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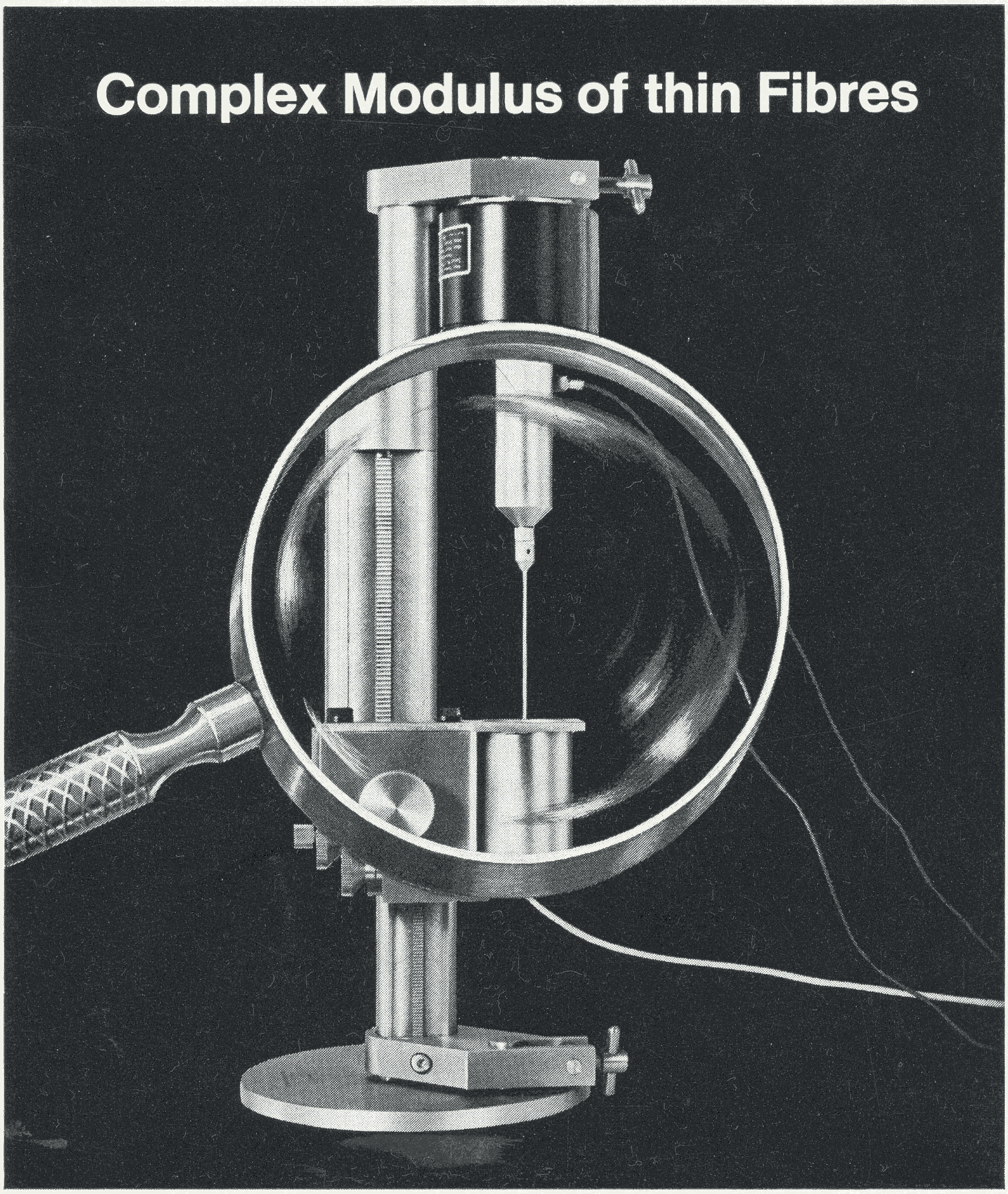

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TECHNICAL REVIEW

No. 2 – 1970

Contents

Measurement of the Complex Modulus of Elasticity of Fibres and Folios

by

B. Stisen

ABSTRACT

A theoretical treatment of a measuring method is given for the determination of the real and imaginary components of the complex modulus of elasticity for thin fibres and strips of folios.
The measurements and calculations for a textile fibre of diameter 0.02 mm are discussed.

ZUSAMMENFASSUNG

Die theoretische Behandlung einer Meßmethode zur Bestimmung des Real- und Imaginärteils des komplexen Elastizitätsmoduls für dünne Fiber und Folienstreifen wird gegeben. Die Messungen und Berechnungen eines Textilfibers mit dem Durchmesser 0,02 mm werden diskutiert.

SOMMAIRE

Exposé théorique d'un méthode de mesure pour la détermination des composantes réelles et imaginaires du module complexe d'elasticité, dans le cas de fibres de faible diamètre et de rubans de feuilles minces. Les mesures et calculs pour une fibre textile de diamètre 0,02 mm sont discutés.

Contents

1. Nomenclature

A	(dimensionless)	Amplification factor.
a	(m^2)	Cross section of specimen.
b	(Nsec/m)	Viscous damping coefficient.
d	(dimensionless)	Loss factor.
d_D	(dimensionless)	Loss factor, calculated by displacement excitation.
d_F	(dimensionless)	Loss factor, calculated by force excitation.
E	(N/m^2)	Complex modulus.
E'	(N/m^2)	Elastic modulus.
E''	(N/m^2)	Dissipative modulus.
F	(N)	Force.
f_r	(Hz)	Resonant frequency.
Δf_r	(Hz)	3 dB resonant bandwidth.
g	(m/sec^2)	Gravitational constant.
k	(N/m)	Stiffness of specimen.
L	(m)	Length of specimen.
M	(kg)	Mass.
O	(mm)	Octave length on recording paper.
P	(mm)	3 dB bandwidth on recording paper.
t	(sec)	Time.
x	(m)	Displacement.

α	(rad)	Loss angle.
ε	(dimensionless)	Strain.
η	(dimensionless)	Normalized frequency.
ϱ	(dimensionless)	Loss factor for $\eta = 1$.
σ	(N/m^2)	Stress.
φ	(rad)	Phase angle.
ω	(rad/sec)	Angular frequency.

2. Introduction

The equipment for the measurement of the damping capacity of materials has up to this day been developed to fit specific applications. Items which have physical characteristics based on the control of the parameters of the production process (cold or hot treatment, process time and temperature) have therefore been tested as complete specimens to maintain specific properties. Other specimens have such large or small physical dimensions that normally used methods are impractical if not impossible.
When a new method is used, care must be taken to understand the way in which the specimen is affected.
The damping capacity of a material is derived from measurements on a vibrating or oscillating system in which, ideally, all the elastic restoring forces are contributed by the specimen.
To have a single degree-of-freedom system for the measurements, three methods are used [1]:

A. The Resonant Bar Method
A bar having a uniform cross section is vibrated in a longitudinal or transverse mode to measure the damping capacity for tension-compression excitation. In the torsional mode the damping capacity for shear excitement can be determined. The bar is suspended to allow vibration in the natural frequency mode or in a higher one. The resonant frequencies can be fairly high.

B. Specimen with Integral Masses
The specimen is machined into three sections, thus two seismic masses are joined together by the third part acting as a spring element. The three parts, properly designed, allow measurements at high stress levels. Both longitudinal and torsional modes can be used. The energy dissipation due to the clamping of the specimen is avoided.

C. Specimen with Attached Masses
In a simple oscillating system the specimen acts as the spring element rigidly suspended at one end and loaded by a seismic mass clamped to the other end. The natural frequency in either the longitudinal or the torsional mode can only be used. The damping properties at different static stress levels can be determined.

For measurement of the damping capacity of fibres and folios method c will be used.
The commonly used terms describing the damping phenomena of materials and systems exposed to cyclic excitation are listed in the Appendix. The definitions are taken from references [2] and [3] and will be used in this article.

3. Principle of Measurement

The fibre to be examined is attached underneath a small shaker at one end and loaded by a mass at the other, see Fig. 1. The mechanical system consisting of the fibre and the mass is excited into vertical oscillations by the shaker (displacement excitation). By measuring the resonant frequency it is possible to calculate the elastic modulus of elasticity, i.e. the real component of the complex modulus. By recording the displacement resonant curve of the oscillating mass the loss factor is found. The displacement is measured by means of a capacitive transducer mounted below the mass.
The Complex Modulus Apparatus Type 3930 is widely used for measurement of the damping capacity of high damping materials. The excitation is created by the Magnetic Transducer Type MM 0002 acting as a force generator. A theoretical treatment of the single degree-of-freedom system is given in the two following sections. In the first section the system is excited by an oscillating force of constant amplitude acting on the mass. The second section relates this excitation to the displacement excitation which is used in the actual measurements. The results show that for normal values of the damping capacity the measured loss factor is valid for both systems of excitation.

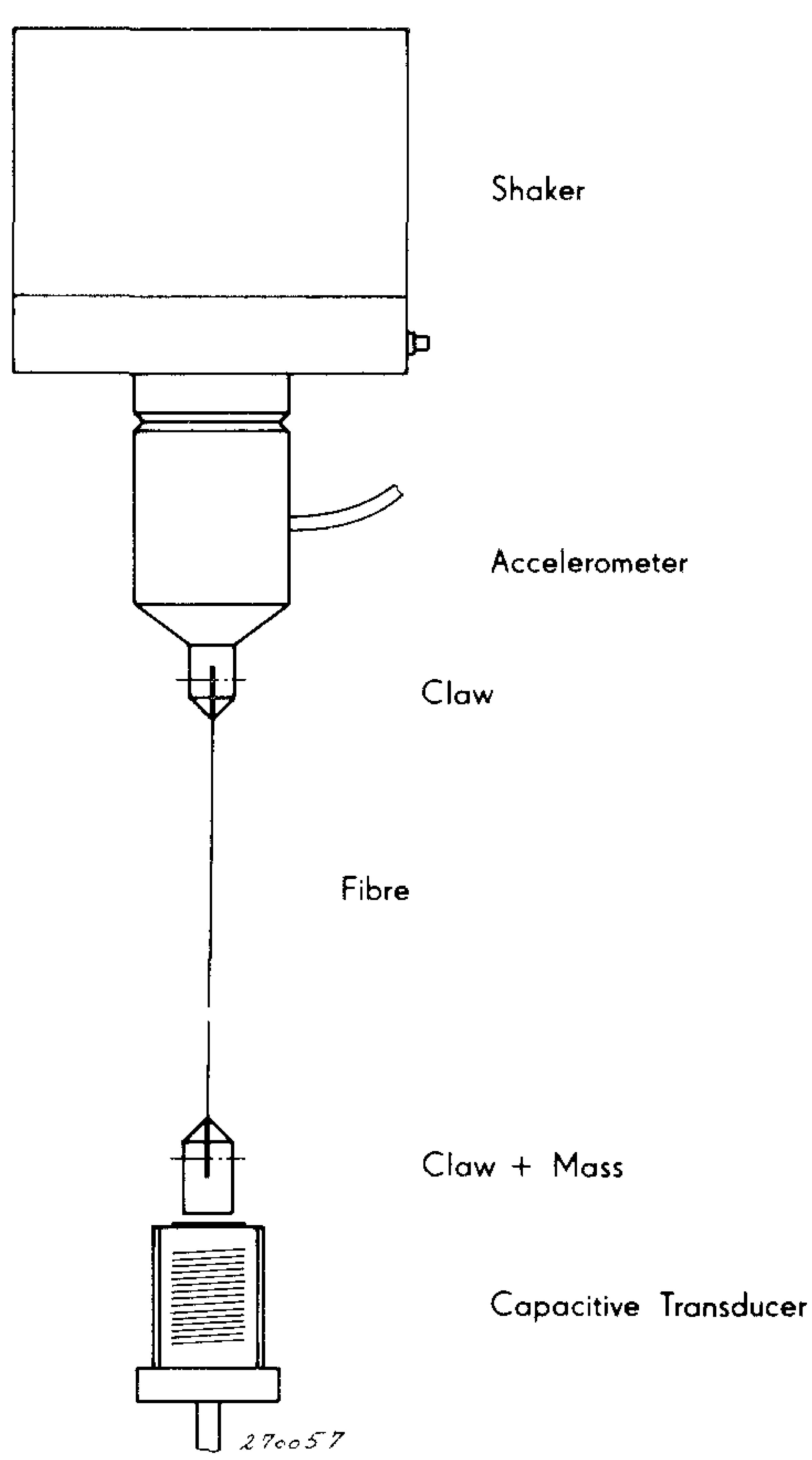

Fig. 1. Mounting of the fibre.

4. Analysis of a Force Excited Single Degree-of-Freedom System

For a single spring-mass system with viscous damping, Fig. 2, oscillating vertically the sinusoidal movement is described in the steady-state condition by:

$$x = x_o\, e^{j\omega t} \qquad \text{(I)}$$

excited by

$$F = F_o\, e^{j(\omega t + \varphi)} \qquad \text{(II)}$$

The force vector leads the displacement vector by an angle φ.

The equation of motion is given by:

$$M \frac{\delta^2 x}{\delta t^2} + b \frac{\delta x}{\delta t} + kx = F$$

By substituting for $\frac{\delta^2 x}{\delta t^2}$, $\frac{\delta x}{\delta t}$, x and F the equation becomes:

$$(-M\omega^2 + jb\omega + k)\, x_o = F_o\, e^{j\varphi}$$

or

$$(1 - \frac{M}{k}\omega^2) + j\frac{b}{k}\omega = \frac{F_o}{x_o k} e^{j\varphi} \qquad \text{(III)}$$

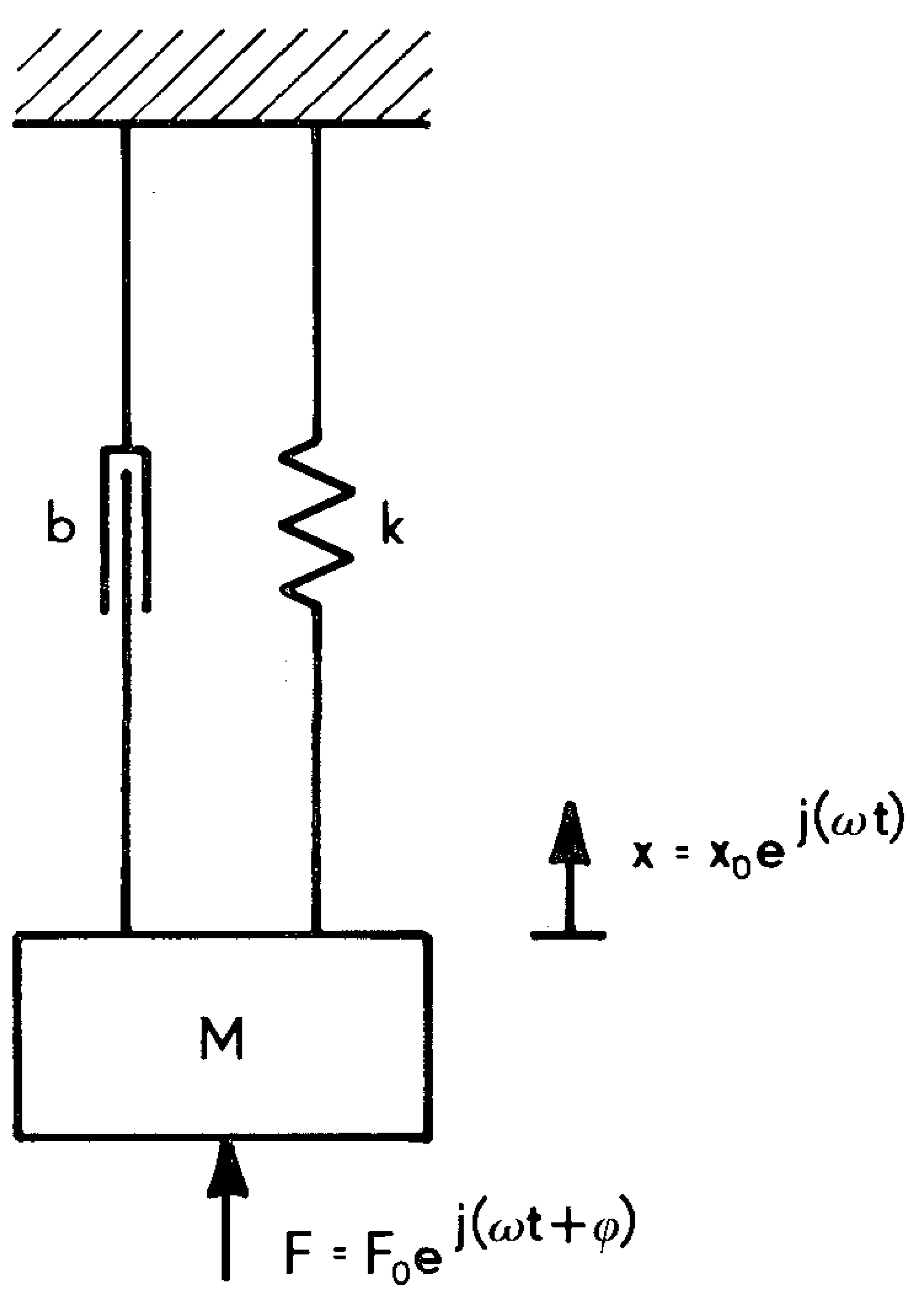

Fig. 2. Principle of the force excited system.

The static deflection $x_{o,s}$ is found by setting $\omega = 0$:

$$kx_{o,s} = F_o(\cos\varphi + j\sin\varphi)$$

whence:

$$\varphi = 0$$

$$x_{o,s} = \frac{F_o}{k}$$

To normalize equation (III) the following terms are used:

Amplification Factor:

$$A = \frac{x_o}{x_{o,s}} = \frac{x_o k}{F_o}$$

Resonant angular frequency of undamped system:

$$\omega_* = \sqrt{\frac{k}{M}}$$

Normalized angular frequency:

$$\eta = \frac{\omega}{\omega_*}$$

Loss Factor:

$$d = \tan\alpha = \frac{b\omega}{k} = \frac{b\omega_*}{k}\eta = \varrho\eta$$

Equation (III) now takes the form:

$$(1 - \eta^2) + j\varrho\eta = \frac{1}{A}\, e^{j\varphi} \qquad \text{(IV)}$$

The real and imaginary components are:

$$1 - \eta^2 = \frac{1}{A}\cos\varphi$$

$$d = \varrho\eta = \frac{1}{A}\sin\varphi$$

Equation (IV) relating the above defined terms can be graphically expressed by a parabola in a complex plane [5], see Fig. 3. For $\eta = 0$ the system is in static equilibrium corresponding to $\frac{1}{A} = 1$ and $\varphi = 0$. For increasing frequency the modulus of the vector $\frac{1}{A}$ decreases until minimum at *OR*, where *A* is maximum. This is the displacement resonance. For a further increase in frequency the amplification factor decreases and the loss factor still increases.

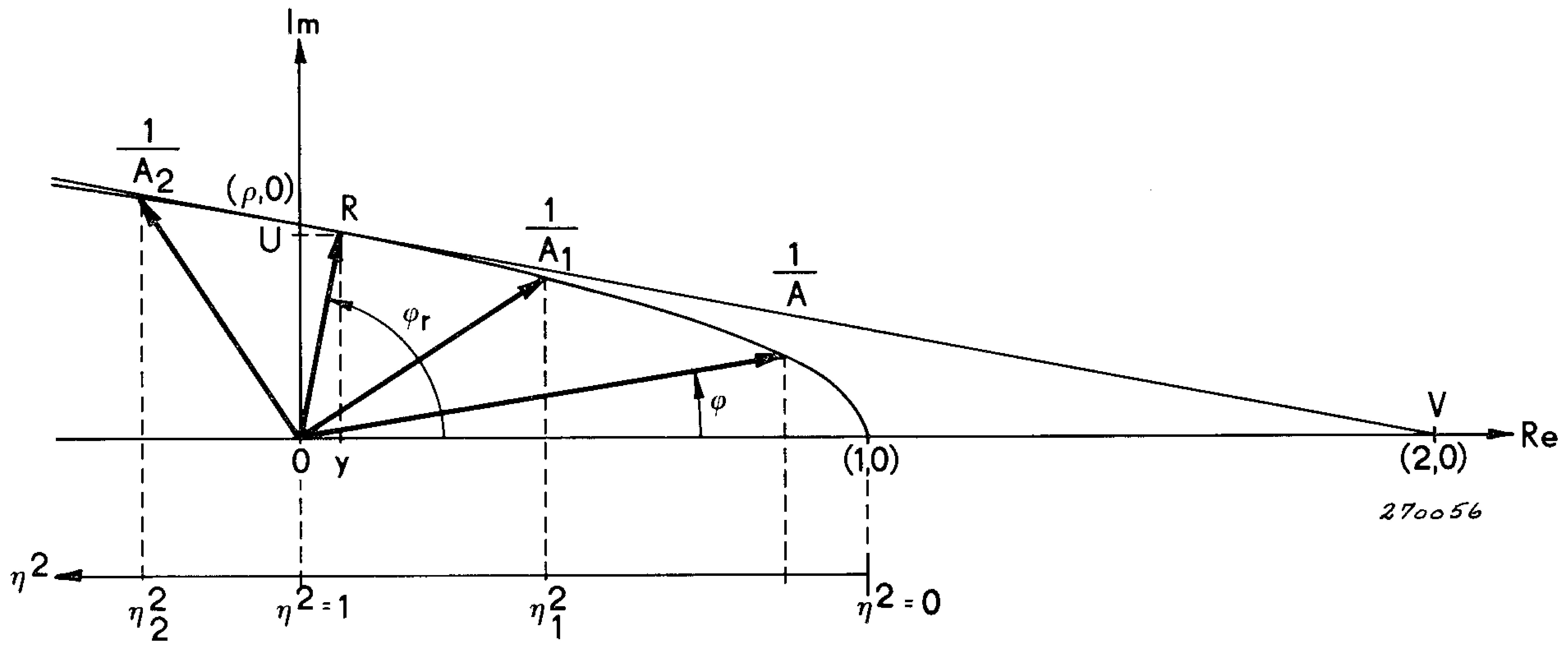

Fig. 3. Graphic representation of the damping phenomenon.

The 3 dB resonant bandwidth is indicated by the subscripts 1 and 2.

From the two similar triangles *OUR* and *VYR* in Fig. 3 the resonant frequency of the damped system can be found:

$$\frac{1-\eta_r^2}{d_r} = \frac{d_r}{1+\eta_r^2} \quad \text{where } d_r \text{ is the loss factor at resonance.}$$

$$\eta_r = \sqrt[4]{1-d_r^2} \approx 1 - \frac{d_r^2}{4} \qquad \text{(V)}$$

5. Analysis of a Displacement-Excited Single Degree-of-Freedom System

The excitation created by a sinusoidal displacement of the point of suspension (see Fig. 4) is described by:

$$s = s_o e^{j(\omega t + \varphi)}$$

The resulting movement of *M:*

$$x = x_o e^{j(\omega t)}$$

The equation of motion (not incorporating the constant gravitational force) is:

$$M\frac{\delta^2 x}{\delta t^2} + b\,\frac{\delta\,(x-s)}{\delta t} + k\,(x-s) = 0$$

By substituting for $\frac{\delta^2 x}{\delta t^2}$, $\frac{\delta x}{\delta t}$, x, $\frac{\delta s}{\delta t}$ and s we obtain

$$\left[\left(1-\frac{M}{k}\,\omega^2\right) + j\,\frac{b\omega}{k}\right] x_o = \left(1 + j\,\frac{b\omega}{k}\right) s_o e^{j\varphi}$$

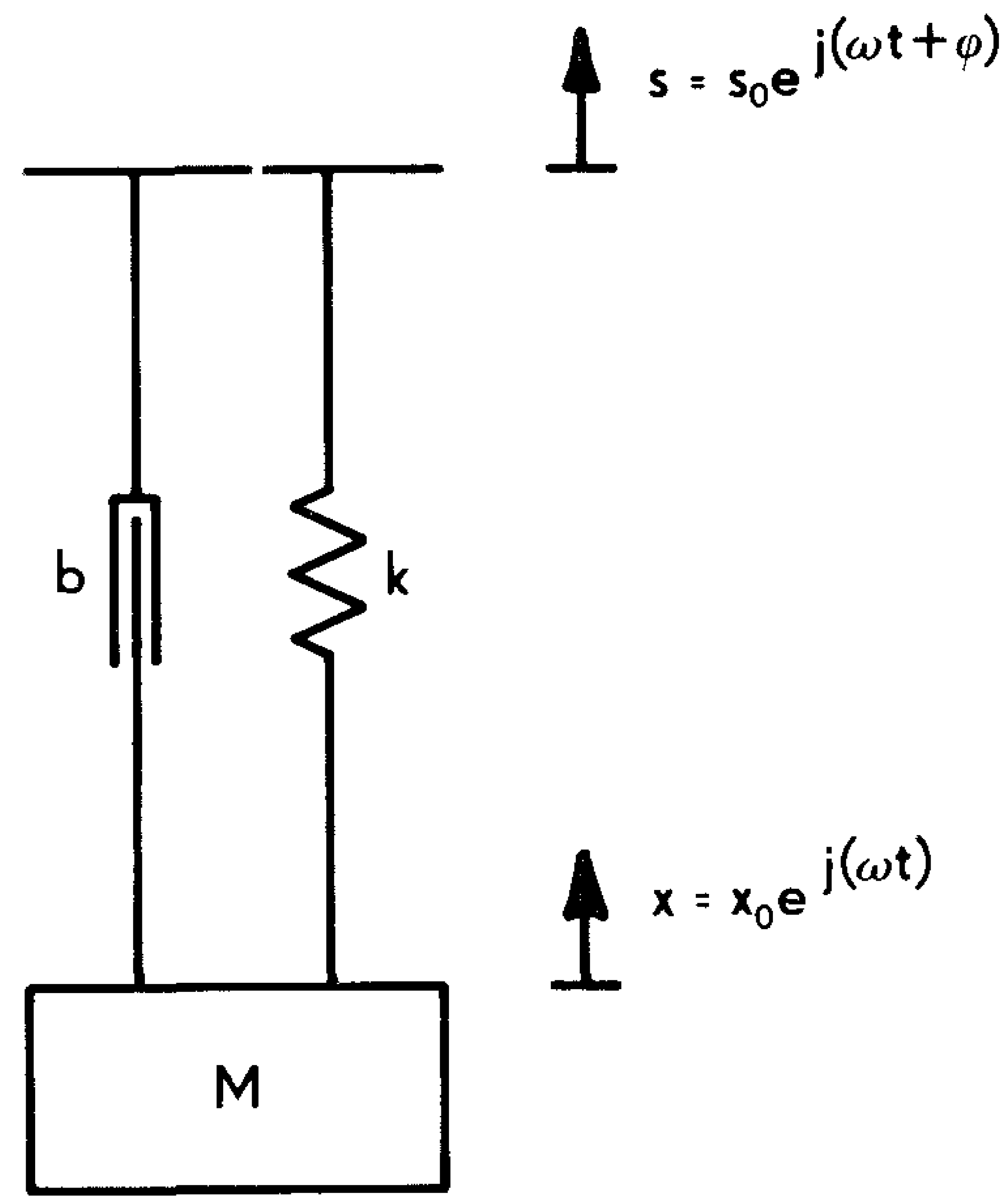

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Fig. 4. Principle of the displacement excited system.

The amplification factor

$$A = \frac{x_\circ}{s_\circ}$$

together with the terms previously used gives:

$$(1 - \eta^2) + j\varrho\eta = (1 + j\varrho\eta)\,\frac{1}{A}\,e^{j\varphi}$$

which can be written as:

$$(1 - \frac{\eta^2}{1 + \eta^2\,\varrho^2}) + j\,\frac{\varrho\eta^3}{1 + \eta^2\,\varrho^2} = \frac{1}{A}\,e^{j\varphi} \qquad \text{(VI)}$$

Equation (VI) corresponds to equation (IV) and relates the parameters describing the oscillation. For fixed value of ϱ the two equations are compared in Fig. 5. For low damping, $\varrho = 0$, equation (VI) can be substituted by

$$(1 - \eta^2) + j\varrho\eta^3 = \frac{1}{A}\,e^{j\varphi}$$

For both curves resonance is determined by the minimum value of the modulus of the vector $\frac{1}{A}$. The point of interest when comparing the two excitation systems is obviously the deviation between the two calculated loss factors.
A general picture can be achieved by looking at the curves. The decrease in the resonant frequency for the displacement excited system is quite evident at $\varrho = 0.316$.

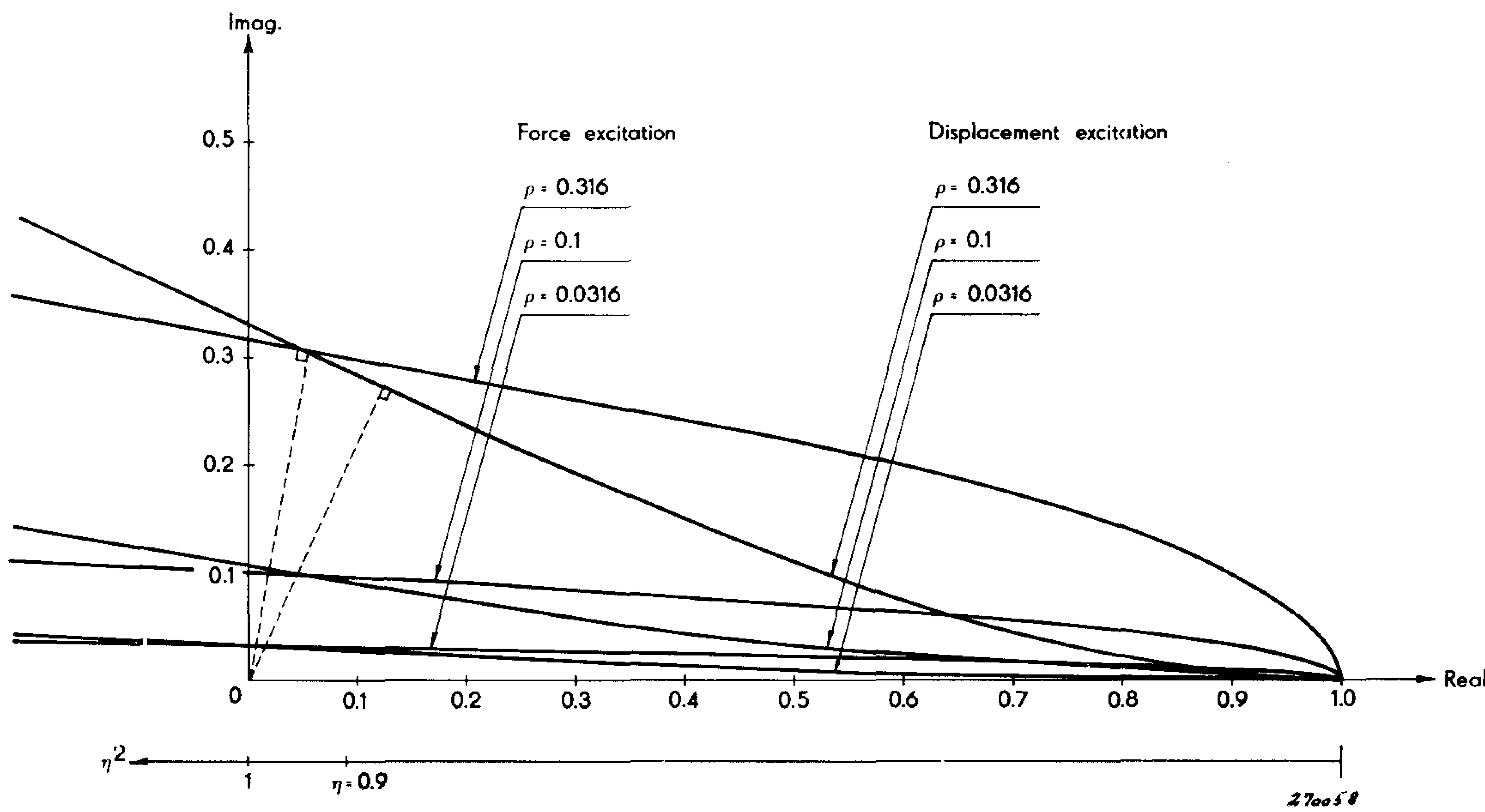

Fig. 5. Graphical comparison of force and displacement excitation.

For a particular value of ϱ the deviation in loss factors can be seen from the two corresponding resonant points on the curves. The imaginary components of the vector $\frac{1}{A}$ are the loss factors:

$$d_F \text{ and } d_D$$

where

$$d_F > d_D$$

The deviation $d_F - d_D$ has been calculated on a digital computer for different values of ϱ.

Fig. 6 shows the percentage deviation

$$\frac{d_F - d_D}{d_F} \times 100\,\%$$

as a function of d_F.

For loss factors smaller than 0.07 the deviation is less than 1 %. In this case the theory for the force excitation of the same system can be applied on account of the error being less than experimental uncertainties.

i.e. in the expression for η_r (V)

$$\frac{d_r^2}{4} \ll 1.$$

Therefore $\eta_r \approx 1$

and $\omega = \sqrt{\dfrac{k}{M}}$

For the fibre $k \times x = \sigma \times a$, where σ = stress and a = cross-sectional area.

From Hookes Law

$$\sigma = E' \times \varepsilon = E' \frac{x}{L}.$$

The dynamic elastic modulus is now found by substitution to be

$$\underline{E' = (2\pi)^2 \frac{f_r^2 \times L \times M}{a}}. \qquad \text{(VII)}$$

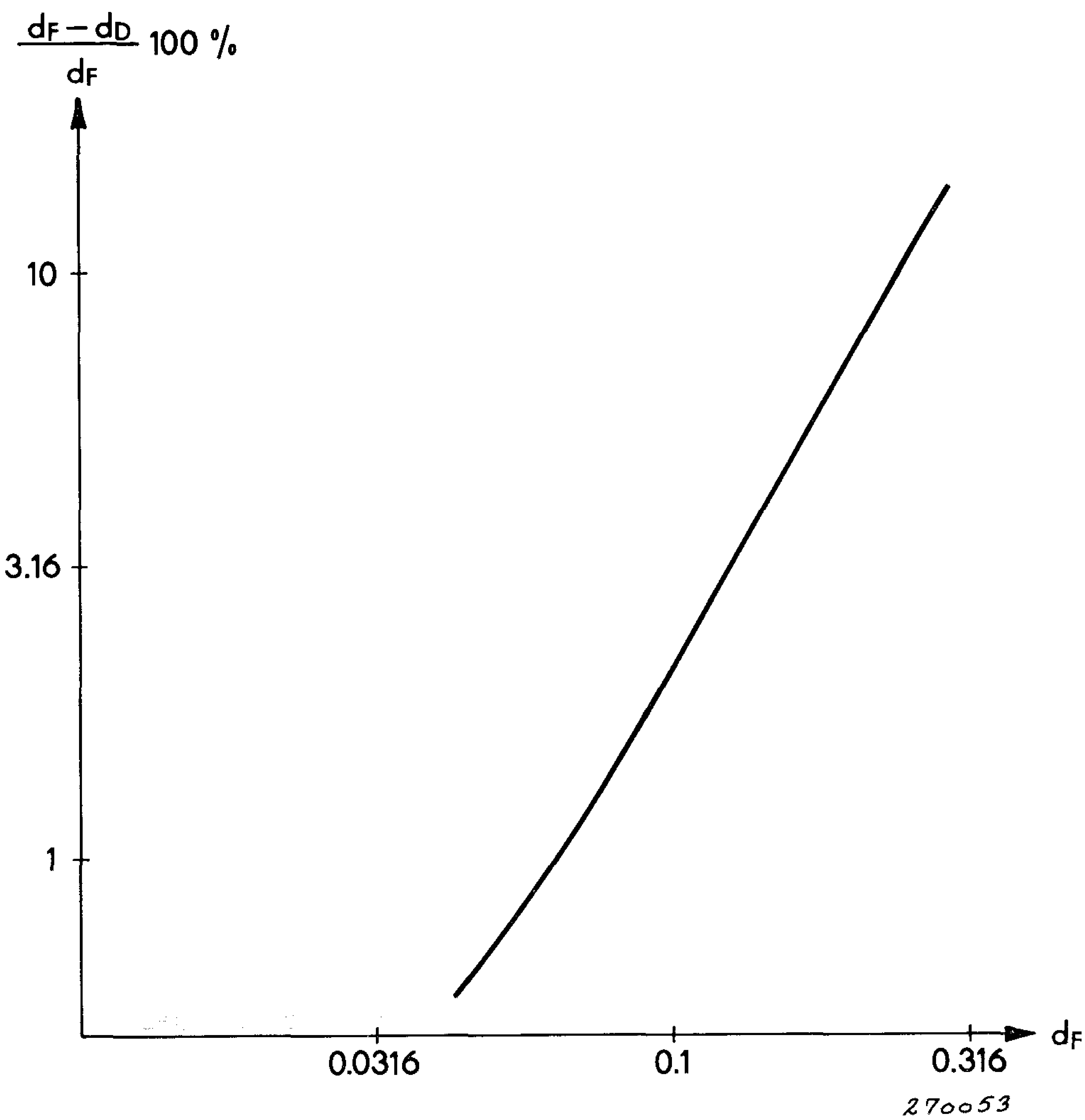

Fig. 6. Difference of the calculated loss factor for force and displacement excitation.

The loss factor can be calculated from

$$d = \frac{\Delta f}{f_r}.$$

When the resonance curve is plotted on a logarithmic scale, this expression for d can be shown to be

$$\underline{d = 2^{\left(\frac{P}{O}\right)} - 1} \qquad \text{(VIII)}$$

where O = octave length in mm on recording paper and
P = 3 dB bandwidth in mm, see lit. [7].

6. Experimental Set-up

The experimental set-up is shown in Fig. 7. In Fig. 8 the Complex Modulus Apparatus Type 3930 with the Mini Shaker Type 4810 is shown. A house containing an accelerometer and a claw for the attachment of the specimen is screwed to the moving part of the Shaker. The acceleration signal controls the excitation level of the point of suspension of the fibre. For constant acceleration, which is obtained by using the compressor circuit, the amplitude of displacement is given by

$$s_o = \frac{1}{\omega^2} \times \text{constant.}$$

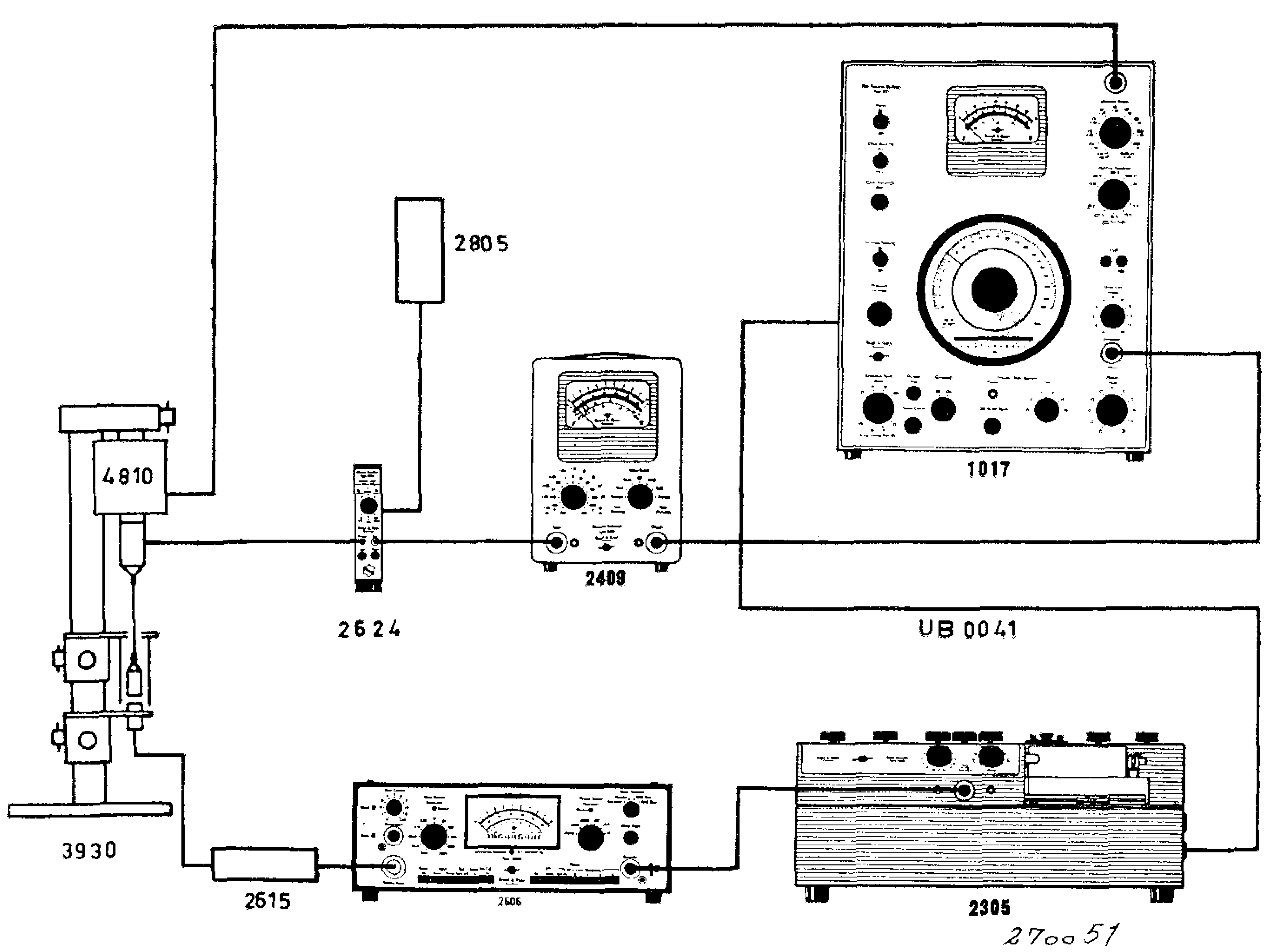

Fig. 7. Experimental set-up.

When a resonance curve is recorded ω changes and consequently so does s_o. The effect of frequency sweeping can be calculated mathematically. However, practical measurements have shown that the effect is negligible for $d < 0.1$.

For $d > 0.1$ double integration of the acceleration signal is recommended to have a signal proportional to displacement. In this way s_o itself is kept constant.

A claw is used for attachment of the mass to the specimen. To this claw further mass-units can be screwed in order to obtain the desired pre-stressed condition. If the mass is not grounded electrically, a shield can be placed around the mass and the top of the transducer. In that case the mass must be given zero potential before the measurements are carried out. For correct displacement detection, a DC voltage on the central electrode of the capacitive transducer, must exist relative to the mass.
The transducer is mounted approx. 1 mm beneath the mass. The Level Recorder is fitted with a 10 dB Potentiometer and synchronized with the Beat Frequency Oscillator.

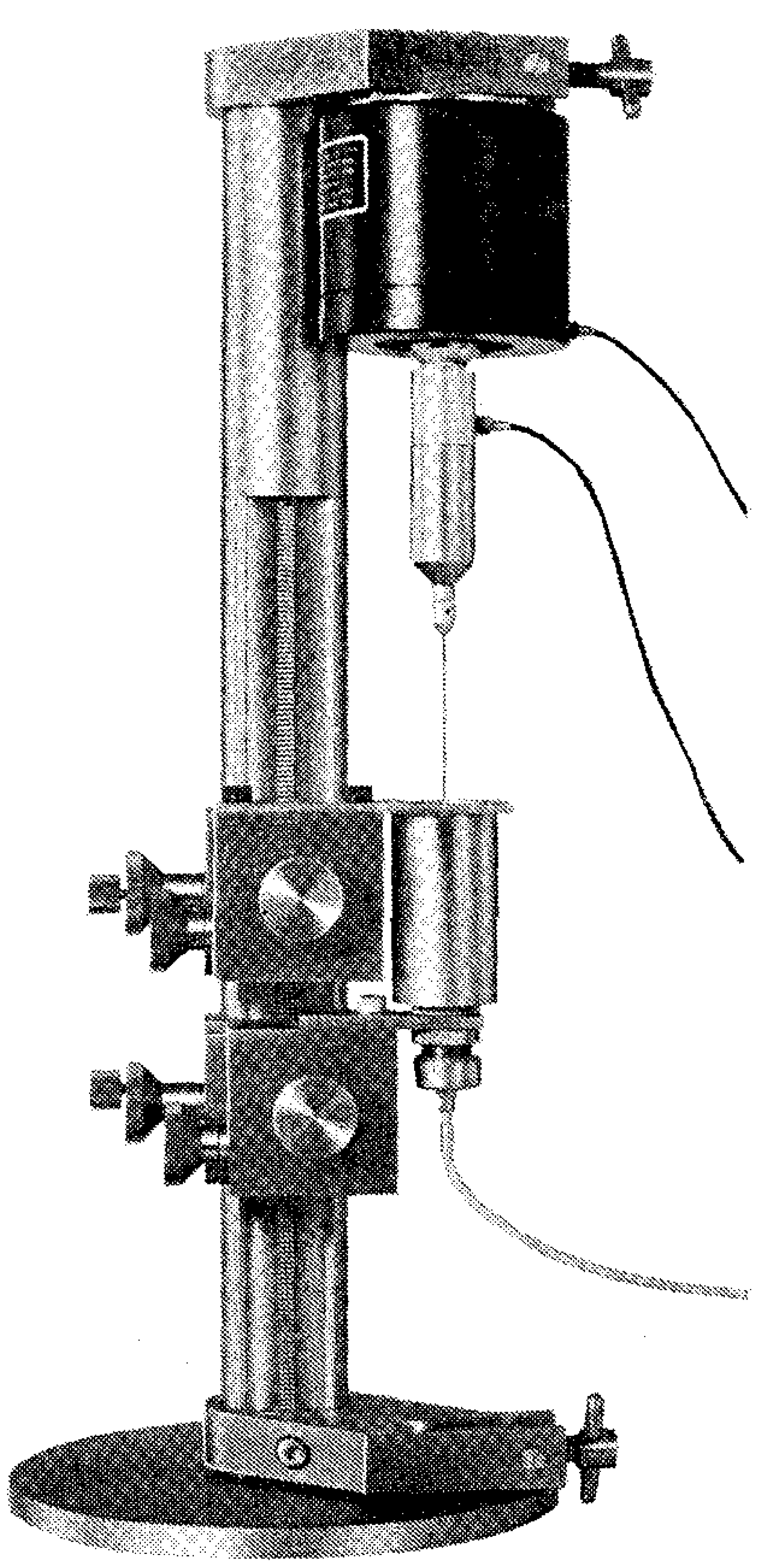

Fig. 8. The modified Complex Modulus Apparatus Type 3930.

7. Limitations of the Principle

The resulting force acting on the mass is given by

$$F = M \frac{\delta^2 x}{\delta t^2} = Mg - \left(kx + b \frac{\delta x}{\delta t} \right)$$

As the motion increases at resonance the following condition must be fulfilled to avoid non-linearity

$$kx + b \frac{\delta x}{\delta t} < Mg.$$

For a specimen not having linear stiffness and damping properties, (which is quite normal) the motion must be even smaller.
It is important to isolate the vertical oscillation from other possible oscillations of the system. The most disturbing one is rotational oscillation of the mass about a horizontal axis. This oscillation can be prevented if the distance between the point of suspension of the mass and the centre of gravity is minimized.
To avoid pendulating oscillations excited by environmental vibrations the Modulus Apparatus must be placed on a rigid table isolated against low frequency vibrations.

8. Measurement on a Textile Fibre

In Table 1 are shown 4 typical measurements of a textile fibre. The fibre diameter was 2.1×10^{-2} mm $\pm$ 3 %. A slow scanning speed is used for the recordings of the resonance curves, namely

Paper Speed 1 mm/sec.
Drive Shaft Speed 0.36 r.p.m.

E' is calculated from equation (VII).

Measurement no.	M	L	f_r	E'	d
1	3.459×10^{-3} kg	0.120 m	20.5 Hz	2.03×10^{12} N/m^2	0.031
2	11.532×10^{-3} kg	0.120 m	11.9 Hz	2.28×10^{12} N/m^2	0.031
3	11.532×10^{-3} kg	0.039 m	20.6 Hz	2.22×10^{12} N/m^2	0.034
4	3.458×10^{-3} kg	0.039 m	36.3 Hz	2.07×10^{12} N/m^2	0.054

Table 1. Measurements on a textile fibre.

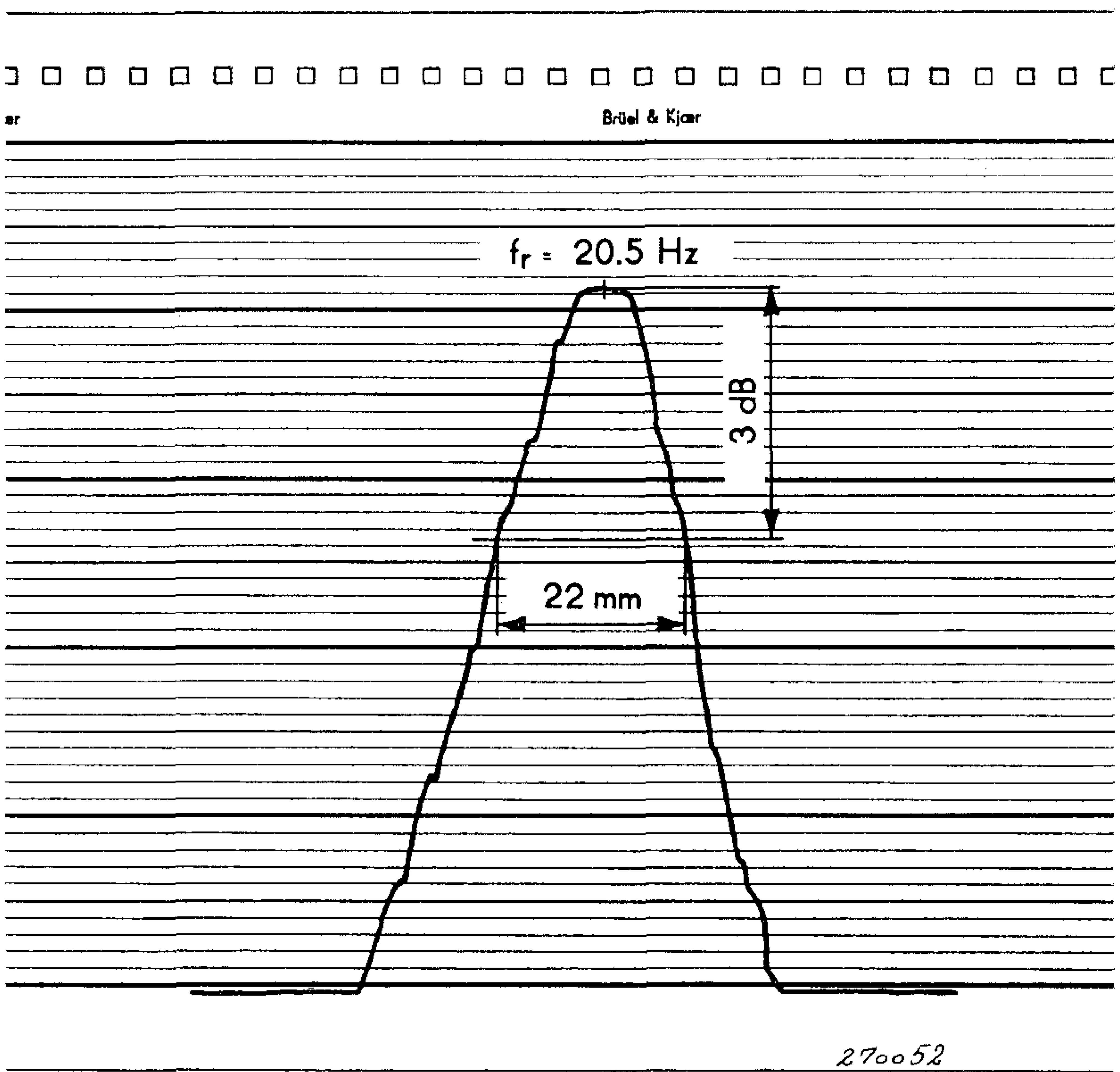

Fig. 9. Resonance curve.

The resonance curve for measurement No. 1 is shown in Fig. 9.

The loss factor is calculated from equation (VIII).

$$P = 22 \text{ mm}$$

$$O = \frac{1 \text{ mm/sec} \times 60 \text{ sec/min} \times 3 \text{ r/oct.}}{0.36 \text{ r/min}} = 500 \text{ mm/oct.}$$

$$d = 2^{\left(\frac{22}{500}\right)} - 1 = 0.031$$

From further calculated results the following characteristics were evident:

1. The loss factor increases with increased pre-stressed condition.
2. The loss factor increases with increased excitation frequency.

9. Conclusion

A theoretical treatment of an oscillating system with attached mass has been given for ideal viscoelastic materials. Most materials do not have damping properties exactly proportional to the exciting velocity which was assumed in the theoretical treatment of the force and displacement excitation systems. However, it is hoped that the article will give an idea of the inaccuracies involved in using one set of formulas and constants for both systems. For loss factors smaller than 0.2 the inaccuracy using the simple relationship is negligible compared to the uncertainties involved in the measurements.

In the expression for the dynamic modulus of elasticity the resonant frequency appears in equation (VII) to the second power, and should therefore be measured by a frequency counter to increase the accuracy. The cross section of a specimen cannot be measured to an accuracy better than 2–5 %. However, for production control and development it might be unnecessary to incorporate the cross section in the calculations and thus avoid the involved indetermination.

The relative changes in the modulus of elasticity due to different preloads are measured without knowledge of the cross sectional area.

The experimental set-up described is a result of research and development of a special equipment for measurement of the dynamic characteristics of thin fibres in the longitudinal direction. A magnetic velocity sensitive transducer was also used to measure the response, but as the magnetic attraction affects the pre-stressed condition of the specimen it is not recommended. The environmental damping, i.e. air friction and damping due to clamping has no influence for $d > 0.001$.

Although more measurements show a change in the elastic modulus for different pre-stressed conditions a complete non linear stress-strain curve cannot be drawn from this information. As regards the damping capacity, the combined influence of excitation frequency and the pre-stressed condition can be determined.

The theory of displacement-excited system with attached mass might be used advantageously for other oscillating systems. The Danish Concrete Research Laboratory has carried out measurements of the damping capacity of wet and hardened concrete samples. The concrete sample (being the combined spring and damper element) was mounted on a shaker and loaded by a heavy mass at the top. The displacement of the mass was measured by an accelerometer.

10. Literature

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Appendix

Listed below are the definitions of the commonly used terms describing the damping phenomena of materials and systems exposed to cyclic excitation. Most definitions are taken from [2] & [3].

Amplification Factor, A

The ratio of the displacement amplitude ($x_{o,\ resonance}$) of a vibration excited by an alternating force at the resonant frequency of a system to the deflection ($x_{o,\ s}$) produced by the same force applied statically:

$$A = \frac{x_{o,\ resonance}}{x_{o,\ static}}$$

Bandwidth, Δf_r

The difference between the two frequencies, one above and one below the resonant frequency, which has a displacement amplitude of $1/\sqrt{2}$, (– 3 dB), of the resonant displacement.

Complex Modulus, E

The complex modulus (E) for a material is defined in terms of the steady-state response to force excitation. The real part (E′) is the portion of the stress in the spring element acting in phase with the displacement divided by the total displacement. The imaginary part (E″) is that part, which leads the displacement by 90° divided by the total displacement.

$$E = E' + jE''$$

Since the real and imaginary terms refer to the components of a phasor, these concepts only have direct meaning for sinusoidal motion.

Linear System

For a linear system the amplitude of motion (displacement, velocity or acceleration) is proportional to the exciting force, both expressions being complex quantities. The linear dynamic range of an oscillating system is identically defined as the dynamic range exhibiting linearity between the applied force and the resulting motion.

Logarithmic Decrement, δ

The natural logarithm of the ratio between two successive displacement amplitudes of the decaying resonance oscillation without excitation:

$$\delta = ln \frac{x_n}{x_{n+1}}$$

Loss factor, d

The loss factor is by definition equal to the ratio of the imaginary part of the spring force, which leads the displacement by 90° to the real part in phase with the displacement. Accordingly, the loss factor is only usable for sinusoidal vibration.

Material Damping

Material damping as treated in this paper defines the energy dissipation from mechanical energy to heat energy under cyclic stress.

Quality Factor, Q

The quality factor is a measure of the sharpness of a resonance or frequency selectivity of a resonant oscillating system. For the use in mechanical analysis the following definition is used:

$$Q = 2\pi \frac{W_{store}}{\Delta W}$$

ΔW is the dissipated energy during one cycle and W_{store} is the stored energy at the beginning of the cycle.

Resonant Frequency, f_r

The resonant frequency of a forced damped single degree-of-freedom system for steady-state conditions is the frequency at which the displacement amplitude is maximum. This frequency is lower than the resonant frequency of the corresponding undamped system.

Specific Damping Capacity, C

The percentage ratio of the energy dissipated during one cycle to the energy stored at the beginning of the cycle:

$$C = \frac{\Delta W}{W_{store}} \times 100\,\%.$$

The amplification factor, quality factor, loss factor and logarithmic decrement are related by

$$\frac{1}{A} = \frac{1}{Q} = d = \frac{\varrho}{\pi}$$

For loss factors smaller than 0.2 the inaccuracy is less than 1.5 %.

Brief Communications

The intention of this section in the B & K Technical Reviews is to cover more practical aspects of the use of Brüel & Kjær instruments. It is meant to be an "open forum" for communication between the readers of the Review and our development and application laboratories. We therefore invite you to contribute to this communication whenever you have solved a measurement problem that you think may be of general interest to users of B & K equipment. The only restriction to contributions is that they should be as short as possible and preferably no longer than 3 typewritten pages (A 4).

Automatic Recording-Control System for Tape Recorders Type 7001

by

Vladimir Kop

The System is intended to control the automatic recording of short duration phenomena which occur with unknown (often long) intervals. The initial objective of the system was the recording of sonic bangs, but it can readily be used for other purposes.

Principle of Operation (see Fig. 1)
The signals are to be recorded on a tape recorder supplied with a full reel of tape. In order to record the signals (occurring at various intervals) closely spaced on the recording tape the tape recorder must be started and stopped once for each signal. As the start of a tape recorder takes some time, it is necessary to use another tape recorder as a memory element. The second tape recorder is supplied with a closed tape loop and is continuously recording the output from a transducer. A detection unit monitors the output of this tape recorder and delivers a logic "one" output when the event to be recorded has occurred. This starts the recording procedure:
The "loop tape recorder" is shifted to the playback mode after a time delay which determines recording time after the detected event.
The "reel tape recorder" is started in the recording mode after another time delay from detection time and it records the signal played back on the "loop

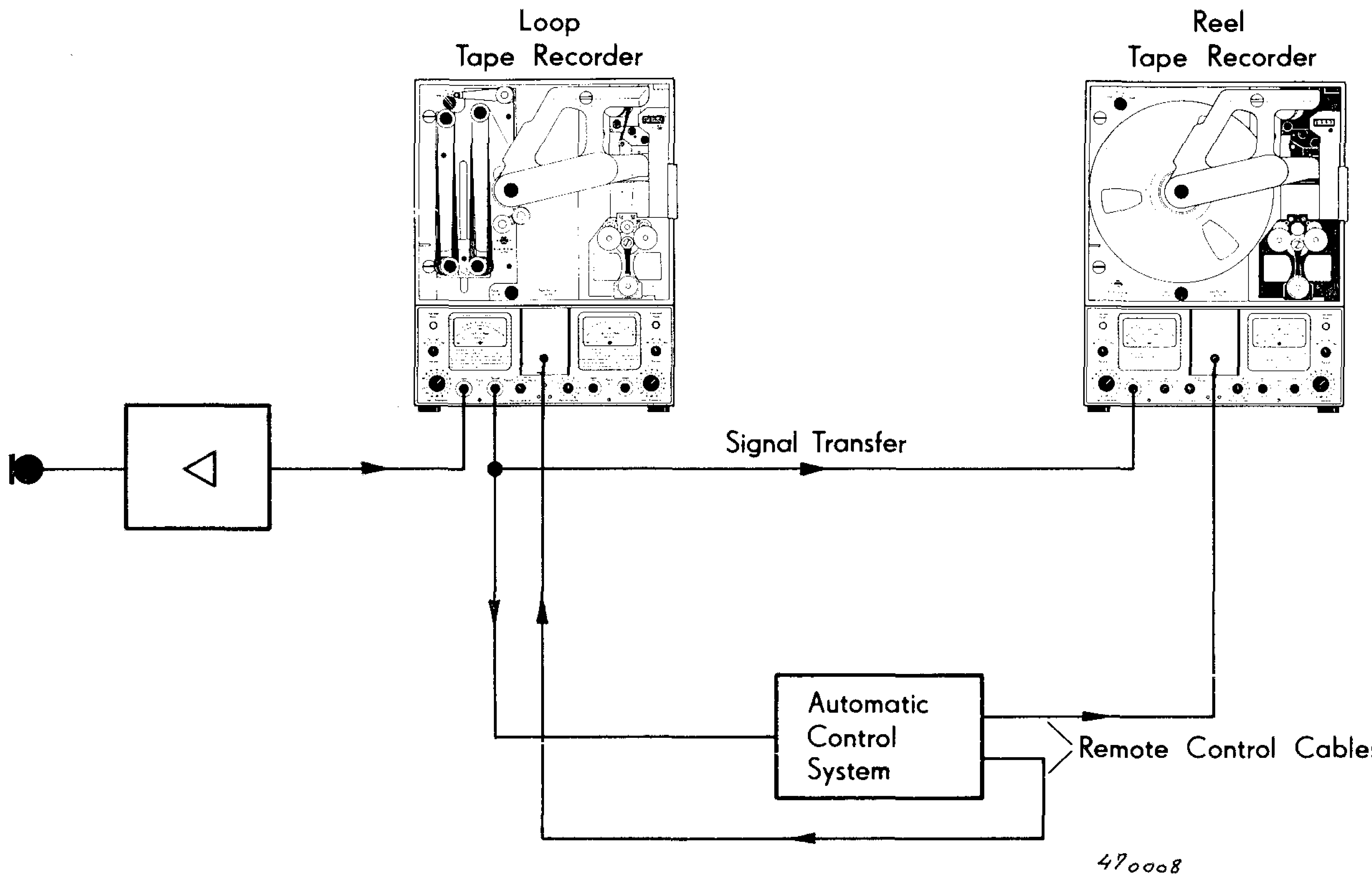


Fig. 1. Block-diagram of the measuring set-up.

tape recorder". This time delay determines the recording time before the event on the final recording (in combination with starting time of the tape recorder and with the passing time for the tape loop on the "loop tape recorder").
When the signal on tape loop passes the play back head a second time, the detection unit again delivers a logic "one" output. After a time delay (recording after event) this resets the system to its original state ("reel tape recorder" stopped and "loop tape recorder" in the recording mode).
Note: The tape loop has to be erased before another signal can be correctly recorded. This is done during one complet passing of the tape loop over the recording head without input signal. In the present version of the control system, however, no measure has been taken to prevent input signals in this time.
Figs. 2 and 3 give a detailed block-diagram and a time-diagram of the signal sequence in the control system. The system consists of a number of logic modules which can be realized in different ways. In the experimental set-up, however, the functions were obtained by using TTL logic integrated circuits combined with discrete components and relays.
The system is set into operation with the "loop tape recorder" in the recording mode and the "reel tape recorder" at stand still. When a signal occurring on

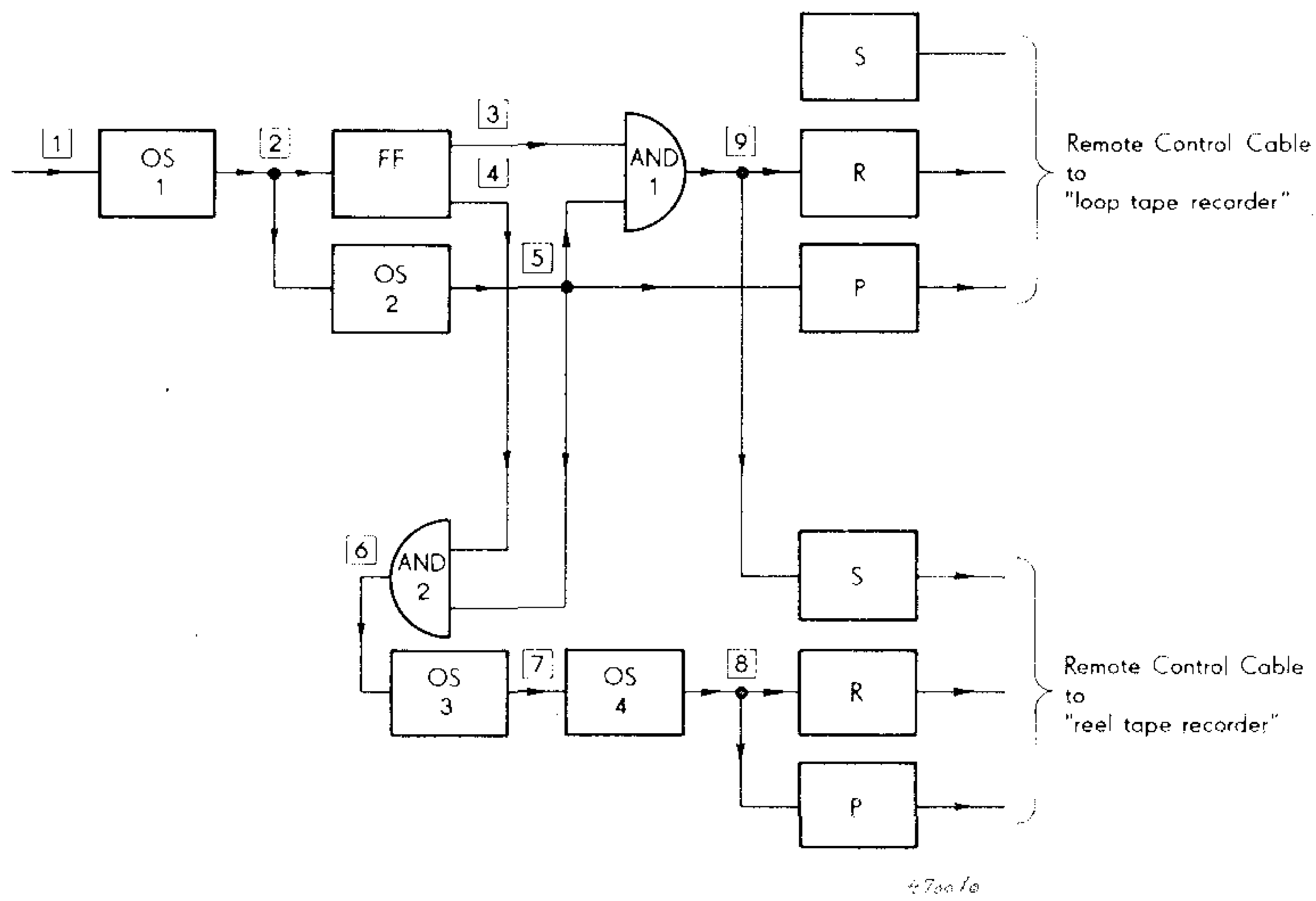


Fig. 2. Detailed block-diagram of the automatic control system.

the output of the "loop tape recorder" is detected the following series of control signals is activated:

1. Input signal from the detection circuit. Occurs when a signal has just been recorded and also after one tape loop revolution.
2. Output signal from OS 1.
3. Output signal from FF.
4. Inverse output signal from FF.
5. Output signal from OS 2. "Loop tape recorder" changes to the playback mode.
6. Output signal from AND 2.
7. Output signal from OS 3.
8. Output signal from OS 4. "Reel tape recorder" starts in the recording mode.
9. Output signal from AND 1. "Reel tape recorder" stops and "loop tape recorder" changes to the recording mode.

The abbreviations used in the diagrams and the text are

OS = One shot multivibrator.
FF = Flip Flop.
AND = And-Gate.
R = relay driver and relay with the same contact configuration as the recorder-button of the Tape Recorder.
P = relay driver and relay with the same contact configurations as the playback-button of the Tape Recorder.
S = Relay driver and relay with the same contact configuration as the stop-button of the Tape Recorder.

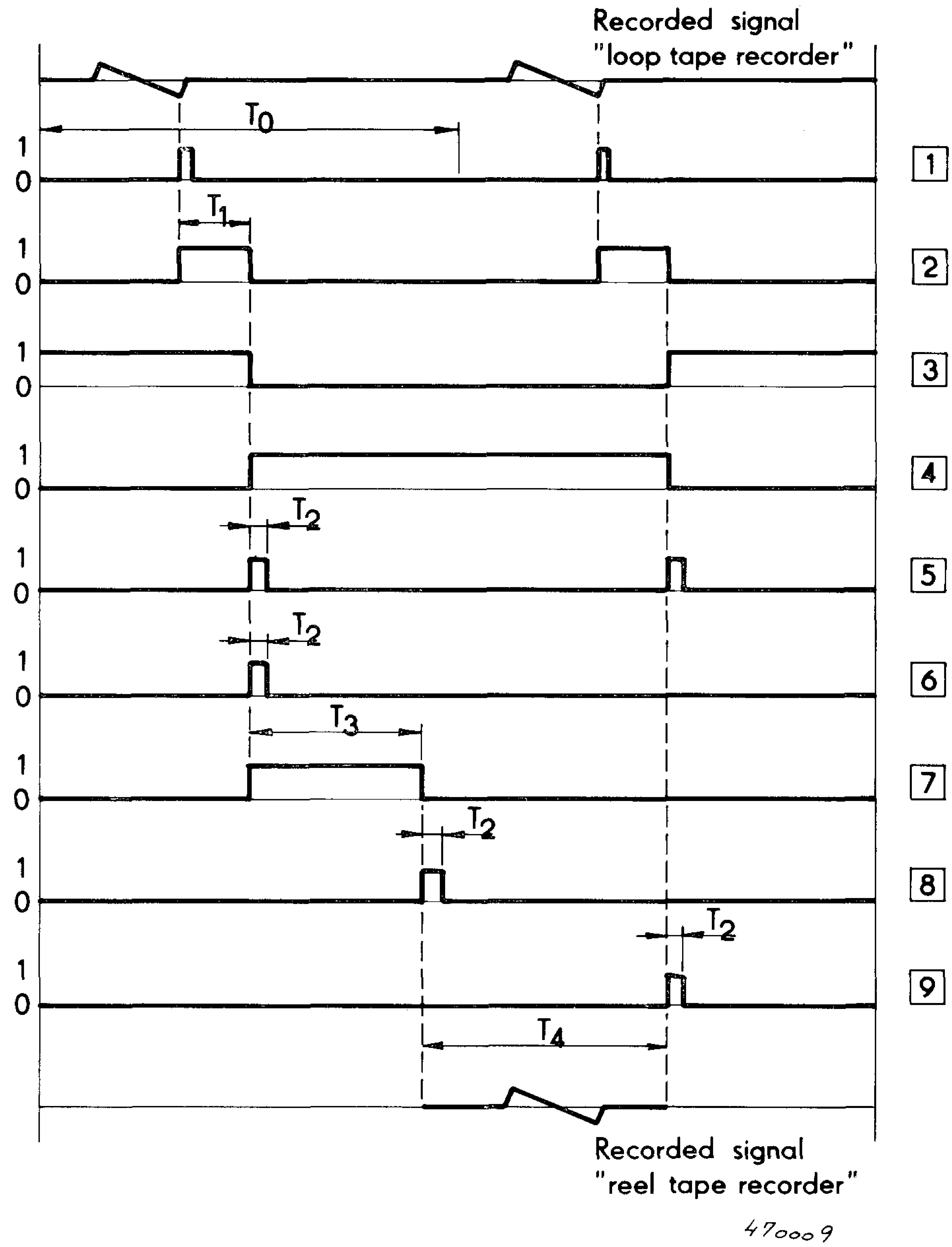


Fig. 3. Time-diagram of the control signals.

Activation of P changes a Tape Recorder to the playback mode from either stand-still or from the recording mode.
If P and R are activated simultaneously a Tape Recorder changes to the recording mode from either stand-still or from the playback mode.
The time delays shown in the time diagram are:

T_0 = Tape loop cycle time (3,5 sec – 20 sec/15 ips).
T_1 = Time interval to be recorded on the "reel tape recorder" after detection time (changeable).

T_2 = Activating time for relays in the tape transport systems of the tape recorders (about 20–50 ms).

T_3 = Time delay before start of ''reel tape recorder'' (changeable).

T_4 = ''Reel tape recorder'' running time.

T_4 = Starting time of the tape recorder + time before occurrance of signal + signal time duration + time after occurrance of signal to be recorded. (min. T_4 about 3,5 sec, because the starting time is about 3 sec).

Remarks

The control system is applied to standard tape-recorders using only their normal remote control connections (manual operation can be performed by turning a switch). No limitations have been made to the two measuring channels or the voice channel of the tape recorders. Various tape loop cycle times can be applied as time constants are easily changed. The limits at 15 ips are approximately 3,5 sec. (tape recorder starting time) and 20 seconds (max. tape loop length).

The control system seems to be very promising for various applications. The system described above could be modified to take samples of a signal at fixed intervals. In such a case only one tape recorder would be needed. In both cases further improvement could be the recording of a digital time indication from an electronic clock.

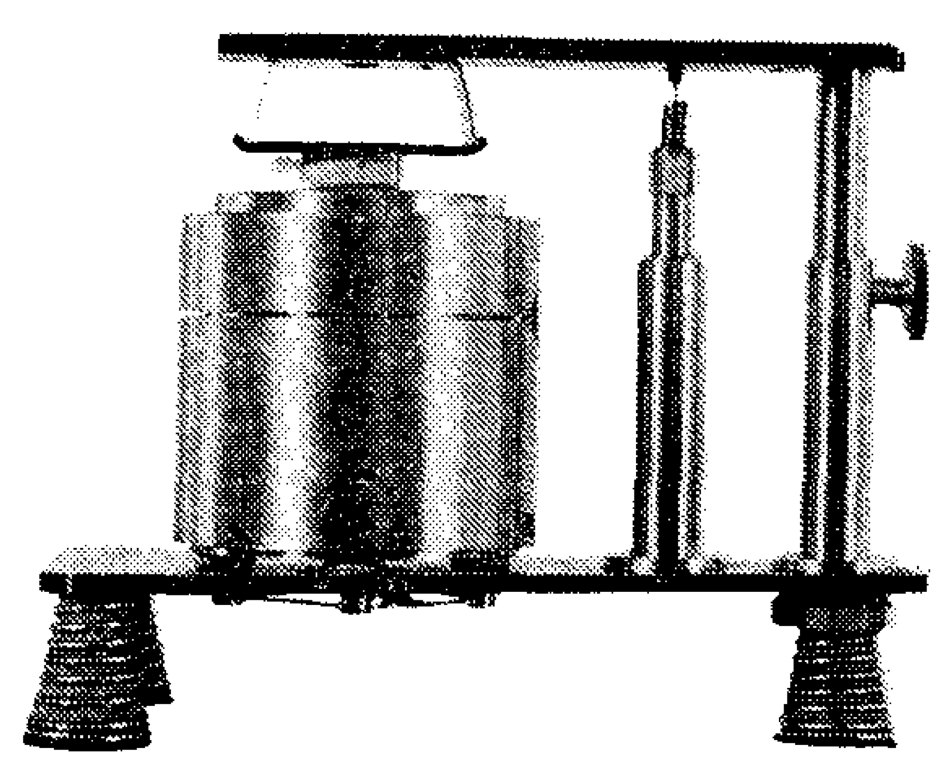
The described control system must be considered as a prototype only. In final systems the primary functions must be supplemented by various safety functions and alarm functions to operate if break down or malfunction should occur.

News from the Factory

Type 4930 Artificial Mastoid

Type 4930 Artificial Mastoid is made for objective calibration of bone conduction hearing aids and bone conductors. It conforms with BS 4009-1966. The mechanical impedance of the human mastoid can be simulated over the frequency range from 50 Hz to 8 kHz. Static force of application of the vibrator can be adjusted from 2 N to 8 N directly measured on a calibrated force gauge. Inertia mass is 3.5 kg and the built-in accelerometer has sensitivity 0.7 mV per m/sec^2 (-63 dB re 1 volt per m/sec^2), capacity is 3000 pF. Output can be connected to the Voltmeter Type 2409, one of the Measuring Amplifiers Type 2603, 2606 or 2607, or to the Precision Sound Level Meters Type 2203 or 2204.

The Artificial Mastoid is also supplied as Type 3505 which consists of Type 4930, The Mini Shaker Type 4810 and the Impedance Head Type 8000 for calibration of the Artificial Mastoid and for measurement of the mechanical impedance of mastoids.



Deviation Bridge Type 1521

Type 1521 is a versatile solid state instrument which makes use of three individual measuring frequencies: 100 Hz, 1 kHz and 10 kHz.

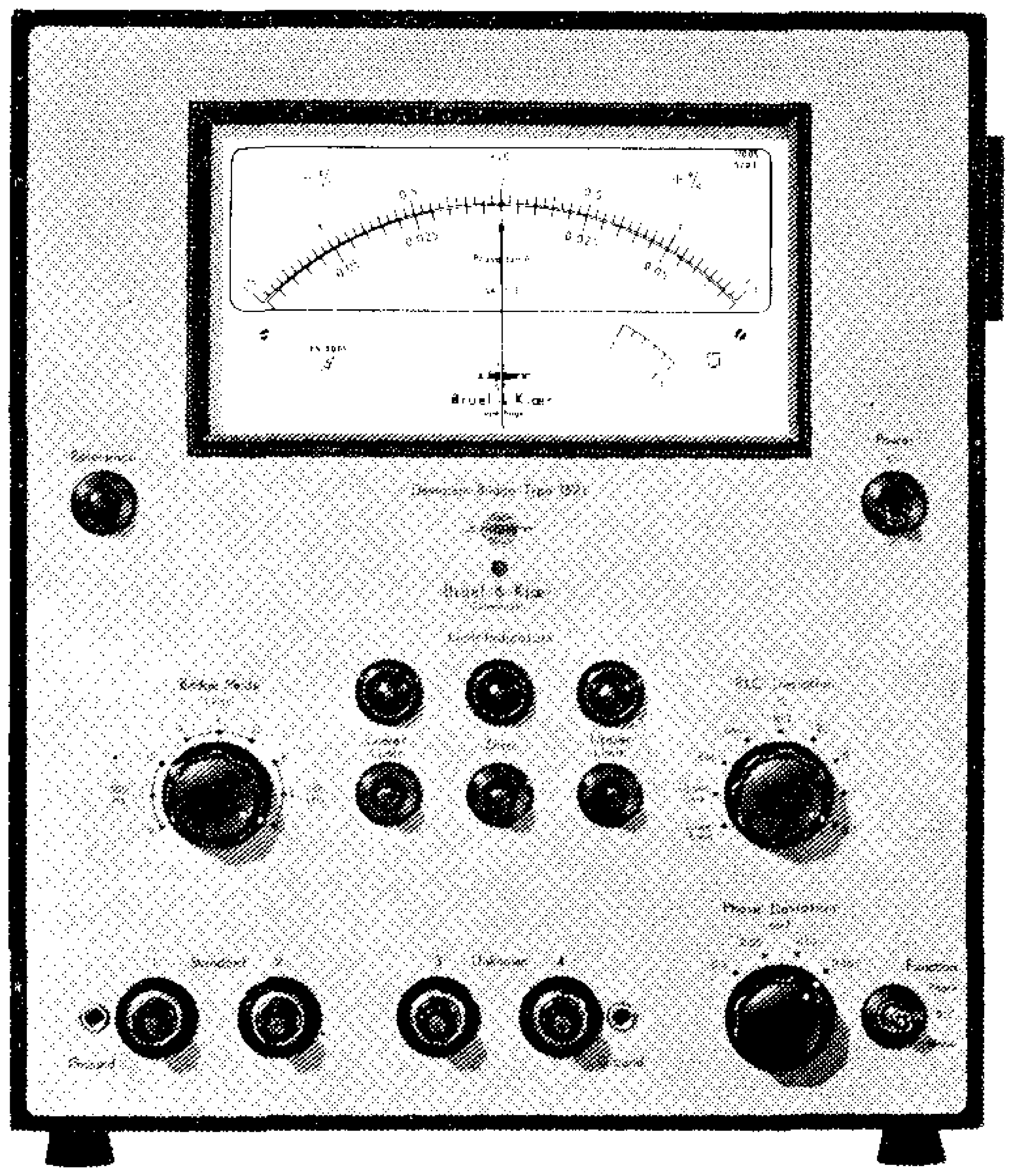
Thus it combines in one instrument the functions of the Type 1503, 1504 and 1505 Deviation Bridge.

The new Deviation Bridge yields a number of improvements of which a few can be mentioned:

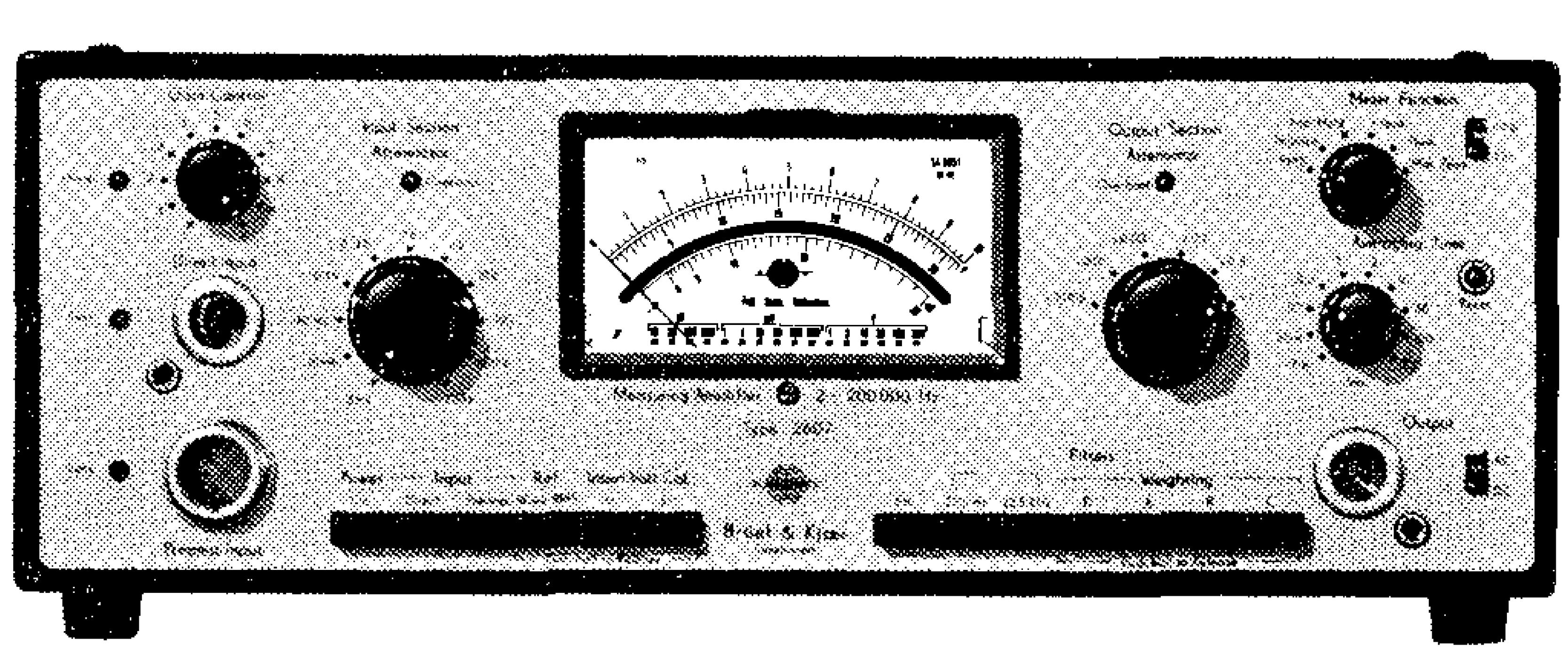
Measuring ranges are extended to 0.3 ‰ impedance and to $\tan \delta = 0{,}003$ for full scale deflection.

Impedance and phase angle ranges are selected by separate stepped attenuators thus maintaining fixed calibration for each measuring range.

Similar to the 100 kHz Deviation Bridge Type 1519, Type 1521 has built in tolerance indicator lamps which can be supplemented by external lamps as well as an analog DC output suitable for servo-directed sorting arrangements.



Measuring Amplifier Type 2607
The 2607 is an extremely versatile instrument which is intended primarily for narrow band analysis at low frequencies in conjunction with the Type 1614 Band Pass Filter Set and with the Type 2020 Heterodyne Slave Filter for impulse analysis.
It combines the function of the previously released Type 2606 Measuring Amplifier with a number of additional features which are described below.
The meter circuits and the recorder DC output are provided with variable time constants fro 0.1–300 sec. in discrete steps including "fast and slow" according to IEC 179 for the meter circuit. Dynamic range for the meter output is 64 dB.
A built in linear to logarithmic converter provides a DC voltage proportional to the measured dB value on the DC recorder output.
In this way a dynamic range of 64 dB is covered by the DC output, which facilitates direct dB recording over a large range by digital instruments or by linear x-y recorders. In this mode the meter range is linear dB reading over 50 dB.



Positive, negative or max. peak measurements can be carried out over d dynamic range of 50 dB with a max. charging time constant of 20 μsec – a feature very valuable for impulse analysis.

Motion Analyzer Type 4911

This instrument is the successor of the Stroboscope Type 4910. The many features of these instruments make them very usefull for many types of motion analysis, besides plain stroboscopy – that is the reason to name the Type 4911 a Motion Analyzer.

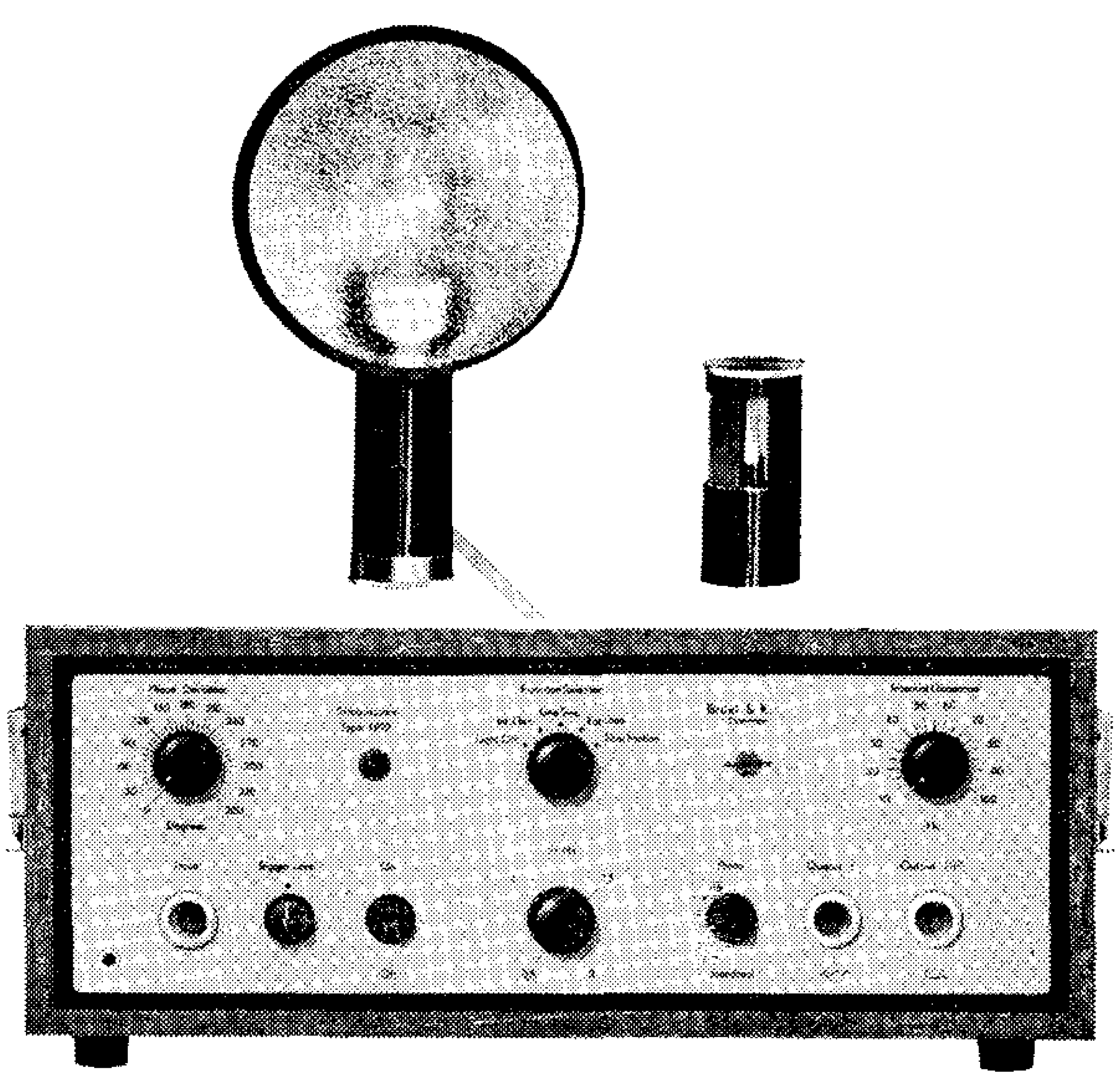
Its most important improvements compared to the Type 4910 are:

The automatic "Slow Motion" feature is now obtained by subtracting in stead of adding an internal frequency Δf to the signal frequency. In this way the apparent motion of the observed systems will be in the actual direction of motion. This gives a more correct impression of possible accelerations and decelerations to the observer.

Another feature now added is line syncronisation minus Δf. This mode is very convenient for slip measurements on asynchronous motors.

The automatic "Slow Motion" frequency range is extended to $\Delta f = 0.3 - 5.7$ Hz.

Synchronisation is now obtained by a direct connection from the camera shutter contact to the stroboscope.



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