FULL LOAD, FULL SPEED TEST OF TURBOEXPANDER-COMPRESSOR WITH ACTIVE MAGNETIC BEARINGS

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INTRODUCTION

The application of the turboexpander in natural gas processing and the petrochemical industry had its beginning at a small gas plant in Southwest Texas where Dr. Judson S. Swearingen installed the first natural gas turboexpander (Swearingen, 1999).

Turboexpander technology has developed considerably in the last 40 years. For example:

- Advances in fluid dynamics theory and computational fluid dynamics have made it possible to design a turboexpander with high isentropic efficiency and performance predictability;
- Progress in rotordynamics evaluation and modern finite element analysis capabilities have resulted in more reliable turbomachinery.
- Increase in demand and economies of scale have resulted in natural gas processing and petrochemical plants becoming larger and larger (Figure 1) (Agahi, 2003).

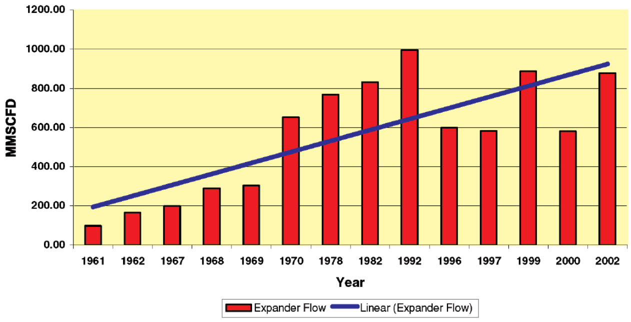

Figure 1. Turboexpander Flow Development Since 1960.

Three large size turboexpander compressor (TEC) units with active magnetic bearings (AMB) were installed in a gas plant in the early 1990s (Agahi, et al., 1994). Because of some difficulties with the operation of the inlet guide vanes (IGV) at the commissioning stage caused by hydrate formation, oil and gas companies and engineering-procurement companies (EPCs) were somewhat skeptical about AMB technology and its reliability. Furthermore, with more offshore platforms using turboexpander compressor units, ease of commissioning and reliability were among the major concerns of end users.

In order to address these concerns, the end users of turboexpander compressor units began to demand testing of the mechanical and control designs of custom designed equipment, and full load, full speed testing (FLFST) of turboexpander compressor units. The first tests were carried out in the late 1990s (Bergmann, et al., 1996). These FLFSTs used a hydrocarbon gas mixture that was intended to simulate the actual process gas as closely as possible.

Full load, full speed testing of a turboexpander compressor with hydrocarbon gas proved to be somewhat difficult and too expensive. Furthermore, obtaining permits to conduct such tests is a challenge for most test facilities. To circumvent some of these issues, a mixture of nitrogen and helium was used in some turboexpander compressor FLFSTs in early 2000. In this approach, a mixture of nitrogen and helium formulated to simulate the molecular weight of the actual process gas is used in a closed test loop.

BACKGROUND

A major Middle East gas production company was contemplating the development of six large liquefied natural gas (LNG) plants. The designs were to be similar, all based on the use of turboexpander compressor units for maximum production of LNG. The turboexpander compressor power was estimated to be in the range of 9.0 to 10.0 MW, assuming that some of the plants would have units in parallel operation to limit the size of the expanders to modules already demonstrated.

The company had experience with similar plants and had identified a large turboexpander compressor as a major contributor to plant outages or production limitations. A thorough review of the problems experienced in the existing plants was conducted with the responsible operating and rotating equipment engineers. The objective of the review was to identify specific design features of the turboexpander compressor that required upgrading from previous designs.

Several specific features of the existing turboexpander compressor were identified as responsible for the great majority of equipment outages. In order of importance, these were:

- Dilution of lubricating oil by process gas and subsequent loss of oil viscosity resulting in excessive vibration and bearing wear.
- Inadequate design of the thrust balancing provisions and poor operation of the thrust balance mechanism resulting in thrust bearing overload.
- Clamping of inlet guide vanes due to lack of controls resulting in erratic unit performance.

The review group concluded that a new, large turboexpander compressor application should incorporate features to address those deficiencies and that the turboexpander compressor should be tested as thoroughly as possible prior to shipment.

Turboexpander compressors now employ magnetic radial and thrust bearings as a solution to the major problem of oil contamination. Thrust calculations and thrust balancing designs have improved and the load can be measured with the active magnetic bearing system. Inlet guide vane controls were updated by the addition of an electric actuator but it was not certain that the control would be as accurate as desired. The active magnetic bearing system, however, lacks redundancy and presents the new concern of a component failure resulting in a loss of levitation under loaded conditions. In order to demonstrate the ruggedness of the design, an FLFST was planned in series with the residue gas compressor for the same plant.

The turboexpander compressor package for this project used 240 mm (9.45 inch) active magnetic bearings. Table 1 shows the turboexpander compressor wheel characteristics. This package was to be installed in a closed loop with other LNG train equipment such as a gas turbine (GT) driven residue gas compressor. The project GT provides the power required to drive the residue gas compressor, which in turn delivers the required flow and pressure to the turboexpander compressor to bring the latter up to the full speed, full load condition. Table 2 shows the design guaranteed conditions and Table 3 depicts the simulated conditions for the FLFST.

Table 1. Turboexpander Compressor Wheel Characteristics.

	Expander	Compressor
Wheel type	Open	Open
Wheel Diameter (Inches)	18.875	21.625
Number of blades	11 / 11	6 / 6
Rated RPM	11250	11250
Tip speed (Ft/sec)	927	1062
Material	TI-6Al-4V	AL 7050
IGV Type	Variable	N/A
Number of IGV vanes	4	N/A
Design flow coefficient	0.084	0.143
Design head coefficient	2.25	0.94

Table 2. Turboexpander Compressor Design/Guaranteed Operating Conditions.

Components	Expander	Compressor	Units
Helium	0.04	0.04	mol%
Methane	87.54	90.29	mol%
Ethane	5.46	5.65	mol%
Propane	1.95	0.01	mol%
i Butane	0.32		mol%
n Butane	0.48		mol%
i Pentane	0.14		mol%
n Pentane	0.11		mol%
n Hexane	0.04		mol%
n Heptane	0.02		mol%
COS	0.00013	0.00001	mol%
CS2	0.0009		mol%
Methyl Mercaptane	0.0009		mol%
Helium			mol%
Nitrogen	3.89	4.01	mol%
Carbon Dioxide	0.005	0.00517	mol%
Hydrogen Sulfide	0.0003	0.00031	mol%
Process Conditions			
Mol Weight	18.34	17.31	
Inlet Pressure	64.16	21.1	bar a
Inlet Temperature	-14.4	19.7	°C
Outlet Pressure	22.06	29.7	bar a
Outlet Temperature	-64	48.8	°C
Flow	525,700	671,100	kg/hr
Speed	11,250	11,250	rpm
Efficiency	88	81	%
Power	15,156	14,936	hp

Table 3. Turboexpander Compressor FLFS Operating Conditions.

Components	Expander	Compressor	Units
Helium			
Methane	88.8	88.8	mol%
Ethane	8.1	8.1	mol%
Propane	1.8	1.8	mol%
i Butane	0.2	0.2	mol%
n Butane	0.3	0.3	mol%
i Pentane			
n Pentane			
n Hexane			
n Heptane			
COS			
CS2			
Methyl Mercaptane			
Helium			
Nitrogen	0.8	0.8	mol%
Carbon Dioxide			
Hydrogen Sulfide			
Process Conditions			
Mol Weight	18.02	18.02	
Inlet Pressure	65.0	21	bar a
Inlet Temperature	50	-2.4	°C
Outlet Pressure	21.06	32.0	bar a
Outlet Temperature	-20.8	43.4	°C
Flow	440,826	440,826	kg/hr
Speed	11,250	11,250	rpm
Efficiency	81	78	%
Power	15,156	14,936	hp

FULL LOAD FULL SPEED TEST OBJECTIVES

The main objectives of the FLFS test that were agreed to by the customer and expander manufacturer are as follows:

- Verification of the mechanical integrity of the turboexpander compressor with active magnetic bearings
- Verification of the control functions related to the inlet guide vanes of the turboexpander compressor unit, automatic thrust balancing, and the antisurge valve while operating at FLFS
- Identification and correction of any faults or defects of the turboexpander compressor system and repeat of the FLFS test to verify that the issues were indeed rectified
- The FLFST sequence is shown in Figure 2

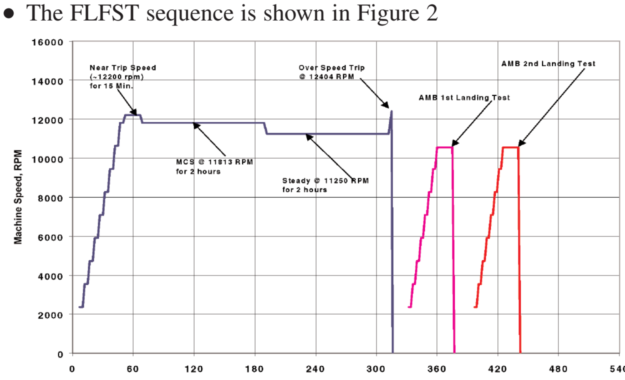

Figure 2. Full Load/Full Speed Test Sequence.

The following conditions were to be monitored and recorded:

- Startup
- Near trip speed operation

Full load operation, turboexpander compressor operates at maximum continuous speed (MCS)

- Partial load operation
- Design/normal speed operation
- Normal shutdown
- Emergency shutdown (ESD) due to process upset
- Two coastdown auxiliary bearing landing tests
- Verification of critical functions such as ESD and process shutdown system trips

THE CLOSED LOOP TEST SETUP

Figure 3 shows a schematic of the test setup and Figure 4 shows the FLFST piping around the turboexpander compressor. The turboexpander compressor unit was integrated into the residue gas compressor (RGC) test loop. The project residue gas compressor was driven by its dedicated GT and delivered test fluid to the expander at 64 barg (928.2 psig) and 50ºC (122ºF). The expander extracted energy from the gas stream by expanding to 20 barg (290.1 psig) and cooling down to −21ºC (−5.8ºF). The cold gas from the expander discharge was then fed to the recompressor, which used the expander power and boosted the test fluid to a discharge pressure of 31 barg (449.6 psig) before returning it to the RGC to repeat the cycle.

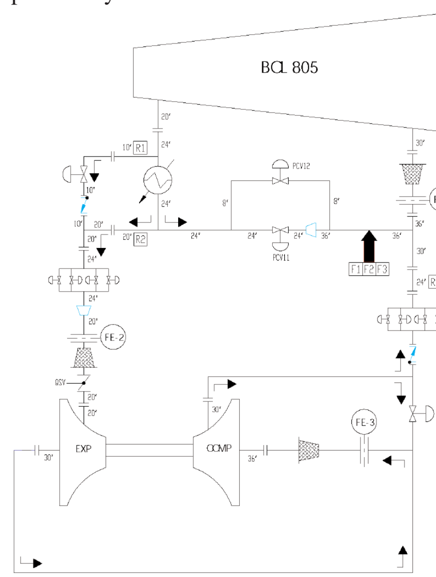

Figure 3. Closed Loop Process and Instruments Diagram for Full Load/Full Speed Test.

Figure 4. Test Loop Around Turboexpander Compressor.

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EQUIPMENT

The major components and supporting equipment in this test loop are listed below:

- Residue gas compressor, centrifugal compressor driven by a GT
- Residue gas compressor aftercooler
- Residue gas compressor auxiliary support system
- Test loop piping
- Flow measuring devices
- Pressure measuring devices
- Temperature measuring devices
- Pressure transducers, cabling, data acquisition/analysis equipment
- Startup seal gas supply, instrumentation air, etc.
- Gas analyzer and reporting system
- Vibration recording and analyzing equipment compatible with active magnetic bearing system
- Residue gas compressor inline inlet strainer (60 mesh)
- Noise meter

The auxiliary equipment and components were as follows:

- Turboexpander compressor package with control system, including active magnetic bearing signal interlock with test facility and startup seal gas supply
- Expander bypass, Joule Thompson (JT) valve
- Expander inlet quick shutoff valve
- Compressor surge control system and recycle valve
- Check valve downstream of the recompressor
- Expander inline inlet strainer (60 mesh)
- Compressor inline inlet strainer (20 mesh)

The majority of the operating parameters such as flow rates, pressures, temperatures, vibration, etc., were monitored and logged by the automatic data acquisition system.

Considering the practical aspects of the FLFST, some design parameters could not be simulated. Figure 5 shows process parameters that were different during the FLFST compared to the normal site conditions.

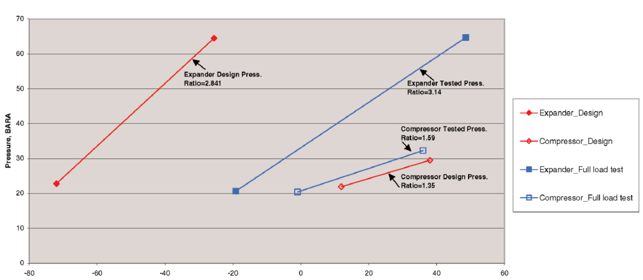

Figure 5. Comparison Between the Design and Test Parameters.

FLFST Operation

Before startup, the test loop was pressurized to 24 barg (348.1 psig). The test header pressure reached 52 barg (754.2 psig) after the residue gas compressor developed a stable pressure. Flow was admitted to the turboexpander compressor by opening the inlet guide vanes and closing the JT valve simultaneously. The recompressor antisurge valve was at the full open position at the startup. The antisurge valve began closing to load up the recompressor when the turboexpander compressor speed reached approximately 5000 rpm.

The TEC speed was ramped up at 10 percent increments until the speed closely approached the trip speed of 12,400 rpm and remained at that speed for 15 minutes. Then the turboexpander compressor speed was reduced to a maximum continuous speed (MCS) of 11,813 rpm for two hours. The test loop equilibrium state was achieved by slowly adjusting the recycle valves of the recompressor and residue gas compressor. The turboexpander compressor speed was further reduced to the normal speed of 11,250 rpm and operated at this speed for another two hours. At the end of this test run, the turboexpander compressor speed was gradually increased to the shutdown speed and the turboexpander compressor tripped on high speed. The turboexpander compressor rotor coasted down under normal conditions, i.e., the active magnetic bearing system was in levitating mode during coastdown. Table 4 shows a sample of the FLFST parameters that were monitored.

Table 4. A Sample of Test Parameters Monitored/Recorded.

Data point	9	10	11	12	13	14	15	16
Exp_Comp Speed (RPM)	12140	12150	11820	11740	11690	11764	11781	11781
Baro (BAR)	1.019	1.019	1.019	1.018	1.018	1.018	1.018	1.018
Exp_P1(BARG)	50.567	50.285	64.849	64.516	64.084	64.769	64.726	64.711
Exp_P2(BARG)	15.698	15.444	20.463	20.672	20.885	20.54	20.569	20.579
Exp_T1(C)	44.11	42.15	46.527	46.822	47.067	47.672	48.097	48.206
Exp_T2(C)	-23.236	-25.512	-21.614	-20.716	-19.636	-20.225	-19.746	-19.603
Comp_P1(BARG)	15.512	15.262	20.212	20.429	20.643	20.297	20.335	20.331
Comp_P2(BARG)	25.467	25.193	32.232	32.341	32.4	32.283	32.31	32.304
Comp_T1(C)	-4.332	-6.825	-3.499	-2.813	-1.963	-2.055	-1.582	-1.457
Comp_T2(C)	34.081	31.91	33.293	33.718	34.09	34.716	35.178	35.313
Seal gas flow (NM3)	2819	2111	2871	2841	2830	2826	2843	2847
Exp_PWB (BARG)	16.7	16.4	21.9	22	22.1	22.1	22	22
Comp_PWB (BARG)	16.3	16	21.2	21.3	21.5	21.5	21.2	21.3
IGV_instruction (%)	75	75	74	74	74	74	74	74
IGV_feedback (%)	74	74	74	74	74	74	74	74
Vent gas temp. (C)	72.6	73	68.8	69.1	69.1	69.1	70.1	70.5
Anti_Surge_Valve positio	21	21	24	24	24	24	24	24
Z1_Thrust curretn (AMP)	12.1	12.1	11.9	12.3	12	12	12.2	12
Z2_Thrust curretn (AMP)	7.3	7.3	7.4	7.1	7.3	7.4	7.2	7.4
ATB % opening	54.1	54.1	55.9	55.9	55.9	55.9	55.9	55.9

The turboexpander compressor was restarted and the speed was increased to 10,550 rpm for about 15 minutes in order for the system to reach thermal equilibrium. A special method was applied to bypass the active magnetic bearing controller and completely disable the AMB amplifiers in order to activate delevitation.

INLET GUIDE VANE SENSITIVITY AND FLOW CONTROLLABILITY

The turboexpander compressor flow is linearly proportional to the opening of its inlet guide vanes except in small opening and full open positions. By controlling the sensitivity of the inlet guide vanes to within 1 percent, i.e., deviation between process signal to inlet guide vanes and feedback signal to actuator system, it could be demonstrated that the expander flow controllability is within 1 percent of the total flow. As the trended data in Figure 6 show, the differences between the inlet guide vane input signals and the corresponding feedback signals were mostly less than1.0 percent from ramp up to the FLFST condition. It is interesting to note that the injection of additional fluid to increase the test loop pressure did not influence the inlet guide vane sensitivity or flow controllability. The inlet guide vane sensitivity remained within 1.0 percent even when the expander inlet pressure was increased to 64 barg (928.2 psig).

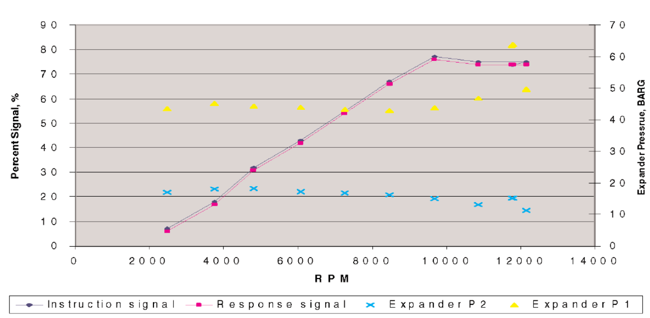

Figure 6. IGV Sensitivity Analysis.

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AMB AUXILIARY BEARING LANDING TEST

Auxiliary bearings support the turboexpander compressor rotor when it is not levitated. Another function of the auxiliary bearings is to catch the rotor upon loss of the magnetic field resulting in delevitation as the machine coasts down from full load and full speed (FLFS) to a full stop. To demonstrate the functionality of the auxiliary bearings and show their ability to support the rotor upon delevitation at FLFS, landing tests were performed during turboexpander compressor shop tests. To implement this test, the turboexpander compressor speed was increased to 10,550 rpm, and then the expander was delevitated by intervening with the active magnetic bearing control system. The delevitation signal shut off the expander inlet quick shutoff valve and opened the JT valve. The active magnetic bearing controllers were bypassed by introducing jumpers in the control cabinet. As a result, all radial and axial amplifiers were disabled, and the rotor landed on the auxiliary bearings and coasted down to a complete stop. For both landings, it took approximately 2.4 seconds from the rotor landing until the quick shutoff valve shut off; it took 4.6 seconds for the turboexpander compressor to coast down to a complete stop; the rotor landed in the auxiliary bearings for a total of 7.0 seconds; and rotor whirling stopped within 4.0 to 4.2 seconds. Figures 7 and 8 provide detailed records of these tests.

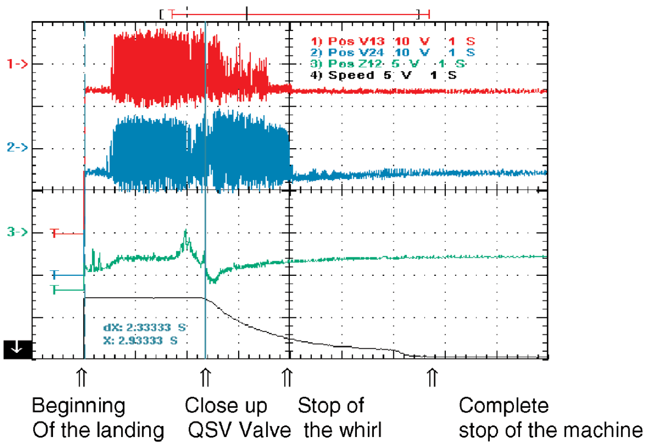

Figure 7. First Landing Test.

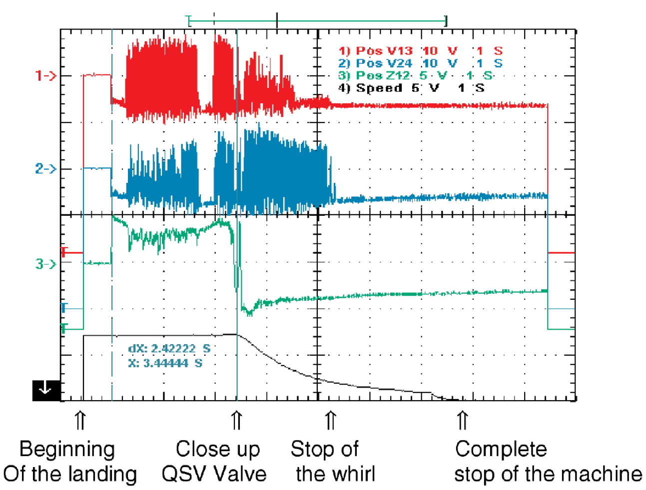

Figure 8. Second Landing Test.

Before and after the landing test, both the radial and axial air gaps between the rotor and auxiliary bearings were measured, compared, and contrasted (Table 5). The data showed that there were no changes in gap dimensions after two landing tests.

Table 5. Comparison of Radial and Axial Air Gap Before and After Landing Tests.

2 Axis	V13	W13	Z12	V24	W24
Nominal gap (+) in microns Nominal gap (-) in microns	180 180	180 180	300 300	180 180	180 180
Before the landing test	+5,36V 178μm -5,36V 178μm	+5,60V 186μm -5,20V 173μm	+3,92V 326μm -4,00V 333μm	+5,28V 176μm -5,20V 173μm	+5,68V 186μm -5,60V 186μm
After the 1st landing test	+5,68V 189μm -5,44V 181μm	+5,68V 189μm -5,52V 184μm	+3,92V 326μm -3,60V 300μm	+5,28V 176μm -5,28V 176μm	+5,28V 176μm -5,60V 186μm
After the 2 landing test	+5,52V 184μm -5,52V 184μm	+5,60V 186μm -5,44V 181μm	+3,64V 303μm -3,64V 303μm	+5,36V 178μm -5,20V 173μm	+5,36V 178μm -5,68V 189μm

The tear down and inspection showed that there were light touches on the compressor impeller blade tips. The rest of the rotor and its corresponding stator parts such as the expander wheel, shaft seals, sensor rings, thrust disk, and magnetic bearings were found to have no touch marks and were in excellent condition.

There were light marks on the ball bearing inner rings and landing sleeves in both radial and axial surfaces but the ball bearings could roll freely.

ACTIVE MAGNETIC BEARING ROTOR VIBRATION

Before spinning the residue gas compressor for the FLFST, fine tuning of the active magnetic bearings, clearance, and tuning checks were carried out to ensure that the air gaps were consistent with the design values, transfer functions were up to date, and all the required securities were set correctly. The bearing system was equipped with antivibration rejection and automatic balancing system logic. The antivibration rejection activation deactivation limits were set at 3400 and 4600 rpm, respectively. The rotor first critical speed was estimated to be approximately 37 Hz. At higher speeds the automatic balancing system took over the control function. These two systems ensured that the active magnetic bearing rotor always rotates around its inertia center. Figure 9 shows the turboexpander compressor rotor vibration throughout the course of the FLFST including both landing tests. The higher rotor vibration was observed during ramp up, at a speed range between 6,000 to 9,000 rpm. The highest vibration reached 50 μm and was mainly in the subsynchronous spectrum, from approximately 37 percent to 44 percent of the synchronous frequency, and occurred only during the startup period. The vibration at this level was considered normal compared to the alarm setting of 90 μm. The tangential velocity component, i.e., exit from the inlet guide vanes, could cause swirling around the turboexpander wheel and resonate at subsynchronous frequencies. At FLFST conditions, subsynchronous displacements were almost nonexistent and overall vibration levels were about 15 μm. The axial vibration was about 13 μm.

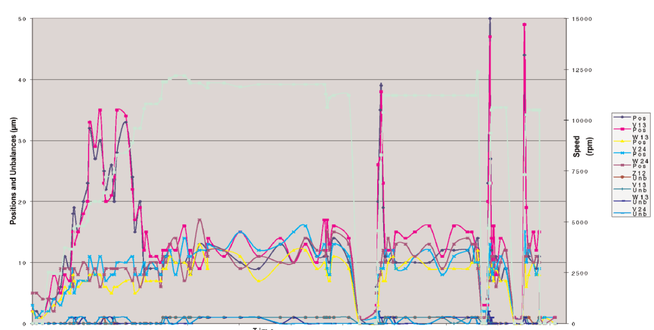

Figure 9. Rotor Total Displacement/Vibration.

The active magnetic bearing current chart, Figure 10, shows the current for each bearing during the FLFST. The relatively flat curves indicate that the rotor was quite stable at the FLFST conditions.

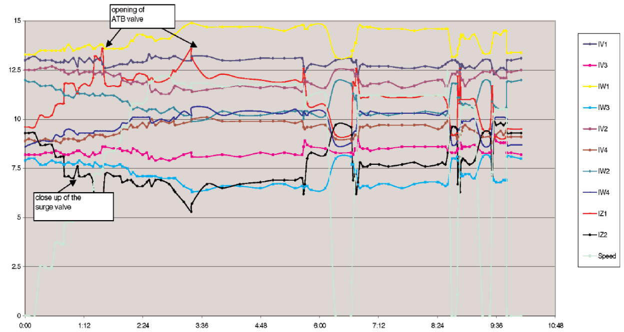

Figure 10. AMB Current.

The bearing temperature chart, Figure 11, shows that the bearing coil temperatures were normal. The highest temperature of 70ºC (158ºF) was observed when the turboexpander compressor was operated near the trip speed. The rotor length expansion was measured at −150 µm. This measurement shows that the relative distance of the rotor and housing was increasing during the test (Figure 11).

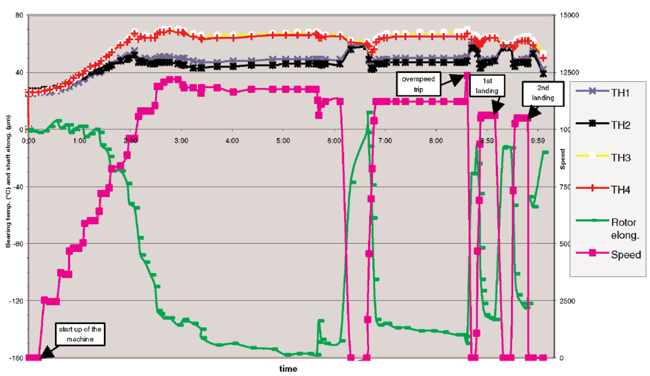

Figure 11. Shaft Extension.

ROTOR VIBRATION DURING TURBOEXPANDER TRIP

One of the FLFST criteria was to observe the rotor behavior during shutdown and coastdown to stop. Upon a request for a turboexpander compressor trip, the normal sequence of the events is to close the quick shutoff valve and open the antisurge valve. Following this procedure, during turboexpander shutdown and coastdown while the rotor was levitated, there were no significant changes in vibration levels, unbalance, temperatures, or bearing currents.

ACTIVE MAGNETIC THRUST BEARING AND AUTOMATIC THRUST BALANCING SYSTEM

The turboexpander automatic thrust balancing system operated flawlessly in conjunction with the active magnetic bearing thrust bearing control system. There were no indications of axial thrust biases throughout the FLFST. The interlock logic between the active magnetic bearing and turboexpander automatic thrust balancing system was set such that when the axial current reached 14 A, the turboexpander system would take action to open or close the automatic thrust balancing valve to relieve the thrust load. The action automatically stopped when the bearing thrust current was reduced to 12 A. During the FLFST, the valve opening varied from 50 percent to 60 percent. Figure 12 shows the axial bearing current during the FLFST.

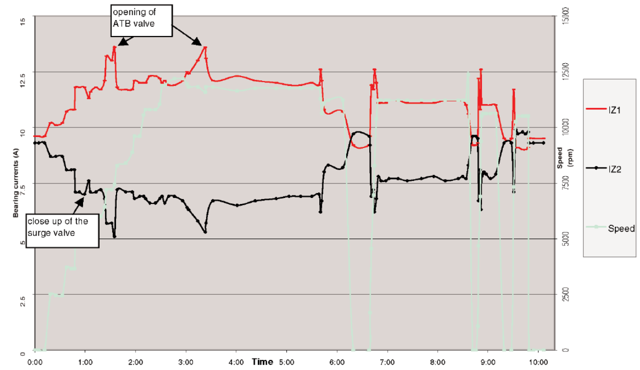

Figure 12. Axial Bearing Current.

EXPANDER INLET QUICK SHUTOFF VALVE TIMING

The project quick shutoff valve was used during the FLFST. Three trips were initiated. One trip was triggered by an overspeed trip at the end of the four-hour mechanical running test, and two trips were performed during landing tests. It took about 2.4 seconds to complete the valve closing process. This duration included the signal transmission time, the valve actuator response time, and the stem traveling time. Based on the factory bench test report, the valve closing time was 0.602 seconds. Therefore, the signal transmission time in the test loop control system took approximately 1.8 seconds.

BEARING HOUSING, SEAL GAS, AND VENT GAS TEMPERATURE MONITORING

The turboexpander compressor seal gas control system was not in operation during the FLFST. The seal gas pressure was controlled manually with a bypass valve. The bearing housing vent gas temperature alarm was initially set at 55ºC (131ºF). This setting was too low for the FLFST conditions and had to be revised. The active magnetic bearing high temperature alarm was set at 110ºC (230ºF) and shut down was set at 130ºC (266ºF). Therefore, the vent gas temperature setting was revised to 95ºC (203ºF).

TEAR DOWN INSPECTION

After completion of the four-hour mechanical running and two landing tests, the turboexpander compressor was disassembled for inspection. All parts were in good condition. The auxiliary ball bearings were replaced with a new set despite being in good condition. The turboexpander variable inlet guide vane assembly was also removed and inspected. There were scratch marks on the interfacing surfaces between the guide vanes and nozzle clamping rings as well as on the nozzle cover. These parts were sent to the metallurgical laboratory for determination of the root cause. The conclusion was that debris carried by the gas stream of the test loop was trapped in the nozzle grooves and was dragged into the interfacing surfaces causing the scratch marks when the vanes moved. All scratches were removed and the surfaces were restored before the inlet guide vanes were reassembled.

FLFST RESULTS

The turboexpander compressor operation during the FLFST was without any major problems or unexpected events. Uninterrupted FLFST continued for four hours at 12.5 MW and 11,250 rpm. The rotordynamics performance was stable and all dynamic parameters were within the design limits and as predicted. The normal trips at FLFS and ESD trips with landing at FLFS into auxiliary bearings were carried out successfully. Rotor coastdown and vibration levels for all trips were smooth and at safe levels. The inlet guide vane test for ease of operation under full pressure, FLFS, ramping up, and closing down conditions did not show any faults, or indications of blow by or clamping problems. The response of the inlet guide vanes to process signals and controllability were demonstrated to be consistent with other controls in the plant and hence could control plant flow with the desired precision. The automatic thrust balancing systems of the bearing and turboexpander maintained the axial position of the rotor in the desired center position during the various tests and operating conditions. Gas dynamics performance test results were as expected and consistent with the predicted values. Expander isentropic and compressor polytropic efficiencies were better than the guaranteed values.

LESSONS LEARNED

The FLFST requires detailed planning and coordination with the various facilities that are involved. The expander vendor had an earlier FLFST at a facility outside their group and this one was within their group. It turned out that more planning and attention to details were necessary in the latter FLFST because each team tended to leave something out anticipating that the other team would pick it up.

Turboexpander compressor units are normally skid-mounted packages ready to be installed on a foundation. This package should be complete with all auxiliary and support systems before it is installed for the FLFST. There were delays, confusion and manual (in lieu of automatic) operation because some components were not shipped to the test site.

The turboexpander compressor shutdown loops should be dedicated and have absolute minimum response time to guarantee the safety and security of the turboexpander and its processes.

The noise level for this turboexpander compressor was estimated at 85 dBA with a noise jacket installed on the casings. The turboexpander, bearing housing, and compressor did not have a noise jacket during the FLFST and no background noise correction was applied. Noise levels were measured at 115 dBA, some 20 percent higher than calculated/expected. This test helped to highlight the need to review and revise formulas and algorithms used for noise level estimation.

The integrity and ruggedness of the auxiliary bearings were tested and demonstrated by multiple landing tests. Inspection of rotor parts after landing tests showed that there was no damage. This could be considered as justification to delete landing tests that normally are requested during open loop air tests.

Gas dynamics performance tests of the expander and compressor produced the same performance that was predicted at the design stage and that would have resulted from extrapolation of the open loop air test. Therefore one may conclude that the FLFST does not provide any additional information through the gas dynamics performance tests.

PRESENT CONDITION

The turboexpander compressor package was installed, commissioned, and put into normal operation in early 2006. The unit has been in normal operation since then.

CONCLUSIONS

The end user and EPC for a large LNG facility in the Middle East had specific requirements for the design, manufacturing, and FLFST of a turboexpander compressor package with an active magnetic bearing system. The turboexpander vendor worked closely and diligently with them and incorporated all the special requirements that were consistent with the turboexpander compressor design. The FLFST was carried out in a loop where the residue gas compressor for the same project supplied the required boost for this package. The FLFST was conducted successfully and with relatively few problems. All the specified FLFST criteria were fulfilled satisfactorily. The turboexpander compressor package is in normal operation at the present time.

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