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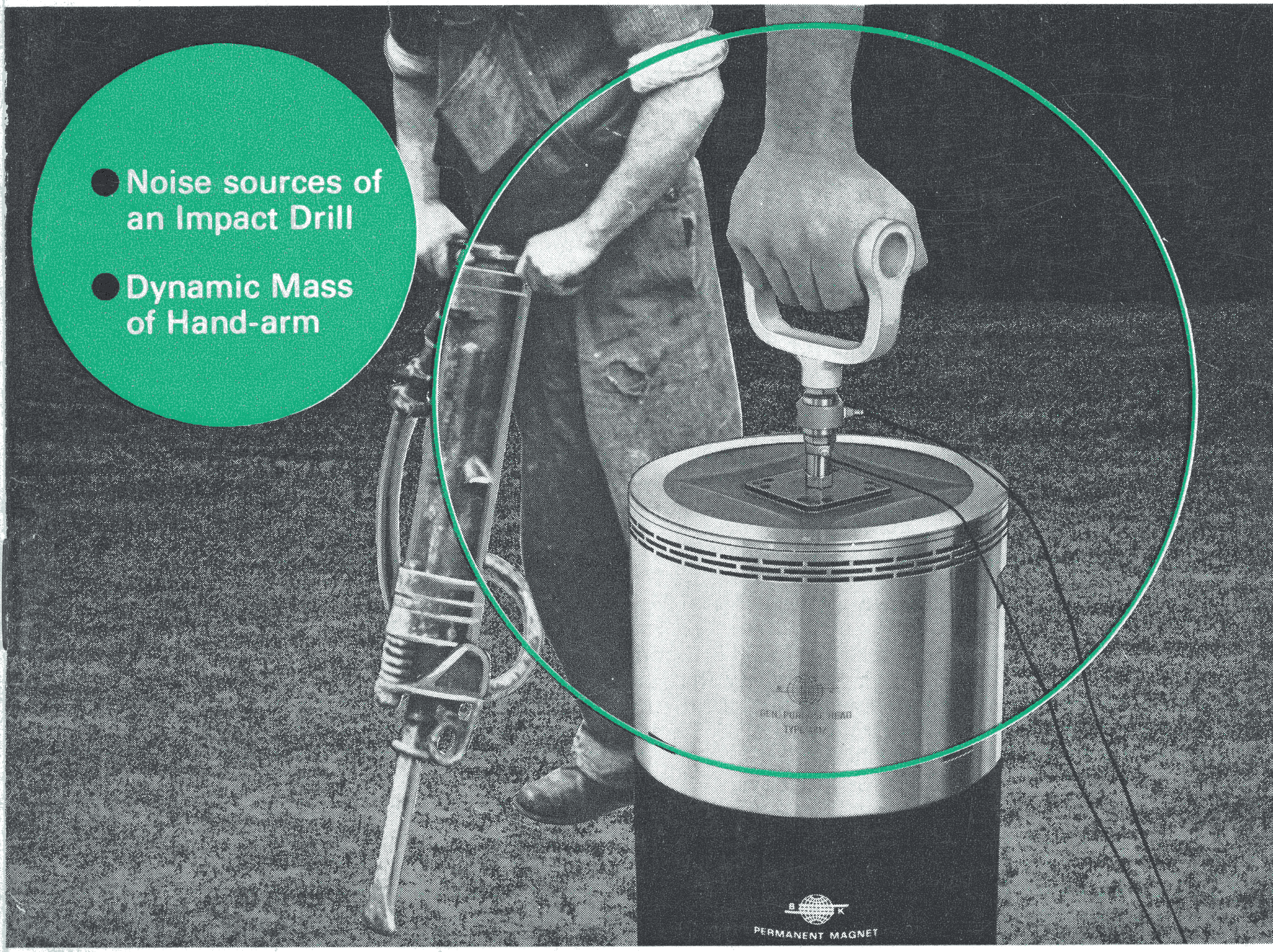

- Noise sources of an Impact Drill
- Dynamic Mass of Hand-arm

Brüel & Kjær

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Acoustical Investigation of an Impact Drill

by

K. Zaveri, M. Phil

ABSTRACT

Acoustical investigation of an impact drill was carried out to determine its main sources of noise by the use of narrow band frequency analysis technique. Sidebands around tooth meshing frequencies, indicating malfunction of gears and pinpointing sources of noise, have been determined also with the use of cepstrum Analysis — a technique especially useful for separating different modulation frequencies.

SOMMAIRE

Une étude acoustique a été effectuée sur une perceuse à percussion pour en déterminer les principales sources de bruit. Pour ce faire, on a utilisé l'analyse spectrale à bande étroite. Les bandes latérales autour des fréquences des engrenages, signes d'un mauvais fonctionnement des réducteurs et permettant de localiser les sources de bruit, ont aussi été étudiées par l'analyse du "cepstrum", technique particulièrement utile pour séparer différentes fréquences de modulation.

ZUSAMMENFASSUNG

Akustische Untersuchungen an einer Schlagbohrmaschine wurden durchgeführt, um die Hauptgeräuschquellen mittels schmalbandiger Frequenzanalyse aufzuspüren. Bestimmte Getriebefehler, die Seitenbänder zu den Zahneingriffsfrequenzen verursachen, ließen sich mit der Cepstrum-Analyse feststellen — einer Technik, die ganz besonders zur Trennung verschiedener Modulationsfrequenzen geeignet ist.

Introduction

Pinpointing sources of noise in the offending machinery and modifying the designs in it's early stage of development, have often proved in practice to be far more economical than taking remedial measures in the form of isolation and damping treatments. One of the diagnostic tools at the hands of an acoustic and vibration engineer is the use of a narrow band frequency analysis technique whereby the significant frequency components in the spectrum can lead to the identification of noise sources in the machinery. This technique was utilized in the investigation of an impact drill, where the identification of the components causing the most noise was desired.

When considering the acoustical investigation of the drill it is important to keep the operating conditions of the drill constant as far as possible, to obtain a stationary noise signal. It was observed that as the voltage supplied to the drill was varied, the speed of the drill was seen to be affected which not only altered the amplitude of noise but also the frequency structure of the noise spectrum. Other factors that affect the overall sound level are the kind of tool used (whether drill bit or chisel) and the kind of loading presented to the drill (whether drilling in concrete, wood or metal). It is therefore pertinant to state the operating conditions of the drill for every measurement.

The drill is operable basically in two modes; drill and impact, and impact only. During the experiments the drill was operated under free running conditions without the tool bit and then in the impact mode only with the drill and chisel bit successively.

Measurement Procedure

A cross-section of the drill is illustrated in Fig.1 where the wheel and pinion shown, have 44 and 9 teeth respectively. In order to relate the frequencies in the noise spectrum to the meshing frequencies of the gears, it is imperative to determine the rotational speed of the rotor with reasonable accuracy. To achieve this a non-contact Magnetic Transducer Type MM 0002 was mounted on the drill casing as shown in Fig.1. A thin strip of metal of very high permeability was attached on one of the blades of the rotor fan. Each time the strip passes the magnetic transducer, a pulse is generated and passed on to a frequency counter via a Motion Analyzer Type 4911 to measure the rotational speed of the rotor. Since it is difficult to keep the rotational speed of the drill constant over a sufficient length of time to carry out a sequential analysis of the sound signal, use was made of a Digital Event Recorder Type 7502 where a part of the sound signal is recorded while the speed of the rotor is simultaneously noted down from the frequency counter. The signal sample is rotated in the memory of the Digital Event Recorder and appears as a continuous signal at its output which is later analyzed by means of a constant bandwidth Heterodyne Analyzer Type 2010 shown in Fig.2. Other advantages with the use of the Digital Event Recorder are that large frequency transformations are possible by which saving in time of narrow band frequency analysis is achievable.

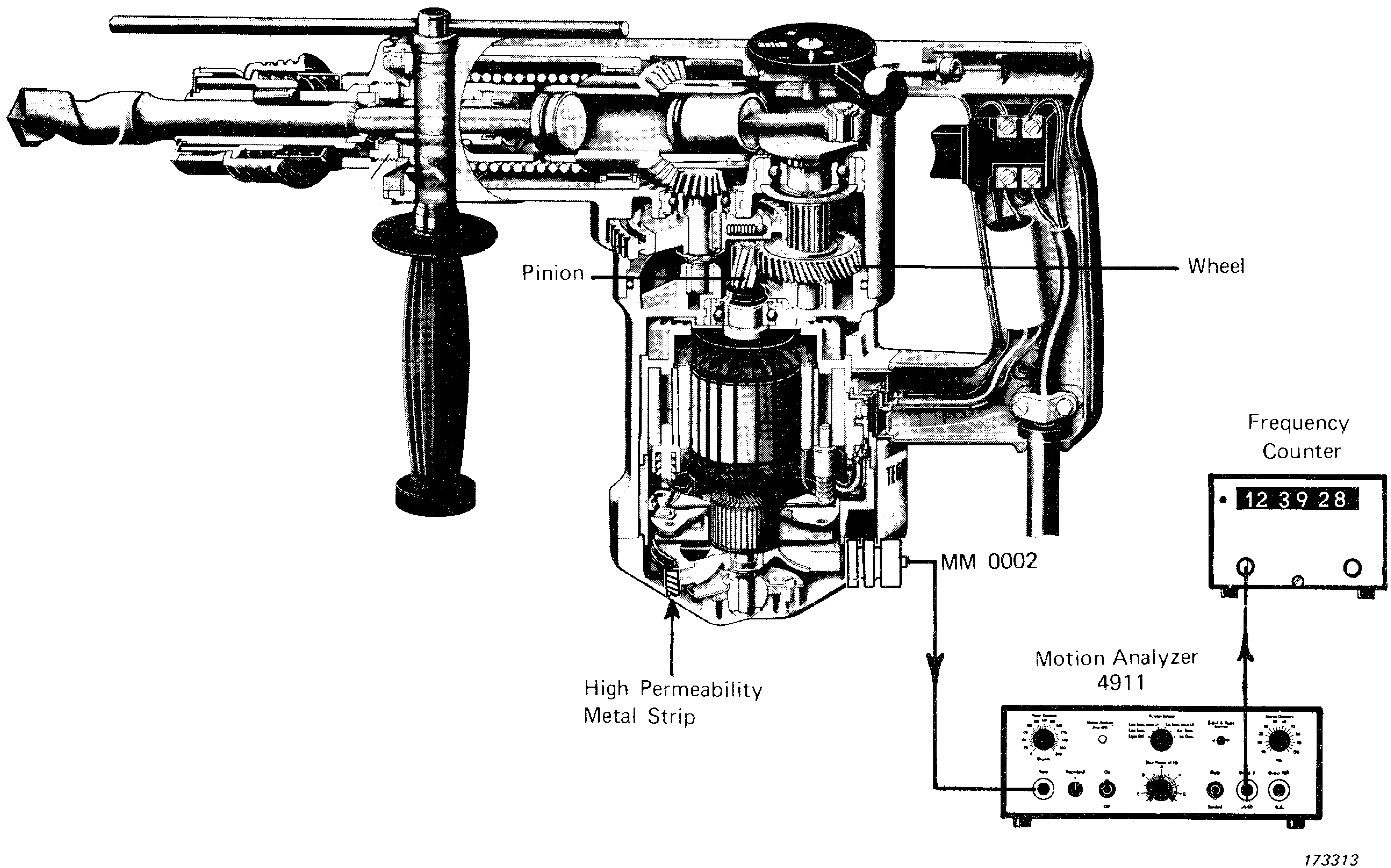

Fig.1. Inside view of Drill

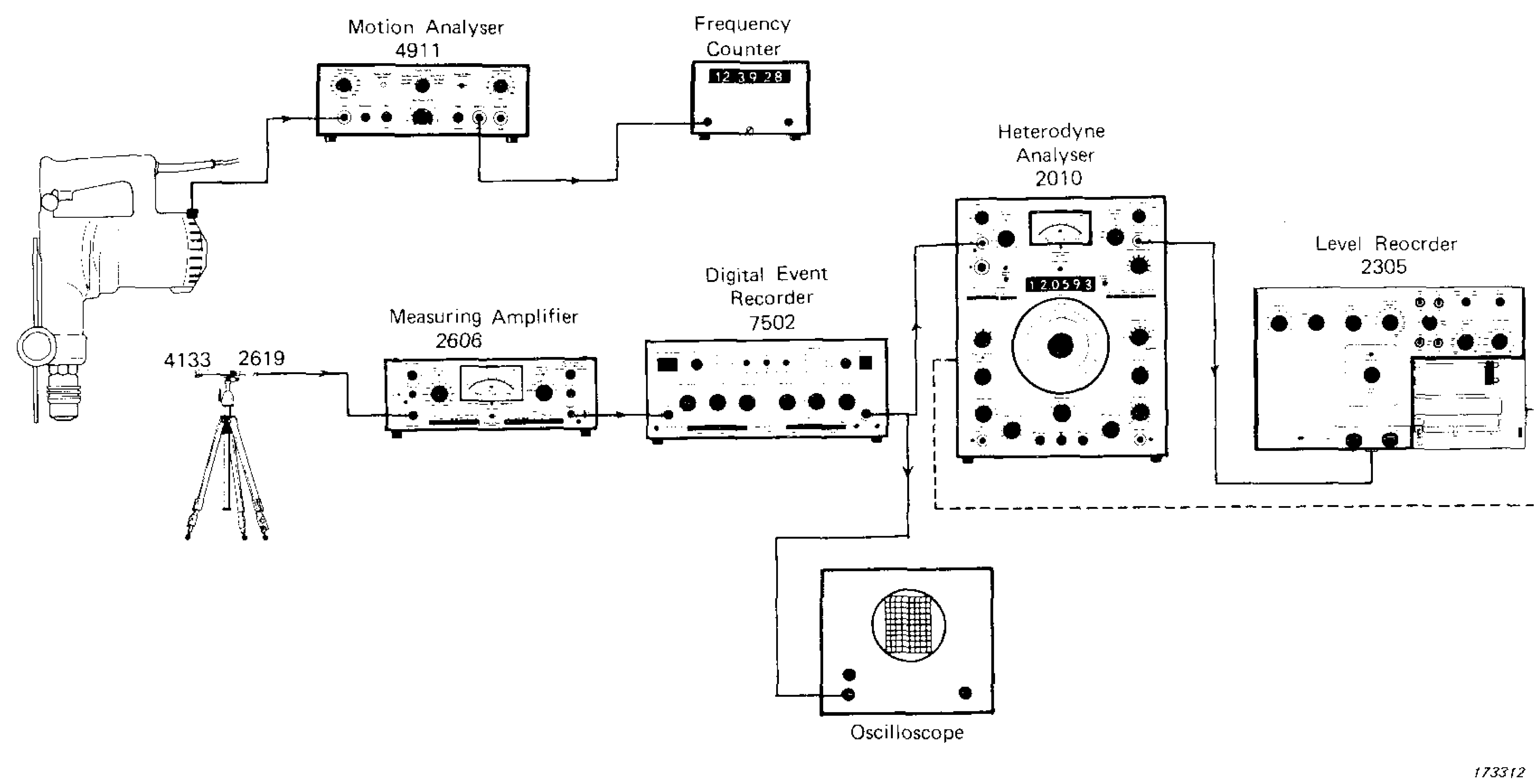

Fig.2. Recording and Analysis Instrumentation Set-up

a) Drill operated under free running conditions
As a preliminary investigation, the drill was made to run free without a tool at 220V. A half-inch microphone was placed at 1 meter distance from the drill with no reflecting surfaces close to the drill or the microphone, apart from the floor. A sample of the sound signal was recorded using the apparatus shown in Fig.2 and the speed of the drill rotor was noted to be 296Hz. A 10Hz bandwidth frequency analysis was carried out giving a spectrum shown in Fig.3 and the overall noise level was measured to be 93dB. Since the spectrum was to be used for diagnostic purposes a linear frequency sweep was carried out.

The pinion on the rotor shown in Fig.1 has 9 teeth. This gives a tooth meshing frequency f_c of 296 x 9 = 2664Hz which is observed in the spectrum*, though the amplitude at this frequency is not predominant. On the other hand, if pinion eccentricity causes it's teeth to be driven into and away from the wheel teeth, fluctuations result on the load between the pinion and wheel. The amplitude of tooth meshing noise is increased and decreased giving an amplitude modulation process, see Ref.[1]. When complex modulation process occurs, sidebands are generated around the meshing frequency which may have larger amplitudes than those at the meshing frequency. This is so, as can be seen from Fig.3 where the first lower sideband is the most predominant one given

* In the spectrum the peak is marked as 2660 Hz. Better resolution cannot be achieved by a 10Hz constant bandwidth filter and especially when the filter is being scanned.

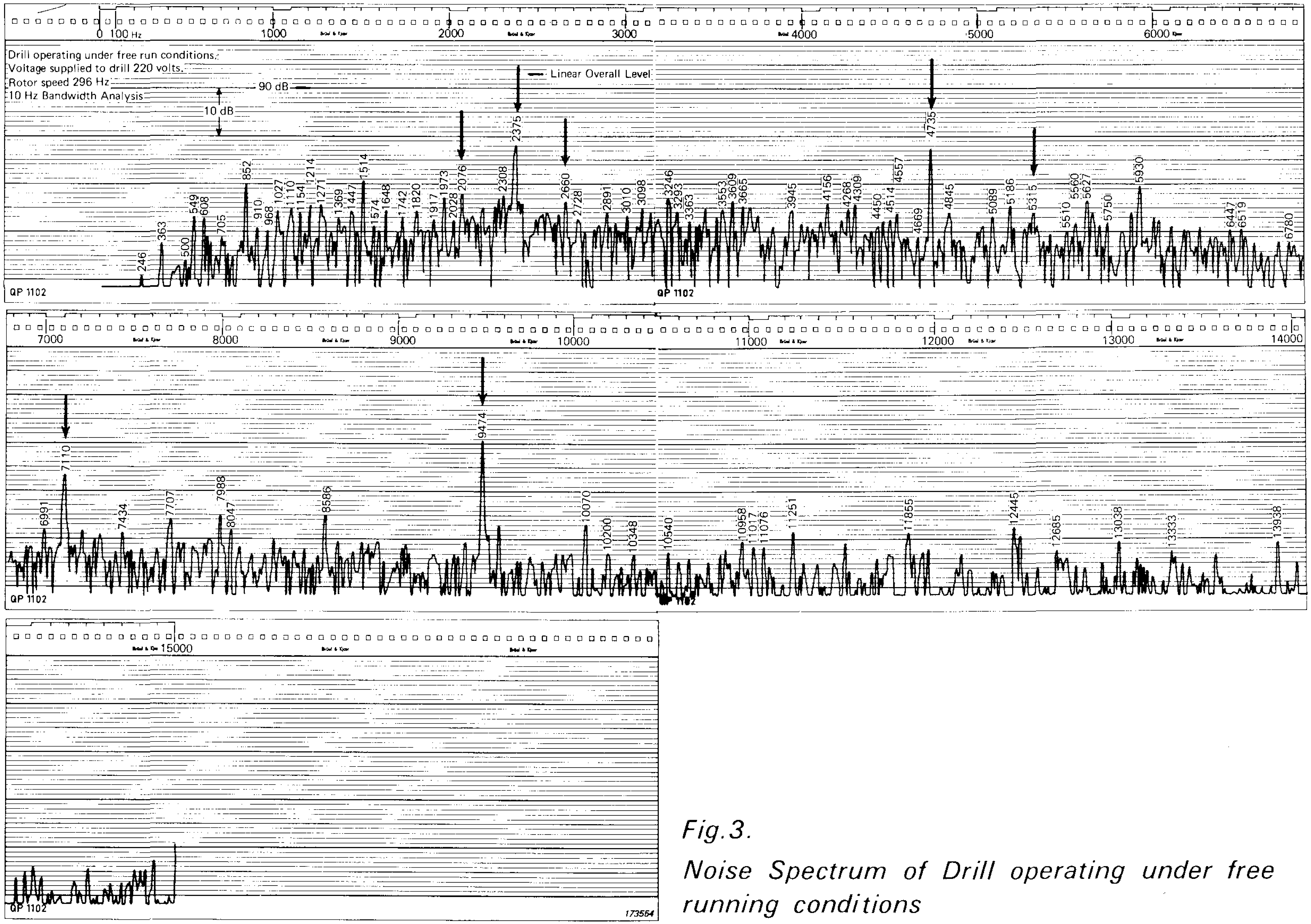

Fig.3.

Noise Spectrum of Drill operating under free running conditions by $(f_c - f_p) = 2664 - 296 = 2368$ Hz where f_p is the pinion rotational frequency. It should be pointed out here, that although 2368 Hz is also the eighth harmonic of the rotational frequency (8 x 296 = 2368), the significance of this frequency is associated with the sideband as the causes of its predominant existance is due to the pinion. The second lower sideband is given by $(f_c - 2f_p) = 2664 - 592 = 2072$ Hz which can also be seen in the spectrum, though its amplitude is not as significant as the first lower sideband.

The second harmonic of the tooth meshing frequency is given by $f_c \times 2 = 2664 \times 2 = 5328$ Hz which is not significant in the spectrum and neither is the first lower sideband given by $(2f_c - f_p) = 5328 - 296 = 5032$ Hz associated with the second harmonic. On the other hand the second lower sideband is quite predominant given by $(2f_c - 2f_p) = 5328 - 592 = 4736$ Hz.

Similarly, the third lower sideband, 7104 Hz, associated with the third harmonic of the meshing frequency and the fourth lower sideband, 9472 Hz associated with the fourth harmonic, are the most significant components.

In other cases similar sidebands could also exist for the wheel frequency (given by meshing frequency ± wheel frequency) but are not found to be significant in this spectrum.

The amplitude modulation theory applied here has been adopted from electrical technology and has been described in references [2 & 3].

b) Drill operated under loaded conditions

The drill was now operated with the drill bit in concrete in the impact mode only. The sound signal was again recorded from a microphone at 1 meter distance from the drill using the same apparatus as shown in Fig.2. The spectrum obtained by a 10 Hz bandwidth analysis is shown in Fig.4. When the recordings were made, two factors were immediately noticeable that were different from those when the drill was operating under free running conditions. They were, firstly that the speed of the drill was considerably reduced for the same voltage and secondly, the overall noise level was much higher. From the spectrum, Fig.4, the peaks at 12150 Hz, 17620 Hz, and 17960 Hz are seen to be most significant, however, those at 14540 Hz and 14970 Hz should not be overlooked. The overall sound level was also measured and was found to be 117 dB. In order to determine where these five frequencies were generated, it was decided as a first step to determine the natural fre-

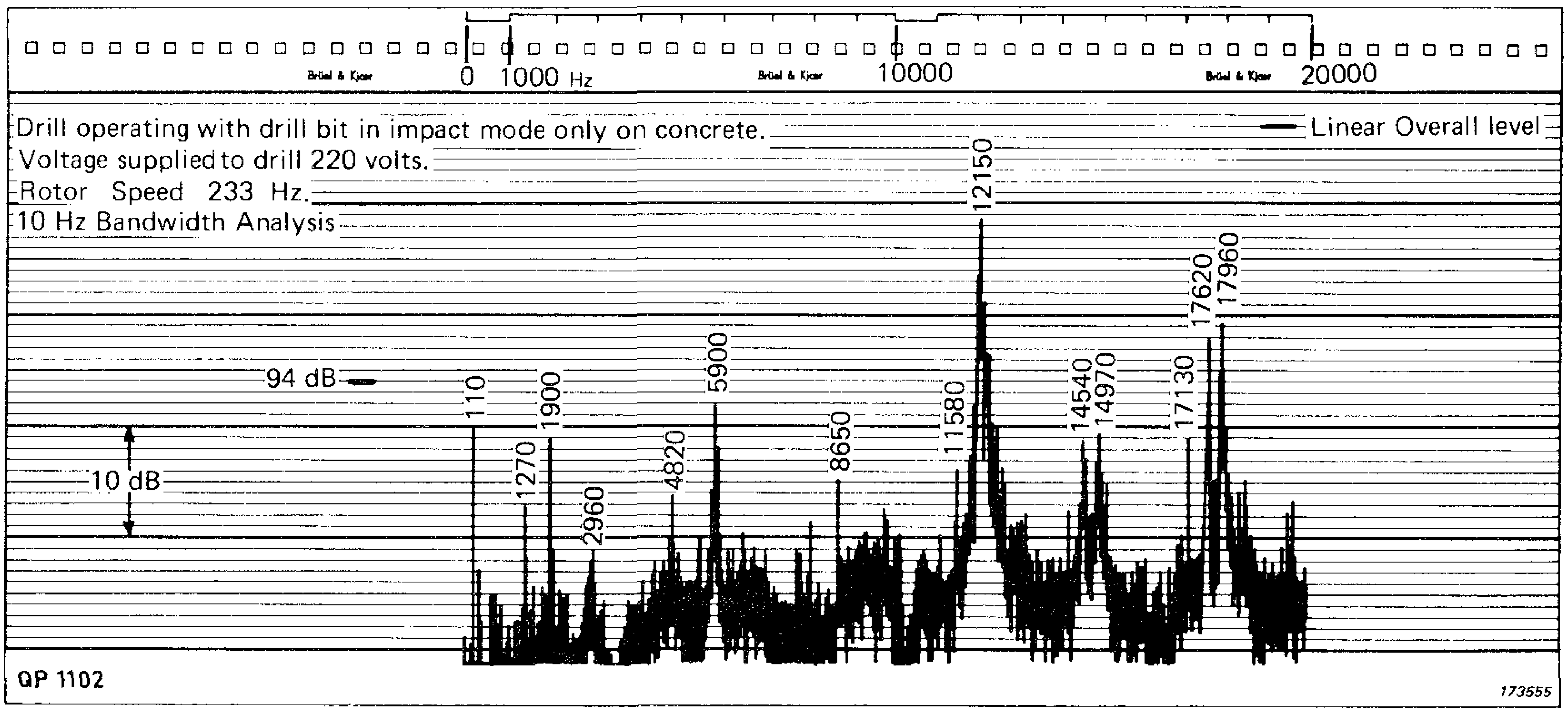

Fig.4. Noise Spectrum of Drill operating with drill bit in impact mode only, in concrete

quencies of the drill bit. This was carried out by placing the drill bit in concrete and tapping it with a hammer as shown in Fig.5. The impact sound signal was recorded and analyzed, the spectrum of which is shown in Fig.6. Here the natural frequencies of the drill bit are excited and are found to be approximately the same as in Fig.4, namely 12180 Hz, 14590 Hz, 14970 Hz etc.

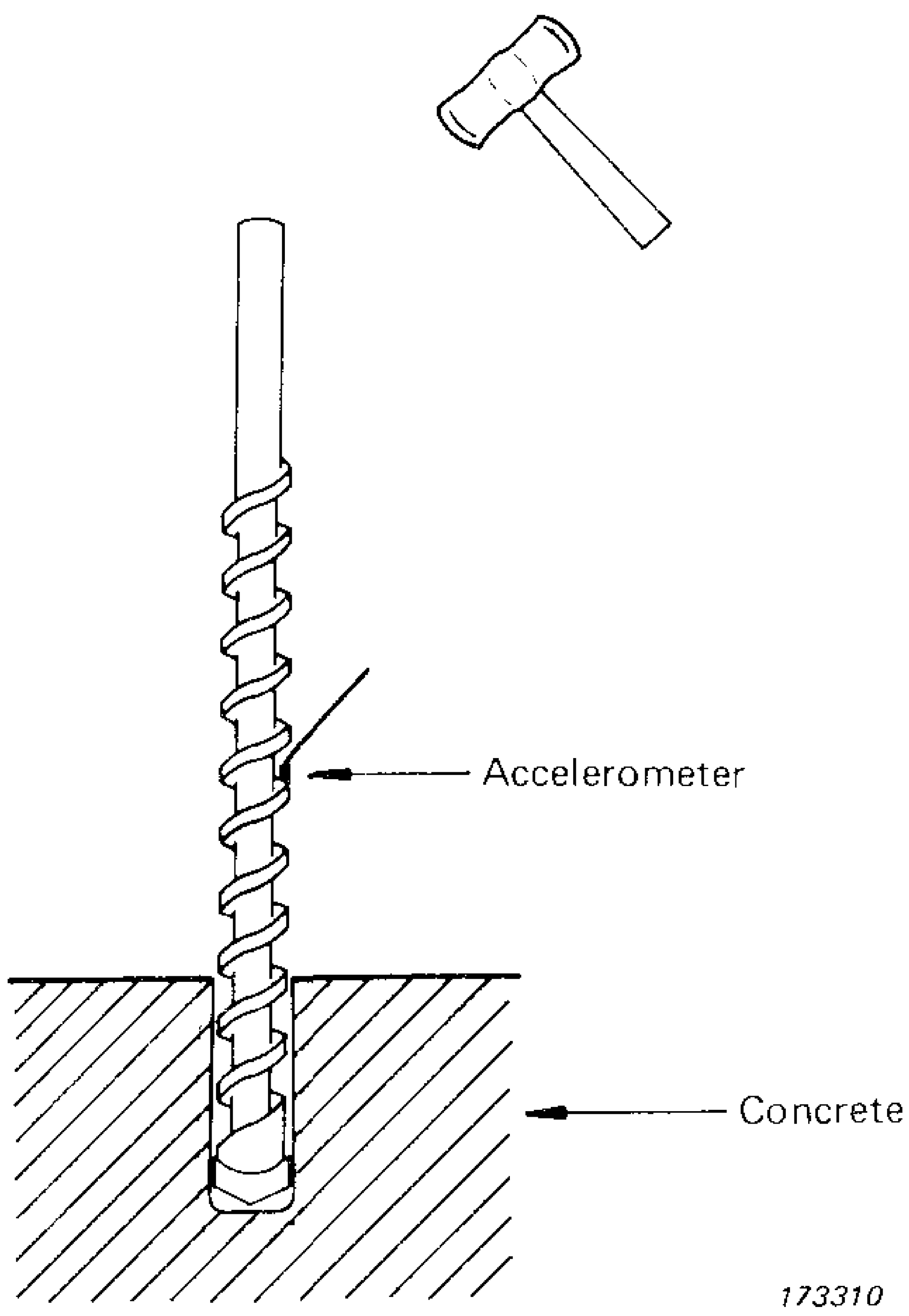

Fig.5. Method of Excitation of natural frequencies of drill bit

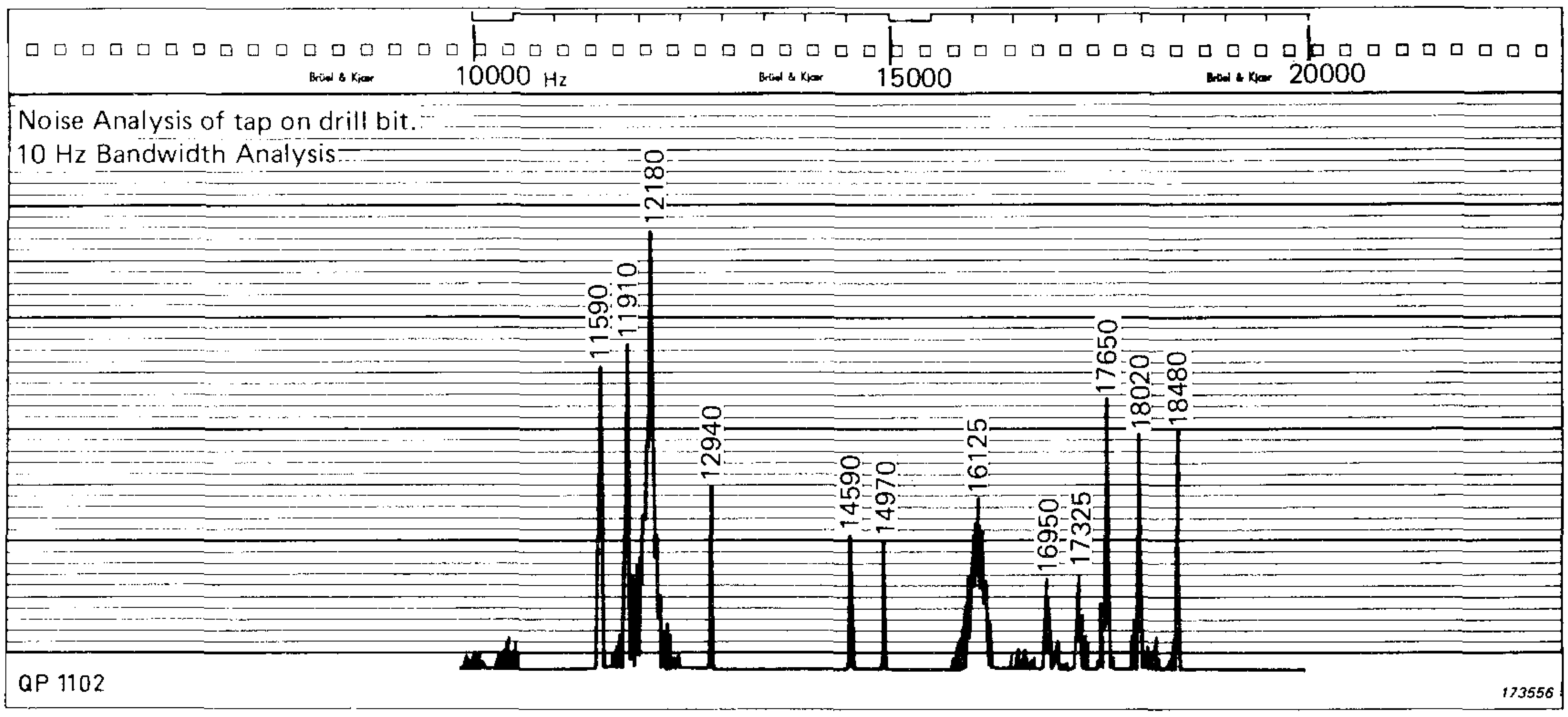

Fig.6. Noise Spectrum of tap on drill bit

A similar **vibration** impact spectrum was also obtained by analyzing the signal from a subminiature-accelerometer Type 8307 placed on one of the flanges of the drill bit (shown in Fig.5) and tapping the drill bit. The signal displayed on the oscilloscope is shown in Fig.7 while the spectrum obtained is shown in Fig.8 where again the natural frequencies of the drill bit are exhibited and seen to be very similar to those of Fig.6.

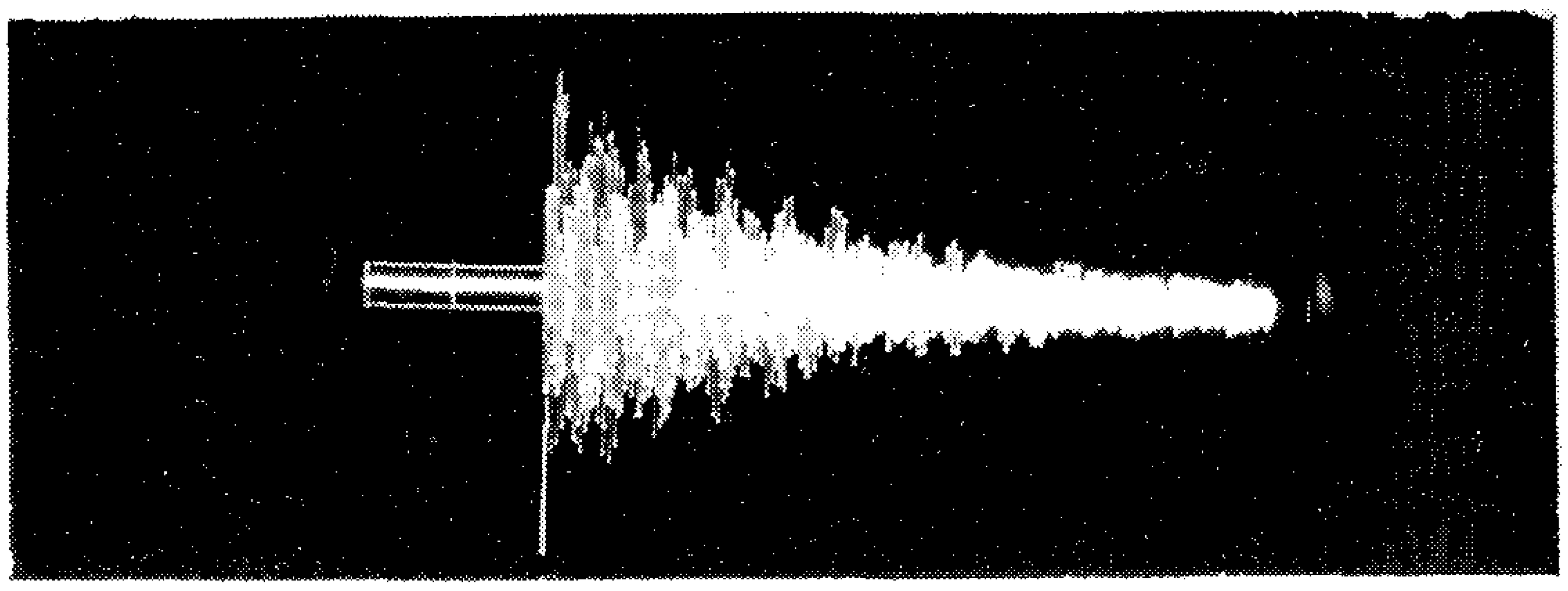

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Fig.7. Signal captured by 7502 from mini-accelerometer

The chisel bit was now used in the impact mode in concrete. The sound signal was recorded at a distance of 1 meter from the drill and analyzed again with a 10 Hz bandwidth filter, the spectrum of which is shown in Fig.9. When other signal samples were analyzed for the same conditions, it revealed that the 13190 Hz and 14670 Hz components were significantly higher (approximately 15 dB) than any of the other components. Expecting again these two frequencies to be generated by the chisel bit, the same experiment with a tap on the chisel bit

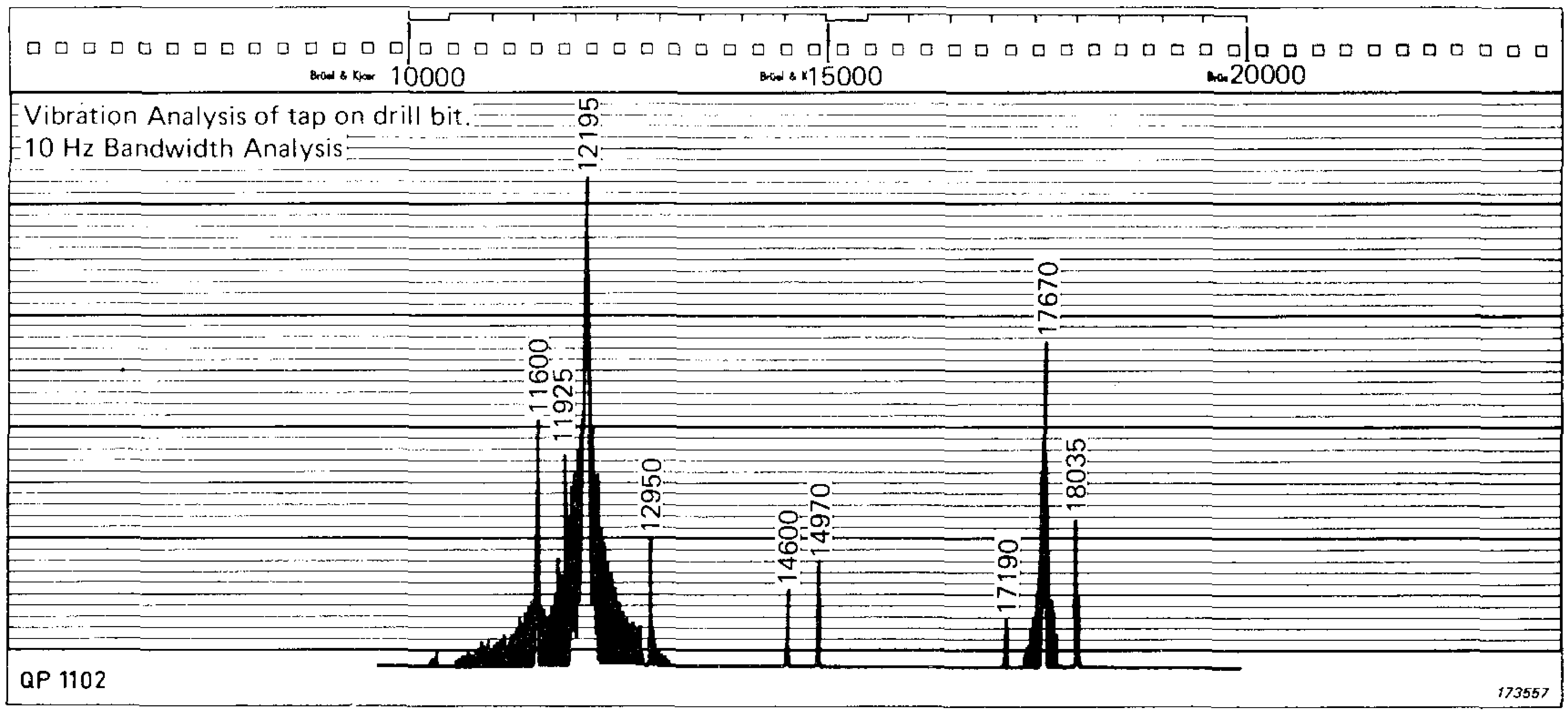

Fig.8. Vibration spectrum of tap on drill bit

(as was carried out on the drill bit) was performed. However, the natural frequencies generated by the chisel impact sound were different to those from the spectrum in Fig.9. Mounting the accelerometer on the chisel bit to obtain a vibration spectrum was also not practicable on account of the geometry of the chisel bit. Indirect methods for determining the natural frequencies of the chisel bit therefore had to be sought for. For the preliminary investigation it was argued that if the two frequencies 13190 Hz and 14670 Hz were generated by the chisel bit,

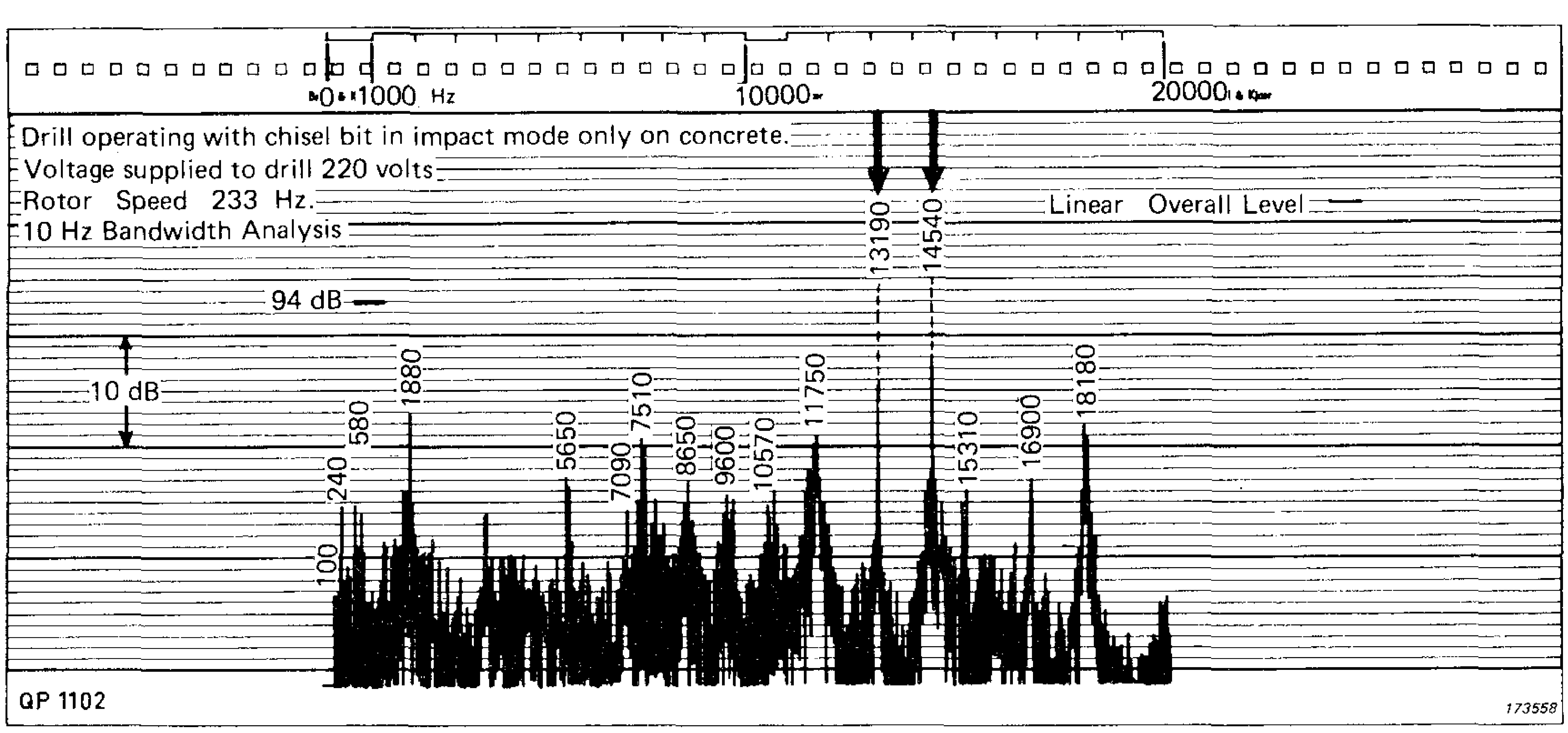

Fig.9. Noise Spectrum of Drill operating with chisel bit in impact mode only, in concrete

they could be damped out if the drill was made to operate with the chisel bit in the impact mode on a block of wood embedded in sand as shown in Fig.10. The sound signal was recorded and analyzed for the drill operating at 220 V and rotational speed of the rotor being 233 Hz. From the spectrum obtained, shown in Fig.11, it is seen that the 13190 Hz and 14670 Hz frequency components are considerably damped out and are quite comparable with the other peaks in the spectrum. The reason why the peaks are not exactly at the same frequencies in Figs.9, & 11, is probably due to the fact that the "free length" of the chisel bit in the two operating conditions has not been the same and neither the end fixing conditions. The overall sound level has also been reduced from 103 dB to 96 dB.

It is interesting to note that the peaks in the spectrum again occur at the tooth meshing frequency 233 x 9 = 2097 Hz, first lower sideband

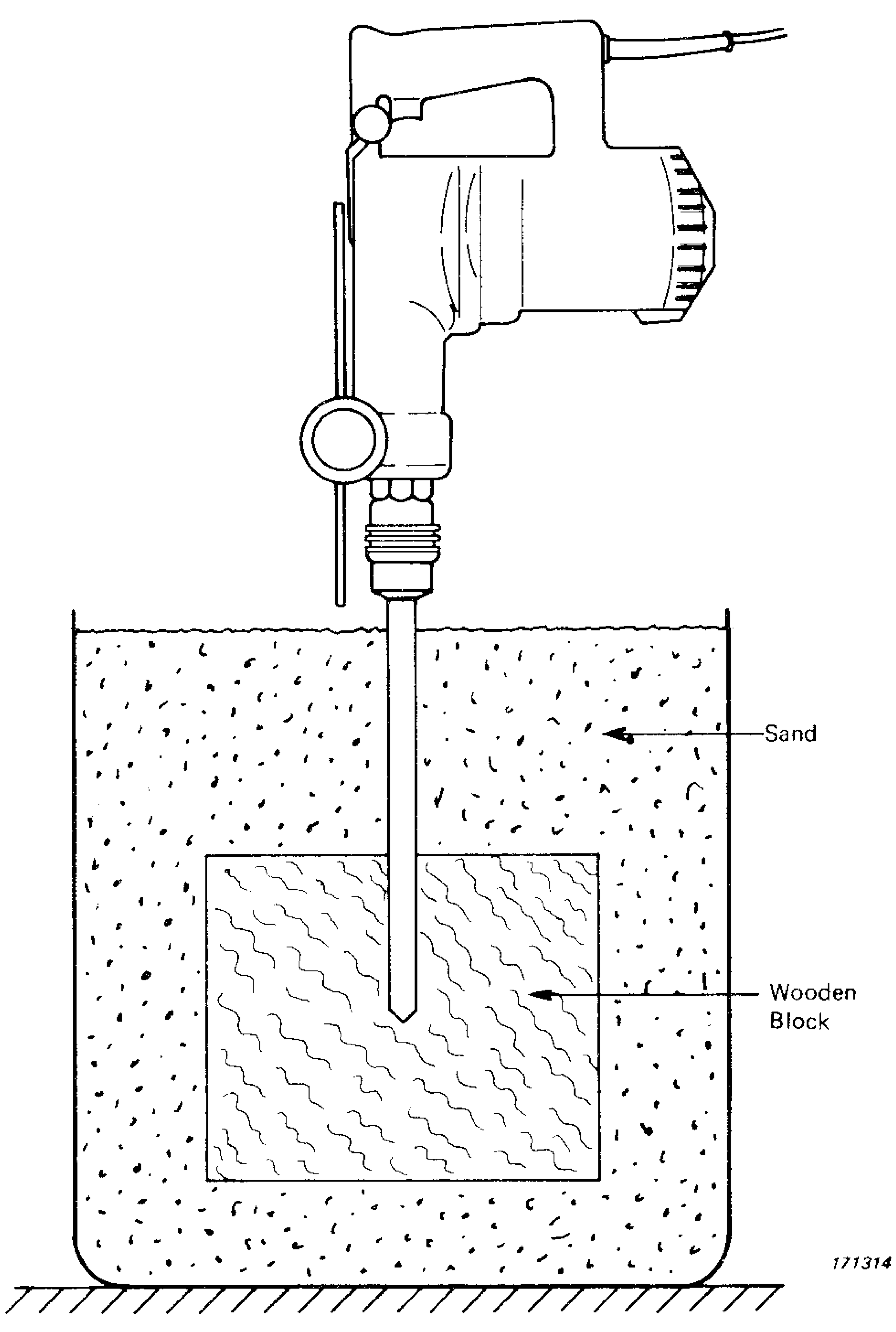

Fig.10. Drill operating with chisel bit on a block of wood immersed in sand

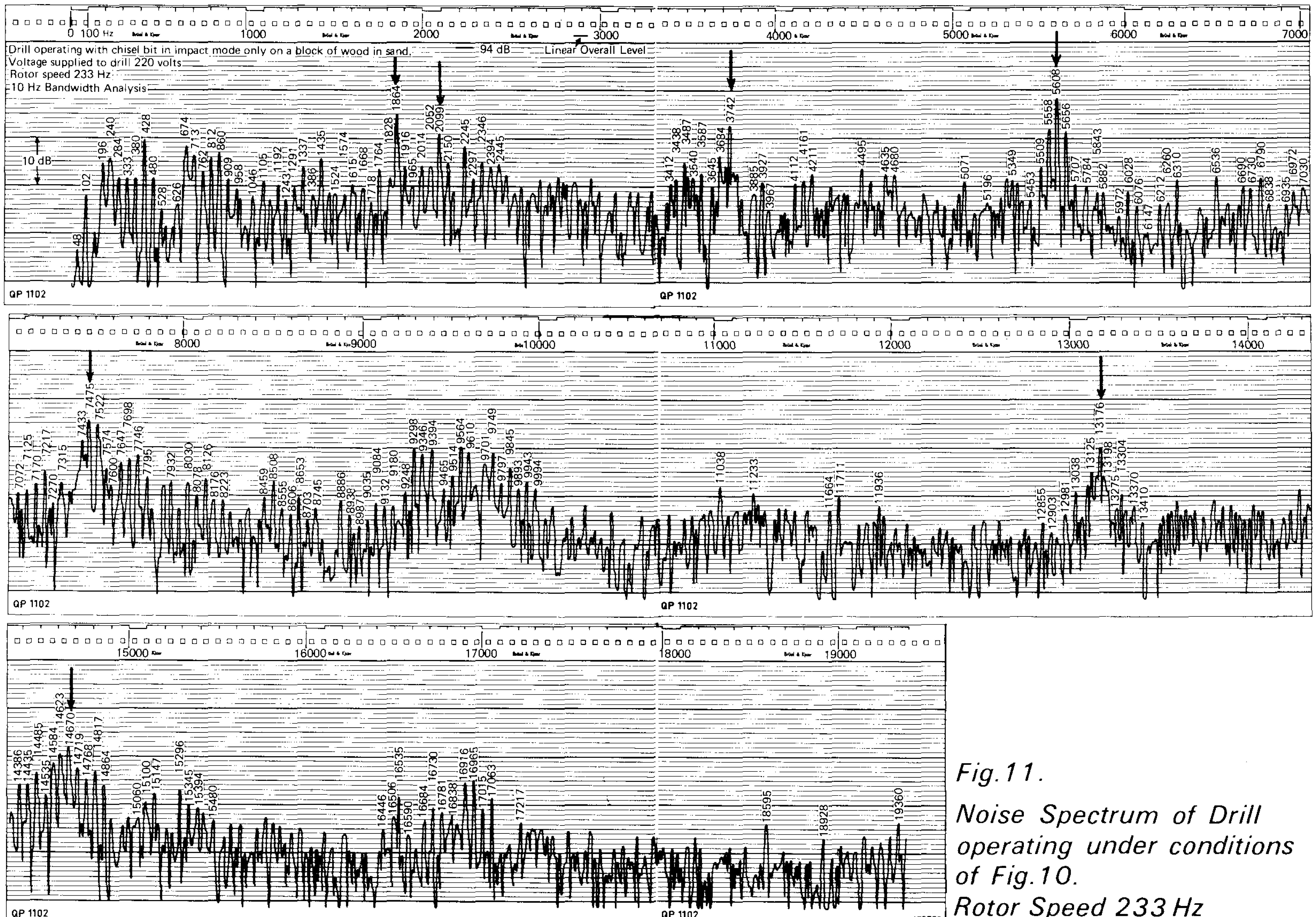

Fig.11.

Noise Spectrum of Drill operating under conditions of Fig.10.
Rotor Speed 233 Hz

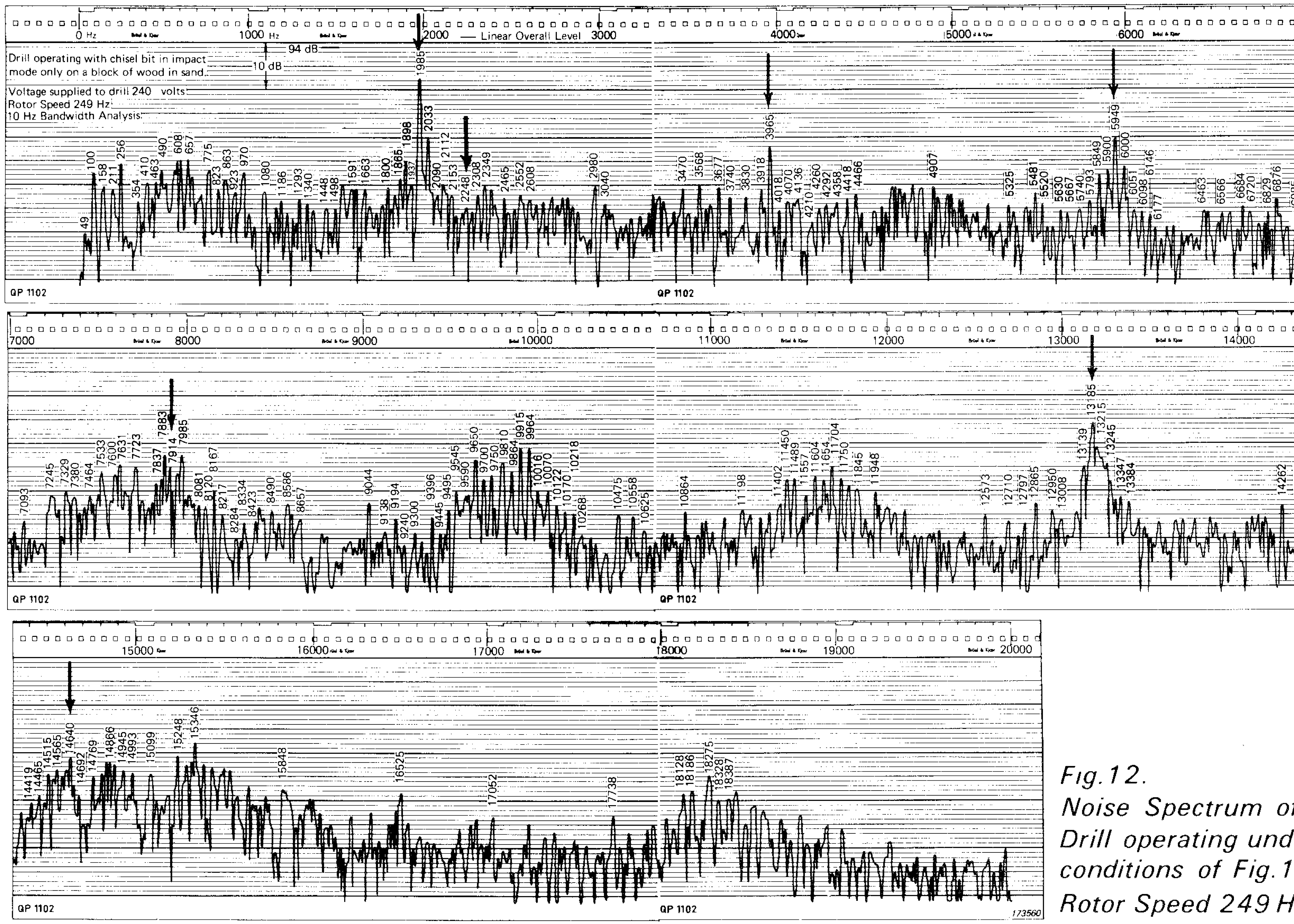

Fig.12.
Noise Spectrum of Drill operating under conditions of Fig.10.
Rotor Speed 249 Hz

2097 — 233 = 1864 Hz, second lower sideband for the second harmonic of the meshing frequency 2 x 2097 — 2 x 233 = 3728 Hz and so on just as in the case of the free running condition. Also from the gear ratios it can be seen that the number of piston impact strokes per second is given by the frequency 9/44 x 233 = 47,6 Hz where 9 and 44 are the number of teeth on the pinion and wheel respectively. This frequency is again seen in the spectrum as the spacing between each individual peak.

In order to distinguish between frequency amplitudes of structural resonances and those due to rotational speeds of gears, another spectrum was obtained for the same loading conditions except that the drill was operated at a higher voltage 240 V with the drill rotor having a rotational speed of 249 Hz. In this case the frequencies excited by structural resonances would not be altered, while those due to gears would be. From the spectrum obtained, shown in Fig.12 it is seen that all the peaks at the frequencies related to rotation of gears, i. e. the lower sidebands of meshing frequency and its harmonics, are shifted upwards in frequency proportionately. However, the peaks at the frequencies 13190 Hz and 14670 Hz are not affected significantly. Having established indirectly that these two frequencies were generated by the chisel bit, a final attempt was made to determine the natural frequency of the chisel bit directly.

The chisel bit was embedded in a wooden block and the drill was placed inverted on the floor as shown in Fig.13. The chisel bit was lifted manually and struck on the piston inside the drill. The resulting impact sound was recorded by the Digital Event Recorder and analyzed by a 10 Hz bandwidth filter. The spectrum obtained is shown in Fig.14, where the resonance peak around 13190 Hz is revealed.

The reason why different natural frequencies were excited by a "hammer" blow on the chisel bit was that the holding conditions of the chisel bit were not the same as when it was under operation with the drill. The free length of the chisel bit as well as the end fixing conditions tend to play a significant role (as should be expected) in determining the natural frequencies of the chisel bit.

From the above discussion it is seen that the noise from the drill is generated by structural resonances of the drill bit and chisel bit as well as from the rotation of gears. The former components are significantly higher in frequency and amplitude and are often known as the "ring" of the steel tool.

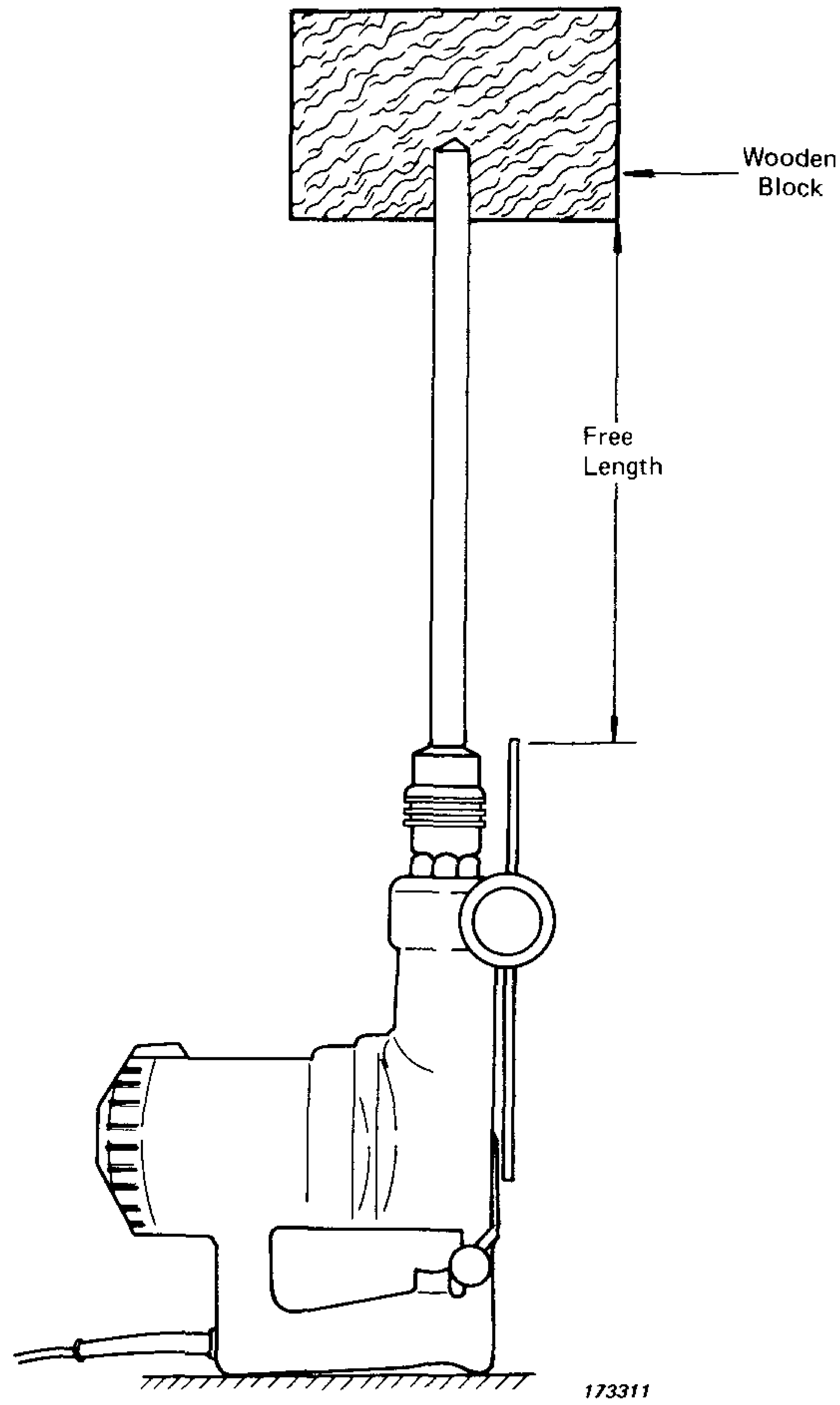

Fig. 13. Method of Excitation of natural frequencies of chisel bit

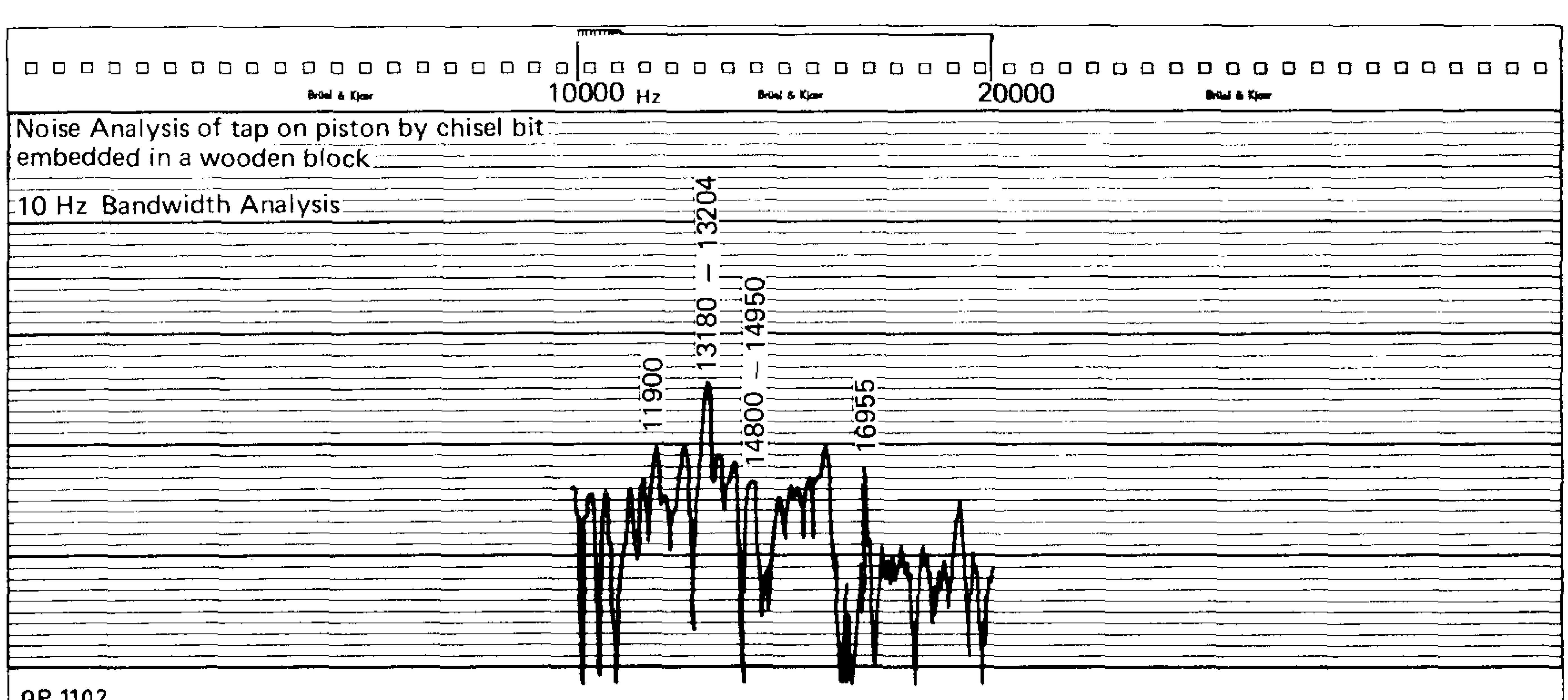

Fig. 14. Noise Spectrum of chisel bit excitation

In Appendix A the frequency spectrum of Fig.12 is further analyzed using the technique of *Cepstrum* Analysis to establish sidebands that may be buried in the noise.

For the sake of completeness the overall sound pressures emitted by the drill in 1/1 octaves were measured in an anechoic chamber and the sound power evaluated, as shown in Appendix B.

Conclusions

The investigation of the drill has shown that the major component of noise is generated by the tool bit and is found to be around 15dB higher than other components in the spectrum. Remedying this noise is often difficult though some tools in industry are found with damping treatments applied to them.

The next important source of noise seems to be generated by the gears. Figs.11 and 12 illustrate the sidebands generated by the pinion around the tooth meshing frequency. Also the wheel rotational frequencies of 9/44 x 233 = 47,6 Hz and 9/44 x 249 = 51 Hz in Figs.11 and 12 respectively seem to modulate major portions of the frequency spectra. Since the larger components are, however, due to the pinion, correction of pinion eccentricity would seem to be the logical primary step for reducing gear noise.

The technique of narrow band frequency analysis, though very useful in detecting harmonics and sharp resonances, can be rather time consuming if analysis in very narrow bandwidths is required. However, it can be speeded up by frequency transformations utilizing the Digital Event Recorder Type 7502, Ref. [8]. If further information of the "periodicity" in the frequency spectrum is desired the technique of cepstrum analysis, as described briefly in Appendix A and Ref. [6, & 7], could then be attempted. Since the cepstral content is found to be sensittive to the sidebands generated around tooth meshing frequencies of gears, the technique sounds promising as a diagnostic tool.

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Appendix A

Cepstrum Analysis

The increase in amplitudes of sidebands and harmonics of rotational frequencies of gears is a generally accepted indication of the deterioration in the condition of gears. e. g. Ref.[5]. Just as the periodicity of a signal in the time domain manifests itself as discrete frequencies in the frequency domain (obtained by Fourier Transform), similarly the "periodicity" in the frequency spectrum (e. g. the components of the rotational speed of gears and its harmonics) can be identified by computing the spectrum of the frequency spectrum. It is well known that the (inverse) Fourier Transform of the mean square spectral density gives the autocorrelation function. If, however, the Fourier Transform of the *logarithm* of the mean square spectral density is taken we obtain what is termed as the *cepstrum* as a function of the independent variable *quefrency* having the dimensions of time.

One advantage of the cepstrum over the autocorrelation function lies in taking the logarithm of the spectral density before the transformation whereby the influence of the low amplitude components is increased. Also with the autocorrelation function, products in the frequency domain imply convolution in the time-lag domain, whereas with the cepstrum the additive relationship (resulting from taking the logarithm before the transformation) is maintained. For further description of cepstrum analysis, see Refs. [6 and 7].

Fig.12 which is reproduced here illustrates the frequency spectrum obtained for the drill rotor having a speed of 249 Hz. From the gear ratios one would expect the side bands due to the piston strokes or wheel rotational frequencies to appear at intervals of 51 Hz. It is seen that it is not always easy to distinguish them from noise peaks.

The Fourier Transform of this spectrum was now obtained using an FFT program and plotted out as the cepstrum shown in Fig.A1. The independent variable quefrency plotted as the abscissa has the units of ms. It should be noted that low quefrencies correspond to high frequency spacings since they are reciprocal values.

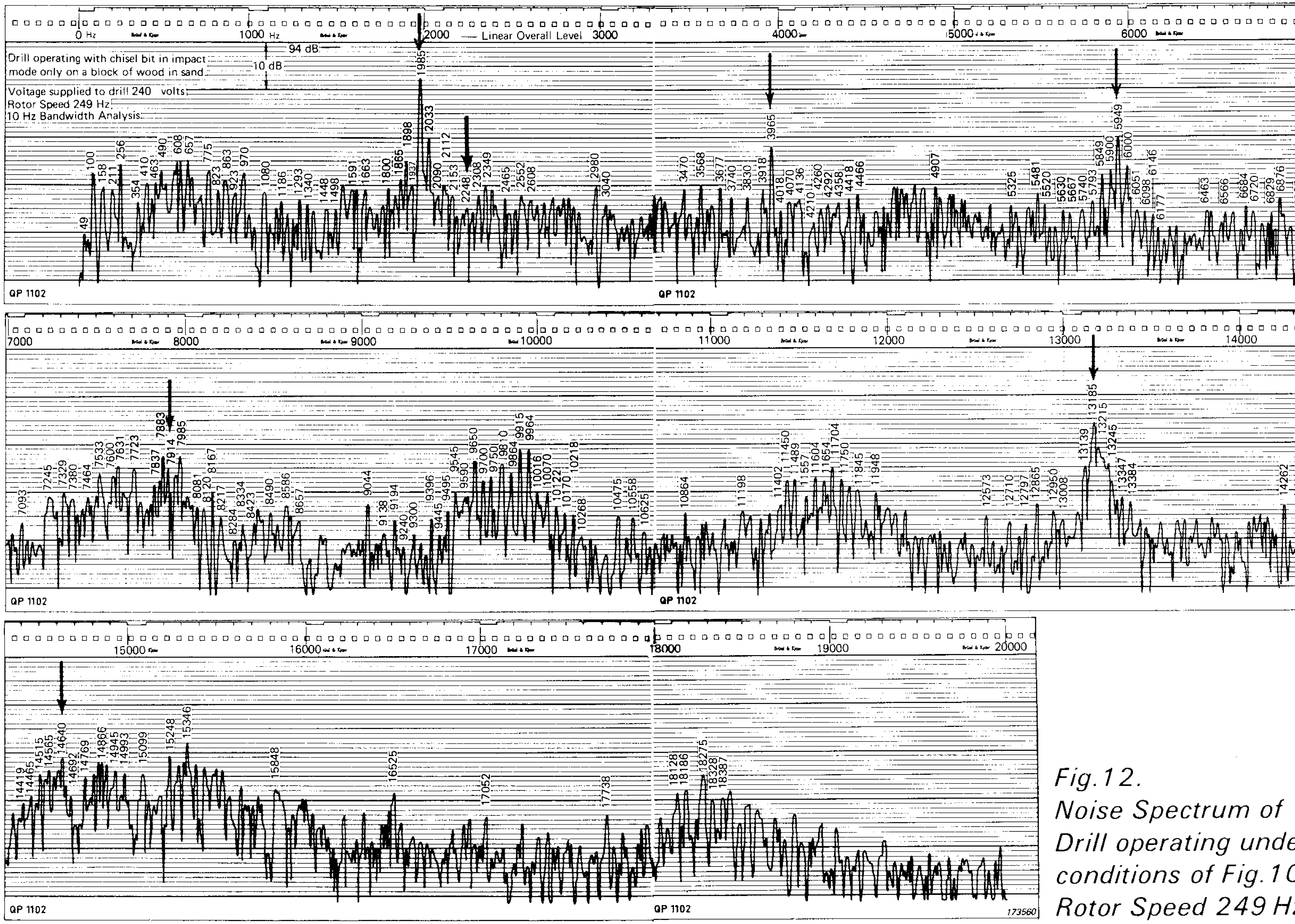

Fig.12.
Noise Spectrum of Drill operating under conditions of Fig.10.
Rotor Speed 249 Hz

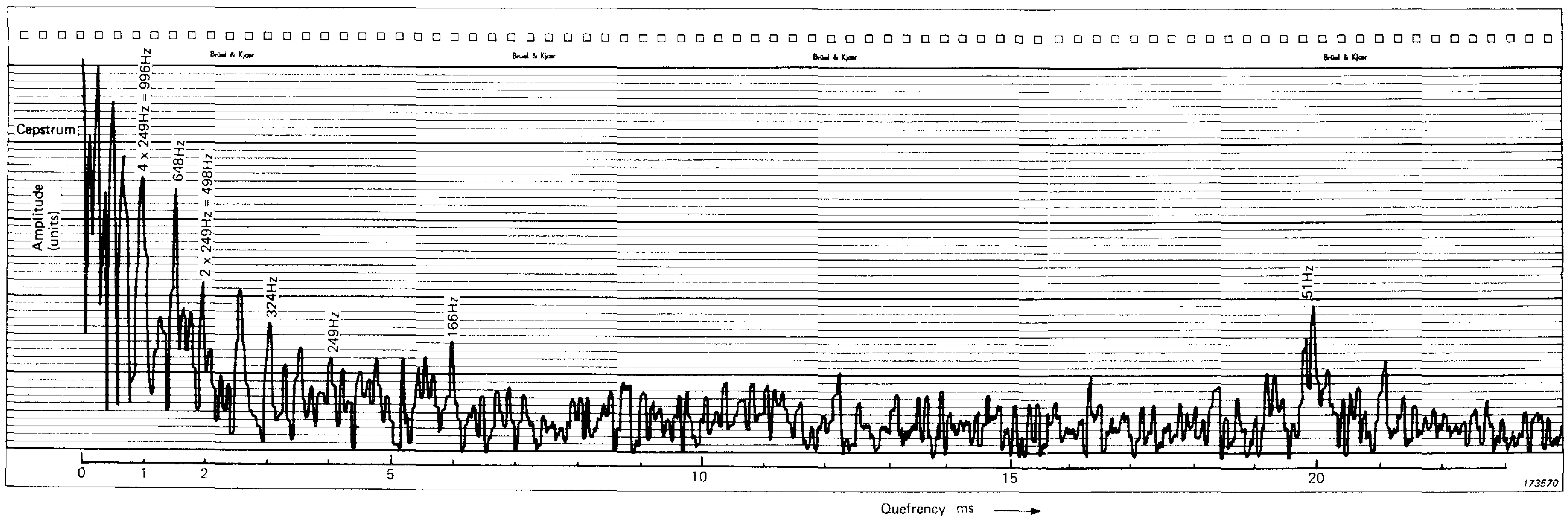

Fig.A 1. Cepstrum Analysis of the spectrum shown in Fig.12

From the cepstrum, quefrencies corresponding to 51 Hz, 249 Hz and its higher harmonics can be picked out relatively easily. However, there are other peaks at quefrencies corresponding to frequencies of 166 Hz, 324 Hz and 648 Hz. To investigate how these quefrencies are generated, the complete set of gears in the drill should be examined and is illustrated in Fig.A2. From the number of teeth on different gears, their rotational speeds as well as their tooth meshing frequencies are evaluated and shown in the figure. As can be seen the pair of bevel gears have a tooth meshing frequency of 161 Hz. The bevel pinion and bevel wheel have rotational speeds of 11,5 and 4,6 Hz respectively. Therefore the quefrencies corresponding to 166 Hz, 324 Hz and 648 Hz in the cepstrum seem to result from this tooth meshing frequency and possible sidebands caused by the rotational frequencies.

It is interesting to note that these frequency spacings though they must be present in the frequency spectrum cannot be detected visually.

At present, work is being carried out to establish whether the change in amplitudes in the cepstrum can be related to the measure of the deterioration of gearboxes. Since the increase in the amplitude of the sidebands can be seen more easily in the cepstrum than in the spectrum a relatively powerful method could lie in the hand of the preventive maintenance engineer.

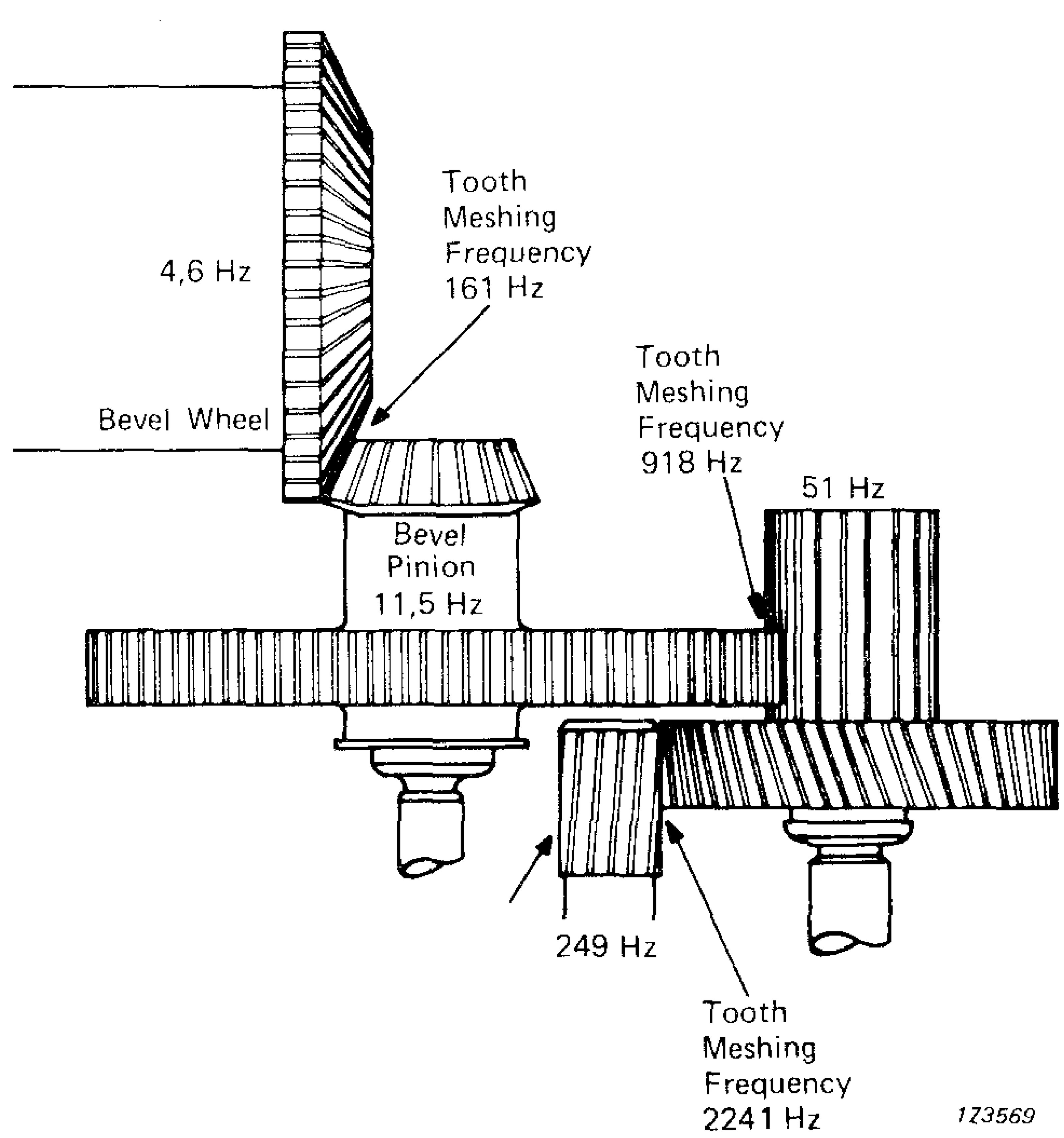

Fig.A 2. Set of gears of the drill

Appendix B

Overall sound pressure level and sound power of the Drill
For the sake of completeness the overall sound pressure level and sound power level of the drill were determined utilizing the guidelines given in ISO Draft Proposal 169.

Since the sound generated by the drill was of greater interest than that generated by the chisel or the drill bit, the sound pressures were measured while the drill was operating with the chisel bit in the impact mode on a block of wood immersed in sand, to damp the frequencies of the chisel bit, as shown in Fig.10. The drill was operated at 230 V with the rotor speed of 241 Hz, in an anechoic chamber. As a reflecting surface, chip-wood board of dimensions 3 m x 3 m was used on which the drill was operated. The recordings of the sound signal were carried out again using the apparatus shown in Fig.2 except that the analysis was carried out in 1/1 octave bands utilizing the Filter Set Type 1615. The details of the microphone positioning is shown in Fig.B1 while Fig.B2 illustrates the sound pressure levels measured in 1/1 octave bands at the five microphone positions. The values of the sound pressure levels are tabulated in Table 1 as well as the mean sound pressure levels in each octave band.

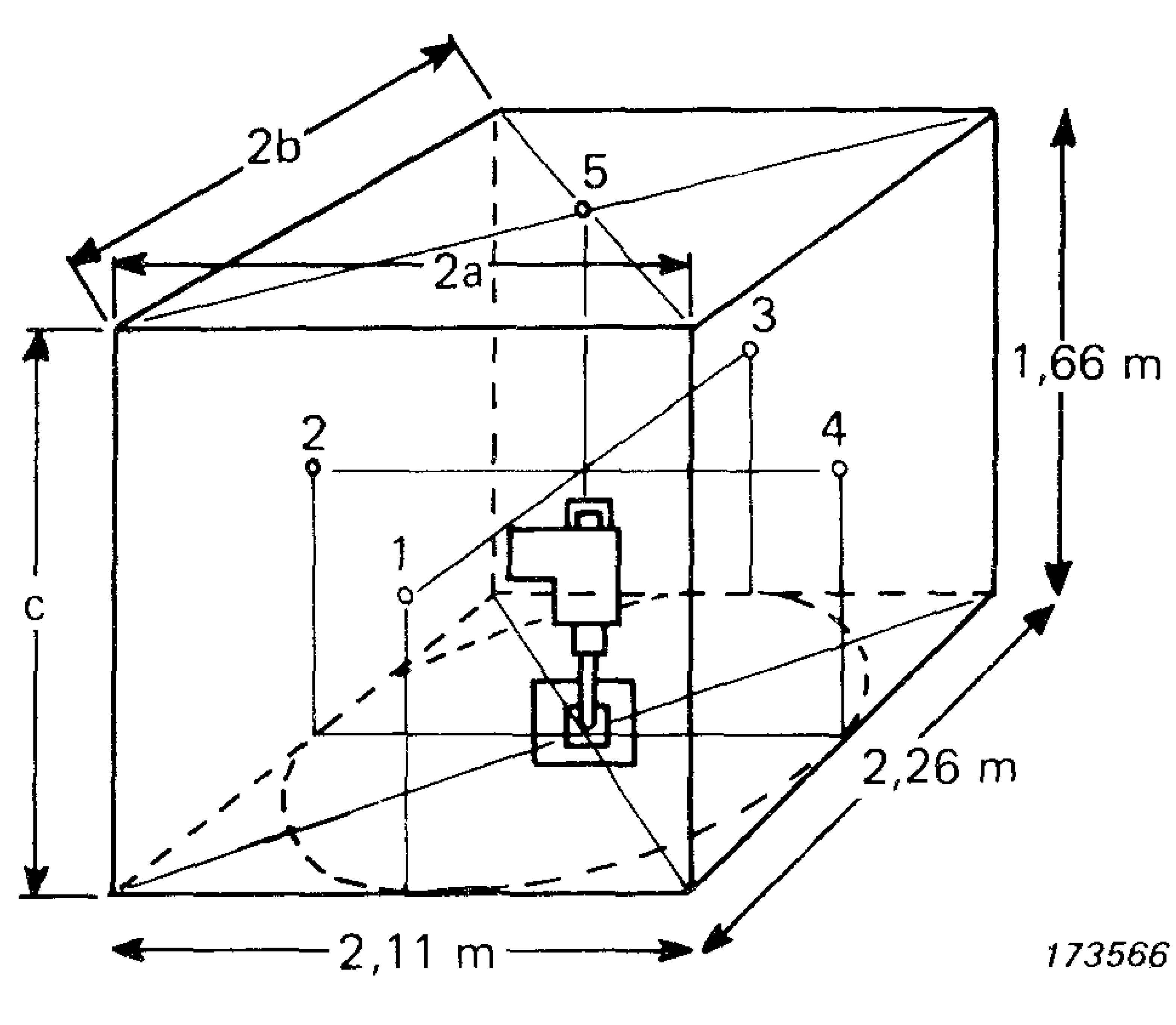

Fig.B 1. Microphone positions and measurement surface around the drill

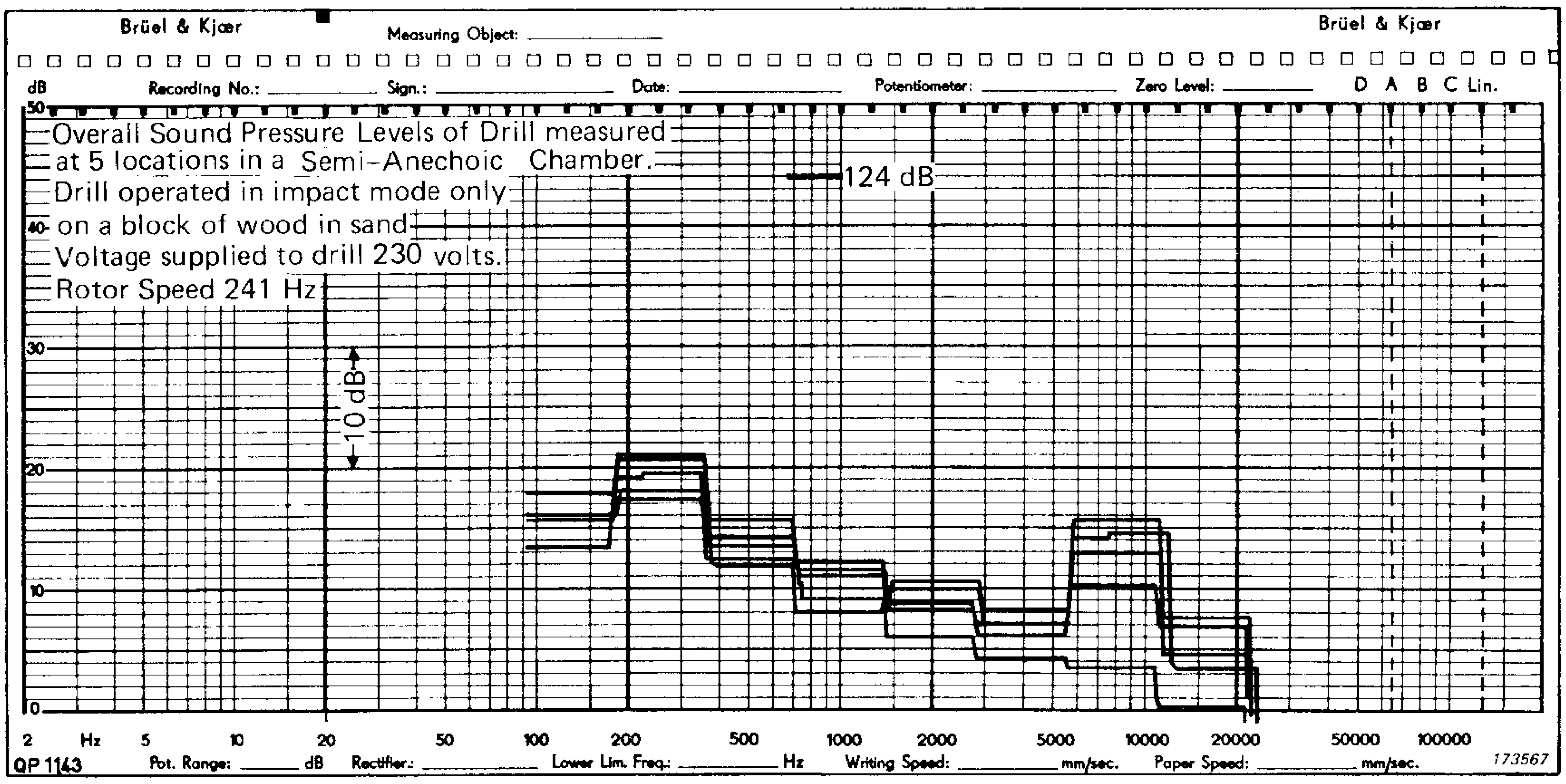

Fig.B 2. Sound pressure levels measured in octave bands at five positions

MICRO-PHONE POSN	BAND CENTRE FREQUENCY (Hz) 125	250	500	1000	2000	4000	8000	16000
1	96	98	92	89	90.5	88	93	87.5
2	96	101	93.5	91.5	90	87	95	86.5
3	93.5	99.5	92.5	88	88.5	87	95	83.5
4	96	101	94.5	91	88	86	90.5	84.5
5	98	97.5	95.5	92.5	86	84	83	80.0
MEAN(dB)	95.9	99.4	93.6	90.4	88.6	86.4	92.8	85.2
SOUND POWER (dB) LEVEL	108.7	112.2	106.4	103.2	101.4	99.2	105.6	98.0

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Table 1.

The surface area S enclosing the drill in Fig.B1 is given by

$$S = 4(ab + ac + bc) = 19{,}26\ \mathrm{m}^2$$

where c is the height, in m, of the measurement surface (normally equal to the height of the machine above the ground plus 1 m).

2 a is the width, in m, of the measurement surface (normally the width of the machine plus 2 m).

2 b is the depth, in m, of the measurement surface (normally the depth of the machine plus 2 m).

The surface area is used for determining the sound power level of the drill from the formula

$$10 \log_{10} \frac{P}{P_o} = 20 \log_{10} \frac{p_m}{p_o} + 10 \log_{10} \frac{S}{S_o}$$

where P_o = reference sound power 10^{-12} W

p_o = reference sound pressure $20\,\mu$Pa

and S_o = reference surface area 1 m^2

The sound power levels evaluated in each octave band are plotted in Fig.B3.

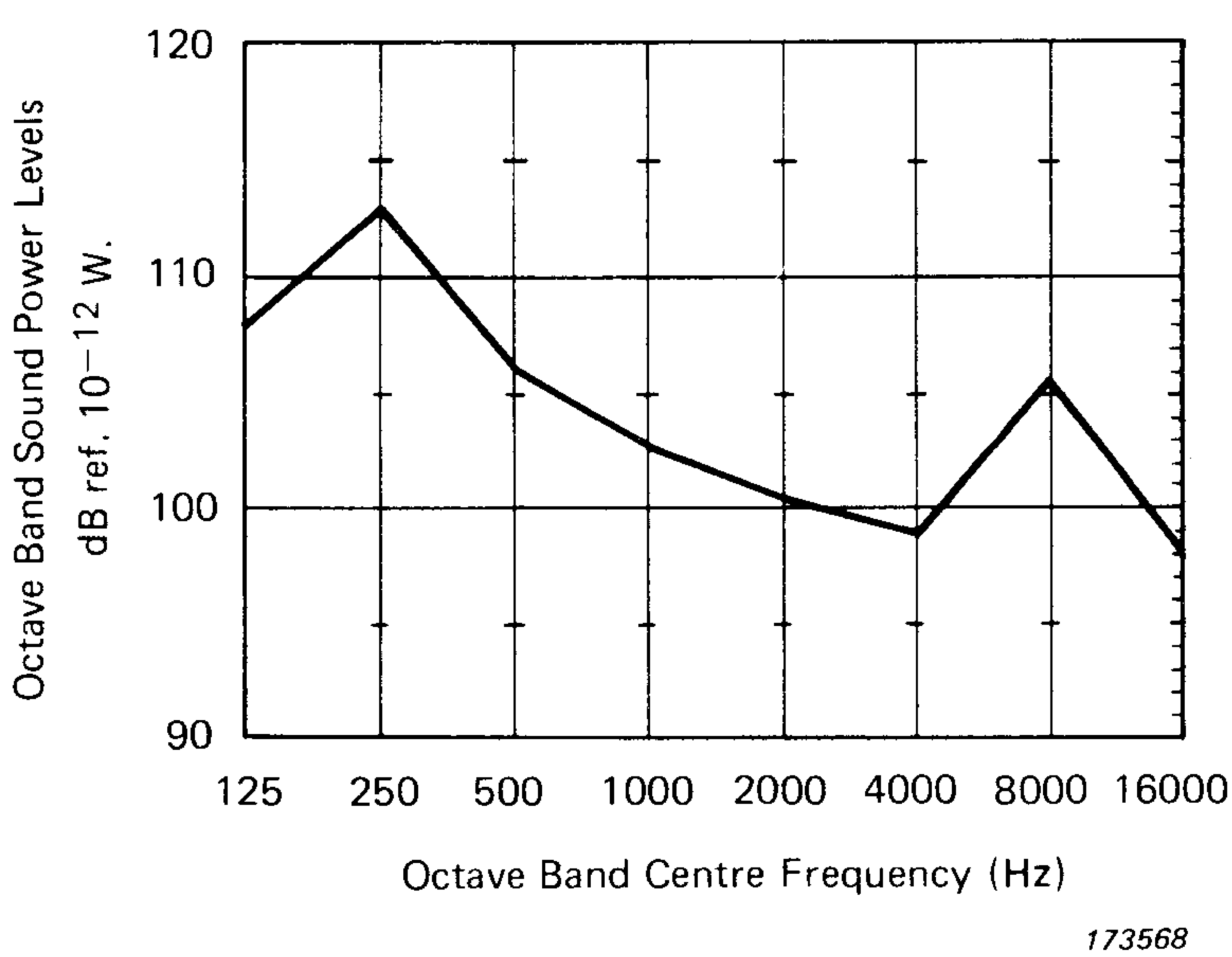

Fig.B 3. Sound power levels in eight octave bands

Measurement of the Dynamic Mass of the Hand-arm System

by

K. Zaveri, M. Phil.

ABSTRACT

To determine the dynamic mass of the hand-arm system, a measuring system has been described where the mass of the handle (required to transfer the excitation to the hand-arm system) is electrically compensated. By exciting the hand-arm at constant acceleration and recording the force required to do so, the dynamic mass of the hand-arm system is obtained as the excitation frequency is scanned over the frequency range of interest. The results obtained are found to be in reasonable agreement with those of other published data.

SOMMAIRE

L'article décrit un système de mesure permettant de déterminer la masse dynamique du système main/bras. Dans ce système, la masse de la poignée (nécessaire à la transmission de l'excitation au système main/bras) est compensée électriquement. En excitant l'ensemble main/bras à accélération constante et en enregistrant la force nécessaire à cette opération, on obtient la masse dynamique du système main/bras en fonction de la fréquence. Les résultats obtenus correspondent assez bien à ceux publiés par d'autres auteurs.

ZUSAMMENFASSUNG

Es wird ein Meßverfahren für die Bestimmung der dynamischen Masse eines Hand-Arm-Systems beschrieben, wobei die Masse des Handgriffes (erforderlich für die Übertragung der Erregung auf das Hand-Arm-System) elektronisch kompensiert wird. Durch Erregen des Hand-Arm-Systems mit konstanter Beschleunigung bei gleitender Frequenz innerhalb des interessierenden Bereichs, erhält man die dynamische Hand-Arm-Masse, indem man die zur Erregung erforderliche Kraft über der Frequenz aufzeichnet. Die auf diese Weise erhaltenen Resultate sind in guter Übereinstimmung mit anderen veröffentlichten Werten.

Introduction

Sound and vibration levels produced by various hand-held power tools currently in use, are in practice not only annoying but could be directly hazardous to health. The hand-arm system subjected to excessive vibration levels is widely known to suffer from diseases such as Raynaud's phenomenon and white fingers not to mention the side effects of fatigue, pain and numbness, (1 — 4). To investigate the interaction between the hand-arm and various power tools, a proper knowledge of the vibration response characteristics of the hand and arm is required. When the interaction is known, the necessary criteria for acceptable vibration levels can be established.

Such investigations have been carried out by, for example, Abrams (5) and D. D. Reynolds & W. Soedel (6). Reynolds and Soedel excited the hand-arm system through a T bar handle at constant force and measured the resulting displacement with an impedance head mounted under the T bar. The resistive force of the hand-arm alone was measured by subtracting out the inertia force due to the handle with the aid of an analogue computer.

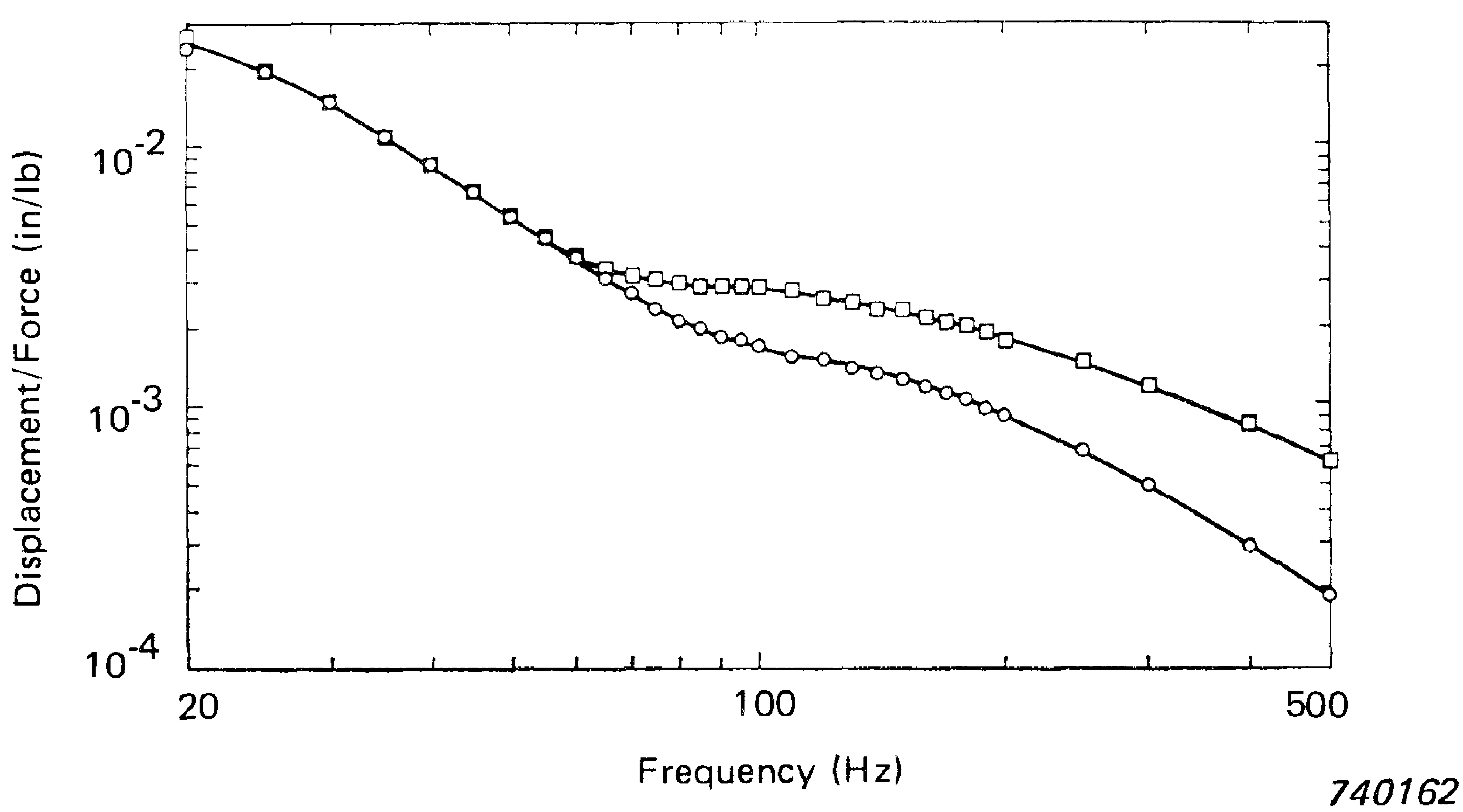

Fig. 1. Effects of handle mass upon impedance tests
□—□ Handle mass compensated for
o—o Handle mass not compensated for

The force required to excite the handle mass, as shown by their results reproduced here in Fig. 1, is significant above approximately 60 Hz and has therefore to be taken into consideration.

In this article, a measurement procedure for the determination of the dynamic mass of the hand-arm system is outlined where the influence of the mass of the handle is compensated for by a simple adding circuit. Practical details to be taken care of during experimentation, as well as the calibration procedure of the system used, is outlined.

Measurement Arrangement

Fig.2, shows a human hand gripping the handle used for the experiment. Also shown are the two directions of excitation relative to the hand-arm system.

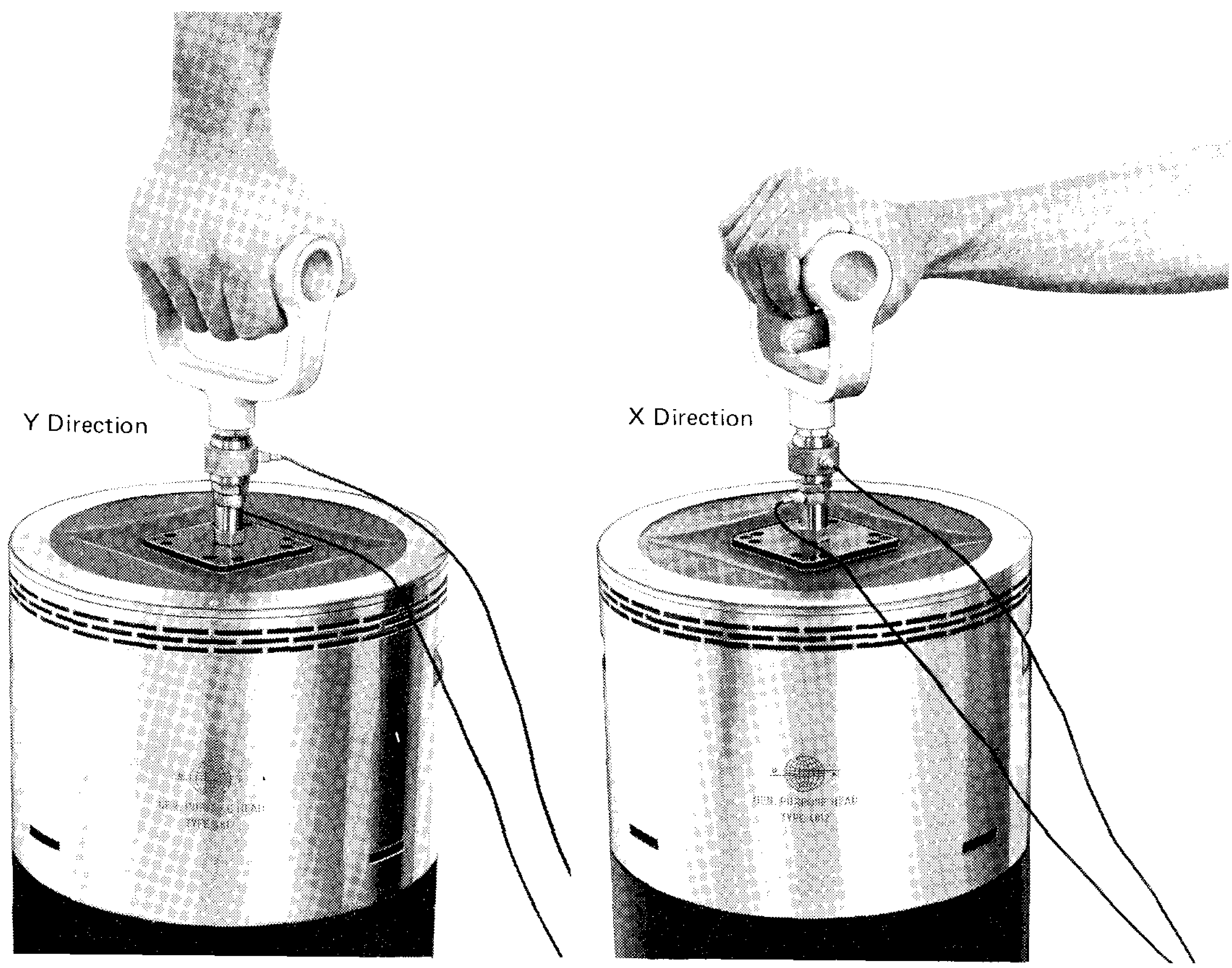

Fig.2. Directions of Motion of Excitation relative to the hand-arm system

The measurement set-up used is shown in Fig.3. A Force Transducer Type 8201 and a double ended Accerometer Type 8305 were mounted between the handle and the Vibration Exciter Type 4805/4812, such that the signals from the two transducers would be 180° out of phase with each other in the frequency range where the handle behaves as a pure mass. The accelerometer signal was led to the Vibration Exciter Control Type 1047 via a Conditioning Amplifier Type 2626 to keep the acceleration level on the shaker table constant. The acceleration signal

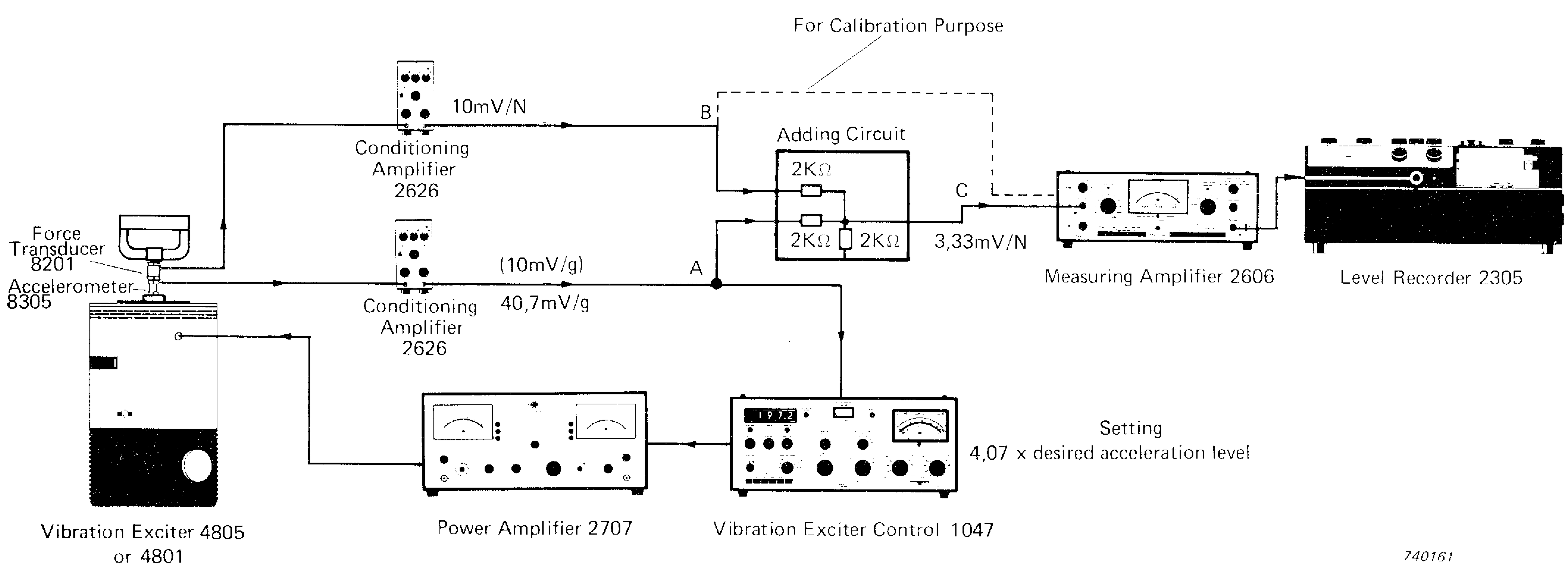

Fig. 3. Measuring Instrumentation Set-up

from point A as well as the force signal from point B are taken to a simple adding circuit. The output from the adding circuit, which represents the force required to excite the hand and arm only, is taken to a Level Recorder Type 2305 via a Measuring Amplifier Type 2606.

Calibration Procedure

The handle mass compensation procedure is carried out by exciting the handle alone at constant acceleration and adding the acceleration signal in antiphase, with appropriate gain, to the force signal at a suitable frequency. The gain is adjusted until the sum of the two signals from the adding circuit is reduced to a minimum, which will practically be the noise floor of the system. When an extra mass (e.g. hand-arm system) is placed on the handle, the voltage output from the adding circuit will represent the force required to excite the extra load. Therefore, by recording the voltage output from the adding circuit while gripping the handle and scanning the excitation frequency at constant acceleration, a continuous recording of the dynamic mass of the hand-arm system is obtained.

When calibration of the system is carried out, a couple of practical points should be kept in mind. First, the simple adding circuit used here reduces the input signals (and thereby their sum) to a 1/3 and secondly, the handle behaves as a simple mass only over a limited frequency range.

The following steps show how calibration of the system is carried out in practice.

1. When the accelerometer and force transducer sensitivities are set on the corresponding conditioning amplifiers, their output sensitivities are normally calibrated to be 10 mV/g and 10 mV/N respectively. For these output sensitivities, the vibration meter channel of the Exciter Control Type 1047 will directly read the acceleration levels in g and the level on the shaker table can be set, for example at 1 g to excite only the handle.

2. Since the total mass of the handle (0,372 kg) + mass on top of the quartz discs of the force transducer (0,043 kg) was (0,372 + 0,043) = 0,415 kg and the acceleration level was 1 g, the voltage output from the conditioning amplifier at point B would be

   10 mV/N x Force required to excite the handle mass,

   i. e. 10 x 0,415 x 9,81 = 40,7 mV (corresponding to 4,07 N).

This voltage can be taken to the Level Recorder from point B, see Fig.3 via the measuring amplifier and marked on its paper to calibrate a force of 4,07 N. The voltage output from the accelerometer conditioning amplifier at point A would be 10 mV since the acceleration level is set at 1 g.

The force and acceleration signals, from points B and A respectively, are now taken to the adding circuit, the output voltage of which at point C would be

$$V_{out} = \frac{1}{3}[\text{Force Signal} - \text{Acceleration signal}]$$

$$\text{i. e. } V_{out} = \frac{1}{3}[(10 \times 0{,}415 \times 9{,}81) - 10] \text{ mV}$$

Thus it can be seen that for mass cancellation of the handle the acceleration signal must be multiplied by a factor of (4,07) so that V_{out} would be zero.

3. The amplification of the acceleration signal can be achieved by reducing the accelerometer sensitivity settings on the accelerometer conditioning amplifier. If the sensitivity settings are reduced by the factor 1/4,07, the voltage output from the adding circuit at point C will be reduced to a minimum and mass cancellation of the handle will have been achieved.

   However, since the resulting output sensitivity of the conditioning amplifier is now increased to 10 x 4,07 mV/g the vibration level setting on the Vibration Exciter Control Type 1047 should now be increased by a factor of 4,07 to establish the desired acceleration level on the shaker table.

   Having compensated for the handle-mass the handle is now gripped and the hand-arm excited at the desired acceleration level. The voltage output at point C will now represent the force required to excite only the hand-arm system. Since the adding circuit at point C reduces the force signal at point B by a factor of 1/3, the force level of 4,07 N marked on the Level Recorder from point B will now represent a force of 3 x 4,07 = 12,2 N from point C.

Results

Curve 1 in Fig.4 shows the force response of the handle alone when excited at constant acceleration level of 1 g. When the hand-arm system is excited at 1 g and the mass of the handle is not compensated for, the

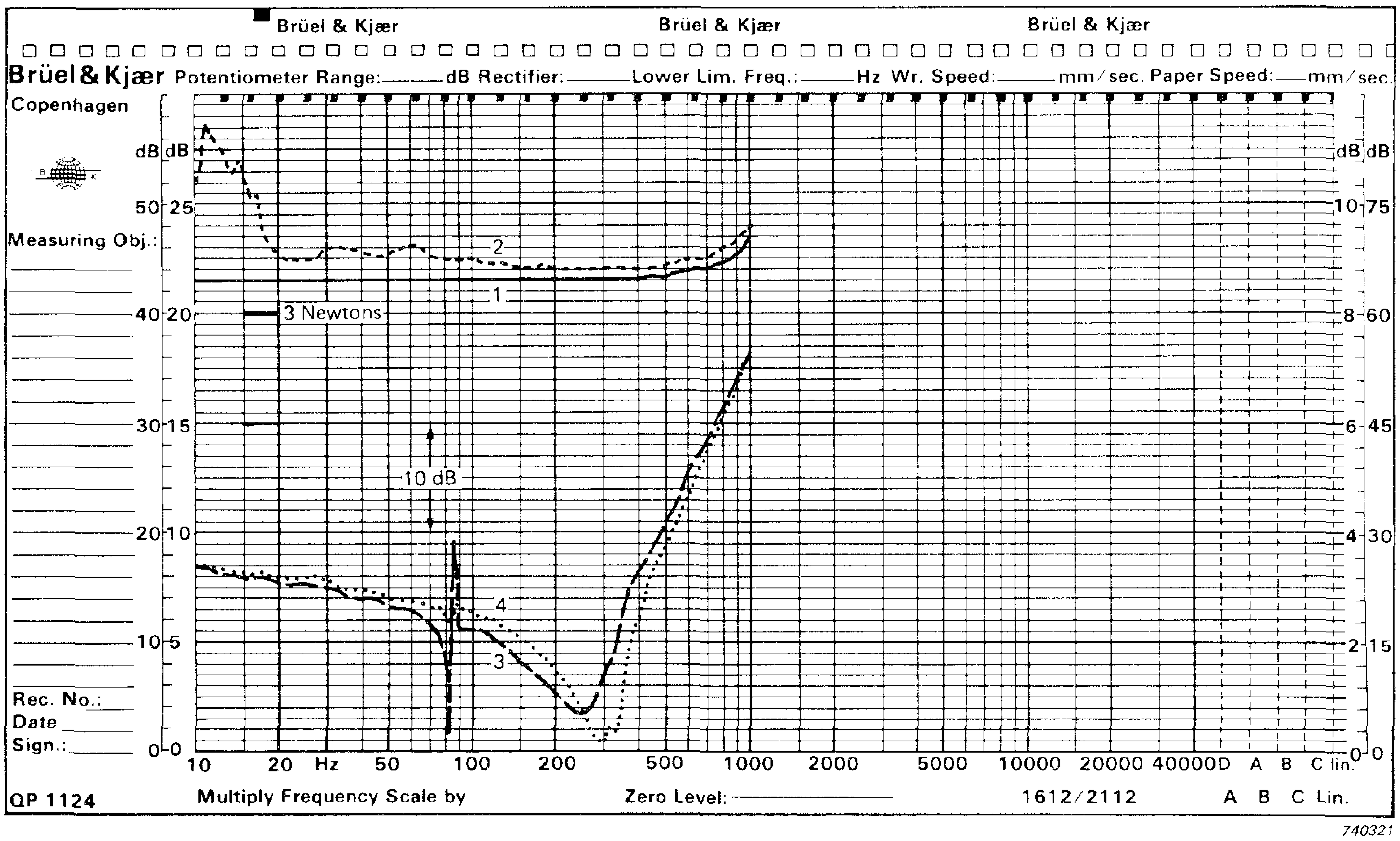

Fig.4. Force Response of
1) Handle when handle mass is uncompensated
2) Hand & Arm + handle when handle mass is uncompensated
3) & 4) Handle when handle mass is compensated

response obtained is shown by curve 2. From the two curves it can be seen that the dynamic mass of the handle dominates over the hand-arm system mass at high frequencies and thus the necessity for mass compensation arises.

When the handle alone is excited and the adjustment of mass compensation is carried out at 250 Hz, the voltage output from the adding circuit is shown by curve 3 in Fig.4. The curve actually gives a measure of how well the mass of the handle is compensated for and the level is seen to be more than 23 dB below the uncompensated curve over the frequency range 10— 500 Hz. The notch-peak occurring at around 83 Hz is due to the rocking motion of the handle caused by slight asymmetry in the distribution of the handle mass around the excitation axis. By ensuring symmetry of the handle, this effect can be minimized as shown by curve 4 in Fig.4. (In this measurement the mass compensation adjustment was carried out at 300 Hz as seen by the dip in the curve at that frequency).

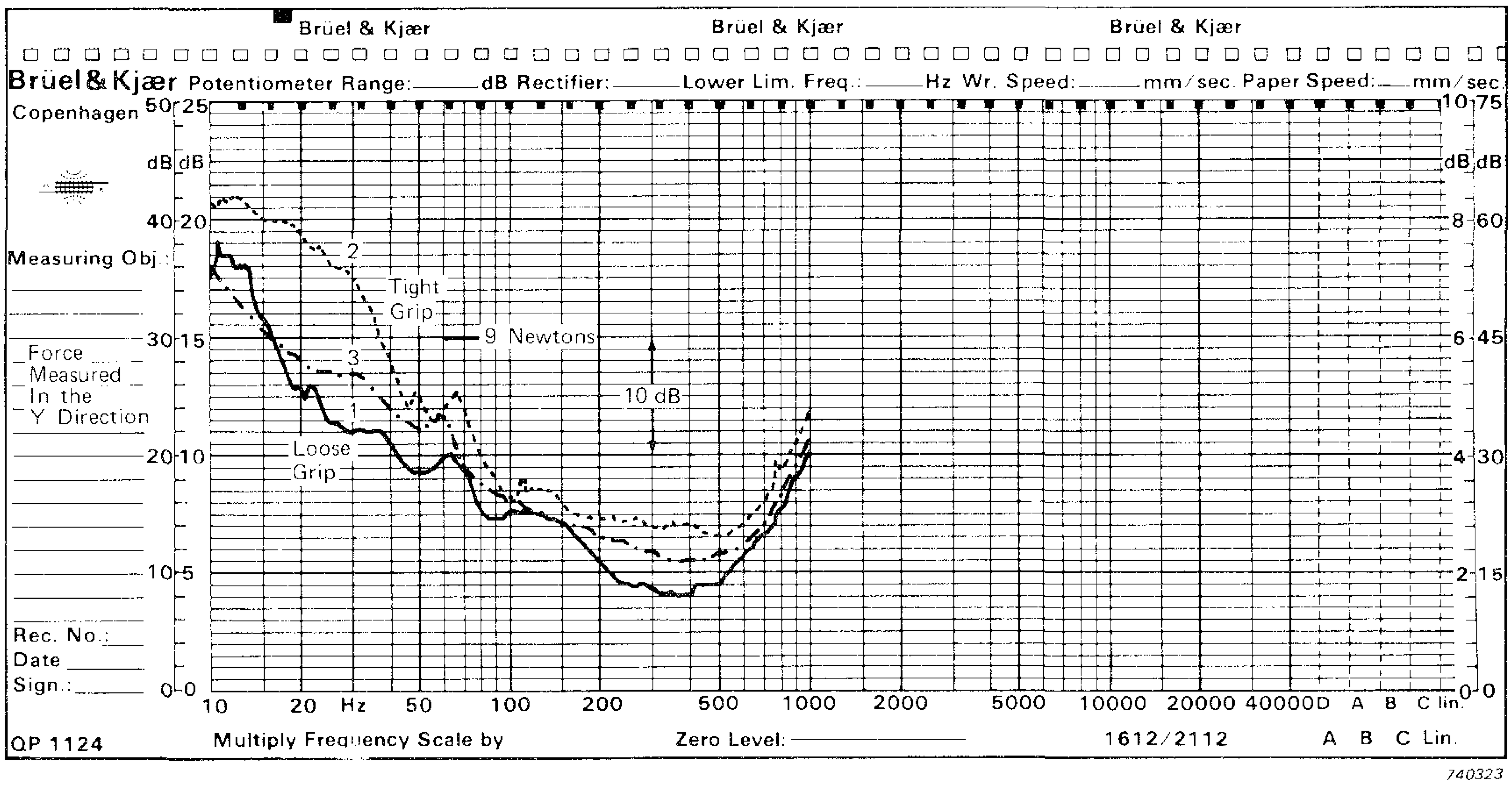

Fig.5. Force Response of Hand-arm system in the Y direction for excitation at 1 g

Curve 1 in Fig.5 shows the force required to excite the hand-arm system of an individual (weighing 55 kg) in the Y direction (see Fig.2) when gripping the handle loosely, while curve 2 is for the case when the handle is gripped tightly. The difference between the two curves is relatively small around 100 Hz and gets larger as the frequency is increased or

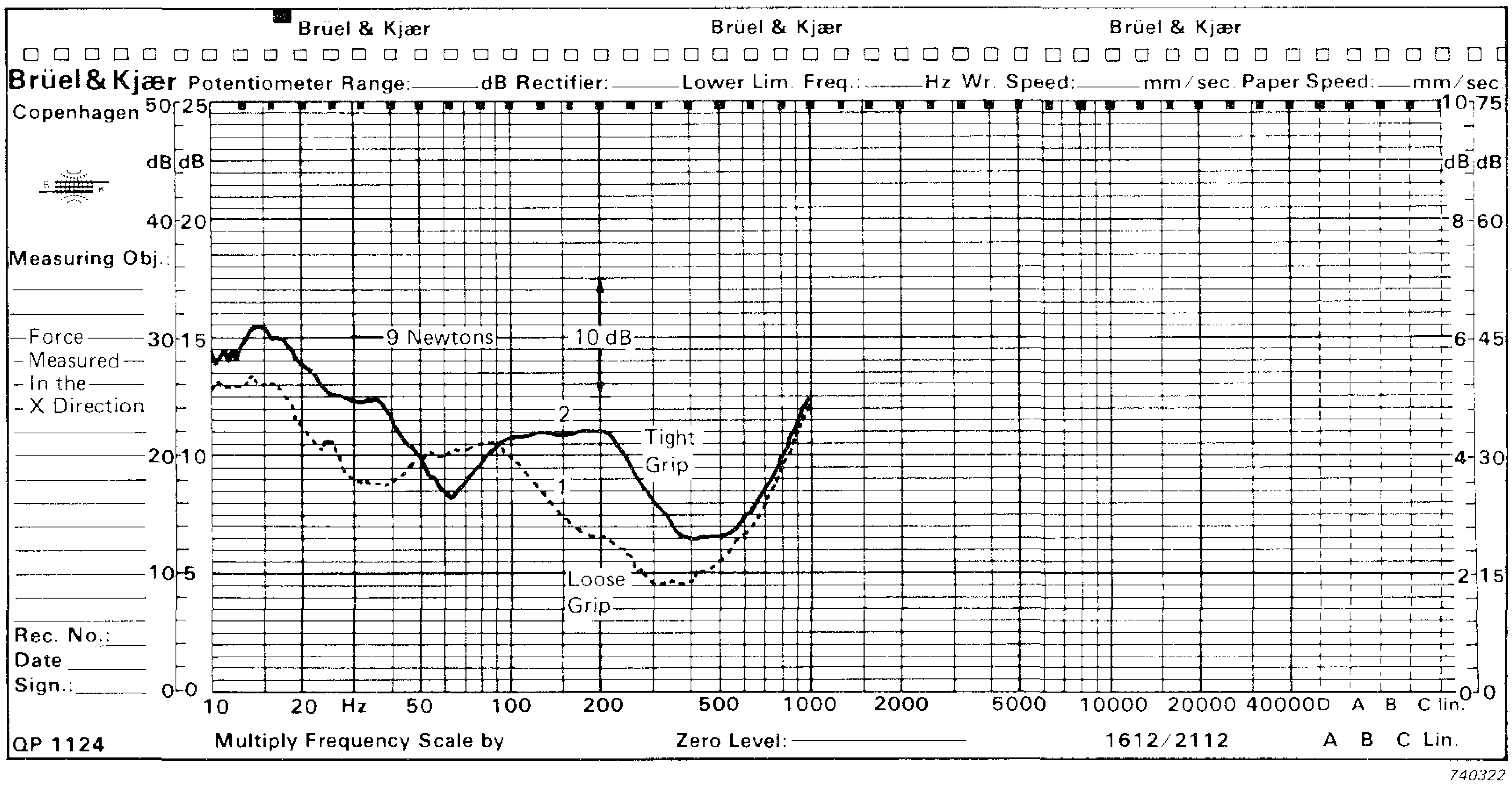

Fig.6. Force Response of Hand-arm system in the X direction for excitation at 1 g

decreased. This effect can be seen in curves of Ref. (6). Curve 3 is of an individual weighing 80 kg when gripping the handle loosely and illustrates what happens when a larger dynamic mass is involved.

Fig.6 shows the response of the hand-arm system of the same individual (weighing 55 kg) in the X direction for gripping the handle loosely, curve 1, and tightly, curve 2.

Discussion of Results

As seen by the notch-peak in curve 3 of Fig.4, it is important to make the handle as symmetrical as possible to minimize rocking disturbances. In addition, the Force Transducer Type 8201 was chosen on account of its comparatively large cross-sectional area and hence its low sensitivity to bending moments.

A T bar handle is relatively easy to construct from a symmetry point of view but unfortunately it is not very convenient to grip.

The sharp rise in all the curves above approximately 500 Hz is caused by the first antiresonance at 1600 Hz of the handle which depends on its mass and stiffness. (The antiresonance frequency being defined as the frequency at which a maximum of force is required for a minimum of motion). Therefore, by optimization in the design of the handle, the antiresonance can be moved to higher frequencies whereby the frequency range of measurements can be extended.

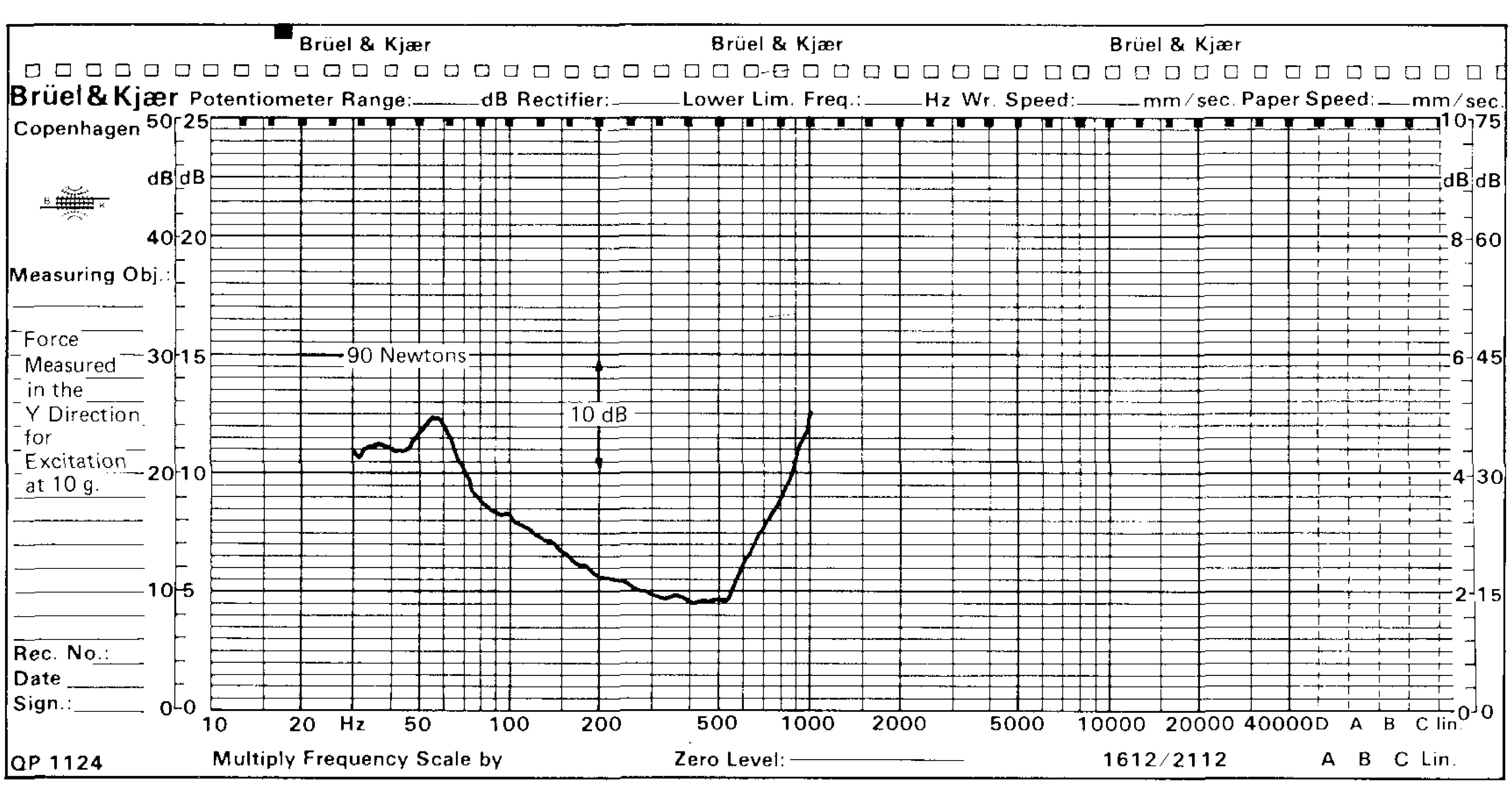

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Fig.7. Force Response of Hand-arm system in the Y direction for excitation at 10 g

Since the measurement of the dynamic mass of the hand-arm system is of interest, the choice of acceleration as the constant excitation parameter is quite obvious. Besides, excitation at constant force is also rather inconvenient as the servo-loop control would be lost, the instant the force due to the hand-arm system was removed from the handle. If, however, the mechanical impedance or dynamic stiffness of the hand-arm was of interest, excitation at constant velocity or displacement could also be carried out. In this case, however, attention should be paid to lower limiting frequencies and phase changes due to integrating circuits.

Fig.7 shows the force response of the hand and arm when excited at 10 g in the Y direction and gripping the handle loosely. At this high level it is rather inconvenient to grip the handle below 30 Hz on account of the large displacements. It is interesting to note that the results obtained above 30 Hz do not show any significant signs of non-linearity. Compare the curve to curve 1 in Fig.5.

Conclusions

A measurement system has been outlined for determining the dynamic response characteristics of the hand-arm system and is relatively easy to operate.

The dynamic response of the hand-arm varies not only as shown for individuals and for a loose and a tight grip, but several other parameters could have influence. They could be for example, light or firm pressure on the handle, bent or straight arm, arm relaxed or stiff, etc. As can be seen, a number of measurements would need to be carried out to obtain a proper knowledge of the hand-arm system before its typical characteristics could be stated and some further work is in progress.

In view of the parameters stated above and the complexity of the hand-arm system the results obtained in the X and Y directions are found to be in reasonable agreement with those from other published data. In Ref.(6) the results obtained by Reynolds and Soedel are compared to those of Abrams, Ref. (5). From the comparison curves it can be seen that the values of displacement/force of the hand-arm obtained by Abrams are slightly lower than Reynolds and Soedels. The corresponding values of the dynamic mass obtained in this investigation are similarly slightly higher than Reynolds and Soedels. Similar measurements can also be carried out for the Z direction which may be necessary, since the hand-arm when gripping a power tool vibrates in all three mutually perpendicular directions.

In the measurement set-up used, the minimum number of instruments have been employed. By using for example, a more sophisticated adding circuit or an additional accelerometer with conditioning amplifier, the calibration procedure may be simplified considerably.

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News from the factory

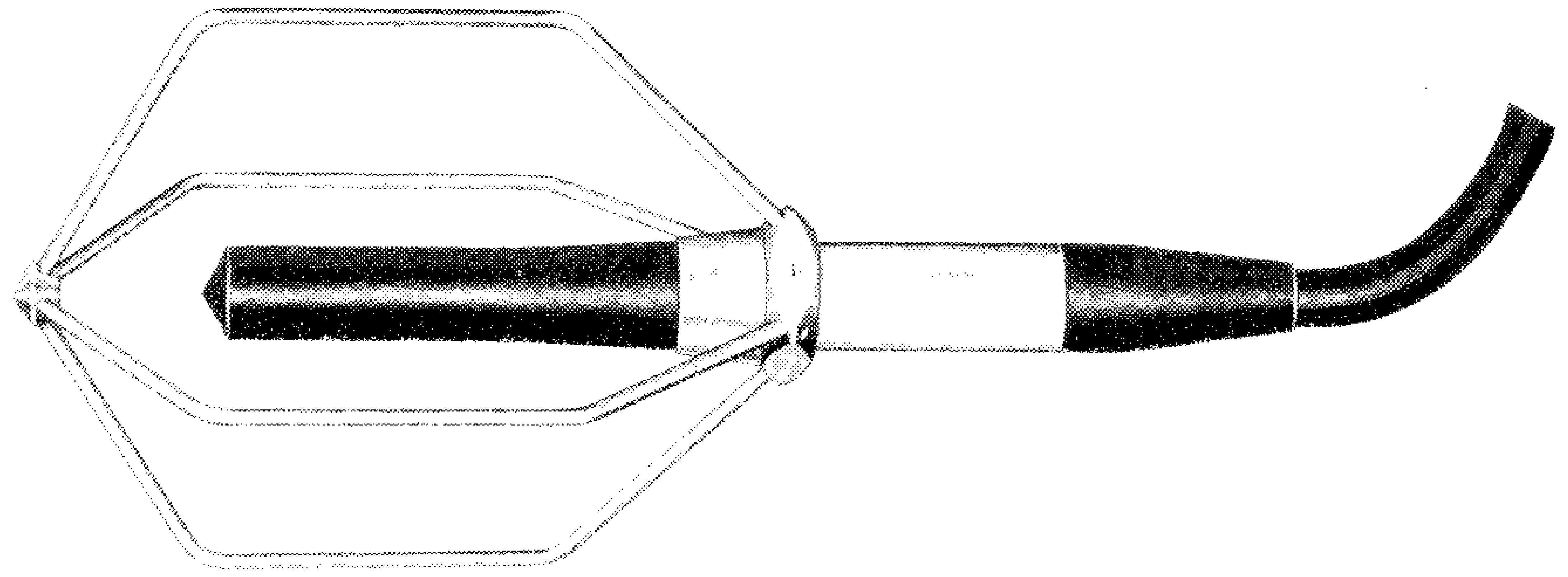
Permanent Magnet Exciter Body Type 4805

The Permanent Magnet Exciter Body Type 4805 forms the base of a highly flexible vibration exciter system capable of exciting any one of the five interchangeable heads from the 380 N rated V system. When the body is driven by the 220 VA Power Amplifier Type 2707 and the exciter heads air-cooled, the force rating and maximum acceleration limit will be between 64% and 71% of those quoted for the same heads when mounted on the electromagnetic Exciter Body Type 4801. However, displacement and velocity limits are unchanged. Forced air cooling of the exciter heads is facilitated by coupling a flexible air-hose into a tapered hole in the base of the body. For full force output, a cooling air flow of 0,42 m^3/min at 0,008 kg/cm^2 is necessary for which the use of an ordinary domestic vacuum cleaner is quite adequate.

For operation without air-cooling the allowable drive current is reduced by half and hence the attainable force and acceleration limits by half.

The body is supported on a sponge rubber annulus whereby its resonance frequency is kept below 20 Hz. The vibration transmission to the work table or floor is thus greatly attenuated. Also quieter operation of the exciter can be achieved, especially if the blower for air-cooling is kept remote from the body.

General Purpose Hydrophone Type 8101



The General Purpose Hydrophone Type 8101 is a wide range transducer for making absolute sound measurements in liquids and gases over the frequency range 1 Hz to 125 kHz with a receiving sensitivity of —184 dB re 1 V per μPa (—84 dB re 1 V per μbar). A built-in low noise FET preamplifier acts as an impedance converter to provide a non inverted signal suitable for transmission over long underwater cables. The hydrophone can also be used in air with minimal electrical interference, since the piezoelectric ceramic sensing element of lead zirconate titanate is well shielded from electrical noise. The sound transparent boot of moulded neoprene rubber is permanently bonded to a highly corrosion resistant monel-metal body which is electrically and vibrationally isolated from the sensing element.

The hydrophone is equipped with a 6 m waterblocked low-noise cable fitted with a waterproof extension converter through which another 1,2 m cable having a 7 pin socket at one end can be connected. The 7 pin socket permits connection to the wide range of B & K Measuring Amplifiers and Spectrometers.

The input circuitry of the preamplifier is self protected against an overdrive up to 140 V RMS without damage while the output circuit is short circuit protected. Mechanical protection of the sensing element is provided by a safety cage fitted to the head of the hydrophone which can be easily detached by removing two clamping screws.

A built-in 50 Ω precision resistor permits insert voltage calibration, while power between 12 to 24 V DC to the hydrophone can be supplied through the cable.

The hydrophone is intended for general purpose underwater sound measurements, for example, ship traffic noise, sea state spectra, sounds of marine animals, etc.

Test Records Type QR 2010 and QR 2011

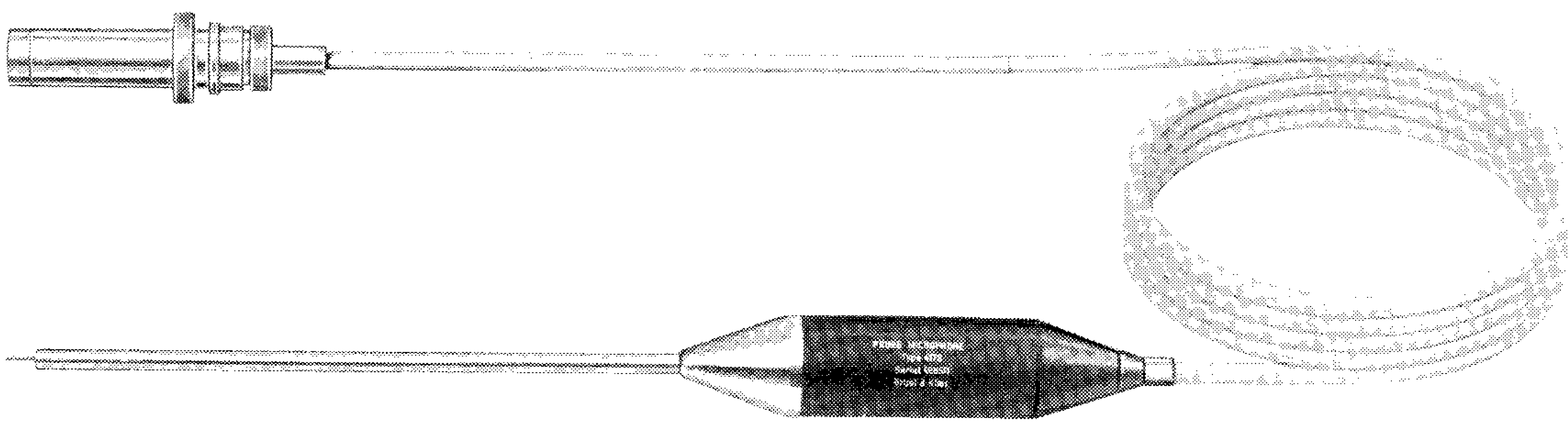
The materials for the records are carefully selected to obtain the best possible mechanical properties and a very low surface noise level. The label material and orientation are arranged to give minimum bending of the recorded surface.

Test Record Type QR 2010 is intended for laboratory measurements on pick-ups and detailed investigations on production samples which have improved considerably in performance on account of lighter construction with increased compliance and extended frequency range. Among the investigations possible with this record are frequency response and cross-table measurements of pick-ups as well as determination of maximum tracking ability. Besides, wow and rumble measurements on turntables and investigation of arm resonances can also be carried out.

Test Record Type QR 2011 is produced for testing and adjustment of Hi-Fi equipment in situ i. e. in homes or in demonstration rooms of dealers. The record provides easy and inexpensive means to users and dealers for evaluating quality of Hi-Fi equipment accurately and reliably. The record provides wide band noise and pink weighted noise in 1/3 octaves over different frequency ranges whereby manual and automatic response measurements as well as phase check of the entire system can be carried out in the listening room. For tracing resonating parts, a sinusoidal signal from 20 Hz to 1 kHz is also included.

All the recordings, it should be mentioned, are in accordance with IEC, RIAA, NAB, BS and DIN recommendation and standards.

Horn-coupled Probe Microphone Type 4170



The Probe Microphone Type 4170 has been designed to cover a wide range of measurements where minimum disturbance of the sound field is the primary criterion or where access to areas of critical sound fields is difficult with normal microphones. This is achieved by making the probe diameter extremely small (1,25 mm) and a very high orifice acoustic impedance. The probe is coupled to a 1/2" microphone (corresponding to Type 4134) through an acoustical exponential horn. By matching the acoustic impedance in the microphone end of the horn, the frequency response of the assembly is equalized and is flat from 30 Hz to 8 kHz within 4 dB.

The probe and the horn which are 60 mm and 120 mm long respectively are covered with a removable stainless steel tube which protects them and gives a uniform outer diameter.

A 1/2" microphone preamplifier similar to Type 2619 is mounted inside the handle and can be connected directly to frequency analyzers and spectrometers via a built-in cable and a seven pin plug.

The small size and low weight of the probe microphone make it suitable for measurements in the ear, inside intricate machinery, sound insulating materials, and for acoustical studies of musical instruments.

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