	90,000	99,000	103,500	116,100

The strain gauges on the HS pinion shaft were located between the high speed torsional resilient coupling and pinion teeth. Therefore, the HS pinion torque measurements were used to infer the dynamic torques in the high speed torsional resilient coupling and gear mesh. Although the measurement location was in close proximity to the high speed torsional resilient coupling and gear mesh, adjustment factors were calculated based on the Phase 2 torsional analysis to account for inertia and mode shape effects.

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The maximum overall values shown in Table 13 represent the worst conditions observed during the speed sweeps and are higher than the average overall values. These peaks occurred due to random excitation of the first TNF. Although the first TNF of 4.2 Hz is below 1× compressor speed, it can be excited by unsteady speed control of the engine, as well as other sporadic behavior of the compressor operation.

Excitation of the first TNF may be related to unsteady speed control. The speed control initially was fairly smooth, while the speed control during these final Phase 1 field tests was more erratic. The operators reported the engine control module (ECM) and/or "personality" module, which contain the engine specific set points, were changed between testing periods. It was suggested that the operating plant personnel should look into the possible causes for this speed fluctuation, including control and engine performance issues.

Gas Engine Driver

Torsional oscillations were measured on the damper end of the engine crankshaft. During the speed sweeps, the highest torsional oscillation occurred during continuous engine misfire as shown in Table 13. The amplitudes at various engine orders were found to be below the engine manufacturer's criteria, even for the engine misfire condition. The maximum overall torsional vibration of 3.1 degrees p-p is due to random excitation of the first TNF.

Comparing the normal operation at 9,500 psi to continuous engine misfire, the maximum overall torsional vibration shown in Table 13 increased by only 11 percent. Misfire caused the torsional vibration at the lower engine orders (0.5×, 1.0× and 1.5×) to increase by 30 percent or more. However, these torsional vibration levels were still below the allowable limits given by the engine manufacturer.

HS Pinion

A strain gauge telemetry system was used to evaluate the dynamic torque in the HS pinion input shaft. The gauges were installed on the HS pinion shaft near the coupling hub, but 90 degrees away from the keyway.

During the speed sweeps, the highest dynamic torque of 129,000 in-lb 0-p occurred during continuous engine misfire at 9,500 psi as shown in Table 13. This is below the allowable level for the pinion shaft in terms of alternating shear stress. However, the inferred dynamic torque of 116,100 in-lb 0-p at the gear mesh (41 percent of rated torque) was slightly above the gear manufacturer's allowable level of 112,184 in-lb 0-p (40 percent of rated torque).

The gear manufacturer reviewed the torsional data and felt that the API gearbox was robust and has a high AGMA service factor of 4. The gearbox manufacturer did not think that the dynamic torque levels were damaging the gear teeth, but was concerned that the levels were increasing with time. They did however request that the unit not be operated with sustained gear tooth separation, which may occur at low loads or with engine misfire. The gear manufacturer also requested that the engine speed control be improved to minimize the excitation of the first TNF.

LS Torque Shaft

A strain gauge telemetry system was used to evaluate dynamic torque in the LS torque shaft. Strain gauges were installed on the LS torque shaft near the intermediate bearing pedestal away from any stress risers. Therefore, the measurements are a direct indication of the dynamic torque in the LS torque shaft.

During the speed sweeps, the highest dynamic torques occurred during continuous engine misfire as shown in Table 13. The highest dynamic torques during engine misfire were not significantly different from those measured during normal operation. As shown, the maximum dynamic torque in the LS shaft was 295,000 in-lb 0-p during misfire at 9,500 psi.

The limiting factor for the torque shaft is the stress at the keyway. Based upon the allowable stress, the dynamic torque in the LS torque shaft should be less than 500,000 in-lb 0-p. Since the measured dynamic torque did not exceed the allowable during normal and misfire operation, the LS torque shaft is considered to be acceptable from a steady-state standpoint. As expected, the dynamic torques were higher during starts and shutdowns compared to steady-state operation.

In some of the data, there also appears to be a pulse occurring once per second. The frequency spectra did not indicate a peak at 1 Hz, but did show three peaks near 4 to 6 Hz with approximately 1 Hz spacing. Therefore, the apparent pulses in the time domain data at one second increments may actually be caused by beating of several closely spaced frequencies. Transfer functions were plotted to determine the relative phase angles at the engine and compressor orders. At the first TNF (4.2 Hz), the HS and LS torsional signals were found to be in-phase (0 degrees), which correlates with the first torsional mode shape. At the second TNF (25 Hz), the HS and LS torsional signals were found to be out-of-phase (180 degrees), which correlates with the second torsional mode shape.

While the unit was down, impact tests were performed on the compressor manifold systems (cylinders, pulsation dampeners), scrubbers, associated piping and support structure to measure the mechanical natural frequencies. The frequency response spectra due to the impact were measured in all three orthogonal directions. Table 14 summarizes the impact test results and the potential excitation sources.

Table 14. Component Mechanical Natural Frequencies. (Courtesy Engineering Dynamics Inc.)

Measurement	Frequency (Hz)	Compressor Excitation
4S Scrubber	13.25, 14.5, 16, 19.2	2x, 3x, 4x
4S Sphere	13.25, 14.5, 16, 19.2, 23.5, 43	2x, 3x, 4x, 8x
4 th Stg Cylinder	13.25, 16, 19.2, 23	2x, 3x, 4x
4D Sphere	23, 26.5, 47.5	4x, 5x, 8x
5S Scrubber	15.75, 16.75	3x
5S Sphere	15.75, 28.75, 89	3x, 5x, 15x, 16x
5 th Stg Cylinder	15.5, 31.5, 47.5	3x, 6x, 8x
5D Sphere	15.5, 18.5, 20.5	3x, 4x
Cooler Platform	11, 11.8, 23.5	2x, 4x
Torque Shaft	47, 60.3, 65, 120, 138, 145	8x6 6 , 10x

One of the significant improvements made was to increase the size of the suction and discharge spheres for pulsation. Figure 17 shows the fifth stage suction pulsation sphere for one of the HP injection gas compressors after it was optimized in Phase 2—increased substantially in size.

Figure 17. Fifth Stage Suction Sphere.

Vibration Survey

Table 15 summarizes the vibration measurements at the various load conditions as the compressor speed was swept from 324 rpm (minimum governor) to 360 rpm (maximum governor). The highest vibration amplitude component measured within the operating speed range was tabulated. The vibration readings are given in mils p-p for each pressure condition tested. The amplitudes are compared to the appropriate allowable and/or the "hay stack" screening criteria curves.

Table 15. Summary of Vibration Readings (Mils P-P) (Unit C-4300).

Measurement Location	Compressor Excitation	Frequency (Hz)	Compressor		5th Stg Disc Press (PSI)			Allowable Vibration
			Harm	RPM				
Compressor Frame								
Flywheel Base	Stretch	6	1x	360	1.9	2.2	2.4	2.5
Flywheel CL	Stretch	6	1x	360	2.3	2.6	2.8	5.0
Oil Pump Base	Stretch	6	1x	360	2.1	1.9	1.7	2.5
Oil Pump CL	Stretch	6	1x	360	3.2	2.9	2.6	5.0
4th Stage System								
4S Scrubber	Stretch	16.2	3x	324	4.5	4.3	5	7.8
4S Sphere	Stretch	18	3x	360	3.5	4.3	5	7.4
4th Stage Cylinder	Horoz	6	1x	360	3.5	4.2	4.0	6.4
5th Stage System								
5S Scrubber	Horoz	17.1	3x	342	5.7	5.5	3.5	7.6
5S Sphere	Stretch	17.1	3x	342	5.4	5.4	3.5	7.6
5S Sphere	Vertical	16.2	3x	324	8.5	7.8	9.5	7.9
5S Sphere	Vertical	28.4	5x	340	15.1	13.3	7	5.9
5S Sphere	Stretch	16.2	3x	324	8	7.5	9.5	7.9
5th Stage Cyliner	Horoz	6	1x	360	4	3.7	3	6.4
5D Sphere	Horoz	17.4	3x	348	4.5	5	6	7.6
Other Structures								
Cooler Platform	Horoz	10.8	2x	324		8	14	9.6
	Stretch	10.8	2x	324		15	29	9.6
Int Brg Pedestal	Stretch	60.3	11x	329	0.4	0.55	0.7	0.85

As noted all three HP compressor trains were installed in Phase 2. Figure 18 shows the installation of the trains.

Figure 18. HP Injection Compressor—Phase 1 Installation.

INTERMEDIATE BEARING PEDESTAL

The intermediate bearing pedestal for the LS torque shaft is shown in Figure 19. This intermediate bearing was needed to support the longer torque shaft. The roller bearing is grease lubricated. The roller bearing has 22 rollers and a pitch circle diameter of 10.125 inches. The roller diameters are 1.063 inches.

Due to the geometry of the roller bearing, several bearing frequencies can be generated. For example, a defect on the outer race could produce a frequency of 53 to 59 Hz depending on the running speed (324 to 360 rpm). The inner race has a high frequency range of 66 to 73 Hz.

Figure 19. Intermediate Bearing for Torque Shaft.

A response of the intermediate bearing pedestal was observed at 60.3 Hz in the stretch direction. The maximum vibration was 0.7 mil p-p due to the 11th compressor harmonic while operating at 329 rpm. Converted to velocity units, the maximum bearing vibration was 0.13 ips peak. The bearing manufacturer stated that it is not unusual for a properly installed bearing to have initial vibration levels as high as 0.16 ips peak. Generally, vibration below 0.1 ips peak does not pose a problem. The recommended set points of 0.6 ips for alarm and 0.8 ips peak for shutdown.

Compressor Frame Unbalanced Forces and Moments

During the initial running tests at the compressor packager's facility it was noticed that the baseplates were exhibiting horizontal movement, from side to size. The packager felt that it must be attributed to the lack of a permanent foundation and the attachment of the baseplate to the shop floor. Dial indicators placed on the sides of the compressor baseplate continued to show movement.

At the permanent jobsite frame vibration was measured. It exceeded the compressor manufacturer's recommended alarm settings. The analysis indicated that the unbalanced forces were quite high. Each crankshaft has four webs but only two counterweights were installed.

The compressor manufacturer's original counterweights were 2250 lb. inch, and they were applied on the two outer webs. Counterweights on these webs have the strongest influence on the shaking moment, because they have a separation of 45.88 inches. The inner webs are separated by only 16.12 inches.

The largest practical counterweight according to the compressor manufacturer is 6279 lb inch. They recommended replacing only the existing outer web weights with the larger weights, because they felt that the further benefit of added weights on the inner webs would be significantly less, and that the torsional natural frequency would be adversely affected by four counterweights.

The reduced shaking moments, the reduced natural frequency, and the influence on torsional stress with the original configuration were calculated, two added counterweights, and four new counterweights. Figure 20 presents their data for shaking moments graphically. This clearly shows the best configuration is four large counterweights.

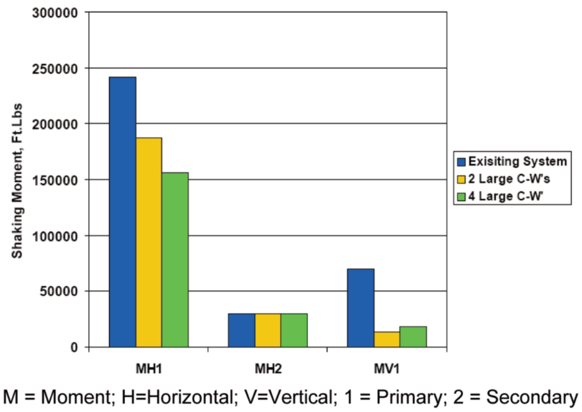

Figure 20. Effects of the Reduced Compressor Counterweights.

Both horizontal and vertical primary moments are reduced in this way. The larger counterweights also resulted in additional reduction in shaking forces, which results from equalizing the reciprocating weights by an addition to the fifth stage crosshead. The primary benefit comes from the four large counterweights, however.

Allowable vibration as measured at the centerline of the frame is 5 mils p-p, according to engineering guidance provided by the compressor manufacturer. Alarm level is 5 mils and shutdown is 6 mils p-p. With the larger crankshaft counterweights installed the maximum compressor frame vibration measured was 2.7 mils p-p at 10,000 psig pressure representing 1000 rpm engine speed and 360 rpm compressor operating speed. The larger counterweights clearly improved the compressor frame vibration significantly.

The revised torsional analysis showed that the system fifth torsional natural frequency, whose mode shape is localized to the compressor, does indeed drop from 7,052 to 5,230 Hz as a result of this counterweight change. However, the forced response stresses are essentially unaffected by the change.

In summary, the clear choice to reduce the source of the vibrations is to replace the two existing small counterweights by large counterweights (6279 lb inch) on every one of the four webs. The reduction factor is 65 percent. With these data available to the team, a rapid decision was made to expedite the delivery of four large counterweights for the first HP unit, and to follow this with similar counterweights for HP Units 2 and 3 (Figure 21).

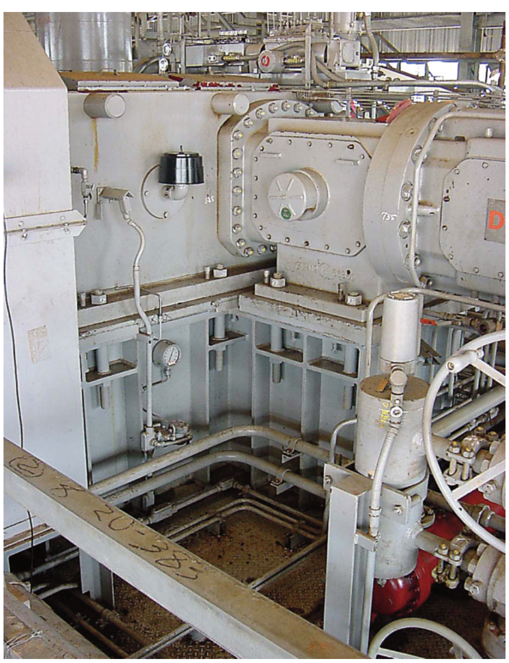

Figure 21. Compressor Frame and Fifth Stage Cross Head.

The compressor frame 1× vibration increases as the rotating unbalance increases with speed squared. The 1× vibration remained fairly constant versus final compressor discharge pressure throughout the load range as tested in the field. Therefore, it was concluded that the compressor frame vibration is more a function of unbalanced inertial forces rather than gas load.

As noted above four large rotating counterweights were installed on each compressor crankshaft replacing the two original smaller counterweights. The purpose of these larger counterweights was to reduce the compressor frame and pedestal vibration. Due to the increased inertia, these weights were also predicted to lower some of the torsional modes, particularly the compressor crankshaft mode (fifth torsional mode of the system). However, the alternating shear stress in the compressor crankshaft was predicted to remain unchanged.

The first torsional natural frequency remains unchanged at 4.2 Hz with the four larger counterweights on the compressor crankshaft. The second TNF appeared to be 25 Hz +, which is just above 1.5× maximum engine speed.

Allowable Strain Levels

One of the numerous field measurements that was acquired included dynamic strain from many of the system components including nozzles of the suction scrubbers and pulsation dampeners.

There are criteria available for determining if vibration-induced stresses would be sufficient to cause piping failures. The most direct method was to install strain gauges at the high stress locations and measure vibration-induced strain directly.

Strain gauges were installed to evaluate the dynamic strain at various vessel nozzles. Based on the consultant's experience, failures seldom occur in most low carbon welded steel piping when measured strain levels are less than 100 micro-strain (μ-in/in) peak-peak. Failures occur frequently when measured strain levels are greater than 200 micro-strain peak-peak. The range of 100 to 200 micro-strain peak-peak is considered marginal with some failures occurring at these strain levels, especially when high stress risers are present. Figure 22 shows strain gauges on the fifth stage suction sphere outlet nozzle of the HP injection compressor.

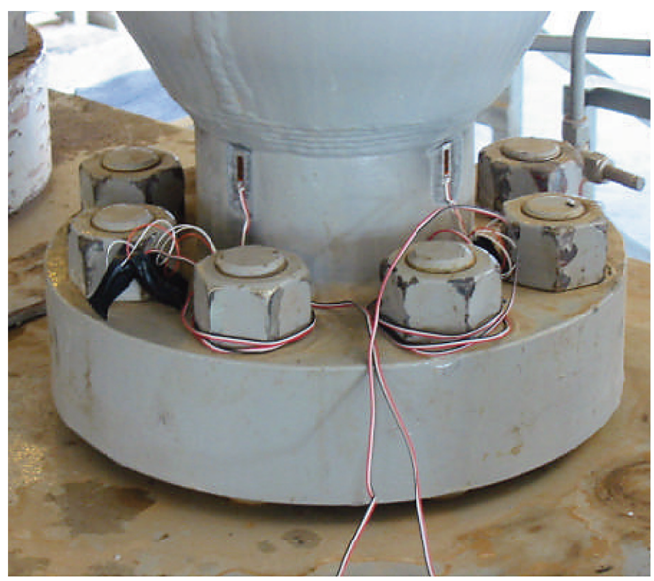

Figure 22. Strain Gauges on Fifth Stage Suction Sphere Outlet Nozzle. (Courtesy Engineering Dynamics Inc.)

The results of the dynamic strain data are shown in Table 16. Locations in red are well above the safe level of 100 micro-strain p-p recommended.

Table 16. Results of the Dynamic Strain Data.

Measurement Location	Freq, Hz	Compressor		5th Stage Discharge Press (Psi)		
		Harm	RPM	8,000 psig	9,000 psig	10,000 psig
4th Stage System						
4S Sphere Cyl Noz Side	22.5 Hz	4x	338	95	60	100
4S sphere Cyl Noz Front	18 Hz	3x	360	80	88	95
4D Sphere Cyl Noz Side	165-179 Hz			75	82	90
4D Sphere Cyl Noz Front	96, 165-169	16x, 28x-31x	360	115	120	135
5th Stage System						
5S Scrubber Out Noz Frt	16.2, 28.4	3x, 5x	324, 341	200	195	170
5S Sphere Inlet Noz Frt	16.2, 89	3x, 15x, 16x	324, 356, 334	100	95	120
5S Sphere Cyl Noz Side				85	82	75
5S Sphere Cyl Noz Front	16.2, 28.4	3x, 5x	324, 341	185	180	160
5D Sphere Cyl Noz Side	17.5	3x	350	110	---	102
5D Sphere Cyl Noz Front	96, 146-148	16x, 26x	360, 337-342	140	170	165

The engineering team and engineering consultants took the field dynamic strain measurements for the HP injection gas compressors and at various injection pressures and times. The data were plotted and are shown in Figure 23 (Figure 24).

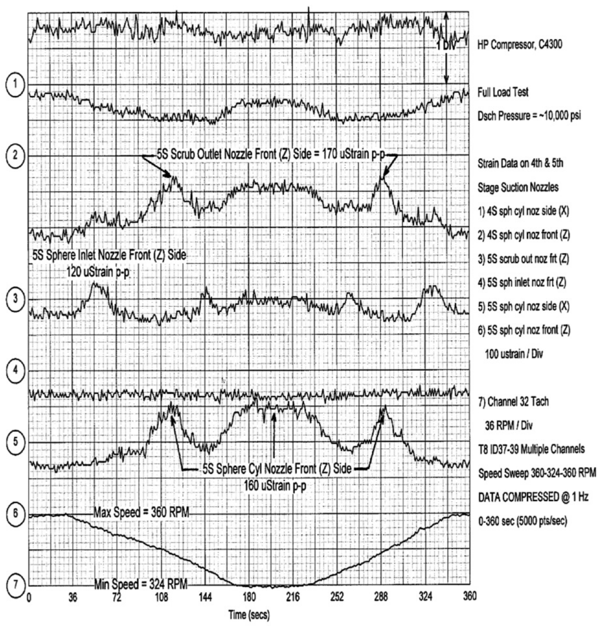

Figure 23. Dynamic Strain Measurements of the HP Injection Gas Compressor at 10,000 PSIG. (Courtesy Engineering Dynamics Inc.)

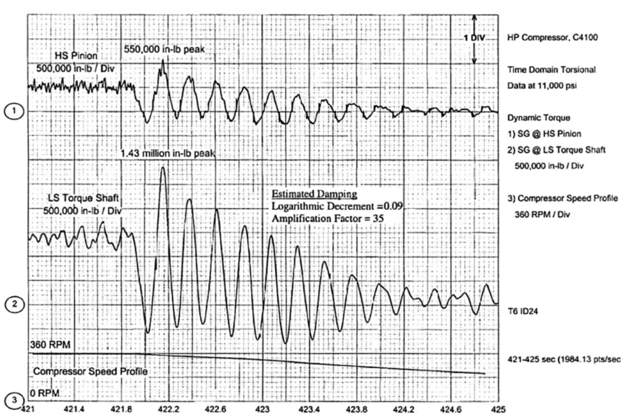

Figure 24. Time Domain Torsional During Shutdown from 11,000 PSIG. (Courtesy Engineering Dynamics Inc.)

By analyzing the field data it was possible to determine the actual shutdown torque and corresponding stresses in the 26 foot long torque shaft, this was important because the authors used the data to calculate the corresponding torqueshaft life for various safety factors. The data are plotted in Figure 25.

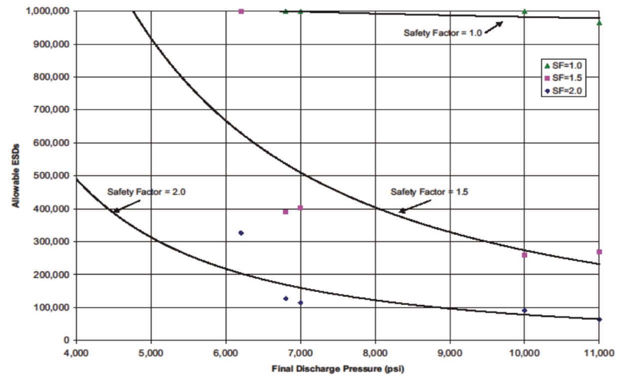

Figure 25. Allowable Emergency Shutdowns Versus Compressor Final Discharge Pressure.

During the design phase of the project the engineering team determined that the minimum safety factor that would be accepted was 1.5, although 2.0 was initially specified. The curve shows that for a safety factor of 1.5 and for emergency shutdowns from 10,000 psig discharge pressure the allowable number of such shutdowns is approximately 275,000. The field test therefore shows that the redesigned 26 foot long torque shaft as specified and designed has an infinite life. That was extremely good news to the owner of the facility and the operators.

OPERATIONAL DATA

The operating data for the Phase 1 hp injection gas compressor was recorded and stored. It is shown in Table 17. This initial data included operating information for 10,000 psig injection gas pressure.

Table 17. Initial Operating Data of HP C-4100 Compressor Unit.

Field Testing C-4100 HP Compressor		C-4100	C-4100	C-4100	C-4100
	Specified Operating Condition	7/29/2001	7/29/2001	7/31/2001	7/31/2001
Time Data Recorded		9:30 am	3:15 pm	2:35 pm	5:50 pm
Ambient Air Temp, F	100 F	87 F	89 F	91 F	87 F
ENGINE PANEL:					
Engine inlet air temp F		125 F	132 F	136 F	129 F
Engine coolant Temp F	179 F	182 F	190 F	190 F	186 F
Engine load, %		82%	88%	97.5%	98%
Engine speed, rpm	1,000 rpm	1,000 rpm	1,000 rpm	1,000 rpm	1,000 rpm
COMPRESSOR PANEL:					
Pressures:					
4th Stage suction pressure, psig	3,069 psig	2838 psig	2907 psig	2906 psig	2904 psig
4th stage discharge press, psig	5,571 psig	5390 psig	5665 psig	5857 psig	5805 psig
5th stage suction pressure, psig	5,414 rpm	5104 psig	5351 psig	5571 psig	5503 psig
5th stage discharge press, psig	10,101 psig	8470 psig	9085 psig	10,121 psig	9,990 psig
Temperatures:					
4th stage suction temperature, F	120 F	111	119	119	116
4th stage discharge temp, F	201 F	181	192	196	192
5th stage suction temperature, F	120 F	106	115	116	111
5th stage discharge temp, F	221 F	158	172	180	175
Compressor Bearing Temp's					
Main bearing #1, F	180 F	139	144	143	147
Main bearing #2, F	180 F	137	143	141	145
Main bearing #3, F	180 F	135	141	139	142
Gear / Bearing Temperatures					
Pinion / radial brg temp, F	225 F	156 F	161 F	163 F	162 F
Pinion / radial brg temp, F	225 F	161 F	166 F	168 F	168 F
Gear / radial brg temp, F	225 F	142 F	147 F	149 F	148 F
Gear / radial brg temp, F	225 F	140F	145 F	147 F	146 F
Intermediate Bearing Temp					
Torque shaft brg oil temp, F	190 F	137 F	140 F	136 F	138 F
Gear Vibration Data					
Pinion radial vibration, mils p-p	2.50 / 5.00	0.5 mils	0.6 mils	0.59 mils	0.60 mils
Pinion radial vibration, mils p-p	2.50 / 5.00	1.7 mils	1.570 mils	1.610 mils	1.650 mils
Gear radial vibration, mils p-p	3.50 / 6.00	0.6 mils	0.60 mils	0.60 mils	0.60 mils
Gear radial vibration, mils p-p	3.50 / 6.00	0.50 mils	1.100 mils	0.710 mils	0.730 mils
Engine & Compressor Vibration					
Engine vibration, ips	0.700 ips	0.50 ips	0.50 ips	0.50 ips	0.50 ips
Compressor vibration, ips	0.60 ips	0.20 ips	0.20 ips	0.20 ips	0.20 ips
Intermediate shaft vibration, ips	0.60 ips	0.40 ips	0.50 ips	0.30 ips	0.30 ips

As noted above the Phase 1 data from the Phase 1 hp injection gas compressor was provided to a third party engineering consultant that performed an audit. Results of the audit were presented in this paper above. Essentially the recommendations did not include anything that would prevent operations at the full rated 10,000 psig discharge pressure. In July, 2001, the first HP injection compressor was operated up to 10,000 psig discharge pressure. Actual operating data from this first unit run is shown in Table 17.

As noted previously there were numerous design changes made in the remaining two HP injection gas compressor trains and then these (Phase 2) units were installed. Operational data were taken; these are shown in Table 18.

Table 18. Operating Data of Phase 2 High Pressure Compressor, C-4300.

Field Testing C-4300 HP Compressor		C-4300	C-4300	C-4300	C-4300
	Specified Operating Condition	3/18/2002	3/18/2002	3/18/2002	3/18/2002
Time Data Recorded		12:39 pm	10:50 am	5:00 pm	5:15 pm
Ambient Air Temp, F	100 F	94 F	95 F	96 F	95 F
ENGINE PANEL:					
Engine inlet air temp F		125 F	127 F	127 F	127 F
Engine coolant Temp F	179 F	175 F	175 F	175 F	175 F
Engine load, %		60 %	81 %	91 %	98 %
Engine speed, rpm	1,000 rpm	1,000 rpm	1,000 rpm	1,000 rpm	1,000 rpm
COMPRESSOR PANEL:					
Pressures:					
4th Stage suction pressure, psig	3,069 psig	2739 psig	2,793 psig	2,892 psig	2,913 psig
4th stage discharge press, psig	5,571 psig	5,634 psig	5,086 psig	5,211 psig	5,568 psig
5th stage suction pressure, psig	5,414 rpm	5,349 psig	4,804 psig	5,105 psig	5,140 psig
5th stage discharge press, psig	10,101 psig	6,310 psig	8,115 psig	9,201 psig	9,955 psig
Temperatures:					
4th stage suction temperature, F	120 F	99 F	100 F	98 F	98 F
4th stage discharge temp, F	201 F	176 F	174 F	170 F	170 F
5th stage suction temperature, F	120 F	114 F	119 F	122 F	124 F
5th stage discharge temp, F	221 F	133 F	164 F	189 F	200 F
Compressor Bearing Temp's					
Main bearing #1, F	180 F	128 F	145 F	147 F	149 F
Main bearing #2, F	180 F	132 F	146 F	148 F	149 F
Main bearing #3, F	180 F	126 F	144 F	146 F	147 F
Gear / Bearing Temperatures					
Pinion / radial brg temp, F	225 F	147 F	155 F	15 F 8	159 F
Pinion / radial brg temp, F	225 F	150 F	160 F	162 F	163 F
Gear / radial brg temp, F	225 F	131 F	141 F	143 F	144 F
Gear / radial brg temp, F	225 F	134 F	142 F	146 F	147 F
Intermediate Bearing Temp					
Torque shaft brg oil temp, F	190 F	105 F	115 F	114 F	115 F
Gear Vibration Data					
Pinion radial vibration, mils p-p	2.50 / 5.00	0.570 mils	0.60 mils	0.50 mils	0.50 mils
Pinion radial vibration, mils p-p	2.50 / 5.00	1.10 mils	1.00 mils	1.20 mils	1.20 mils
Gear radial vibration, mils p-p	3.50 / 6.00	0.70 mils	0.60 mils	0.60 mils	0.60 mils
Gear radial vibration, mils p-p	3.50 / 6.00	0.750 mils	0.80 mils	0.80 mils	0.80 mils
Engine & Compressor Vibration					
Engine vibration, ips	0.700 ips	0.40 ips	0.40 ips	0.50 ips	0.50 ips
Compressor vibration, ips	0.60 ips	0.10 ips	0.10 ips	0.20 ips	0.20 ips
Intermediate shaft vibration, ips	0.60 ips	0.10 ips	0.20 ips	0.20 ips	0.20 ips

One of the most significant changes made to the Phase 2 units was to increase the size of the suction and discharge pulsation dampeners (spheres) in order to reduce the peak-to-peak pulsation level in the connected piping.

Operating Deflection Shapes

During the field testing the engineering consultant performed an operating deflection shape (ODS) at the rated condition noted in Table 19. The engine was held at the maximum speed of 1,000 rpm and the compressor speed was 360 rpm. The final discharge pressure was maintained by opening the recycle valve and pinching back on the gas flow to the injection well.

Table 19. Operating Conditions During the Operating Deflection Shape Measurements.

	Suction Pressure, PSIG	Discharge Pressure, PSIG
4th Stage	2,950 psig	5,400 psig
5th Stage	5,050 psig	9,900 psig

Operating deflection shapes are used to visualize the vibration by exaggerating the amplitudes. Vibration data are acquired at each point in the wire-frame model of the structure in three directions. The test point locations are shown in Figures 26 and 27. A keyphasor is used to synchronize the amplitude and phase readings at each point. These data are processed to produce the animated display of the structural vibration shapes using a structural measurement systems program. Avi files were prepared to show the motion at 1×, 2×, and 3× compressor speed.

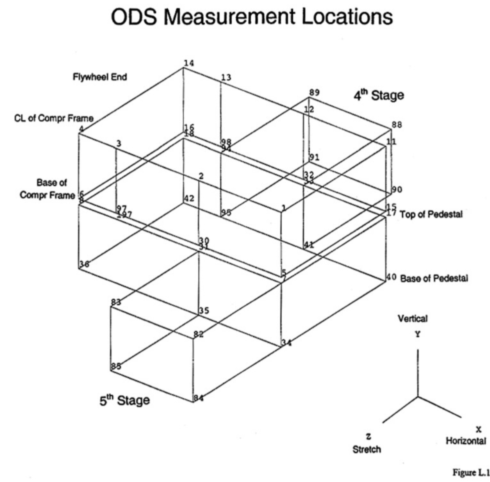

Figure 26. ODS Measurement Locations, C-4300 HP Injection Gas Compressor (A). (Courtesy Engineering Dynamics Inc.)

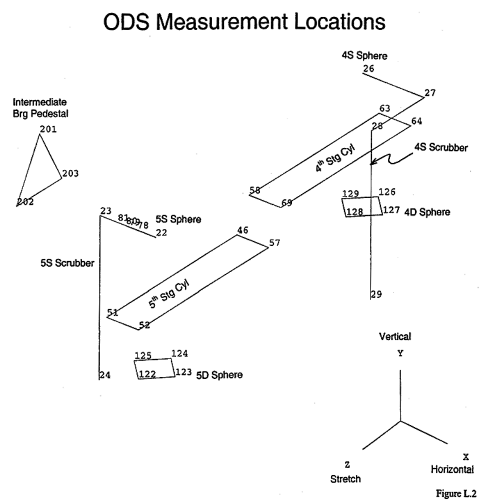

Figure 27. ODS Measurement Locations, C-4300 HP Injection Gas Compressor (B). (Courtesy Engineering Dynamics Inc.)

The measured 1× vibration levels at each point are listed in Figure 26 and 27. The maximum 1× vibration at various locations and elevations is summarized in Table 15. In the stretch direction, the vibration levels on the base of the compressor frame and pedestal were 2.3 mils p-p or less. The vibration at the centerline of the compressor frame was 3.2 mils p-p or less in the stretch direction. These levels meet the compressor manufacturer's maximum allowable vibration criteria and were considered acceptable.

Selected single frame representations of the 1× operating deflection shape animations illustrate the vibration characteristics of the compressor. The deflections at 1× running speed show some twisting of the compressor frame about the vertical axis due to the primary horizontal moment. Compressor cylinder vibration in the horizontal direction can also be observed. The compressor frame and cylinder vibrations are below allowable levels and considered to be acceptable.

A 1× operating deflection shape clearly shows the motion of the cylinders and compressor frame in Figure 28. The data and deflection diagram are for data plotted with the HP injection gas compressor operating at 10,000 psig injection pressure. The team wanted to ensure that the deflection and vibration at the highest injection pressure were within the manufacturer's design limits.

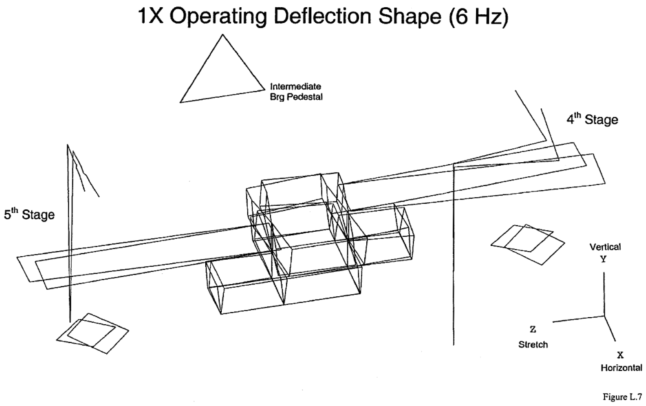

Figure 28. Operating Deflection Mode for C-4300 Unit at 10,000 PSIG Discharge Pressure. (Courtesy Engineering Dynamics Inc.)

CONCLUSION

The success of the high pressure gas injection project can be attributed to identification of design issues with the original gas compression train and the subsequent resolution and implementation of the changes. When the EPC design engineering phase of the project was initiated the equipment had already been purchased in the front end phase and had been put into manufacturing.

Once the design issues were identified a comprehensive engineering program commenced almost immediately and a plan implemented to provide early high pressure injection gas with one HP injection gas train while engineering and manufacturing the two (2) remaining trains with all design changes implemented.

The field testing verified and proved the design changes were successful, as the design life of the infamous 26 foot torque shaft connecting the low speed gear shaft to the compressor went from 11 cycles (1 shutdown cycle) to approximately 275,000 shutdown cycles.

High pressure natural gas injection trains using natural gas fired engines and low speed reciprocating gas compressors can be desiged to operate safely while injecting gas at 10,000 psig. The project stressed engineering, operations, safety, and environmental concerns. This technology is available for use for projects that require high pressure injection of natural gas and where the application of a synchronous electric motor driver cannot be accommodated or is not practical.

NOMENCLATURE

0-p = 0 to peak, vibration displacement
API = American Petroleum Institute
Bhp = Brake horsepower
Bopd = Barrels of oil produced per day
C = Centigrade, degrees
cpm = Cycles per minute
F = Fahrenheit, degrees
HP = High pressure system or compressor
Hz = Hertz, frequency units cycles per second
IP = Intermediate pressure system or compressor
ips = Inches per second, vibration measurement
KO = Knockout drum
LP = Low pressure system
LLP = Low low pressure system
ODS = Operating deflection shape
psi = Pounds per square inch
psig = Pounds per square inch gauge
mmscfd = Millions standard cubic feet per day, gas handled
p-p = Peak-to-peak, vibration displacement
P&ID = Process and instrumentation diagram
rpm = Revolutions per minute
TNF = Torsional natural frequency, Hz
Z = Compressibility of gas

REFERENCES

ASME B106.1M, 1985, *Design of Transmission Shafting*, American Society of Mechanical Engineers, New York, New York.

Feese, T. and Hill, C., 2009, "Prevention of Torsional Vibration Problems in Reciprocating Machinery," *Proceedings of the Thirty-Eighth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 213-238.

MIL STD 167-1 (SHIPS), May 1974, "Mechanical Vibrations of Shipboard Equipment (Reciprocating Machinery and Propulsion System and Shafting) Type III, IV and V."

Peterson, R. E., 1974, *Stress Concentration Factors*, New York, New York: John Wiley & Sons.

Shigley, J. E. and Mischke, C. R. 1989, *Mechanical Engineering Design*, New York, New York: McGraw-Hill Inc.

Taylor, C. F., *The Internal Combustion Engine in Theory and Practice*, "Volume 1: Thermodynamics, Fluid Flow, Performance, Second Edition, Revised," and "Volume 2: Combustion, Fuels, Materials, Design, Revised Edition."

Wachel, J. C. and Szenasi, F. R., 1993, "Analysis of Torsional Vibrations in Rotating Machinery," *Proceedings of the Twenty-Second Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas pp. 127-152.

Wilson, W. K., 1958, *Practical Solution of Torsional Vibration*, British Internal Combustion Engine Research Association.

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