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ROTOR-TO-STATOR RUB VIBRATION IN CENTRIFUGAL COMPRESSOR

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This paper discusses one example of excessive vibration encountered during loading of a centrifugal compressor train (H type compressor with HP casing).

An investigation was made of the effects of the dynamic load on the bearing stiffness and the rotor-bearing system critical speed. The high vibration occurred at a "threshold load," but the machine didn't run smoothly due to rubs even when it had passed through the threshold load.

This paper shall present the acquisition and discussion of the data taken in the field, as well as a description of the case history which utilizes background information to identify the malfunction condition. The analysis shows that the failures, including full reverse precession rub and exact one half subharmonic vibration, were caused by the oversize bearings and displacement of the rotor center due to foundation deformation and misalignment between gear shafts, etc.

The corrective actions taken to alleviate excessive vibration and the problems which remain to be solved are also presented in this paper.

INTRODUCTION

Modern rotating machines, if taken off-line, can cost hundreds of thousands of dollars in revenue due to production loss caused by the machine downtime. Therefore, machinery condition monitoring can prevent mechanical failures by warning operators of problems in the early stage, so that corrective action can be taken to prevent catastrophic failures. Detection and early diagnosis of potential failures of rotating machinery is a developing technology (Ref. 1).

Spectrum analysis, which is very effective in diagnosing rotating machinery problems and in quality control of new machinery, is widely used and can do an excellent job. However, in many cases it is difficult to pinpoint the cause of a failure in rotating machinery by spectrum analysis only. When combined with more information, for example, dynamic phase angle, shaft centerline position, orbits of dynamic motion and so on, which offer the most insight into the behavior of the machine, it will become more effective. In some cases of detecting the cause of a failure, it is necessary to acquire and analyze those dynamic motion parameters in a variety of operating conditions. This is an effective method often used in "clinical diagnosis" in the field.

According to the results of spectrum analysis, 1× and 1/2× vibrations are usually caused by unbalance and oil whirl. Often this is true, but this paper describes one case history wherein excessive vibration in the centrifugal compressor, detected by "clinical diagnosis," was caused by rotor to stator rubs.

FAILURES

The subject machine is the H type compressor with one HP casing, one LP body including two shaft-lines mounted on a gear box with 3 compression stages, and one HP casing with 3 aligned impellers, mounted at shaft-line end of third LP stage (see Fig. 1).

This machine was commissioned on August 1, 1980, and put into operation smoothly. On October 28, 1981, the machine incurred an excessive vibration condition on #3 bearing. Usually high rotor vibration occurs near the #3 bearing. Maximum amplitude of the rotor vibration was 45 μm and continued 45 seconds with increase in opening the guide vane after start-up, then soon dropped. But in the malfunction condition, maximum rotor vibration amplitude was more than 125 μm and didn't drop with continuous increase in opening the vane. As a result, the machine couldn't be put into operation (Fig. 2).

The same failures had occured in August 1982, and August 1984, after overhaul. It took ten days to dismount and remount the machine many times to alleviate the malfunction.

THRESHOLD LOAD

To detect the exact moment and cause of this high rotor vibration, the spectrum cascade during start-up and the curve of amplitude vs. degrees of the opened van had been recorded (see Figs. 3 and 4). Obviously, there is no amplitude peak in machine speed from 0 rpm to 13987 rpm. This means that the machine did not pass through a critical speed during start-up. But, with an increase in opening the vane, an amplitude peak occurred.

The results of calculation and analysis show that the high speed rotor and the #3, #4 bearings in the LP casing belong in the variable stiffness bearing system. It is different from most compressors. With an increase in dynamic load during opening the guide vane, both machine power and load of the #3, #4 bearings are increased. Fig. 5 shows radial load of the #3 bearing and direction angle vs. power of the machine. Because the bearing stiffness directly affects the critical speed of the rotor-bearing system (Fig. 6), at the moment when the dynamic load reaches a certain value, the critical speed can coincide with the operation speed, and the resonance will occur. This value of dynamic load is called "threshold load." In normal cases the machine runs smoothly as soon as it passes through the threshold load (Ref. 2).

DATA ACQUISITION AND REDUCTION

In the malfunction condition, excessive vibration did not drop even when A passed through the threshold load. To determine the exact cause of the failure, the dynamic data of the rotor vibration on the #3 bearing had been acquired and processed by Bently Nevada instrumentation in the field while assessing the troubles.

(1) rotor orbit plots

The change course of the orbit plots during loading the machine is shown in Fig. 7.

(2) vibration waves

In the excessive vibration condition, there are two types of waves and orbit plots (A), (B) (see Fig. 7).

(3) comparative spectrum plots

Corresponding to waves and orbit plots (A), (B), different spectrum plots are shown in Fig. 7.

(4) polar plots

At the moment when excessive 1× rpm vibration was dropping, the phase angle changed 720° (see Fig. 8).

DIAGNOSIS

The above-mentioned measurement provides us with information to recognize the causes of the malfunction condition.

(1) 1× rpm vibrations

At the moment of passing through the critical speed on the threshold load, 1× rpm high vibration occurred and soon dropped, but in the malfunction condition, excessive vibration did not drop even if passed through the threshold load. As shown in the Fig. 7 (A), the rotor motion orbit plot was elliptical with one fixed bright spot. During dismounting the machine, some wearing plots were discovered in the exact same direction on the #3 bearing. And at the moment when the excessive 1× rpm vibration was dropping, the change of 720° in phase angle had been recorded by chance (see Fig. 8). It shows the excessive vibration was reverse precession due to full rub, because high friction forces may cause a dramatic change of the precession direction from forward to continuous backward whirl (Ref. 3). This is self-excited vibration. If harmful energy acculates long enough, the orbit plots will diffuse (see Fig. 7 (A)). At this moment the machine must be shut down immediately.

(2) exact 1/2× rpm vibration

Oil whirl (hydrodynamic instability of bearings) is the most popular forced subrotative speed mechanism, but in this case according to the instability threshold curves of offset halves bearing (Ref. 4) and the results of the calculation of speed term and clearance term, the operation condition point on the #3 bearing was far from the region of instability. And it was exact 1/2× rpm but not less than 1/2 rpm vibration, which often occurs during oil whirl. In addition, the vibration wave was different from oil whirl and coinciding with subharmonic vibration (Ref. 5), so that the failure was not caused by oil whirl.

Exact 1/2 subrotative vibration may be caused by partial rub or by a loose nonrotating part (Ref. 3 and 6). During dismounting the machine, no loose part was found on the bearing shell, bearing housing, or pedestal support. As observed in the performance of many machines, the partial rotor to stator rub, or rub in oversized or poorly lubricated bearings, often causes steady subharmonic vibration of the frequency equal to exactly half of the rotating speed (Ref. 3). In the subject machine, the top clearances of the #3 and #4 offset halves bearings was 0.48 mm.

The diameter of the bearings was 120 mm, the ratio of the diameter to the clearance was 0.4%; obviously it is too excessive. For this reason, the failure can be attributed mainly to the oversized bearings.

Rotor rub against the bearing generates very complex rotor vibration. Rubs are created in machines by introducing a contact between the rotor and the bearing. The contact may be initiated by displacement of rotor center to stator center, which is caused by misalignment, foundation deformation, unparallel between gear shafts and eccentricity of the bearings. Displacement is prerequisite to introduction of rubs. That is why the machine may be put into operation by chance after remounting the bearings and exact alignment even in the oversize bearing condition. But in most cases, full rubs or partial rubs in the oversize bearing frequently occur due to rotor center displacement.

CORRECTIVE ACTION

It was impossible to take enough time to correct all the defects and keep machine downtime short, so the following temporary corrective actions were taken:

(1) The top clearances of both #3, #4 bearings were reduced from 0.48 mm to 0.31 mm on November 2, 1984.

(2) To increase oil viscosity the PRESLIA 20 type lub oil was replaced by PRESLIA 30 and oil temperature was reduced from 42°C to 28-30°C on November 3, 1984.

(3) The gear coupling was replaced by a new one and exact alignment was taken in December 1984.

Instant improvement results from modifying the bearings. After reducing the top clearances of the bearings, it was easy to pass through the threshold load and put the machine into operation. But the machine couldn't run smoothly with the same oil viscosity as before, even the #4 bearing had been cracked due to poor alignment condition. The vibration record shows the amplitude of rotor vibration was higher when the figure-8-shaped orbit appeared and was lower when it disappeared. This means the rubs were not eliminated completely. Since corrective action (2), (3) was taken with sufficiently high damping, the subharmonic vibration hasn't recurred at all and the machine has run smoothly up to now.

The problem which remains to solve is that it is necessary to reduce the inlet lub oil temperature below 30°C, otherwise rotor to stator rub vibration will occur again. Therefore, other defects should be eliminated in the future.

SUMMARY AND CONCLUSIONS

This case history shows that it is not enough to diagnose complex rotor vibration by the spectrum analysis method only, because there are several possible causes of failures in the machine corresponding to certain characteristic spectrums. Therefore, for detecting the potential failures of rotating machinery, it is necessary to acquire as much information as possible, for example, orbit plots, polar plots, vibration waves, and other parameters of dynamic motion in a variety of operation conditions. The more information obtained, the more possibilities to eliminate the false and retain the true, and finally pinpoint the real cause.

In the H type centrifugal compressor with HP casing, the high vibration during loading the machine is an unavoidable transient due to passing through the critical speed on the threshold load. But, excessive full rub reverse precession or exact 1/2 subharmonic vibration may be caused by oversize bearings and displacement of the rotor center.

In this case history, the corrective actions, i.e., reduce the top clearances of the bearings, increase the lub oil viscosity and exact alignment, provided instant results to alleviate excessive vibration caused by rotor to stator rubs in the centrifugal compressor.

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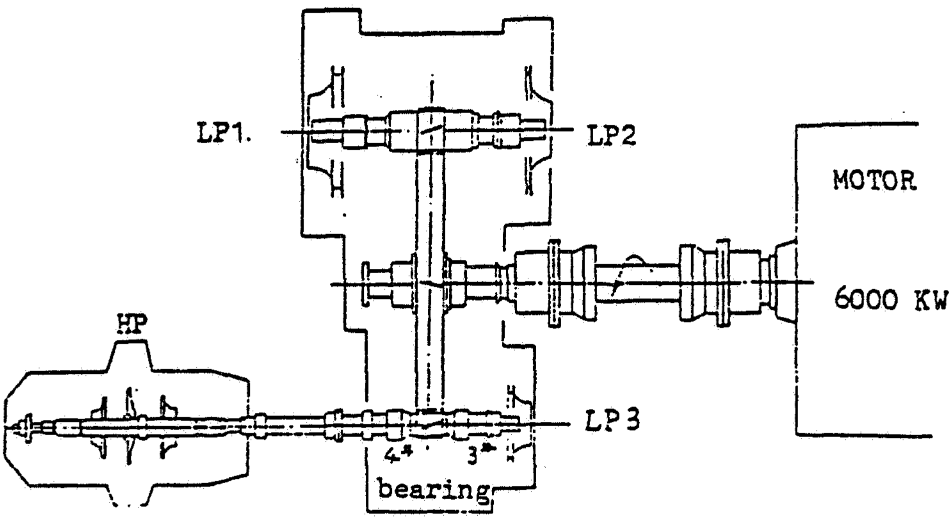


Figure 1.- H type of centrifugal compressor with HP casing.

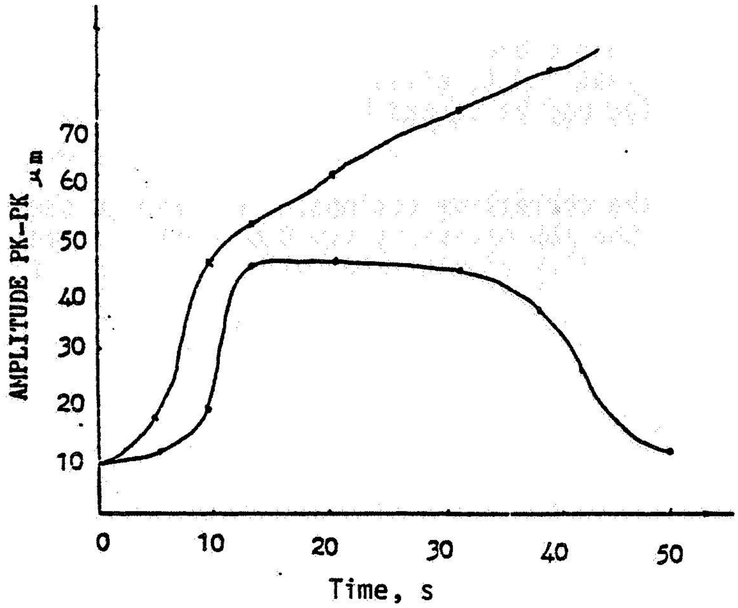


Figure 2.- Vibration versus time during startup. November 30, 1981.

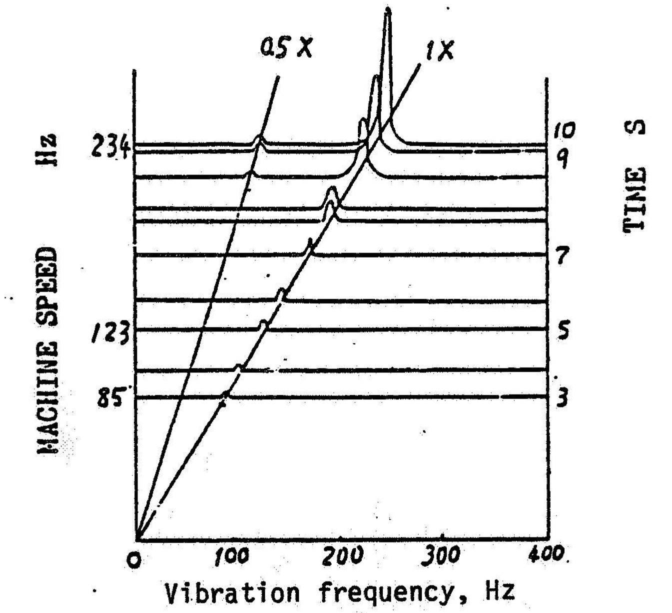


Figure 3.- Spectrum cascade during startup. February 10, 1983.

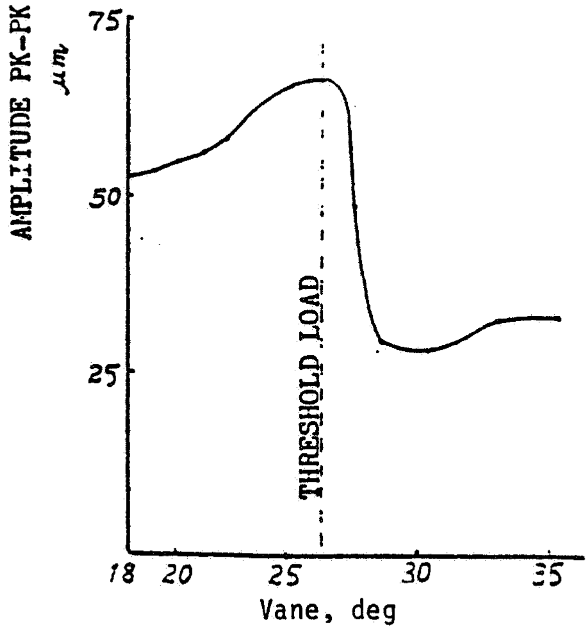


Figure 4.- Effect of dynamic load on vibration. February 10, 1982.

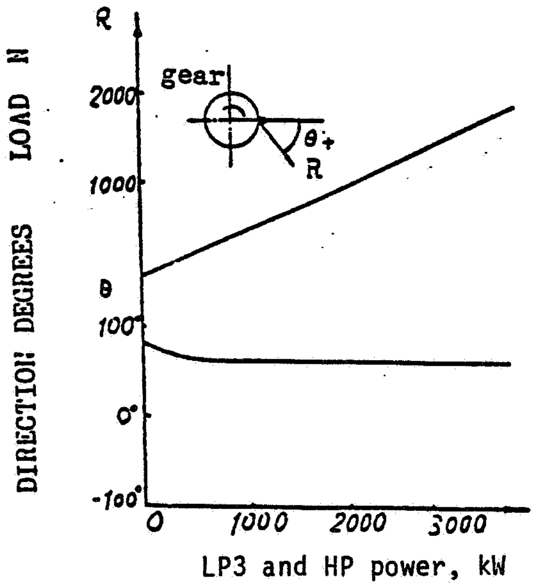


Figure 5.- Radial load versus machine power.

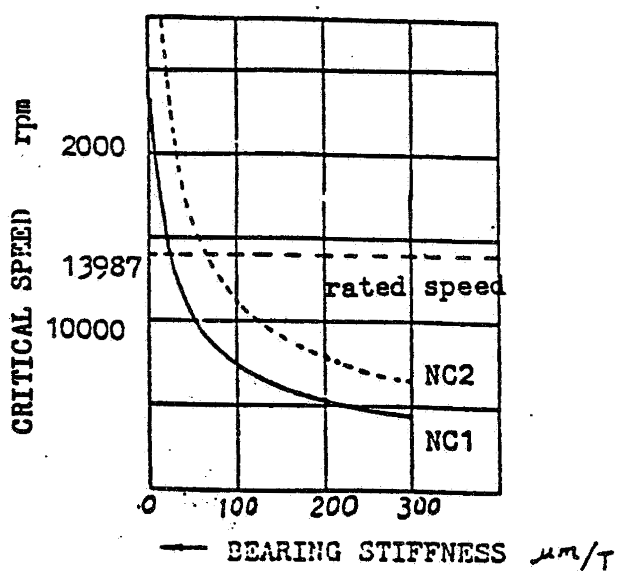
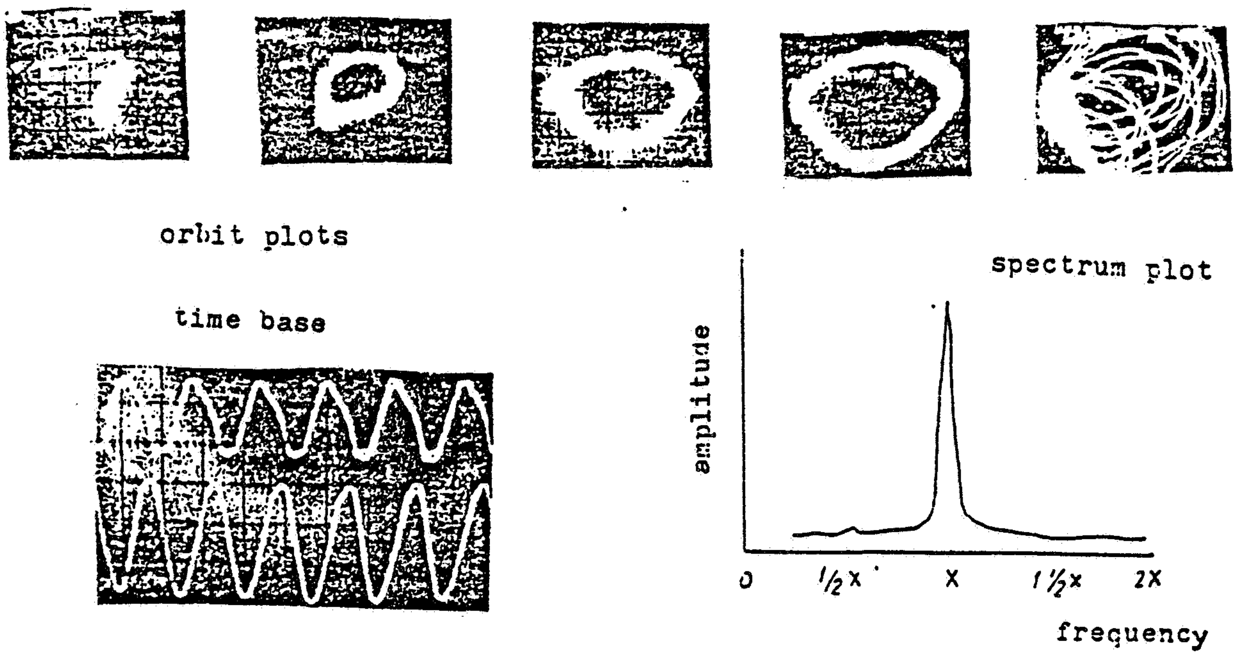
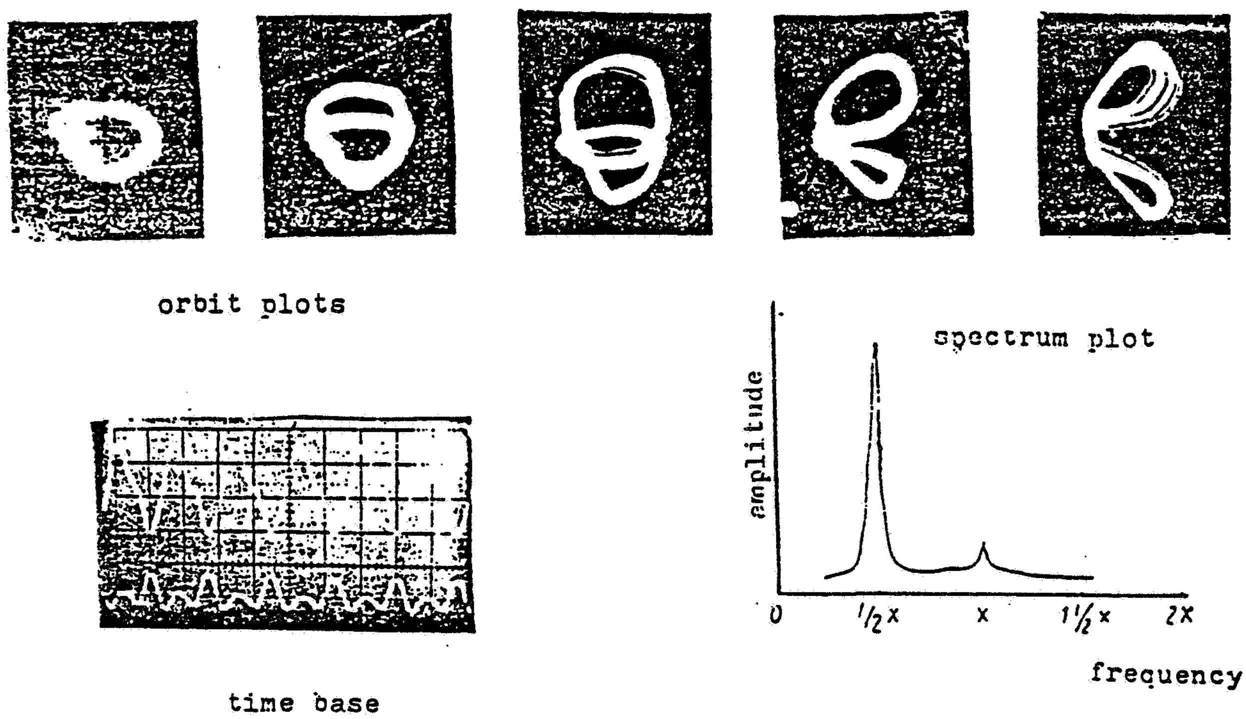


Figure 6.- Critical speed versus bearing stiffness.



(a) Full-rub, reverse-precession vibration.



(b) Exact 1/2 subharmonic vibration.

Figure 7.- Orbit, spectrum, and frequency plots.

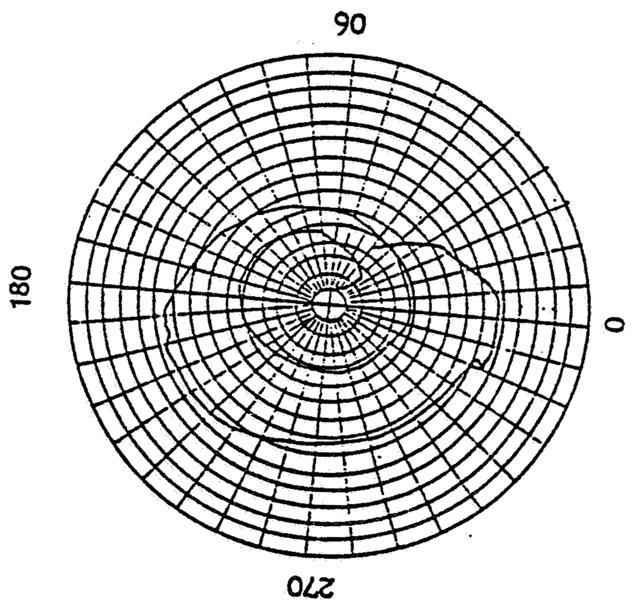


Figure 8.- Polar plot. February 13, 1982.