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# VIBRATIONS IN HIGH PRESSURE CENTRIFUGAL COMPRESSORS OF OFFSHORE PLATFORMS

By

Roberto Firmento de Noronha (PhD)
Petróleo Brasileiro S.A. (PETROBRAS)
and Universidade Federal Fluminense
Rua Ministro Ramos Monteiro 37A/1304
Rio de Janeiro/RJ, Brazil, 22430-100
Phone 55 21 3876 7439
roberto.noronha@petrobras.com.br

## ABSTRACT

During the startup of one of the compression trains of an offshore platform, in August/2006, excessively high vibrations were observed at the high pressure compressor. The same problem was also verified one week later, during the preliminary tests of the String Test for the compressors of other two offshore platforms, which were supplied by the same vendor. The vibration records showed that the high vibration was due to the synchronous vibration component (1X vibration) and that it increased with the gas temperature.

The Vendor's previous experience with this problem, which was designated as *Thermal Induced Unbalance Vibration*, indicated that this could happen when there were mounted components secured to the shaft through two interference fits. This was true for the balance piston of the high pressure compressor of P-50. The design of this component, with a cylindrical shape, was not usual, but was employed because of the small diameter of the component. The same design was repeated on the HP compressors of P-53 and P-54.

After modifying the balance piston of the HP compressor at the test bed, a high level of vibration has been again observed in the following pre-test. The analysis of the vibration records showed that now the problem was more related to pressurization, rather than to a thermal effect. Rotor dynamical simulations confirmed that the high vibration was now caused by the design of the balance piston seal.

The objective of this work is to present a detailed description of the causes of these vibration problems and discuss the ways of avoiding them for future equipment.

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Roberto F. de Noronha, PhD
PETROBRAS and UFF
roberto.noronha@petrobras.com.br

# Objective

- To present a case where the use of the API 617 standard did not prevent the occurrence of high vibrations in centrifugal compressors.

# Compressors Description

- The FPSOs P-50 (in operation since April/2006), P-53 (planned for 2008) and P-54 (operating since Nov/2007) have in common oil production capacity of 180,000 bpd and also the specifications of its centrifugal compressors:
- Three compression trains driven by variable speed electric motors. Capacity of 70,000,000 Ncfpd per set, $P_s$ at 130 psia and $P_d$ at 2800 psia.
- The vendor and the configuration of the compressors ended up being the same for all three platforms, with the LP compressor as *back-to-back* and the HP as *straight-through*.

# Description (cont.)

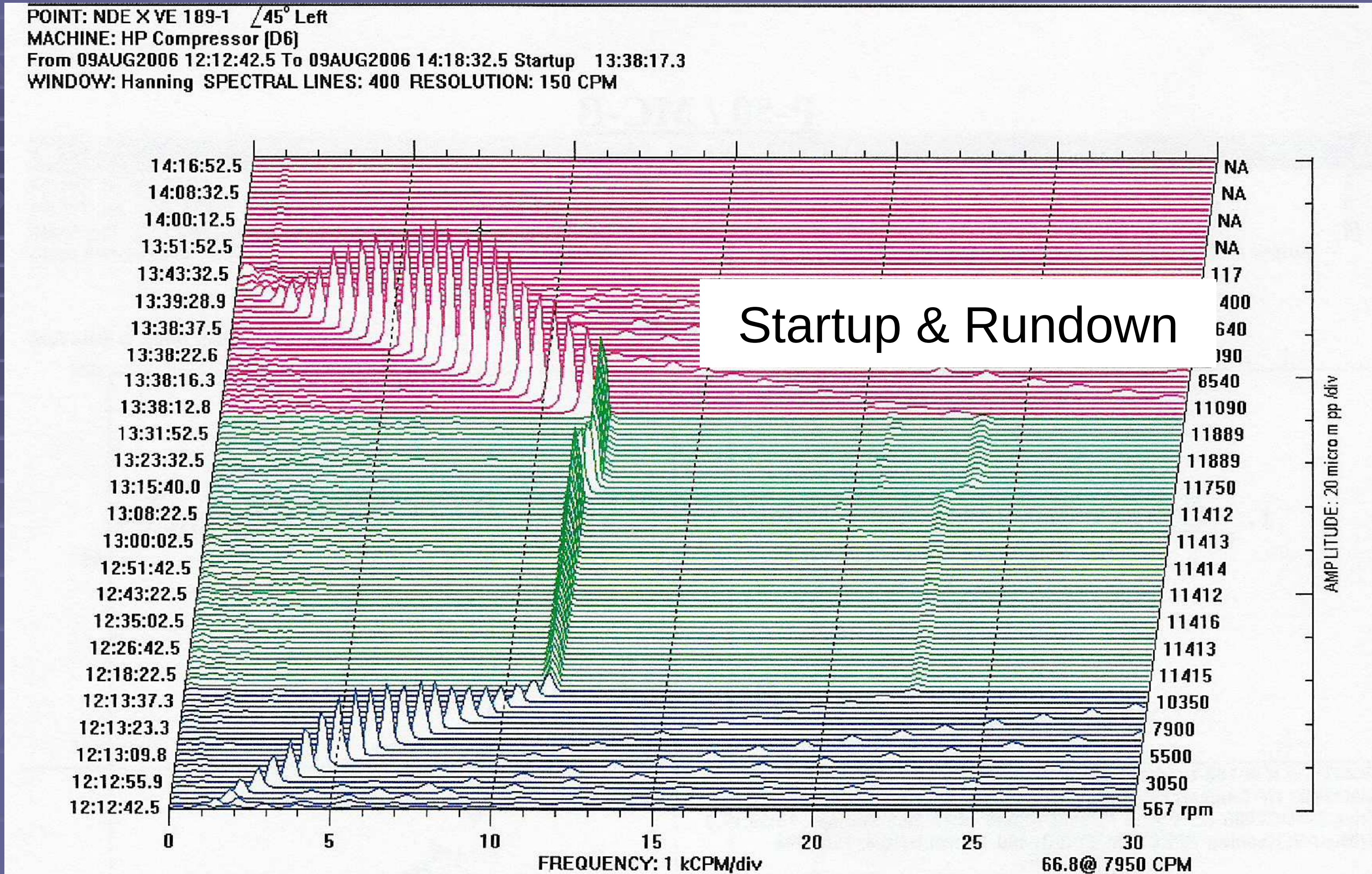
Left - Open bundle of the HP compressor of P-50, P-53 and P-54 FPSOs. Right - The barrel.

# Description (cont.)

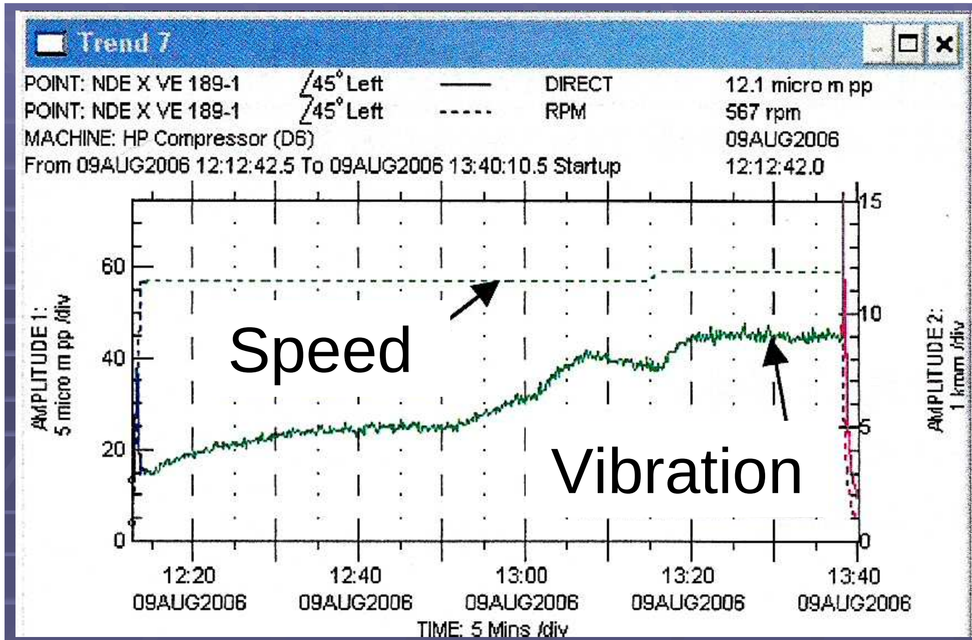
| | Some parameters | | Another manufacturer | |
|---|---|---|---|---|
| Compressor | LP | HP | LP | HP |
| Type | BtoB | StThg | StThg | BtoB |
| Bearing span / hub diam. | 9.96 | 9.89 | 9.19 | 9.52 |
| Impeller diameter (in) | 14.2 | 12.2 | 18.1 | 15.8 |
| *MCS* (rpm) | 14893 | 14893 | 11490 | 11490 |
| Max. pol. efficiency (%) | 82,9 - 81,14 | 76,39 | 80,4 | 76,5 - 62,9 |
| Level I log dec | 0.64 | 0.37 | 0.034 | -0.123 |
| Level II log dec | 0.57 | 0.38 | 0.101 | 0.641 |

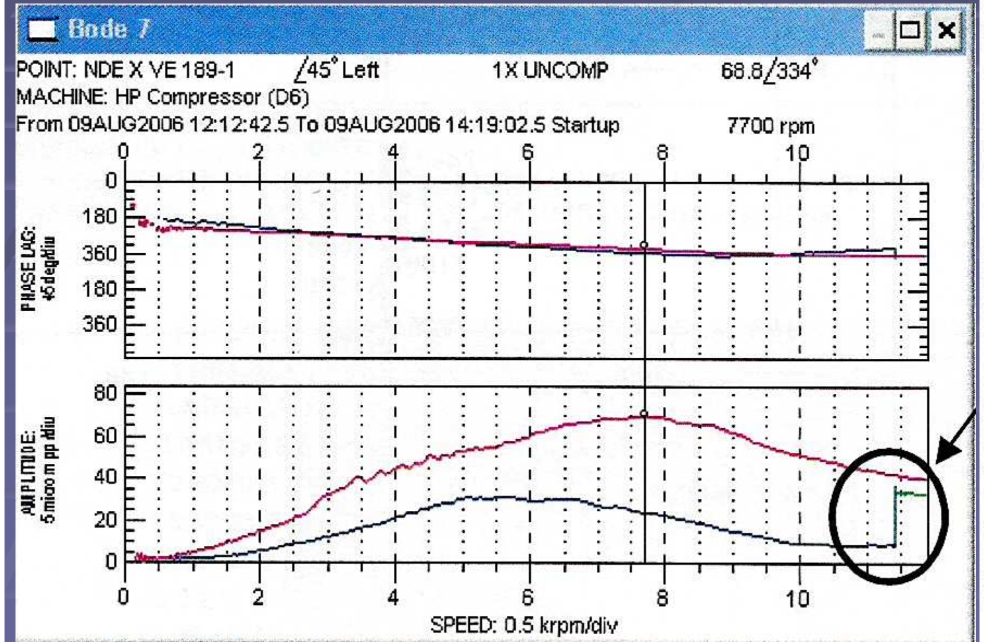
# P-50 Compressors

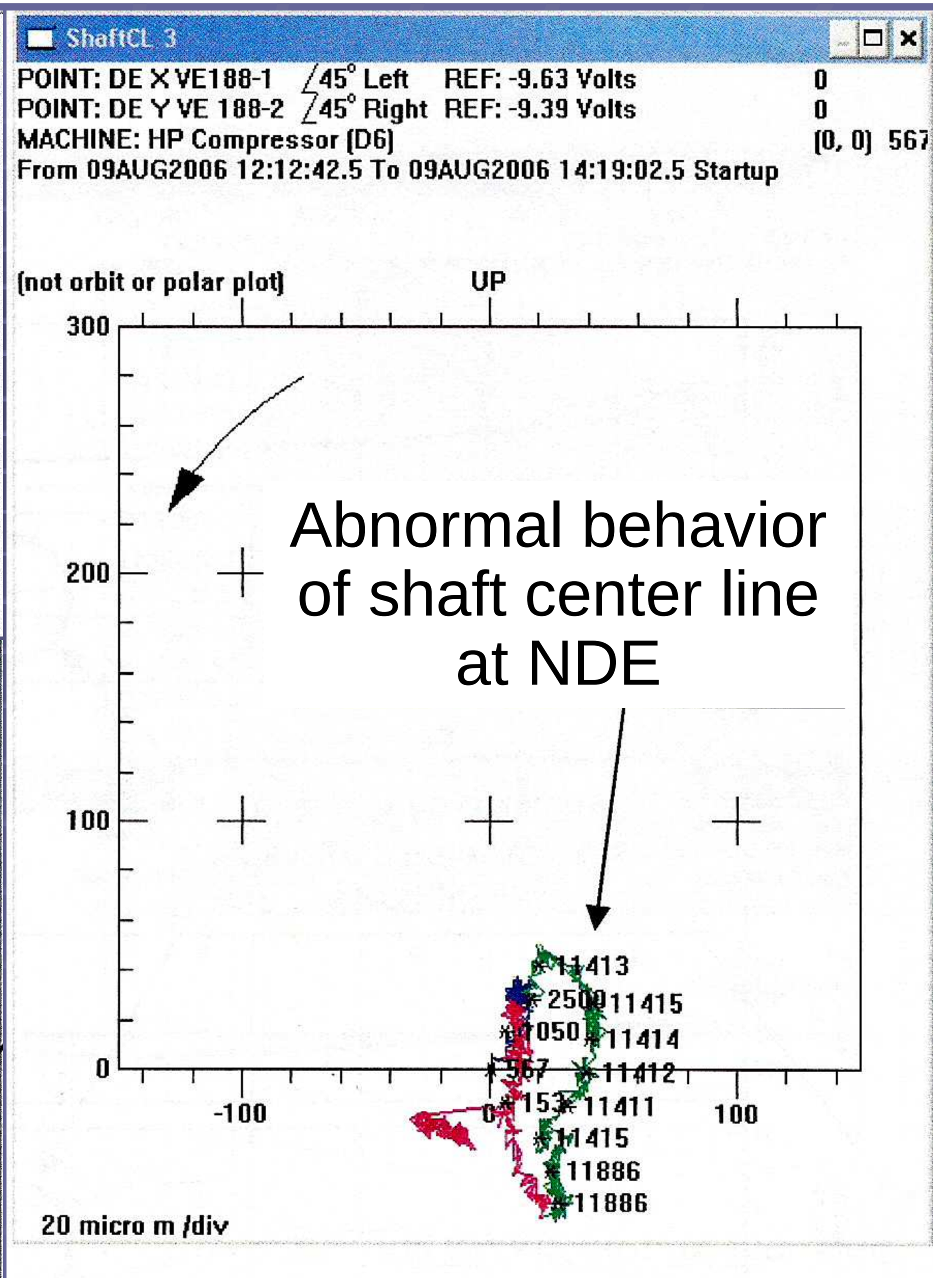
- String Test done in 2002: vibration values below the specified limits (train MCC).
- Startup of the first set of compressors, MCB, in Aug/2006: excessively high vibration at the HP compressor, mainly at the non driven end.
- Records showed that the vibration was basically the synchronous 1X component, which increased with temperature, as the recycle valve was closed.




Waterfall plot of one of the NDE sensors – vibration is basically 1X



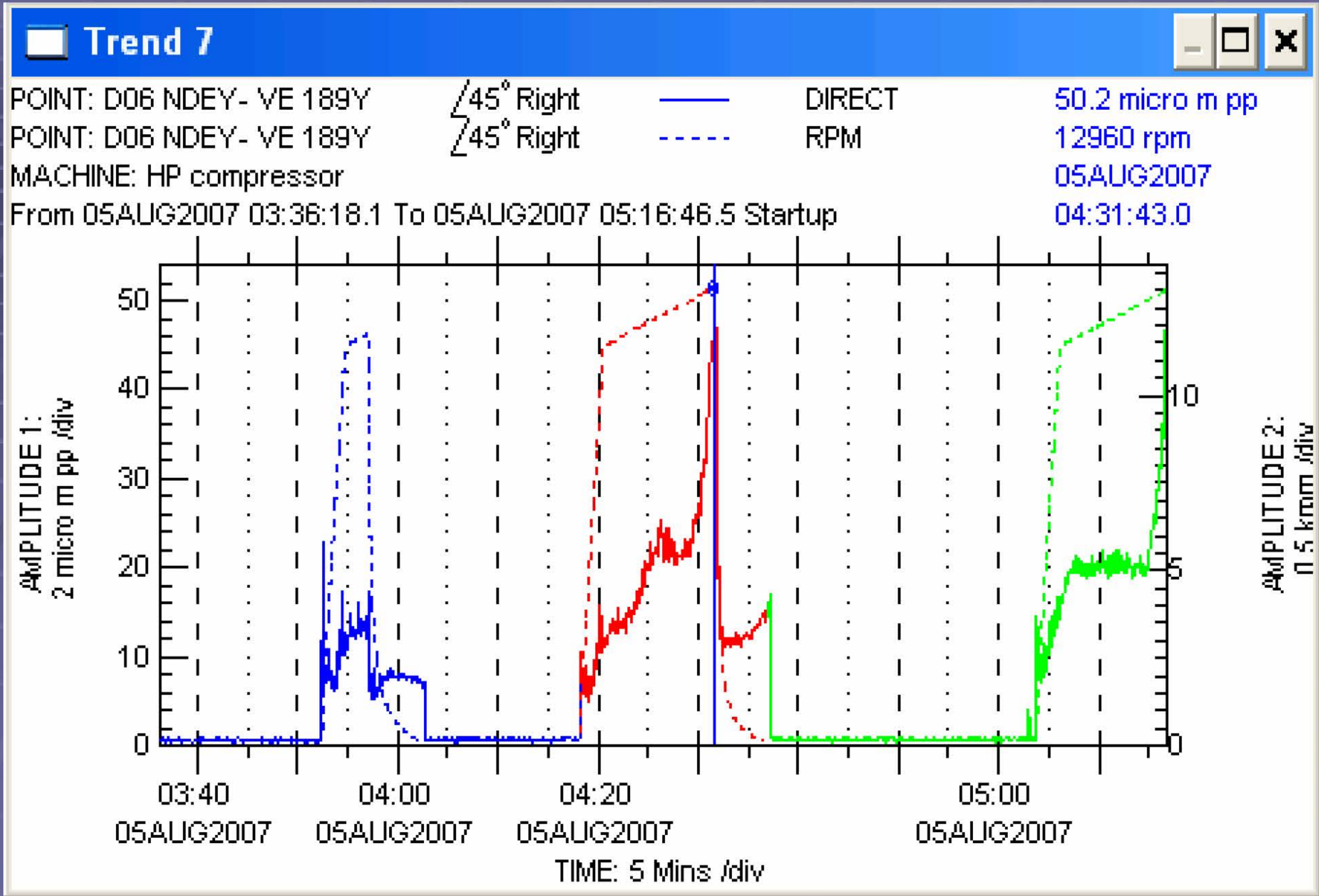




Above: Speed & NDE vibration. Below: Bode plot.

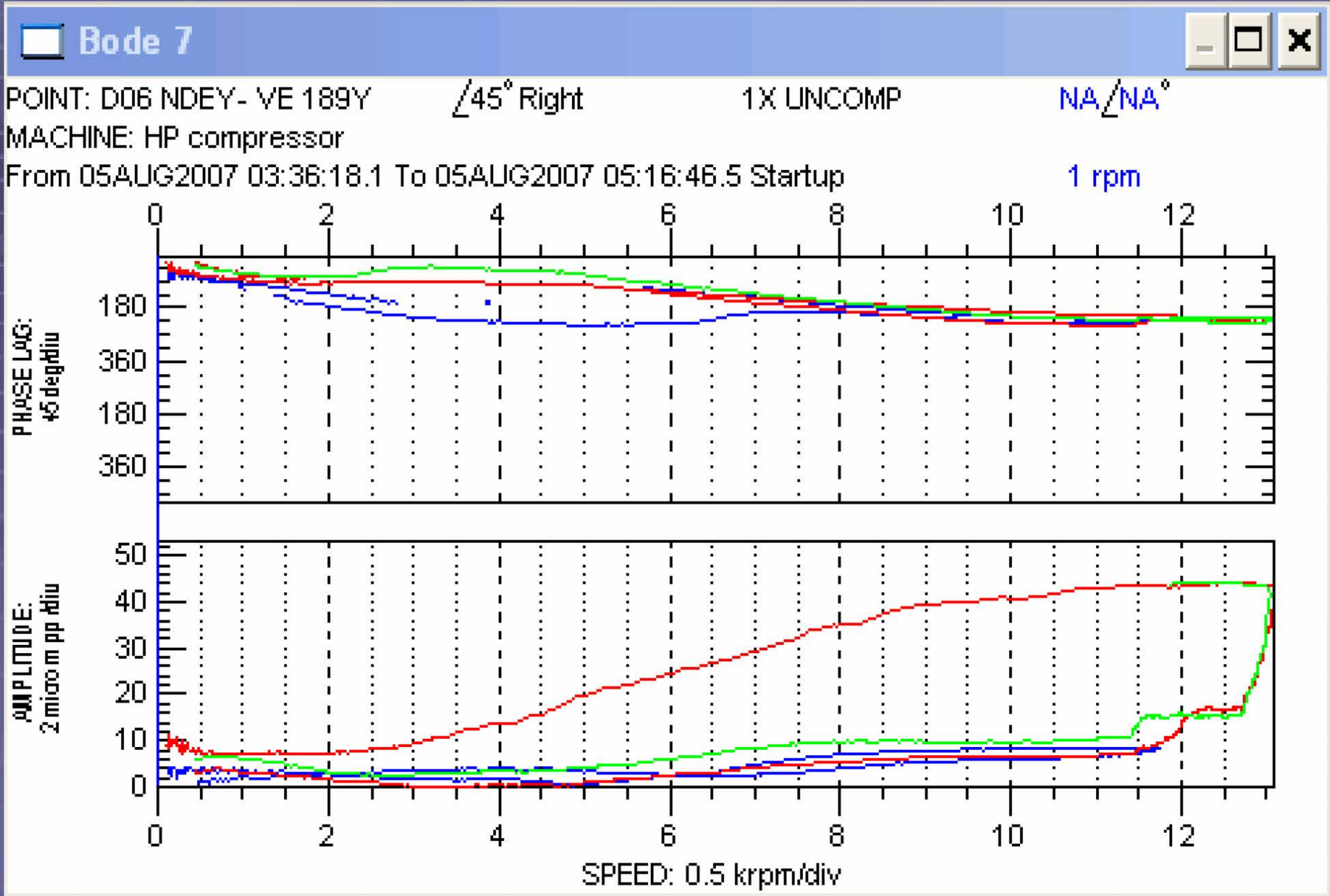
NDE shaft average position: Height changes with constant speed.

# P-50 Compressors (cont.)

- A field balancing reduced somewhat the level of vibration and the MCB train continued to operate precariously until May/07.
- MCA began operation in Sept/2006, also with high vibration on the HP, although smaller than that of MCB. Due to problems in the VSD, the compressor operation was interrupted, which enabled the placement of the MCA HP bundle in the MCB.
- The spare HP rotor, modified as explained below, was placed in the bundle that was previously in the MCB, and inserted into the MCA (Aug/2007).
- The MCC, the train with which the String Test was performed, works smoothly since its startup in Feb/2007.



Trend 7
POINT: D06 NDEY- VE 189Y
∠45° Right
DIRECT
50.2 micro m pp
POINT: D06 NDEY- VE 189Y
∠45° Right
RPM
12960 rpm
MACHINE: HP compressor
05AUG2007
From 05AUG2007 03:36:18.1 To 05AUG2007 05:16:46.5 Startup
04:31:43.0
AMPLITUDE 1: 2 micro m pp /div
50
40
30
20
10
0
AMPLITUDE 2: 0.5 krpm /div
10
5
0
03:40 05AUG2007
04:00 05AUG2007
04:20 05AUG2007
05:00 05AUG2007
TIME: 5 Mins /div



Bode 7
POINT: D06 NDEY- VE 189Y
∠45° Right
1X UNCOMP
NA∠NA°
MACHINE: HP compressor
From 05AUG2007 03:36:18.1 To 05AUG2007 05:16:46.5 Startup
1 rpm
PHASE LAG: 45 deg/div
180
360
180
360
AMPLITUDE: 2 microm pp /div
50
40
30
20
10
0
0
2
4
6
8
10
12
SPEED: 0.5 krpm/div

# String Test of P-53 & P-54 compressors

- Based on the ASME PTC 10 type 1 performance test (full pressure and full load). A mixture of HCs was used, with the same molecular weight as in the design condition.
- Test performed with a string (compressors, gearbox, motor, and VSD) of the P-53, whose design point was higher than that of the P-54, therefore also valid for the latter.
- The test suffered successive delays, mainly due to excessive vibration during the pre-runs, performed with inert gas.
- One week after the P-50 MCB startup, the vibration records of the first pre-run showed that this increase occurred with the 1X component and that it increased with gas temperature.

# String Test (cont.)

- Vendor past experience: Thermal Induced Vibration Unbalance - could occur when components were mounted on the shaft through 2 equal-length and equal-interference fits, which was the current technology of the manufacturer when the P-50 compressors were made.
- Although it did not happen to all rotors (e.g. the successful String Test of P-50 MCC), this type of component mounting could produce an uneven distribution of interference on the shaft during assembly, which, in turn, could lead to a thermal bow during operation.
- This was also applicable to the balancing drum of the String Test HP compressor.

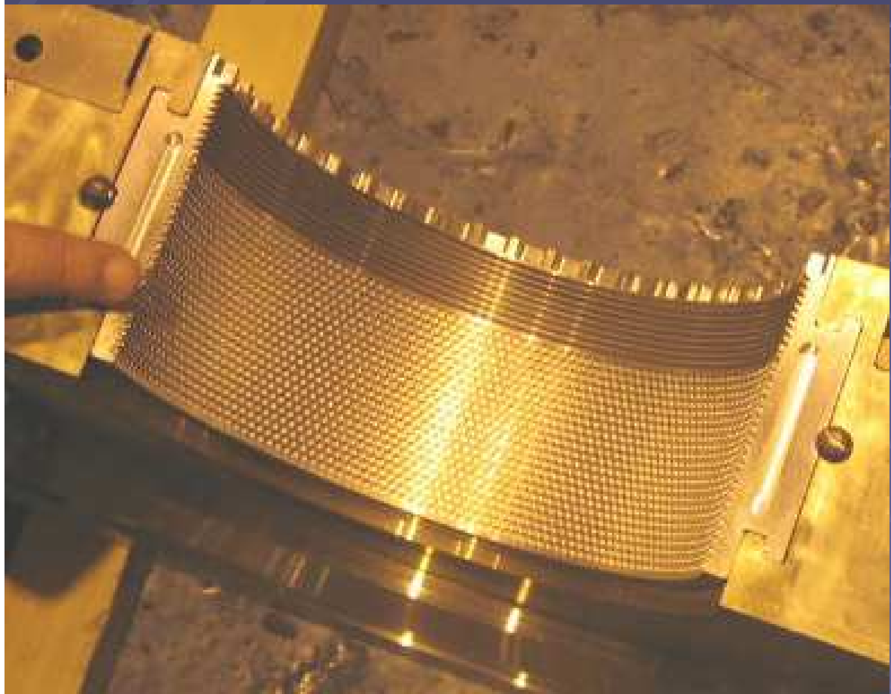
# Balance Piston Design

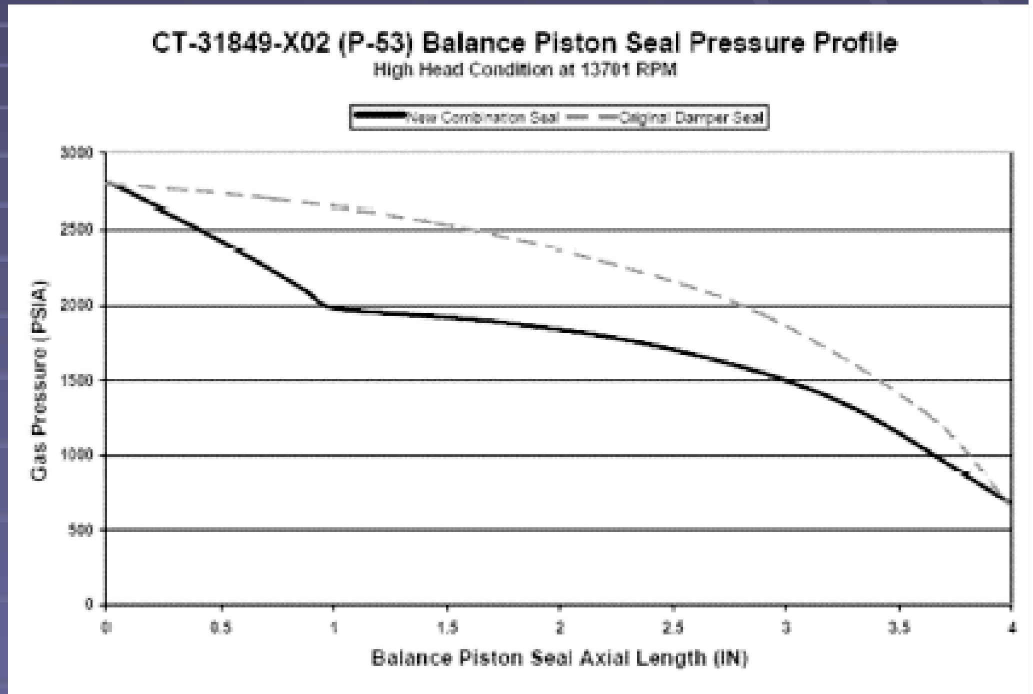
- A cylindrical-shaped design was used for the balance piston of the P-50 HP rotors and was repeated on P-53 and P-54, although the impellers had already a different fit design.
- The axial dimension of one of the interference rings was reduced and a annular slot was created at the face of that side. Thus, this ring was now to act basically as a guide, while the unchanged ring kept the function of holding the drum to the shaft.

# Balance Piston Seal

- At another pre-run, the compressor continued to show a high level of 1X vibration. However, the vibration was now more related to a pressurization effect, caused by the design of the balance piston seal.
- This was of the hole pattern type, to produce damping and ensure the stability of the rotor. However, the stiffness of the seal was of the order of the joint stiffness of the bearing and the squeeze film damper.
- In practice, a large bearing at the coupling end of the rotor. The opposite end, more flexible, became also more susceptible to vibration.
- In order to reduce the stiffness effect of the seal, a stretch of labyrinth seal was added.

# Balance Piston Seal (cont.)






Modified seal and pressure profile over the balance piston

# Unbalance Response

- Third pre-run presented an acceptable level of vibration.
- String test initiated a week later, but interrupted because of problems with the motor. Nevertheless, the vibration values were within the specified limits.
- In view of the P-50 string test, the vendor was asked to perform unbalance response analyses for different conditions of unbalance and clearance.
- To validate the requested analyses, an unbalance response test was carried out before the final String Test, with inert gas, but with full load and pressure.

# Conclusion

- Regarding the HP compressors of the 3 platforms, the vibration problems found on the String test and on operation were considered diagnosed and solved.
- To avoid these problems in future compressors, the author has suggested that the following should be added to the bid specs.:
- (a) An observation that the use of 2 equal-length and equal-interference fits should be avoided.
- (b) For compressors that employ damper seals at the balance piston, these seals should be considered in the unbalance response analysis (by the API 617, they are not) and the unbalance response test should be performed at full load and full pressure.