THE INFLUENCE OF TILTING PAD BEARING CLEARANCE ON ROTOR RESPONSE OF A STEAM TURBINE

by

Brian C. Howes and Donald D. Leung

Abstract:

Field measurements are presented from a steam turbine showing the response during run-up and coast-down. In one case, a bearing was wiped due to prolonged operation close to the critical speed. In another case, the unit was run-up without damaging the bearing.

A computerized data acquisition system was used to display the data on site. Many actual plots of vibration data are presented, including bode, Nyquist, waterfall spectra, orbit cascade, and speed versus time.

The Authors:

Brian Howes is Manager of Engineering with Beta Machinery Analysis Ltd. in Calgary, Alberta. He is active in field trouble-shooting of rotating and reciprocating equipment problems including field balancing and has been with Beta Machinery Analysis Ltd. for two years. Mr. Leung has a B.Sc. in mechanical engineering from the University of Calgary.

INTRODUCTION:

This paper presents a case history of the problems that occurred while attempting to balance a steam turbine using an influence coefficient balancing method.

The authors were called in to balance the steam turbine when its owners became concerned with high vibration levels on the “compressor coupling” end of the topping steam turbine and the turbine’s varying vibration levels while under constant load.

Prior to our arrival the tilting pad bearing at the compressor end of the turbine had been replaced as the old bearing had been damaged. Examination of the old bearing showed that the rotor had contacted the bearing pads at several distinct locations.

MACHINE DESCRIPTION:

The machine train consisted of tandem steam turbines directly coupled to the topping steam turbine which drives two syn gas compressors (high pressure and low pressure). See Figure 1 for a schematic sketch of the machine layout.

The topping turbine has its first critical speed at approximately 6,500 RPM and its second critical speed is at approximately 11,500 RPM. The normal operating speed range for this turbine is from 9,800 RPM to 10,300 RPM.

The topping turbine has a history of high sensitivity to changes in balance and alignment which has been attributed to operation near its second critical speed.

Due to time constraints, balancing was to be attempted only on the two most accessible balance planes which were the couplings at either end of the topping turbine.

BALANCING OBJECTIVES:

The objectives of the balancing were:

1. To reduce vibrations for the topping turbine during normal operation.

2. To reduce vibrations when passing through the first critical during run up or coastdown.

TEST EQUIPMENT:

Pairs of eddy-current proximity probes mounted at 90° as shown in Figure 1 were used for shaft motion measurements with a modified axial thrust proximity probe being used for speed and phase reference purposes.

Data were recorded using an eight-channel FM recorder to allow capture of transient phenomena and were played through a two channel tracking filter interfacing with a computer which allowed rapid processing of the recorded data. The results of the data processing were then displayed in various formats on the computer terminal screen and plotted on a digital plotter as required.

PROCEDURE USED AND RESULTS OBTAINED:

When we had arrived on site the unit was stopped as the tilting pad bearing was being replaced. At this time the topping turbine was uncoupled from the compressors and a ‘running ring’ was used to hold the coupling hub stationary on the shaft.

During the initial start-up to determine the ‘as-found’ vibration readings, the operators used their normal starting procedure which caused the train to be held at a speed of approximately 6,800 RPM for several minutes (see Figure 2). A Nyquist plot of the data recorded at probes ZI-27 is presented in Figure 3, the same data in an orbit plot form are shown in Figure 4.

Comparison of the start-up and coastdown Bode plots with the ‘as-found’ imbalance distribution at the same probes (for probe ZI-27 see Figure 5 & 6; for probe ZI-28 see Figure 7 & 8) showed that the vibration behaviour (both amplitude and phase) during start-up differed significantly from that during coastdown.

This observation was repeated for each of the runs required to determine the influence coefficients of weights located at the couplings. A general comment can be made that for each succeeding run it appeared that the first critical was occurring at a lower speed than for the previous run. Abrupt changes in vibration amplitude and phase were often noted.

Using the influence co-efficients determined from the trial runs and the ‘as-found’ vibrations, it was found that if data recorded at a speed below the first critical were included in calculations that the RMS balancing routine used could not converge on a solution that would leave a lower residual vibration than those ‘as-found’. At this point, although we were concerned over the validity of the data recorded, balance weights were calculated using only data recorded above the first critical speed and these weights were installed as required on the couplings.

PROCEDURE USED AND RESULTS OBTAINED (continued)

The results of this balance weight installation were that vibration levels were still quite high and erratic as slight changes in speed dramatically changed vibration (see Figure 9).

Spectral analysis of this start-up showed significant levels of vibration at frequencies other than one times shaft speed (see Figure 10).

At this point it was felt that the same tilting pad bearing had been wiped again and that the accumulating damage was changing bearing stiffness from run to run thus rendering our influence co-efficient balancing method ineffectual. Unfortunately, time did not allow us to redo any of our previous trial runs to determine if there was any repeatability of bearing stiffness.

A decision was made to recouple the entire compressor train and to determine the coupled response of the turbine with the balance weights left in place on the couplings. The turbine bearing nearest the compressor was found to be wiped and was replaced before the coupled unit was started.

During the start-up (after recoupling), the turbine was accelerated slowly through its first critical speed (as can be seen in Figure 11). Also at this time it appeared that the shaft had used up all of the clearance in the X-probe direction at the ‘compressor’ end bearing (see figure 12).

During a shutdown immediately after the initial start-up and another subsequent start-up, significant variation was again noted in vibration amplitude and phase between runs. It was felt that the bearing was wiped but as the turbine initially had acceptable vibration levels while under load in the normal operating speed range, it was decided to keep the compressor train on line.

After two more unmonitored stop-start cycles, the turbine started to exhibit extreme vibration instability as slight changes in run speed or load produced large changes in vibration. At this time the unit was shut down and the bearing nearest the compressor was again found to be damaged and was replaced.

At this time two changes were made to the start-up procedure:

1) The unit was slow rolled at approximately 600 RPM to allow any gravity and/or thermal bow to work itself out. This slow roll procedure was continued until vibration levels had stabilized which took approximately thirty minutes.

2) After the slow roll vibration had stabilized, the turbine was slowly accelerated to 5,200 RPM. After insuring sufficient steam pressure, the turbine was accelerated as rapidly as possible to its normal operating speed range so as to minimize the time spent near the first critical (see Figure 13).

The results of this start-up were the lowest vibration levels noted during the testing and good vibration stability under load throughout the normal operating speed range (see Figure 14). Monitoring after fourteen hours of continuous operation under load showed that the vibration levels (amplitude and phase) were almost identical to those recorded immediately after the load was initially applied. This was true over the normal operating speed range.

At this point all vibration levels were considered acceptable and no further attempt was made to reduce these levels.

OBSERVATIONS:

1. The determination of influence coefficients requires the assumption of linearity and repeatability. The progressive damage of the bearing from run to run made this assumption wrong. It appears that even the ‘as found’ data was recorded with a damaged bearing, thus no meaningful comparison can be made between ‘as-found’ and ‘as-left’ vibration levels.

   Time did not permit re-running any balance tests to determine the degree of error in the measured influence coefficients.

2. Instability of vibration with the turbine operating under ‘constant’ load conditions tended to be symptomatic of a damaged bearing.

   The original problem of high vibrations and instability related to load was apparently not primarily due to imbalance.

3. The acceleration past a critical speed must be sufficiently high so that the amplitude of shaft motion at the critical speed is not allowed to fully develop.

CONCLUSIONS:

The balancing of high speed rotating equipment has been shown to depend on the repeatability of the experimental data. This conclusion is based on the case history reported here, as well as other balancing experiences.

Consequently, it is important to have methods to confirm the data repeatability (and linearity).

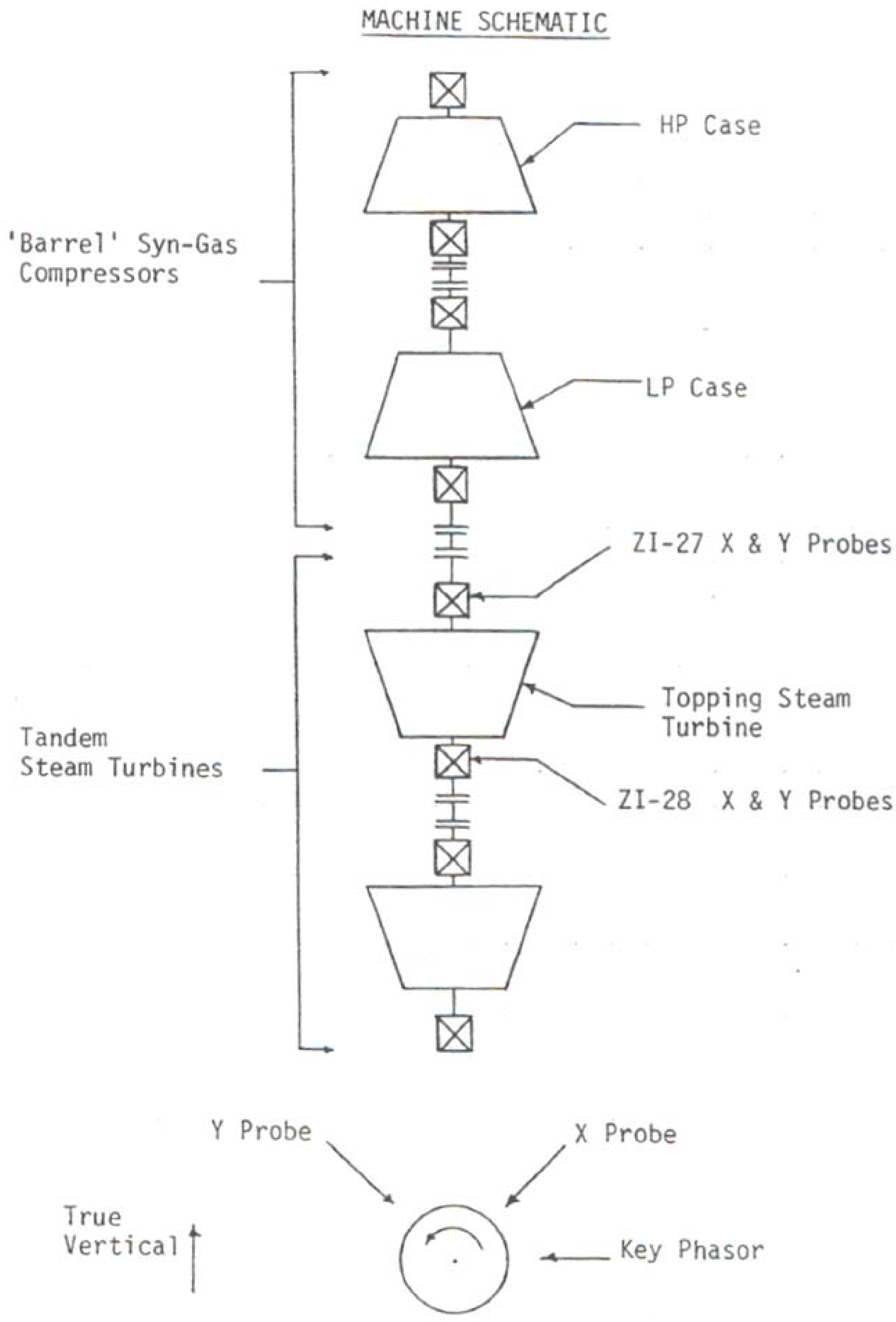
The most obvious method is to repeat the “as found” run after the trial weight runs, or to repeat a trial run (with the trial weight moved 180°, perhaps). The disadvantage of this approach is that it adds time to an already time-consuming analysis.

A second method involves careful data collection, presentation and interpretation. These data should include amplitude and phase plots versus speed for the run-up and coast-down on each test. Although the imbalance will intentionally be different from run to run, the location of the critical speeds and amplification factors should remain fixed. This approach will help to detect changes such as the bearing stiffness discussed. Other causes of “non-repeatability” are not necessarily so easy to detect.

No matter what techniques are used to confirm that the collected data are “good”, it is important to recognize that balancing is not a straight forward “cookbook” procedure.

Balancing should always be preceded by the question “Why are we balancing this machine?”.

FIGURE: 1



'Probe Orentiation'

FIGURE: 2

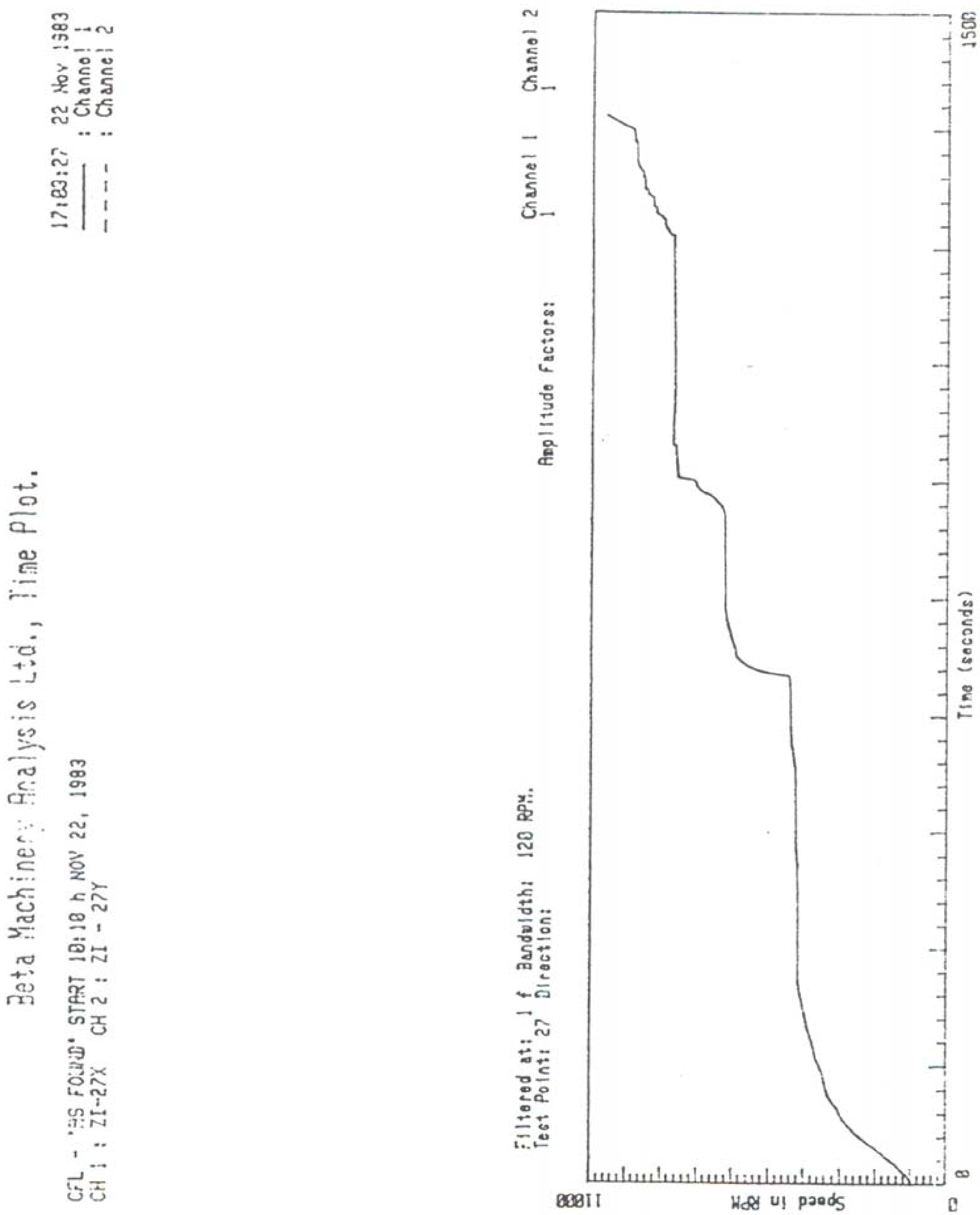

Beta Machinery Analysis Ltd., Time Plot.
CFL - 'AS FOUND' START 10:10 h NOV 22, 1983
CH 1 : ZI-27X   CH 2 : ZI - 27Y
17:03:27 22 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 120 RPM.
Test Point: 27 Direction:
Amplitude Factors:
Channel 1
1
Channel 2
1
11000
Speed in RPM
0
0
Time (seconds)
1500

FIGURE: 3

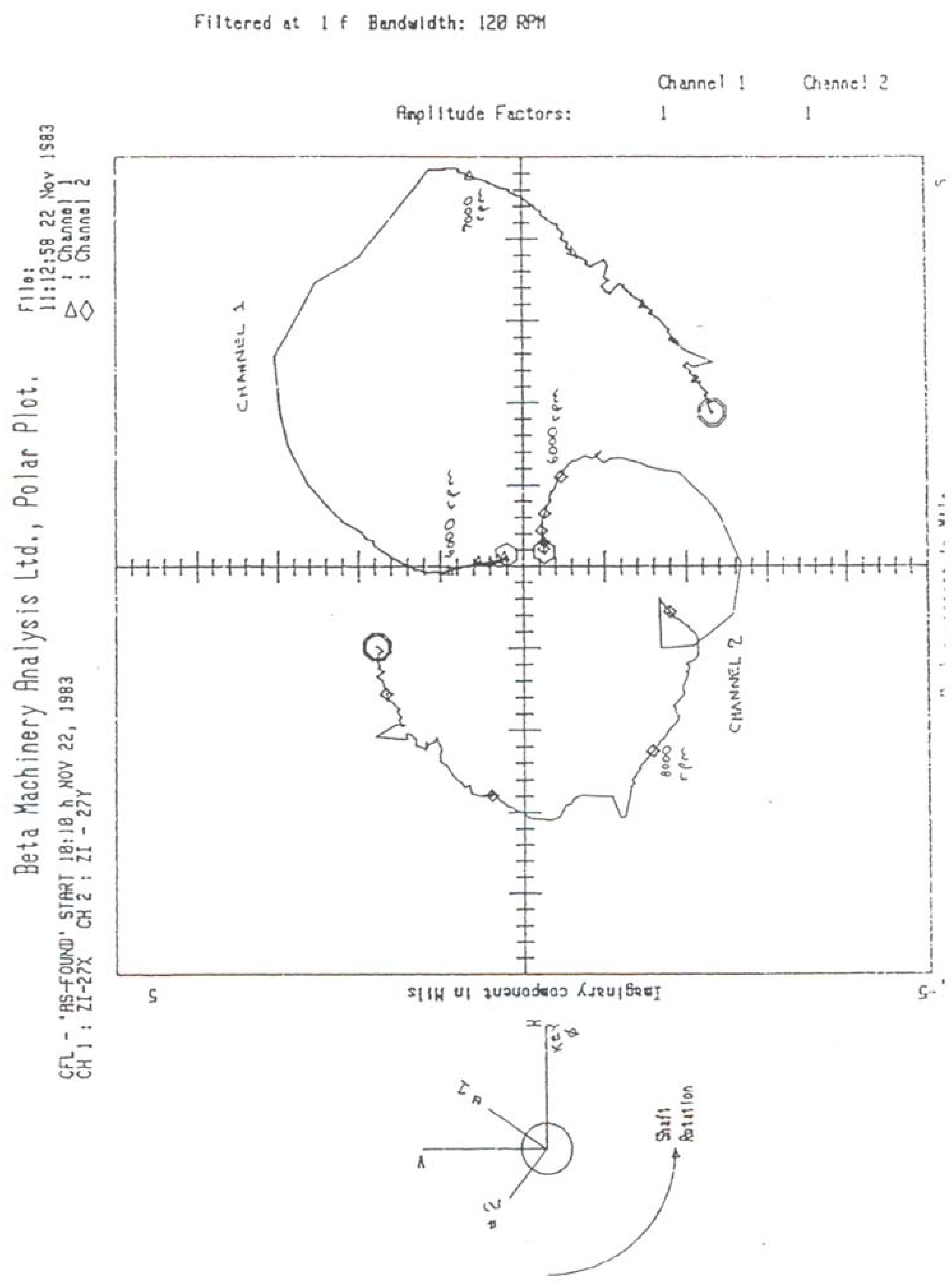

Filtered at 1 f Bandwidth: 120 RPM
Channel 1
Channel 2
Amplitude Factors: 1 1
Beta Machinery Analysis Ltd., Polar Plot.
File:
11:12:58 22 Nov 1983
△ : Channel 1
◇ : Channel 2
CFL - 'AS-FOUND' START 10:10 h NOV 22, 1983
CH 1 : ZI-27X CH 2 : ZI - 27Y
CHANNEL 1
CHANNEL 2
7000 rpm
6000 rpm
8000 rpm
Imaginary component in Mils
Shaft Rotation

FIGURE: 4

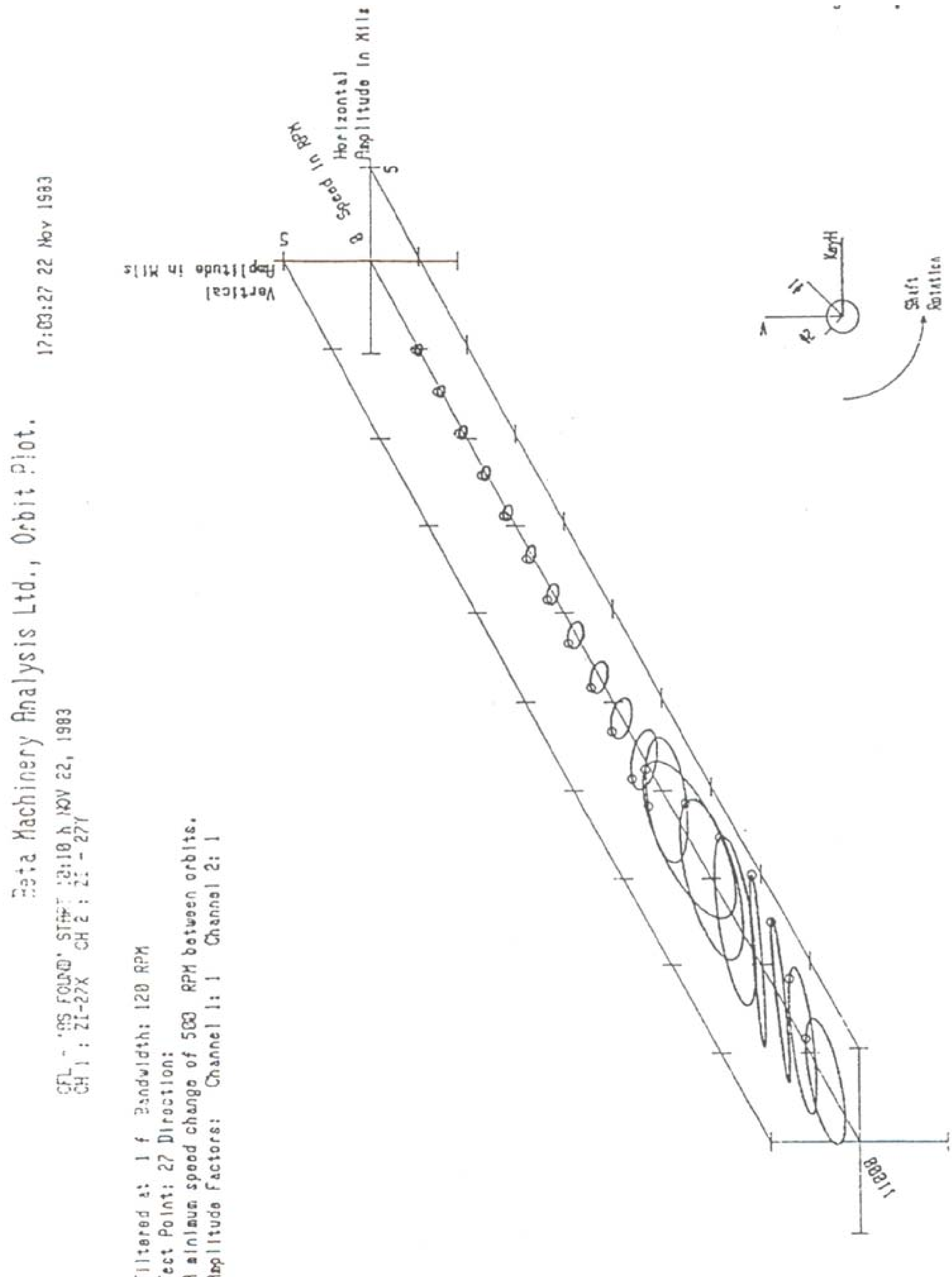

Beta Machinery Analysis Ltd., Orbit Plot.
17:03:27 22 Nov 1983
Filtered at: 1 f Bandwidth: 120 RPM
Test Point: 27 Direction:
A minimum speed change of 500 RPM between orbits.
Amplitude Factors: Channel 1: 1 Channel 2: 1
Vertical Amplitude in Mils
Horizontal Amplitude in Mils
Speed in RPM
Shaft Rotation

FIGURE: 5

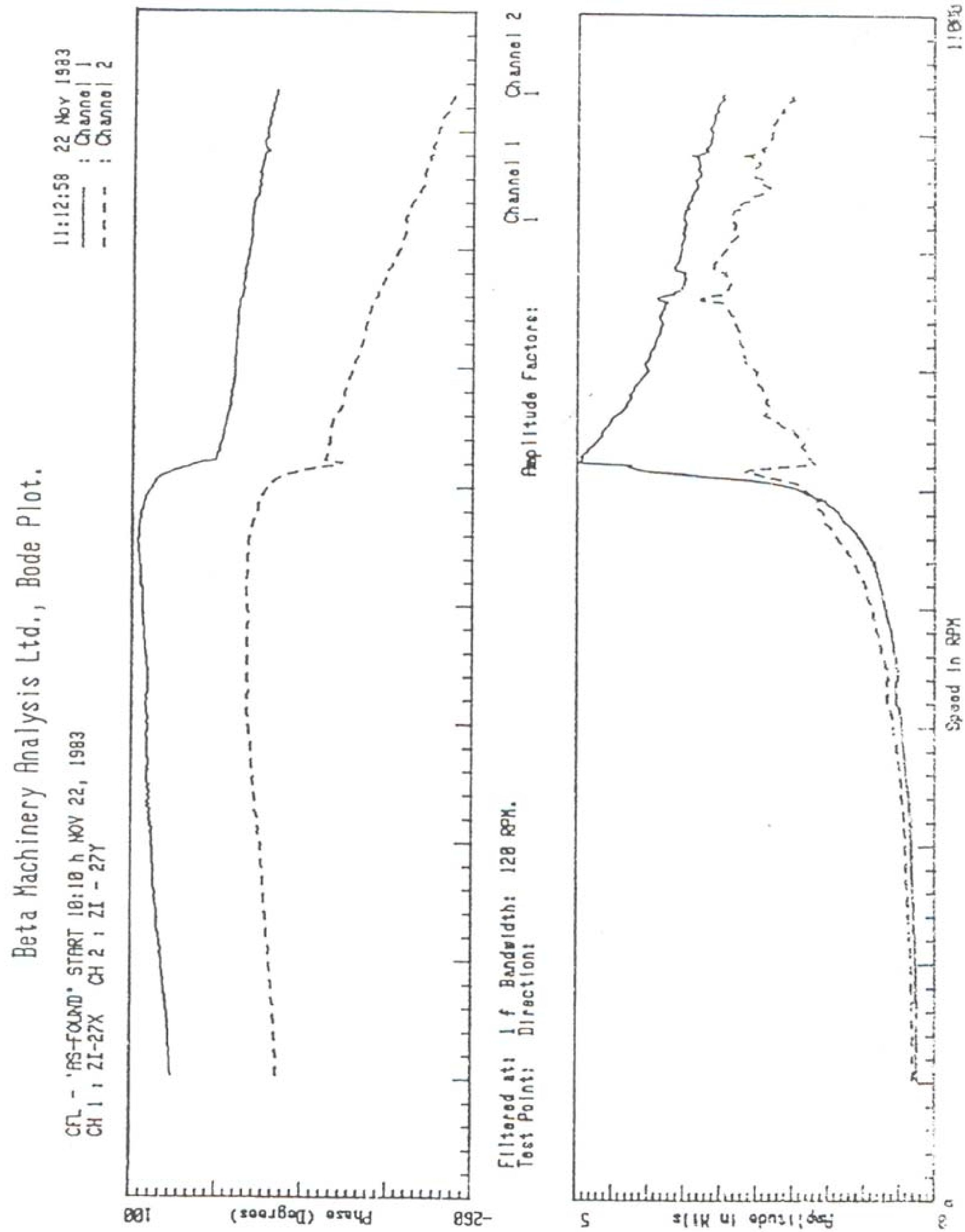

Beta Machinery Analysis Ltd., Bode Plot.
CFL - 'AS-FOUND' START 10:10 h NOV 22, 1983
CH 1 : ZI-27X CH 2 : ZI - 27Y
11:12:58 22 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 120 RPM.
Test Point: Direction:
Amplitude Factors: Channel 1 Channel 2
Phase (Degrees)
Amplitude in Mils
Speed In RPM

FIGURE: 6

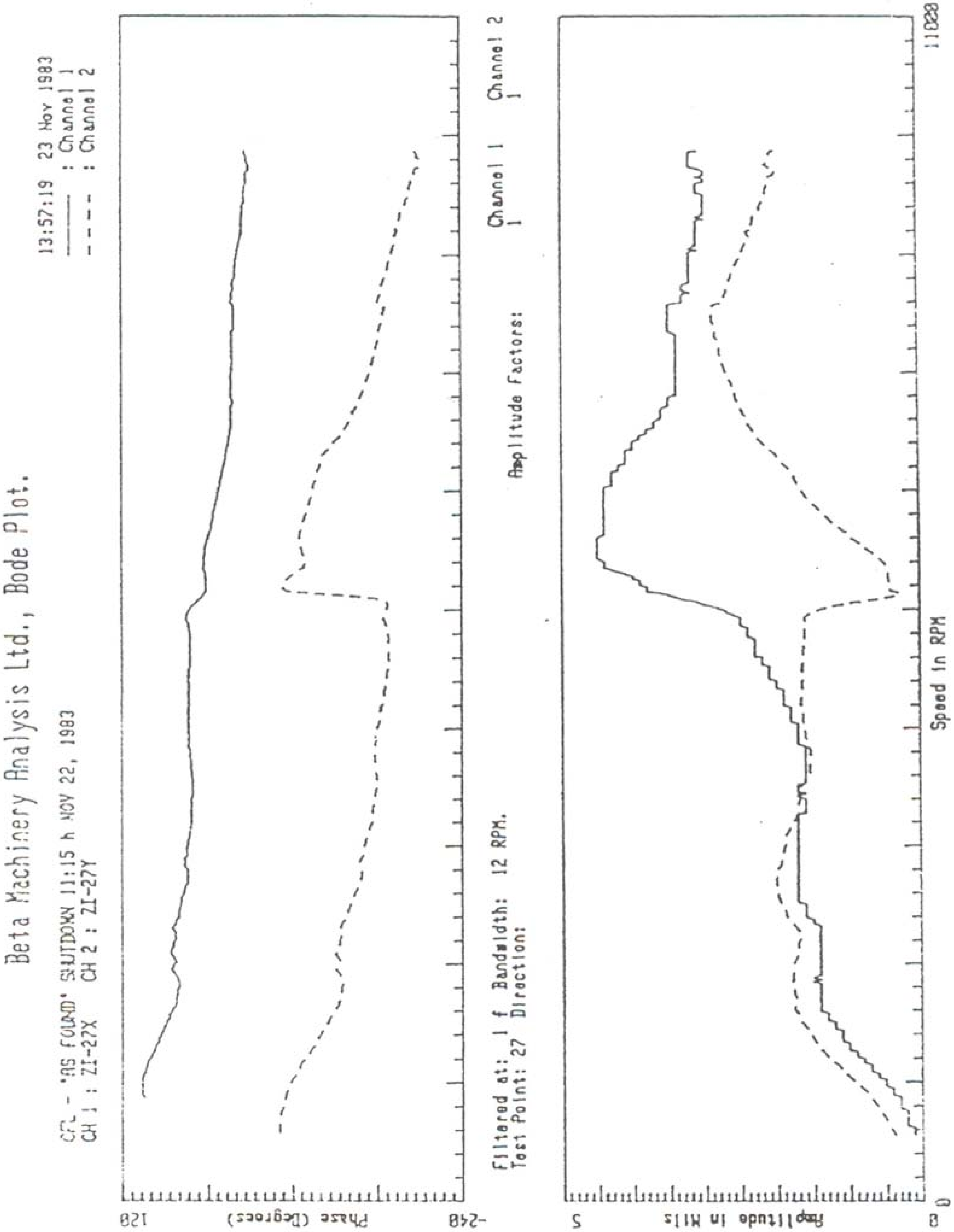

Beta Machinery Analysis Ltd., Bode Plot.
CPL - 'AS FOUND' SHUTDOWN 11:15 h NOV 22, 1983
CH 1 : ZI-27X CH 2 : ZI-27Y
13:57:19 23 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 12 RPM.
Test Point: 27 Direction:
Amplitude Factors:
Channel 1
1
Channel 2
1
128
Phase (Degrees)
-248
5
Amplitude in Mils
0
0
Speed in RPM

FIGURE: 7

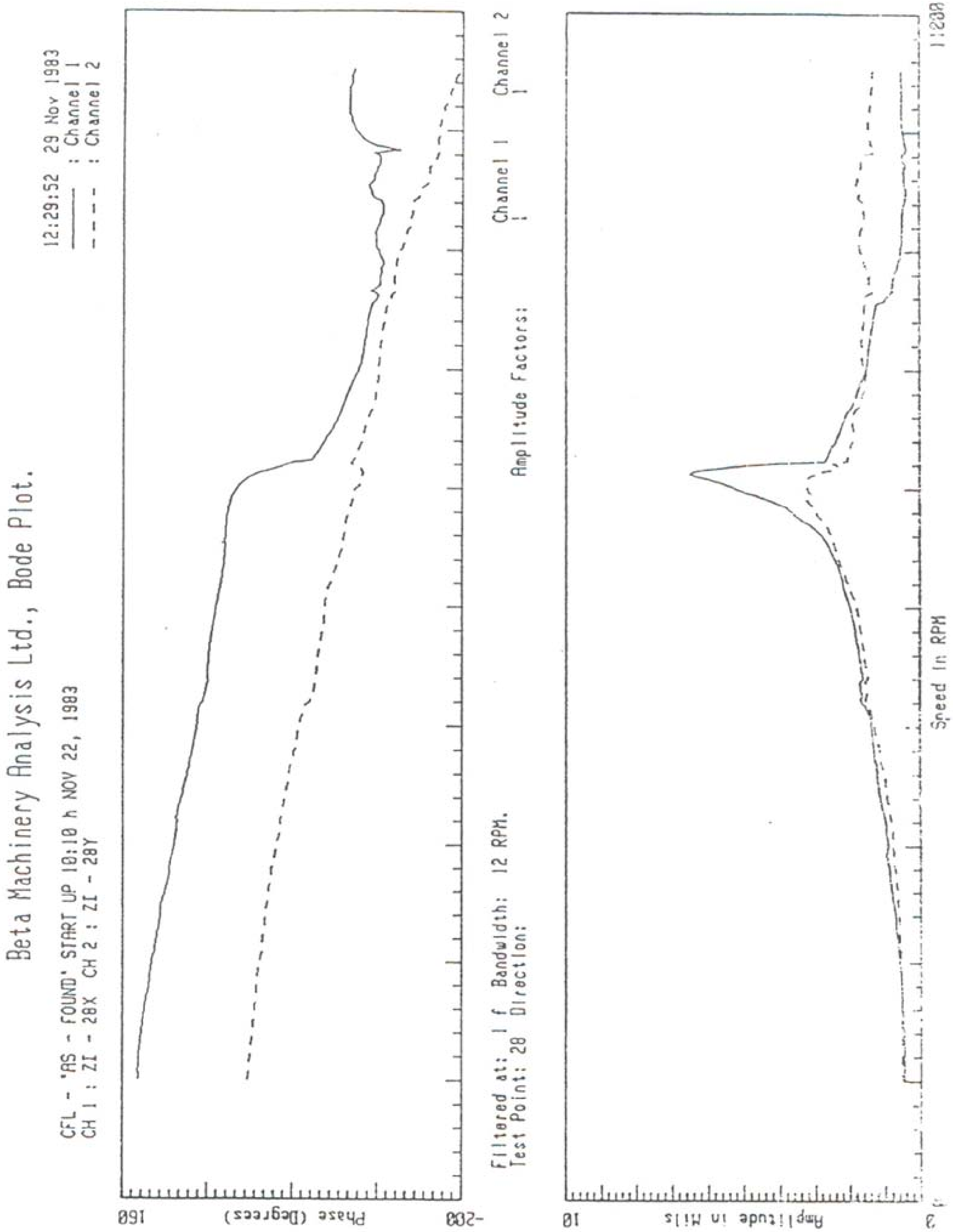

Beta Machinery Analysis Ltd., Bode Plot.
CFL - 'AS - FOUND' START UP 10:10 h NOV 22, 1983
CH 1 : ZI - 28X CH 2 : ZI - 28Y
12:29:52 29 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f. Bandwidth: 12 RPM.
Test Point: 28 Direction:
Amplitude Factors:
Channel 1
Channel 2
160
Phase (Degrees)
-200
10
Amplitude in Mils
Speed in RPM
1200

FIGURE: 8

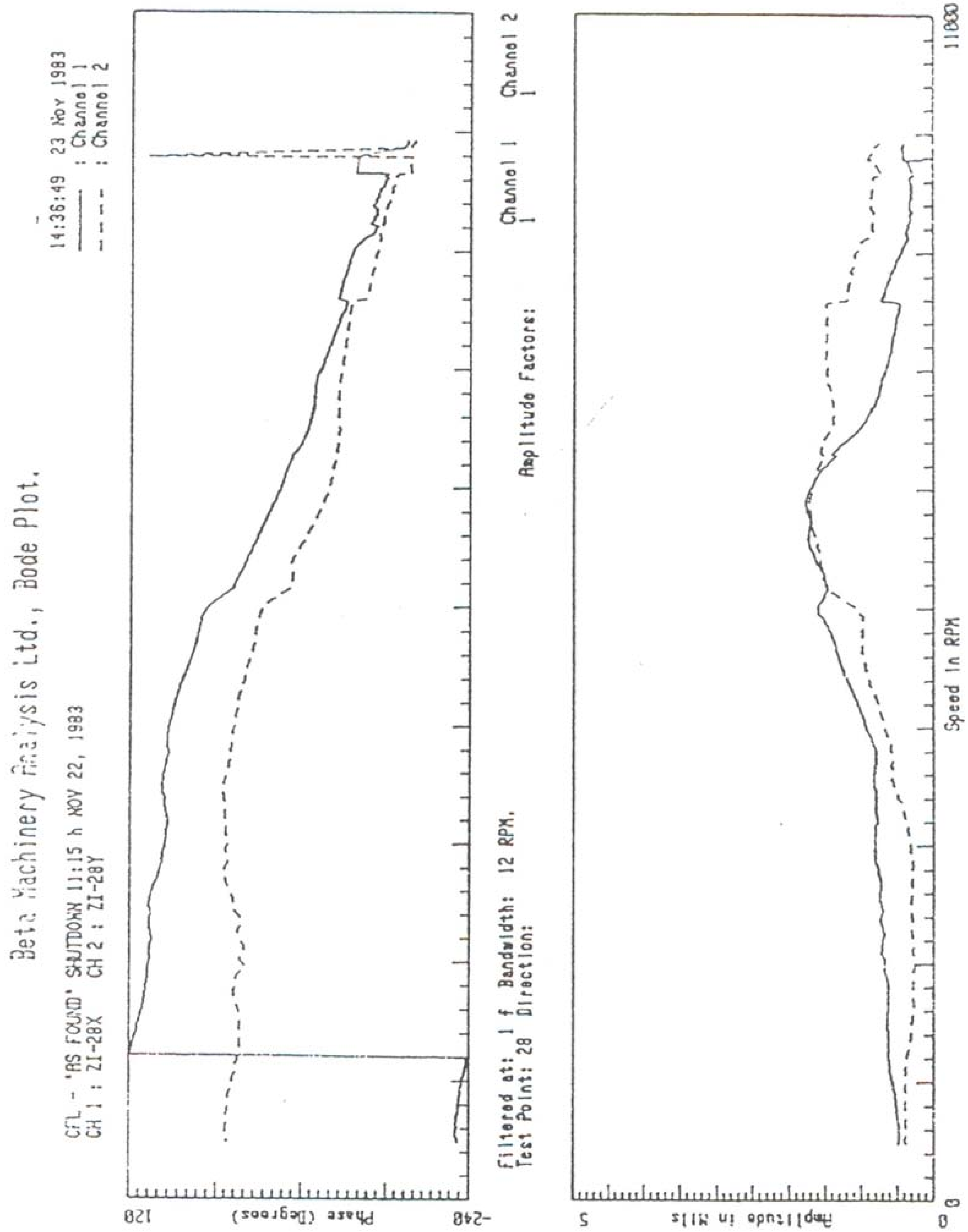

Bode Plot.
CFL - 'AS FOUND' SHUTDOWN 11:15 h NOV 22, 1983
CH 1 : ZI-28X
CH 2 : ZI-28Y
14:36:49
23 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f
Bandwidth: 12 RPM.
Test Point: 28
Direction:
Amplitude Factors:
Channel 1
1
Channel 2
1
120
Phase (Degrees)
-240
5
Amplitude in Mils
0
0
11000
Speed In RPM

FIGURE: 9

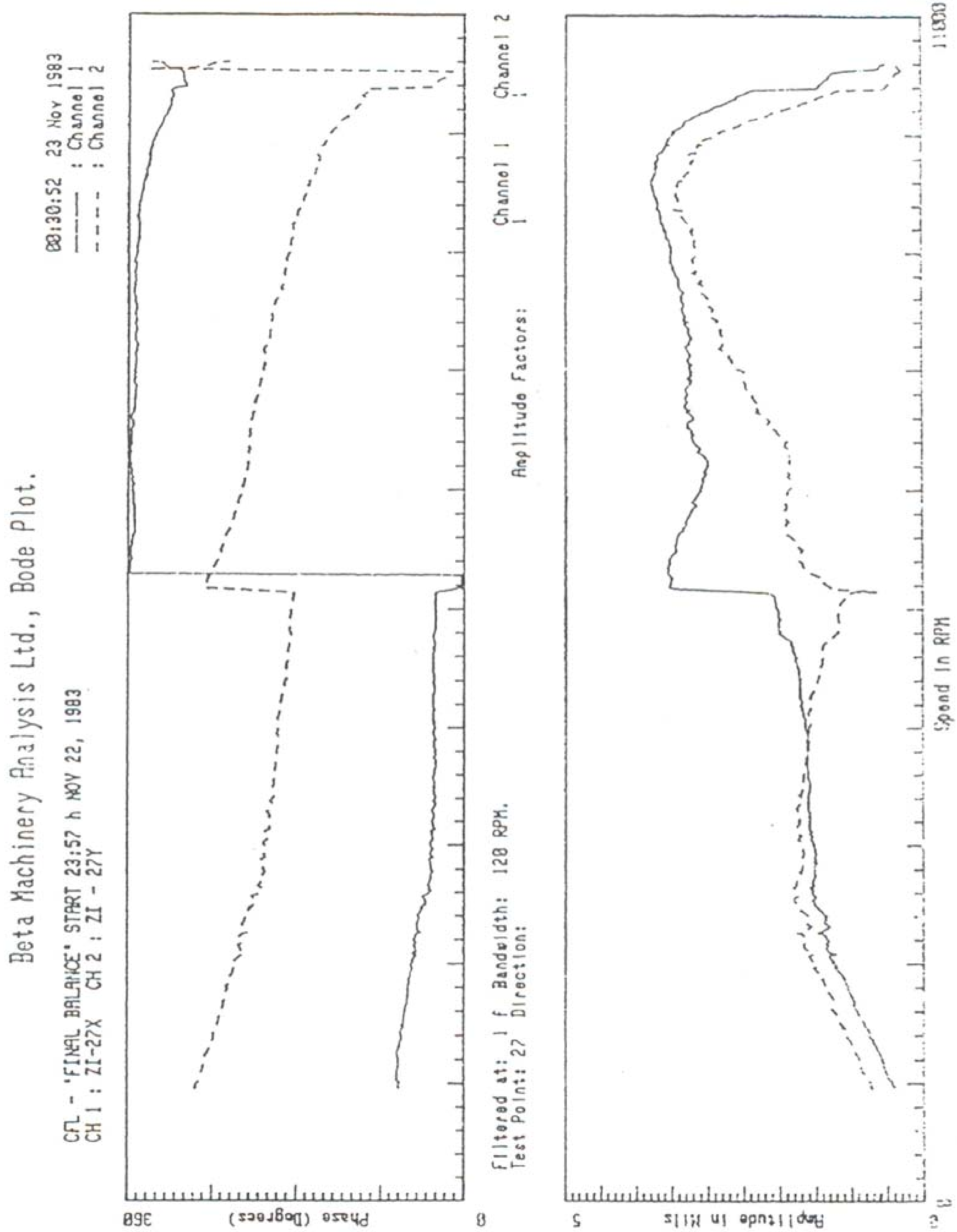

Beta Machinery Analysis Ltd., Bode Plot.
CAL - 'FINAL BALANCE' START 23:57 h NOV 22, 1983
CH 1 : ZI-27X CH 2 : ZI - 27Y
00:30:52 23 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 120 RPM.
Test Point: 27 Direction:
Amplitude Factors:
Channel 1
Channel 2
360
Phase (Degrees)
0
5
Amplitude in Mils
0
Speed In RPM

FIGURE: 10

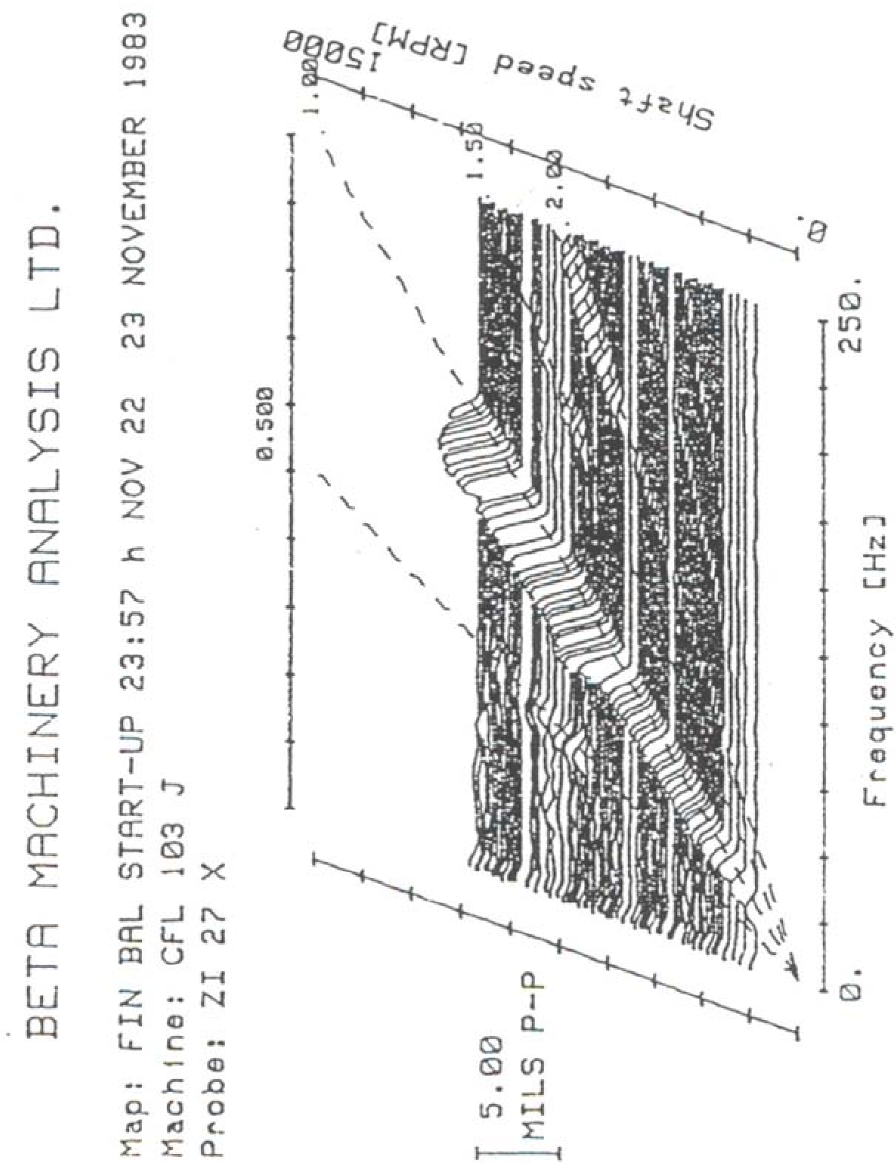

BETA MACHINERY ANALYSIS LTD.
Map: FIN BAL START-UP 23:57 h NOV 22 23 NOVEMBER 1983
Machine: CFL 103 J
Probe: ZI 27 X
5.00
MILS P-P
0.500
Shaft speed [RPM]
15000
1.00
1.50
2.00
0
Frequency [Hz]
0.
250.

FIGURE: 11

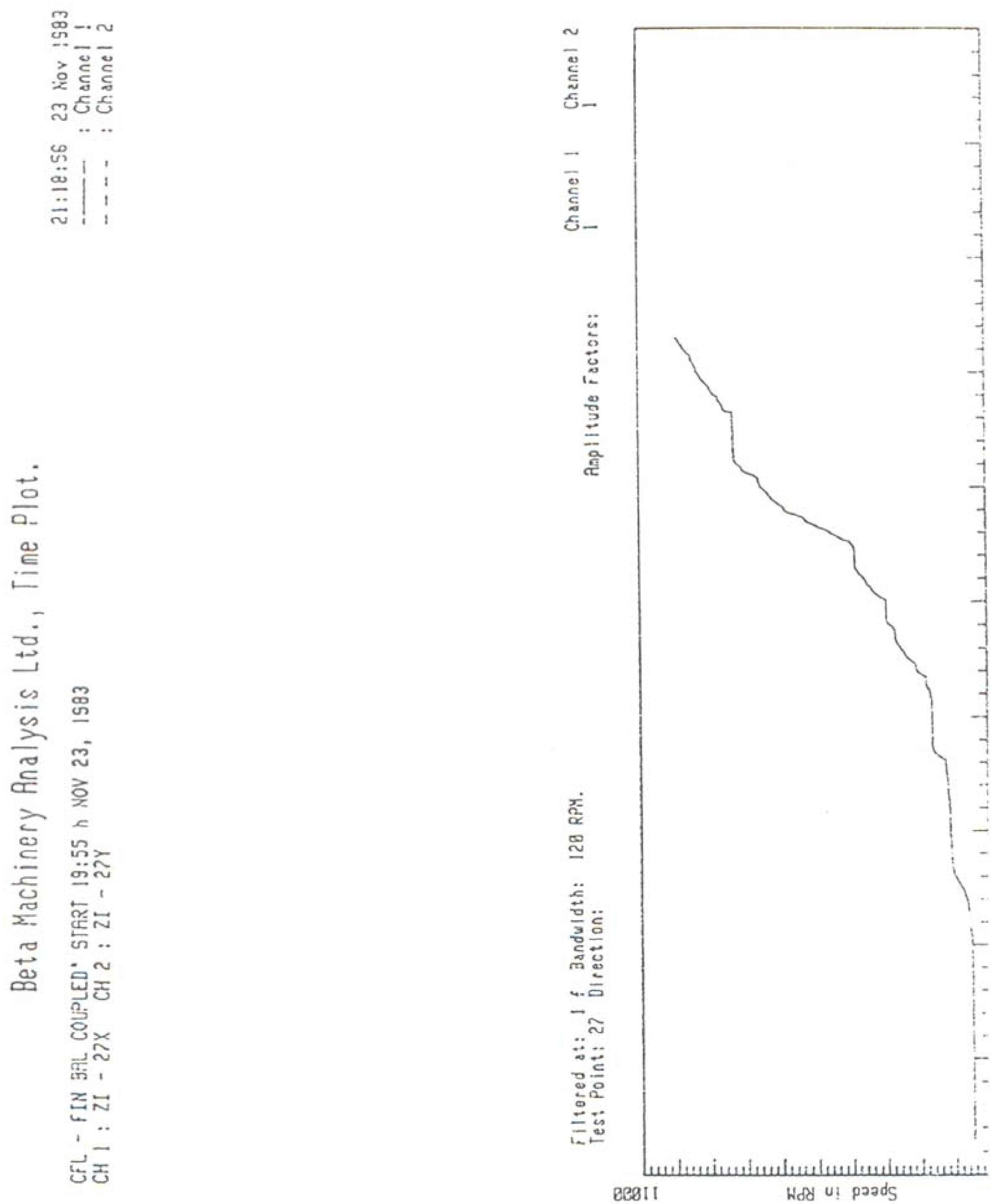

Beta Machinery Analysis Ltd., Time Plot.
CFL - FIN BAL COUPLED' START 19:55 h NOV 23, 1983
CH 1 : ZI - 27X CH 2 : ZI - 27Y
21:10:56 23 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 120 RPM.
Test Point: 27 Direction:
Amplitude Factors:
Channel 1
1
Channel 2
1
11000
Speed in RPM

FIGURE: 12

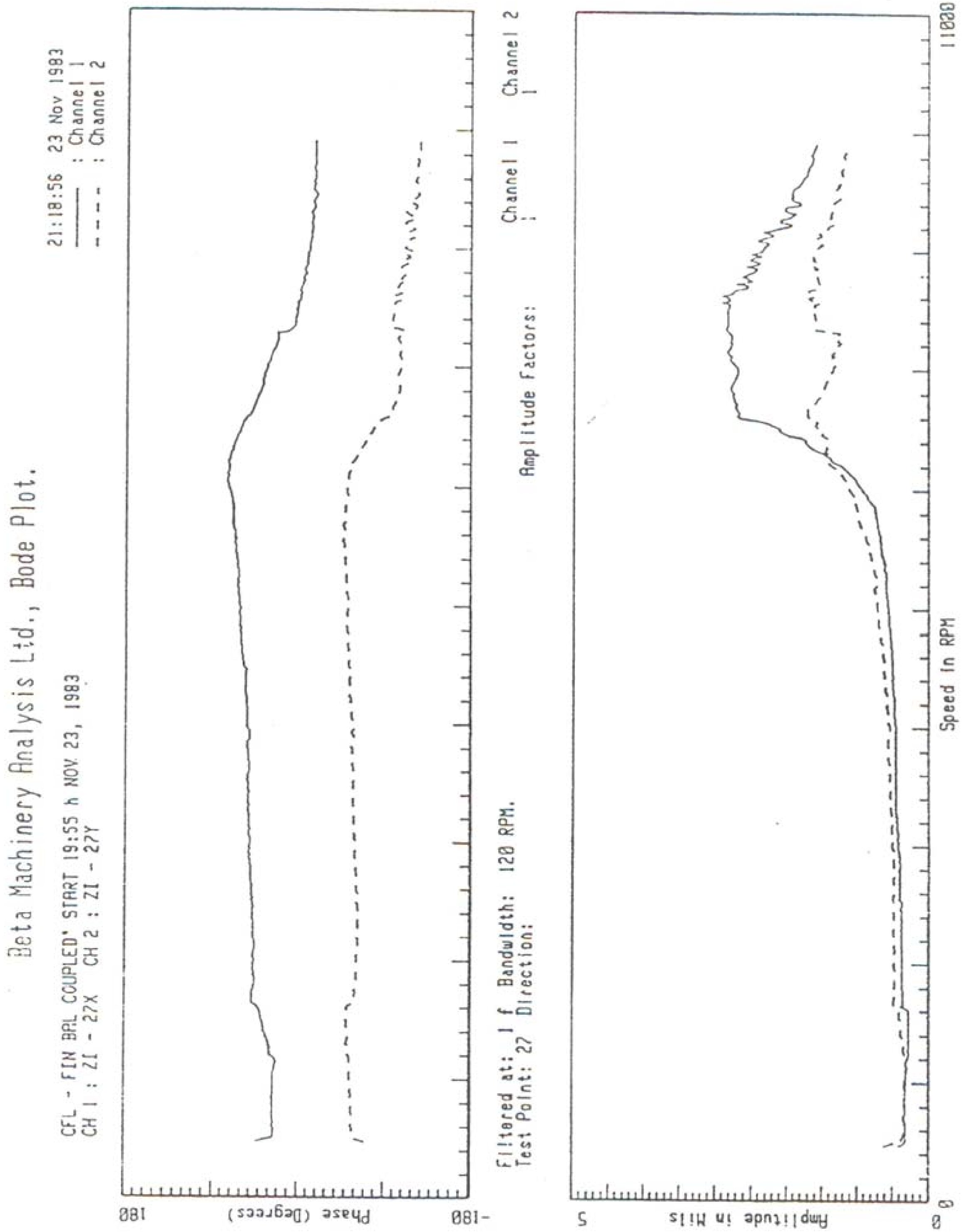

Beta Machinery Analysis Ltd., Bode Plot.
CFL - FIN BRL COUPLED' START 19:55 h NOV 23, 1983
CH 1 : ZI - 27X CH 2 : ZI - 27Y
21:18:56 23 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 120 RPM.
Test Point: 27 Direction:
Amplitude Factors:
Channel 1
Channel 2
180
Phase (Degrees)
-180
5
Amplitude in Mils
0
Speed In RPM
11000

FIGURE: 13

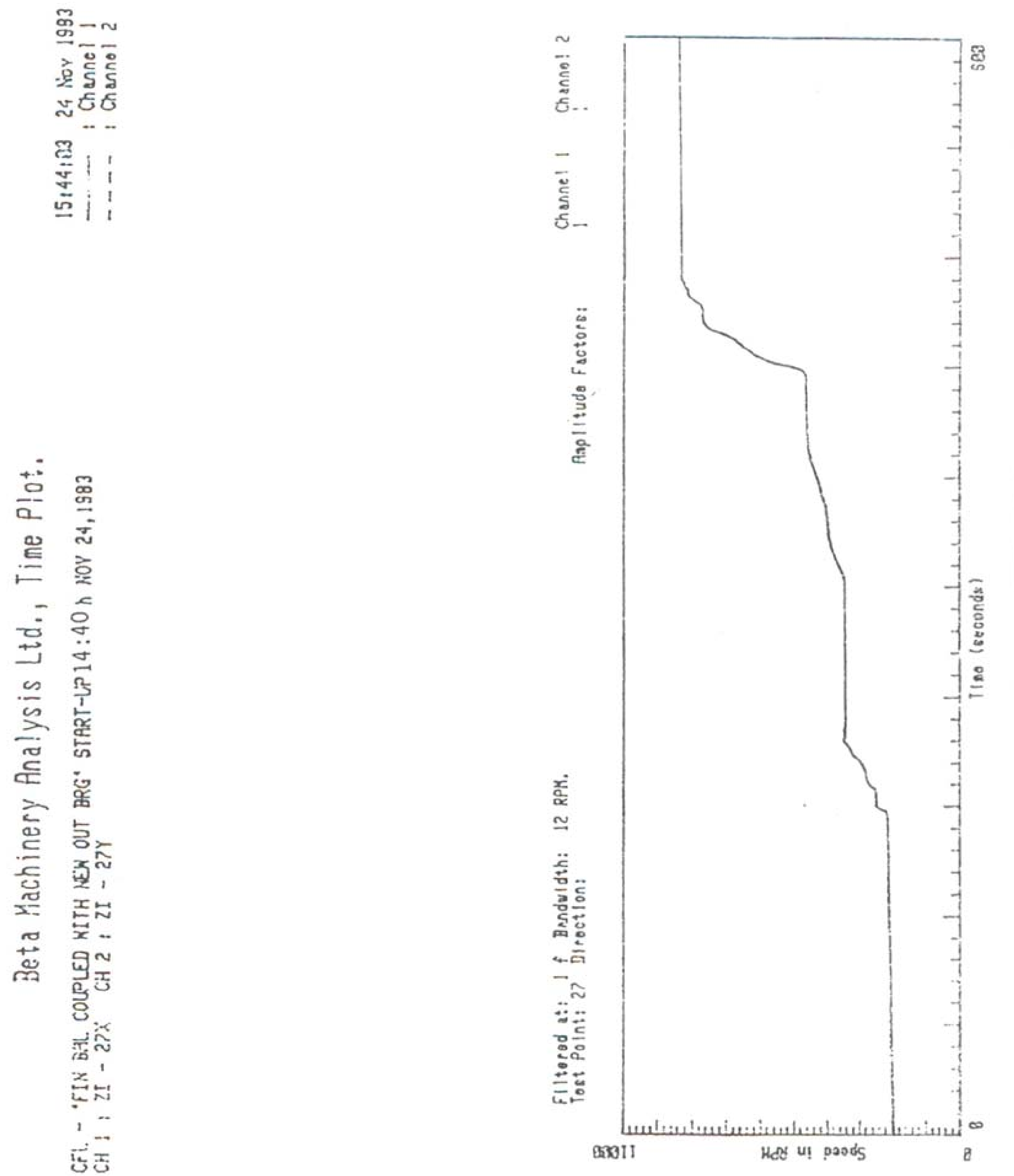
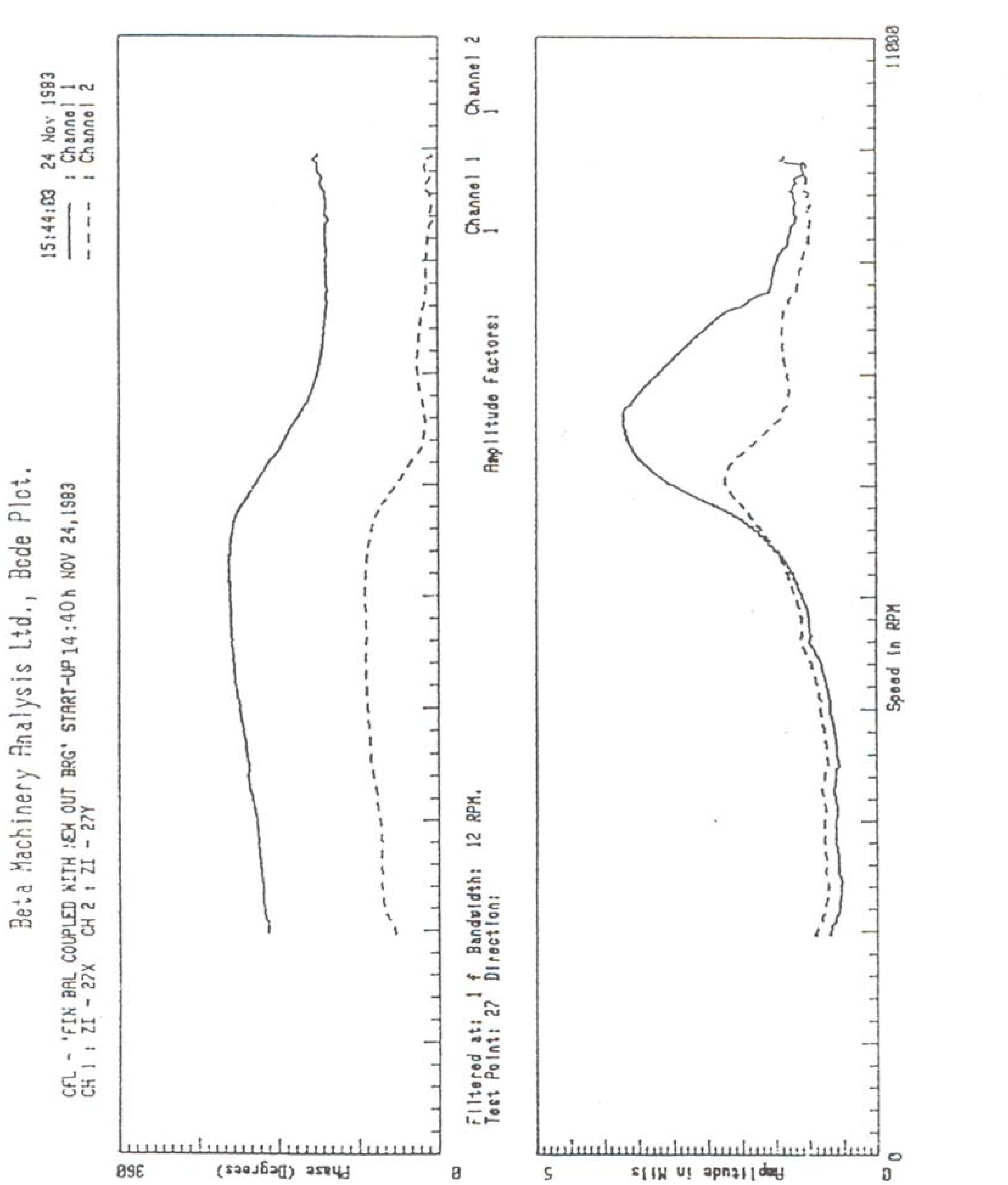

Beta Machinery Analysis Ltd., Time Plot.
CPL - 'FIN BHL COUPLED WITH NEW OUT BRG' START-UP14:40 h NOV 24,1983
CH 1 : ZI - 27X   CH 2 : ZI - 27Y
15:44:03 24 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 12 RPM.
Test Point: 27 Direction:
Amplitude Factors:
Channel 1
1
Channel 2
11000
Speed in RPM
0
0
Time (seconds)
503

FIGURE: 14


Beta Machinery Analysis Ltd., Bode Plot.
CFL - 'FIN BAL COUPLED WITH NEW OUT BRG' START-UP 14:40 h NOV 24,1983
CH 1 : ZI - 27X CH 2 : ZI - 27Y
15:44:03 24 Nov 1983
: Channel 1
: Channel 2
Filtered at: 1 f Bandwidth: 12 RPM.
Test Point: 27 Direction:
Amplitude Factors:
Channel 1
1
Channel 2
1
360
Phase (Degrees)
0
5
Amplitude in Mils
0
0
Speed in RPM
11000