The Rationale of Monitoring Vibration on Rotating Machinery in Continuously Operating Process Plant

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The Rationale of Monitoring Vibration on Rotating Machinery in Continuously Operating Process Plant

This paper presents a factual discourse on the problems associated with vibration monitoring of continuously operating process machinery. The shortcomings of existing vibration criteria are discussed with reference to operational experience and case studies on chemical process plant operated by Shell Chemicals U. K. Ltd. and with particular regard to the wide variations in machine frame impedances which have been measured on a representative number of machines used for ethylene and propylene production. A technique is described which enables bearing impedances to be measured quickly and safely on operational machines on site. The information thus obtained is used in conjunction with operational vibration levels to interpret general vibration criteria in a more realistic manner, having regard to the dynamic characteristics of particular machines. The philosophy of machine wear diagnosis in relation to prediction of component failure is discussed with a view to the extension of running periods between overhaul and to enable predictions to be made of replacement components requirements prior to dismantling machines, and thereby to reduce overhaul periods. This philosophy is being applied at present to a number of operational machines, and details of the results to date are given. Finally, the type of monitoring system required to provide this additional information, as well as providing the conventional day-to-day alarm facility, is discussed.

Introduction

Vibration levels on machine frames and bearing housings provide a useful guide to such things as state of balance and alignment of installations, particularly for commissioning and acceptance trials. In recent years, also, continuous vibration monitoring of process plant has become accepted practice, sometimes by means of permanent electronic installations coupled to alarm systems, but more often by means of periodic checks with hand-held vibration level meters.

However, vibration criteria used for the comparative assessment of plant performance are extremely generalized in conception, being based entirely on subjective opinions and having little regard to the dynamic response of particular machines. Dynamic forces arising from the rotating elements inside a machine produce bearing and seal failure, for instance, and are often disguised by extremely stiff bearing supports. Also, criteria based on rms levels of vibration within unspecified frequency bands can be misleading.

The objectives of the work described in this paper were firstly, to review the general problems of vibration measurement and criteria making full use of the more sophisticated techniques in general use for research and development work, and secondly, to produce routine procedures based on the techniques which would enable operators to predict rate of wear and deterioration of machine components and to diagnose malfunctioning of machines. Thus, it was thought, it would be possible to improve the information supplied to control room operators from continuously monitored alarm systems and, also, to build up sufficient experience on machines with inherent malfunction to enable plant to be operated to the limit of component failure and, thus reduce the number of unscheduled shut-downs which are so costly in lost production on continuously operated process plant.

Vibration Criteria

Operational criteria representing vibration boundary levels for satisfactory or unsatisfactory running conditions are many and varied. Of the more widely quoted criteria, those of Rathbone, Yates, and V.D.I. 2056 are worthy of mention with particular regard to the way in which these criteria were obtained.

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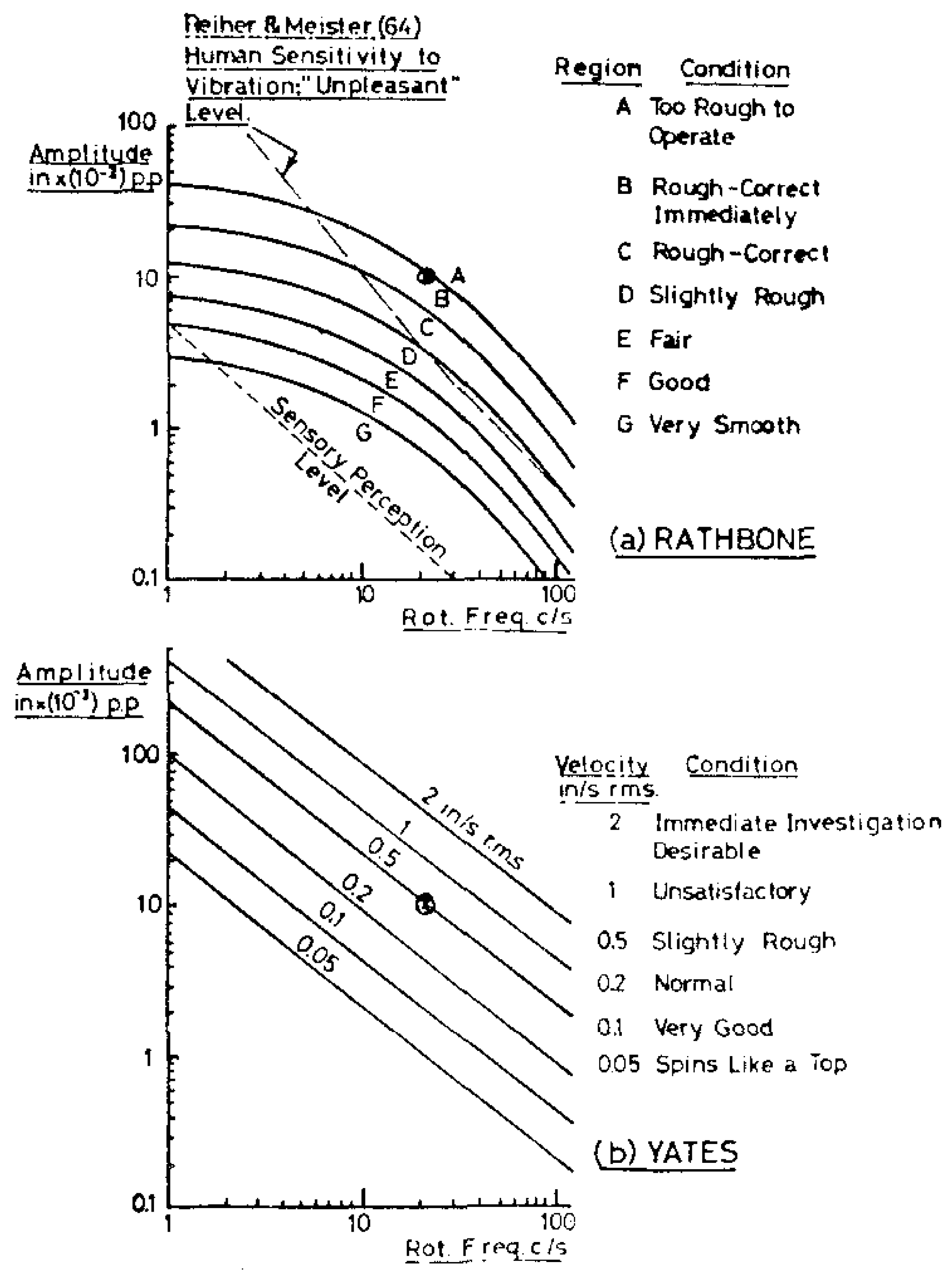

Fig. 1 Vibration criteria for rotating machines proposed by Rathbone and Yates

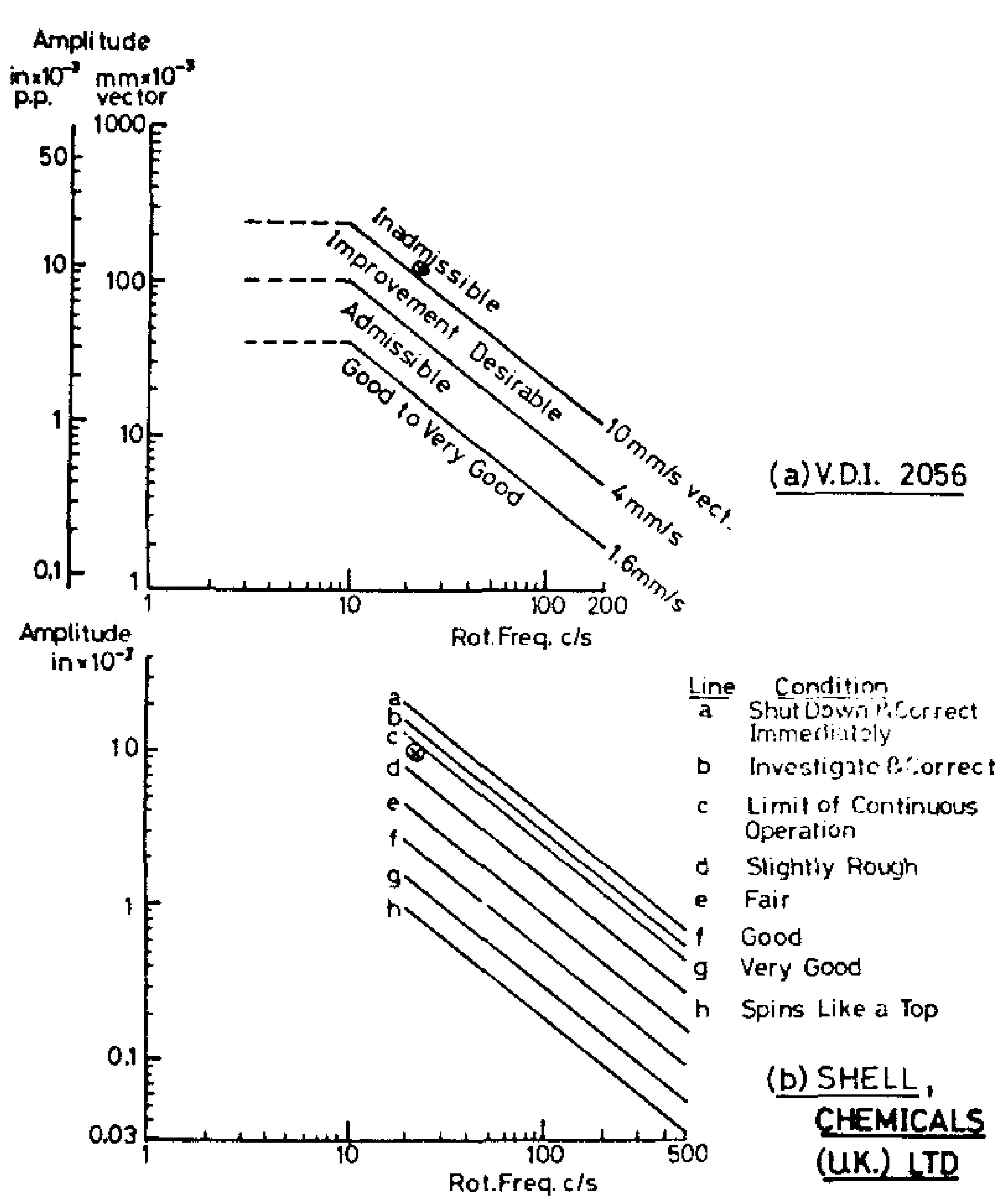

Fig. 2 Vibration criteria for rotating machines; V. D. I. 2056 and Shell Chemicals U.K. Ltd., Carrington

Fig. 1 shows the Rathbone and Yates criteria; in both cases the assessment of acceptability was obtained by subjective opinions of several practical engineers and inspectors, followed by vibration level measurements with relatively crude instruments.

Rathbone [1][1] limited his assessments to machinery running at speeds less than 5000 rpm and considered it reasonable to extrapolate his curves for higher rotational speeds. He was also conscious of the fact that the measurement of bearing levels only would not necessarily produce valid criteria for all machines.

Yates [2] produced his criterion by numerous tests on marine geared turbine installations. The fact that in this case the machinery was installed in relatively flexible steel shells, as opposed to the more massive foundations of Rathbone's machines, would account in part for the differences between the two criteria. For example, at 22 Hz 'too rough' on the Rathbone curve is classified as only slightly rough on the Yates curve.

It is also interesting to note that the line classifying an unpleasant level of vibration from the Reiher and Meister curves of human sensitivity [3] to vibration compare closely with the acceptable boundary levels on machines.

There is also a similarity between these curves and those for building vibrations.

The V.D.I. 2056 specification [4], Fig. 2, is one of the most exhaustive in present use and was compiled from previously published criteria for five separate classes of machines. Group G machines for power generation and process machinery on heavy rigid foundations were taken from Rathbone's work, although his curves were approximated to straight lines and the maximum rotational speed extended to 12,000 rpm.

The directive places no restriction on the type of vibration measured (e.g., displacement, velocity, or acceleration), but suggests that where several harmonic components are present an rms measurement of the complex signal should be used for comparison with the given criteria.

S.C.U.K.L., like many other companies, have produced their own guidelines for acceptable vibration levels (Fig. 2) by modifying criteria such as those aforementioned to suit their own particular machinery in the light of their own experience.

The significant factor regarding all these criteria is, however, their derivation from subjective opinions with little regard to the dynamic systems to which they are applied. Rathbone recognized this factor to some extent when he suggested that resonance of the structure could serve to allow increased tolerances and that machines particularly sensitive to unbalance vibrations might require criteria based on residual unbalance.

The problem can be appreciated better from an examination of a mathematical model. Consider, for example, a relatively simple model of a single machine, Fig. 3.

Here vertical vibrations are considered only, coupling between vertical and horizontal vibrations are ignored, the oil film is represented by a linear spring, and casing modes of vibration are ignored, i.e., coupling between bearing housings is neglected. The system has thus been reduced to seven degrees of freedom if we ignore rotational inertias and consider only linear motion of the masses. To assess the vibratory loads and stresses in such a system by measurements at two points only (i.e., the bearing supports) is therefore meaningless unless additional information is available to supplement the measurements.

Ideally, if the stiffness, damping, and mass parameters for the system were known it would be possible to calculate the loads

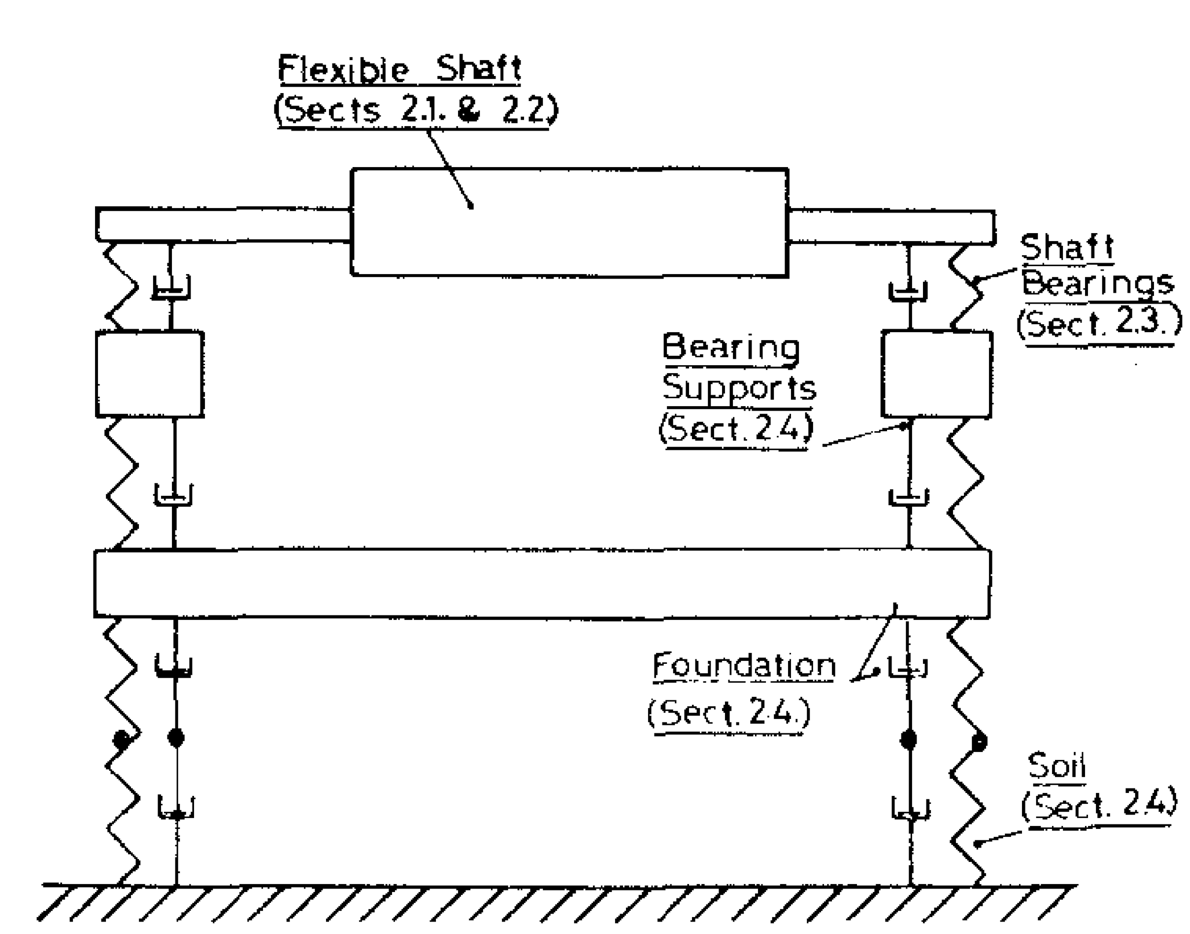

Fig. 3 Simplified model of rotating machine for vibration analysis

[1] Numbers in brackets designate References at end of paper.

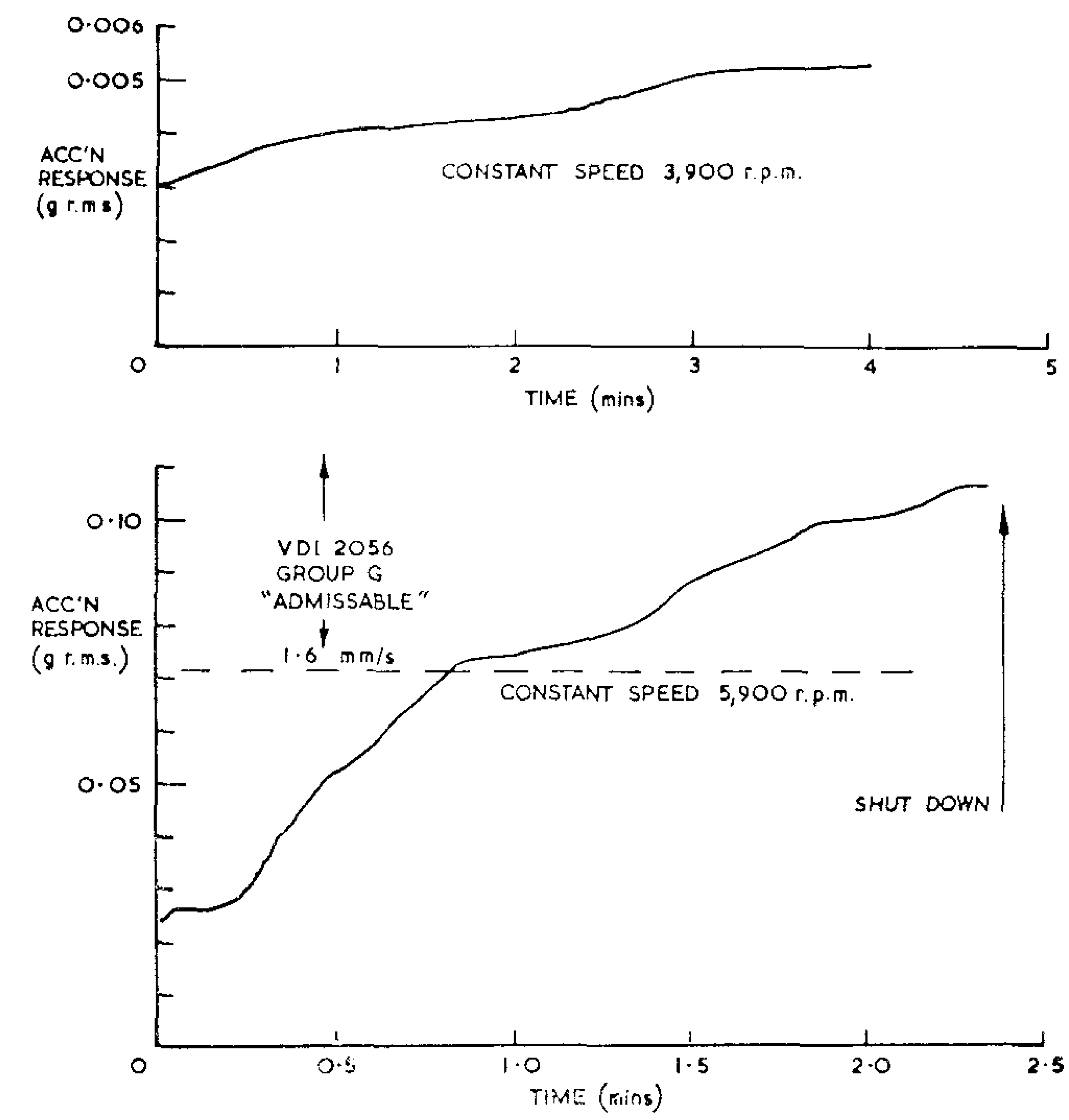

Fig. 4 Increase in acceleration response at D. E. casing support of compressor during bearing failure

and stresses at any point from the available measurements, but this is obviously impractical. It was considered, therefore, that a more practical approach would be to define the dynamic loads arising inside a machine, and to investigate how such loads would affect the levels of vibration at the monitoring points.

Typical cases illustrating the practical significance of these shortcomings in the vibration criteria are described in the following:

In one recent instance the bearing housing vibration levels on a new 10,000 hp, 6000 rpm centrifugal compressor, handling cracked gas from pyrolysis furnaces, were being monitored during commissioning. During a progressive stepwise increase in speed a slight increase in vibration level at a constant speed of 3900 rpm was noted (upper diagram, Fig. 4). Later, near full speed, the drive end bearing was clearly overheating and the compressor was shut down. It was subsequently deduced that the journal-type gas seal at this end of the shaft had been installed with clearances which were too small. The consequent overheating of the seal had caused the white metal in the seal and in the adjacent bearing to melt completely, leaving only the steel backing shell. However, the lower diagram in Fig. 4 shows that although a tenfold increase in vibration level, occurred consequent on this failure, the level was still within admissible limits according to existing, widely used criteria. In fact, previous impedance measurements on this compressor had shown that the bearing housings were extremely stiff, and, hence, the 'standard' criteria grossly underestimate the significance of a given vibration level.

A similar example is provided by the badly fouled 2200-hp, 13,500-rpm, back-pressure steam turbine rotor shown in Fig. 5. Here, maximum measured first-order (unbalance) vibration levels, before shut-down, were 3.4 mm/sec, which according to V.D.I. 2056 is entirely acceptable for this class of equipment. However, the steam consumption for rated power output had risen considerably, and after blade cleaning and subsequent rebalancing the corresponding vibration level was a maximum at only 0.7 mm/sec.

An even more critical case is illustrated in Fig. 6. This is a

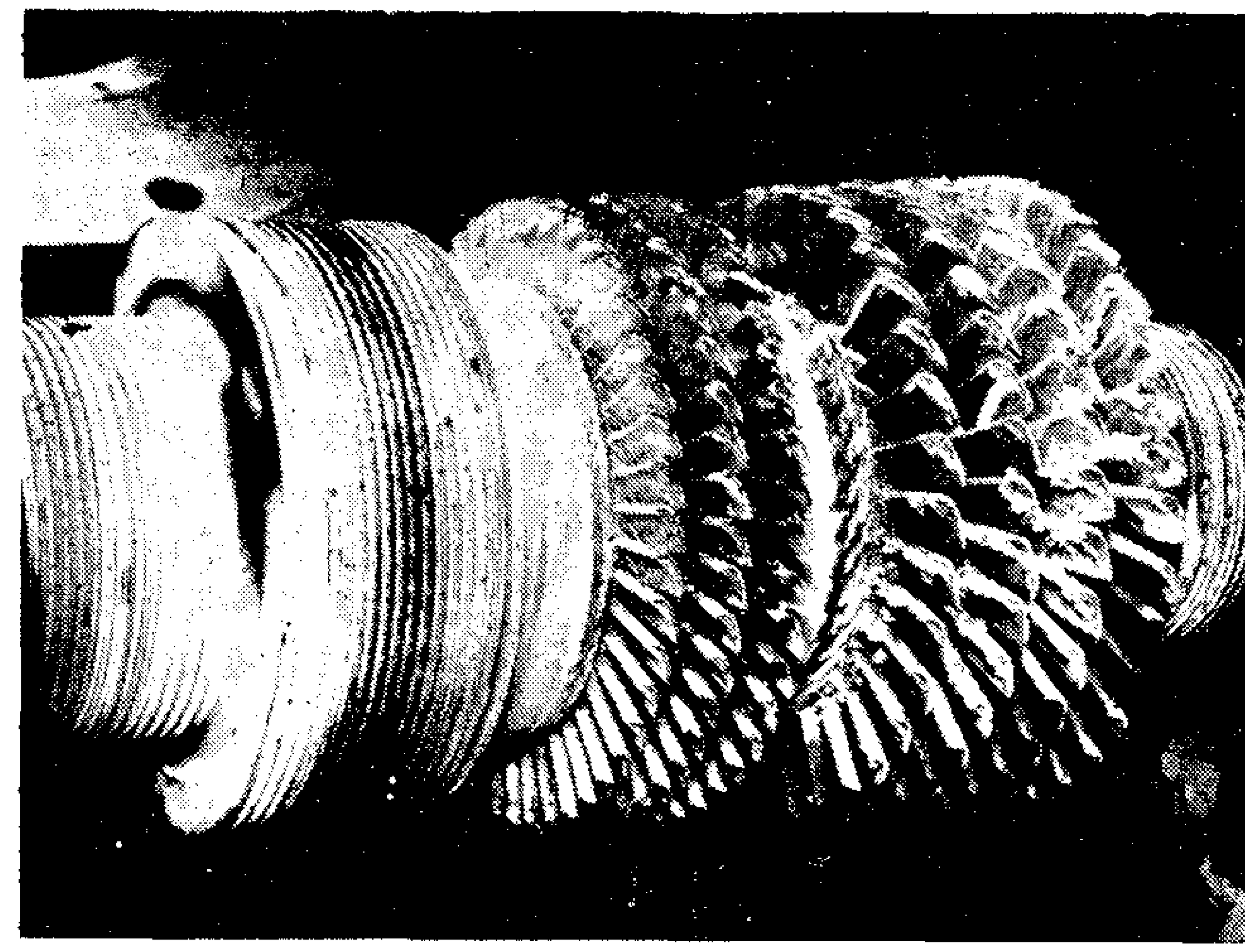

Fig. 5 Rotor-stator fouling in 2200-hp steam turbine

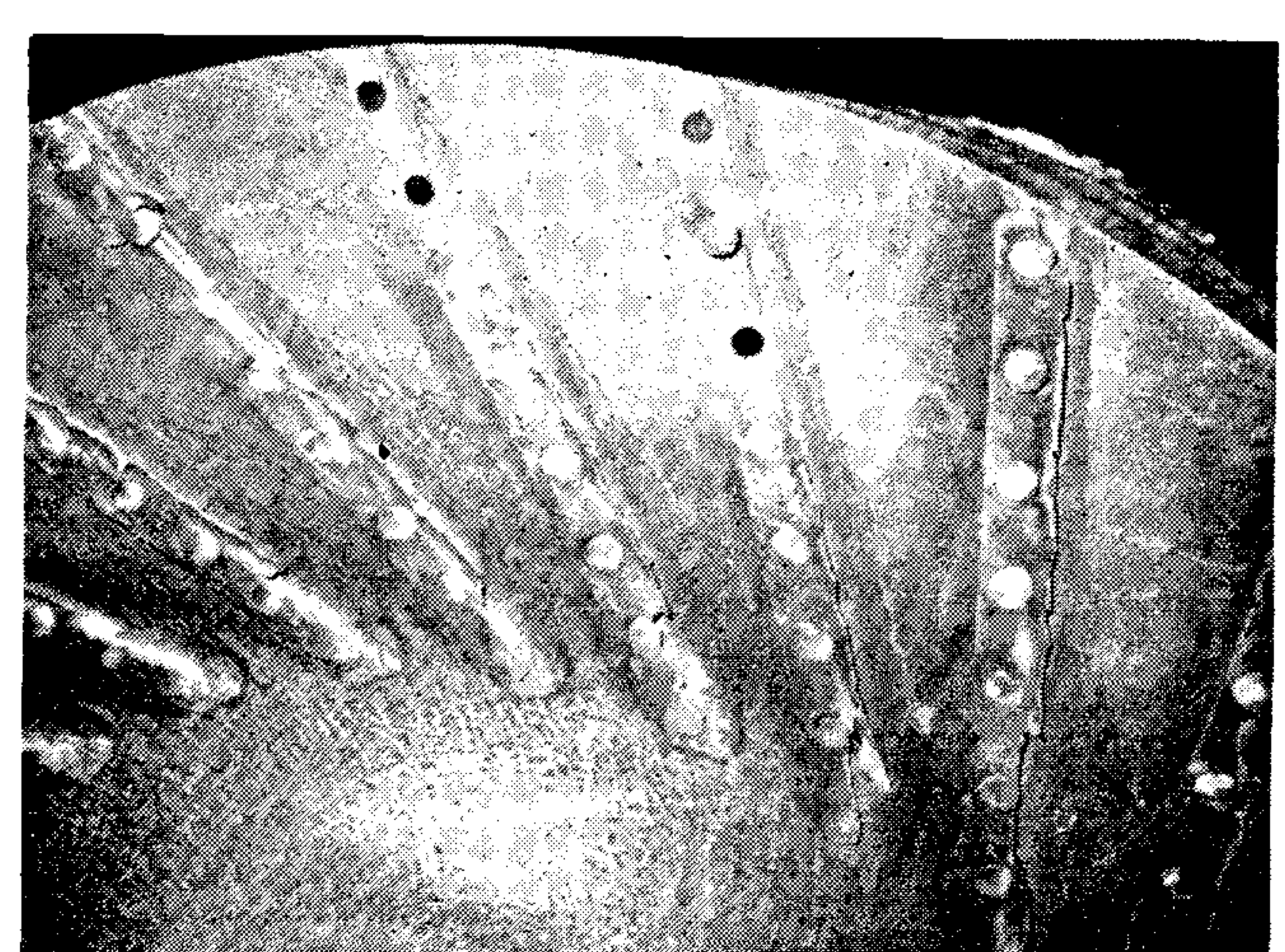

Fig. 6 Rotor failure in centrifugal compressor

view of one impeller of a 5000-hp, 7000-rpm centrifugal compressor which has suffered extensive erosion damage. The impeller was of riveted construction and one cover plate had become completely detached due to failure of the rivet heads and the vanes they affixed. At the time this failure occurred a marked drop in performance was, not surprisingly, observed, and vibration checks were made using hand-held, overall level-measuring equipment. Due to the lack of previous vibration history the levels, which according to the criteria were acceptable, were taken at their face value and the extent of the damage was not realized for some weeks when the compressor was shut down for routine maintenance. Here, again, the considerable unbalance was disguised by a massive, high-impedance casing construction.

The theoretical and practical evidence thus indicates that unless more information is available regarding particular machine, dynamic systems, the application of generalized criteria can be misleading. It is, therefore, important to obtain some knowledge of the response of a machine to particular dynamic forces. This knowledge can be most conveniently obtained, for bearing housing measurements, from a measurement of bearing housing impedances at the monitoring points which would enable the level of dynamic force to be computed directly from a measurement of vibration level under running conditions and, thus, help considerably in qualifying the relative severity of machine vibrations.

The Measurement of Mechanical Impedance

In order to investigate further the feasibility of using measured bearing support impedances in conjunction with corresponding vibration amplitudes as a method of assessing the acceptability of machine vibration, it was planned to investigate the dynamic properties of several turbine/compressor sets at the Shell Chemicals Plant, Carrington.

Due to the nature of the plant and the working fluids passing through the compressors, the use of standard harmonic excitation techniques to obtain dynamic impedance measurements on site was impracticable. Intrinsic safety regulations precluded the use of equipment requiring large electrical voltages, near the machines. Large electro-dynamic vibrators, apart from the problem of mechanical attachment to machines in the limited time available for tests during plant shut-down periods, could not, therefore, be used. Attention was, therefore, turned towards transient excitation as a practical means of measuring impedance of large structures in the field.

The theoretical basis for this approach is well known. In contrast to the Fourier series representation of periodic functions where components occur at discrete frequencies, the Fourier spectrum of a transient is a continuous function of frequency.

Further, if $f(t)$ and $r(t)$ are the input force transient and corresponding time history of the response, respectively, then the mechanical transfer function is given by $Z(\omega) = F(\omega)/R(\omega)$ where $F(\omega)$ and $R(\omega)$ are the moduli of the Fourier transforms of the input and output transients, respectively.

Thus, from the input and output time histories $f(t)$ and $r(t)$ it is possible to calculate the impedance over a wide frequency band. The only real problem, therefore, was to produce the input force transient in such a way as to cover the frequency range of the machines in terms of running speed and two or three multiples of this.

Extensive laboratory experiments were carried out to determine the optimum method for transient excitation. Devices ranging from a captive bolt gun to a 4-lb hammer with spring interface were tried, but for the type of machinery to be tested, the simplest and most effective exciter was found to be a simple pendulum hammer with a copper nose (Fig. 7). Hand hammers were equally effective, but relied on the operator to control the pulse force level, whereas the pendulus system gave controlled force level input from the swing angle. The input forcing function was thus obtained from an accelerometer attached to the rear of the hammer and the response function was obtained from an accelerometer mounted on the machine.

Fig. 7 Transient vibration excitation of machine frames

The complete technique was checked against the standard swept sine technique in the laboratory using a rotary convertor as a test vehicle, and Fig. 8 shows the level of agreement between the two methods used. Complete details of the transient technique and analysis methods have been reported elsewhere [5].

The method has many advantages over the swept sine method, particularly for field work. Once the pendulum was mounted in position it was only necessary to record the signals from two accelerometers by means of an FM tape recorder remotely situated, which required a minimum test time on site.

Analysis was carried out from digital ordinate sampling of the recorded signals, the Fourier spectra being obtained by means of a standard digital program.

Site measurements were obtained from fourteen machines at Carrington, the results of which have been summarized in Fig. 9 which shows the overall impedance envelope over the frequency range 20–500 Hz together with results published by other authors for comparison.

This curve illustrates better than anything else the danger of associating a single vibration level with machine performance. Although the machines investigated were of similar physical sizes and had similar bearing geometries, it is apparent that the impedances at the bearings vary in the ratio 1000–1.

However, once the impedances at the monitoring point are known it is possible to interpret changes in vibration level in a more realistic manner and, also, to modify operational criteria accordingly in relation to the dynamic stiffness of the bearings. With the experience gained during the experiments described above and in the light of the case histories also described, the next consideration in respect of vibration monitoring was to consider improvements in monitoring methods and data acquisition

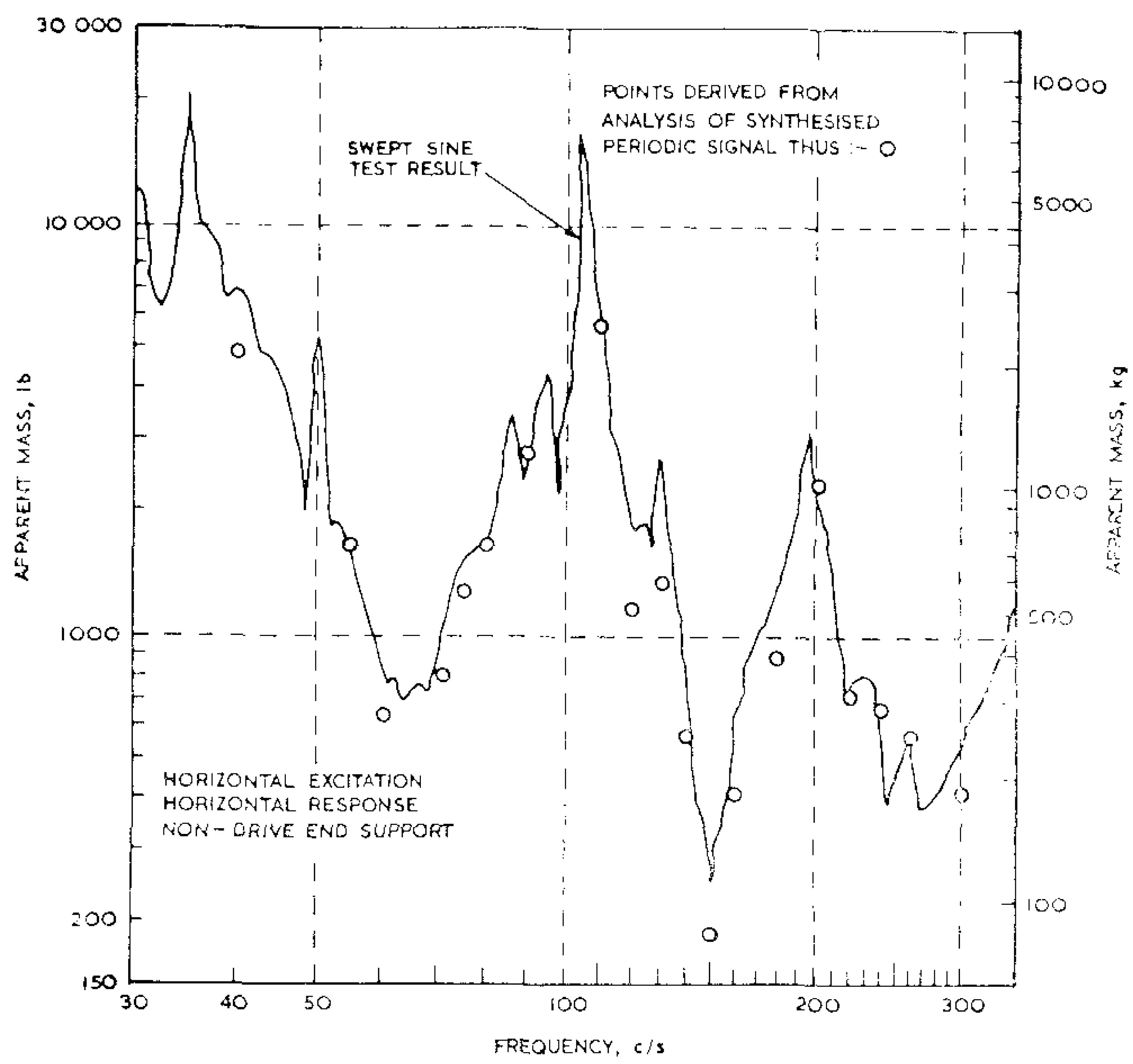

Fig. 8 Tests on rotary convertor; support impedance obtained from line spectrum of synthesized periodic acceleration response signal

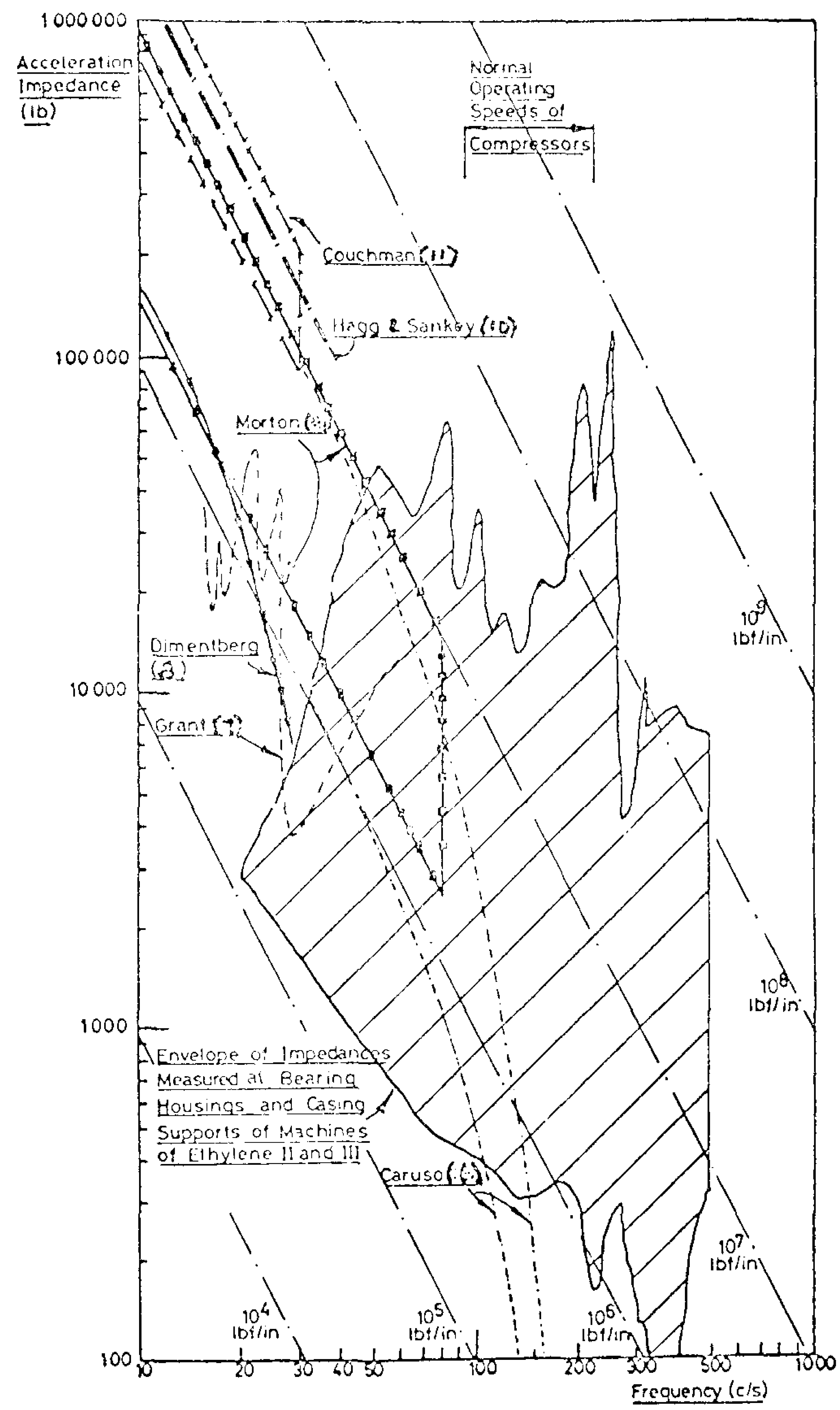

Fig. 9 Measured impedances of bearing support structures of machines of Ethylene II & III compared with values given by several authors for different machines

which would enable operators to obtain more than just an arbitrary alarm at preset vibration levels.

The Technique of Vibration Monitoring

Until recent years, the general technique for continuous vibration monitoring consisted primarily of a transducer detecting either bearing housing level or relative motion between shaft and bearing, ac-dc converter, and an alarm triggering system.

The control signal could therefore be a composite of the fundamental shaft frequency together with harmonics and subharmonics of unspecified number. Experience has shown that if an rms vibration level is composed of say three frequencies of similar amplitudes, the machine will subjectively appear "better" than would a similar machine with a vibration level of the same rms value but composed only of say a fundamental shaft rotation frequency.

This means, in effect, that monitoring on rms levels and ignoring frequency information can be misleading particularly when one is trying to assess the rate of deterioration of continuously operating plant. Modern systems for vibration monitoring have provided great opportunities for acquiring more data on the vibrational behavior of machines, and by making use of this facility it is possible to extend the use of such systems to include, in some cases, diagnostic and wear predictions to be made, as well as to provide day-to-day alarm monitoring.

It has been shown above that a monitoring procedure, in which overall vibration levels are measured and compared with existing vibration criteria, will frequently result in quite major failures going undetected. On the other hand we have irrefutable evidence that a more detailed examination of the vibration characteristics of rotating equipment is capable of providing very worthwhile returns.

In some cases, the introduction of a modest vibration inspection program reveals design faults in rotating machinery which have gone unrealized for years. For instance, on one ethylene plant, constructed in 1959, two separate instances of journal bearing instabilities in flexible-shafted centrifugal compressors have recently been detected. In the first case a 2500-hp, 10,450-rpm compressor had had a history of failed journal bearings, shaft seal failures, and consequent shaft damage. The first detailed vibration survey conducted revealed the presence of the journal instability, with a frequency of 0.47 of the shaft rotational frequency with a velocity level of 9.5 m/sec. At the next planned shutdown the bearings were changed from the existing plain cylindrical design to a tilting-pad type, which cured the instability. Nearly three years trouble-free running have now been obtained following the bearing modification.

In a second instance on the same plant, a 10,000-hp, 6000-rpm centrifugal compressor (see Fig. 10) had a similar history of bearing damage and, also, of shaft coupling hub fretting. Considerable effort had been expended in developing and applying new procedures for the hot alignment of the compressor and its turbine driver, since it was thought that this would solve the problem. Following recommissioning after a planned plant shutdown early in 1970, a vibration inspection program was initiated, the frequency analysis from which revealed the presence of a journal bearing instability ($^1/_2$-order whirl). So a simple vibration investigation immediately established the real reason for the original bearing damage and shaft coupling hub damage, which was one of straightforward bearing instability. However, it was not considered to be economical to shut down the plant for two

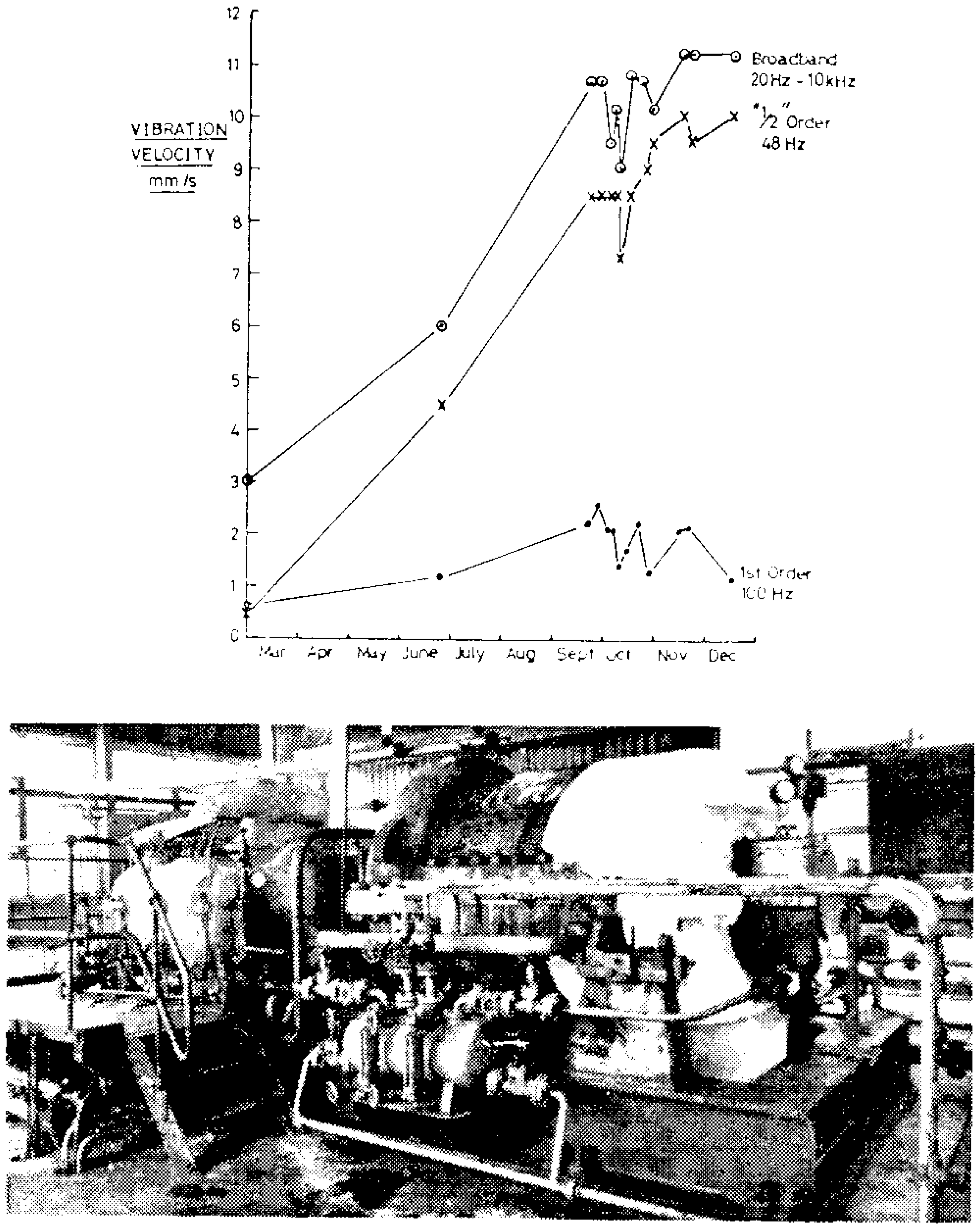

Fig. 10 The progressive effect of bearing instability on $^1/_2$-order vibration level on 10,000-hp centrifugal compressor

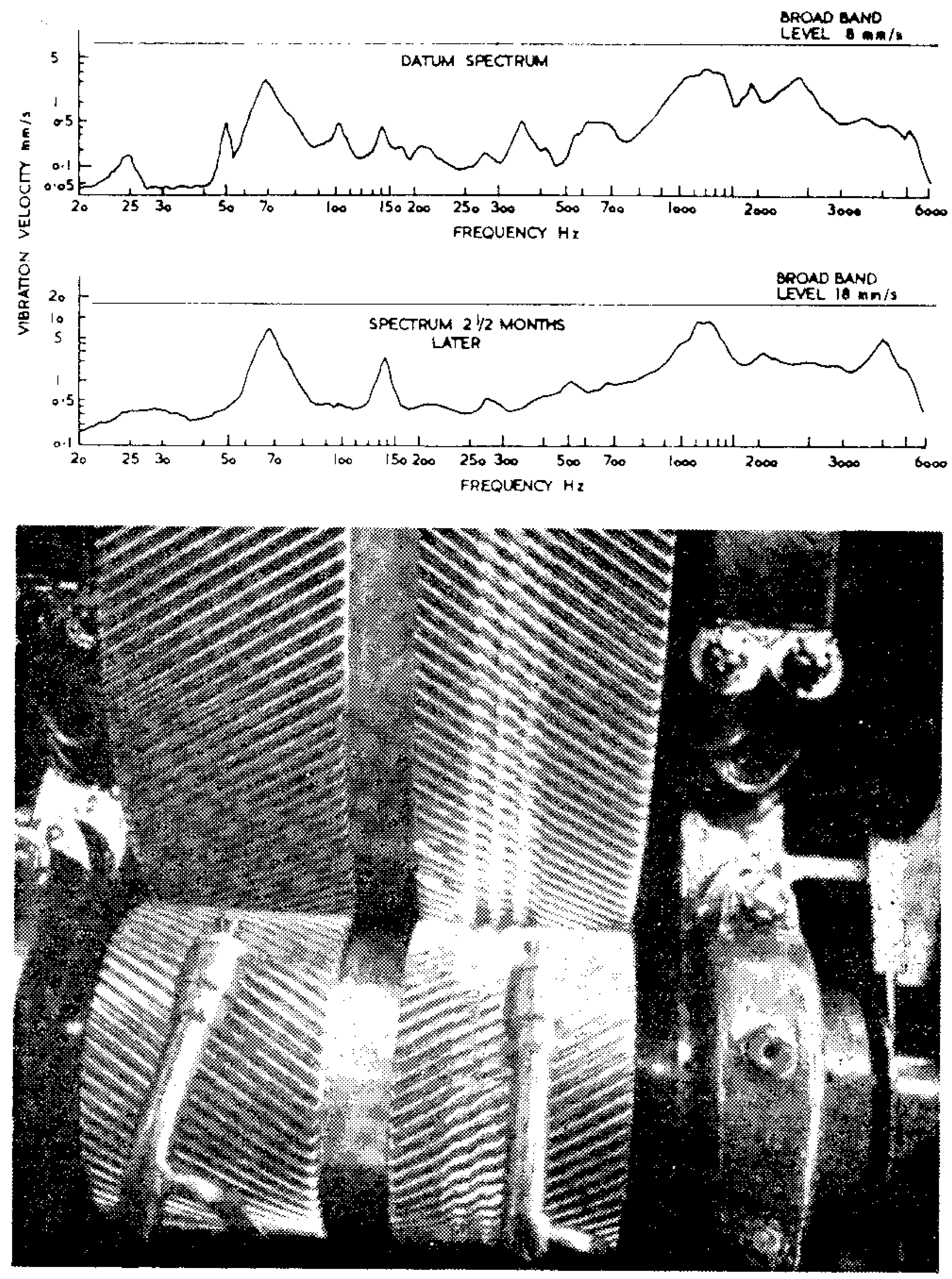

Fig. 11 Gear failure in 1000-hp gearbox due to drive coupling wear

weeks to modify the bearings. It was decided, therefore, to run the machine with weekly checks on the $^1/_2$-order vibration and, thereby, monitor the possible bearing deterioration with a view to running the whole plant until the next planned shutdown, or at least to establish the critical $^1/_2$-order vibration levels for this machine immediately prior to failure.

Fig. 10 shows the increase in $^1/_2$-order vibration to the present time and, although the vibration level has increased dramatically, the machine is still on line.

The potential advantages of vibration monitoring are also demonstrated in the following case during 1970 when the results of regular vibration analyses on the gearbox of a 1000-hp, 1500-lb_f/in^2 boiler feed pump showed an increase from 8 mm/sec broadband to 18 mm/sec in some $2^1/_2$ months running (Fig. 11). Apart from a rise in first-order levels, a significant spectral peak had appeared at about 1250 Hz and another around 4000 Hz, subsequently diagnosed as flexible coupling tooth meshing and gear meshing frequencies. The prime cause was a badly worn drive coupling, which in turn imposed on the gearbox radial loads for which it was not designed. Two months after the last analyses shown in Fig. 11 the gearbox failed, some of the damage being shown in the lower photograph. During this period the broadband level increased to approximately 28 mm/sec.

In the similar example shown in Fig. 12, the lower photograph shows an electric motor, gearbox, centrifugal compressor combination. The double helical gearbox, transmitting 500 hp, acts as a speed increaser from the motor speed of 3000 rpm (50 Hz) to the compressor speed of 13,200 rpm (220 Hz). During a routine vibration inspection a high level (up to 16 mm/sec), high-frequency (1750 Hz) vibration was apparent on the gearbox pinion

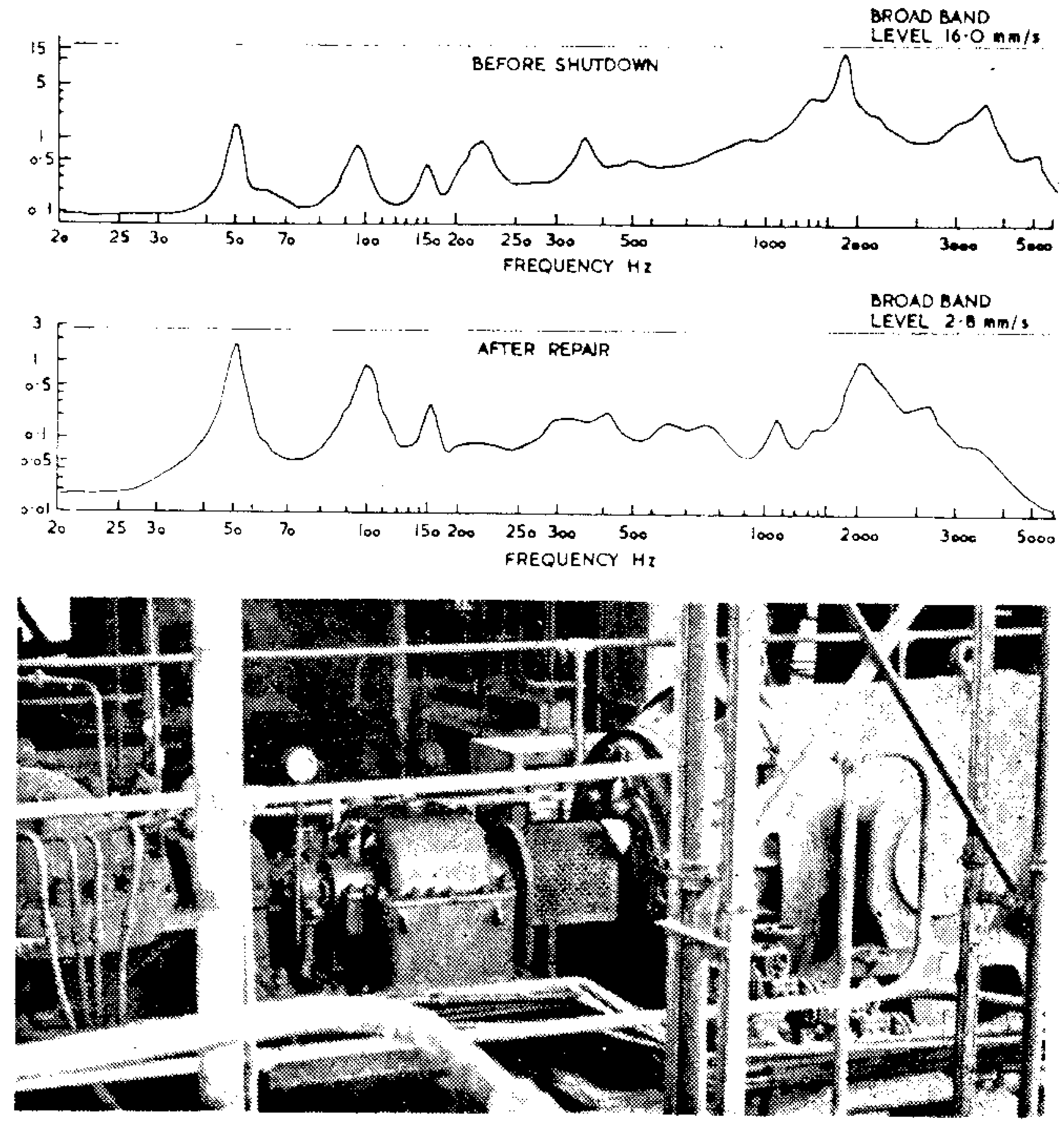

Fig. 12 Effect of gear tooth wear on casing vibration spectra, 500-hp gearbox

bearings as shown in the upper spectrum of Fig. 12. This vibration was apparently a lower sideband of the gear meshing frequency and it was advised that the gearbox be shut down and repaired. This advice was acted upon and, following the installation of a new pair of gearwheels, the broadband level of vibration was reduced to 2.8 mm/sec (lower spectrum Fig. 12), while the 1750-Hz peak was no longer evident.

The significant factor in these investigations is that by extending the range of routine measurements to include frequency response information it is possible to obtain a much clearer view of the behavior of machines over long periods of continuous operation. Unnecessary shutdown of continuous process plant can be very costly with regard to loss of production. It is considered, therefore, that reducing what used to be regarded as a specialized investigation to a routine procedure will enable operators to operate plant in which component deterioration is evident, to the limit of time before enforced shutdown or until some convenient time when planned shutdown has been arranged.

It should be a realizable aim, with the knowledge obtained prior to shutdown, to insure that replacement parts are available in advance, thus avoiding delays during the overhaul period.

Instrumentation for Vibration Monitoring

When deciding on the policy to be adopted regarding vibration monitoring, the first pertinent question to be answered is "Against what type of failure or maloperation are we primarily providing protection?" With some sort of answer to this question the relevant choice of instrumentation becomes more apparent. In the case of an oxygen compressor, for example, where the primary concern is one of safety, the monitoring equipment must be capable of detecting any reductions in shaft/casing clearance (dynamic or otherwise) and the continuous on-line surveillance of this parameter is essential together with automatic trip facilities.

For normal spur gearboxes, shaft, displacement measuring equipment is of little value due to the small tolerance on depth of mesh and, incidentally, the high meshing frequencies normally involved. Here, therefore, bearing housing monitoring over a wide frequency band is more appropriate.

In the case of a stiff-shaft machine, however, with generous clearances, dynamic bearing loading due to unbalance may be considered the most probable fault and either the continuous monitoring of bearing housing vibration level with an alarm level based on a measured housing impedance, or the use of shaft proximity probes with a displacement level alarm based on known journal bearing properties are equally appropriate.

The decision on whether to monitor continuously or by means of regular inspection is also dependent on the type of failure anticipated. It is also clearly dependent, from a cost effectiveness viewpoint, on the capital cost of the equipment protected and on the probable effect on plant production of this equipment being unavailable. In this context, the main compressors and their drivers on large single-stream process plant are undoubtedly the surest candidates for continuous monitoring.

Equipment like boiler feed pumps, which are normally duplicated, is more difficult to categorize, although the initial expenditure on the installation of continuous monitoring equipment is often justified due to the wide dependence of the works on uninterrupted steamsupply.

Even where continuous monitoring is decided upon it is still worthwhile continuing with regular examinations of the vibration spectra (say once monthly) for trend analysis. Much more reliable information on 'normal' and 'abnormal' vibration levels is then obtained, which in turn means further refinement in the setting of alarm levels.

Many organizations now market instrumentation specifically for vibration monitoring, both for continuous 'on-line' operation and for routine inspection. Unfortunately, it appears that many manufacturers are unaware of the frequency, amplitude, or resolution required for particular applications. It is, therefore, essential to be very clear on the precise characteristics required.

The initial choice is between shaft proximity probes and bearing housing 'absolute' vibration measurement. Where the prime function is to warn against loss of working clearances then proximity probes are an obvious choice. A ready-made criterion for alarm or shutdown also exists by comparison with known design clearances.

Where dynamic loading at high frequency is the prime concern (e.g., gearboxes) then bearing housing measurements are more appropriate. Velocity transducers may be used but they have a limited frequency range and are easily damaged by rough handling. Much more reliable are piezoelectric accelerometers with their wide frequency range and robust construction. The authors' preference is for a combined accelerometer and emitter follower, 'potted' as one assembly, which enables normal, robust cabling to be used. The latter helps to remove a major cause of lack of reliability. Cable lengths up to 500 m and frequency response up to 10 kHz are readily achieved using this procedure.

For hazardous-area applications Zener barrier techniques are simply applied to prevent any possibility of excessive electrical energy being dissipated in the hazardous area.

However, it is dangerous to generalize with respect to monitoring systems and it is better to treat each case individually. In fact, for complete reliability a dual system consisting of shaft probes and accelerometers on bearing housings would be advantageous, particularly when interpreting level changes over long periods where the effect of internal forces can affect different systems in different ways.

For example, a bearing test rig in the university laboratory was designed to investigate oil film dynamics. The bearing load is applied pneumatically, the motion of the journal in the bearing is measured by inductive probes and the bearing housing vibration is monitored by accelerometers. We noticed that when the load was applied to the shaft the journal motion decreased as the oil film thickness reduced and the vibration level on the housing increased due to increase in transmissibility of the dynamic forces through the stiffer oil film.

If this bearing had been part of a coupled system one can imagine that a transfer of bearing load due to say foundation level changes between machines would appear as an improvement in vibration on a shaft probe system, but a worsening of the vibration with a housing vibration monitoring system.

The latter system, in this case, gives the desired indication as the tendency would be to accelerate bearing failure due to increased loading.

Conclusion and Recommendations

There is undoubtedly a case for a more sophisticated approach to routine monitoring of machine vibrations in respect of continuously operating process plant where even short periods of unscheduled shutdown can be very costly in lost production.

For continuous alarm monitoring it is important to choose the monitoring system most suitable to the type and function of the machines in use and to consider the composition of the monitoring signal with respect to significant frequencies and harmonics of rotational speed.

In establishing viable vibration criteria, it is also possible to include information regarding the dynamic response of particular machines and to relate criteria to dynamic force levels.

Periodic checks on vibration spectra can produce valuable information with respect to malfunction and wear diagnosis, as well as assisting in the establishment of good operational criteria.

However, experience has shown that the problem of digesting the considerable amount of data which can be obtained requires special consideration. For plant containing large numbers of machines a solution to this problem is to computerize the data, a procedure which is at present being developed by S.C.U.K.L.

Using standard equipment and techniques for vibration measurement and analysis, it is relatively easy to reduce what has usually been considered to be a research approach to vibration problem diagnosis to a routine procedure requiring a minimum of specialist staff to operate the system. It has been established that the routine collection of data on the vibration response of plant considerably simplifies the diagnosis of reasons for component failure as well as providing indications of wear and deterioration of process plant.

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