INTERNAL HYSTERESIS EXPERIENCED ON A HIGH PRESSURE SYN GAS COMPRESSOR

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This paper describes a vibration instability phenomenon experienced in operating high pressure syn gas centrifugal compressors in two Ammonia plants at Qatar Fertiliser Company. The compressors were closely monitored using orbit and spectrum analysis to follow up changes from baseline readings. Tape recordings of run up and run down data presented a clue to the cause of the instability, which was later confirmed through physical examination of the rotor assembly. Internal hysteresis was the major destabilizing force. However, the problem was further complicated by seal lock-up at the suction end of the compressor. Also, a coupling lock-up problem and a coupling fit problem, which caused frettage of the shaft, should not be counted out as contributors to the self-excited vibrations.

INTRODUCTION

Instabilities in high-speed, high-pressure centrifugal compressors are quite common today. These compressors are usually running above two times their first critical speed, are quite light, and have high horsepower and pressure ratings. A recent survey (ref. 1) on causes of Ammonia plant shutdowns reported that syn gas compressor failures caused more downtime than any other major equipment, accounting for 23 to 31% of the major equipment downtime. The average Ammonia plant has a syn gas compressor failure every 7 to 9 months, and of the 88 world-wide plants surveyed in reference 1, 89% suffered a syn gas compressor failure during the survey period. The results of this survey are not surprising to those familiar with the operation of Ammonia plants. The syn gas compressor is subjected to a severe operating environment which is much tougher than that demanded of other compression units in the plant.

GENERAL CONFIGURATION

A schematic of the high pressure compressor train is shown in fig. 1. An aircraft derivative gas turbine (gas generator) powers a free power turbine which then drives the high pressure compressor through a speed increasing gear. The normal start-up sequence is to start the gas turbine and hold it at warm-up speed of 1500 rpm, which corresponds to 4500 rpm on the compressor. Upon completion of the warm-up time, the unit is then quickly accelerated to the minimum governed speed, which is equivalent to 11,000 rpm for the compressor. A sketch of the high pressure rotor is shown in figure 2, outlining the major components. The rotor consists of two stages: the high pressure stage consisting of 7 impellers, and the circulator stage with one impeller. The two stages are separated by a long interstage labyrinth seal, and are housed in a common barrel casing. The rotor is supported on tilt pad bearings with 5 pads in a load on pad configuration. The oil seals are of the floating ring type, where the sealing is applied through lapping of the housing lip with the mating face of the outer seal ring. An "O" ring seal is used on the housing face of the inner seal ring.

HISTORY

The syn gas high pressure compressor has always been the bottle neck in the operation of the Ammonia plants at Qatar Fertiliser Company. It has been plagued by several design problems that were detrimental to the safe operation of the unit, and consequently it had to be run below its rated speed and horsepower. The unit was running at too high an axial thrust during the first two years of operation. This required the resizing of the balance piston twice before we were able to run it at full load. Another design deficiency which also caused interrupted operation and damage to the rotor, was the improperly heat treated tie rods which hold the aerodynamic assembly together. These would often break during start-up at the threaded section of the rod and fall through the thrust balance line opening into the suction of the compressor damaging labyrinth seals and shifting impellers on the rotor shaft.

Recognizing the problems caused by this compressor and others in the plant, attention was directed towards acquiring more sophisticated diagnostic instrumentation to enable plant personnel to follow up developments on the machines.

INTERNAL HYSTERESIS

The normal vibration spectrum for the rotor obtained after an overhaul always contained a subsynchronous component approximately equal to 50% of the running speed frequency. As seen in figure 3 it was bounded, and the synchronous component due to residual unbalance was the dominant component in the baseline spectrum. Figure 4 shows the unfiltered orbit which agrees with the information designated by spectrum analysis, where the subsynchronous component forms an internal loop and is smaller in magnitude compared to the synchronous component. However, after being in service for some time, we always noted an increase in vibration. Figure 5 shows a spectrum of the same rotor taken 7 months later where the subsynchronous component became the dominant frequency. The orbit analysis also confirmed this fact. As noted in figures 4 and 6, the subsynchronous component has increased and predominates the total motion masking the effect of unbalance. Gunter (ref. 2) describes investigations which produced results similar to what we have experienced.

Our suspicion of internal friction damping to be the driving force of this instability was later verified during the annual shutdown when the rotor was pulled out and inspected. All 7 impellers of the high pressure stage, including the sleeves, had travelled along the shaft in a direction towards the suction end of the compressor. It was later hypothesized that the shrink fit was not adequate to start with , which explains why we had subsynchronous vibrations right after starting up with a newly overhauled rotor. After running for some time, the impellers apparently start to travel towards the suction end of the compressor. As they begin to make contact with the sleeves, this gives rise to more friction and subsequently increases the contribution of the internal friction forces acting on the rotor. This causes the subsynchronous precession to predominate over the unbalance response.

TRANSIENT DATA

Through the use of a tape recorder, transient data were obtained and further confirmed our previous findings. Figure 7 shows the run-up data of a newly overhauled rotor that was installed during an annual turnaround. At a speed of 5700 rpm (95 HZ), we noted a peak in amplitude of the synchronous component indicating the critical speed of the rotor. The synchronous component continued to be the only apparent signature until the rotor crossed the threshold speed and a subsynchronous component appeared at exactly 50% of running speed frequency. The subsynchronous component traced the running speed frequency until it reached the critical speed frequency and locked on it. At this point, changes in the running speed did not affect the instability which remained locked on to the rotor's first critical speed.

This verification eliminated the possibility of oil whirl in the seals as being the exciting force, since the seals were also replaced during the annual turnaround and could not have been locked-up at this stage. We have had the seals lock-up after several months of operation, and noticed that the vibration characteristics in such a case are quite distinct from what we witnessed here. Fig. 8 shows a frequency spectrum display of the rotor with a subharmonic component at 72 HZ which amounts to about 40% of running speed frequency. This was the result of the seals locking-up and acting as a sleeve bearing.

MAJOR RECONFIGURATION

The information from our findings was reported to the manufacturer of the compressor, who quickly confirmed that the rotor design was marginal and very susceptible to the form of instability just described. They claimed that knowledge to predict such a phenomenon was not available at the time this compressor was designed, and proposed a major redesign (3) to solve the problem. This involved a change in rotor stiffness by increasing the shaft diameter and reducing the bearing span. This would eventually mean scrapping all of the five existing rotor shafts, and the replacement and/or reworking of existing bearing and seal assemblies to reduce the bearing span. All spacers would have to be replaced by new ones to accommodate the larger diameter rotor shaft, in addition to reworking all the impellers and balance piston bores. The proposed arrangement also included putting the journal bearings between the oil film bushing seals on each end of the compressor.

This extensive modification was rejected for several reasons. First, it was very expensive; over half a million dollars, not including the cost of scrapped parts and the additionally required spares. Second, it would require a rather long downtime in order to carry out the proposed rework. Third, it was based completely on a computer program simulation, and as stated in Reference 4, this is questionable for the case of rotordynamic simulation since machine parameters such as damping, internal friction, bearing support stiffness, etc., usually cannot be calculated accurately. These parameters must be adjusted in the computer simulation until the behaviour of the machine in the field can be reproduced faithfully. No such attempt to take field measurements was made by the manufacturer. Finally, a single minor modification which we proposed showed good promise of reducing or eliminating the major driving force of the instability.

CORRECTIVE ACTION

The objective here was to determine the most economical way to make the machine run smoothly and reliably. It was apparent that increasing the shrink fit by plating the shaft about one mil oversize should be the first step in such a solution, since it requires a minimum of expenditure. This solution was sound since it does not have an adverse effect on material strength, cause stress concentrations or stress corrosion, and will not exceed the yield strength of the material.

Figure 9 shows a cascade spectrum of a rotor that had the shrink fit improved by plating the shaft 1 mil oversize. No appreciable subsynchronous vibrations were observed throughout the speed range of the compressor. What is also apparent in Figure 9, and of interest to note, is that the rotor's first critical speed was still around 95 HZ. The unit was operating at a very low vibration level allowing the plant to run at loads as high as 125%, a level never before achieved.

It is important to note here that problems arising from loose fits have been known to the manufacturer and in a particular case (5) caused them to thoroughly scrutinize and modify their methods for assembly of spacers and impellers. The fractional frequency problem they experienced was similar in nature to the one described here, and also involved a rotor with a back-to-back impeller arrangement. The sleeve at the center, required to separate the two sections was modified and made integral with the shaft with shoulders at each end to provide impeller stops. This particular rotor was also fitted with a squeeze film damper bearing to enable it to operate through the full speed/pressure range.

SUMMARY AND CONCLUSION

- Instabilities in high speed compressors are numerous; to tackle these problems we must be able to first identify them. This requires both an increased understanding of the causes of instabilities, in addition to the availability of modern sophisticated diagnostic instrumentation.

- Hysteretic whirl is not a rare source of instability (6). It can come from loose impeller and spacer fits, can be caused by poor coupling hub fit, or can result from tooth friction in a poorly misaligned gear coupling. It can also be provoked by a sudden impact or shock and cause an otherwise stable rotor to whirl.

We were the victims of such an occurrence, where one of the tie rods on the aerodynamic assembly broke off and caused impact damage. The rotor was balanced and restarted, but experienced severe subsynchronous vibration diagnosed as hysteretic whirl. The damage caused shifting of the impellers on the shaft and started relative internal slippage in the fit.

- It has apparently been assumed by many turbomachinery users that tilt pad bearings are the solution to all rotordynamic problems. While this might hold true in many cases involving oil whirl excited instabilities, it is certainly not a solution to all problems. The case just presented is a good example of this since the rotor was supported by tilt pad bearings.

- A large percentage of centrifugal compressors in the process industry today are operating at speeds higher than two times their first critical speed, and a significant portion of these are susceptible to instability. The instability can be the result of several exciting forces (4), which makes it difficult to pinpoint the fault. The solution of increasing shaft stiffness by increasing shaft diameter is not a favorable one, since this will increase the peripheral speed of the shaft's journal, restrict gas flow, and increase horsepower consumption.

- Instabilities like the one described here have been around for some time and still exist in many plants. Many machinery operators, however, still accept these problems as a fact of life, partly because they are not aware of recent advances in this field as well as the availability of easy to use diagnostic instrumentation. As such, the failures are accepted as something normal and they continue to regularly change and replace failed components.

ACKNOWLEDGEMENTS

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REFERENCES

1. Williams, G.P. and Hoehing, W.W., "Causes of Ammonia plant shutdowns, Survey IV", Ammonia plant and related facilities safety symposium, Los Angeles, California, AICHE paper No. 124g.

2. Gunter, E.J., "Dynamic Stability of Rotor-Bearing Systems", NASA SP-113, 1969.

3. Shade, W.N., "RB10-8B HPC Rotor Stability Problems", Cooper Energy Services, Rotating Machinery Engineering Dept., Centrifugal Compressor Design Report No. CCD-108, Oct. 21, 1982.

4. Vance, J.M., "Instabilities in Turbomachinery", Proceedings of Machinery Vibration Monitoring and Analysis Seminar, New Orleans, Louisiana, April 7-9, 1981, pps. 107-113, Sponsored by The Vibration Institute, Clarendon Hills, Illinois.

5. Smith, K.J., "An Operation History of Fractional Frequency Whirl", Proceedings of the Fourth Turbomachinery Symposium, Texas A&M University, College Station, Texas, 1975, pp. 115-125.

6. John Sohre, "Operating Problems With High-Speed Turbomachinery Causes and Correction", Part II, Case Histories, 1976.

7. Vance, J.M. and Lee, J., "Stability Of High Speed Rotors With Internal Friction", ASME Journal of Engineering for Industry.

8. Childs, D.W., "Fractional Frequency Rotor Motion Due To Non-Symmetric Clearance Effects", ASME paper No. 81-GT-145, Gas Turbine Conference, Houston, Texas, March 9-12, 1981.

9. Murphy, B.T., and Vance, J.M., "Labyrinth Seal Effects on Rotor Whirl Instability", Proceedings of The Second International Conference on Vibrations in Rotating Machinery, Churchill College, Cambridge, September 1-4, 1980.

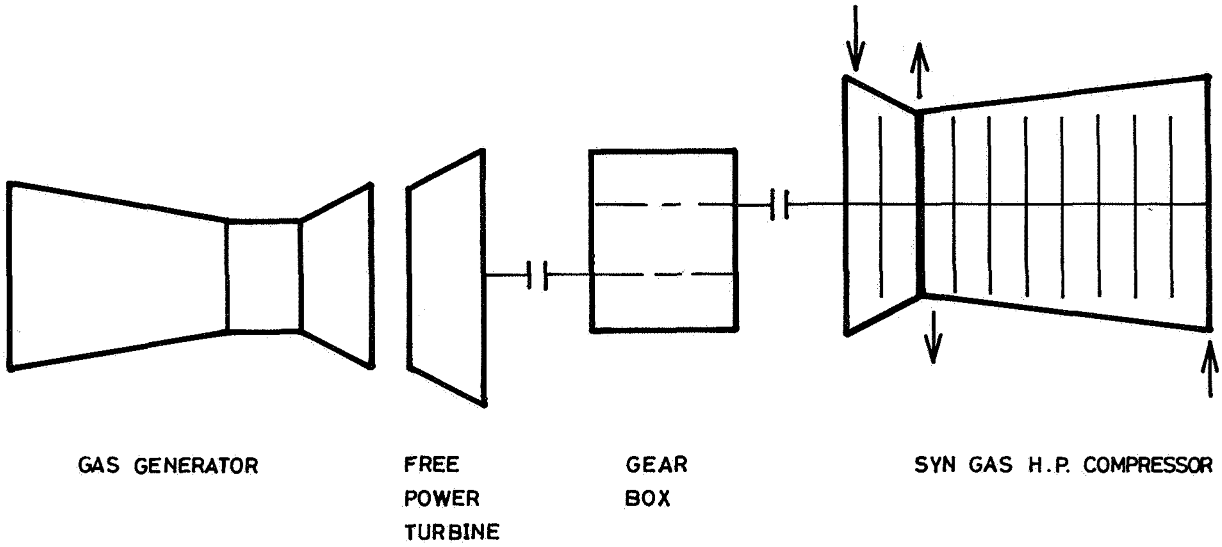

Figure 1. - Schematic of compressor train.

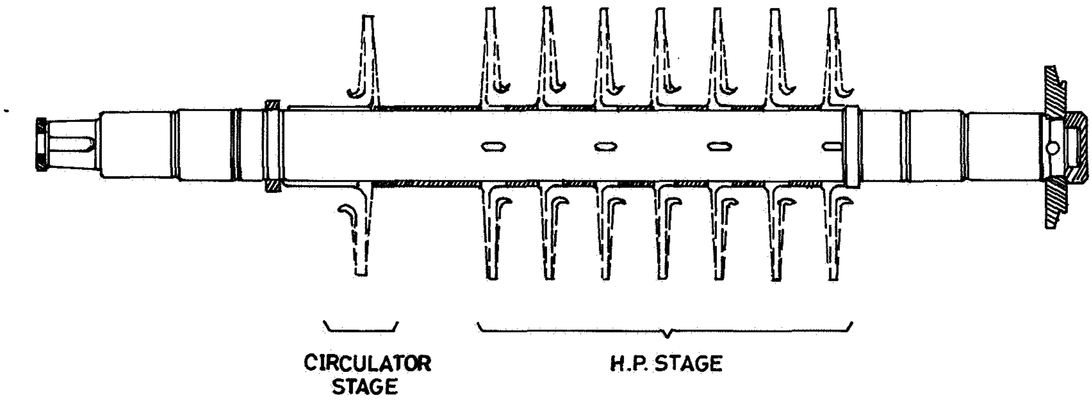

Figure 2. - Syn. gas compressor rotor.

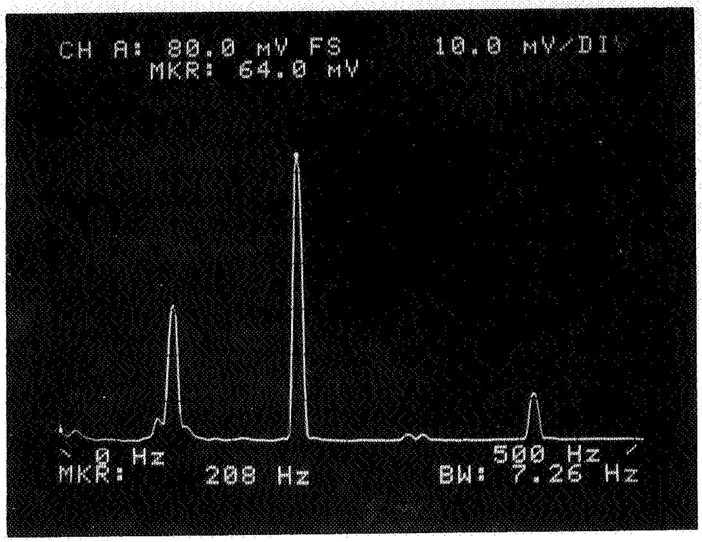

Figure 3. - High-pressure compressor 3101; location, A-II compressor house; date, 28/4/1981; 8 V; 50 mV rms/div.; 200 mV/mil; frequency range, 0-500 Hz.

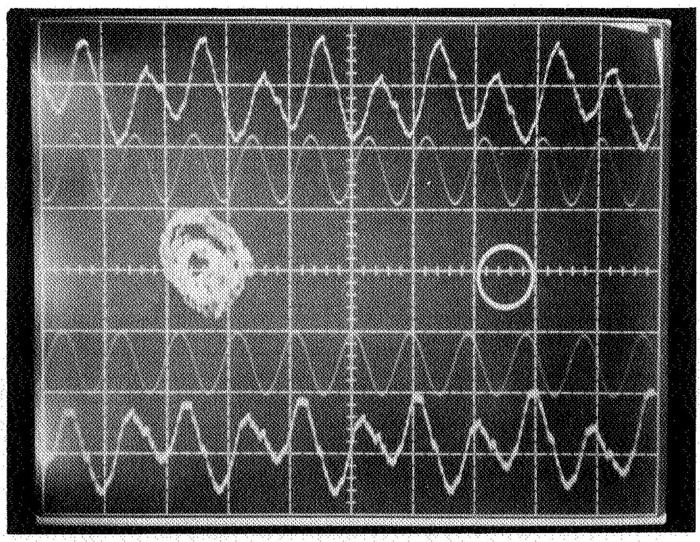

Figure 4. - High-pressure compressor 3101; location, A-II compressor house; date, 28/4/1981; 8 V/8 H; 0.2 V/div.; 1 mil/div.; sweep rate, 5 ms/div.; rotational speed, 12 471 rpm.

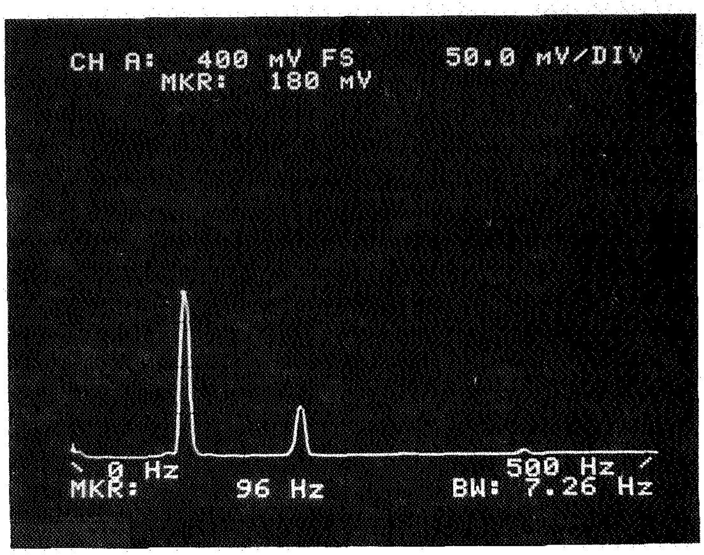

Figure 5. - High-pressure compressor 3101; location, A-II compressor house; date, 31/12/1981; 8 V; 10 mV rms/div.; 200 mV/mil; frequency range, 0-500 Hz.

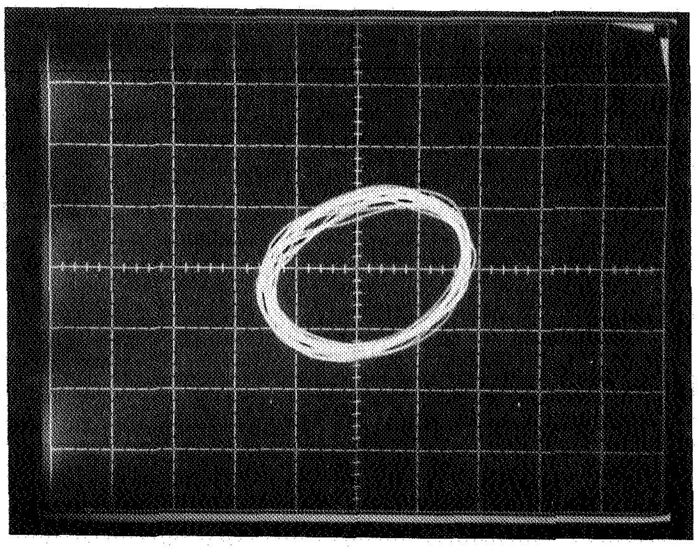

Figure 6. - High-pressure compressor 3101; Location, A-II compressor house; date, 31/12/1981; 8 V/7 H; 0.2 V/div.; 1 mil/div.; sweep rate, 5 ms/div.; rotational speed, 11 500 rpm.

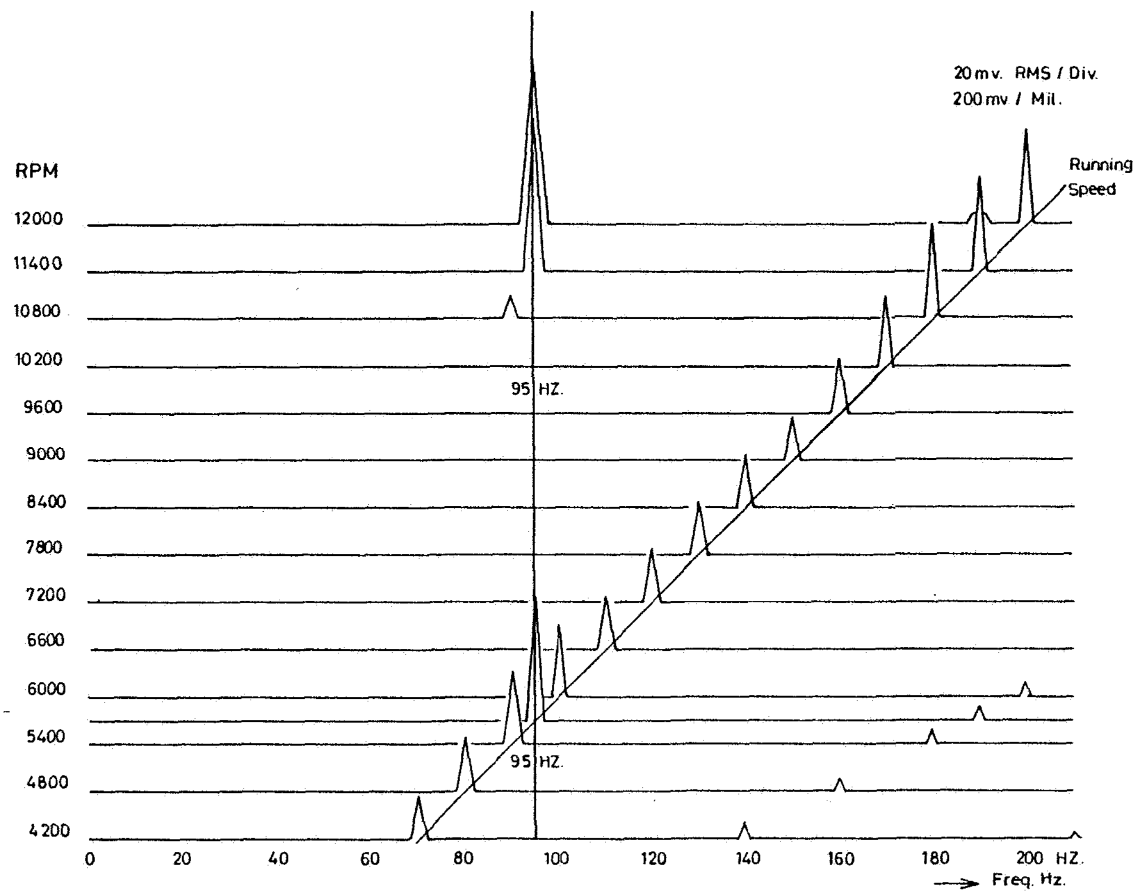

Figure 7. - High-pressure compressor runup.

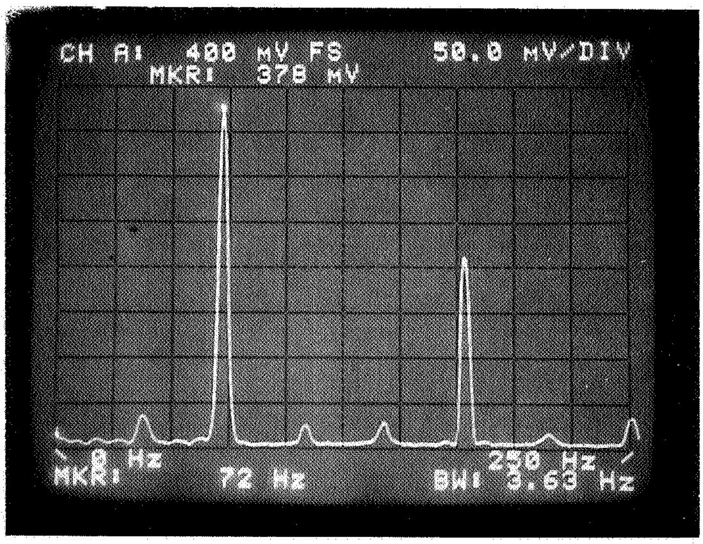

Figure 8. - High-pressure compressor 3101; location, A-II compressor house; date, 8/8/1983; 8 V; 20 mV rms/div.; 200 mV/mil; frequency range; 0-250 Hz.

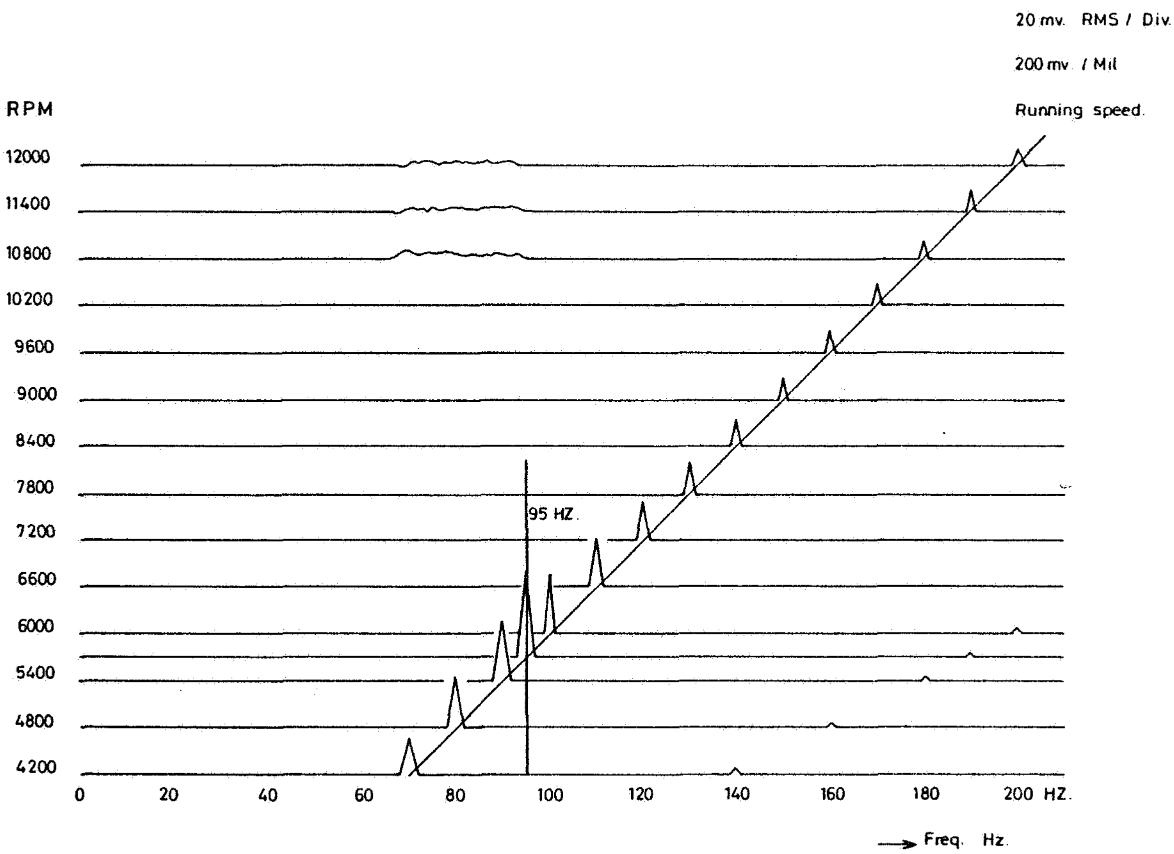

Figure 9. - High-pressure compressor runup with shrink fit improved.