VIBRATION ANALYSIS OF THE SYNCHRONOUS MOTOR OF A PROPANE COMPRESSOR

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Abstract: *This paper aims at describing the Analysis of a synchronous electric motor which presented high vibration levels (shaft displacement and bearing housing vibration) during the commissioning process, as well as propose the best practices for the solution of vibration problems in similar situations. This motor belongs to the propane centrifugal compressor installed at a Gas Compression Station. The methodology used in this study conducted an investigation of the problems presented in the motor through the execution of many types of tests and the analysis of the results. The main evaluations were performed, such as the vibration analysis and the rotor dynamic analysis. The electric motor was shipped back to the manufacturer's shop, where the manufacturer made certain modifications to the motor structure so as to improve the structure stiffness, such as the improvement of the support and the increase of the thickness of the structural plates. In addition to that, the dynamic balancing of the rotating set was checked. Finally, the excitation at a critical speed close to the rated speed was found after Rotor Dynamics Analysis was performed again, because of the increase in bearing clearances. The bearing shells were replaced so as to increase the separation margin between these frequencies. In order to verify the final condition of the motor, the manufacturer repeated the standard tests* - *FAT (Factory Acceptance Tests)* - *according to internal procedure and international standards. As a result of this work, it was possible to conclude that there was a significant increase in the vibration levels due to unbalance conditions. It was also possible to conclude that there are close relationships between high vibration levels and unbalance conditions, as well as between high vibration levels and the stiffness of the system and its support. Certain points of attention related to the manufacturing process of the motor compressor are described at the end of this paper, based on the results of this study.*

Keywords*: motor, vibrations, unbalance, dynamics, analysis*

1. INTRODUCTION

A centrifugal compressor is a component of a machinery train which includes an electrical motor and a transmission gear. These machinery trains are largely used in petroleum and gas industry processes. During the construction and commissioning process of these machinery trains, many kinds of tests and analyses are performed so as to guarantee the reliability, availability, and serviceability of the process plant. One of these is the vibration analysis.

One of the major difficulties in performing vibration diagnosis is to establish and isolate all the variables that work together during the machinery train tests, such as mechanical and/or electrical problems. These problems may result in loss of time and, as a consequence, in a delay in the construction and startup schedule.

During the startup of the propane centrifugal compressor installed at a Gas Compressor Station, the driver presented high values of shaft displacement and bearing housing vibration as shown in Tab.1, despite the fact that the values expected were lower than 50 μm (p-p) and 1.8 mm/s (rms).

Table 1. Vibration levels (unfiltered) of the motor during commissioning process

Coupled Running	Shaft Vibration (μm p-p)				Bearing Housing Vibration (mm/s rms)					
	DE(1)		NDE(2)		DE(1)			NDE(2)		
	X	Y	X	Y	H	V	A	H	V	A
Unfiltered	66.5	46.1	78.9	52.9	3.50	3.27	3.74	2.83	4.59	1.70
Filtered	66.5	46.1	78.9	52.9	3.50	3.27	3.74	2.83	4.59	1.70

(1): Drive End Side
(2): No Drive End Side

The services listed below were performed in the field in order to solve this vibration problem:

- Replacement of the damaged bearing shells;
- New alignment between motor, gear and compressor;
- Balancing of the motor;

Vibration levels remained high and still increasing after these services as shown in Tab.2.

Table 2. Vibration levels (unfiltered) of motor after services

Condition		Shaft Vibration (µm p-p)				Bearing Housing Vibration (mm/s rms)					
		DE(1)		NDE(2)		DE(1)			NDE(2)		
	Date	X	Y	X	Y	H	V	A	H	V	A
Coupled	02/12/2010	87.0	69.1	88.1	110.5	3.7	4.4	5.7	2.1	3.8	2.8
Coupled	02/13/2010	89.0	69.0	94.0	121.9	3.9	4.3	6.7	2.2	3.6	2.7
Coupled	02/14/2010	88.2	70.7	90.3	114.6	3.7	4.5	5.8	2.1	3.8	2.7
Uncoupled	02/19/2010	63.6	52.0	61.2	65.0	2.6	2.6	3.3	1.7	3.6	2.2

(1): Drive End Side
(2): No Drive End Side

After exhausting all the possible solutions at the site, the manufacturer shipped the electric motor back to the shop so that it could investigate the root causes of the problem and make the necessary repairs and corrections. In addition to the high vibration levels during start up, synchronization problems were also verified between the motor and the power supply. This problem was supposed to be investigated at the factory as well.

Prior to the services, tests were performed as references, including vibration level measurements. Table 3 shows the results of the tests performed.

Table 3. Vibration levels of the motor at the manufacturer's facilities prior to the services

	Shaft Vibration (µm p-p)				Bearing Housing Vibration (mm/s rms)					
	DE(1)		NDE(2)		DE(1)			NDE(2)		
	X	Y	X	Y	H	V	A	H	V	A
With coupling and balancing weight	35.4	25,8	50.8	26.5	1.8	3.2	3.5	1.8	2.4	3.2
Without coupling and balancing weight	54.9	54.8	71.9	67.4	1.9	3.9	3.3	0.8	3.2	1.8
Acceptance Criteria	50 µm p-p				1.8 mm/s (rms)					

(1): Drive End Side
(2): No Drive End Side

At the shop, the manufacturer made certain modifications to the motor structure, such as the improvement of the support and increase of the thickness of the structural plates, so as to improve the structure stiffness. In addition to that, the dynamic balancing of the rotating set was checked. Some apparently insignificant damages to the coupling substantially contributed to the unbalance of this component. Additionally, the bearing shells were replaced in order to increase the separation margin between the rated speed and the first critical speed found in the Rotor Dynamics Analysis. In order to verify the final condition of the motor, the manufacturer repeated the standard tests - FAT (Factory Acceptance Tests) - according to internal procedures and international standards.

2. MODIFCATIONS AND SERVICES

The major services and tests performed at the manufacturer's facilities with the intention of improving shaft and bearing housing vibration levels for the motor are presented below due to the fact that they are considered to have more impact on the results.

2.1. Structural modifications of the motor

Structural modifications of the motor were made in order to increase frame stiffness and consequently improve shaft and bearing housing vibration levels, due to the excitation of a structural natural frequency (structural resonance). Among the main services, the following are pointed out:

- Two additional feet, which were not included in the original motor project, were installed below the external fan.
- End covers were reinforced in order to improve the system's stiffness. Their thickness was increased at the point where the bearing housings are supported. After this modification, the end cover was thicker than the frame plates, making the frame stronger.
- The exciter base feet were machined again in order to correct the lack of parallelism resulting from machining mistakes. A new foot was manufactured to support the exciter. This problem was interfering in the centralization of the rotor and the stator within API Standard 546 3rd edition limits.

- Guide pins were installed on the end covers, the frame and the exciter, in order to avoid misalignment during assembly and disassembly services.
- Lock washers and double nuts were added to the fixation bolts so as to avoid their loosening. This modification was made because several bolts were loosening or breaking as a result of the high vibration levels during running of the machine in the field.

While these services were being performed at the manufacturer's facilities, the constructor carried out Model Analysis of the system's motor and base. This analysis showed an excitation of natural frequencies of the base close to running speed. The central base parts were grouted to increase base stiffness and ensure the separation margin of these frequencies - base natural frequency and motor running frequency.

2.2. Repairs of the exciter rotor and the diode wheel

The exciter presented a short circuit during the Zero Power Factor Test. The manufacturer informed that the short circuit might have been caused by small mechanical damages as a result of the assembly and disassembly of this component. This component was rewound and individually balanced.

It was verified that an epoxy putty was used as a balancing weight in the exciter rotor and diode wheel instead of a steel weight, in disagreement with API Standard 546 3rd edition. This epoxy putty was removed and balancing of the component was performed. In addition to the disagreement with API Standard 546 3rd edition, there was a chance of putty detachment during motor running due to thermal expansion differences between materials.

2.3. Runout runway

During the tests, shaft runout above the limits allowed by API Standard 546 3rd edition was detected. The shaft probe runway was burnished in order to reduce runout values (electrical and mechanical). The runout values remained above API Standard 546 3rd edition, mainly due to the distribution of metallurgical material (electrical runout). The manufacturer reported that it did not have resources for its reduction. The runout values before and after services are presented in Tab. 4.

Table 4. Motor runout values before and after services

Runout	Measurements of the Motor				Measurements of the rotor			
					Mechanic		Mechanic + Electric	
	DE-X	DE-Y	NDE-X	NDE-Y	DE	NDE	DE	NDE
Before services (µm p-p)	15.8	14.2	19.4	19.1	2	2	6	8
After services (µm p-p)	13.5	14.9	19.6	23.1	3	3	5	10
Acceptance Criteria	12,5 µm							

2.4. Coupling damages

During assembly of the coupling for tests, damages to the bolt threads of the sleeve and ring cover were detected, as shown in Fig. 5, which caused the increase of vibration levels.

Figure 5. Coupling damages

Since the coupling manufacturer recommended against using these damaged components in their current conditions or repairing them, new ones were supplied to replace the damaged items. The replacement was supervised by the manufacturer's representative to ensure no new damages to the coupling.

2.5. Dimensional deviation of the shaft

The shaft was machined again in order to correct marks in the bearing seats which caused a modification in the shaft dimensions and tolerances. Consequently, the clearances between the shaft and the bearings were increased. The other parts of the shaft were machined again and the components (fans, exciter and diode wheel) were replaced in order to enable the assembly on the shaft. Despite the new machining, the manufacturer reported that it was not possible to increase shaft flexibility because the decrease of the dimensions was insignificant, as shown in Tab. 6.

Table 6. Fitting dimensions of the shaft and components

Description	Shaft Dimensions			Internal Diameters		
	Project		Actual (mm)	Project	Actual	
	Nominal (mm)	With Tolerance (mm)		With Tolerance (mm)	With Tolerance (mm)	Fitting (mm)
Front Bearing	180	179.652 179.627	179.628	180.000 179.040	180.035	0.407 (clearance)
Rear Bearing	225	224.561 224.532	224.550	224.000 224.048	225.040	0.490 (clearance)
External Fan	230 h6 – H8	229.079 229.050	229.070	229.072 229.000	229.060	0.010 (interference)
Exciter Fan	180 h6 – H7	179.068 179.043	179.060	179.040 179.000	179.020	0.040 (interference)
Exciter rotor	172 js6 - E7	170.568 170.043	170.558	170.625 170.585	170.600	0.042 (clearance)
Diode wheel	160 h6 – h7	159.000 158.975	158.996	159.000 159.040	159.018	0.022 (clearance)

2.6. Critical speed

The calculations and Rotor Dynamics Analysis of the motor were performed again due to increased clearance of the bearings caused by burnishing of the shaft in the bearing seats. During this analysis, excitation of the first critical speed (1784 rpm) close to rated speed (1800 rpm) was detected, as shown in Fig. 7, representing a clearance of only 1%, while the API Standard 546 3rd edition requires a minimum separation margin of 15%.

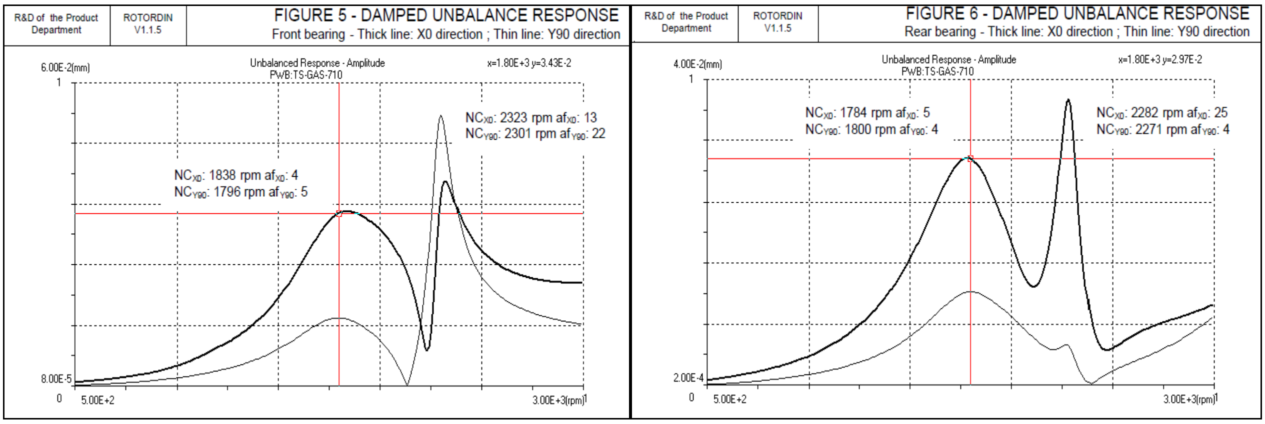

Figure 7. Rotor Dynamics Analysis – front and rear bearings with increased clearances

The manufacturer performed an internal Unbalance Response Test to confirm the Rotor Dynamics Model. However, this test was not considered valid because the unbalance weights were added at points different from those indicated in the Rotor Dynamics Analysis. The balanced condition of the motor prior to testing was not duly proven either. In addition to that, API Standard 546 3rd edition establishes maximum limits for the amplitudes of shaft displacements for the Unbalance Response Test (maximum 98 mm). These limits have been exceeded during the test, as shown in Fig. 8.

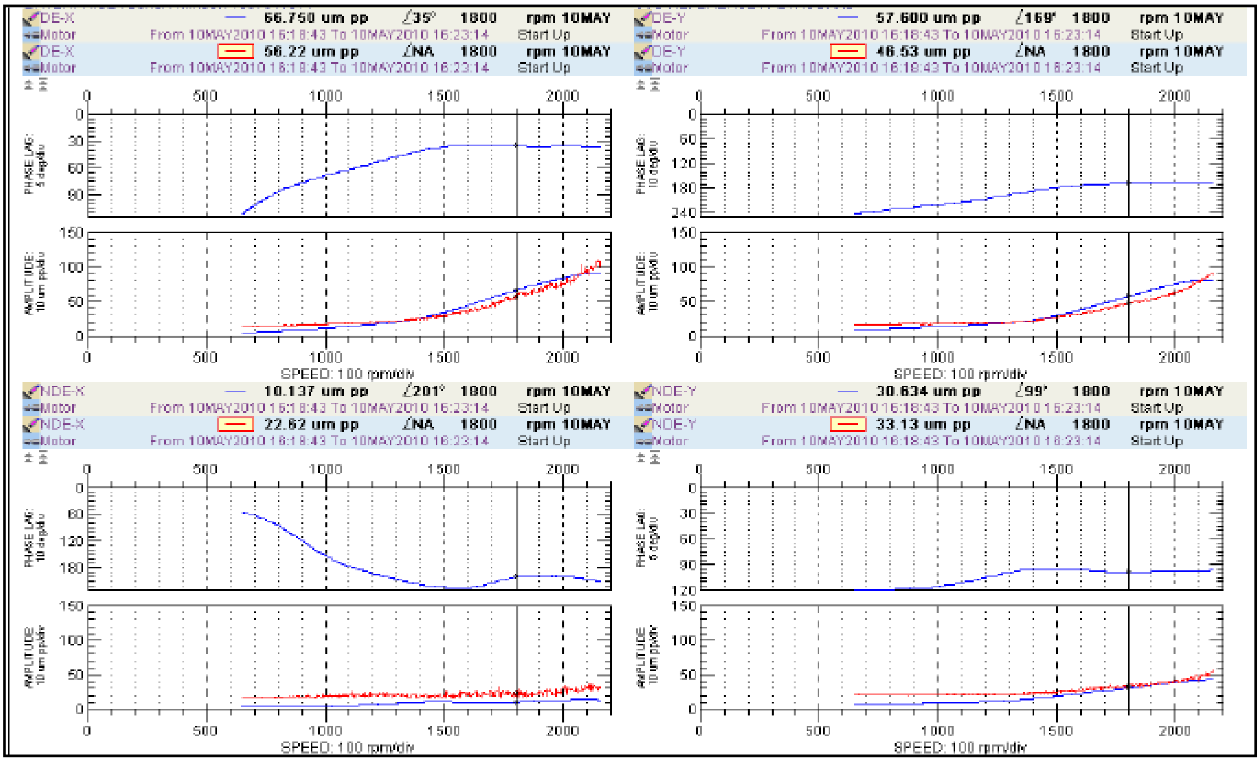

Figure 8. Bode Plots of the internal Unbalance Response Test performed by the manufacturer

The solutions proposed by the manufacturer in order to increase the separation margin between the first critical speed and the rated speed are described below:

- Replacement of the cylindrical bearing shells for new ones with a special dimension (reduced internal diameter) which would reduce clearances between the shaft and the bearings and consequently increase the stiffness of the system.
- Modification of the bearing design and replacement of the bearings for new ones with four lobes. The use of four lobe bearings implies changes in the parameters of the lubrication system, such as:
  - ✓ Oil flows due to the higher temperature levels of this kind of bearing, resulting in the need to resize and replace the calibrated orifices.
  - ✓ Oil pressures due to the different design of these bearings.
- Replacement with a new shaft to restore the dimensions of the original project.

It was decided that the replacement of the bearings for new cylindrical ones with reduced dimensions (clearances) would be carried out. The new Rotor Dynamics Analysis presented very close results for both types of bearings (cylindrical ones with reduced dimensions and four lobes). Additionally, changes would be avoided in the lubrication system required to adopt the four lobe bearing. After bearing replacement, a new Unbalance Response Test would be performed.

2.7. Bearing shell replacement

The bearings shells were replaced. The dimensions of the new bearings and bearing seats were taken for the calculation of current clearances. The clearances presented were within the specified tolerances and were far less than the clearances measured before the bearing replacement, as shown in Tab. 9.

Table 9. Bearing Clearances

	Actual Dimensions			Previous	Project Dimensions		
Description	Shaft	Bearing	Clearance	Clearance	Shaft	Bearing	Maximum Clearance
Drive end (mm)	179.633	179.840	0.207	0.480	179.652 179.627	179.870 179.822	[3] 0.243
No drive end (mm)	224.550	224.823	0.273	0.480	224.561 224.532	224.855 224.795	[3] 0.323

[3]: Clearance values used in the Rotor Dynamics Analysis.

3. TESTS AND BALANCING

3.1. Mechanical Running Tests

The Mechanical Running Test performed after replacement of the bearing shells did not present satisfactory results. Shaft vibration was above the acceptable limits established in API Standard 546 3rd edition. The test results are presented in Tab. 10.

The manufacturer was unable to provide technical clarifications as to the phenomena observed during the Mechanical Running Test:

- The discrepancy between the Rotor Dynamics Analysis and the results obtained from the test;
- The shaft vibration on the no drive end - NDE increased significantly after the replacement of the bearing shells, clearly diverging from the values predicted in the Rotor Dynamics Analysis. Moreover, this incident was presented only on the no drive end - NDE.
- A large variation in vibration levels between the cold condition (right after start up) and the hot condition (after four hours running).

Table 10. Mechanical Running Test results of the motor before and after replacement of the bearing shells

	Test before replacement of the bearing shells	**Test after replacement of the bearing shells**		**Acceptance Criteria**
	Hot Condition	**Cold Condition**	**Hot Condition**	
Shaft Vibration (µm p-p)				
Drive End - DE -X	19.47	9.36	14.34	50 µm (p-p)
Drive End - DE -Y	20.88	8.86	14.06	
No Drive End - DE -X	61.80	34.86	68.00	
No Drive End - DE -Y	51.50	33.72	60.08	
Bearing Housing Vibration (mm/s rms)				
Drive End - DE -H	0.374	0.392	0.505	1.8 mm/s (rms)
Drive End - DE -V	0.721	0.462	0.650	
Drive End - DE -A	0.818	0.564	0.908	
No Drive End - DE -H	0.470	0.414	0.607	
No Drive End - DE -V	0.486	0.408	0.553	
No Drive End - DE -A	0.427	0.570	0.656	

After analyzing the vibration spectrum of the motor shown in Fig.11, it was found that the amplitudes in the running frequency were very pronounced, suggesting an unbalance condition of the rotating set, according to Bently (2002). Based on these data, the manufacturer was requested to perform further verifications of the balance condition of the motor.

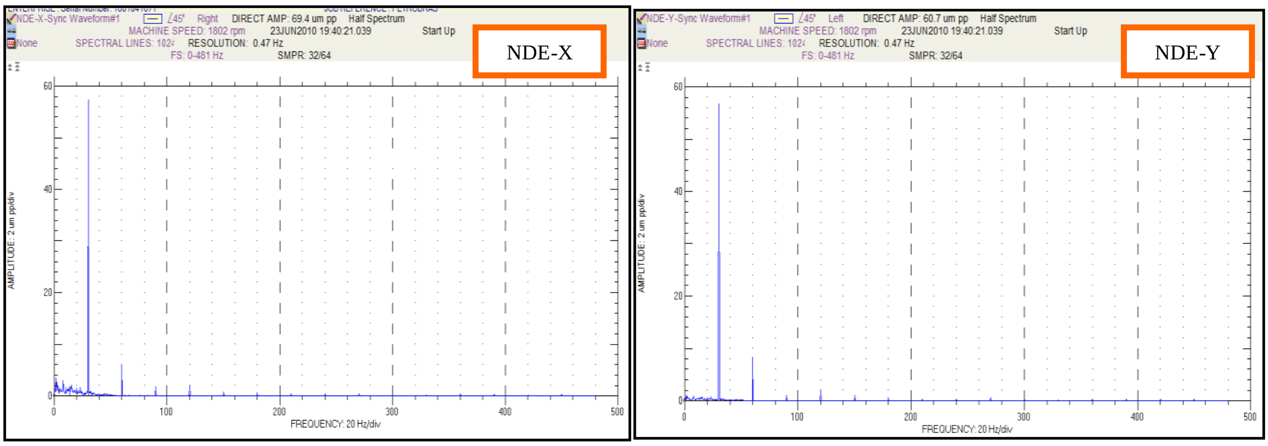

Figure 11.Spectrums of the shaft vibration from the Mechanical Running Test (no drive end - NDE)

3.2. Balancing

The first balancing step was performed with the rotor mounted in the electric motor, in only one plane (exciter fan), based on readings of bearing proximity sensors using the graphic method as proposed by the manufacturer. However, the result of this attempt to balance the motor was not satisfactory: there was a slight reduction in shaft vibration levels, but also a significant increase in bearing housing vibration levels. This process was inefficient due to the fact that the motor has many balancing planes, such as the diode wheel, the fans and the exciter. During this step, the need to perform balancing in more planes and in a more precise manner was evident.

The second balancing step was performed with the rotor mounted in the electric motor, in two planes (Plane 1 - front internal fan and Plane 2 – exciter rotor), based on readings of bearing proximity sensors using software.

Based on the results obtained in the balancing steps, it was observed that the increase in bearing housing vibration levels was more pronounced after the addition of balancing weights on the front internal fan. One by one, the correction weights placed on this plane were removed. As no satisfactory results were obtained, the weight of the exciter was also removed with the intention of restoring the initial balance condition of the motor and therefore achieving the same vibration levels as before balancing. Unexpectedly, the vibration levels were not the same as before balancing. The manufacturer was unable to clarify why it was not possible to restore the initial condition of the motor.

Based on the balancing results, it was found that as shaft vibration was reduced, there was an increase in bearing housing vibration. The manufacturer was unable to clarify the reason for this behavior of the motor. The balancing weight was placed back on the exciter rotor, because, in this condition, a better equilibrium between shaft and housing bearing vibrations was achieved. The motor was left in this new balancing condition. Table 12 shows the vibration levels (overall and filtered 1x) obtained in the main steps outlined above and the final results after balancing.

Table 12. Vibration levels during the balancing of the motor after replacing bearing shells

	Before Balancing	**One Plane Balancing**	**Two Planes Balancing**	**After Removing Weights**	**Weights at the Exciter**
PLANE 1	**NO**	**NO**	**155g - 120° 120g - 105°**	**NO**	**NO**
PLANE 2	**NO**	**135g - 336°**	**115g - 300°**	**NO**	**115g - 300°**
Shaft Vibration (µm p-p)					
Drive End - DE -X	14.25 (9.4 - 34°)	17.90 (11.6 - 94°)	13.81 (8.9 - 257°)	15.78 (11.52 -358°)	14.03 (8.27 - 348°)
Drive End - DE -Y	14.06 (8.7 - 123°)	23.19 (17.2 - 184°)	26.29 (22.37 - 308°)	20.27 (14.22 - 45°)	16.26 (9.94 - 20°)
No Drive End - DE -X	69.90 (67.6 - 357°)	42.97 (35.7 - 47°)	24.49 (16.82 - 55°)	46.19 (38.24 - 30°)	36.73 (27.31 - 43°)
No Drive End - DE -Y	61.5 (56.7 - 93°)	48.88 (43.7 - 135°)	23.74 (15.75 - 149°)	49.79 (34.48 -114°)	42.41 (26.12 -121°)
Bearing Housing Vibration (mm/s rms)					
Drive End - DE -H	0.505	1.05	2.81	1.51	1.04
Drive End - DE -V	0.650	1.17	2.14	1.53	1.29
Drive End - DE -A	0.908	1.62	2.67	1.76	1.46
No Drive End - DE -H	0.607	0.69	1.06	0.60	0.65
No Drive End - DE -V	0.553	0.65	0.65	0.66	0.71
No Drive End - DE -A	0.656	1.22	1.26	1.22	1.15

Based on the test results of the motor, the vibration levels were compared under different balancing conditions. Based on these results, it was concluded that the vibrations of the motor are greatly affected by its balancing condition. Moreover, there is no repeatability and maintenance of the balancing condition of the rotating set. According to the manufacturer, this difficulty was present because of the existence of electric cables in the exciter rotor. Whenever these wires are disconnected and reconnected, they lose their original configuration and the mass distribution of this component is modified, consequently affecting the balancing condition of the rotating set. To eliminate this problem, modifications to the motor design would be required, separating the fan from the rotating set and placing it in the heat exchanger. These modifications would reduce the overall length of the shaft and would permit balancing of the completely assembled rotating set without requiring disassembly for subsequent assembly in the motor.

During the Mechanical Running Test, after balancing, shaft vibration remained virtually the same. However, bearing housing vibration showed variations in relation to results obtained previously, but remained just within the limits allowed by API Standard 546 3rd edition.

3.3. Unbalance Response Test

The Unbalance Response Test was performed after replacing the bearing shells and balancing of the motor, according to API Standard 546 3rd edition and the manufacturer's Rotor Dynamics Analysis of the motor. The same balancing plans were used for the addition of the masses for the Unbalance Response Test, as shown in Fig. 13. The motor was accelerated to 120% of its operating speed (2162 rpm) in each step of the test, as recommended by the manufacturer, and then coasted down for data collection.

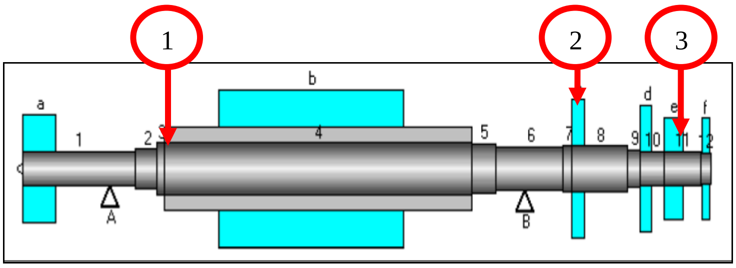

Figure 13. Planes used for Unbalance Response Test

The weights were moved to new positions during the test so as to determine the sensitivity of the rotor response to unbalance weight placement, as recommended by API Standard 546 3rd edition. The weight placed on Plane 1 was increased to the maximum value allowed by API Standard 546 3rd edition in order to check if the amount of weight was sufficient to cause excitation of the rotor. The weights at the other planes were maintained as directed by the manufacturer's engineering team. Table 14 shows the settings used during the various testing steps.

Table 14. Configuration for each step of the Unbalance Response Test

Test Step	Plane 1		Plane 2		Plane 3	
	Mass	Angle	Mass	Angle	Mass	Angle
1	160 g	90 º	110 g	84 º	-	-
2	160 g	90 º	110 g	84 º	23.2 g	90 º
3	160 g	90 º	110 g	84 º	23.2 g	270 º
4	160 g	0 º	110 g	336 º	23.2 g	180 º
5	160 g	0 º	110 g	336 º	23.2 g	0 º
6	320 g	0 º	110 g	336 º	24.2 g	0 º

During the test, characteristic peaks that suggest critical frequency excitation were not observed at any speed within the operating speed range (up to 2162 rpm or 120%), in other words, within the limits of the separation margin. The shaft vibration remained within the limits of API Standard 546 3rd edition for the Unbalance Response Test (maximum 98 µm). The shaft displacement amplitudes at the operating speed were higher than those described in the manufacturer's Rotor Dynamics Analysis. However, they were below the maximum permitted by API Standard 546 3rd edition. Table 15 shows shaft vibration (overall values and 1x Runout compensated) and bearing housing vibrations obtained in step 6. Figures 16 and 17 show Bode Plots of the last test step (STEP 6).

Table 15. Shaft Vibration of Unbalance Response Test (Step 6) after replacing bearing shells

Rotating Speed **rpm**	DE-X **(µm p-p)**	DE-Y **(µm p-p)**	NDE-X **(µm p-p)**	NDE-Y **(µm p-p)**
1800	19,2 (10,3-131°)	29,8 (21,8- 235°)	24,2 (5,3-116°)	32,5 (19,3-170°)
2162	43,4 (33,5-179°)	66,5 (55,8- 265°)	45,2 (26,1-171°)	49,8 (29,5-234°)

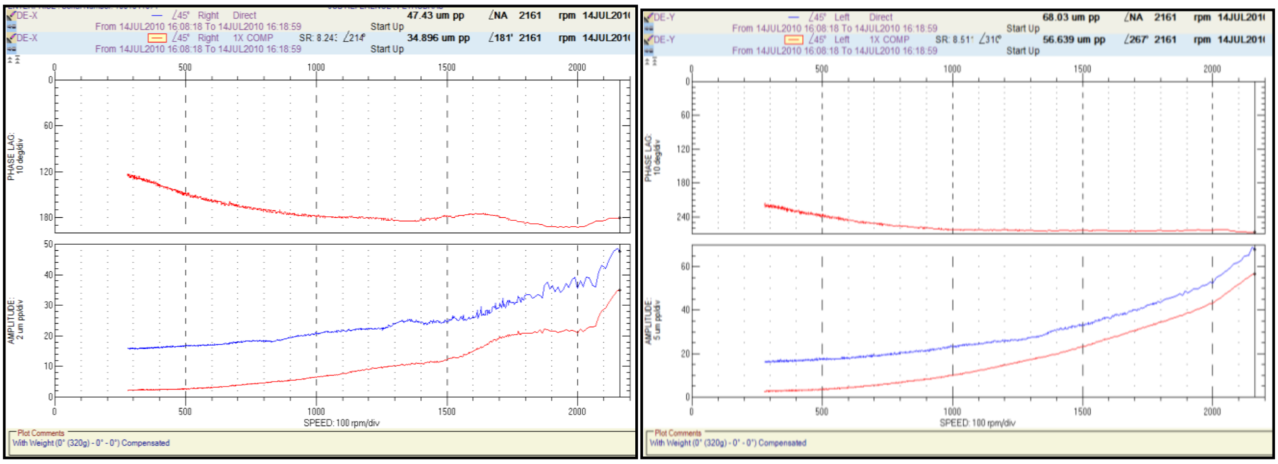

Figure 16. Bode Plots (Runout compensated) of Unbalance Response Test after bearing shell replacement-DE

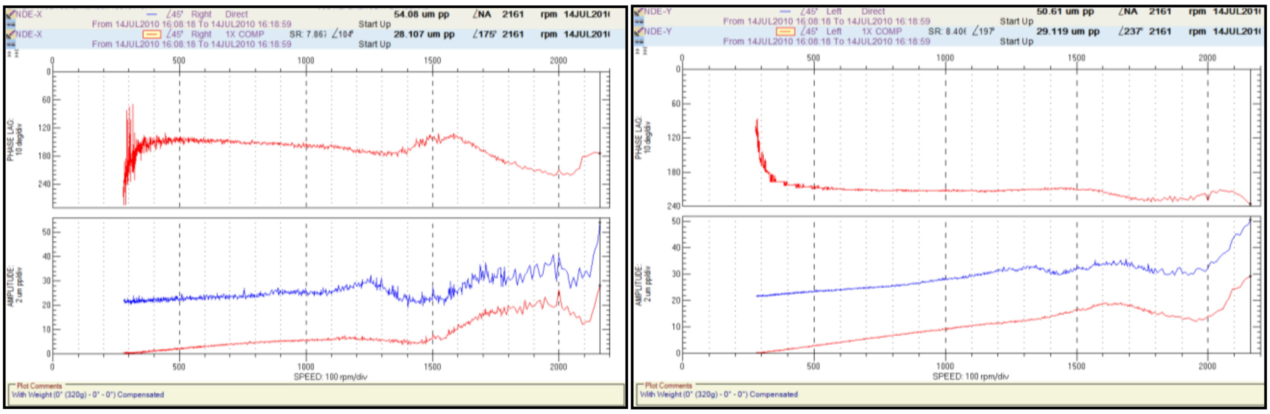

Figure 17. Bode Plots (Runout compensated) of Unbalance Response Test after bearing shell replacement -NDE

After the Unbalance Response Test, the motor was run again. The shaft vibration levels during operation were slightly higher than those of the previous run, but remained within the range allowed by API Standard 546 3rd edition. Table 18 shows the vibration levels during both tests (previous and present) for comparison purposes.

Table 18. Vibration levels before and after the Unbalance Response Test

	Results before Unbalance Response Test	**Results After Unbalance Response Test**	**Acceptance Criteria**
Shaft Vibration (μm p-p)			
Drive End - DE -X	9.62 (3.24 - 24°)	12.87 (6.89 - 62°)	50 μm (p-p)
Drive End - DE -Y	13.08 (6.1 – 338°)	11.61 (3.19 - 252°)	
No Drive End - DE -X	35.43 (26.37 - 54°)	37.08 (29.39 - 48°)	
No Drive End - DE -Y	40.50 (34.76 - 145°)	44.17 (38.03 - 139°)	
Bearing Housing Vibration (mm/s rms)			
Drive End - DE -H	0.77 (0.52- 138°)	0.73 (0.56 - 115°)	1,8 mm/s (rms)
Drive End - DE -V	1.25 (1.15 - 94°)	1.02 (0.96 - 84°)	
Drive End - DE -A	1.72 (1.57 - 291°)	1.51 (1.38 - 281°)	
No Drive End - DE -H	0.40 (0.25 - 297°)	0.38 (0.18 - 291°)	
No Drive End - DE -V	0.76 (0.68 – 266°)	0.79 (0.71 - 264°)	
No Drive End - DE -A	1.40 (1.28 - 105°)	1.43 (1.30 - 99°)	

4. TIMING AND ACCELERATION PROBLEMS

The manufacturer submitted data from on-site electrical tests on the motor and compressor and reported that the timing issue presented during the motor start up was caused by the motor's long acceleration time. The manufacturer also reported that the delay in motor acceleration may have been caused by variations in the unit's process conditions. However, process data collected concomitantly to the electrical data was not presented in order to support this latter affirmation.

The manufacturer reported that, during the visual inspection performed after disassembly of the motor at the factory, damages were observed in some parts of the rotor, such as the displacement of bars. The manufacturer reported that the excessive time of acceleration in the field during the commissioning process caused the thermal expansion and consequently the excessive dilation. These damages were not considered severe enough to disable the rotor, but it was necessary to correct them. The manufacturer replaced the bandages and damaged bars which were displaced.

The constructor presented a computer simulation based study of the unit process which explained that the actual motor torque during start up was greater than the theoretical torque informed by the compressor manufacturer. The study compared the start up conditions of the compressor system with pressures from 4 kg/cm2 to 7 kg/cm2 (current process). The study concluded that the ideal pressure for the compressor start up process was 4 kg/cm2. This pressure reduction process for compressor start up involves the burning of propane, which also generates problems for system operation and also increases operating costs.

5. CONCLUSIONS

After the execution of all the services and tests described previously, there was a significant reduction in engine vibration levels compared to the results presented when the motor returned to the manufacturer's facilities. As shown in Tab.19, shaft vibration and bearing housing vibration are within the limits allowed by the API Standard 546 3rd edition.

Table 19. Vibration levels before and after services at the manufacturer's facilities

Condition	Shaft Vibration (μm p-p)				Bearing Housing Vibration (mm/s rms)					
	DE-X	DE-Y	NDE-X	NDE-Y	DE-H	DE-V	DE-A	NDE-H	NDE-V	NDE-A
(4): Before Services	**35,4**	**25,8**	**50,8**	**26,5**	**1,8**	**3,2**	**3,5**	**1,8**	**2,4**	**3,2**
(4): After Services	**12,9**	**11,6**	**37,1**	**44,2**	**0,7**	**1,0**	**1,5**	**0,4**	**0,8**	**1,4**
Acceptance Criteria	50 μm p-p				1,8 mm/s (rms)					

(4): Measurements with coupling.

Despite achieving vibration levels within the acceptance criteria of API Standard 546 3rd edition, there was no repeatability of the results and questions as to the behavior of the motor were not technically elucidated by the manufacturer, such as:

- The reason for the variations in vibration results without any intervention in the motor;
- The reason for the reduction in shaft vibration levels being accompanied by an increase in bearing housing vibration levels;
- The reason why there is no repeatability of the balance condition;
- The reason for the wide variation in vibration levels between cold and hot conditions;
- The reason why the vibration of the motor increased after bearing shell replacements, diverging from the Rotor Dynamics Analysis, in which a reduction in vibration levels was expected after the replacement of the bearing shells. Furthermore, this phenomenon was observed only on the no drive end side (uncoupled side);

Due to these technical uncertainties, the level of reliability of the electric motor was not satisfactory. With this, the manufacturer proposed to manufacture a new motor for future replacement of the current one. Conditions for accepting the manufacturer's proposal were presented, including, for instance, the design of the new motor incorporating improvements to avoid the problems presented by the current motor, such as:

- Problems associated with vibration;
- Problems associated with acceleration.
- Another condition presented was that the inspection plan and testing of the new motor should be rethought.

The current motor was considered acceptable for operation at the Gas Compression Station, while the new motor is being manufactured. The constructor will install the new motor when it is delivered to the site.

Factors that may be considered as points of attention for the manufacturing of motors in the future are described below:

- Dimensional examination of the shaft and rotating set components and verification of the clearances / interference fitting between the shaft and components mounted on the same;
- Verification of the concentricity between the front and rear bearing housings. The concentricity of the bearing housings is supposedly achieved only with the dimensional control of the cover lids and the bearing housing guides. The manufacturer, however, did not evidence the existence of procedures for verifying and ensuring the concentricity after assembly;
- The process of balancing individual components and rotor. The balancing process is not being performed in a gradual manner. Moreover, the rotor is not balanced with all the components assembled;
- Final balancing process of the motor on the bench. The standard procedure adopted by the manufacturer uses a measurement based on accelerometers, which can mask the results of the measurements in cases of hydrodynamic bearings because of the oil film. Moreover, the design of the motor blocks access to the rear rotor plane (internal fan), leaving doubts as to the residual unbalance in this rotor plane. Additionally, the existence of many balancing planes hinders the verification of residual unbalance in each plane;
- The stiffness of the frame. It was verified that, during the various testing steps, after placement or removal of air ducts or the heat exchanger, vibration levels of the motor changed;
- Improvements of runout measurements, heat treatments and material selections in order decrease runout values.

6. REFERENCES

API STANDARD 546 2008, "Brushless Synchronous Machines - 500 kVA and Larger" "American Petroleum Institute", 3rd edition, United States of American.

Bently, D.E. and Hatch, C.T., 2002, "Fundamentals of Rotating Machinery Diagnostics", Bently Pressurized Bearing Press, Minden, Nev, United States of American.

7. RESPONSIBILITY NOTICE

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