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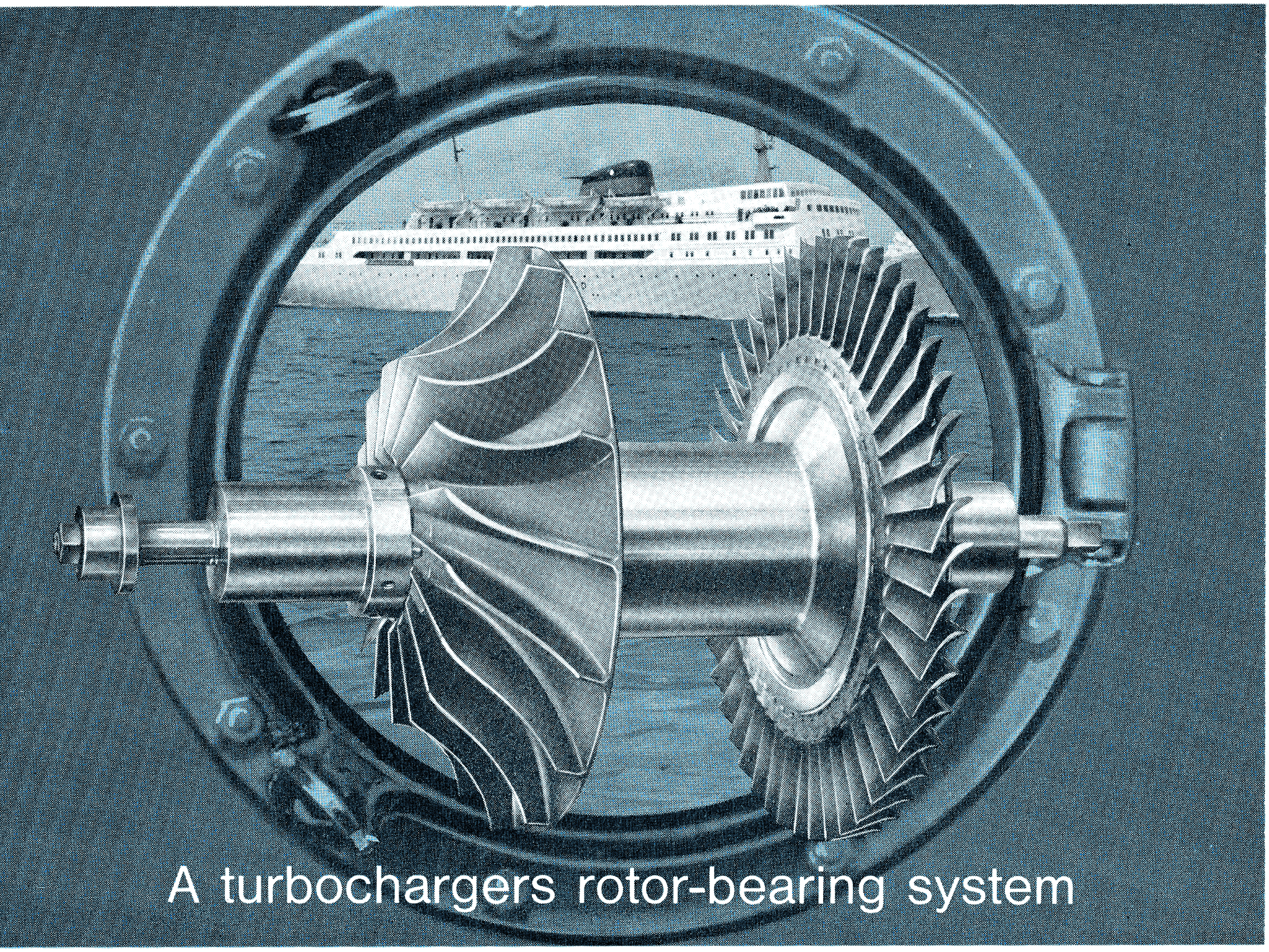

A turbochargers rotor-bearing system

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(Continued on cover page 3)

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Contents

Laboratory Tests of the Dynamic Performance of a Turbocharger Rotor-Bearing-System

by

Gerhard Westphal)*

ABSTRACT

Exhaust turbochargers for marine-diesel-engines are in a steady increasing degree equipped with elastic bearing mountings, where the dynamic performances are matched to modern high speed design. Properties in connection with rotor centering and damping of vibrations are of high interest — especially at critical speeds.

This paper describes research equipment and measuring procedures used for the evaluation of laws of the rotor bearing dynamics. Some parameters depending on the damping performance are recorded as a function of rotational speed.

SOMMAIRE

Les turbochargeurs d'échappement des moteurs diesel de marine sont de plus en plus souvent équipés de montages de portée élastiques lorsqu'on adapte les performances dynamiques aux vitesses élevées des réalisations modernes. Les propriétés relatives au centrage du rotor et à l'amortissement des vibrations sont d'un très grand intérêt, surtout aux vitesses critiques.

Cet article décrit l'équipement de recherche et les procédures de mesure utilisés pour déterminer les lois relatives au comportement dynamique du système portée/rotor. Certains paramètres dépendant de l'amortissement ont été enregistrés en fonction de la vitesse de rotation.

ZUSAMMENFASSUNG

Abgasturbolader für Schiffsdieselmotore werden zunehmend mit elastischen Lagereinbauten ausgerüstet, deren dynamischen Eigenschaften den stetig steigenden Betriebsdrehzahlen angepasst sind. Dabei sind Rotorzentrierung und Schwingungsdämpfung in kritischen Drehzahlbereichen von besonderem Interesse.

Der Aufsatz beschreibt Versuchseinrichtung und Meßverfahren zur Erforschung von Gesetzmässigkeiten in der Rotor-Lager-Dynamik. Verschiedene von der Lagerdämpfung abhängigen Meßwerte werden als Funktion der Drehzahl dargestellt.

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Introduction

Supercharged marine diesel engines need exhaust turbochargers of special construction, the development of which aims at increasing power density. Thereby the rotary speed goes up, thermal loads increase and demands to material properties are made more stringent. In addition the demand for minimum maintenance and highly reliable operation becomes ever more important.

In the evaluation of the technical dependability of an exhaust-turbocharger with high running speeds the dynamic relations of the rotor-bearing system plays an important role. The bearings for the rotor may be either roller or plain bearings. Where long life is required the plain bearings are the immediate answer, however, in this case one must have an absolutely dependable auxiliary lubrication system. For the normal working conditions in the machine rooms of Diesel-motor ships, however, such a system with cooled and filtered lubrication with pressurized oil would demand a considerable technical effort.

The possibilities for service monitoring and maintenance have become very much reduced on account of the generally increasing lack of trained technical personnel. Therefore, in most cases, exhaust turbochargers with roller bearings and self-lubrication are installed in diesel motor ships today. Such roller bearings, however, can only be optimally utilized in those cases where most of the essential working conditions on board a ship can be taken into consideration.

This paper describes rotor-bearing systems with roller bearings exclusively and describes parts of a research programme at HSM-Turbo, a department of the Elsinore Shipbuilding and Engineering Company Ltd.

The rotor-bearing system

The bearings for a modern exhaust-turbocharger must cope with very high rotational speeds as well as large axial loads. For values of d x n $> 1{,}2 \times 10^6$ an acceptable service life is only attainable when the loads can be distributed evenly on all rollers of a bearing. That places a first demand on the construction that the center-line of the rotor and that of the inner race of the bearing must coincide with the center-line of the outer race.

In operation the housing of exhaust-turbochargers deforms under the influence of temperature and inside pressure. Therefore the bearing seats cannot be flush even if they were manufactured with absolute precision. The demands for even load on all rollers of a bearing can therefore only be met when the bearing is installed so that it can move and thereby be able to centre itself on the rotor.

Relative to plain bearings the roller bearings do not particularly have good damping properties. Furthermore, at very high rotational speeds non-synchronous vibration may appear which can disturb the smooth running of the machine. Such disturbances may be reduced to a large degree by installation of precision bearings which are preloaded axially. The preload can, however, not be increased at will, as it adds to the bearing loads and reduces the mobility needed for self-centering.

Other disturbances are introduced from the environment. The exhaust-turbochargers being built together with the diesel motors are subjected to violent vibrations during operation. Elaborate internal investigations on board different ships have shown how the housings of the exhaust-turbochargers connected to the exhaust pipes are excited to certain vibrations which are detrimental to the bearings. Simultaneously the rotor is loaded in the axial direction with shocks generated by the pulsating outlet of the exhaust gas.

These excitations which are unfavourable for the bearings can only be isolated by effective external damping devices which, however, demand mobility.

The rotor of a fast running turbo machine must be well centered in the housing with due consideration to clearance losses and shaft seals.

Rotor displacement must be kept small, which might be difficult when there is unbalance and at operation in the vicinity of a critical bending frequency. Under all circumstances, however, the rotor should always tend to return to its central position.

A satisfactory solution to all these demands can only be obtained in the construction when the bearing housing acts both as a spring and a damper and stiffness and damping are tuned with respect to the mentioned influences on operation.

Under these conditions the bearing system has a dominating influence on the dynamics of the rotor. Operation below critical speed is not possible any more. The critical bending frequencies and the corresponding rotor displacements can be calculated from the assumptions of certain stiffness and damping values. However, the parameters of the elastic bearing system are mostly dependent on operation conditions and they must be found by experiments.

The experimental set-up

For the experiments to be as close to reality as possible they should be carried out on a quite normal exhaust-turbocharger of medium size. The experimental technique should not influence the dynamic relations of the rotor, the bearings or the housing. The rotor should be driven by its turbine and reach rotational speeds which are well above its second critical bending speed.

Between the turbine disc and the compressor impeller the rotor displacement in the X and Y axes should be measured with a resolution of at least $\pm 2\,\mu m$. The measurements should be carried out by non-contact transducers and not influenced by the rotational speed. The position of measurement should be chosen at a point where both the first and second critical speed would provide a distinct displacement. In addition measurements should be carried out to determine the reaction at the bearings.

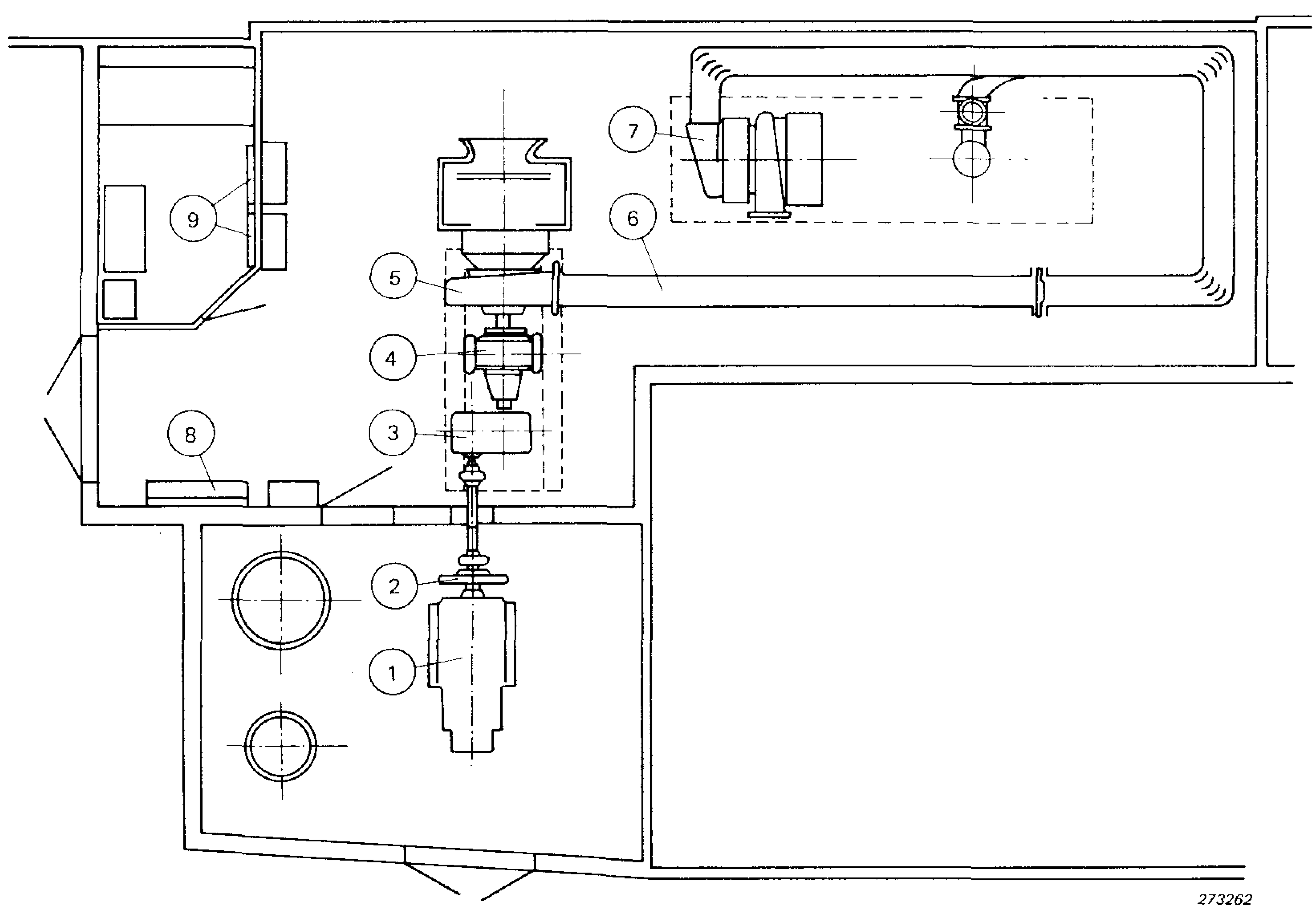

Fig. 1. Plan of the test installation used for investigation of the exhaust turbocharger

All the experiments were carried out on an HSM exhaust turbocharger type TH 22K. In order not to endanger the transducers at the rotor by hot gases or to obtain false measurement values on account of temperature the turbine was driven by pressurized air at a temperature under

100°C. The pressurized air was taken from a compressor test arrangement normally used for flow measurements whereby the speed of the turbocharger could be well regulated.

Fig.1 shows a plan of the test arrangement. The installation is driven by the motor (1) which is a 12 cylinder Rolls Royce Merlin 724 aircraft motor with a power output of 1500 BHP. This motor has proven itself to be very useful for experimental work on account of its reliability and its ability for rapid speed changes. To the flywheel (2) a pneumatically operated clutch is connected. The power is led to a first stage gearing (3) in which the transmission ratio can be changed by change of gear wheels. From (3) the power goes to a planet gear (4) which until now has been driven to a maximum speed of 41000 rpm. Different experimental compressors may be installed at (5). At full power operation pressurized air (generating more than 1000 HP) may be taken from the test line (6). This pressurized air is used to drive the experimental turbocharger (7).

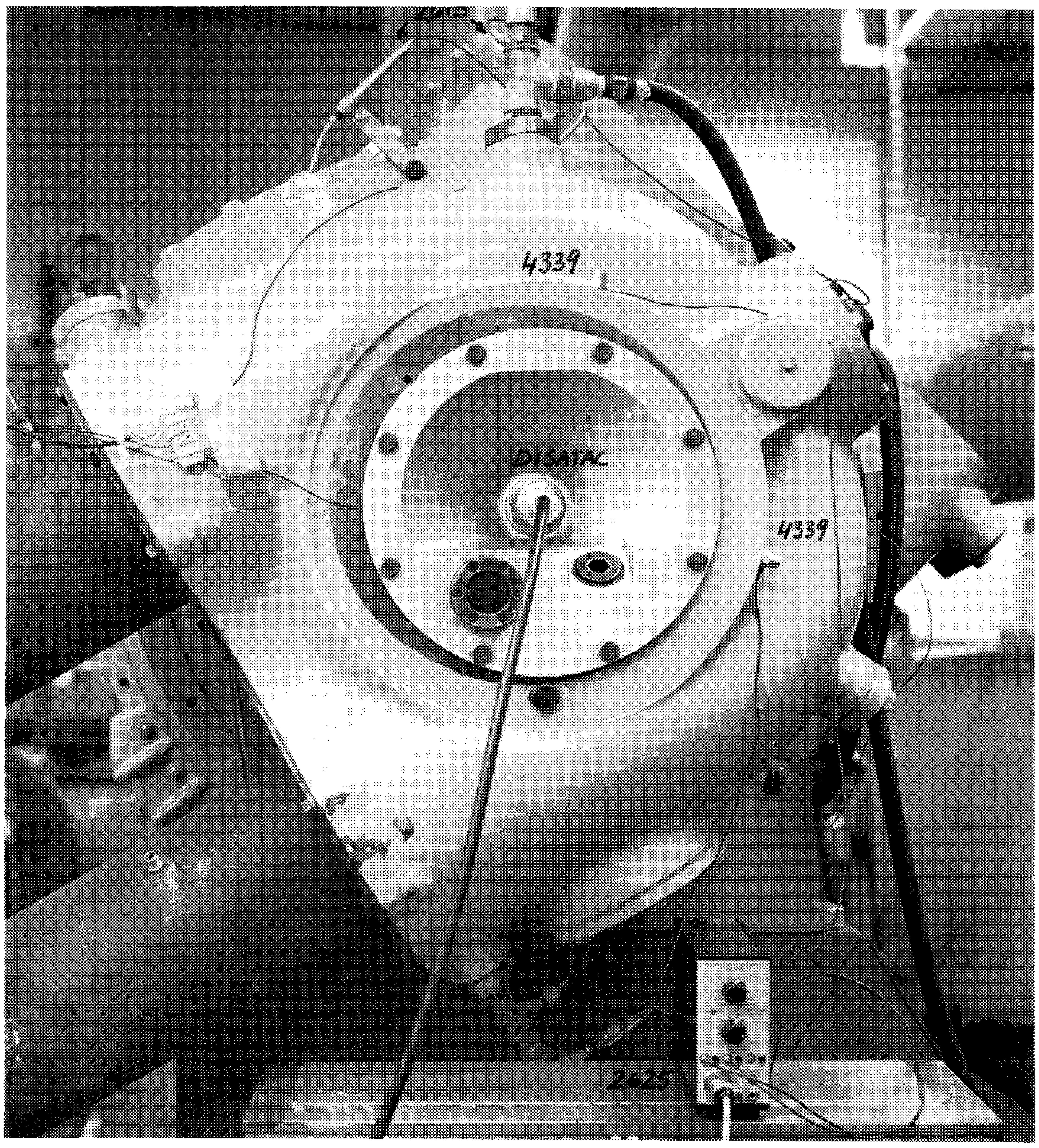

Fig.2. End view of the turbine side of the turbocharger

The motor (1) is electrically regulated from the control desk (8). Power changes can be fully automatically regulated as a function of time by means of simple relay-circuits, whereby reproducible test programmes may be carried out.

All measured values are transformed to electrical signals and are indicated and recorded by the instruments and recorders shown at (9). Fig.2 gives a view of the turbine-side of the experimental turbocharger and Fig.3 shows part of the measurement room.

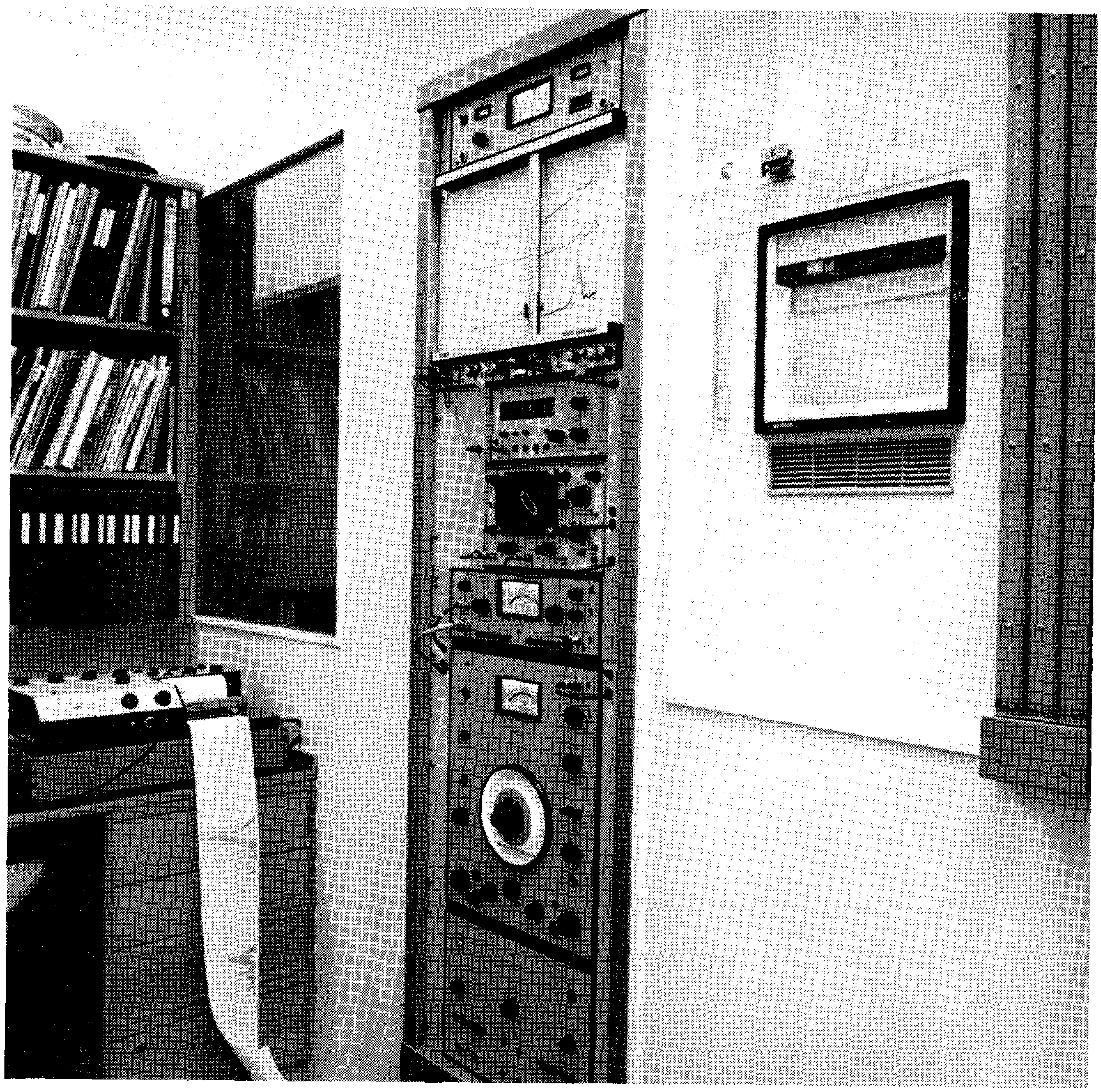

Fig.3. The instrument panels in the measuring room

The measurement arrangement

The motion of the rotor was measured in the X and Y direction each by a capacitive displacement transducer B & K Type MM 0004. These transducers are found to be very rugged and have been experienced to give good results.

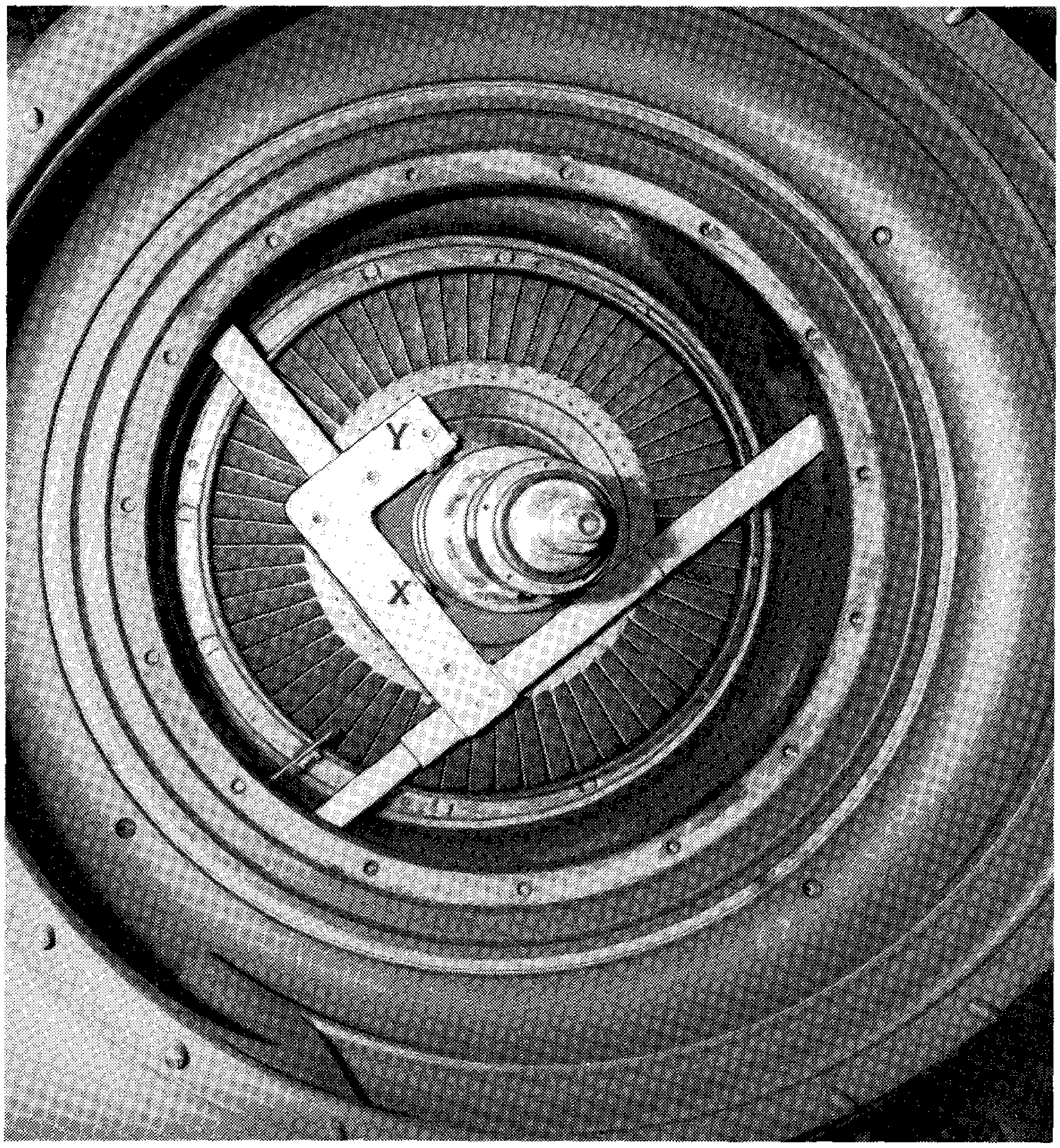

Fig. 4. The supporting rig with the displacement transducers

The displacement transducers were mounted in a stiff supporting rig and the measurement cables were incapsulated and led through a supporting tube to the outside. Thereby, disturbances of the measurement circuits due to the high air velocity at the turbine outlet were fully eliminated. The rig was fixed by means of a 3-tube system in the turbine outlet housing. To damp the vibrations of this system the tubes were made of a sandwich construction with plastic in-between the outer and inner tubes. The tubes were tightened sufficiently in the housing so that the motion of the displacement transducers became negligible as the resonance frequency of the supporting system was over 2000 Hz and had very small displacements. Fig.4 shows a photograph of the supporting rig. The X and Y axes were oriented according to the principal axes of symmetry of the housing. The experiments revealed that the careful installation of the displacement tranducers was of utmost importance, as with unfavourable bearing damping and operation at the second critical rotational speed considerable vibration was transferred to the housing. The dynamic calibration of the displacement transducer is described under a separate section.

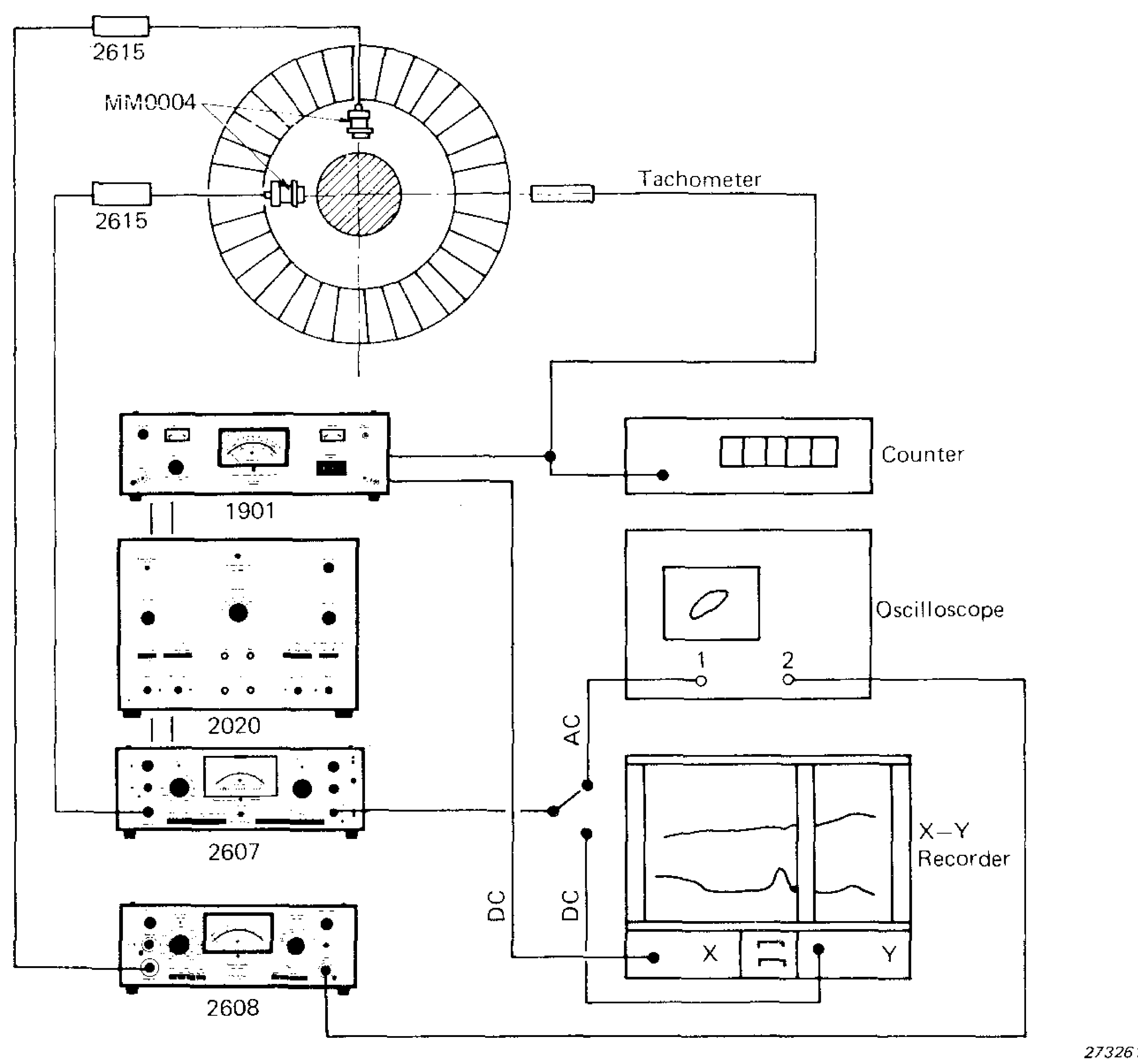

Fig.5. Measurement arrangement for displacement monitoring and recording

Fig.5 shows the measurement set-up for measurement of the rotor motion. The signal from the two transducers Type MM 0004 were led via B & K Preamplifiers Type 2615 to the Measuring Amplifiers Type 2607. The X and Y signals from the Measuring Amplifiers were used to deflect the beam of an oscilloscope in the X and Y directions respectively to obtain directly the path followed by the rotor center line.

The rotational speed of the rotor was measured at the rotor end with a magnetic tachometer (Disatac), and an electronic counter. The tachometer signal was also used to trigger a B & K Tracking Frequency Multiplier Type 1901.

The rotor displacement "A" (of each channel in turn) was directly recorded on an X-Y recorder as a function of rotational speed or angular velocity. To obtain these DC analog signals proportional to displacement and frequency they were taken from the Measuring Amplifier Type 2607 and the Tracking Frequency Multiplier Type 1901 respectively and led to the X-Y recorder inputs.

In the displacement measurement circuit a Heterodyne Slave Filter Type 2020 could be inserted which was tuned with the speed of the rotor from the Tracking Frequency Multiplier.

The measurement arrangement for the recording of acceleration values on the bearings is shown in Fig.6. Accelerometers Type 4344 were used directly on the bearing assemblies of the exhaust turbocharger (see

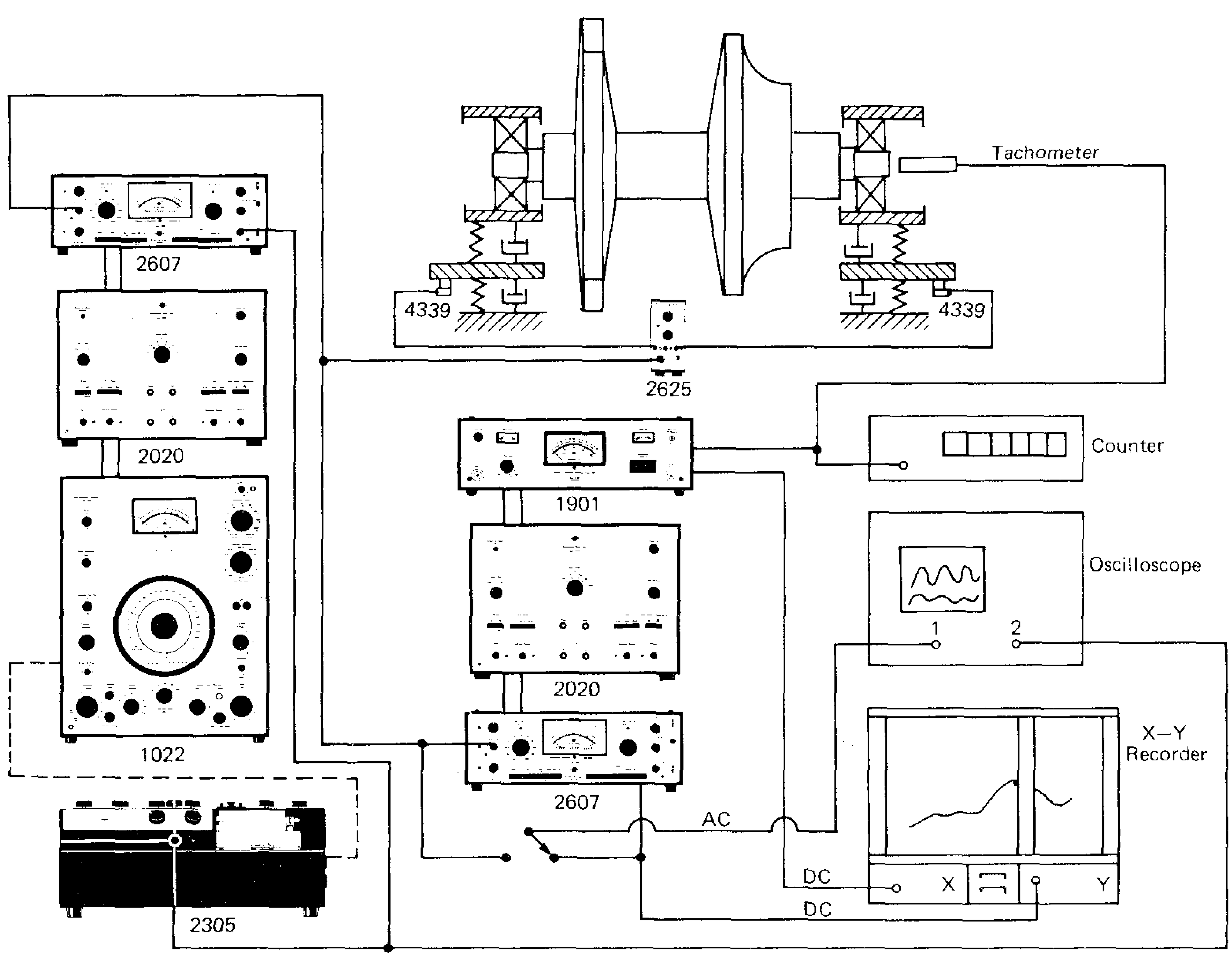

Fig.6. Measurement set-up for acceleration measurement and frequency analysis

Fig.7). Other Accelerometers Type 4339 were mounted on the outside of the turbine inlet housing and the compressor inlet housing respectively so that the ribs inside the housings provided the best possible transmission path between the bearing flanges and the measuring points on the housings. The signals from the accelerometers were led successively via a Preamplifier Type 2625 to a Measuring Amplifier Type 2607. The acceleration signal was narrow band filtered by means of the Heterodyne slave Filter Type 2020 tuned to the speed of the turbocharger by means of the Tracking Frequency Multiplier triggered by pulses from the tachometer. The acceleration signals were recorded as functions of the rotary speed or angular velocity ω on the X-Y recorder and the measured signal could be displayed on the oscilloscope and compared with the unfiltered signal.

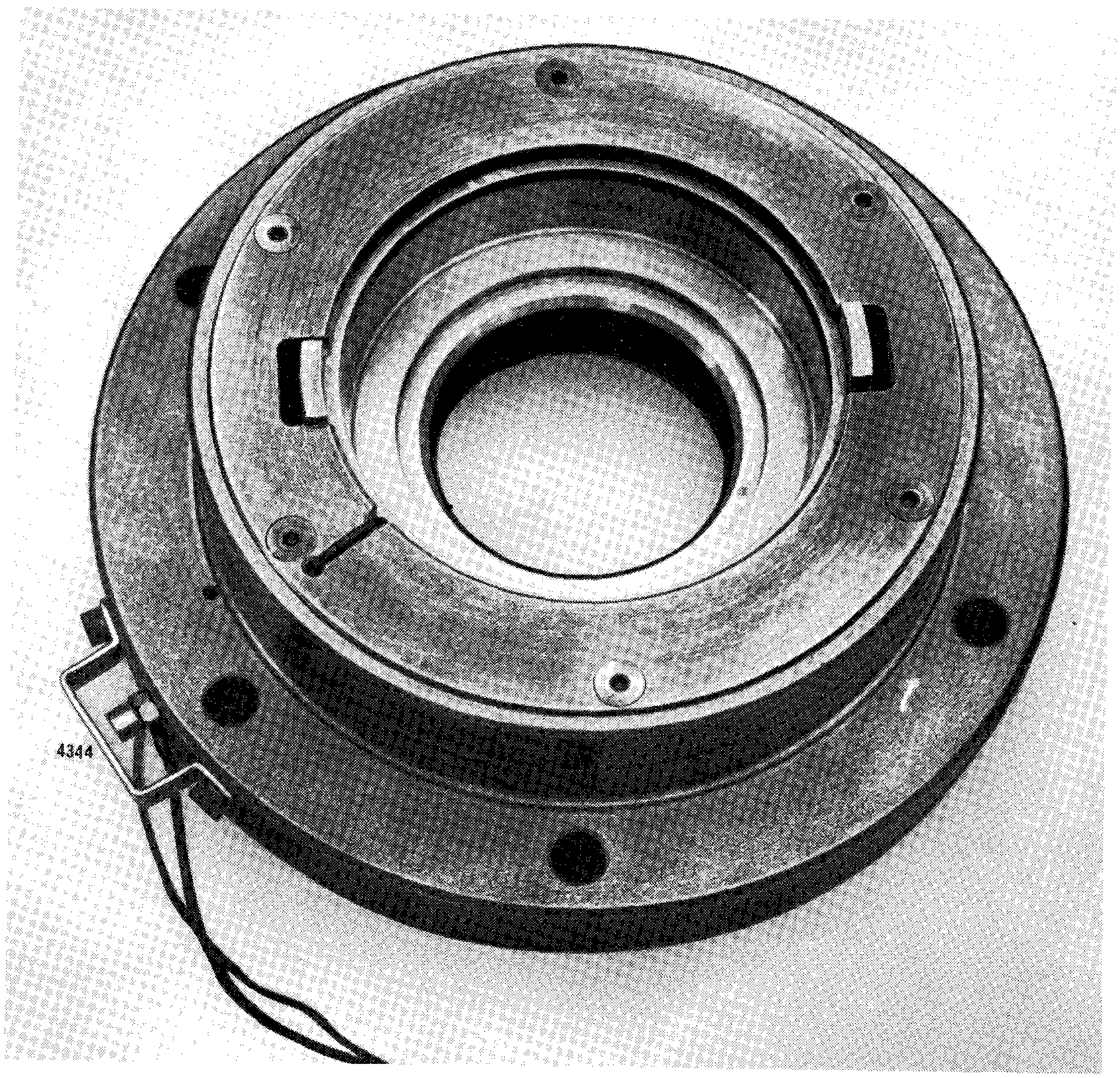

Fig.7. The mounting position of mini accelerometer on the bearing housing

At particularly interesting speeds frequency analysis was carried out on the acceleration signal or on the rotor displacement signal. Here the Heterodyne Slave Filter Type 2020 was tuned by means of a Beat Frequency Oscillator Type 1022, which was connected to a Level Recorder Type 2305 by means of a Bowden Cable (UB-0041) which synchronized the frequency sweep of the oscillator and the filter with the precalibrated recording paper of the Level Recorder.

Other parameters which were essential for a reproducible experimental technique were measured by separate measuring circuits and recorded on a 12 channel point-recorder.

Calibration
The Capacitive Transducers Type MM 0004 were dynamically calibrated before installation on the turbocharger. For this purpose five cylindrical steel discs of diameter 60 mm and length 25 mm were manufactured. In each of these discs holes of dimensions 10H6 were drilled on a precision drilling machine. The holes had the following eccentricities relative to the centres of the discs:

$e_1 = 0{,}05$; $e_2 = 0{,}30$; $e_3 = 0{,}60$; $e_4 = 1{,}00$; $e_5 = 1{,}30$ mm

During rotation around the centres of the holes the discs had an unbalance dependent on the eccentricities. The simple geometrical form, however, allowed a precise calculation of the unbalances which were eliminated by symmetrical holes drilled in both sides. The discs were mounted in turn on the shaft of an electric motor which could be regulated gradually from 0 to 18000 rpm, and which had precision plain bearings. The displacement transducers Type MM 0004 were mounted at a distance of $2{,}00 \pm 0{,}02$ mm from the cylindrical surface of the disc. The exact eccentricity was measured by means of a dial gauge with an accuracy of 0,01 mm. The signal from MM 0004 was led via a Preamplifier Type 2615 to the Measuring Amplifier Type 2607.

The measured value from MM 0004 was only dependent on the disc eccentricity and it was constant in the range from 1000 to 10000 rpm. Over 15000 rpm the measurement arrangement was disturbed by vibration caused mainly by unbalance in the motor. The calibration curve (Fig.8) is linear and goes through the origin.

After installation of the displacement transducers in the experimental turbocharger the eccentricity of the rotor was measured with a dial gauge.

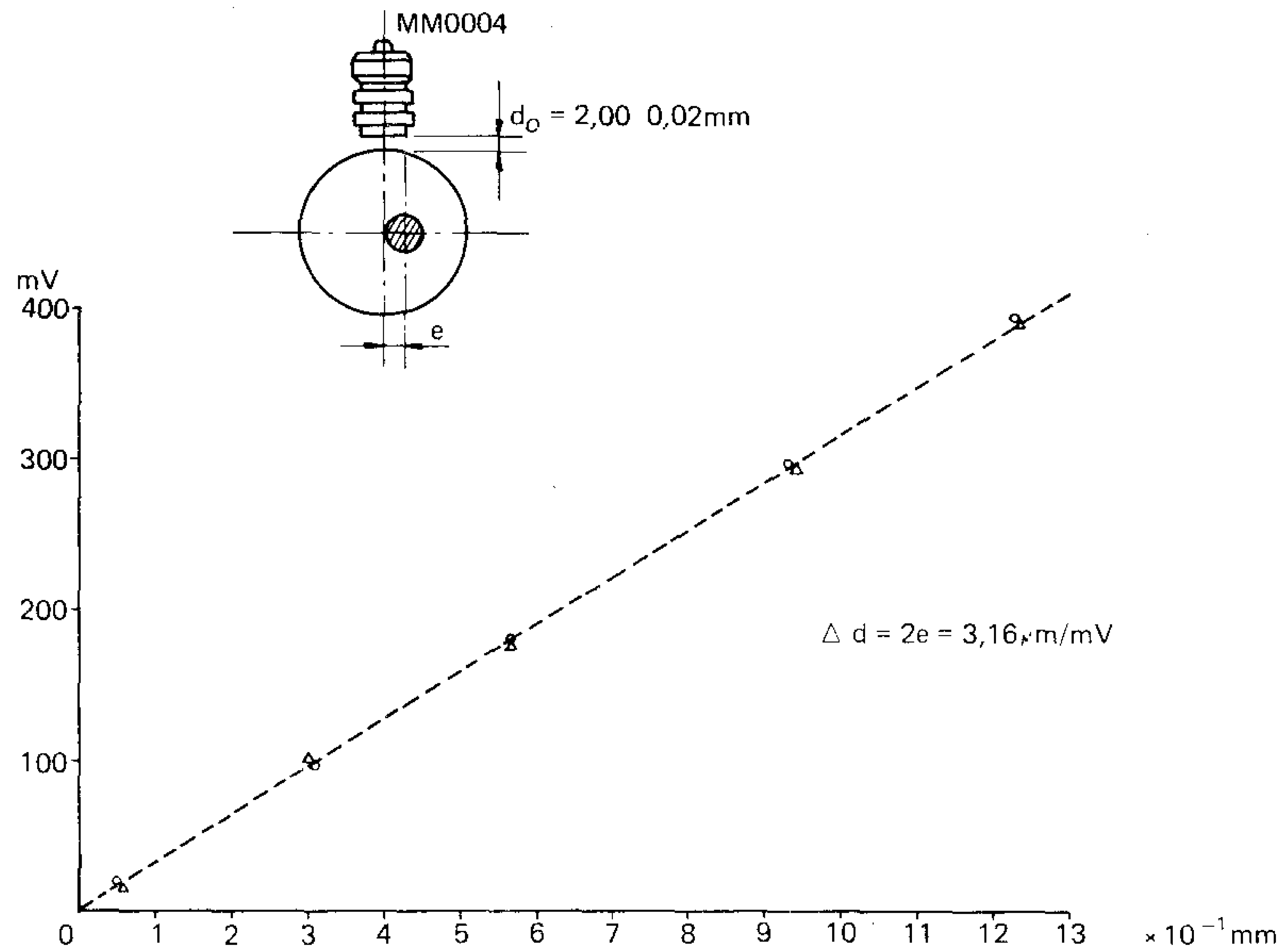

Fig.8. Calibration curve for the Capacitive Transducer Type MM 0004 in displacement measurements

The measured eccentricity $2e = A_0$ was used as reference level for the measurements and the measurement set-up was adjusted accordingly for angular velocities much below the first critical bending frequency i.e., $\omega/\omega_{k1} \ll 1$.

The measurement results

The dynamic properties of different rotor-bearing systems were investigated and, among others, the parameters of bearing construction which affected the damping were systematically changed. The measurements proved, in accordance with previous calculations, that two conditions would seem to appear. First, a still sufficient centering of the rotor might be accompanied by violent vibrations of the bearing housings and second, that a smooth run could be obtained even with inferior centering. Naturally, neither of these conditions could be accepted.

By measurement of the rotor displacement in two planes at right angles to each other the characteristics of the bearing system could be determined with respect to rotor centering.

To evaluate the smoothness of operation acceleration was measured at the bearings, as the measured values are then proportional to the bearing reactions.

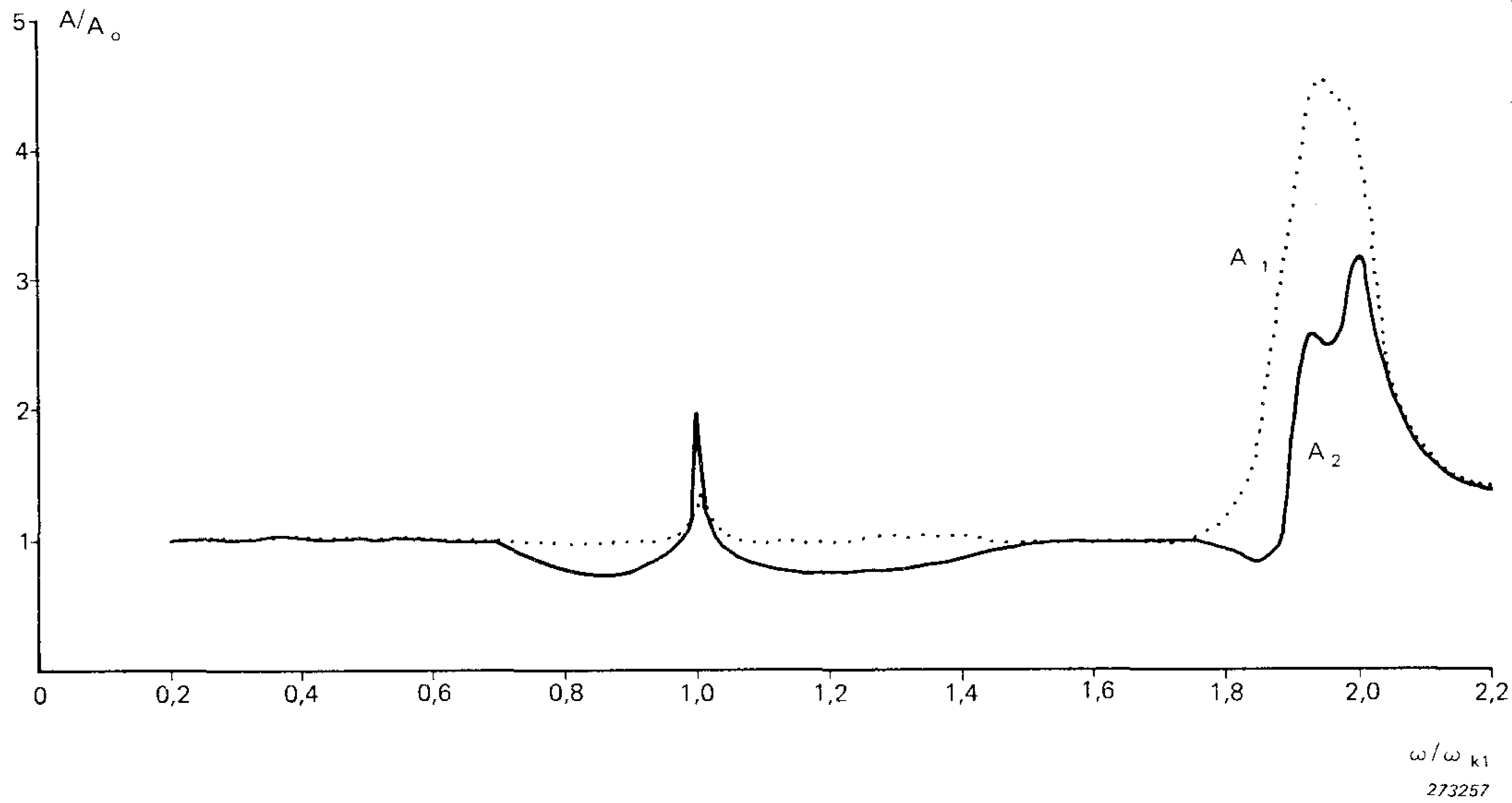

Fig.9. Rotor displacements in the Y axis for dampings D_1 and D_2

Fig.9 shows a recording on the X-Y recorder of the displacement A = f(ω). Here

A	=	rotor displacement in the Y axis
A_0	=	A for $\omega/\omega_{k1} \ll 1$
A_1	=	A for damping D_1
A_2	=	A for damping D_2
ω	=	angular velocity of the rotor
ω_{k1}	=	ω for the first critical frequency in bending
U_1	=	rotor unbalance for A_1 and D_1
U_2	=	rotor unbalance for A_2 and D_2

Fig.10 shows the acceleration b = f(ω) corresponding to Fig.9. Here

b	=	acceleration in the y axis measured with tuned filter Type 2020 and bandwidth 10 Hz
b_1	=	b for damping D_1
b_2	=	b for damping D_2

b_1 and b_2 were measured on the same bearing housing and $U_1 = U_2 = U_{12}$.

With the exception of D_1 and D_2 all other operational parameters concerning Figs.9 and 10 were constant.

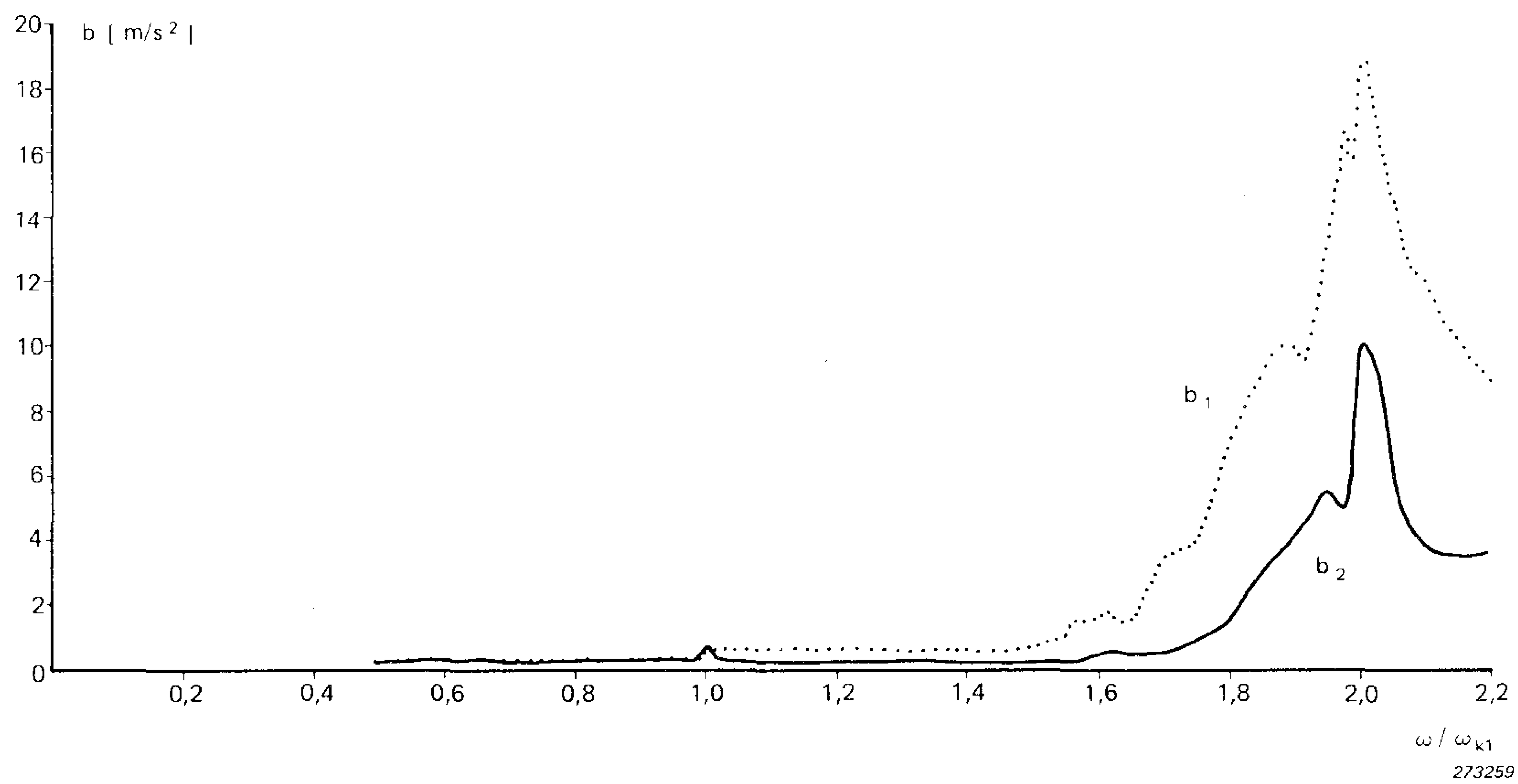

Fig.10. Accelerations on bearing housing for dampings D_1 and D_2

Fig.11 shows the rotor displacements, A_3 and A_4 for dampings D_3 and D_4. Here

$$D_1 > D_2; \quad D_3 > D_4; \quad D_3 \gg D_2$$

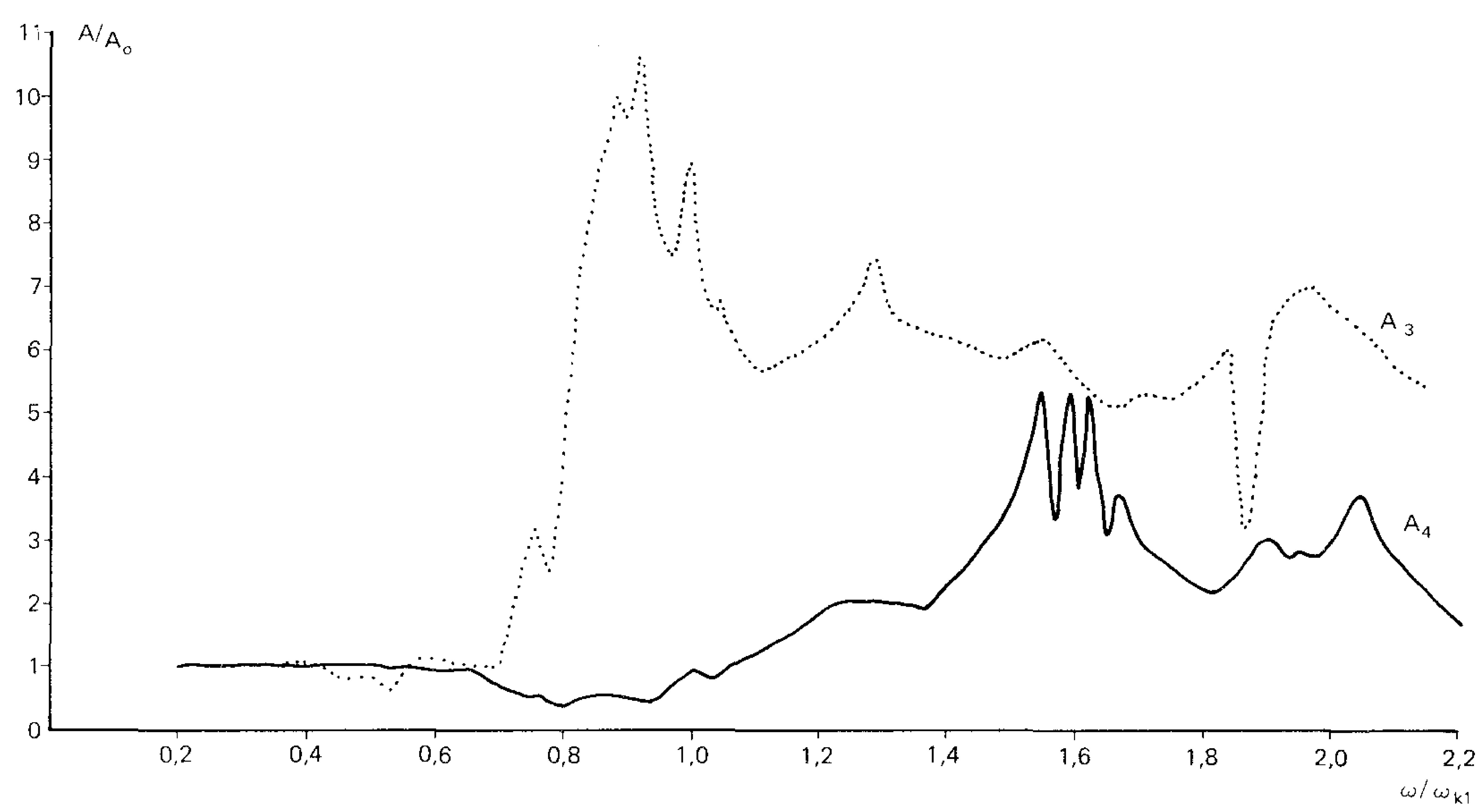

Fig.11. Rotor displacements in the Y axis for dampings D_3 and D_4, and large unbalance U_{34}

The measurements which are recorded in Fig.11 were carried out with a rotor unbalance $U_{34} = 10\ U_{12}$. This unbalance is so large that the service life of a normal plain bearing operating at $\omega/\omega_{k1} = 1{,}8$ would only be a few minutes.

Fig.12 shows a frequency analysis of the displacement A_3 for operation at a constant speed between the first and second critical frequency of bending. Here the rotor displacement signal was analyzed with a 10 Hz filter bandwidth. From the analysis in the X and Y directions the mean path of motion of the rotor can be constructed for the particular angular velocity. As it would be too lengthy to explain in detail about this method it should only be said that the frequency analysis provides a useful way to obtain a result when the path of motion is too disturbed to produce a useful display on an oscilloscope with X and Y deflection of the beam.

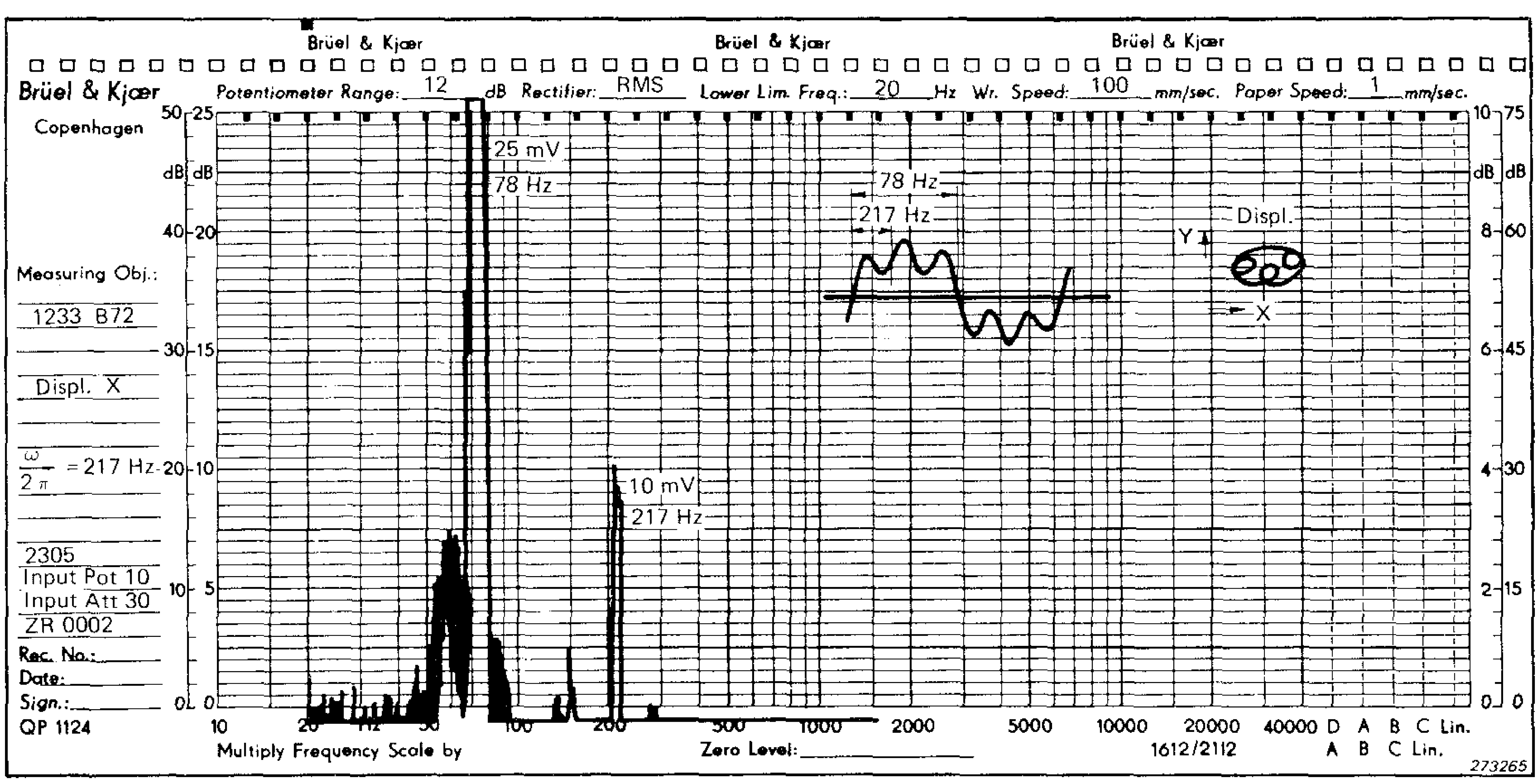

Fig.12. Frequency analysis of displacement A_3 at a constant speed between the first and second critical speed. The constructed path of motion of the rotor is given at the right of the figure

Conclusion

Although this paper can only point to very few measurements it seemed important in the introduction to explain in some detail the technical reasons which led to the measurements described and other quite elaborate experiments.

During the measurements app. 450 test runs were carried out of which more than 200 included operation with speeds $\omega/\omega_{k1} > 2$. On and off operational conditions were met which were quite serious and which came very near to the limit of total break-down of the experimental machine. Therefore, especially under critical situations the measurement arrangement used proved itself to be very reliable and accurate in function. The evaluation of measured data was facilitated considerably by using instrumentation which could directly record the functions $f(\omega)$.

In the experiments, 5 rotors of different geometry and mass were systematically investigated for 3 standard unbalances and about 30 different bearing configurations.

Hereby the design basis for the rotor geometry in conjunction with damping bearing systems was ascertained. The optimized bearing systems would guarantee the following properties:

a) Self-centering of the bearing
b) Centering of the rotor at all speeds
c) Optimal damping and smooth running in operational speed range
d) Simple technological construction
e) Small consumption of lubricant

The measurements have confirmed that by observance of certain criteria in the development, the dynamic properties of the rotor-bearing system are characterized by exact reproducible regularities.

Measurements of the Resonance Frequencies on a Turbocharger Rotor

by

Gerhard Westphal)*

ABSTRACT
A high-speed turbocharger rotor was supported to vibrate as a free-free beam. The rotor was mechanically excited for measuring the resonance frequencies of the 1., the 2. and the 3. mode. The paper describes the measuring set up and procedure, and results are given and discussed.

SOMMAIRE
Le rotor d'un turbochargeur à grande vitesse a été suspendu de façon à vibrer comme une poutre libre-libre. Le rotor a été excité mécaniquement pour mesurer les fréquences de résonance des 1er, 2e et 3e modes. L'article décrit le montage et la procédure de mesure, donne les résultats et les discute.

ZUSAMMENFASSUNG
Der Rotor eines hochtourigen Abgasturboladers wurde so unterstützt, daß dieser als frei-freier Balken schwingen konnte. Der Rotor wurde mechanisch erregt und mit Hinblick auf seine 1., 2. und 3. kritische Biegeschwingung durchgemessen. Der Aufsatz beschreibt Meß-aufbau und Meßverfahren, die Messergebnisse werden diskutiert.

Introduction
At very high rotational speeds the operation of rotors supported on ball bearings is often disturbed by violent vibrations of the rotor-bearing system. These vibrations can, however, be controlled when the inferior damping capacity of the ball bearings are compensated by specially tuned elastic bearing housings.

The dynamic characteristics of such rotor-bearing systems can be determined with good accuracy by means of computer calculations provided that the bearing parameters are defined and that good agreement exists

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between the rotor and its mathematical model. In order to obtain such agreement the rotor was excited as a free-free beam in bending and the first, second and third resonance frequencies were measured. From the measurements ratio numbers could be calculated by which the mathematical model was corrected. The corrections affect the dynamic modulus of elasticity of the rotor which was found to be 10 — 20% higher than the value determined by a static test.

The computation of such a rotor-bearing system is found in Ref.(1). It should be pointed out, that here the first resonance frequency of the rotor under free-free conditions is coincident with the third critical rotational speed in bending of the rotor for normal bearing configurations and bearing stiffness k = 0.

In the following, vibration measurements are carried out on the rotor of a modern high speed exhaust turbo-charger built at Helsingør Skibsværft og Maskinbyggeri (HSM).

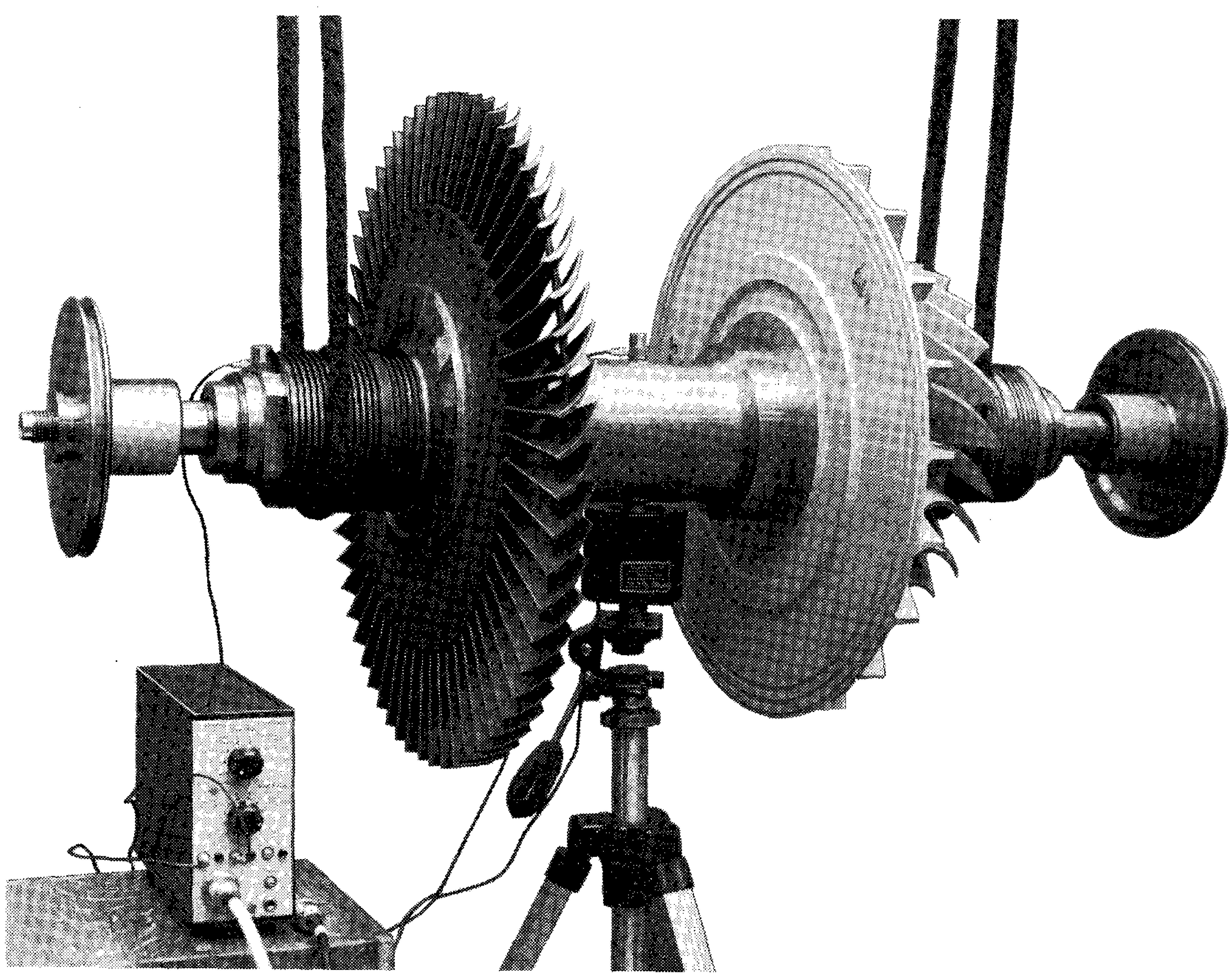

Fig.1. Test rig

Measurement arrangement

A completely assembled rotor of total length 950 mm (see Fig.1) has a weight of approximately 100 kg. On account of the varying moments of inertia, the distribution of mass and the external forces, the rotor was divided into several segments by measurement stations (see Fig.2). The rotor was supported by elastic ropes at stations 6 and 11 which coincided approximately with the nodes of the first resonance frequency. At station 9 a Brüel & Kjær Vibration Exciter Type 4810 was mounted with the axis of excitation vertical and in the plane of the rope supports.

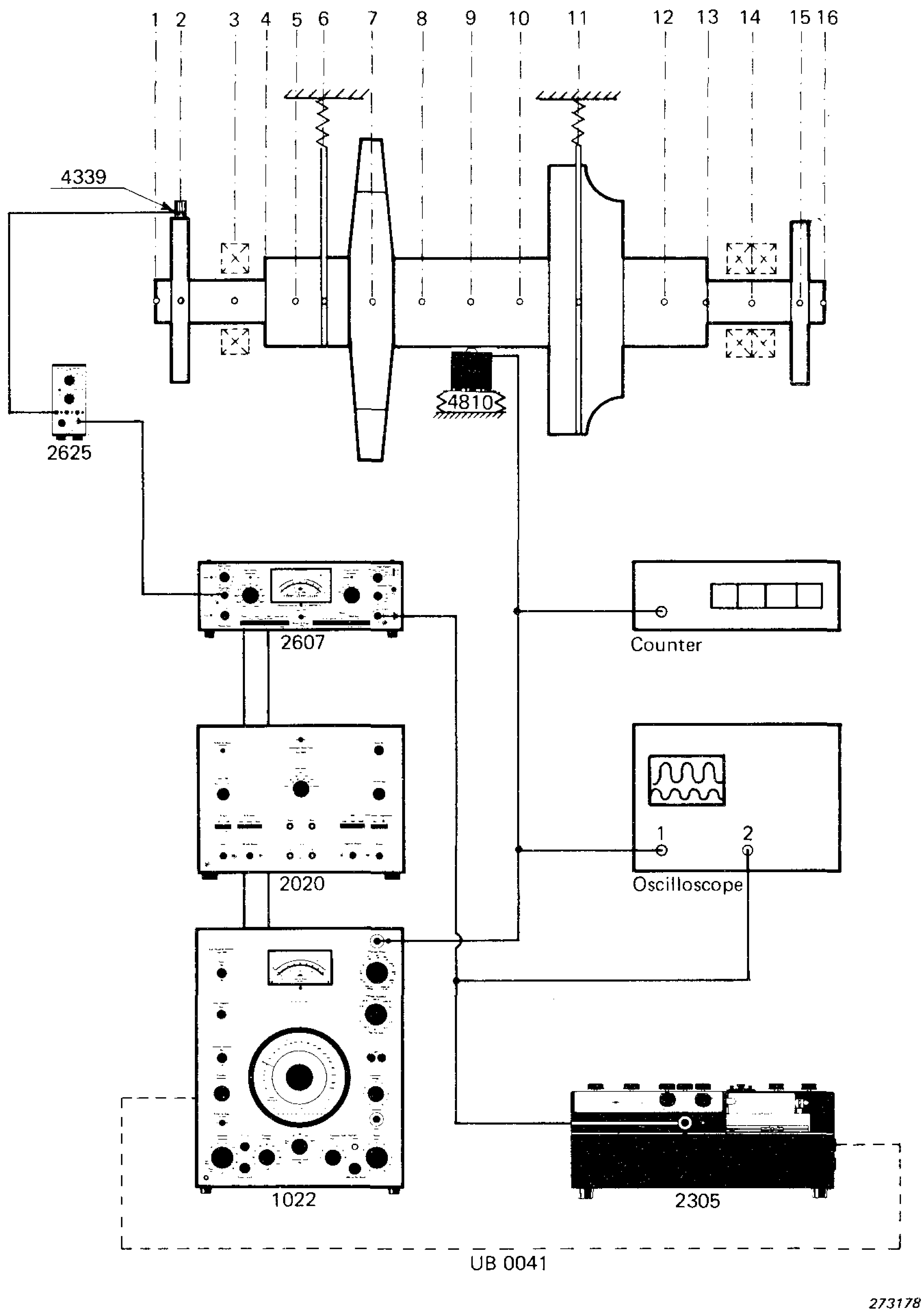

Fig.2. Measurement set-up

Accelerometers Type 4339 were mounted with wax at different stations on the other side of the rotor but in the same plane. The signals from the accelerometers were successively led via a Preamplifier Type 2625 to the Measuring Amplifier Type 2607.

A 10 Hz constant bandwidth narrow band analysis of the signals was carried out using the Heterodyne Slave Filter Type 2020. The centre frequency of the Heterodyne Slave Filter was synchronized to the Beat Frequency Oscillator signal frequency exciting the vibration exciter. The signal from the Beat Frequency Oscillator was also led to one of the channels of a double beam oscilloscope and to an electronic counter to measure exactly the excitation frequency digitally. From the output of the Measuring Amplifier Type 2607, the response signal was led to a Level Recorder Type 2305 as well as to the second channel of the oscilloscope. The Level Recorder was connected to the Beat Frequency oscillator by means of a Bowden Cable Type UB 0041 to synchronize paper speed with the frequency sweep speed.

Measurement Results

The vibration level on the accelerometer Type 4339 was found to be low and it was adjusted at measurement station to be approximately 0,005 g at 100 Hz. As the mechanical amplification factor Q was considerably larger than 10 at resonance, even significantly lower excitation levels would be sufficient for detection of the resonances. The recordings from measuring station 9 are given in Fig.3.

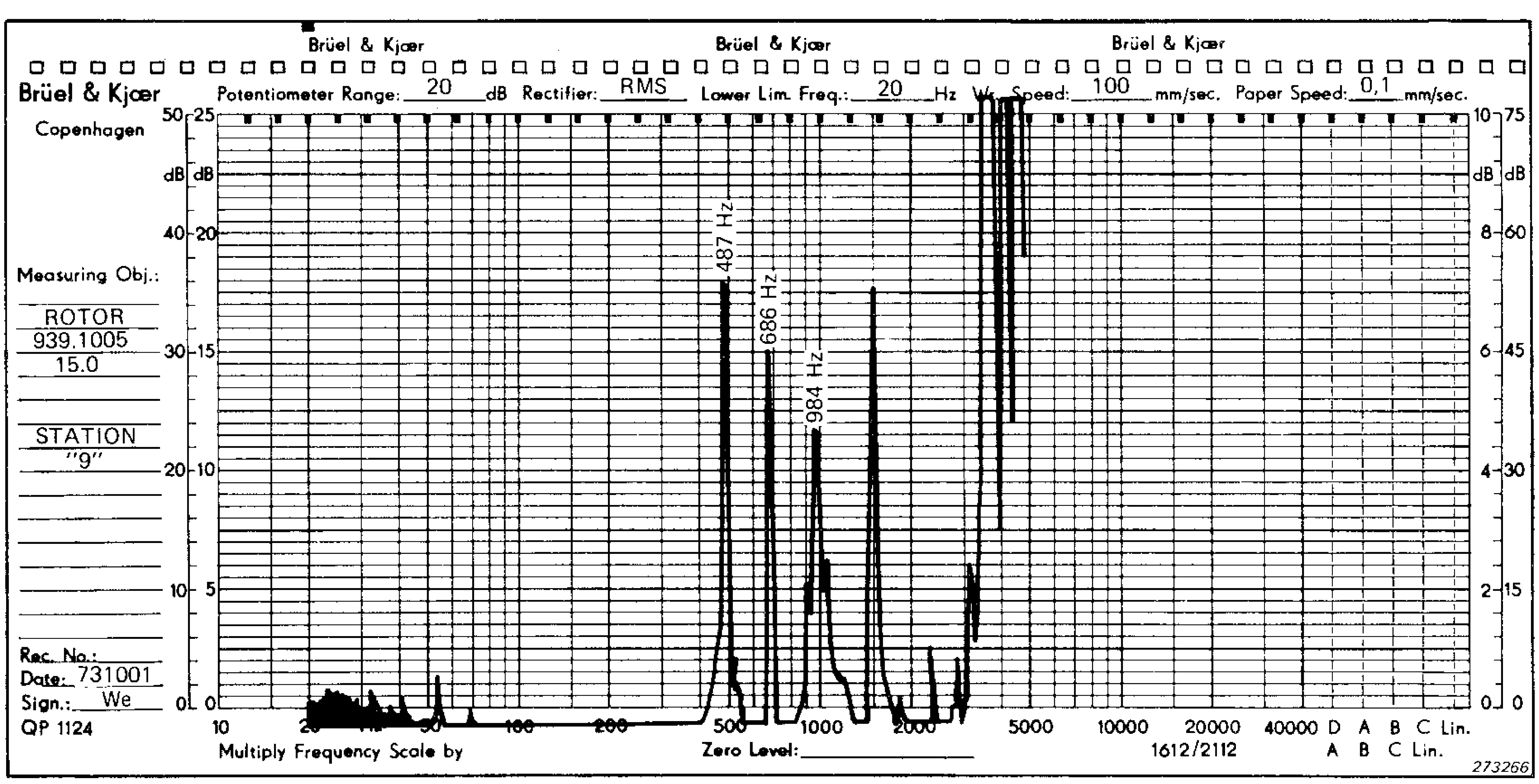

Fig.3. Recorded signal at station 9

The excitation frequency was kept constant at 487 Hz which generated the first mode of resonance and measurements were made along the rotor. Thereby the change in vibration amplitude as well as the positions of the nodes could be found. As the exciter signal was permanently displayed with fixed phase on one of the oscilloscope channels a 180° phase change of the response was easily observable when a node was passed.

By this method each vibration resonance mode could be recorded as a standing wave and the first second and third critical bending frequencies could be positively identified (see Table 1). The table lists measured accelerometer output voltages in microvolts as read from a Measuring Amplifier Type 2607 for measuring stations 1 — 16 at three frequencies f_1, f_2 and f_3. The 180° phase changes are indicated by the change of sign in the table.

Station	f_1 = 487 Hz	f_2 = 686 Hz	f_3 = 984 Hz
1	+ 840	+2500	+ 450
2	+ 510	+1100	− 190
3	+ 155	+ 305	− 360
4	+ 150	+ 48	− 300
5	+ 118	− 130	− 220
6	+ 58	− 250	− 130
7	− 20	− 330	− 10
8	− 98	− 320	+ 50
9	− 135	− 165	+ 182
10	− 142	− 4	+ 150
11	− 90	+ 230	− 30
12	+ 40	+ 450	− 260
13	+ 108	+ 500	− 380
14	+ 310	+ 350	− 390
15	+ 680	− 100	− 120
16	+1420	−1550	+ 420

Table 1. Measurement results at resonance frequencies of the 1., 2. and 3. mode

In the measurements described, resonances above 3000 Hz were found in which amplitude measurements along the rotor disclosed other modes than expected. Further investigations, however, proved the vibrations were caused by resonances in the impeller which were transferred to the rotor.

Conclusion

The measurement method outlined here to determine bending vibration has yielded satisfactory results and can be used for other applications. For example the same instrumentation was used to investigate the foundations for a HSM-Roots compressor for steam. From the vibration measurements the optimal points for application of stiffeners were determined with due regard to the need for elastic parts in the structure to allow the thermal expansion of the compressor housing. By a similar method the measurement of blade vibrations in turbo machine construction was carried out (Ref. 2).

The measurement of bending vibrations is increasingly used in practice especially in cases where the limiting conditions cannot be verified by theoretical considerations alone, thus leaving significant uncertainties about the calculated results.

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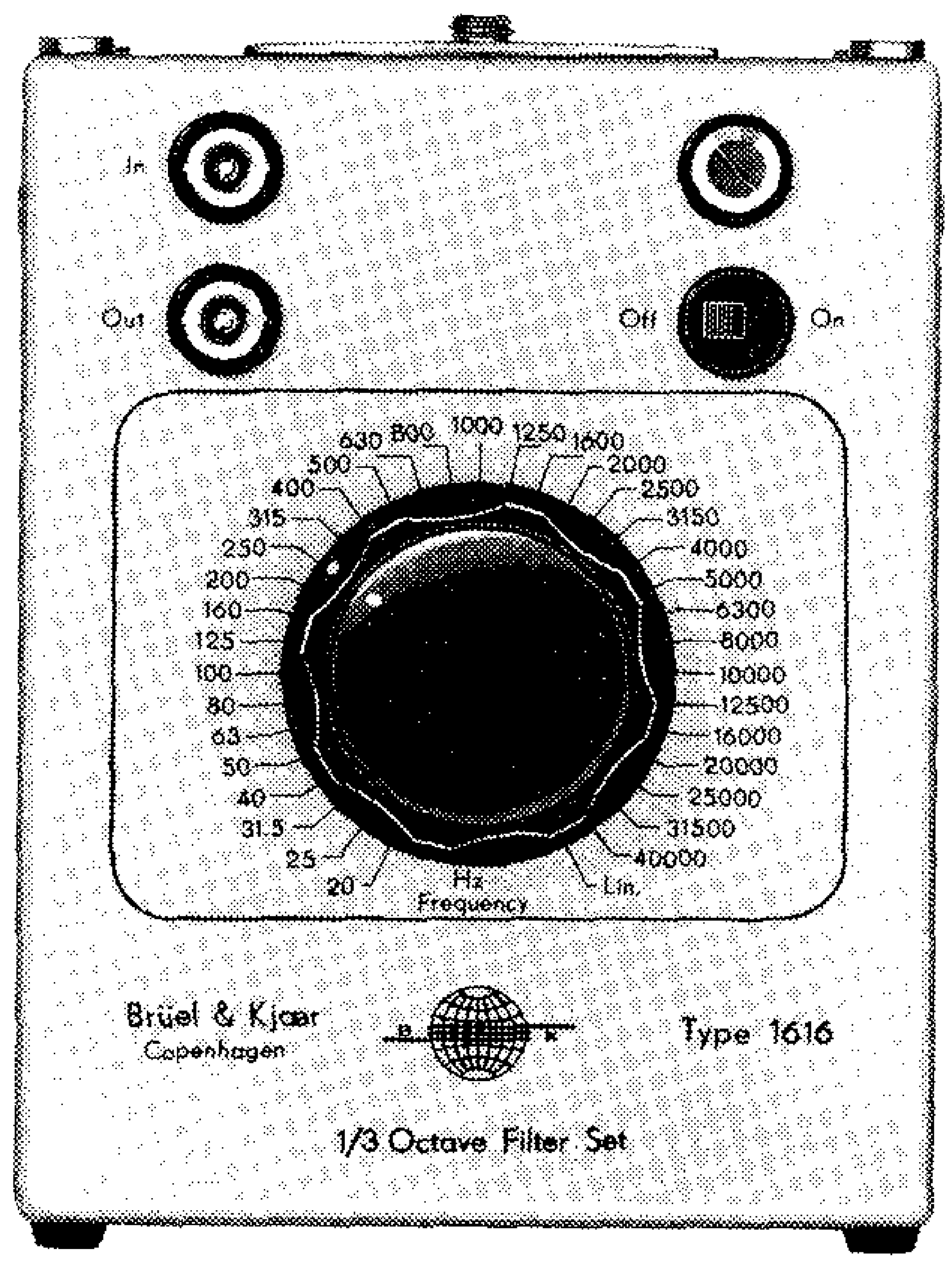
News from the factory

Third-Octave Filter Set Type 1616

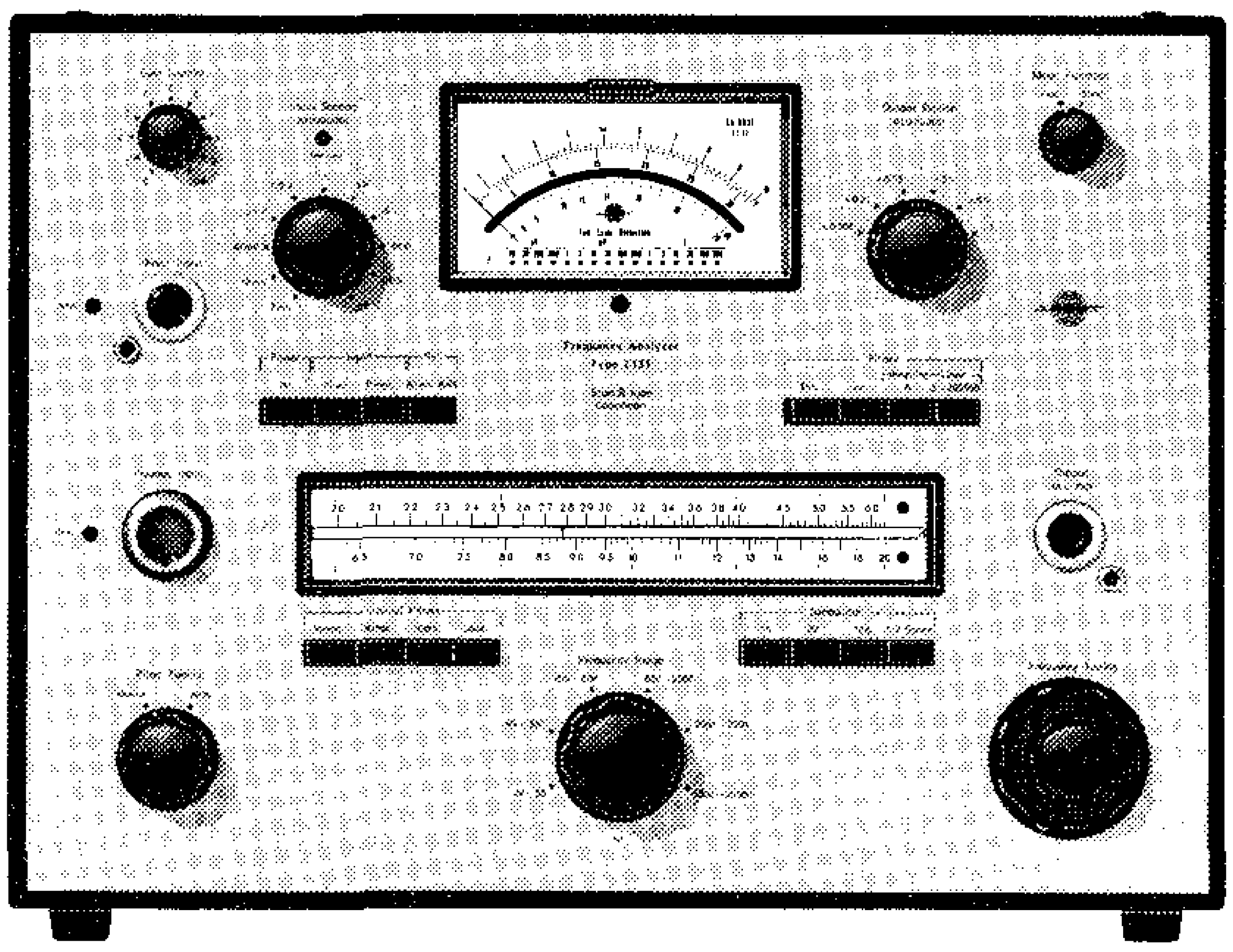


To carry out noise and vibration analysis using the Precision and Impulse Precision Sound Level Meters Types 2203 and 2209 a Third-Octave Filter Set Type 1616 has been designed. It combines mechanically and electrically by a screened connecting bar with either of the Sound Level Meters to form a compact portable unit. The filter set which covers the frequency range 18 Hz — 44 kHz contains 34 active third octave filters with centre frequencies from 20 Hz to 40 kHz in accordance with ISO R 266, DIN 45401 and ANSI S1.6-1960 standards. Each third-octave filter in itself fulfils the requirements of IEC Recommendation 225, DIN 45652 and ANSI S1.11, Class III.

For frequency analysis the filters may be switched in successively with the selector switch. In the linear position which is also included, a 22,4 Hz high pass filter and 40 kHz low pass filter are incorporated to suppress disturbance from low frequency signals and undesired high frequency signals.

Finally a battery monitoring meter on the front panel indicates when the batteries are run down.

Audio Frequency Analyzer Type 2121



This constant percentage bandwidth analyzer is a simplified version of the Frequency Analyzer Type 2120 and has a selective frequency range from 20 Hz — 20 kHz. It consists of a combination of the Measuring Amplifier Type 2608 and an active, continuously variable RC-filter complex which can be used in four basic modes.

1. As a constant percentage bandwidth analyzer having four selectable bandwidths, 1%, 3%, 10% and 1/3 octave.

2. As a tunable bandstop filter where the suppression of a single frequency f_0 is more than 60 dB, and at 0,5 f_0 and 2 f_0 is less than 1 dB. The bandstop mode can be used for distortion measurements down to 0,1%.

3. As a tunable high pass filter.

4. As a tunable low pass filter.

The internationally standardized A-weighting network for sound measurements as well as the linear frequency range from 2 Hz to 200 kHz are also included.

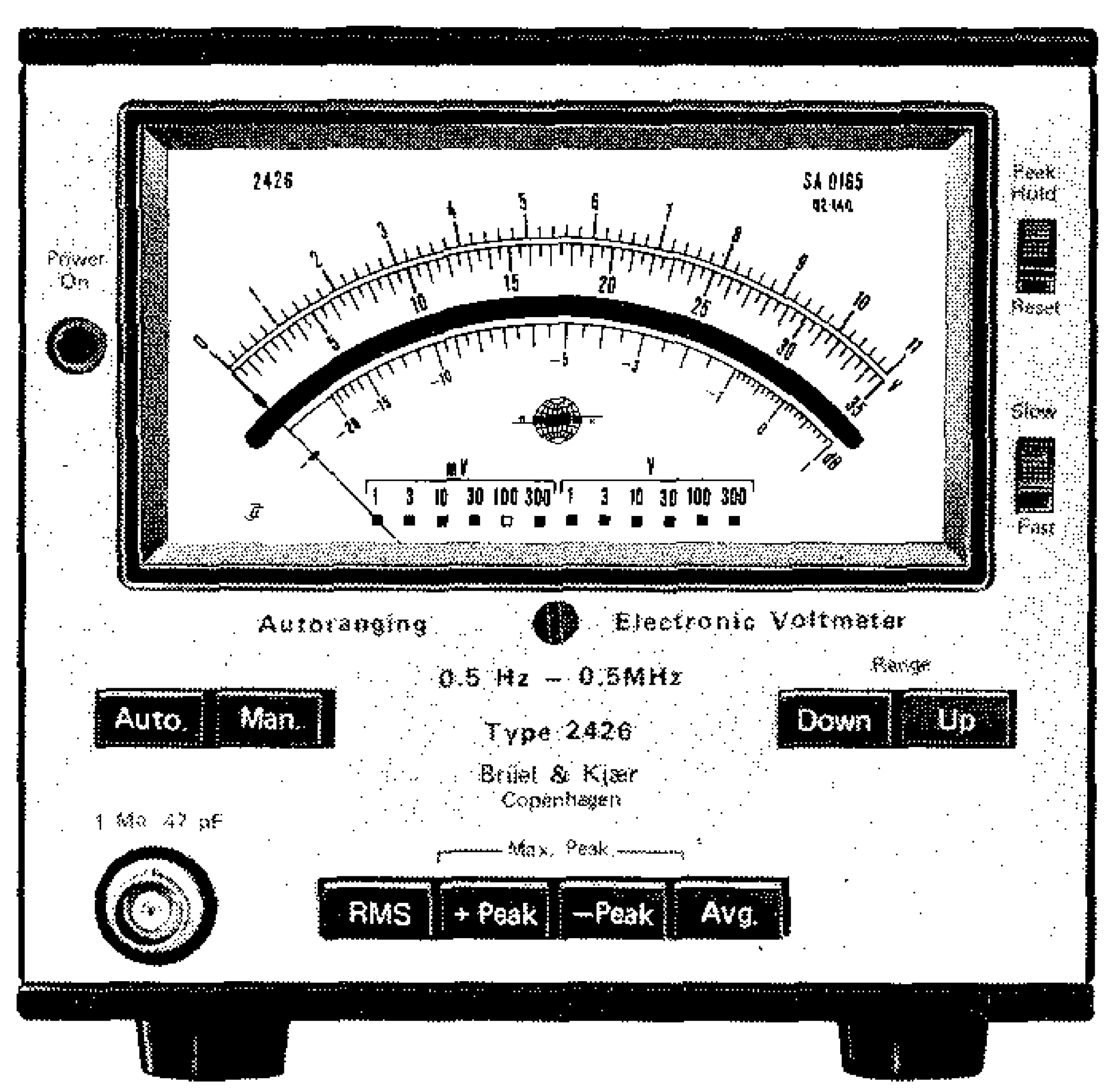
Provision is made for the connection of external filters to be used alone or in series with one of the internal filters. Utilizing, for example, the band pass 1/3 octave Filter Set Type 1614 the selective frequency range can be extended from 1,8 Hz to 180 kHz.

The analyzer contains "Fast" and "Slow" meter averaging times by which it can perform as a Precision Sound Level Meter in accordance with IEC Recommendation 179 and DIN 45633 part 1 when used with one of the B & K condenser microphones and preamplifier.

When used with a Level Recorder Type 2305 or 2307 a continuous frequency sweeping analysis can be carried out and an automatic recording of noise and vibration spectrograms can be obtained.

The measuring range of 10μV to 300V RMS (700V peak) together with a low noise level of 0,4μV makes the analyzer quite suitable for noise and vibration analysis over a wide frequency and dynamic range.

Autoranging Electronic Voltmeter Type 2426

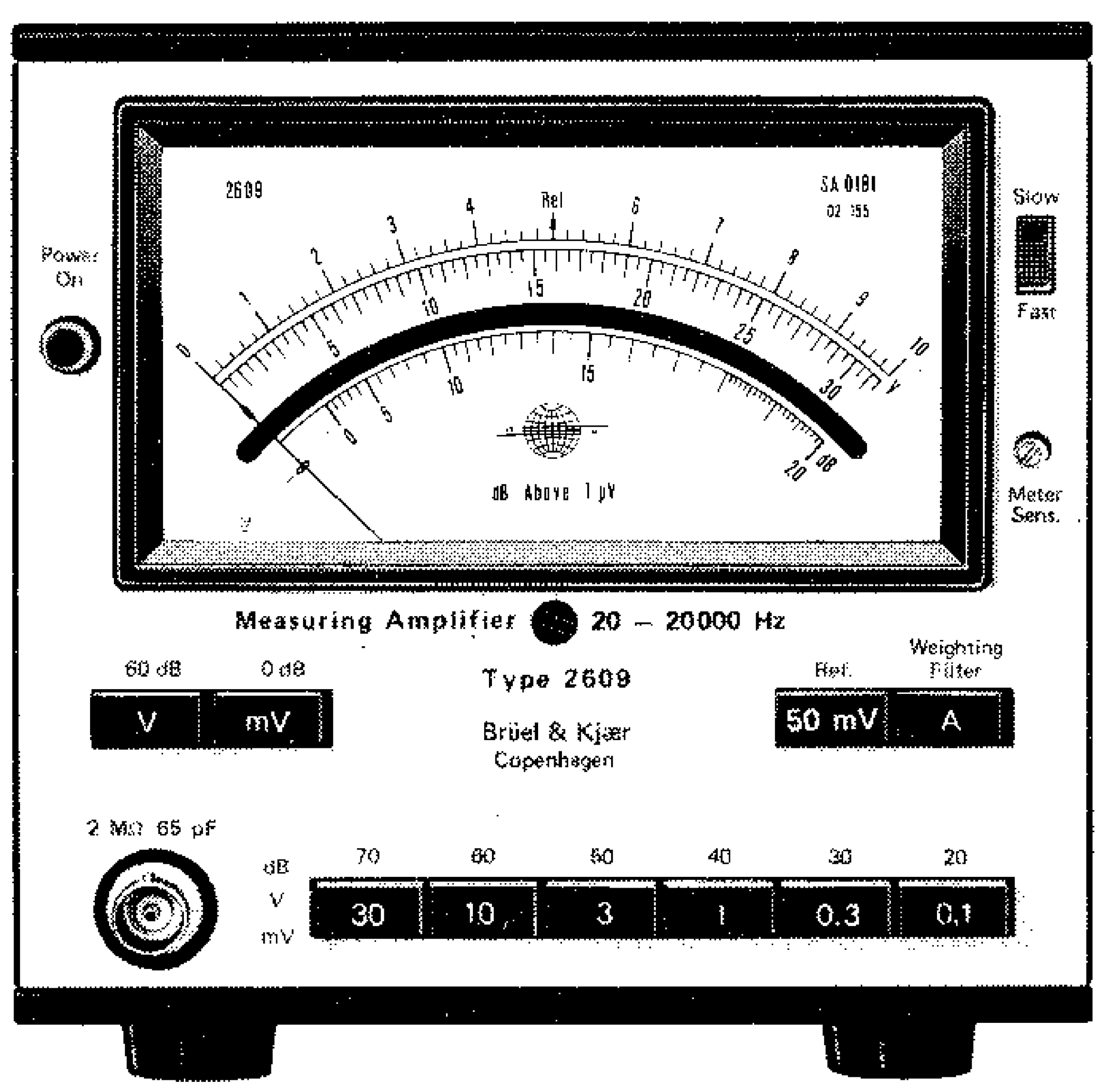


This voltmeter which is a small size general purpose voltmeter for indication of + Peak, —Peak, Max. Peak, true RMS and Average value is well suited for measurements on signals with complex waveforms up to a crest factor of 5. A Peak Hold function is also included for measurements on impulse signals. The instrument is extremely easy to operate, as in the "Auto" mode it selects the correct range according to the input signal automatically. The selected range is indicated on the meter of the instrument and also in the form of an 8-4-2-1 BCD signal at a socket on the rear of the instrument. The range may also be selected manually by push-buttons "Down-Up" or may be controlled remotely by a BCD signal.

The voltmeter has an amplification of 60dB in 10dB steps which is useful in set-ups where a calibrated amplifier is needed for which purpose linear AC and DC outputs are also incorporated.

The meter damping "Fast" is in accordance with the standards for VU measurements and the "Slow" is intended for use on low frequency signals and signals of varying amplitude. For measurement on narrow band random signals facilities are provided for varying the time constants by connection of external capacitors.

Measuring Amplifier Type 2609

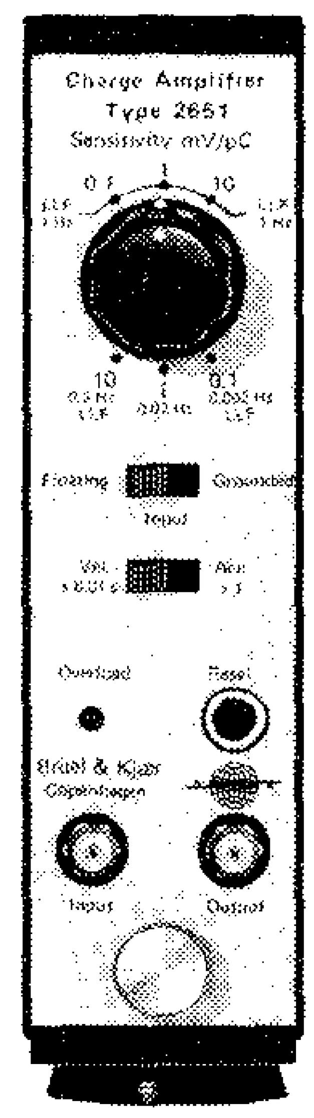


The Measuring Amplifier Type 2609 is a simplified version of the B & K line of Measuring Amplifiers intended for use exclusively in the audio frequency range 20 Hz to 20 kHz. On account of its limited number of features the instrument is relatively easy to operate and is economically ideal for multi-channel noise monitoring systems or less demanding applications requiring only limited capability. It is also intended for use in audio frequency compressor feedback loops and as an input amplifier for the Noise Dose Meter Type 4423, Tape Recorders Types 7003 and 7004, and Level Recorders Types 2305 and 2307.

The overall amplification is 90 dB calibrated in 10 dB steps while a further —4 to + 6 dB continuous adjustment is available to calibrate the meter scale for various microphone sensitivities. The voltage measuring range of the amplifier is from 100 μV to 30 V full scale deflection. "Fast" and "Slow" meter averaging times are in accordance with IEC 179 and will measure the true RMS value of most signals met in normal audio frequency work.

An A-weighting network may be selected by push-button though no provisions are made for connection of external filters to the amplifier.

Charge Amplifier Type 2651



The Charge Amplifier Type 2651 is a general purpose wide frequency range (0,003 Hz — 200 kHz) accelerometer preamplifier that eliminates the reduction of system sensitivity caused by the use of long accelerometer cables and makes recalibration of system unnecessary when accelerometer cables of different lengths are used. It has three fixed gain settings making it especially suitable for use with the B & K range of Uni-Gain® accelerometers. A dual FET input stage ensures a high input impedance, very low internal noise level and a low input current to secure a satisfactory low frequency response.

For low frequency vibration measurements and shock applications the lower limiting frequencies can be chosen to be either 0,003 Hz, 0,03 Hz or 0,3 Hz for which the output sensitivity of the amplifier will be 0,1, 1,0 and 10 mV/pC respectively. For these lower limiting frequency positions a "Reset" push-button is provided to clear the input stage when desired. However, for normal vibration measurements the lower limiting frequency can be chosen to be 1 Hz and the same output sensitivities can be chosen depending on the range of signal levels and transducer sensitivities.

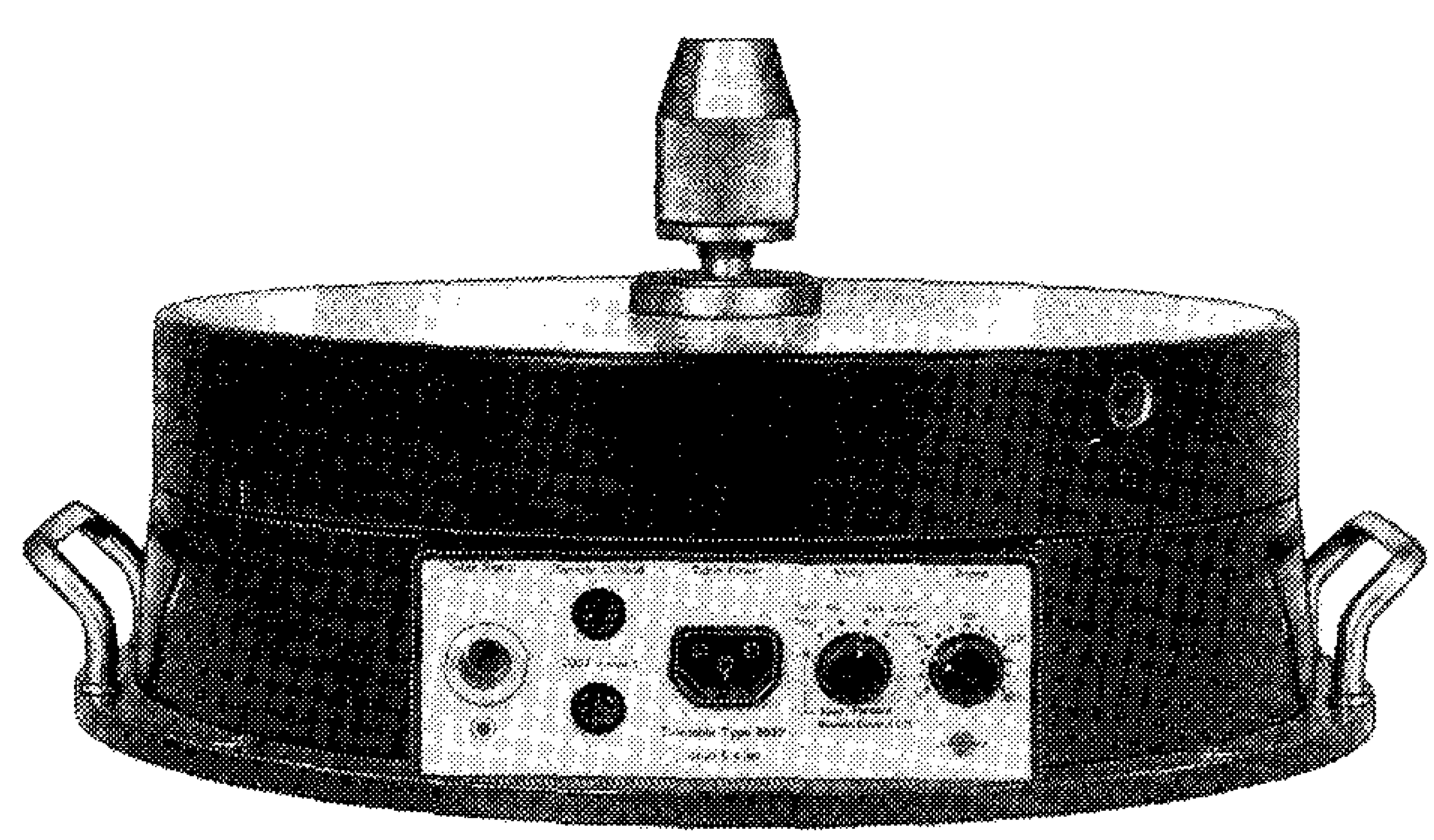
To overcome the often encountered problem of ground loop interference in measurement set-ups, a switch is incorporated to electronically separate the input and output grounds thus breaking the loop.

An active integrator is included in the amplifier to give the velocity output in the frequency range 10 Hz to 20 kHz. A fast acting overload indicator is included to respond to positive and negative overloads of duration as short as 20 μs. It is especially useful during measurements of complex signals in the velocity mode where the integrator will cut off the high frequency end of the signal.

To cater for plug-in usage in a rack system all the connections (power supply, input, ground and simultaneous outputs of velocity and acceleration) are available on a multipin connector on the rear panel.

The power supply for the charge amplifier may have either single or dual polarity. Dual polarity is, however, preferred as the output is then centred at ground potential with negligible DC offset (typically 10 mV), making it convenient to display a low frequency signal on an oscilloscope or a DC recorder. Any power supply of dual polarity with output ± 6 V to ± 18 V DC may be used without adjustment. With single polarity, power supplies of any output between + 12 V and + 35 V DC may be used and the DC offset will in this case be approximately half the supply voltage.

Turntable Type 3922



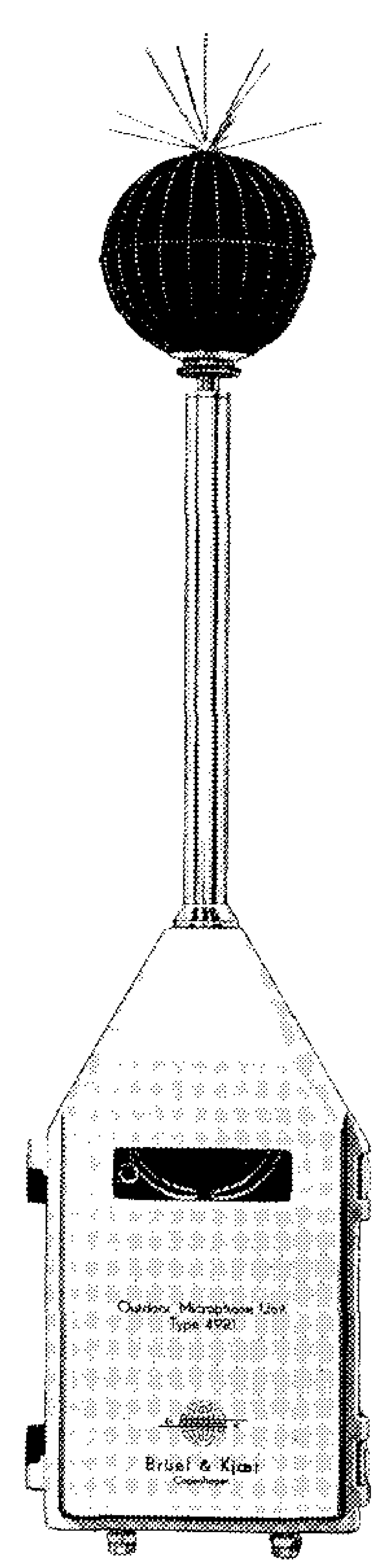
The directional characteristics of test objects such as antennae, loudspeakers, microphones, hydrophones etc. are often required in practice. To facilitate these measurements Brüel & Kjær have developed a Turntable Type 3922 to rotate a test object mounted on it in synchronism with the rotation of the polar diagram recording paper on the Level Recorders Types 2305 and 2307. The instruments when used in combination record automatically the directional characteristics of the test object on a 100 mm wide polar paper Type QP 5102 which is preprinted in graduations of 10° from 0° to 360°.

The turntable which is driven at 0,75 RPM by a synchronous motor has a diameter of 354 mm with six equispaced tapped holes (UNF 10— 32 and M5) for fixing test specimens and will take a load up to 100 kg. A removable chuck in the centre of the table will accommodate shafts up to 16 mm in diameter. Both manual and remote start of the equipment are provided and a friction clutch on the centre spindle allows positioning of the test object relative to the recording paper.

The function selector of the turntable offers 4 modes of operation. In the mode "Stop" the turntable and recorder will run as long as the start button on the level recorder is depressed. In the "Automatic Stop" mode the turntable and recorder run continuously and stop at a predetermined point. In the "Continuous Forward" mode the turntable and level recorder continuously run after the start button is depressed. With the Level Recorder Type 2307 only, the turntable may be made to sweep synchronously through a preset angle in the "Sweep" mode.

Two heavily screened connections via slip rings are available from the base plate of the turntable, one of which will supply power to the test object while the other will either apply or take signals from the specimen.

Outdoor Microphone Unit Type 4921



With the outcome of outdoor Microphone Unit Type 4921 complying with IEC 179, new standards of accuracy and reliability have been made achievable for outdoor noise measurements and permanent noise monitoring systems in the audio frequency range 20 Hz to 20 kHz.

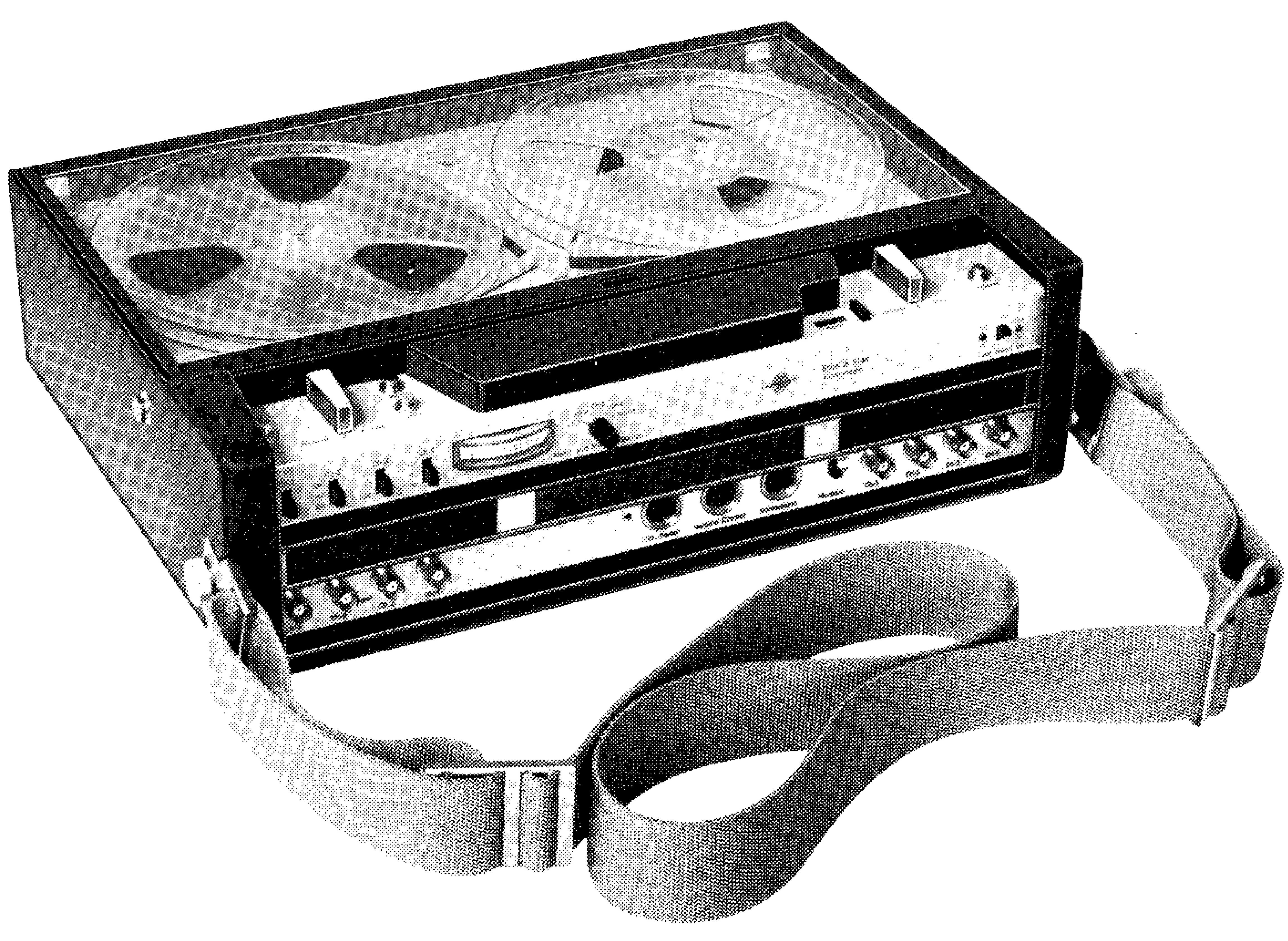
The unit which consists of two components connected by a steel tube has a quartz coated 1/2" Condenser Microphone Type 4149, a preamplifier similar to Type 2619, a rain cover, windscreen and an electrostatic actuator in the upper component.

Although quartz coating of the microphone diaphragm provides effective protection against moisture, pressure equalization of the microphone takes place via a dehumidifier in the lower component which also contains a control box in a weather-proof case. As a further protection against moisture a heating element is incorporated in the preamplifier to keep the microphone warm.

The control box which is a plug-in unit for facilitating servicing in the field, contains a conditioning amplifier (adjustable gain up to 60 dB), a calibration oscillator for the electrostatic actuator and an A-weighting network.

Built-in batteries provide power for 30 hours of continuous operation. A power supply to replace the battery box is also available.

Portable Tape Recorders Type 7003 and 7004



These multichannel instrumentation tape recorders giving optimum performance and reliability are designed especially for recording acoustic and vibration data in the field while still maintaining laboratory accuracy. They are small lightweight both battery or mains operated and employ all solid-state circuitry giving accuracy and reliability of performance. The rugged electrical and mechanical design enables them to be used under severe environmental conditions without loss of accuracy.

As the Tape Recorders Type 7003 and 7004, which employ FM and Direct recording techniques respectively, are mechanically and in some respects electrically similar, they have the unique feature that by, changing a few plug-in units and head assembly Types 7003 and 7004 can be interchanged. With FM recording the frequency range from DC to 10 kHz is available while for direct recording the frequency range is from 25 Hz to 50 kHz.

Type 7003 has four tracks and two tape speeds 1,5 i.p.s. and 15 i.p.s. giving a tape speed transformation ratio of 1:10, permitting low frequency vibration signals to be transformed to higher frequencies capable of being handled by conventional analyzers. The magnetic recording and reproducing heads are mounted separately on a precision plug-in support and are made of glassbonded ferrite giving extremely long life (10,000 h). The high signal to noise ratio, of 39 dB and 42 dB at 1,5 i.p.s. and 15 i.p.s. respectively and very low distortion, high linearity and low noise contribute as other attractive features of the instrument.

Type 7004 which is intended for acoustical measurements and applications demanding an extended frequency range, has two tracks and can be operated in two modes. In the direct mode tape speeds of 1,5 i.p.s. and 15 i.p.s. are available while in the "Audio" mode a maximum signal to noise ratio of 60 dB is obtained at tape speeds of 15 i.p.s. and 7,5 i.p.s. for recording within the audio frequency range.

Other features of the tape recorders include a voice circuit, overload indicator and facilities for remote control. Finally a tape loop casette is also included.

Following is the missing page 51 of references for article by Stryjenski published in TR 2 — 1973

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