VIBRATION PROBLEMS WITH AN ELECTRIC MOTOR FOR A CENTRIFUGAL PUMP INSTALLATION

by

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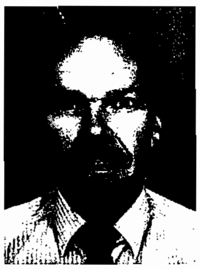

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ABSTRACT

A case history is presented in which vibration related problems with two pole electric motors have delayed acceptance of the complete pump packages by the end user. Noncompliance with contract specifications led to a major "redesign and test" program for the motors. Parameters measured included:

- Bearing housing absolute velocity.
- Shaft relative vibration as measured by proximity probes.
- Glitch (combined electrical and mechanical runout).
- Critical speed separation margins.
- Dynamic response due to deliberate unbalance.
- The relationship between shaft relative vibration and bearing housing absolute vibration.

The initial nonconformances and the corrective actions are described in detail. The design and contract specifications and results of the tests are discussed relative to the requirements of API 541 parts 1 and 2 [6] along with a European standard [8]. Some of the issues examined are relevant to all rotating equipment.

INTRODUCTION

In many large pump installations, contracts for the supply of all rotating equipment in the drive train are placed with the pump manufacturer. The vibration and balance specification for each major item is usually imposed by the end user or contractor. The pump manufacturer will expect this to be correctly assessed by his subsupplier.

The relevance of conformance or non conformance with vibration specifications to "real" problems under actually installed operating conditions is very subjective [1]. Vendors standards will not, and cannot possibly, match every user's requirements.

As a minimum, the subsupplier's "standard" should protect the user from inadequate specification. Correct interpretation of any "nonstandard" requirement is essential.

Misinterpretation, however, can be caused by any of the following:

- Confusing, multiple, contradictory specifications.
- Language barriers.
- Inadequate review of test procedures.
- Inadequate inspection.

This may lead to noncompliance and add unforeseen costs due to redesign, rework, reinspection, or delays. The time of discovery varies as follows:

- Potential noncompliances can be discovered and avoided at an early stage during review of the test procedures.
- Actual noncompliances can occur, but are detected and addressed during tests at manufacturer's works because of comprehensive inspection.
- Actual noncompliances can occur without detection during unwitnessed shop tests. These may be found, prior to release for shipment, by adequate review of documentation.
- Actual noncompliances can occur without detection and the units are shipped to site. These may be detected later if a problem manifests itself at site, or upon delayed review of documentation.

This case history will describe how some large two pole electric motors were found to be outside the customer's specification on the pump test stand, and how this led to a major "redesign and retest" program. Along with describing the design modifications and the results in detail, the vibration aspects of the specification will be examined relative to "real" site problems and will be compared to International Standards for such motors. The presentation will also highlight some clarifications to the API 541 specification and summarize its vibration related requirements for unfamiliar users.

CASE HISTORY

The Packaged Equipment

This order concerned four complete pump packages for use in a refinery on "hydrocracker" service.

The first two trains (Figure 1) contained the following baseplate mounted items:

- High pressure barrel pump.
- Speed increasing gearbox.
- Electric motor with double extended shaft.

The remaining two trains (Figure 2) included the above items, plus an additional "bolt on" baseplate containing:

- Over-running clutch
- Hydraulic power recovery turbine (manufactured by pump supplier)

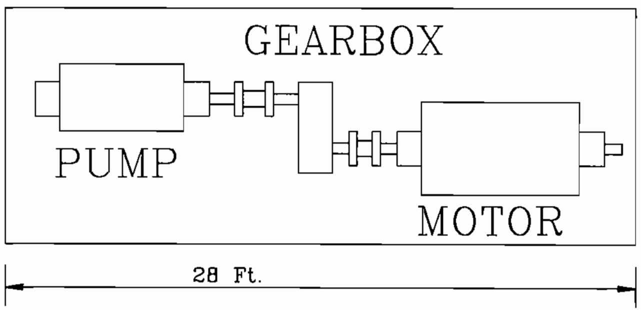


Figure 1. Schematic Showing Pump/Gearbox/Motor Package.

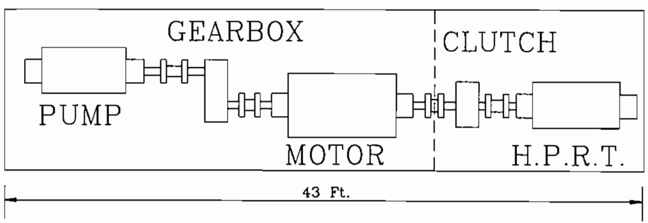


Figure 2. Schematic Showing Pump/Gearbox/Motor/Clutch/HPRT Package.

With the exception of the over-running clutches, all items required an oil supply to the bearings. For this purpose, a force feed lubrication system, manufactured and tested to API 614, and mounted on a separate baseplate, was provided for each pump package.

All four contract gearboxes were witness tested at full load and speed at the gearbox manufacturers works with vibration and bearing temperatures within the agreed limits. The gearboxes were then shipped to the pump manufacturer. The contract specification for the gearboxes was API 613.

All four contract motors were of identical design and were of the squirrel cage induction type with rated power of 1630 kw, 6.6 kv, running at 3000 rpm.

The motors were of the sleeve-bearing type and all were supplied with double-extended shafts. The drive end (DE) refers to the shaft end which delivers power to the gearbox, and the non drive end (NDE) refers to the shaft end which receives power from the hydraulic turbine. Provision was made for proximity probes, which were to be supplied and fitted by the pump manufacturer.

All motors were tested at the motor manufacturers works, and then shipped to the pump manufacturer. One motor was tested "full load," while the others were tested "no load." The contract specification for the motors did not invoke any international motor standards but had similarities to API 541.

Discovery of Initial Problem at the Pump Manufacturer's Works

All four pumps, to API 610, were witness mechanical and performance tested in pump vendor's shop using slave motors and contract gearboxes on contract baseplates. The contract motors, sized for low specific gravity hydrocarbon could not be used for this test since the power required at "end of curve" pump flow, on specific gravity of water, exceeded that available from the motor.

In accordance with the contract requirements, the two packages shown in Figure 1 were then assembled and shipped to the plant site without further testing.

Mechanical and performance testing of a hydraulic turbine required assembly of a complete package shown in Figure 2. The contract motor could be used here because the flowrate of the turbine was less than that required during pump tests.

One set of contract pump, gearbox, motor, and clutch was to be used to witness test two turbines sequentially. The second turbine performance test would be followed by a four hour mechanical run of the complete train, measuring vibration and bearing temperatures on each item of rotating equipment.

During the inhouse mechanical and performance testing of the first hydraulic turbine, the turbine, pump, and gearbox vibration levels were within the specified limits. The motor vibration levels were not. Motor shaft vibration levels of 2.2 mils (*vs* 2.0 mils allowed) were measured by the contract proximity probes and control panel. At this point the problem appeared to be marginal.

The motor was then disconnected from the driven equipment and ran "no load" on the baseplate with similar results. To eliminate possible influence of the contract baseplate, the other remaining motor was tested "no load" off the baseplate on a concrete floor, again revealing similar vibration levels.

A check of the motor manufacturer's records revealed that the motor rotor had been balanced with a full key at the turbine end of the shaft and a half key at the gearbox end; the string test had been the first test with coupling hubs fitted.

The motor supplier carried out trim balancing on couplings at the pump manufacturer's works, and the motor again ran unloaded. Data acquisition instrumentation was used to capture the shaft vibrations during "no load" steady state test and rundown.

The steady state unfiltered vibration levels were marginally improved, but glitch readings (slow roll combined electrical and mechanical runout) of up to 0.8 mils were found to be outside of the allowable limits of 0.5 mils.

Analysis of the rundown traces indicated that there may also be a critical speed between 2600 and 2900 rpm.

This check was not made on the motor.

Attempts to reduce glitch by micropeening did not improve the situation, and these two motors were returned to the motor supplier for rectification.

The Initial Solution

Discussions with the motor supplier determined that:

- Their "standard" method for assessment of vibrations was based on bearing housing velocity criteria.
- The glitch problem had not been identified at the motor manufacturer's workplace.
- The contract required a separation margin of 15 percent between any critical speed and the operating speed. This equated to criticals below 2550 rpm and above 3450 rpm.
- Their calculations showed values of 2450 rpm and 4680 rpm for the first and second critical speeds.
- The rotordynamics calculations had considered the rotor without coupling hubs.
- The overhang of the shaft beyond the bearing housing was unusually long (Figure 3).
- Revised calculations with coupling masses showed the first and second critical speeds reduced to 2270 rpm and 3040 rpm.

It was concluded that the critical observed on test was the second critical.

Further calculations were completed by the motor supplier to determine if, and by how far, the second critical could be raised above the 3450 rpm limit.

They established that the second critical speed was sensitive to the overhang length at the NDE coupling.

The presence of an acoustic baffle plate at the NDE of the motor, which had influenced the positioning of the coupling hub on the shaft, is shown in Figure 3. It was possible to move the coupling hub back 6.0 in towards the bearing so that the flange was behind the baffle plate. Intermediate positions were not possible without restricting access to the coupling bolts. Combining this modification with a reduction in coupling hub weight would result in predicted critical speeds of 2380 rpm and 3920 rpm.

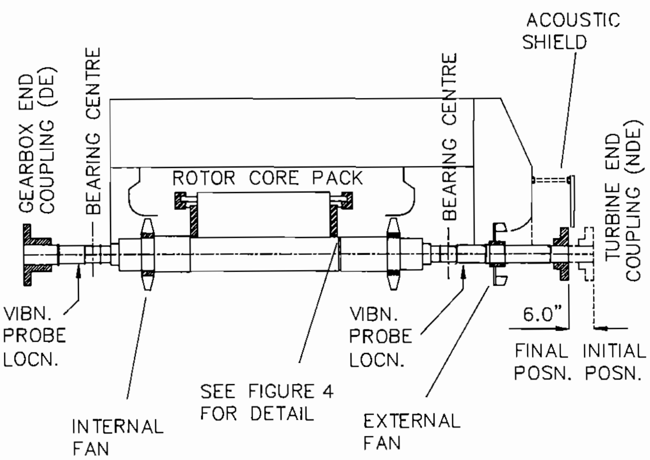


Figure 3. Motor Rotor Arrangement.

This issue, along with the glitch problem, was addressed as follows:

- The shaft surface at the probe areas was slightly remachined, burnished, and demagnetized in accordance with a third party's recommendations.
- The coupling hub mass at the NDE was reduced by special machining.
- The shaft was shortened at the NDE by approximately 6.0 in.
- New, longer coupling spacers were sourced for connection to clutch couplings.

Unfortunately, this fix was unsuccessful. Results of "no load" vibration tests on the two modified motors are shown in Table 1.

Although the measured vibration levels were within the acceptance limits, the motors were rejected for reasons including:

- While mechanical runout was less than 0.25 mils, the glitch was high in both motors. This indicated nonhomogeneous electrical properties within the shaft material.
- Motor Number 1 now showed a first critical speed within 10 percent the running speed. The two different criticals exhibited on this motor implied anisotropic stiffness in the rotor (i.e., the stiffness of the system was changing with angle of rotation). This could possibly be explained by the method of attachment of corepack endstops to shaft (Figure 4). If keys were of a different fit in the groove, such that there was a gap between one or two keys and the shaft, and a close fit of the others, it would behave dynamically in a similar manner to a cracked shaft.
- The two motors exhibited different criticals, which implied different rotor stiffnesses. This could be caused by differences in shrink fit between the rotor core pack and the shaft.

Table 1. Vibration Levels Following Initial Shaft Modification.

Parameter Measured	Readings at Probe Ref. Nos. 1 NDEX	2 NDEY	3 DEX	4 DEY	Allowable limit
Motor No. 1					
Unfiltered Shaft Vibration (Mils) at running speed of 3000 rpm.	1.57	0.90	1.61	0.83	2.00
Glitch - Mechanical & Electrical runout. (Mils)	0.63	0.59	0.71	0.63	0.50
1st Critical Speed	2200	2750	2200	2750	<2550
2cnd Critical Speed	>3600	>3600	>3600	>3600	>3450
Motor No. 2					
Unfiltered Shaft Vibration (Mils) at running speed of 3000 rpm.	1.38	1.69	1.38	1.42	2.00
Glitch - Mechanical & Electrical runout. (Mils)	0.98	0.91	0.75	0.98	0.50
1st Critical Speed	2500	2500	2500	2500	<2550
2cnd Critical Speed	>3600	>3600	>3600	>3600	>3450

- Some of the specified test records had not been completed during original testing. This was due to misinterpretation of a complex specification; the problem was magnified by translation to a different language.
- Failure to recognize the omissions until this late stage meant that the exact status of the motors which had already been shipped to the plant site was not known.

THE FINAL SOLUTION

With testing of the pump/turbine packages behind schedule, the situation was discussed in detail with the end user. The basic task was to provide a "new" surface for the proximity probes to monitor and adjust the shaft design in order to fine tune the first and second critical speeds. It was agreed to return the two motors from site to the motor manufacturer and design and manufacture four new shafts for use with existing rotor corepacks. Before the final design was agreed, the end user required reviews of:

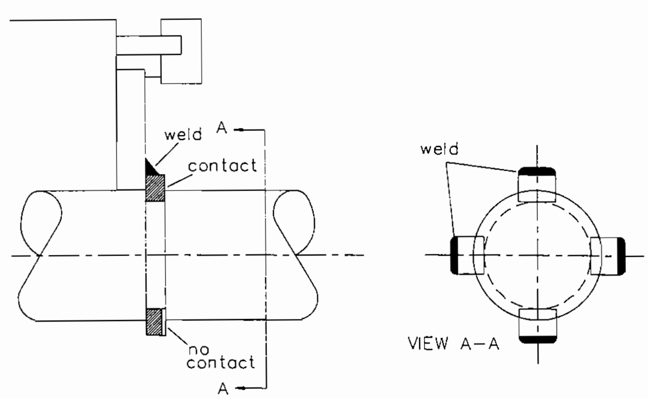


Figure 4. "4 Key" Attachment of Corepack to Shaft—Old Design.

• The relevance of glitch to "real" problems.

• Which surface the probes should monitor. The end user wanted to investigate chrome plating.

• The methods, and assumptions made, in predicting new critical speeds.

These points are discussed in detail as follows.

The Relevance of Glitch to Real Problems

The job specification had adopted what appears to be the "industry standard" of API 670 [12]. This limits the *glitch* to 25 percent of the allowable vibration or 0.25 mils, whichever is the largest. It equated to 0.5 mils for this contract. Before reviewing the proposals for attaining these glitch levels, it is worth discussing the "real" problems that would be caused by supplying a machine with higher levels.

Irrespective of whether glitch is caused by mechanical runout or electrical properties, it is constant for a given probe location. Suppose that a machine had a pure sinusoidal *glitch* reading of 0.8 mils *vs* a specified maximum of 0.5 mils. Then, with no "real" vibration whatsoever, the proximity probe would still register 0.8 mils. If there were an "indicated" vibration of 1.0 mils due to unbalance, the "real" vibration could be anywhere between 0.2 mils and 1.8 mils, depending on the phase angle between the glitch and the unbalance.

A second machine with a lower glitch reading of 0.5 mils and an "indicated" vibration of 1.3 mils would have a "real" vibration between 0.8 mils and 1.8 mils. While the first machine is above specification for glitch, it is at least as good as, and possibly better than, the second machine in terms of "real" vibration. Therefore, high glitch readings serve to reduce the accuracy of the known state of the machine; acceptability is dependant on the philosophy of the user.

Which Surface Should the Probes Monitor

The depth of material "detected" by the probe is of the order of 10 mils and, to ensure satisfactory operation, requires an actual depth of electrically homogeneous material of 20 mils radial. The electrical properties can be influenced by any discontinuities in the material close to the surface, and also by magnetism or residual stresses in the material [2]. The choices for probe detecting surfaces were reviewed as follows:

• Use of the shaft base material was rejected. The new shafts would not be finish machined until after assembly into the rotor corepacks. Since the final glitch could not be measured until then, the motor manufacturer was very concerned about the possibility of scrapping more shafts, and wanted a solution with potential for adjustment. Although there is plenty of published material available to theorize on methods of reducing glitch [2], attempts on the first motor shafts had eventually failed due to electrically non-homogeneous material at probe area.

• Chrome plating onto the base material was rejected. It was explained to the end user that 20 mils was near to the limit for easy deposition of chrome, and that chrome plating is itself porous.

• Adding a shrunk on, nonmagnetic, 316L glitch ring was accepted. While there were potential difficulties, this method allowed scope for correction, by replacement if necessary. To minimize problems, the effect of temperature on the shrink fit of two materials with different thermal properties was evaluated and the rings were made of reasonable length (2.25 in) to keep stress concentration effects at ends of shrink fit away from the probes. The latter was possible because the probes were located in a separate enclosure, external to the bearing housing. Finally, the proximeters that had already been purchased for the contract were changed out due to different calibration constants.

With respect to API 541, paragraph 2.4.5.1.1 implies that glitch problems have been encountered with hot rolled steel shafts larger than 6 in diameter and better results are obtainable with forged steel shafts. Motors to PART II criteria are forbidden by paragraph 2.4.5.1.2 from using sleeves or plating. Paragraph 2.4.7.3.2 implies that this is also applicable to PART I motors.

It is the writer's experience that some customers request AISI 4140 glitch rings to be shrunk on to pump shafts that are made of more exotic material, in order to standardize on proximeters at site.

Review of Critical Speeds

Even though new shafts could be used, the areas for change were limited to minimize redesign of the other rotating parts. The old shaft (with length reduced by 6 in), is shown in Figures 5 and 6 *vs* new shaft machinings that were finally adopted. Glitch ring positions are also shown. To reduce the potential for anisotropic shaft stiffness, the axial positioning of the core pack was effected by a "full circle" split ring rather than the original "four key design" (Figure 7).

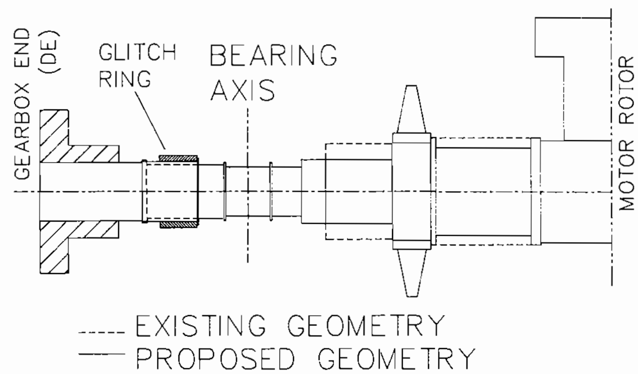


Figure 5. Old Vs *New Motor Shaft Geometry at Gearbox End (DE).*

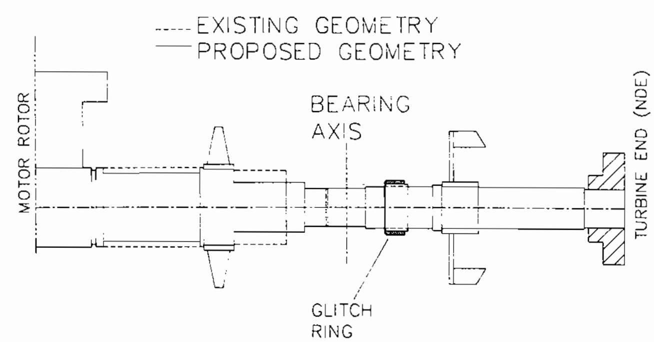


Figure 6. Old Vs New Motor Shaft Geometry at Turbine End (NDE).

Also of major concern was the stiffness contribution of the rotor shrink fit at the rotor corepack/shaft interface. The effects of this, together with the modifications, were evaluated as follows:

• Curve number A1 of Figure 8 shows calculated values of the first critical speed for the existing shaft, *vs* bearing stiffness. The range of stiffnesses obtained from the bearing supplier would result in a predicted critical speed of 2420 rpm to 2450 rpm. The actual criticals as measured on Motor 1 and Motor 2 were between 2500 rpm and 2700 rpm.

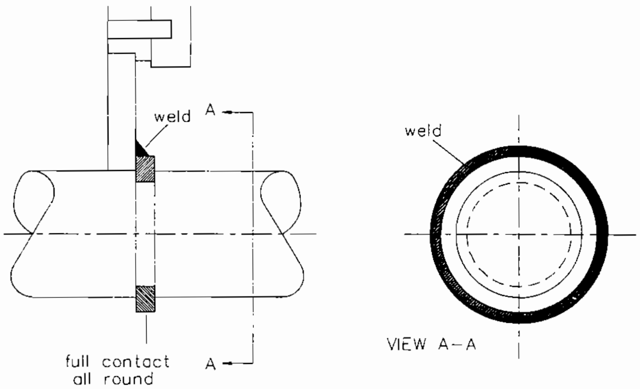


Figure 7. "Split Ring" Attachment of Corepack to Shaft—New Design.

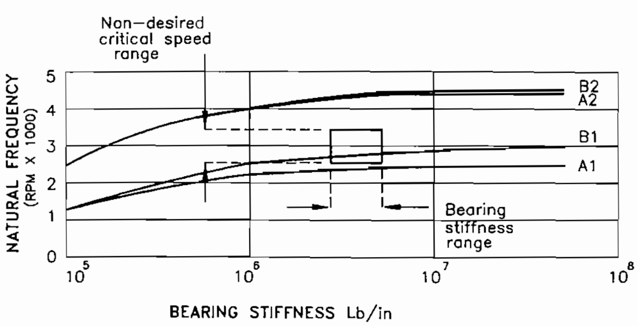
• The preceding prediction was based on the fact that the shrink fit between shaft and rotor core pack had no effect on stiffness of the rotor. In practice, the shrink fit will contribute to the stiffness.

• Curve Number B1 was produced by artificially increasing the diameter under the core pack to obtain a modelled stiffening effect. The resulting first critical speed at the same bearing stiffnesses (between 2680 rpm and 2700 rpm) now matched that found on the test.

• The second critical on test could only be established as greater than 3600 rpm; this could be compared with predicted values of between 4580 rpm and 4650 rpm (Curves A2 and B2). The modelled effect of shrink fit was, therefore, considered satisfactory for the next step.

• Using the same modelled effect of shrink fit, the proposed new shaft geometry was examined. The new predicted criticals are shown in Figure 9. Curves C1 and C2 show first and second criticals for modelled shafts with coupling hubs fitted. Curves D1 and D2 show the effect of adding half the coupling spacer weight to the couplings.

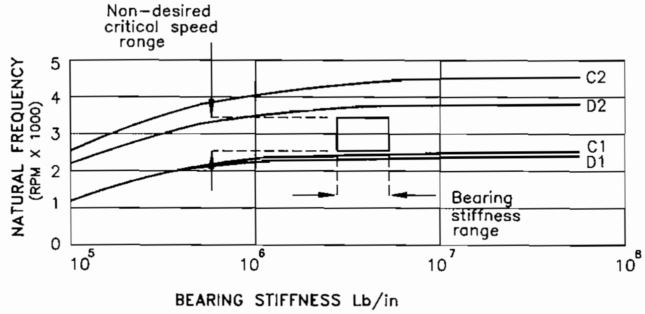
• The final geometry was chosen to obtain the criticals below 2550 rpm and above 3450 rpm under all perceived operating conditions (i.e., a margin of ± 15 percent of running speed).



CODE FOR CURVES

A1 – 1st. Natural frequency shaft unstiffened
A2 – 2nd. Natural frequency shaft unstiffened
B1 – 1st. Natural frequency shaft stiffened
B2 – 2nd. Natural frequency shaft stiffened

Figure 8. Critical Speed Map for Old Shaft with and without Modelled Shrink Fit.



CODE FOR CURVES

C1 – 1st. Natural frequency new shaft plus coupling hubs
C2 – 2nd. Natural frequency new shaft plus coupling hubs
D1 – Same as C1 except 1/2 spacer mass added to couplings
D2 – Same as C2 except 1/2 spacer mass added to couplings

Figure 9. Critical Speed Map for New Shaft with and without Coupling Transmission Units.

• The same geometry was used in the pump vendor's computer program and this verified the motor supplier's predictions.

TESTING AND TEST RESULTS OF REDESIGNED MOTORS

As a result of the non-conformances it was agreed to:

• Increase scope of testing at the motor manufacturer's works to include full "no load," "load" and rotordynamic tests for *all* four motors (original requirement was for one complete test plus three "no load" steady state tests).

• Increase inspection by employing independent third party to witness all balancing and testing.

All four redesigned motors were successfully tested at the motor manufacturer's works with very similar vibration characteristics. The following relates to the test setups.

• Cross references to probe reference numbers used on all vibration traces are given in Figure 10.

• The motor was fitted with both "gearbox end" and "turbine end" coupling hubs during all tests.

• "No load" tests were carried out on solid concrete foundations.

• "Load" tests were carried out on a dynamometer test stand where the motor was mounted on pedestals and connected to a load via a universal-joint coupling, gearbox and motor running as an alternator/generator.

• "Soft foot" checks were carried out at each test location prior to tests. A maximum movement of 0.001 in was allowed.

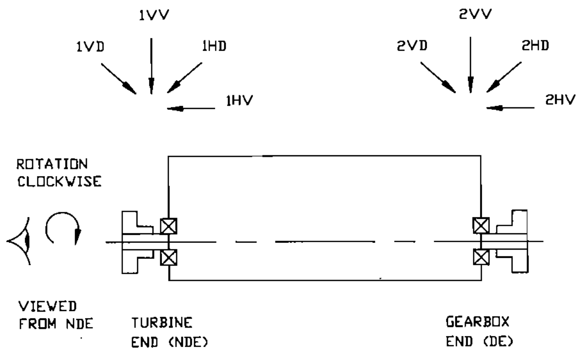
• All vibration readings were taken after appropriate stabilization of bearing temperatures.

The following describes the vibration related tests that were completed on each motor to satisfy the contract.

• "No load" steady state vibrations. These levels were virtually the same as for the "load" test.

• "No load" slow roll readings at 300 rpm to check *glitch*. Glitch levels of 0.33 mils to 0.49 mils were typically obtained.

• "No load" rundown from 3600 rpm, recording vibrations on the shaft probes to ensure that the critical speeds are outside of



REF.NO	CHANNEL NAME	WHERE MEASURED	ANGLE deg.
1	1VD	VERTICAL SHAFT NDE	45
2	1HD	HORIZONTAL SHAFT NDE	315
3	2VD	VERTICAL SHAFT DE	45
4	2HD	HORIZONTAL SHAFT DE	315
5	1VV	VERTICAL HOUSING NDE	0
6	1HV	HORIZONTAL HOUSING NDE	270
7	2VV	VERTICAL HOUSING DE	0
8	2HV	HORIZONTAL HOUSING DE	270

Figure 10. Key to Location of Vibration Probes on Motor.

separation margins. Typical results for NDE and DE bearings are shown in Figures 11 and 12.

• "No load" deliberate unbalance check. A mass of approximately 12 × W/N was added to each of the coupling hubs at about the same reference angle (where W is mass of rotor overhanging the bearing). The motor was allowed to run down from 3600 rpm, while recording vibrations on the shaft probes. Typical results for NDE and DE bearings are shown in Figures 13 and 14.

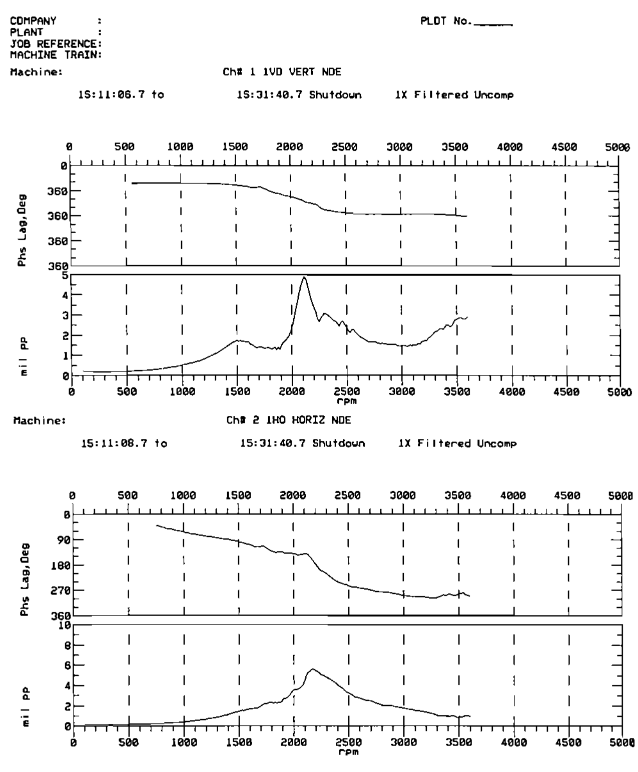


Figure 11. Typical Critical Speed Rundown Check. NDE Bearing.

• With the motor in the "no load" position, it was subjected to five direct on line starts with 80 percent voltage applied. Vibration levels immediately after reaching rated speed were recorded. These levels were unchanged from the "no load" tests.

• "On load" steady state vibrations. Typical frequency spectrums for shaft relative displacements, and housing absolute velocities are shown in Figures 15 and 16.

• After final running tests, the bearing shells were examined and found to be undamaged.

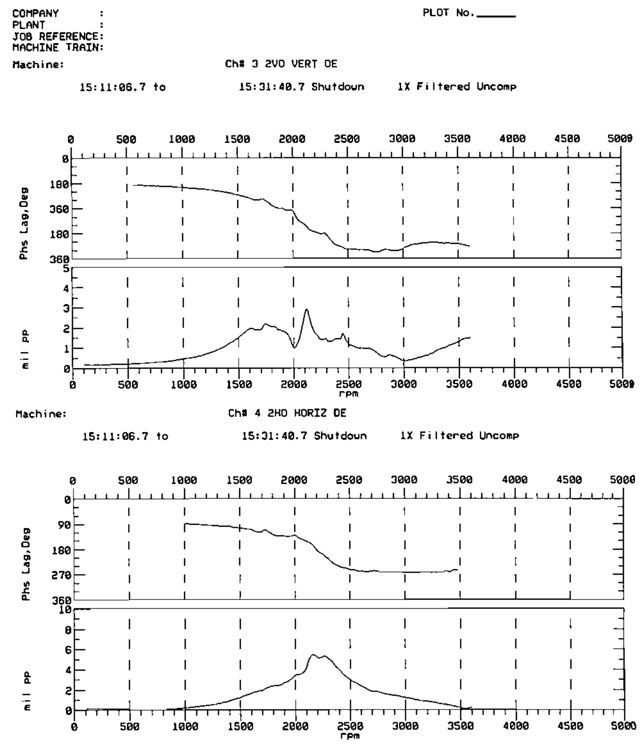


Figure 12. Typical Critical Speed Rundown Check. DE Bearing.

Two of these redesigned motors were eventually string tested in the complete package at the pump manufacturer's works without further problems. Unfiltered, steady state vibration levels, measured during the four hour run, were substantially the same as those found during the "no load" and "load" test at the motor manufacturer's works.

The complete packages have been shipped to the plant site and are currently being installed for commissioning.

DISCUSSION

The following relates to interesting issues that were encountered during manufacturing, building, and testing of the redesigned motors. Where relevant, API 541 is included in the discussion.

Machining

Some problems did occur in obtaining low mechanical runouts at the glitch rings. A target value of 0.2 mils was thought necessary to ensure total glitch values below 0.5 mils. These levels were obtained eventually by subcontracting the final machining; it would appear that center lathes to handle these accuracies on such large rotors are at a premium.

Balancing

The specification called for two plane balancing at 1000 rpm; because the rotor was flexible (running above first critical speed),

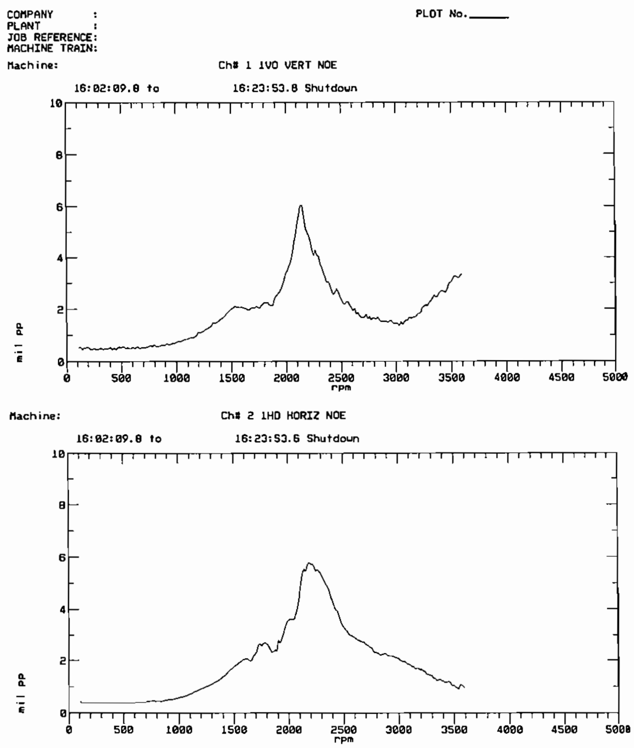


Figure 13. Typical Rundown with Deliberate Unbalance. NDE Bearing.

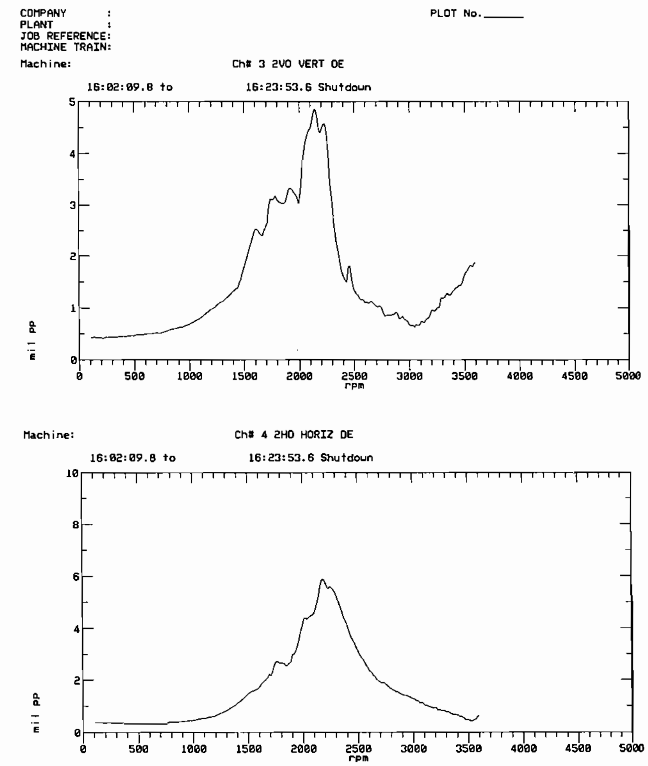


Figure 14. Typical Rundown with Deliberate Unbalance. DE Bearing.

the motor supplier had gained agreement from the end user to balance at the running speed. While this ensures good balance at the running speed, it does not guarantee low vibrations passing through subsynchronous criticals [4, 10]. This is illustrated by the results of the deliberate unbalance response check (Figures 13 and 14).

Ideally, flexible rotors should be balanced at two speeds; first by correcting at one or two planes in the middle of the rotor well below critical speeds, and then at supercritial speeds in two planes further apart [11].

The rotor was balanced in steps. Corrections were made on the ends of the rotor core pack, at the internal fans, and then at the external fan, with final adjustment on coupling hubs. The rotor design was such that compensation could not be made at the center of the rotor.

API 541 allows the buyer to specify whether three plane balancing is to be used without making recommendations as to the positions or speed of balance. If the intent is to specify the third plane as the center of the rotor, this rotor design would not have allowed compliance.

API 541 requires a sensitivity check of the balance machine to be made. The contract specification stipulated that, in addition, the residual unbalance was to be verified by adding a known unbalance at six different angular locations at each of the correction planes in turn. A similar method can be found in appendix J of API 610, 7th Edition [5]. Thus, the balance verification test alone involved 12 different runs up to 3000 rpm. Since the run time is proportional to the balance speed, the additional costs of this test should not be underestimated.

Another minor problem was to determine a location for the known unbalance weight attachment. The manufacturer had made provision for 12 tapped holes on the fans, but some of these had been filled during normal balancing; therefore, removal/replacement of original balance masses was sometimes necessary.

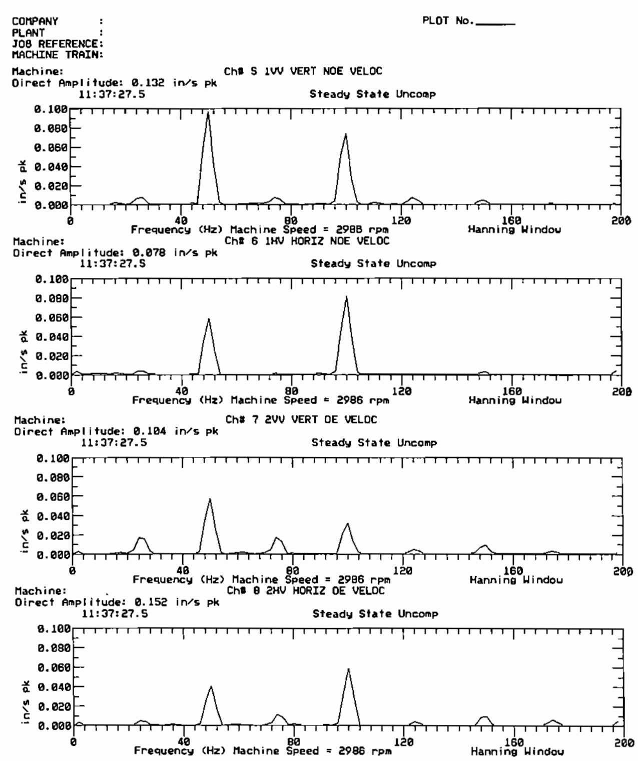


Figure 15. Typical Vibration Spectrums on Shaft During Load Test.

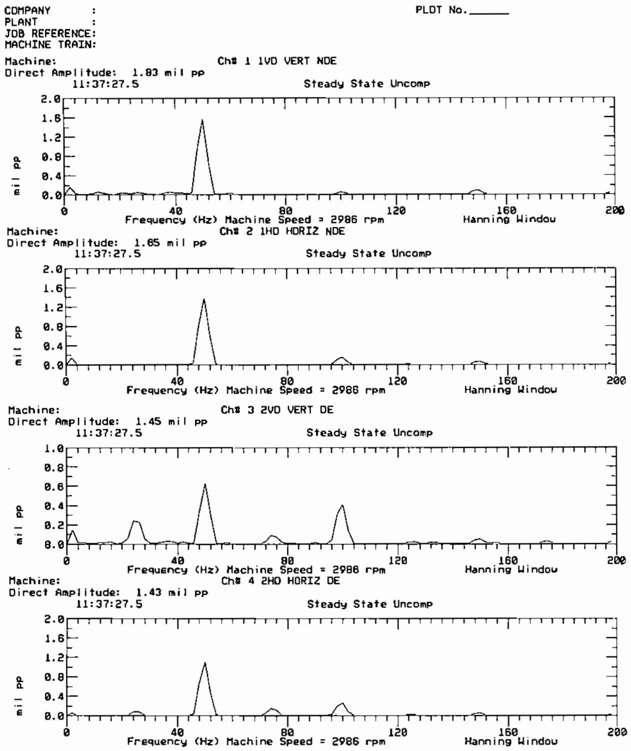


Figure 16. Typical Vibration Spectrums on Housing During Load Test.

Response during Rundown with Deliberate Unbalance at Coupling Hubs

Typical response curves for one of the motors with a deliberate unbalance placed at the coupling hubs is shown in Figures 13 and 14. The contract specification limited the amount of vibration that could be tolerated within ± 15 percent of the operating speed and also as it passed through the first critical speed. One of the motors came very close to the limits that were set. A two speed balancing procedure, as discussed above, may have improved this situation.

However, the relevance of this test, compared with actual site conditions is debatable. While the motor took more than 15 minutes to run down freely during this test, the same level of vibrations would not be expected to develop at the site, since the dwell time at any speed would be much reduced by the braking effect of the coupled centrifugal pump.

Acceptance Criteria and Relationship Between Shaft Relative Readings and Bearing Housing Absolute Readings

To illustrate how opinion varies with regard to acceptable vibration, it is pertinent to make comparisons between the vibration related test requirements of various international standards. For European standards [7, 8]:

- The tests are to be carried out on "no load" with the machine on resilient mounts.
- Shaft relative vibration is not even mentioned as an acceptance criteria in various standards [7 and 8].
- In one set of standards [8], acceptances are dependant on the *frame size* (height from centerline of shaft to foot of motor). Acceptance is based on housing *RMS velocities* for frame sizes below 16 in and housing *amplitudes* for frames above 16 in.
- Both British standards [7, 8] "recognize" that axial vibration acceptance criteria should be *less arduous* than the radial direction.

For API 541:

- The tests are completed on solid foundations.
- The bearing type dictates whether measurements are made on shafts or housings.
- The acceptance criteria are not entirely clear. This is discussed further in APPENDIX 1.
- Axial vibration limits are to be 0.8 ∞ radial limits.

Although not required by the contract, a test was made to see how the *steady state* vibrations responded to a deliberate unbalance, placed at different angular locations, on each of the couplings. The following observations suggest that low levels, as indicated on bearing housings, may not necessarily mean low levels on shaft readings and vice versa.

The response of one of the motors to a deliberate unbalance of 4.06 oz in at the turbine end (NDE) coupling is shown in Figures 17, 18, 19, and 20. This unbalance, which was chosen arbitrarily as the mass of a small bolt, represented a ratio of 60 × W1/N where W1 is the mass of the overhanging shaft, plus external fan, plus turbine end coupling hub and was more than 15 × the unbalance expected from the coupling spacer that would be fitted for the string test. The filtered, 1 × shaft relative vibrations are shown in Figures 17 and 18, as measured at the turbine end (NDE) and gearbox end (DE) bearings, respectively. The corresponding filtered 1 x absolute velocities on the bearing housings are shown in Figures 19 and 20.

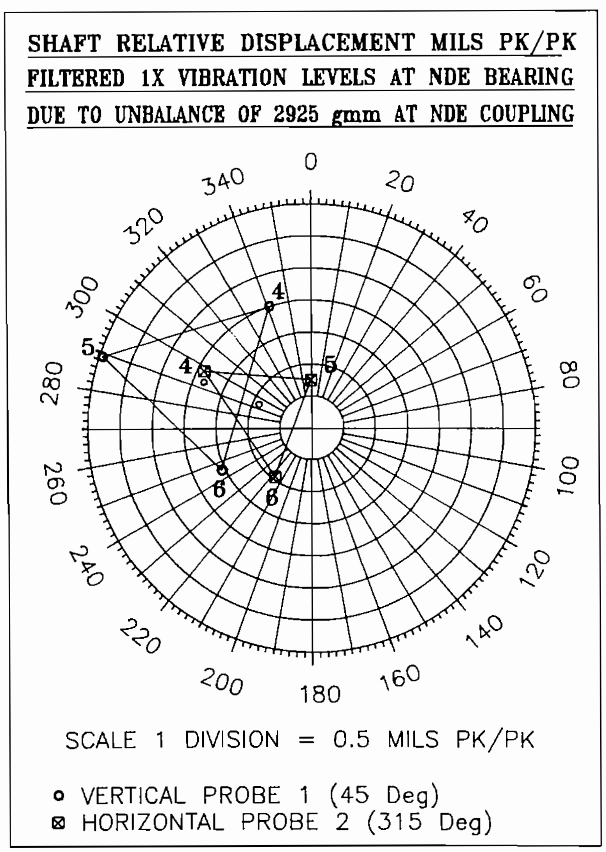


Figure 17. Shaft Vibration at NDE Bearing Due to Unbalance at NDE Coupling.

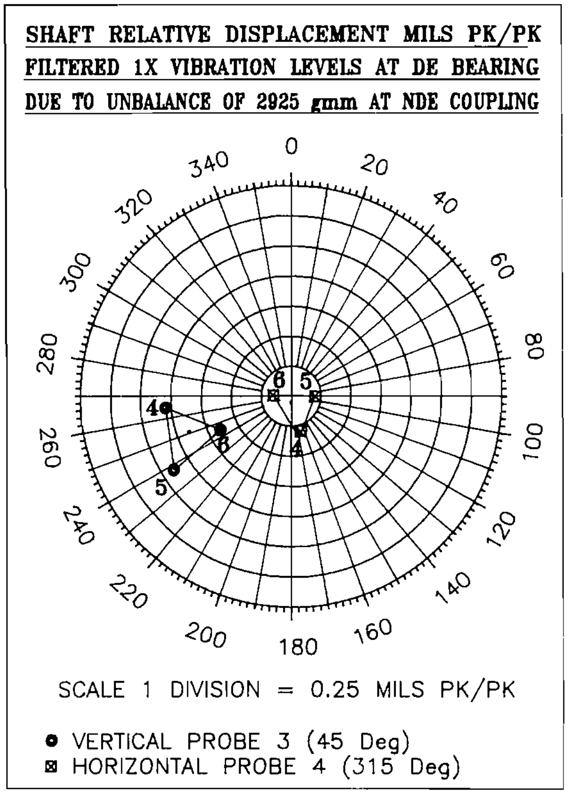


Figure 18. Shaft Vibration at DE Bearing Due to Unbalance at NDE Coupling.

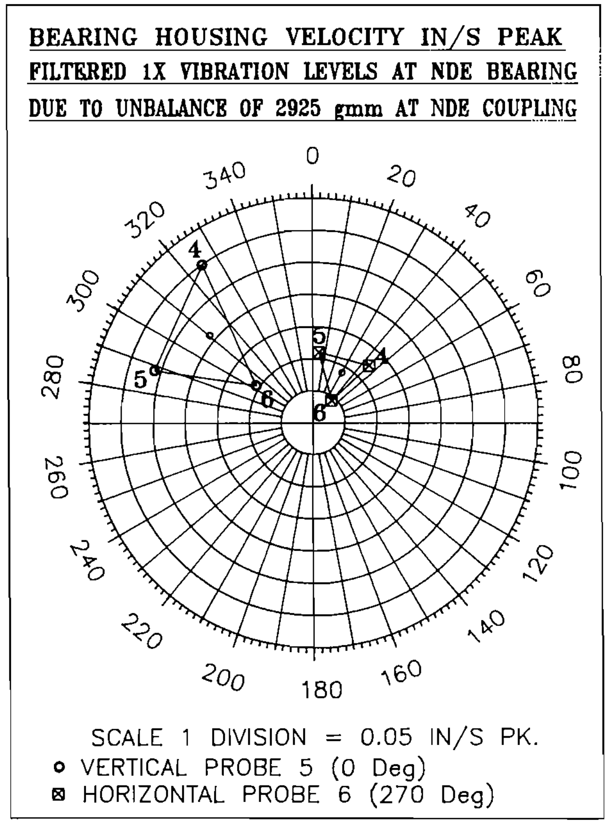


Figure 19. Housing Vibration at NDE Bearing Due to Unbalance at NDE Coupling.

In Figures 17, 18, 19, and 20, the points labelled 4, 5, and 6 represent the resulting 1 x filtered readings with the unbalance placed at 0, 120, and 240 degrees around the coupling hub. The two different symbols represent the two probes that were monitoring at that location. The points can be seen to rotate around the value for which the unbalance is not present.

Of some interest in these plots is the cross coupling effect, where addition of unbalance at one end of the motor affects the readings at the other end. As expected, addition of unbalance at the NDE affects the NDE readings more than the DE readings. Of more interest is the interrelationship between shaft relative displacement and housing absolute levels. It would appear from first glance that mass could be added at a particular location to improve the vibration levels. It is apparent, however, that a reduction in one probe reading may lead to an increase in another probe reading.

For example, suppose that probe number 2 had been used in isolation to determine an optimum position for addition of corrective masses; the plot of Figure 17, probe 2 would suggest an angular location midway between points 5 and 6 (i.e., 180 degrees).

Close examination of Figures 17 and 19 reveals that the reading on probe 1 would actually *increase* while the other probes would have similar levels at different angles. Similarly, while addition of mass at location 6 would be beneficial for readings from probes 3, 4, 5, 6, 7, 8, the readings on probes 1 and 2 would increase.

Obviously the influence of the electromagnetic forces within such machines tends to complicate and distort simple unbalance logic.

CONCLUSION

The basic lessons learned can be applied to most rotating equipment as follows:

- Ensure that the intent of the specification is appropriate.

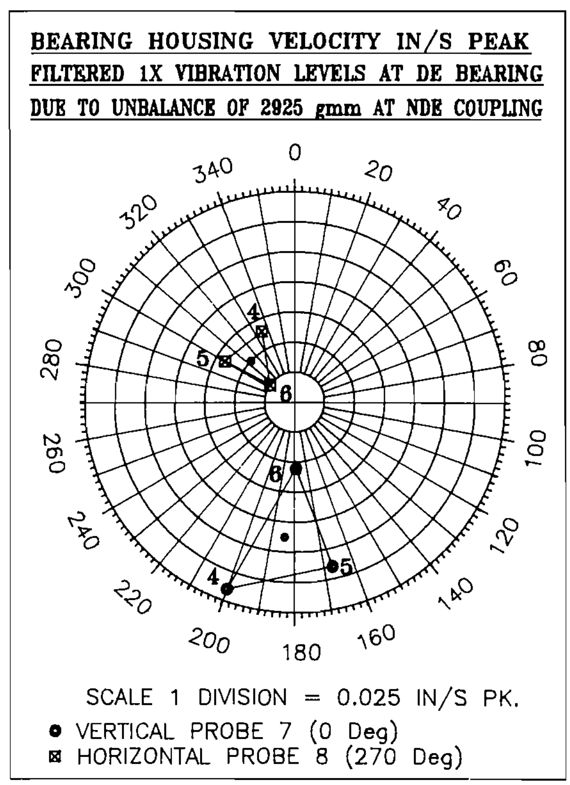


Figure 20. Housing Vibration at DE Bearing Due to unbalance at NDE Coupling.

VIBRATION LIMITS FOR MOTORS WITH SLEEVE BEARINGS
MEASUREMENT ON HOUSING IN RADIAL DIRECTION

MOTOR SPEED (RPM)						
	@ 60 Hz →	720	900	1200	1800	3600
	@ 50 Hz →	600	750	1000	1500	3000
DISPLACEMENT	UNFILTERED IN. PK/PK	0.0015	0.0015	0.0015	0.0015	NM
	FILTERED 1X IN. PK/PK	0.001	0.001	0.001	0.001	NM
	FILTERED 2X IN. PK/PK	0.001	0.001	0.001	0.001	NM
VELOCITY	UNFILTERED IN/S PK	NM	NM	NM	0.14	0.14
	FILTERED 1X IN/S PK	NM	NM	NM	0.09	0.09
	FILTERED 2X IN/S PK	NM	NM	NM	0.09	0.09

NM INDICATES NOT TO BE MEASURED
AXIAL LIMIT IS 0.8x RADIAL LIMIT

Figure 21. Proposed Acceptance Criteria to Replace Figure 1 of API 541.

• Ensure that the requirements are understood and carried out.

These lessons, as applied specifically to electric motors are expanded below.

Ensure That the Intent of the Specification Is Appropriate

The above case history has demonstrated that adherence to arduous specification can be achieved. However, the level of specification should be related to the criticality of the service conditions. The following points should be considered when selecting a specification.

• It has been shown by comparison of different International Standards that there is a large variation in opinion as to what vibration tests and acceptance criteria should be applied to electric motors.

• It has been found that low vibration levels on shaft may not necessarily mean low levels on bearing housings, and vice-versa. Where proximity probes are fitted, it is recommended to set limits on both shaft relative vibration AND bearing housing velocity.

• It has been discussed how some test requirements, such as the unbalance response test in this case history, may not be relevant to conditions that will be experienced at site.

• Results after the "initial solution" demonstrated that two motors of seemingly the same design can exhibit different rotordynamic behavior if manufacturing tolerances, such as shrink fits, are inadequate. Therefore, specifying rotordynamic tests on one motor out of four may not be sufficient where the declared design criticals are near to the acceptance limit.

• Steady state vibration levels on motor during "no load" and "load" test at motor manufacturer's works and during "string" test at pump manufacturer's works were not substantially different. In this instance, it could be concluded that the "no-load" tests would have sufficed. In general, however, it is known that the thermal effects of a load test can greatly influence the rotordynamic behavior of a motor.

Ensure that the Requirements Are Understood and Carried Out

Irrespective of the merits of the specifications, once the requirements have been imposed, it is necessary to ensure that they are interpreted correctly. This case history has showed how an initial misunderstanding by the motor manufacturer led to a delay in finding nonconformances. It is recommended that:

• The specifications are clear; simple reference to international standards may not be sufficient since they themselves may need clarification. A discussion about API 541 requirements is given in APPENDIX 1.

• Account is taken of language and translation difficulties.

• "Proof positive" of manufacturer's understanding is obtained early in the contract. This can be achieved by requesting and reviewing a detailed manufacturing or test procedure as appropriate. A statement of "compliance with specification" by the manufacturer is not sufficient.

• For contracts with arduous specification, comprehensive inspection should take place at the motor manufacturer's works.

APPENDIX 1

Discussion of the Vibration Requirements of API 541 Form Wound Squirrel Cage Induction Motors—250 Horsepower and Larger

The requirements of API 541 are not clear for anyone who is interested only in the vibration related parts of the specification. For example, the various acceptance criteria are revealed under the headings of "Frame and mounting plates" (para 2.4.2.1), "Shaft" (2.4.5.1.2), "Resonances" (2.4.7.1.1), "Vibration" (2.4.7.3), "Bearings and bearing housings" (2.4.8.3) and "Special tests" (4.3.3.3.3). Similar confusion occurs when trying to define what to measure and where. To translate this specification into another language may prove problematic. This appendix is a personal interpretation and would act as a starting point for anyone reading the specification for the first time.

Where Are Measurements to be Made And Which Acceptance Criteria Is to be Used?

Paragraph 2.4.8.15 states that, for 2 and 4 pole motors that are not fitted with proximity probes, a provision shall be made to measure shaft vibration with hand-held pickups. Paragraph 2.4.7.3.2 states that for sleeve-bearing motors, measurements should preferably be made on shaft and that, if fitted, proximity probes should be used. Therefore,

For Two and Four Pole Motors with Sleeve Bearings:

• All will have readings taken on shaft, either with probes if supplied, or on a shaft surface suitably prepared for hand held pickups.

• No axial readings need to be taken.

• Acceptance criteria will be to Figure 2 in the API 541 standard, if the motor is to meet PART I requirements. For PART II motors, it would appear that paragraph 2.4.5.1.2.2 supersedes Figure 2.

• If a "special test" is specified, paragraph 4.3.3.3.3 would require a frequency sweep and the allowable filtered levels would be reduced to 20 percent of the unfiltered allowables.

For Motors with Six Poles and Higher with Sleeve Bearings

• Provision for hand held pickups or proximity probes is not mandatory, and if they are not provided, readings will be taken on the housing in the radial and axial directions.

• Acceptance criteria for the radial direction will be based on Figure 1 if the motor is to meet PART I, or Figure 5 to meet PART II. Figure 5 is the most arduous.

• The allowable axial vibration levels will be 0.8 ∞ those in the radial direction.

For Motors with Antifriction Bearings

- Readings will be taken on the housing in the radial and axial directions.
- Figure 3 would always apply.
- The allowable axial vibration levels will be .8 x those in the radial direction.

What Are the Units of Measurement?

Figures 1, 3, and 5 in API 541 refer to readings on the housing. Figure 2 deals with shaft relative readings. These figures are the same in format showing graphs of peak velocity (in/s) *vs* speed (cycles per minute) for various lines of constant peak-to-peak displacements (in). Heavy lines are meant to indicate acceptance criteria for filtered and unfiltered vibrations. It is not *explicit* whether it is velocity, displacement, or both that are to be measured. Referring to Figure 1 of the API 541 specification:

- The boundary for acceptance of unfiltered readings is defined by a constant *Displacement* line (0.002 in), which meets a constant *velocity* line (0.14 in/s) at a frequency of 1800 cycles/min.
- At first glance, it would appear that the most arduous unfiltered criteria would be *displacement* below 1800 cpm and *velocity* above 1800 cpm. However, because low frequency vibrations show up as large displacements and high frequency vibrations show up as large velocities, confusion can occur. For example, suppose that a motor running at 900 rpm had unfiltered displacement readings of 0.0015 in, with filtered 1 × and 2 × readings of 0.001 in. This would pass the displacement criteria, but would it pass the velocity criteria? The 2 × filtered reading would correspond to approximately 0.09 in/s peak velocity which is allowed, but it is greater than the IMPLIED unfiltered allowable of 0.07 in/s. In such a case, if common sense prevailed, the motor would be passed. First of all, it is not sensible to allow 2 × filtered levels higher than the unfiltered. Secondly, it could be *implied* from the chart that overall velocities of 0.14 in/s and filtered values of 0.09 in/s are acceptable.
- For motors running at 3600 rpm there does not appear to be an acceptance value for 2 × running speed frequency.

Similar comments can be made for Figures 3 and 5.

Referring to Figure 2 of the specification,

- The issue is confused by introducing velocity.
- Signals from the shaft, taken with hand held shaft riders, measure velocity; displacement can be obtained by integration.
- Signals from proximity probes indicate the shaft displacement directly. They can ONLY measure displacement since the signal cannot be differentiated. This is the preferred method of measurement and therefore the units of measurement for shafts should be displacement.

A final observation is that the figures only seem to concern 60 Hz speeds. Many contracts are sold ex USA to 50 Hz markets. Since there are not too many speeds at which motors can run, and the concern seems to be for 1 × and 2 × filtered readings, it may be better to describe the limits with tables, rather than graphs. An example of such a table is shown in Figure 21, which reflects the requirements of API 541 Figure 1.

Alternative to Hot Correction Factor

A correction factor, as described in API 541 paragraph 2.4.7.3.3, is allowed to be used for vibration levels during a "load" test. The intent is to allow for the effect of the test set-up on the readings.

To eliminate this correction factor approach, "load" tests for this contract were completed after "balancing" the universal jointed coupling hub (UJC hub). This was achieved by running the motor disconnected, then connected to the load, and in each case, recording the filtered 1 × vibration readings and phase angle at the drive end bearing. The vector difference between the readings was attributed to an unbalance in the UJC hub. The amount of unbalance was calibrated by adding a trial mass to the UJC hub and again measuring the vector change in vibration. Final masses to compensate the unbalance were then added to the UJC hub. To verify the correction, the hubs were then rotated relative to each other in 120 degree intervals with only minimal change of vibration. It was now considered unnecessary to apply a hot correction factor.

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