

FIELD EXPERIENCES WITH ROTORDYNAMIC INSTABILITY
IN HIGH-PERFORMANCE TURBOMACHINERY

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Instability in centrifugal compressors is a very serious problem and one that has caused the Phillips Petroleum Co., and our partners, considerable consternation and eventually resulted in lost production, extended construction periods and costs, and heavy maintenance expenditures.

Phillips Petroleum has been involved in two major incidents relative to rotordynamic instability of centrifugal compressors. Both incidents have resulted in serious problems for all parties concerned. In one instance we sustained a substantial loss of revenue on crude oil sales since we had no method of disposing of the gas which is associated with oil production. In the second instance we faced a possible shortfall in delivery of gas to meet contractual commitments. Fortunately, with the combined efforts of the manufacturer and our own people we were able to avert a major shortfall in gas delivery.

EKOFISK OIL FIELD PROBLEM

The Ekofisk oil field in the Norwegian Sector of the North Sea was developed by the Phillips Petroleum Norway Group (1) on the premise that early delivery of some of the crude oil would be possible by temporarily reinjecting all of the gas produced. This type of installation would allow production of a part of the oil prior to completion of the gas pipeline to shore. Accordingly, process and compression facilities were installed offshore to separate oil from gas and to compress the gas to approximately 625-bar pressure for injection into the formation. This arrangement would permit producing oil to the equivalent gas capacity of the injection compressors. Although this represented only a portion of the ultimate capacity of the field, nevertheless it was very important to the principals involved since it was the beginning of a return on a very large investment.

The reinjection compressors receive gas from the separator area at 68-bar pressure and boost it through two parallel trains to 625-bar pressure. Each train consists of two 15 000-kW units in series, with the first unit discharging at 240-bar pressure and the second unit at 625 bars. Each casing contains 8-stage rotors with back-to-back impeller construction. The flow was from the suction through the first four impellers in series out the center of the casing to an interstage cooler and then return to the opposite end of the compressor with flow through the final four impellers and discharge from the center of casing. This arrangement necessitated a long labyrinth in the center to break down discharge pressure from 625 bars to 440 bars.

An attempt was made in March 1974 to commission the reinjection compressors. By June it was evident that we were not going to be able to operate them in

their existing state and further that we had a full-grown rotordynamic instability problem with no immediate solution in sight. What followed was a long period of testing which, for the most part, yielded negative results and which has become characteristic of similar situations. First the manufacturer tried the relatively simple changes such as adjusting seal clearances, seal lockup, adding seal grooves, adjusting lube oil temperatures, and numerous bearing configurations.

All these many changes required that we operate the compressors to determine their effectiveness. At the conclusion of this period, several months had elapsed with no solution in sight. At this time it was decided by the manufacturer to design a squeeze film bearing that in a subsequent test proved to be successful and was adopted as an interim solution. On Christmas Eve 1974 we started gas injection into the formation, and shortly thereafter we commenced producing crude oil close to the anticipated design rate for that period.

Concurrently with the design and manufacture of the new squeeze film bearings the compressor manufacturer started work on a new rotor design incorporating a larger diameter shaft and a slightly shorter bearing span. At about the same time we decided to hedge our position and secure new compressors for the final stage of each string which incorporated a change in design by using two compressor bodies rather than one. Each set of compressors, which were manufactured by different firms, was designed, built, and full-load tested at actual operating conditions in approximately 1 year, a very laudable accomplishment. It is interesting to note that the calculated payout of these new compressors was somewhat less than 1 week in terms of lost crude oil production.

We were able to operate the compressors successfully with the squeeze film bearings and continued injecting gas and producing crude oil until the summer of 1975, when the new design rotors were installed. The machines have operated successfully since. They operate at reduced head now since the formation pressure is much lower. They are actually only needed when the gas pipeline is out of service for some reason or when it is operating at restricted capacity.

HEWETT GAS PLANT PROBLEM

The Hewett Partners (2) operate a gas plant on the East coast of England that furnishes natural gas for use in Britain. The gas comes from wells located approximately 17 miles offshore. All compression equipment is located on shore. During the initial operation of the plant no compression was necessary but, as the field pressure declines, compressors must be added to deliver gas at a constant 68-bar pressure. Three identical 3000-kW centrifugal compressor units were installed in 1973, two additional identical units in 1976, and finally one additional identical unit in 1979.

The compressors are divided between two parallel trains. There are other compressors of different sizes in each train, but they have no bearing on the problem under discussion. The compressors in question are fitted with single-stage, back-to-back, parallel-flow impellers. They operate at 13 750 rpm. Bearings are the 5-lobe, pressure-pad type.

Events concerned with this problem can best be presented in a chronological order as this gives a much better feel of the time frame required to identify and solve this type of problem.

Chronological History

June 1973: Units 1, 2, and 3 were installed. Unit 3 compressor rotor was removed during commissioning, and the original spare rotor was installed. The rotor has remained on this unit through current date.

January 1975: Unit 1 rotor was damaged by foreign object in compressor. Unit 1 rotor was removed and replaced with original unit 3 rotor.

January 1976: Unit 6 compressor was commissioned.

April 1976: Unit 7 compressor was commissioned.

November 1976: Unit 6 was shut down because of high vibration, presumably caused by foreign object damage, and replaced with the rotor removed from unit 1 in January 1975. The compressor has experienced numerous intermittent tripouts from that time to present date. Numerous attempts were made to correct the problem: Alignment was checked, and even the gearbox and coupling were changed in an effort to determine the cause of the problem.

At this stage no one thought that the problem could have been instability, particularly since many hours of running time had been logged on this and similar compressors without difficulty. In retrospect, it would appear that this was the beginning of our instability problems, although on this compressor we have never had our diagnostic instruments connected during a tripout to confirm this. A check somewhat later than the onset of tripouts indicated a 0.4-mil amplitude at 2/3-running-speed frequency. On this particular compressor the instability has always remained bounded when instruments were in place, but there is certainly no reason to believe now that the many tripouts were anything other than instability.

November 1978: Unit 7 started to experience random intermittent tripouts similar to those experienced on unit 6. In late November the unit would not go back on line because of constant high-vibration tripouts. Unit 7 rotor was removed, and unit 6 rotor was installed.

2 December 1978: Unit 7 continued to experience tripouts. Unit 6 rotor was pulled and rebalanced, a new coupling assembly was fitted, and the rotor was reinstalled.

19 December 1978: With little or no improvement in operation of the compressor of unit 7, unit 6 rotor was removed and unit 7 rotor, which had been repaired and rebalanced, was installed.

22 December 1978: There being no improvement in the operation of unit 7 and with high-vibration tripouts continuing, the unit 7 rotor was pulled, rebalanced, and reinstalled. The unit was operable, and once again on-line operation was established at reduced speed and head.

11 January 1979: Frequent tripouts in unit 7 again required a shutdown. Unit 6 rotor was installed, and an acceptable level of operation was finally established although at something less than normal head.

5 February to 7 March 1979: During this period numerous attempts were made to commission the unit 8 compressor, but we were unable to do so because of high vibration and the resulting tripouts. Because of damage from foreign objects and frequent tripouts, it was necessary to rebuild the compressor rotor. On 7 March a short run was attempted with the rebuilt rotor in position, but this ended with a severe tripout because of high vibration on both bearings.

12 March 1979: For the first time a run was attempted with unit 8 with diagnostic instruments in place, and the results clearly showed a subsynchronous vibration at 2/3 running speed. The compressor tripped at 13 200 rpm as a result of this subsynchronous vibration. It was a typical example of an instability trip accompanied by a slow buildup of the 2/3-running-speed component, which then suddenly became unbounded and reached several mils amplitude.

Figure 1 is a plot of speed versus amplitude of vibration for the unit 8 inboard horizontal bearing probe. This figure shows the first indication on instruments of the presence of instability. Instability is building up slowly around 8000 to 9000 cycles per minute, and at this point the amplitude is quite low, only about 0.25 mil. Figure 2 is a plot of speed versus amplitude of vibration for the inboard horizontal bearing probe and shows the instability tripout that occurred immediately following the plot in figure 1. Tripout occurred at a frequency of 9300 cycles per minute and an amplitude of 12.5 mils.

This was the first clear indication of the reason for our inability to commission unit 8, and in all probability was the cause of all the troubles on the other units, particularly unit 7. Six months had elapsed since the beginning of serious troubles on unit 7, and two months since starting to commission the unit 8 compressor. As an example of the amount of work encountered on this problem, rotors were changed eight times during this interval.

15 March 1979: Unit 2 had been operating since commissioning in June 1973 with the original rotor and during this period displayed no problem with tripouts from high vibration. It was decided to remove this rotor to make minor repairs of a nature not related to the instability problem. Unit 7 compressor rotor was installed, and immediately upon startup the compressor began experiencing high-vibration tripouts. Examination with diagnostic instruments in place a few days later revealed a strong subsynchronous component at 2/3 running speed. However, the compressor was able to operate continuously at heads somewhat below design.

Where operation is possible but limited by instability, as was the case with this compressor as well as others, an area of the performance curve between the surge line and the design point and extending roughly from 80 percent speed to 100 percent speed was observed to be particularly sensitive to instability. As a general rule, but certainly not in all cases, operation was feasible in the section well to the right of this area. This permitted the plant to operate but at much lower head than always needed. An upset on another compressor in

the same train would often force the unit to operate in or near the sensitive area and result in an instability trip.

21 March 1979: Continued high vibration of the unit 6 rotor in the unit 7 compressor required a shutdown to remove, rebalance, and reinstall it. On starting up after shutdown with instruments in place, the machine tripped at a running speed of 13 000 rpm. Once again instability was present, and an unbounded component at 2/3 running speed was observed.

9 April to 14 May 1979: During this period four rotor changes were made in the unit 7 compressor in an effort to obtain some level of acceptable operation. Numerous tripouts were experienced, many of them as a result of coupling unbalance. However, at the end of this period unit 7 was still exhibiting instability and was unable to operate for any length of time, and then only at reduced head.

Figure 3 is a plot of speed versus vibration for the unit 7 inboard bearing probe. This plot shows an instability tripout that reached 4 mils.

Mid-year 1979 summary: By this time it was recognized that we had a serious instability problem with no solution in sight. With regard to the six identical units the following conditions existed at this time:

(1) Units 1 and 3 never experienced any difficulties with instability, and a check with a frequency analyzer showed only a very small pip at 2/3 running speed. These compressors continued to run satisfactorily the entire time.

(2) Units 2 and 6 had some instability at all times. For the most part these units had operated satisfactorily but at a reduced head for the preceding 6 months.

(3) The instability experienced on unit 8 has prevented this unit from operating since its commissioning on 5 February 1979. Unit 7 had operated for only brief periods since the beginning of its serious problems in November 1978, which were later identified as instability.

(4) The manufacturer had been actively involved in arriving at a satisfactory solution and had undertaken the following:

(a) Performed a complete rotor stability analysis

(b) Arranged for a consultant to study and conduct field tests to determine if a problem existed external to the compressors. No problem external to the compressors was identified.

(c) Collected and analyzed a large accumulation of data from various field tests to determine a solution. This included several bearing and seal configurations and even impeller changes. Each of these changes required a run to determine results.

(d) Made an exhaustive study to determine any differences between the six units which would explain why some units did not experience instability whereas others did. Nothing of any significance was determined.

13 July 1979: A complete new rotor had been manufactured and was installed in unit 8. There were no changes in this rotor over previous ones; however, because of the large number of tripouts, rubs, and foreign object damage experienced on Units 7 and 8, it was believed prudent to obtain a new one. The initial run on this rotor was very similar to previous runs, with a strong instability indicated at 2/3 running speed. Thus it was necessary to operate at low speed. After having operated at low speed for several hours an instability tripout occurred. Upon restarting unit 8, a major lessening of instability was detected; and it was possible, at least for a period, to operate unit 8 at full speed and load. The machine was dismantled and completely inspected and careful measurements made of all clearances and fits to try to determine what change had occurred during the tripout which could account for the improvement, but no reasons could be found.

The compressor was rebuilt and started up once again. At first it operated satisfactorily, but its performance gradually deteriorated and it suffered a number of instability tripouts during August. Finally, it was observed to be in much the same condition as on previous runs.

18 September 1979: The manufacturer fabricated a new bundle for unit 8 with two major changes. The shrink fit of impellers was reduced in areas by shortening the length of the fit. To do this it was necessary to key impellers to the shaft. The diaphragm wall was extended into the area between the backs of the two impellers and gas was introduced by two holes to induce a laminar flow along the back plates. Although there were several high-vibration tripouts upon startup, enough data were collected to indicate a change in instability frequency in that instability no longer tracked running speed but remained at one frequency regardless of running speed. The instability frequency appeared to be locking on the critical frequency of the shaft. Figure 4 is a plot of vibration for the outboard vertical probe made just before a tripout. Note that instability is not tracking running-speed frequency. Figure 5 is a plot for the inboard vertical bearing probe at tripout. Tripout amplitude was 8.2 mils at 8300 cycles per minute.

The bundle was rebuilt with the same diaphragm arrangement as previously but by reverting to the original rotor design by eliminating the keyed impeller and the reduced shrink fit. The combination proved successful and on Oct. 10, 1979, the unit was placed on the line with the instability completely bounded. Even after a violent surge the instability remained bounded. The compressor continues to run to date in this manner with no instability difficulties. Figure 6 is a plot of the inboard and outboard vertical and horizontal probe vibration representing unit 8 as finally modified. Maximum instability is only 0.1 mil.

Units 2 and 7 have been fitted with the new diaphragm arrangement. Unit 7 has operated satisfactorily without instability since this change. However, unit 2 still has had some vibration tripouts but at present is operating satis-

factorily. Some instability appears to be present, but it remains bounded. Unit 6 will be modified as soon as plant schedules permit.

From our standpoint it would appear that the combination of the new diaphragm arrangement, whereby laminar flow was induced on the impeller back plates, along with better than normal rotor balance results in successful operation with the 2/3-running-speed component remaining bounded. The plant is now operating smoothly, although you can understand that after having gone through the experiences of the past 18 months everyone is not convinced that similar problems will not recur. Only successful operation over a long period will provide the final proof.

FIELD SOLUTION OF INSTABILITY PROBLEMS

My purpose in leading you through this rather detailed description of a major instability problem is to point out forcefully to all concerned - designers, manufacturers, consultants, contractors, and users alike - the very serious consequences of an instability problem.

Field solution of instability problems may be characterized as follows:

(1) Difficulties are encountered and time is lost in properly identifying the problem. A clearer understanding of the phenomenon surrounding instability by users, along with more widespread use of diagnostic instruments, will undoubtedly help this situation. I strongly advocate users owning or at least having ready access to such equipment along with trained personnel to operate and interpret results. Signature analysis and good records of operation and maintenance are also very important.

(2) Ineffective methods are used and excessive time is required in determining the cause of instability. Far too much time is consumed in making minor changes and operating the compressors to determine results. Endless combinations of bearing designs, preloads, etc., along with seal configurations, clearances, grooves, etc., can require weeks if not months to check out. Bear in mind that this is even before we have encountered the heavy-artillery-like effects of hysteresis at the impeller shaft mating surface, new diaphragms, larger diameter shafts, reducing bearing span, squeeze film bearings, and others. Add to this the time required to determine if the cause of the instability is external or internal to the compressor and you may have encompassed a very long costly period of time. More effective means of making analytical determinations of the results of various modifications is needed in order to reduce the time required to obtain a satisfactory solution.

Finally, I believe there is a great need for better methods of identifying potential instability problems in the design stage. Maybe closer cooperation between all parties will eventually lead to this. Conferences such as this will help. Better specifications by users and contractors in defining all aspects of service in which the compressor will be used will also help. Wider use of rotordynamic analysis by designers will be an aid. I am certain that many improvements have been made, but there is need for many more. I hope that such improvements will be forthcoming because the need is great and the potential penalty very high.

(1) The Phillips Norway Group consists of the following companies:

Phillips Petroleum Company Norway (Operator)	36.960%
American Petrofina Exploration Company Norway	30.000%
Norsk Agip A/S	13.040%
Elf Aquitaine Norge A/S	8.094%
Norsk Hydro A/S	6.700%
Total Marine Norsk A/S	4.047%
Eurafrep Norge A/S	0.456%
Coparex Norge A/S	0.399%
Cofranord A/S	0.304%

(2) The Hewett Group consists of the following companies:

Phillips Petroleum Company (Operator)	18.97%
American Petrofina Exploration Company	16.26%
Agip (UK) Ltd.	8.13%
Century Power & Light Ltd.	3.91324%
Plascom Ltd.	2.30892%
Halkyn District United Mines Ltd.	2.30892%
Oil Exploration Ltd.	2.30892%
ARCO British Ltd.	4.58%
ARCO Oil Producing Inc.	15.26667%
Canadian Superior Oil (UK) Ltd.	1.52666%
North Sea Exploration & Research Co. Ltd.	4.58%
North Sea Sun Oil Co. Ltd.	10.68667%
Superior Overseas Development Co. Ltd.	9.16%

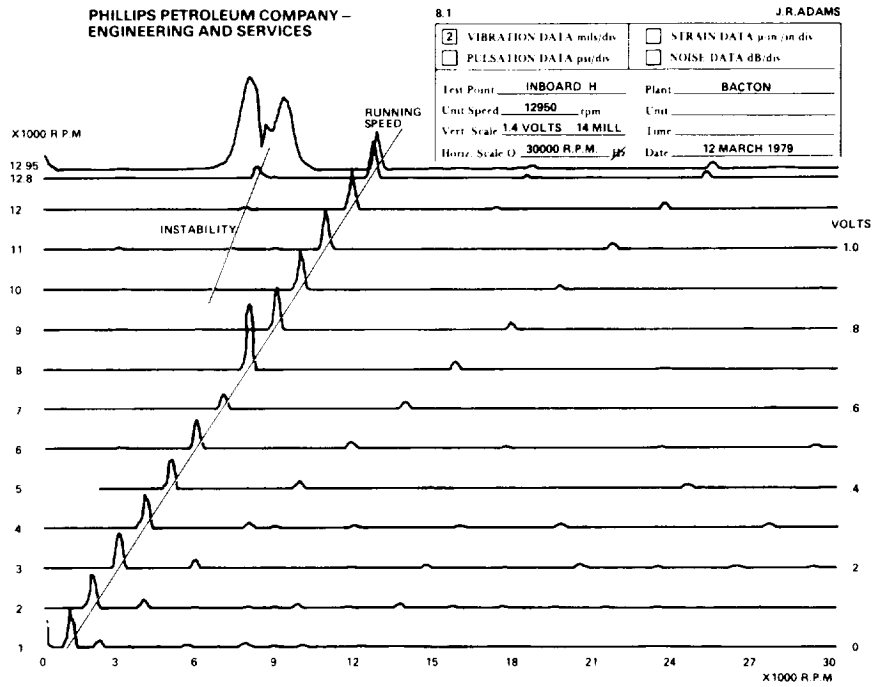


FIGURE 1

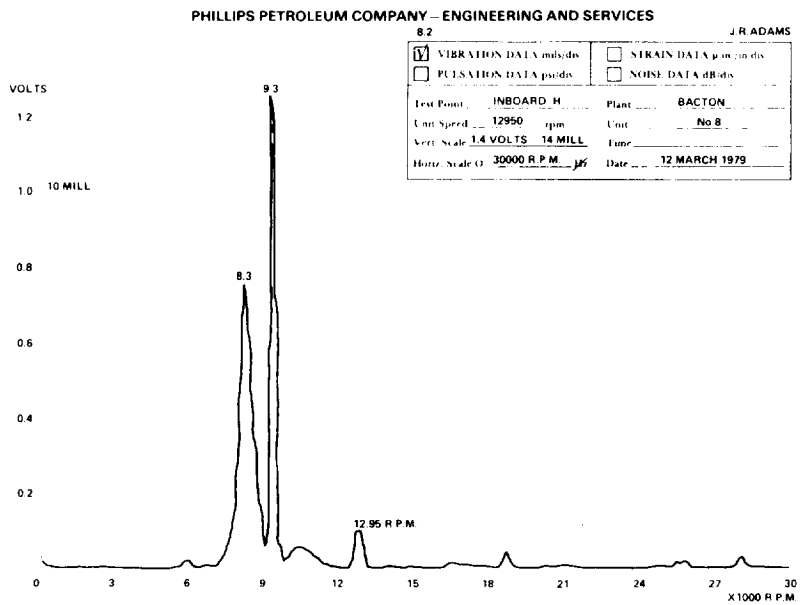


FIGURE 2

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<input checked="" type="checkbox"/> VIBRATION DATA mils/div	<input type="checkbox"/> STRAIN DATA μ m/in/div
<input type="checkbox"/> PULSATION DATA psid/div	<input type="checkbox"/> NOISE DATA dB/div
Test Point: INBOARD	Plant: BACTON
Unit Speed: 12960 rpm	Unit: No. 7
Vert. Scale: _____	Time: _____
Horiz. Scale (): 30000 R.P.M. μ s	Date: 10 APRIL 1979

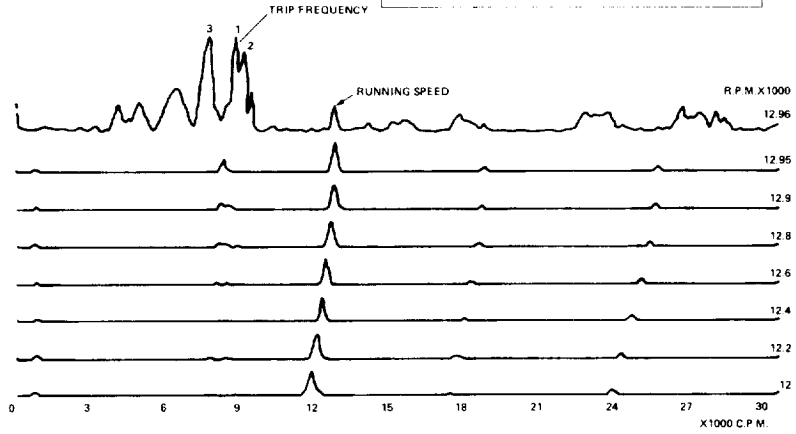


FIGURE 3

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<input type="checkbox"/> PULSATION DATA psid/div	<input type="checkbox"/> NOISE DATA dB/div
Test Point: O.B.V.	Plant: BACTON
Unit Speed: _____ rpm	Unit: _____
Vert. Scale: 100 MV/MILL	Time: _____
Horiz. Scale (): 30000 R.P.M. μ s	Date: 18 SEP 1979

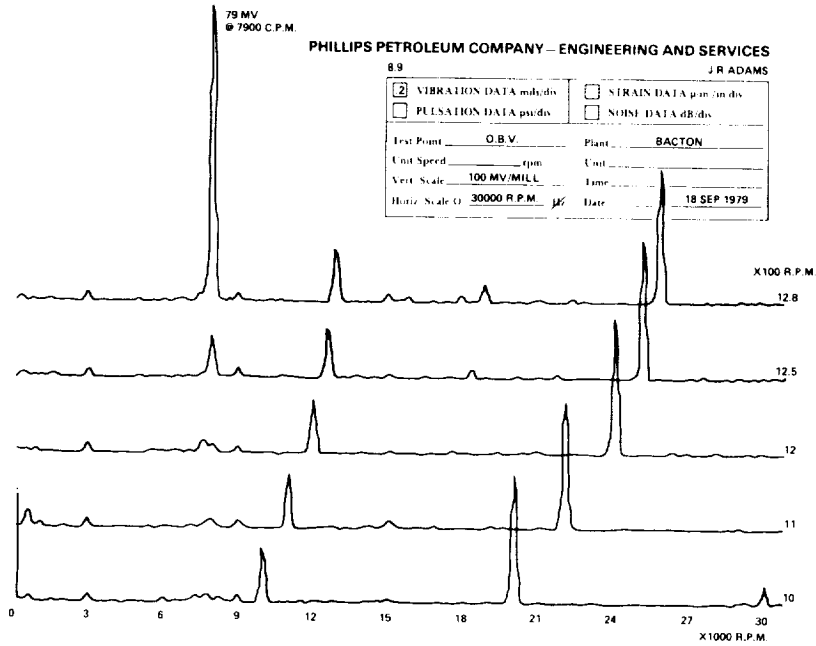


FIGURE 4

8 10 J.R. ADAMS

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<input type="checkbox"/> PULSATION DATA psid/s	<input type="checkbox"/> NOISE DATA dB/d/s
Test Point: <u>1 B.V.</u>	Plant: <u>BACTON</u>
Unit Speed: <u>12900</u> rpm	Unit: _____
Vert. Scale: <u>1" = 2 MILL</u>	Time: <u>1231 HRS</u>
Horiz. Scale: <u>30000 R.P.M.</u>	Date: <u>18 SEP 1979</u>

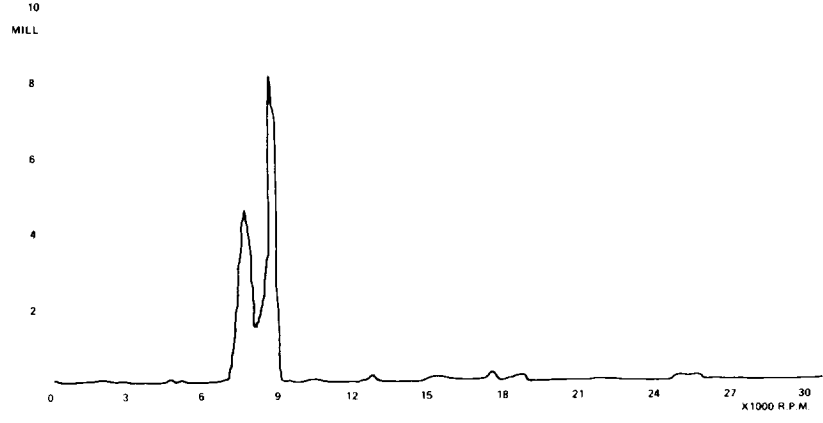


FIGURE 5

8 19 J.R. ADAMS

<input checked="" type="checkbox"/> VIBRATION DATA mils/dis	<input type="checkbox"/> STRAIN DATA μ m/in/dis
<input type="checkbox"/> PULSATION DATA psid/s	<input type="checkbox"/> NOISE DATA dB/d/s
Test Point: _____	Plant: <u>BACTON</u>
Unit Speed: <u>14000</u> rpm	Unit: _____
Vert. Scale: <u>1" = 0.2 MILL</u>	Time: <u>1056 HRS</u>
Horiz. Scale: <u>30000 C.P.M.</u>	Date: <u>18 OCT 1979</u>

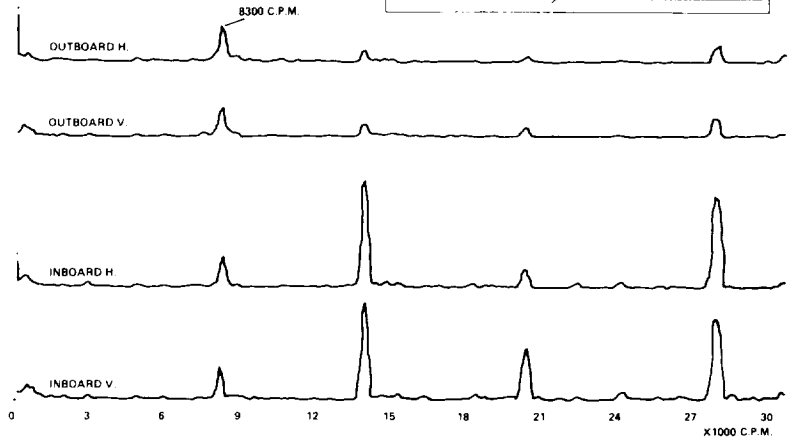


FIGURE 6

