B.I.O.S. Final Report No. 1856
Item No. I.

INFORMATION OBTAINED DURING AN INTERROGATION OF DR. FEDERN (CARL SCHENCK G.M.B.H.) ON MATTERS CONNECTED WITH VIBRATION AND SHOCK.

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(B.I.O.S. Trip No.1140)

Target No:- 1/283

TECHNICAL INFORMATION AND DOCUMENTS UNIT,
40, CADOGAN SQUARE, LONDON, S.W.1.

INTERROGATION OF DR.FEDERN AT CARL SCHENCK G.M.B.H. DARMSTADT ON MATTERS CONNECTED WITH VIBRATION AND SHOCK.

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1. OBJECT OF VISIT

No information concerning the mechanical durability testing of radio equipment for the armed forces of Germany had been collected by previous investigators, and very few targets associated with such work were known. Accordingly, the main purpose of this visit was to discover details of individuals, experimental and other organisations (service and industrial) who have carried out investigation, development, design or production work in connection with the following subjects, together with as many details of their work as possible.

(1) Vibration and shock conditions in all types of aircraft, naval vessels, road and rail vehicles, directed missiles etc., in which radio or radar components and equipment may be operated or transported.

(2) Machines for the production of conditions of vibration or shock etc., for laboratory or factory test purposes.

(3) Accelerometers, vibration pick-ups, measuring, recording, or analysing equipment suitable for examining the conditions encountered during the forms of transport or use detailed in (1), or created by the machines referred to in (2).

(4) Stroboscopes and other test gear auxiliary to the machines detailed in (2) above.

(5) Troubles encountered with radio and radar components and equipment due to the shock and vibration conditions encountered during the forms of transport or use detailed in (1) above.

(6) The correlation of the performance of the machines detailed in (2), and the test schedules devised for radio and radar components and equipment, with the nature and severity of shock and vibration conditions encountered in the forms of transport or use detailed in (1).

(7) The protection of radio and radar components and equipment from the shock and vibration conditions encountered in the forms of transport or use detailed in (1).

(8) The physical proper ties of the protective devices (anti-vibration mountings etc.), detailed in (7) above.

Dr.Federn of Carl Schenck G.m.B.H. was known to be an expert on the design of balancing and material vibration testing machines and therefore likely to provide a great deal of constructional design data and experience relating to subject (2) above.

2. SUMMARY

A description of the only vibration machine produced by Carl Schenck G.M.B.H. is given, together with nominal performance data, and this is followed by a reçord of the actual characteristics of the machine observed in both a short preliminary test on a machine installed in the factory and in a more detailed investigation of the machine carried out at a later date in this country. The machine was found to be free from all resonances except at the maximum frequency of 100 c.p.s. with heavy table loads when the agitating system developed a troublesome resonance, resulting in a considerable increase of table amplitude at the higher frequencies. Suggestions to overcome this trouble are given. It should be noted, however, that apart from the motor, the machine was originally designed for a maximum frequency of only 70 c.p.s.

Much useful information relating to the structural design of vibration machines was obtained from Dr.Federn, including details of new agitating system. These systems are discussed at length in the following pages. Systems are described for varying the amplitude of a vibration table in a predetermined cyclical manner between two selected values and for rotating the direction of a linear vibration through 360° at a predetermined rate.

Various methods of vibration table suspension for both horizontal and vertical movements and methods of driving the table are discussed.

Vibration measuring devices and frequency indicators are described, and brief details of work carried out on anti-vibration mountings are given. Some notes on foundations for vibrating machines are included.

Appendix I to this report contains the names of 34 organisations and individuals who have carried out work on the subjects enumerated in paragraph 1 above, apertaining to the durability testing of radio equipment for the armed forces of Germany.

Appendix II is a short account of an interrogation of Dr.Petersen of the Technische Hochschule, Darmstadt and is included as it has some bearing on the subject of this report.

3. INTRODUCTION

Carl Schenck, G.M.B.H. of Darmstadt,Essen in Germany, were engaged prior to 1939 mainly in the development and production of materials testing machines and balancing machines. Amongst other instruments the firm developed in 1936 a Reverse Bending Testing Machine (Wechselbiegemachine) employing a rotating eccentric and crank drive, the first

3. <u>INTRODUCTION</u> (continued)

machine being sold a year later. At the end of 1937, on their own initiative, they designed a vibration table using the same driving mechanism as in the Reverse Bending Tester. The machine, known simply as the Schenck "Rütteltisch" (Vibration Table), was developed because there was no satisfactory machine in existence at the time and it offered good commercial prospects. Prior to this date only two manufacturers were producing vibration machines. Losenhausen of Dusseldorf (See Appendix I, para.2.) made one type of vertical vibrating table resting on a rotating cam and guided in the vertical direction and another energised by out-of-balance rotating masses. List of Berlin (See Appendix I, para.3) also produced an out-of-balance type vertical machine. These machines were used for testing connection of flexible tubes subjected to vibration in normal use and for other similar purposes. The Schenck "Rütteltisch" provided a very satisfactory testing machine capable of giving reproducible results. The movement of the table related to time was sinusoidal, and the machine was free from all resonances over its operating range from 0 - 70 cp.s. frequency of vibration.

Difficulties encountered in obtaining special driving motors for higher speeds up to 6,000 r.p.m. delayed the raising of the maximum frequency until 1942. In this year the Reichsluftfahrtministerium (R.L.M.) Department G.L.C.-5, (See Appendix I, para.1) having difficulties with their own machines, asked Carl Schenck to supply a vibration table better than their existing machines and with a maximum frequency of 100 c.p.s. This Department G.L.C.-5 of R.L.M. was responsible for the approval testing of aircraft components. Schenck's used their standard table with a higher speed motor to satisfy this requirement, and this is the machine at present in production. Subsequently they received large orders for this type of machine and in 1944 they sold 150 machines. During the period 1942-1945 they supplied two or three machines each year to the firm Luftfahrtgerätewerk Hakenfelde G.M.B.H. (Berlin-Spandau, Streitstrasse, 5-17). (See Appendix I, para.5).

The following report includes a detailed description of the Carl Schenck "Rütteltisch" (Vibration Table) together with general comments on design of vibration machines arising from an extensive interrogation of Dr.Federn the present chief design authority on the Carl Schenck "Rütteltisch".

4. THE DESIGN AND PERFORMANCE OF THE CARL SCHENCK "RÜTTELTISCH"

4.1. Description of the Machine

The machine consists essentially of a table forced to execute a linear vibrating motion in the plane containing its top surface.

4.1. Description of the Machine (continued)

This motion is imparted to the table by means of a simple lever coupling operated by means of a motor driven crank. These parts may be easily identified in Plate.1, which shows an early model of the complete machine mounted on a suitable test bed.

The driving motor is a 220 volts D.C. shunt type of 1,000 watts rating. It is protected by means of an overload and no voltage cut-outs and its speed may be varied between 420 and 6,000 r.p.m. by rheostats connected in series with the field and armature windings. The field rheostat of 1,2000 ohms is intended for attaining the highest speeds only. Three aramature rheostats have values of 27, 200 and 630 ohms and are connected in series. The high resistance control, 630 ohms, is used to obtain speeds less than 1,800 r.p.m. One end of the driving motor is fitted with a revolution counter while the other end carries the driving cam. This counter may be used in conjunction with a stop watch for measuring the table vibration frequency, since this quantity is equal to the motor speed in revolutions per sec. The motor is supported by a welded channel iron framework in earlier models, and by a hollow box-shaped casting in The latest models of the machine. (See Plates 1 and 2).

The driving crank consists of two eccentrics whose relative position, when locked together determine the radius of the circular motion imparted to the big end of the lever connecting rod. This operation may be understood by referring to Figure 1., which shows the two eccentric discs having circular profiles, one fitting within the other. C_I is the centre of the inner eccentric I, C_s is the centre of a spigot on I, to which the big end of the connecting rod is fitted. C_R is the centre of O and also the centre of rotation of the whole. The eccentricity of the spigot C_s about C_R is the vector sum of $C_R C_I$ and $C_I C_s$. The eccentrics are designed so that $C_R C_I = C_I C_s$, hence the eccentricity of Cs about C_R may be varied between zero and 2 $C_R C_I$ by rotating I through 180° with respect to O.

Figure 2 shows the mechanical details of the eccentric mechanism. The outer cam is rigidly attached to the tapered end of the driving motor spindle by means of a keyway and a retaining nut. The motor selected for this machine was one having a specially short spindle of large diameter ($\frac{3}{8}$"). The fillet on this motor spindle eliminated the fatigue failures in this region which were experienced on earlier machines without it.

4.1. Description of the Machine (continued)

The driving crank and the connecting rod constitute a balanced system. The balancing has been carried out by removing material from suitable points of both the inner and outer eccentrics. The balancing recesses have been deduced mathematically and are identical for all machines, i.e., no experimental adjustment is made to individual machines.

The clamping ring secured by four 'Allen' screws, serves a dual purpose. Firstly it maintains the selected angular displacement of the inner and outer eccentrics, secondly it prevents any play between these eccentrics from introducing harmonic distorion of the table vibration waveform. The inner eccentric carries an index mark on its periphery whose position relative to the scale engraved on the outer eccentric indicates the selected eccentricity or radial throw of the lever connecting rod big end. Adjustment of the amplitude is carried out by slackening the clamping ring screws and rotating the inner eccentric by means of a special spanner engaging two holes in its surface.

The lever connecting is of conventional design, made of Elektron alloy, and houses a self-aligning double row roller bearing within its big end. The little end is a fixed double row roller bearing, and passing through this is a pin which is clamped to the upper end of the lever. The mean position of this rod and the lever are horizontal and vertical respectively. The lever is a single machines Electron casting hinged and supported at its lower end by means of the crossed spring strip arrangement illustrated in Figure 3. The merits of this particular hinge are:-

(i) there is no play present to introduce harmonic distortion.

(ii) rocking of the lever produces a uniform bending moment in the strips.

(iii) rocking of the lever takes place about the point of intersection of the strips, which is fixed.

(iv) the stiffness opposing motion of the lever in any unwanted direction is very high.

Rotation of the driven crank forces the lever to execute a rocking motion whose amplitude at the upper end of the lever is equal to the eccentricity of the driving cam. The horizontal projection of the motion of the lever at certain heights above the hinge may be directly imparted to the table by means of the table connecting rod.

4.1. Description of the Machine (continued)

Figure 5 shows how this is achieved and the table amplitude may be set to $\frac{1}{2}$, 1/5 or 1/10 of the driving crank eccentricity. These fractions have been referred to as the lever displacement ratio.

The table connecting rod consisted of two mild steel strips rivetted together, each of which had been pressed into a shape providing increased stiffness in a plane perpendicular to its horizontal axis. The occurrence of fatigue failures in a region close to the end clamping points necessitated the replacement of this particular rod by filleted design illustrated in Figure 4.

The table consists of a 40 cm. square machined casting of silumin alloy having a plane upper surface provided with tapped holes for the attachment of test on their jigs. The table thickness is 0.5 cm. or approximately 0.2" except in the region of the tapped holes and the stiffening webs on the underside. Stiffening is provided by the marginal flanges and two equally spaced webs extending from one end to the other where the table connecting rod is attached. The table is supported by means of a thin steel strip extending over the whole width of each end of the table. Three pairs of strips are provided so that the table may be supported at the heights corresponding to the three driving points of the lever.

All parts of the machine, with the exception of the 630 ohms armature rheostat, are mounted on a rigid cast iron bedplate measuring 85 x 65 cms. which is wet floated and bolted down at eight points to a concrete cube weighing approximately 1.5 tons. This concrete block is supported on four rubber blocks, symmetrically placed near each corner, so that its high resonant frequency in any mode of vibration is about 3 c.p.s.

4.2. Performance of the Machine

4.2.1. Nominal characteristics.

The handbook supplied with each machine contains the following information:-

The displacement waveform of the table is practically sinusoidal throughout the frequency range of all load values. The upper frequency limit of 100 c.p.s. and the lower limit of 7 c.p.s. are determined by the design of the driving motor used and the magnitude of the maximum series control resistance inserted in its armature circuit.

The vibration amplitude of the table is claimed to be independent of frequency, and is equal to the selected eccentricity of the driving cam multiplied by the lever displacement ratio used.

4.2.1. Nominal characteristics. (continued)

A dial gauge or microscope for amplitude measurements were only supplied with the machine on special request. Although the eccentricity of the driving crank may be adjusted to as high a value as 20 mm eccentricities greater than 12 mm should not be used. Eccentricities between 10 and 12 mm may be used occasionally for tests lasting only a few minutes, while values between 0 and 10 mm may be used for long duration tests. These rules apply irrespective of which lever displacement ratio is used. Amplitude ranges 0-5 mm, 0-2.0mm, 0-1. 0mm are therefore obtainable corresponding to the lever displacement ratios, 1:2, 1:5, 1:10. In view of the reduced bearing loads the lever ratio should be selected so that the amplitude required is contained in the smallest range.

The table load, which includes the weight of any devices for securing the test body to the table, must not exceed 50 kilograms. This limit, imposed by the strength of the table suspension strips, is identical for all lever displacement ratios.

The capabilities of the machine are summarised in Figure 7, whose curves for various lever ratios indicate the relation between the maximum permissible acceleration which the table can apply and the weight of the table load.

It will be noticed that the reduced bearing loads consequent upon the use of the lever ratios 1:10 and 1:5 permits the provision of higher test accelerations up to 30g.

Curves are given in Figure 8, 9 and 10 showing the maximum vibration frequency permissible for various loads over the three amplitude ranges available by using the three different lever ratios 1:2, 1:5 and 1:10 respectively. It should be noted that amplitudes on the right of the vertical dotted line on each chart may only be used for tests lasting a few minutes.

4.2.2. Observed Characteristics during preliminary test.

Tests were carried out on a machine set up in the factory Darmstadt. The performance of this machine was examined throughout the frequency range of 30-100 cp.s. permitted by the design of the D.C. shunt motor and its controlling rheostats. The waveforms of amplitude, velocity and acceleration of the table itself were investigated by means of an electromagnetic pick-up and using a Mullard Type E.805 oscilloscope together with a special differentiating and integrating unit produced by Phillips of Eindhoven. By feeding the velocity waveform into the unit it is possible to obtain the acceleration waveform by differentiation or the amplitude waveform by integration.

4.2.2. Observed Characteristics during preliminary test (continued)

Representative waveform taken at a number of different frequencies for lever ratios of both 2:1 and 10:1 are shown in Figures 11 and 12 respectively. An inertia operated electro-magnetic vibration pick-up manufacturerd by Phillips was used to provide the velocity waveform. The derived acceleration waveform is of the most interest and it will be noticed that some distortion is apparent at certain frequencies. This is due to some slight resonance in the machine either on the base or on the table, but it will be seen that the waveform is approximately sinusoidal. The high frequency ripple on the waveform, most noticeable on the acceleration waveform, is due to interference from the ball-races on the double-eccentric and on the vertical lever.

Ideally the amplitude of vibration should remain constant for any given setting of the eccentric drive over the entire frequency range. Dr.Federn stated that the machine had been designed from Stress rather than Constant Amplitude considerations. No complaints had been received from firms which had purchased the machine and the variation of amplitude with frequency due to insufficient stiffness had not been investigated. Users of the machine were concerned only with obtaining reproducible results, the machine being used for production testing and not as a calibrator of vibration instruments. During the above tests trouble was experienced due to the motor fixing bolts becoming loose and causing the Günther Tachometer, which was mounted on the opposite end of the motor shaft to the Double Eccentric, to vibrate excessively. This happened twice but was found to be due only to insufficient tightening of the fixing bolts.

4.2.3. Results of further investigation

Further tests have since been carried out with various table loads and amplitude settings. The machine used for these tests is shown in Plate 2.

During one test the amplitude, velocity and acceleration waveforms of the table were checked by means of an electro-magnetic vibration pick-up of the type used by Carl Schenck G.M.B.H. on their balancing machines. The velocity waveform produced by this pick-up was integrated and differentiated to produce amplitude and acceleration waveforms respectively, and these were displayed on an oscilloscope and photographed. The results are shown in Plates 3 and 4 attached, the oscilloscope time base being locked at half table frequency, thus showing two successive cycles.

4.2.3. Result of further investigation (continued)

It can be seen that all the waveforms, especially those for the loaded condition, approximate very closely to a pure sine wave. The integrating circuit was substantially linear between 20 and 100 c.p.s. and as the gain of the oscilloscope amplifier was the same for each displacement waveform shown the peak amplitude of the sine wave may be assumed to be approximately proportional to the actual table amplitude. Unfortunately the gain of the amplifier used was sufficient to give only a small amplitude sine wave on the photographs but it is clear that under "No load" conditions the table amplitude is approximately constant. This has since been checked by visual means and found to be true. Under the 55 lbs. load conditions however, the amplitude increases by 100% between 50 and 100 c.p.s., being ± 0.005 c.m. from 20-50 c.p.s. and increasing to ± 0.010 c.m. at 100 c.p.s. Compared with other existing types of machines this is by no means a poor performance.

Tests carried out at the same amplitude ± 0.005 c.m., but with a table load of 110 lbs. show a greater increase of table amplitude. It is interesting to note that the maximum permissible table amplitude at 100 c.p.s. for a load of 110 lbs. is given by the makers to be ± 0.007 c.m., so that the machine was being operated well within its specified limits.

In order to investigate the cause of the increase in table amplitude we must consider the individual strength of the various items constituting the moving parts and their mountings, and the possibility of resonances in the parts, occurring at frequencies close to or below 100 c.p.s.

The concrete mounting block and the cast iron base-plate were both checked and found to be rigid and to have negligible movement in any direction. The concrete block is mounted on four synthetic rubber blocks and has a resonant frequency below 5 c.p.s. in any direction. The motor and its mounting block were also found to be completely rigid relative to the base-plate under loaded running conditions.

Considering the moving parts, the table itself and its load were observed to have the same amplitude of vibration in the intended direction at all points on their surface and showed no movement in the plane perpendicular to this. We may reasonably suppose therefore, that no resonance of the table, its load or the table suspension strips was occurring.

The remaining possible sources of the trouble are now the table connecting strip, the strip clamping arrangement, the vertical lever, and the motor shaft. The clamping screws were all checked for tightness and no slipping movement could be observed with the aid of a stroboscopic lamp. The table connecting strip was known to be a weak link and is in fact over-stressed in tension by a factor of 100% when operating at 100 c.p.s. with an amplitude of ± 0.005 c.m. and a load of 110 lbs. It was observed that the strip was buckling at the end nearest to the vertical lever, and had at this point an abnormally large

4.2.3. Result of further investigation (continued)

vertical movement thus allowing the table amplitude to increase in the horizontal direction. The split steel tube tended to spread at the ends allowing the supposedly stiff portion of the strip to bend along part of its length.

A new connecting strip suggested by the designer of the machine, Dr. Federn, was manufactured and fitted. It consists of acute angle crossed steel strips at each end and is supported along its centre section by thick strips of light alloy. It is shown in detail in Figure 6 and may be seen fitted to the machine in Plate 5. This strip proved to be more successful than the original, although there was still a slight tendency to buckle at the lever end. Instead of being by far the weakest part of the system, however, it had now become relatively strong and other resonances showed up, namely in the vertical lever and the motor shaft.

With 110 lbs. load and a table amplitude of $\pm$ 0.005 c.m. the lever, when viewed by stroboscopic light, was seen to be bending along its length as a frequency of 100 c.p.s. was approached. This was confirmed by measuring the amplitude at a number of points on the lever as the frequency was varied. The amplitude of the centre point of the lever remained constant up to 70 c.p.s. and then increased until approaching resonance, at about 97 c.p.s., it was approximately twice its previous value. The eccentric was then set to an almost negligible throw, nominally zero, and on running the motor through the whole frequency range once more the amplitude of the centre of the lever appeared to increase fourfold at resonance, the throw of the eccentric having increased about 30%. Above the resonant frequency of the lever the amplitude of the centre point decreased to 200% of the undistorted movement whilst the top of the lever and the eccentric became almost stationary.

This latter effect pointed to resonance of the motor spindle overhang carrying the eccentric. The theoretical self resonant frequencies of the lever and the motor spindle were calculated for a table load of 110 lbs. and were found to be 111 c.p.s. and 99.3 c.p.s. respectively. These values compare very reasonably with those found experimentally. It should be noted however that in calculating the latter value, 99.3 c.p.s. the gyroscopic effect of the eccentric disc on the configuration of the shaft under conditions of whirling was not taken into account. This effect might in practice produce a first critical speed different from the figure of 99.3 c.p.s. given above. The self resonant frequency of the connecting rod between the eccentric and the vertical lever is relatively high and it may therefore be assumed that no resonance occurred in this rod.

Resonance of the driving system was also encountered with a table load of only 75 lbs. The test loads consisted of 14" square sheets of mild steel in two thicknesses $\frac{1}{4}$" and $\frac{1}{2}$" built up to required weight and bolted to the table with five bolts one at each corner and one in the centre. It is therefore interesting to note that in testing a standard

4.2.3. Result of further investigation (continued)

radio chassis weighing 70 lbs. no resonance effects were noticed, this being due to the effectively lighter loading of the table caused by the resilience of the radio chassis taking up part of the accelerating forces of the heavy components contained therein. This in effect means that although the machine is unsatisfactory at maximum frequency and load with the new connecting link due to resonance in the driving system when vibrating a dead load, it may safely be used for testing standard types of radio chassis in which, by reason of their method of construction, a certain amount of resilience is inevitable.

4.2.4. Suggested Improvements

The machine shown in Plate 1 has a motor mounting made up from two separate sections of channel girder. This was later replaced by a solid box shaped casting which gives much greater rigidity to the driving system. The lever is also shown as having only two different ratio connections 10:1 and 2:1, but this has now been modified to provide the three ratios 10, 5 and 2:1.

In view of the experiences described in the above paragraph, it would seem most desirable to increase the strength of the vertical lever to raise its self resonant frequency well above the operating range of the machine, that is to at least 140 c.p.s. The skirt of the resonance curve would then not cause any appreciable increase in amplitude such as occurs above 50 c.p.s. at present with moderate loads.

The motor shaft carrying the eccentric should also be increased in diameter and if possible the eccentric shaft overhang from the motor bearing should be reduced, in order to raise the first critical speed of the shaft overhang to the order of 7,800 r.p.m. (130 c.p.s.). The increased weight of the vertical lever would necessitate a stronger shaft in any case, and it might even be advisable to mount the eccentric on a separate large diameter shaft either overhung or slung between the bearings and driven by the motor through a torsionally rigid coupling such as a metal bellows. If the eccentric were to be slung between bearings then a change of eccentric design would be necessary. This is not desirable and difficulty would also be found in providing a suitable support for the outer bearing. There would, however, appear to be insufficient clearance on the present bedplate of the machine to move the motor in order to accommodate a coupling and bearings for an overhung eccentric, and therefore a special motor with a large diameter spindle presents the most convenient solution.

The performance of the table connecting strip would definitely be improved by replacing the acute angle or crossed strips by right-angle crossed steel strips at each end, the centre portion being made very stiff. This would entail slight modifications to the vertical lever and its present crossed strip pivot arrangement to allow the greater clearance required by the crossed strips of the connecting rod. The point of intersection of the crossed strips on the connecting strip should also lie on the line joining the point of pivot of the vertical lever and its upper fulcrum point in order to reduce vertical movement of this end of the connecting strip to a minimum.

4.2.4. Suggested Improvements (continued)

The connecting rod between the eccentric and the lever might also with advantage have the present ball race at the top of the lever replaced by a crossed strip pivot providing only small movements are required. This would eliminate any interference due to slackness or wear in the ball race.

For general vibration testing purposes where the frequency of vibration is only required to a moderate degree of accuracy a tachometer indicating cycles per second or per minute is preferable to the counter fitted at present.

The motor speed is varied by a combination of three separate variable resistances in series with the armature and one resistance in the field circuit. The speed control for the whole range would be more conveniently effected by one armature and one field rheostat.

5. GENERAL COMMENTS ON VIBRATION TESTING AND BALANCING OF ROTATING ASSEMBLIES BY DR. FEDERN

5.1. Detailed design considerations of vibration machines

5.1.1. Methods of driving the table

(i) Mechanical methods

The two usual methods of driving a vibration table mechanically are in the first place a direct link between a rotating crank or eccentric and the table and secondly an out-of-balance rotating weight or weights attached to the table itself. The latter method has the disadvantage that the amplitude of vibration will vary with the load on the table.

The first method, using a direct drive will provide a constant amplitude of vibration, providing the agitating system is not over-stressed in any way, but still suffers from the disadvantage that bearing play interferes with the vibration waveform at small amplitudes and also that bearing loads are high.

Both these latter disadvantages may be overcome in part by employing lever coupling to the table which has the advantage that great accelerating forces can be produced without damage to the bearings of the machine, and as in the case of the Carl Schenck machine the table amplitude may be set very accurately over a wide range by making use of different lever ratios. Adjustment of the eccentric also provides the easiest method of regulating the table amplitude. Bearings of all types should be avoided wherever possible in the driving system and spring loaded bearings should not be used as they have the serious disadvantage that the bearing load is doubled in one direction.

5.1.1. Methods of (continued)

(ii) Electrical

The table may be a y by the use of a loudspeaker
type of moving coil unit ut an iron core is used a
large mass of copper is must be firmly anchored but
the system has the advan no hysteresis effect and the
winding can be arranged soidal motion, Phillips of
Eindhoven (See Appendix nvestigated spacing the turns
and the effect of layer

With a flat faced d an air gap the motion is not
sinusoidal since the gap ging. One system suggested
was to use two electro-m side of the armature attached
to the table. Counter balance any load on the table
so that the centre of gr vibrating mass lies on the
axis of the magnets. A has been built for the
dynamic calibration of S

5.1.2. Driving mo drive systems and motor
couplings

Driving motors sho e their speed is less
dependent on variations voltage than it is with
smaller types. Direct ot have a very long life at
speeds in excess of 6,00 ubles encountered with the
carbon brushes. Altern es wound motors appear to be
in luenced to a large ex ltage variations and the best
method of driving seems ronous alternating current
motor and a variable fre ge.

It is possible to eeds by using gear drives.
Metal gears however prod resonances on all parts of the
machine, although this m rt by using helical type
gears. Dr. Federn had f fibro gears in connection
with vibration machines. d no high speed bearing
troubles, although in hi bearings gave less trouble
than ball bearings in hi The ball bearings used on
the "Rutteltisch" have m lised fabric.

Carl Schenck G.M.B niversal motors for
vibration tables, and if s not available then a
rectifier from 3-phase A erred. Speed control
troubles occur if a numb perated simultaneously off
the same D.C. supply.

5.1.3. Table susp orizontal movement

On the standard Sc e the suspension is effected
by using two steel strip Plates 1 and 2. This is a
simple method and was ad n. A multiple strip system
as shown in Figure 13(a) offers possibilities of

5.1.3. <u>Table suspension for linear horizontal movement</u> (continued)

reducing the weight of the table which would not then have to be so stiff as the distance between supports would be greatly lessened. A disadvantage of the strip method of suspension is that curvilinear motion of the tables takes places. If the height of the strip is increased to minimise this then it is not stiff enough to withstand high loads without buckling. If stiff lengthening plates are introduced as shown in Figure 13(b) then resonance within the suspension system itself becomes more likely, several modes of bending of the strips being possible, two of which are shown in Figure 13(c).

In order to avoid stress concentration at the clamping points on the strip it is found very useful in small machines to perforate the strip near these points as shown in Figure 13(e). For larger machines it is better to remove material from the strip as shown in Figure 13(f), grinding being carried out in the direction shown.

The best suspension is considered to be the cruciform strip arrangement shown in Figure 13(d), horizontal resonance being avoided because of high stiffness in the horizontal direction of the central mass. The best results are obtained here when the thickness of the strip is one tenth of the length. The length, or distance between the clamping points must be such that with maximum deformation no stresses are produced which exceed half the overstrain value. Small lengths should not be used as they are subject to shock damage due to overstrain at very small displacements.

The table suspension also has to bear the vertical rocking stresses which are produced when the centre of gravity of the load is much above the table level. This condition does not influence the stresses in the agitating system of the machine unless the amplitude of vibration of the top of the load increases due to resonance or a near approach to it.

The relative sizes of clamping strips and bolts, when the strip method of suspension is used, are shown in Figure 14. The recessed hexagon bolts, of high tensile steel, should be as tight as possible and the holes in the clamping strip and the suspension strip should give sufficient clearance to the bolt to ensure that the suspension strip is clamped entirely by friction and that no vertical forces act upon the bolts. The bolt shank diameter should be 40% of the width of the clamping strip and the angle θ on the diagram should be 45°. If the angle is greater than this value, the bolt shank will be larger and insufficient material of the strip will be left for clamping. The lateral spacing of the bolts should be equal to the width of the clamping strip. The bottom edge of both the clamping and suspension strips should be machined so that end support is provided by a good butt contact along the whole of their length.

In the present vibration machine the suspension strips are of cold rolled steel work hardened, and the clamping strips are of soft iron. No cement or varnish is used in the assembly of the strips.

5.1.3. Table suspension for linear horizontal movement(continued)

In cases where forced drive is not used care must be taken to ensure that buckling of a plain or cruciform strip suspension does not occur by the table "flopping" to one side under load. The table displacement should be limited so that at the maximum possible displacement the bending moment in the strip is not exceeded by the moment of the total load on the strip about the top edge of the clamping arrangement at the bottom of the strip. This also applies to spring supports in balancing machines.

5.1.4. Table suspension for Circular Horizontal Movement

In order to produce this movement it was suggested that the table should be supported by four bars at four corners and energised by an unbalance rotor under the table, operating in the plane of the centre of gravity of the table and the load combined in order to avoid precession. This system would necessitate a heavy bedplate and is probably not suitable for frequencies in excess of 100 c.p.s. It would possibly be better in this case to suspend the table on springs to isolate the foundation from the vibrations.

5.1.5. Table suspension for Vertical Movement

Vertical guide arrangements are unsatisfactory. Trouble had been experienced by the firm of List, Berlin (See Appendix I, para.3) with plunger guides and roller guides wearing quickly because the balls run on a very small fraction of the guide especially at low amplitudes. The wear thus produced will cause play in two directions perpendicular to each other and to the required direction of motion.

Spring suspension may be used, or preferably the Carl Schenck vibration table could be turned through 90° and a suitable right angle bracket used. An example of this type of machine was seen at the Carl Schenck Factory.

A very promising form of suspension was suggested by Dr. Federn as shown in Figure 15(a). The plate is of spring steel and is attached to the bedplate by the four outer holes and to the vibration table by the four inner holes or vice versa, the plate being in the horizontal plane in its unstressed condition. A number of these plates are arranged around the edge of the table, which should be of deep cast construction, and a number are also arranged in a lower horizontal plane to prevent rocking of the table. With this arrangement as shown in Figure 15(b) a perfectly linear vertical movement should result with no tendency towards rocking or horizontal movement of any kind. The dimensions shown on Figure 15(a) for the plate are those suggested by Dr. Federn.

5.1.6. Foundations for Vibration Machines

In the case of the Carl Schenck Machine a foundation block of concrete weighing $1\frac{1}{2}$ tons was used. This mass was based on experience, as a heavy foundation was necessary in order to obtain a linear movement.

5.1.6. Foundations for Vibration Machines (continued)

The table should be wet floated on to the top of the block. The resonant frequency of the block on its four synthetic rubber feet for oscillation in a vertical direction is not greater than 150 c.p.m. and the rocking resonant frequency is lower than this. A slight preference was stated for suspension of the block from above rather than support from below owing to the possible danger of the foundation falling over due to tilting in some cases.

The alternatives of isolating the foundation or embedding it in the ground are immaterial at frequencies up to 3,000 c.p.m. but above this frequency isolation is essential, as floor springiness gives rise to resonance troubles. Spring suspension has been found necessary to meet the severest requirements of accuracy but rubber blocks are best otherwise. Friction devices allow small amplitudes to pass and a great deal of trouble may result with nearby machinery even if only one vibration machine is concerned.

If it is desired to have the top of the foundation at floor level it should rest on rubber blocks at the bottom of a suitable pit as shown in Figure 16.

It is recommended that the foundation block should be made of a material called "schwerspat" which has a density comparable to that of granite (schwerspat = heavy spar, barytes). The foundation block must be in one solid piece and must not be composed of several separate slabs.

Holding down bolts for the table should be bedded into the block, passing right through if possible, and they should be quite free from grease before coming into contact with the concrete. Another method of holding down might be to tap the bolts into an inverted channel girder embedded in the block.

5.1.7. Adjustment of Amplitude

The Schenck double eccentric cannot be applied to a spindle between two bearings. This difficulty might be overcome by using an arrangement such as is shown in Figure 17. The outer eccentric which forms the inner race of the ball bearing at the end of the connecting rod, is clamped to the inner eccentric by three set screws. The screws pass through three slots in the outer eccentric into three of six tapped holes equally spaced on the face of the inner eccentric, which is keyed to the driving shaft. Thus by loosening the three screws the eccentrics may be moved through 60° relative to each other, and by moving the three screws to the other three tapped holes they may be moved a further 60°, and similarly again for a further 60° thus giving 180° of adjustment from minimum eccentricity of the ball race relative to the shaft.

The above system naturally calls for a large diameter ball bearing which is a disadvantage at the higher speeds, as the circumferential speed is increased and lubrication becomes difficult. Also slackness in the bearings becomes more pronounced in the larger sizes.

5.1-7. Adjustment of Amplitude (continued)

Dr. Federn had no practical experience of methods of adjusting amplitude during motion, but stated that Dr. Peterson (see Appendix I, para.9) of the Technical High School at Darmstadt might have some information on this.

During an interrogation of Dr. Peterson (see Appendix II) two methods of amplitude adjustment during operation of the machine were suggested. These two methods are shown diagrammatically in Figure 18.

The first method is suitable for an out-of-balance type of machine. The out-of-balance weight (or weights) is held against the centrifugal force arising during rotation by a flexible wire or chain which is passed round a bar or pulley wheel on the shaft, the axis of the wheel being at right angles to the latter, and runs axially along the shaft to a collar. The collar is constrained to rotate with the shaft by means of splines but is free to be moved axially by means of a suitable lever running in a slot or on a ball race. In this way the weight may be moved relative to the shaft and at right angles to it thus producing more or less unbalance as required which will in turn vary the amplitude of the vibration table for any given load.

The second method may be used with a hydraulically operated table. The crank assembly, C, produces a fixed oscillatory movement of the lever P Q which is pivoted about the point P. The roller assembly, R, transmits this movement to the piston of the drive pump, S, which produces the movement of fluid in the hydraulic system necessary to drive the vibration table. By sliding the whole pump along the lever P Q the stroke of the piston may be varied thus altering the table amplitude accordingly.

Dr. Federn also put forward some ideas for varying amplitude during operation, although these had not yet been tried in practice. One idea employing out-of-balance rotating weights finally developed into the system illustrated in Figure 19.

The vibrating table, T, is suspended on steel strips, S, as in the standard Schenck "Rutteltisch". The strips are mounted on the foundation block on which is also suspended a heavy mass, M, by any suitable means which will allow all round movement with a low self resonant frequency. To the top of this mass is rigidly fixed a vertical arm, L, which is attached at its top end, through a crossed strip arrangement and connecting strip, to the table. A single small rotating unbalance weight, R, may be positioned at any point along this vertical arm from the centre of percussion, CP, of the combined mass, M, and arm, L, to the top of the arm. When the rotor, R, is positioned at CP, then the effect on M and L combined will be very small and will cause practically no movement. As it is moved towards the top of the arm, L, i.e. away from the centre of percussion, the moment of the accelerating force produced by R at any instant will increase causing M and L to oscillate torsionally about CP. The greater the moment the greater the angle of oscillation will be and thus the horizontal

5.1.7. <u>Adjustment of Amplitude</u> (continued)

movement of the top of the arm, L, will increase accordingly. Most of the vertical accelerating forces produced are absorbed by the mass, M, but any slight vertical movement is taken up by the crossed strip and link assembly between the top of the arm and the table. As the centre of percussion will vary with the position of the rotor, it is not possible to pivot the mass, M, and in any case this would result in the vertical unbalance forces being transmitted to the foundation.

Another idea he had in mind was to use twin rotors running at slightly different speeds so that the "beating" effect would produce a cyclic variation of amplitude in any given direction, say horizontal. Vibration in the vertical direction could be eliminated in any convenient way. If an independent drive was used for each rotor the faster would tend to pull the slower to it and thus belt or gear coupling between the two drives is necessary.

In using multiple exciters without a positive coupling between them it is possible to get synchronism in phase opposition giving zero amplitude. From energy calculations it may be determined whether the exciters will pull into phase opposition or into phase synchronism.

Dr. Federn had not used any constant amplitude device on vibration machines, although electrical contact devices were used for this purpose on Fatigue Testing Machines such as the Schenck "Pulser". He stated that Phillips of Eindhoven (see Appendix I, para.23) might have information on a magnetic coil pick-up for control purposes.

5.1.8. <u>Adjustment and Central of Frequency</u>

Dr. Federn had carried out no measurements on frequency stability. Where troubles arise due to instability automatic control should be used. He is opposed to the use of gears as they produce high frequency resonances in different parts of the machine. For large machines where gears are used these should be of the double helical type to provide the smoothest possible running. Dr. Federn had no experience of fibre gears used under these conditions.

As large a motor as possible should be used to obtain maximum stability. Synchronous alternating current motors are useful for high speeds and the frequency is fairly easily adjusted. Alternating current series motors are greatly influenced by supply voltage variations and should thus be avoided.

5.1.9. <u>Large Vibration Machines</u>

For this type of machine an out-of-balance exciter is preferred with the load suspended. There are many design difficulties in the construction of a large forced drive machine and the cost is prohibitive. Troubles due to torsional oscillations have been encountered where two exciters are linked by loaded rubber couplings. Metal bellows (see Appendix I, para.22) are preferred as couplings as there is no backlash with this type.

5.1.10. Constant Acceleration

Dr. Federn had not carried out any work on this subject, but he stated that there had been a machine for producing known accelerations at the Technical High School (Technische Hoschschule) Charlotenburg, Berlin (see Appendix I, para.28). A Professor Cornelius had been engaged on investigation of torpedo gyros and had carried out tests on gyro apparatus.

It was considered that impact type machines gave no information regarding the actual conditions, and, except for apparatus subject to severe shock conditions such as in torpedoes, a vibration test was considered adequate in all cases.

5.1.11. Variation of Direction of Vibration

Vibration tests carried out to detect loose contacts and wiring in apparatus using a Cathode Ray Oscilloscope as an indicator require the direction of vibration to be varied. Dr. Federn had little experience of actual results of such tests, but he stated that linear movement was preferred because the acceleration could be calculated.

He suggested a system of producing linear motion using rotating unbalance weights such that the direction of the accelerating force applied to the table was rotated continuously in the horizontal plane. One out-of-balance mass with its axis vertical is driven by the motor and two others are positioned also with their axes vertical on opposite sides of the first and gear driven to rotate in the opposite direction to it and at a slightly different speed. The masses are such that the vibromotive couple is the same for the outer masses as for the inner. The direction of the resultant accelerating force will thus rotate at a rate determined by the ratio of the gear drive between inner and outer shafts.

Using this system to apply a vibratory motion to the testing table the efficiency of the check on electrical connections of an apparatus is greatly increased.

5.2. Vibration Testing of Materials

Concerning endurance testing Dr. Federn stated that investigations carried out at the Kaiser Wilhelm Institute, Dusseldorf, (see Appendix I, para.29) had shown that the endurance limit of steel was no different over the normal range of frequencies around 1,000 c.p.m. than it was at frequencies of 30,000 c.p.m. On damped materials, however, the frequency does have an effect as for instance in rubber in which internal heating results in fracture by heat and strain. It is possible to get trouble in steel due to the fact that damping is greater in torsional strain, and the requirement therefore arose for a torsion testing machine.

Plastics behave in a similar manner to rubber although there is no initial elastic deformation. After working, however, they tend to become

5.2. Vibration Testing of Materials (continued)

elastic and therefore their endurance depends on the frequency employed.

The Technical High School, Darmstadt, state that long term investigations on endurance of materials is unavoidable. Shorter tests are unreliable as the type of failure curve is not known. Static investigations had been made under heat conditions, and plastics and steels had been under test in ovens for two years.

The necessity for high and low temperature vibration testing was considered to depend upon the materials and was probably not necessary for metallic articles. Impact test results, however, definitely depended on temperature and low temperature tests were carried out on materials at different temperatures in several places.

In connection with the high frequency vibration testing of materials, Dr. Federn stated that a machine had been developed in 1928 for this purpose. The specimen under test was clamped between two iron end pieces, one of which was fixed, the other being suspended so that it was free to move axially with respect to the specimen under the influence of a large electro-magnet. The whole machine was installed in a special soundproof room. The magnet was D.C. polarised and supplied with alternating current at 500 c.p.s. The amplitude normally used was 0.2 m.m. deflection controlled to ± 1% and stresses up to 100 Kg/mm were produced.

5.3. Anti-Vibration Mountings

Dr. Federn mentioned two firms concerned with the production of anti-vibration mountings. The first was the Continental Caoutachouc Co., Hanover (see Appendix I, para.31) who manufacture mountings similar to the metalastik type, the second being "Getefo" (Gesellschaft für Technischen Fortschritt G.M.B.H. - Company for technical progress) of Schwarzach, formerly of Berlin (see Appendix I, para.30). Mr. Clemens-A. Voigt of the latter firm was engaged on investigation of new methods of isolation and new materials.

It was considered that better results were obtained by using a large number of small mountings rather than a few large ones.

Non-linear mountings could be designed to give a constant resonant frequency for different loads, although this frequency varied for different amplitude vibrations, being low for small and high for large amplitudes.

Dr. Klotter (see Appendix I, para.26) of the Technical High School, Berlin has carried out work in connection with this aspect of vibratory movement. Two firms Zeiss-Jena (see Appendix I, para.25) and Leitz of Wetzlar (see Appendix I, para.32) were experimenting on microspope isolation mountings. Dr. Bayerling of Anschutz (see Appendix I, para.4) has investigated the isolation of gyros from vibration (see Appendix I).

5.3. Anti-Vibration Mountings (continued)

Metal anti-vibration mountings using steel springs with oil damping allow vibrations of the foundation to be transmitted through the oil damping. However they are considered to be good for absorbing energy where vibrations in a machine have to be reduced. Steel springs with friction damping are unsatisfactory because if the mechanical forces due to vibration are smaller than the damping frictional forces then the mounting becomes a stiff connection.

Rubber mountings were preferred to steel because they absorb shock and damping is provided by the rubber itself.

5.4. Measurement of the Amplitude and Frequency of Vibration

Much information on vibration measuring devices is contained in "Archiv für Technisches Messen (ATM)" and also in "Electrische Messung Mechanisches" (Ref.1). The accuracy of different electrical measuring methods is described, covering photo-electric, capacity and induction types of pick-up and also strain meters of various kinds.

A more theoretical work on the subject is "Schwingungsmessgeräte" (Ref.2) (Vibration Measuring Devices) by Dr. Klotter late of the Technische Hochschule, Karlsruhe.

5.4.1. Vibration Pick-Ups

Until 1938 Carl Schenck G.M.B.H. manufactured a vibrometer consisting of two pendulums, one to indicate vertical and the other horizontal movement on a scale on which 1 m.m. deflection represented 0.0025 m.m. vibration amplitude. The pendulum not in use was clamped and the resonant frequency of the indicating system was of the order of 40 c.p.m. Production of these instruments was discontinued in view of the advantages of electrical types.

A moving coil pick-up is now produced, intended mainly for use on balancing machines. It is a direct drive type and is illustrated in section in Figure 20. A 4 m.m. diameter rod passes axially through the centre pole piece and is supported at each end by a diaphragm of high radial stiffness. Axial flexibility is achieved by a number of spiral slots in each diaphragm each slot continuing through 360° as shown in the lower half of Figure 20. The moving coil is attached rigidly to one end of the axial driving rod between the diaphragm and the circular gap between the pole pieces. Attached to this end of the rod is a thin pick-up probe which is clamped to the vibrating object. The whole moving assembly has a very small inertia and this has the minimum possible effect on the vibratory movement to be investigated. Two types of pick-up are produced with coil impedances of 3,200 and 200 ohms respectively. The former is for use with a cathode ray oscillograph whilst the lower impedance coil is used in one case to provide the input power for a wattmeter method of balancing employed by Dr. Federn. Two sizes are available for each type, one small size for frequencies of the order of 50 c.p.s. and another larger size containing a double coil for frequencies of the order of 8 c.p.s. The diameter of the wire used for the coils is 0.03 m.m. for the 3,200 ohm coil and 0.07 m.m. for the 200 ohm coil. One of the small size units with a 3,200 ohm coil may be seen mounted above the table connecting

5.4.1. Vibration Pick-Ups (continued)

strip in Plate 2.

Messrs. Phillips of Eindhoven produce an inertia operated type of moving coil pick-up with a sensitivity approximately half that of the Schenck type. Phillips carry out an ageing process on the magnet in order to obtain a good calibration curve; this is not done with the Schenck pick-up.

In comparing the inertia and direct drive moving coil types, the former suffers from the disadvantage that the whole mass must vibrate and in being clamped to the vibrating object the properties of the latter may be changed due to the clamping strain. The direct drive type is naturally preferable for low frequencies, and there is no need for damping devices in the pick-up as there is no low resonant frequency. Dr. Federn had also found that the Phillips type of pick-up was insensitive to high frequency ball bearing disturbances of 50 to 80 times the normal vibration frequencies, these being absorbed in the case of the instrument, thus showing that results of investigations depend largely upon the type of pick-ups used. Phillips, Eindhoven, have also built a relative type of pick-up as compared with the inertia type which measures absolute movement. Mr. Van Veen was the leader of the pick-up department of this firm, and should have information on the calibration of these instruments.

Bosch of Stuttgart (see Appendix I, para.7) produce a contact type of accelerometer with which D.V.L. (see Appendix I, para.6) have carried out tests to investigate aircraft undercarriage landing shocks. Shock measurements are also carried out with dynamic strain meters.

A number of firms have concerned themselves with various types of vibration analysers of the quartz crystal, photo-electric and sonometer varieties and these are listed together with other firms concerned with vibration investigation in Appendix I to this report.

5.4.2. Amplitude Measurements

The firms of Zeiss-Jena and Leitz, Wetzlar (see Appendix I, paras. 25 and 32) produce microscopes for the measurement of amplitude vibration. A photographically produced slit 0.001 m.m. wide is used with magnifications of X300 and X600.

Dual dial gauges may also be used mounted on micrometer slides. They should not be allowed to vibrate but should be set on opposite sides of the vibrating body so that they just make contact with it at each end of its travel. The micrometer readings compared with those taken with the body at rest then give the overall movement during vibration.

5.4.3. Frequency Indicators

A revolution counter was used on the standard Schenck "Rütteltisch" since in a large number of cases it was desired to know the total number of reversals carried out during a test.

5.4.3. Frequency Indicators (continued)

Messrs. Hartman and Braun of Frankfurt-am-Main (see Appendix I, para.19) manufacture resonant reed tachometers which have proved to be very satisfactory. They also make complex instruments covering 0-300 c.p.s. on one unit and 300-600 c.p.s. on another unit. The scale is marked in intervals of ½ c.p.s. and the instrument is driven by alternating current derived from a coil and magnet arrangement. They also make purely mechanical frequency indicators and they were concerned in the development of the wattmeter used by Dr. Federn on balancing machines.

Messrs. A.E.G. and Messrs. Siemens are the only other firms in Germany largely interested in the manufacture of frequency measuring devices, apart from a small firm in Dresden whose name was not known. Messrs. Telefonbau and Normalzeit (see Appendix I, para.27) of Frankfurt-am-Main, however, produce a single type of tachometer which is of interest. The instrument consists of a small 3-phase A.C. generator, driven by the rotating body, whose phase frequency is equal to the speed of rotation, but whose output voltage remains constant for all speeds. This generator is connected to an elaborate indicator consisting of a 3-phase synchronous motor, operated from the output of the 3-phase constant voltage variable frequency generator, driving a tachometer for speed indication and also a separate revolution counter. A stop watch is also incorporated in the instrument for timing purposes in conjunction with the counter for calculation of mean speed over a period of time and for other measurements. The generator consists of a stationary commutator of special design whose segments are connected to a resistance network fed by a 60 volt D.C. supply. Three brushes spaced 120 degrees apart are rotated over this commutator by the test shaft or body, and the spacing of the commutator segments and the resistance values associated with each are such as to give an alternating voltage supply to each brush, the wave form being a series of steps following an approximately sinusoidal law. The voltage on one brush leads or lags that generated in each of the other two by 120 degrees. Connections from these brushes to the synchronous motor indicator are made by slip-rings. A suitable rotary converter operating on the 220 volt A.C. supply and providing 60 volts D.C. can be supplied if required.

The standard frequency range of the above instrument is 0-2,000 r.p.m. but a range of 0-10,000 r.p.m. can be provided. The generator may not be run at speeds in excess of 1,000 r.p.m. and therefore a gear box is normally used to drive it. Its phase voltage is independent of speed and equal to 60 volts peak, the maximum current output being 2.6 amps.

This instrument is unnecessarily elaborate and expensive as a tachometer quite apart from the high driving torque required and the need of D.C. and A.C. power supplies. It may, however, have useful application in electronic speed or frequency control circuits, and it has been incorporated in the electrical equipment of certain balancing machines made by Carl Schenck G.M.B.H. to indicate the phase of unbalance with respect to a fixed point on the rotating body.

5.4.4. Stroboscopic Equipment

Dr. Federn demonstrated a stroboscope manufactured by Zeiss Ikon of Dresden (see Appendix I, para.8). It is purely mechanical in operation and consists of a variable speed universal motor driving a disc in which are punched a number of slots. The whole instrument is held before the eyes and the object to be investigated is viewed intermittently through the slots as they pass the viewing aperture. The instrument is supplied with seven interchangeable discs having 1, 2, 3, 6, 10, 12 and 24 slots. The calibrated speed range of the motor is 5-50 r.p.s. and therefore the effective frequency coverage of the stroboscope is 5-1,200 c.p.s.

The Technische Physikalische Werkstatten, department of Messrs. A.E.G., Berlin (see Appendix I, para.21) produce an electronically operated stroboscope using a high pressure mercury vapour lamp with a short flash duration, the instrument being known as the Lichtblitz-Stroboskop. A resistance capacity oscillator is used to control the flash frequency, and this may be synchronised by holding a pick-up coil near a small piece of iron attached to the rotating body. The phasing can be altered by moving the position of the pick-up coil. This type of instrument has been used in a high speed wind tunnel for photographic purposes.

6. CONCLUSION

Considering once again the Carl Schenck "Rütteltisch" its main features may be summarised as follows:-

(i) The machine is foolproof and can be used by unskilled personnel. The balanced reaction type of machine needs much more care in use.

(ii) The motion is substantially sinusoidal.

(iii) The movement is linear.

(iv) There is no resonance in any part of the machine between 0 and 100 c.p.s. vibration frequency for rigid loads up to approximately 75 lbs.

During 1944 and 1945 the "Rütteltisch" had been equipped with a calibrated microscope and also dial gauges, using the contact method described in paragraph 6.2, for measurement of table amplitude.

Messrs. Hartmann and Braun of Frankfurt-am-Main purchased a Schenck "Rütteltisch" in 1941. Since 1937 they had used a machine manufactured by Losenhausen, Düsseldorf, for testing aircraft instruments produced for the armed forces. This machine was of the out-of-balance rotating weight type and suffered badly from bearing wear after a time and was finally discarded in 1943 due to a broken gearwheel. On purchasing the Schenck machine they found it to be better than the Losenhausen 1940/1 model and to be satisfactory under all conditions of use. Representatives of Hartmann and Braun also stated that List of

CONCLUSION (continued)

Berlin-Trepslow make a vibration table but that it is a very unsatisfactory machine. It apparently operates on a resonant spring assembly method energised by a motor driven out-of-balance weight. If a supply voltage variation alters the speed of the motor a very large change in amplitude results as the machine is working on the resonance curve of the spring assembly.

It can be stated that much thought has been put into the design of the Carl Schenck "Rütteltisch" and it is probably the best machine of its kind in Germany. It has a good performance bearing in mind that it was designed for a maximum frequency of only 70 c.p.s. There is, however, room for improvement in order to overcome the slight increase of amplitude with frequency, although this probably only entails slight strengthening of the connecting strip assembly between the table and the lever, and also the lever itself.

Two Appendices are attached. Appendix I contains a list of a number of firms and also individuals whose work is connected with the subject of this report, together with details of research and investigation or production work carried out.

Appendix II is a brief report on an interrogation of Dr. Petersen of the Technische Hochschule, Darmstadt.

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3rd May, 1948

APPENDIX I

LIST OF FIRMS AND INDIVIDUALS WITH DETAILS OF WORK CARRIED OUT IN CONNECTION WITH VIBRATION

1. Reichsluftfahrtministerium (R.L.M.) Dept. G.L.C.5.

Location: (i) Berlin (Ober Kommand Lufthahrt)

(ii) Peenemunde (later known as Karlshargen)

Approval testing of aircraft components. Placed order with Carl Schenck G.M.B.H. in 1942 for Vibration Table with maximum frequency increased from 70 to 100 c.p.s. (See Report para.3).

2. Losenhausenwerke Maschinenbau A.G.

Location: Dusseldorf

Manufacturers of various vibration machines of the cam-operated and out-of-balance rotating mass type. (See Report para.3).

3. List

Location: Berlin - Trepslow

Manufactured out-of-balance type vibration machine. (See Report paras. 3 and 5.1.5.).

4. Anschutz A.G.

Location: Kiel - Neumühlen

(Herr Fischer and Dr. Bayerling)

Connected with vibration investigation in torpedoes. Quartz crystal pick-ups were used to actuate recorders. Experiments were carried out on vibration tables in connection with the development of gyro compasses, and both experimental and mathematical investigations were conducted to assess the effect of vibration on gyros in general. Dr. Bayerling has experience of isolation of gyros from vibration. (See Report para. 5.3).

5. Luftfahrtgeratewerke Hakenfelde G.M.B.H.

Location: Berlin - Spandau, Streitstresse 5 - 17.

This firm has purchased several Carl Schenck "Rütteltischen". They are concerned with automatic controls of various types and have a sub-group, Siemens, engaged mainly in manufacturing gyros. (See Report para.3).

APPENDIX I (continued)

6. D.V.L. (Deutsche Versuchsanstatt für Luftfahrt e.V.)

Location: Berlin - Adlershof

(Dr. H. Viehmann, Elektr - Geräte (Dept. for Electronic Devices)

D.V.L. carried out investigations on vibration and shock conditions in aircraft using many types of accelerometers and both piezo-electric and electromagnetic vibration pick-ups including those manufactured by Phillips, Eindhoven (see 22 below). The Bosch accelerometer had been found to be the best although it suffered from the disadvantage that it gave a discontinuous reading (see 7 below).

D.V.L. have produced reports on the above and also on shock investigations in connection with landing gear. Thought had been given to methods of producing constant acceleration. (See Report para.5.4.1.)

7. Bosch

Location: Stuttgart

Manufactured vibration pick-ups and accelerometers used by D.V.L. (See 6 above). The Bosch accelerometer consisted of spring supported masses carrying contacts which were set to operate at 1g, 2g, 3g, 4g, etc. (See Report para.5.4.1.)

8. Zeiss-Ikon

Location: Dresden

Produce stroboscopes and acceleration measuring devices including a quartz crystal pick-up vibration analyser. (See Report para.5.4.4.)

9. Materialprüfungsanstatt (Materials Testing Laboratory)

Location: Technische Hochschule, Darmstadt, Essen.

(Dr. Petersen)

This laboratory have carried out vibration investigations on autocars and engine suspensions using a vibration meter made by Askania (see 10 below).

In this laboratory there is a reverse bending materials testing machine in which adjustment of the degree of bending may be effected during operation by varying the throw of a double eccentric. This is achieved by adjusting the second eccentric by means of a motor carried round on the shaft of the first eccentric. (See Report paras. 5.1.7. and 5.2.)

10. Askanin

Location: S.W. Berlin

Manufacturers of vibration measuring instruments using piezo-electric and other types of pick-up. (See Report para.5.4.1.)

11. Junkers

Location: Dessau

This firm have made investigations of vibration conditions caused by both jet-propelled and conventional engined aircraft. They had trouble with the variable pitch airscrew in dive bombers and were very much concerned with balancing of crankshafts and turbines. Dr. Federn of Carl Schenck G.M.B.H. was requested to develop a machine for balancing propellers. One such machine was produced from a standard machine by using special grips.

12. Forschungsinstitut für Segelflug

Location: Ainring/Obb. (Upper Bavaria)

Shock and vibration investigations on gliders.

13. Kraftfahrtforschung

Location: Stuttgart

(Professor Kamm)

Professor Kamm has carried out very extensive investigations on motor-cars on "autobahn" and other roads.

14. Naval Problems

Location: Danzig (Gdingen or Gotenhafen)

Investigations on torpedoes. Laboratories probably all destroyed.

15. Torpedoversuchsanstatt

Location: Eckernförde (bei Kiel)

Torpedo investigations.

16. Technische Hochschule, Darmstadt

Location: Darmstadt, Essen (Professor Vieweg (Physicist) and Dr. Haardt (Faculty of Mechanics)).

Professor Vieweg was concerned with the conditions of flight of the V2 rocket and research problems related thereto.

APPENDIX I (continued)

17. M.A.N.

Location: Augsburg, Nurmburg

Special Laboratory available for naval vibration testing problems. They have published several technical articles in "Schwingungstechnik". (Julius Springer - Berlin 1934).

18. Dr. Lehar (deceased)

Location: Augsburg

He has carried out investigations on strain in wheels and axles of railway rolling stock and in connecting rods and chassis of locomotives. He developed electrical dynamic stress meters capable of measurement down to 0.001g using simple non-electronic equipment.

19. Hartmann und Braun

Location: Frankfurt-am-Main

This firm have experience of the Carl Schenck "Rütteltisch", one of which was supplied to them in 1940. They manufacture indicating and recording instruments of many types. (See Report para.5.4.3. and 6).

20. Erprobungstelle

Location: Rechlin (bei Berlin)

This establishment was responsible for all approval testing for the German Air Force. They investigated the Carl Schenck "Rutteltisch", and as a result an order was placed for 150 of these machines.

Each sub-division of R.L.M. (see 1 above) had a sub-group in Erprobungstelle.

21. A.E.G.

Location: Berlin

Concerned with stroboscopes, and frequency control tachometers. Their department, Technische Physikalische Werkstatten, produced the A.E.G. Lichtblitz Stroboskop. (See Report para.5.4.4.)

22. Samson Apparatebau A.G.

Location: Frankfurt/Main

Manufacturers of metal bellows with both parallel and spiral convolutions and various temperature and pressure controlling devices. (See para.5.1.9.)

APPENDIX I (continued)

23. Phillips

Location: Eindhoven, Holland

Manufacturers of piezo-electric and electromagnetic type vibration pick-ups and analysers (see 6 above). This firm may have information on electromagnetic pick-ups for control purposes (see Report para.5.1.7).

They have also carried out investigations on moving coil methods of producing vibration (see Report para.5.1.1 (ii)).

24. Majhak

Location: Hamburg

Have manufactured a sonometer type analyser.

25. Zeiss-Jena

Location: Jena

Manufacture microscopes for small vibration amplitude measurements. A slit of 0.001 m.m. nominal width, made photographically, is used with microscope magnifications of X300 and X600. They have also carried out investigations on microscope isolation mountings (see Report paras. 5.3. and 5.4.2).

26. Dr. Klotter

Location: Formerly at Die Technische Hochschule, Karlsruhe

Later probably at D.V.L. and also at Die Tech. Hochschule, Berlin.

Concerned with vibration measuring instruments and published a book called "Schwingungsmessgeräte" ("Vibration Measuring Devices"). See bibliography.

He has also carried out investigations on non-linear anti-vibration mountings (see Report paras. 5.3. and 5.4.).

27 Telefonbau und Normalzeit

Location: Frankfurt-am-Main

Manufacturers of telephone and light electrical equipment. They produce an elaborate frequency indicator using a constant voltage variable frequency 3-phase generator (see para. 5.4.3).

APPENDIX I (continued)

28. Technische Hochschule, Berlin

Location: Charlottenburg, Berlin

Professor Cornelius was concerned with investigations on torpedo gyros.

There is in this university an impact type shock testing machine producing a known shock acceleration (see Report para. 5.1.10).

29. Kaiser-Wilhelm Institut

Location: Dusseldorf

Vibration endurance testing investigations carried out at normal frequencies around 1000 c.p.m. and also at high frequencies of the order of 30,000 c.p.m (see Report para. 5.2).

30. Getefo (Gesellschaft für Technischen Fortschritt G.M.B.H.)

Location: Schwarzach (formerly Berlin)

Dr. Clemens A. Voigt investigated new methods of vibration isolation and new materials for mountings (see Report para. 5.3).

31. Continental Caoutachouc Co.

Location: Hanover

This firm manufactures anti-vibration mountings (see Report para. 5.3).

32. Leitz

Location: Wetzlar

Carried out investigations on isolation of microscopes from vibration and produces microscopes, for measuring vibration amplitude, of a cheaper variety than those of Zeiss-Jena mentioned in 25 above (see Report paras. 5.3. and 5.4.2).

33. Dr. Heymann

Location: Formerly of Carl Schenck G.M.B.H., Darmstadt
Later in Zwingenburg or Auerbach

Dr. Heymann patented an idea for producing vibration using multiple exciters operating at different frequencies. He was also one of the first to use electrical methods for balancing.

34. Professor Houwick

Location: Delft, Holland

Expert on rubber problems including those connected with vibration isolation.

APPENDIX II

NOTES ON AN INTERROGATION OF DR. PETERSEN TECHNISCHE HOCHSCHULE, DARMSTADT

1. INTRODUCTION

Dr. Petersen was concerned mainly with materials investigation. He had studied vibration stresses in structural parts of aircraft and stress amplitude curves had been drawn.

He considered it necessary to simulate vibration conditions both to meet practical requirements and to investigate the scientific fundamentals of the whole problem. When investigating the cause of an effect linear sinusoidal motion should be applied, all factors which affect the behaviour of the test specimen being investigated. An attempt should be made to initiate the phenomena, and conditions should then be varied in order to isolate and identify the cause.

2. VIBRATION MACHINES

2.1. Driving Methods

Considering mechanically driven machines the direct drive produces an amplitude of vibration which is independent of the mass of the test speciman. With the out of balance rotating weight drive the amplitude depends on the load which must therefore be maintained at a constant value for repeatable vibration conditions. The latter method, however, has the advantage that negligible forces are transmitted to the immediate surroundings.

2.2. Bearings

Good bearings are very necessary with both types of machines. Irregularities in roller or ball races cause higher harmonics to be superimposed on the fundamental waveform and this effect is aggravated in the case of ball races if the material in the race is not perfectly homogeneous as irregular wear will result. From this aspect, therefore simple journal bearings or better still conical joint bearings are to be preferred. At high speeds, however, cooling of the latter types becomes a difficult problem, and even complete immersion may not solve it. Also comparatively high friction torque has to be overcome at start. Ball and roller bearings have the advantage that friction is very small at low speeds and they could if necessary run for a certain time with very little lubrication. Of the two latter types the ball bearing is more easily worn out as mentioned above and therefore roller bearings designed for optimum conditions with the correct lubricant would appear to be best.

APPENDIX II (continued)

2.3. Waveform

Asymmetrical waveforms may be caused by parts of the machine or joints between parts, yielding more in one direction than in another. Thus a joint between two parts may yield little in compression due to butt contact of two faces, but may yield very much more in tension due to stretching of the joint.

2.4. Variation of Vibration Amplitude

Dr. Petersen suggested two methods of varying vibration amplitude during operation of the machine. These methods have been described in paragraph 5.1.7. of the Report and are illustrated diagrammatically in Figure 18.

The first method is suitable for an out of balance type of machine. The out of balance weight (or weights) is held against the centrifugal force arising during rotation by a flexible wire or chain which is passed round a bar or pulley wheel on the shaft, the axis of wheel being at right angles to the latter, and runs axially along the shaft to a collar. The collar is constrained to rotate with the shaft by means of splines but is free to be moved axially by means of a suitable lever running in a slot or on a ball race. In this way the weight may be moved relative to the shaft and at right angles to it thus producing more or less unbalance as required which will in turn vary the amplitude of movement of the vibration table for any given load.

The second method may be used with a hydraulically operated table. The crank assembly, C, produces a fixed oscillatory movement of the lever P Q which is pivoted about the point P. The roller Assembly, R, transmits this movement to the piston of the drive pump, S, which produces the movement of fluid in the hydraulic system necessary to drive the vibration table. By sliding the whole pump along the lever P Q the stroke of the piston may be varied thus altering the table amplitude accordingly.

2.5. Reduction of forces transmitted by direct drive types of machine to their surroundings

Normally a heavy foundation resting on rubber cushions will suffice, the whole having a resonant frequency below the lowest operating frequency. Another method of avoiding interference from machines of the direct drive type would be to arrange two tables to be driven in exact phase opposition with identical loads on each.

APPENDIX II (continued)

3. MEASUREMENT OF VIBRATION

When questioned on measuring methods, Dr. Petersen was familiar with the standard devices such as the Föppel triangle, the narrow slit used by Carl Schenck G.M.B.H., the mirror rotating through an angle dependent on the vibratory displacement and strain gauges. He recommended a book by E. Siebel called "Handbuch der Werkstoffprüfung" published by Julius Springer.

4. ENDURANCE TESTING OF MATERIALS

If a specimen withstands ten million stress reversals then it will never break at that stress. For a good specimen this figure may be reduced to one or two million, but if corrosion or notching is likely to be present then the test should be carried to fifty million reversals. The effect of small temperature changes on fatigue was not considered to be great.

A synthetic material such as Hardstpapier has a practically constant fatigue strength over a large temperature range but its creep strength decreases with increase of temperature.

Under high frequency vibration conditions some materials generate very great internal heat, so great with some metals that partial melting occurs. If the damping energy is small then the heat generated is small. With metals only a 10-20% change is noticed between 30 and 30,000 cycles per minute.

With rubber fatigue fracture is a heat phenomenon where the fracture starts inside the material due to the heat generated. Materials with a high damping capacity are not safe for use under conditions of vibration. Rubber is satisfactory below 100°C.

Tests carried out on rubber to metal joints had shown that under static test conditions a fracture would run through the rubber where under fatigue tests the fracture occurred at the cemented area. During 1940 and 1941 many firms submitted samples of metal-rubber bonding for tests to determine the fatigue strength of the bond.

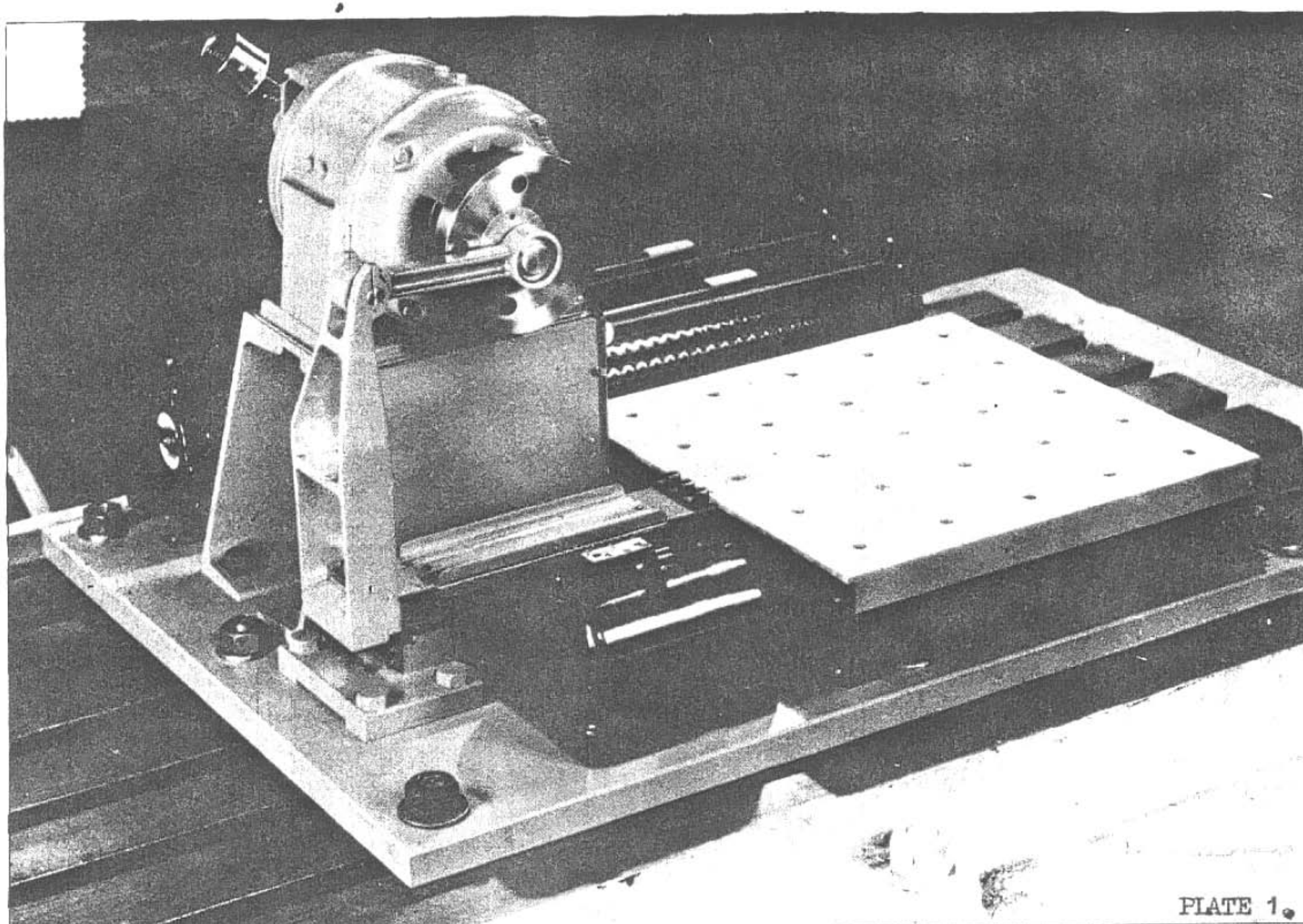

PLATE 1.

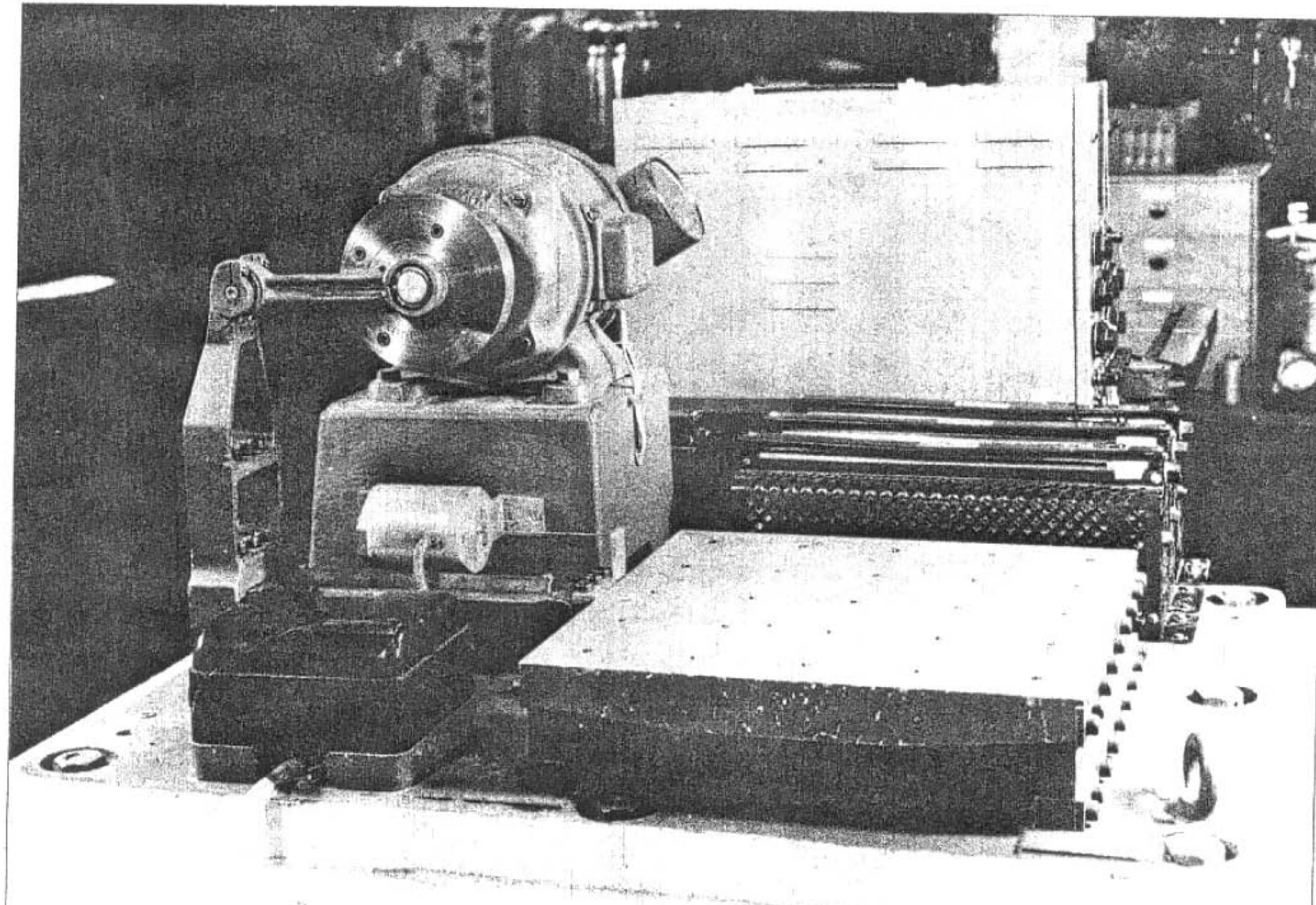

PLATE 2.

CARL SCHENCK MACHINE.

Horizontal Vibration. + 0.002"

Table unloaded.

Frequency c.p.s. | Velocity Waveform. | Acceleration Waveform. | Displacement Waveform.

20

50

80

100

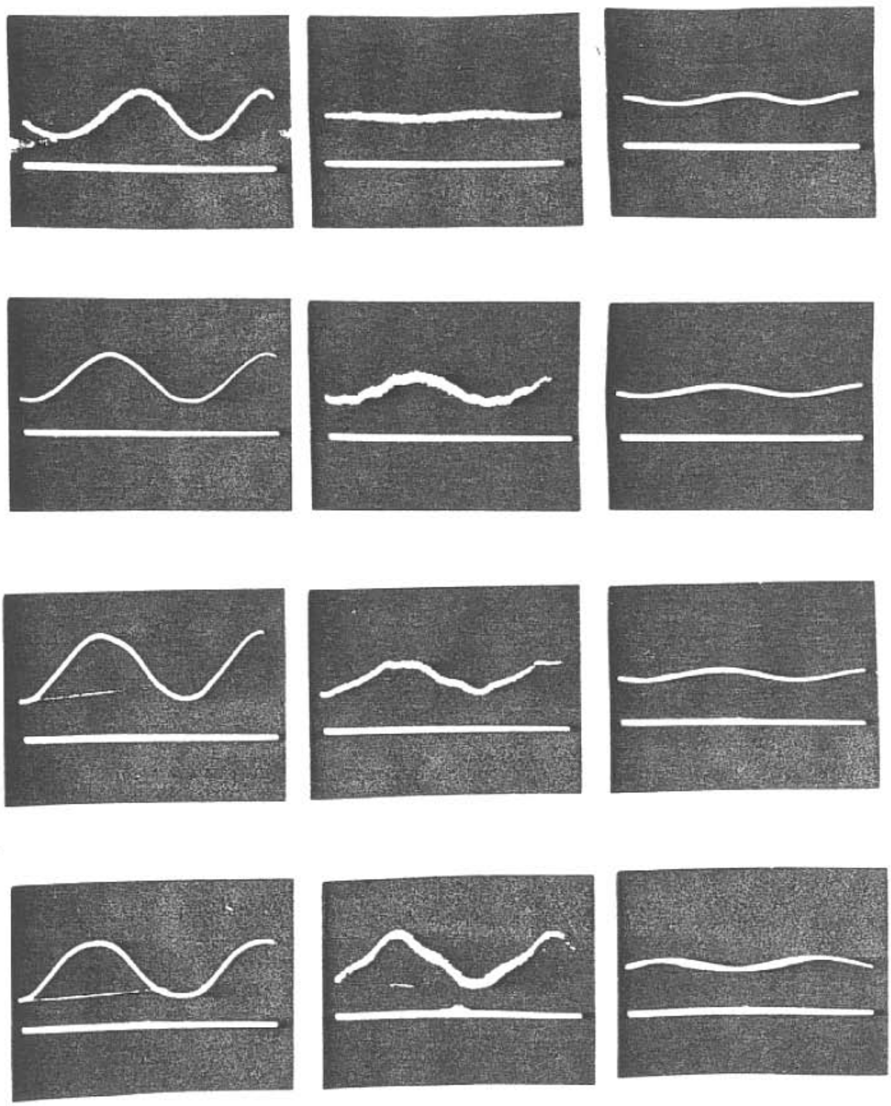

PLATE 3.

CARL SCHENCK MACHINE.

Horizontal Vibration. ± 0·002"

Table loaded 55 lbs.

Frequency c.p.s.	Velocity Waveform.	Acceleration Waveform.	Displacement Waveform.

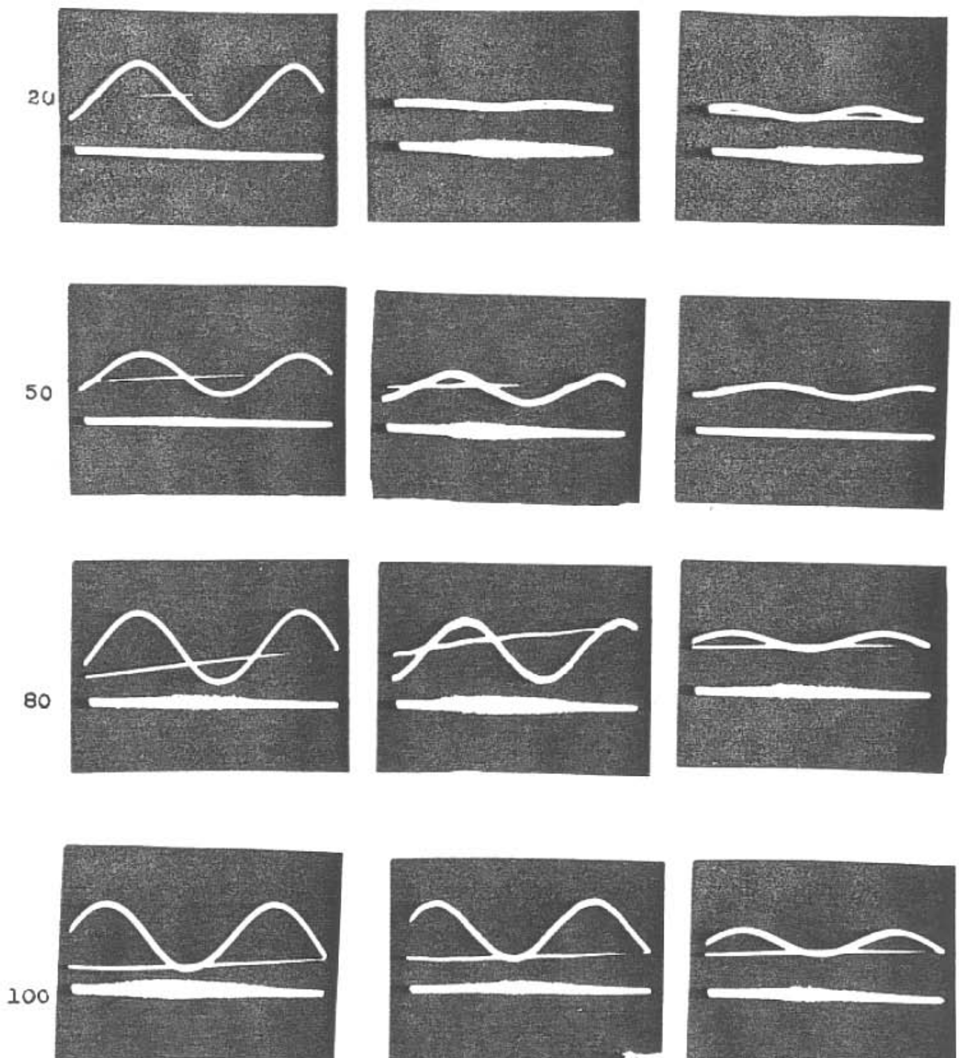

PLATE 4.

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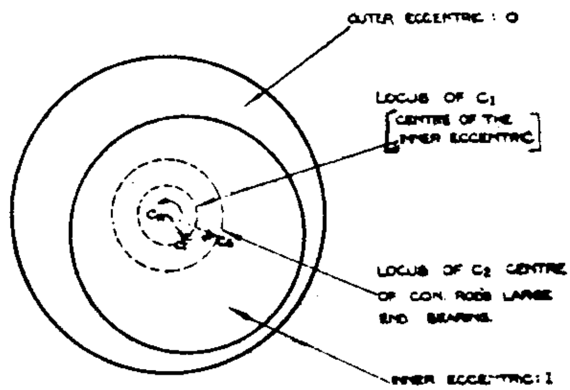

FIG. I

DIAGRAM OF THE ADJUSTABLE DRIVING ECCENTRIC

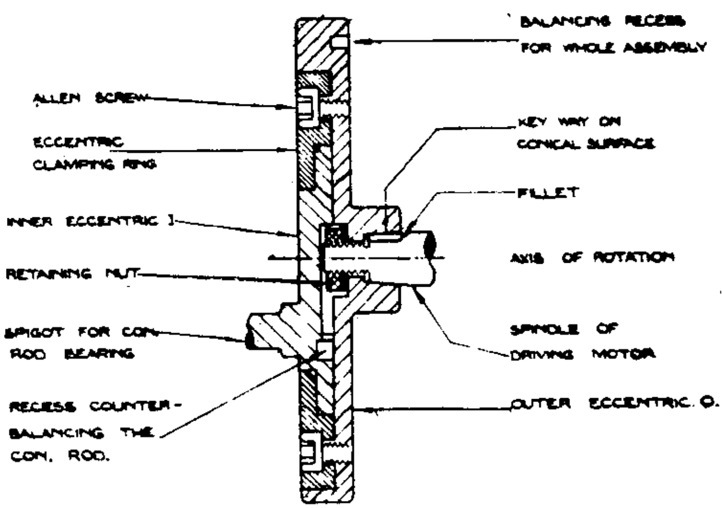

FIG. 2

CONSTRUCTION OF THE ADJUSTABLE DRIVING ECCENTRIC.

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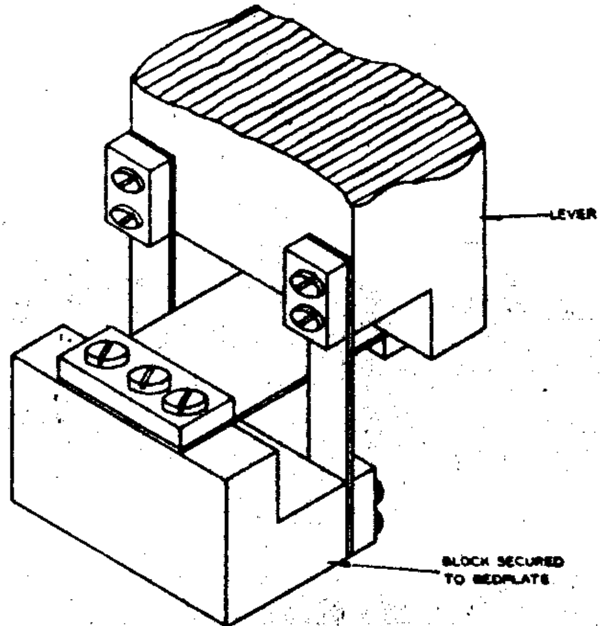

FIG. 3.
CONSTRUCTION OF THE LEVER SUPPORTING HINGE.

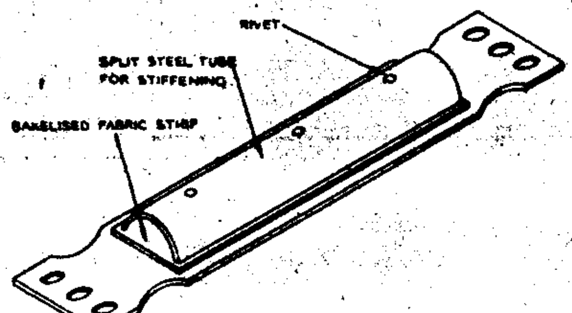

FIG 4.
IMPROVED TABLE CONNECTING ROD.

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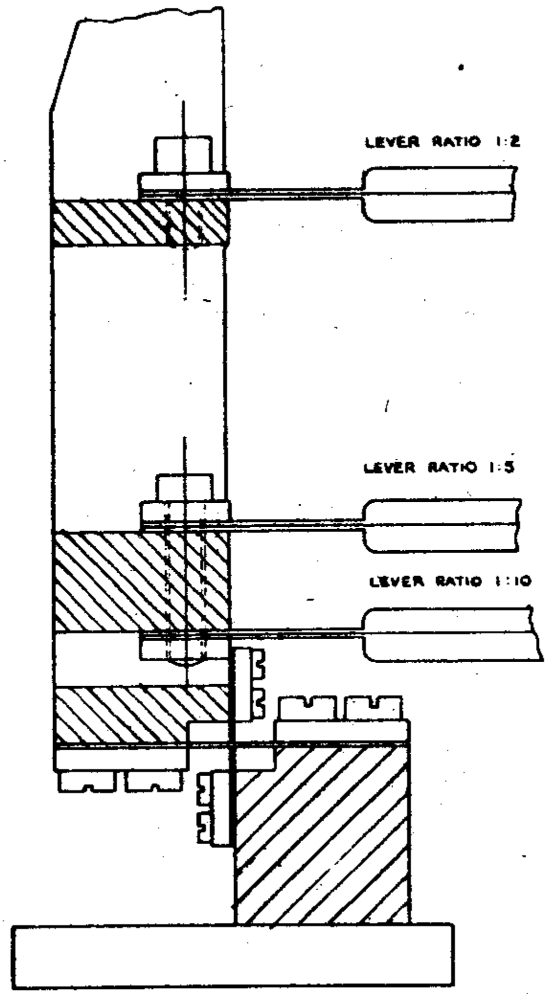

FIG. 5 TABLE CONNECTING ROD POSITIONS
FOR VARIOUS LEVER DISPLACEMENT RATIOS.

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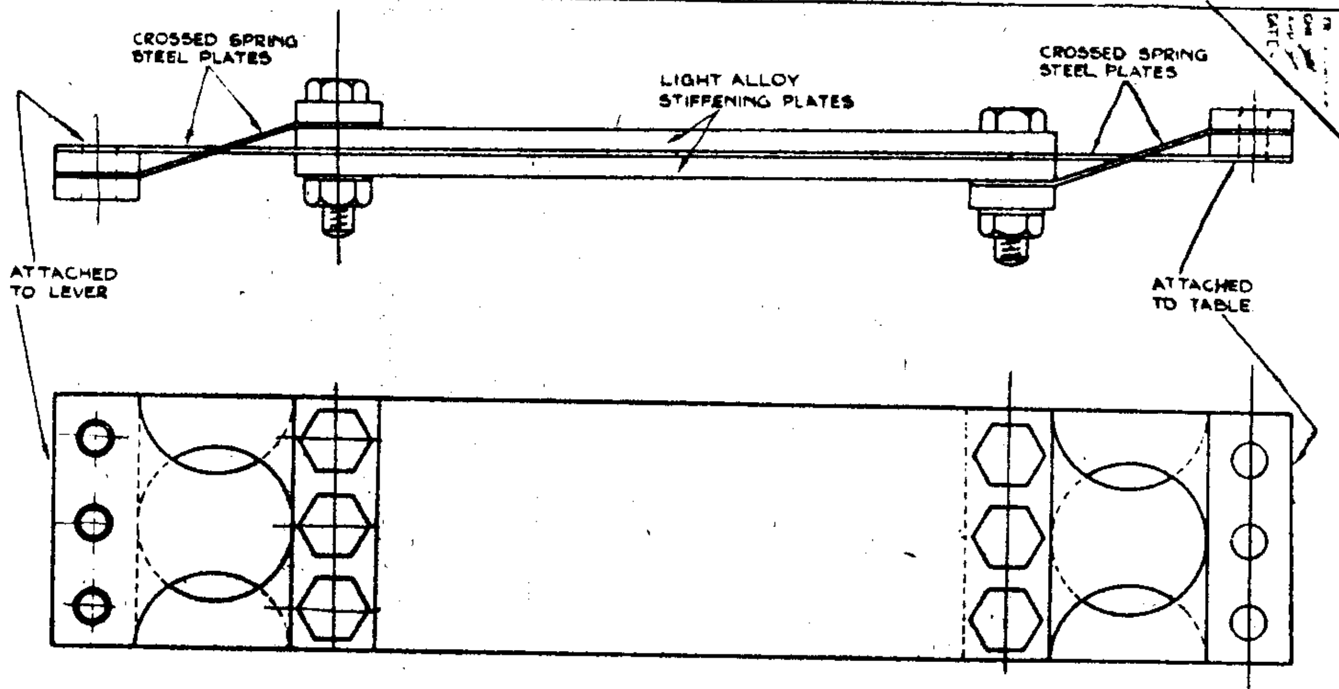

FIG.6. EXPERIMENTAL TABLE CONNECTING STRIP WITH INCREASED STIFFNESS.

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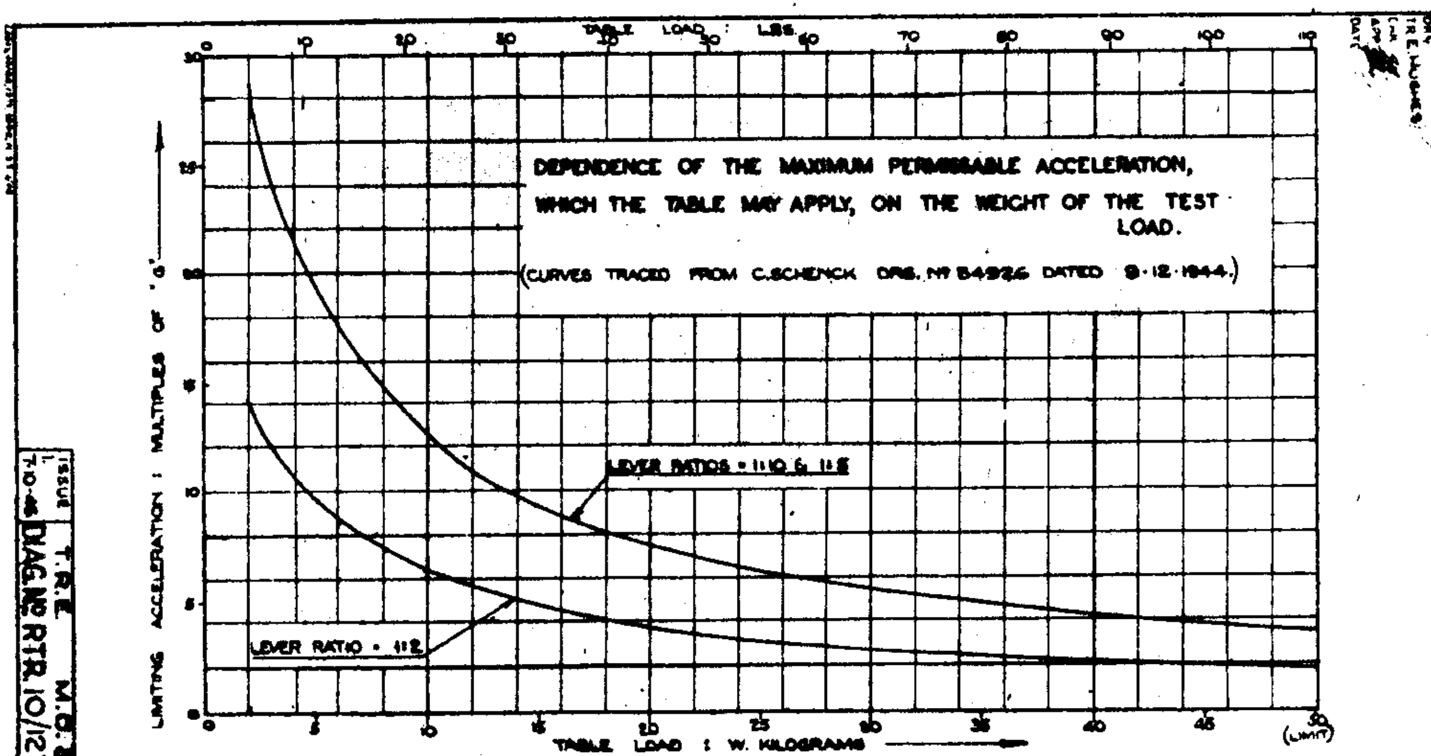
DEPENDENCE OF THE MAXIMUM PERMISSABLE ACCELERATION, WHICH THE TABLE MAY APPLY, ON THE WEIGHT OF THE TEST LOAD.
(CURVES TRACED FROM C.SCHENCK DRG. Nº 84926 DATED 9·12·1944.)
TABLE LOAD : LBS.
LIMITING ACCELERATION : MULTIPLES OF 'G'
TABLE LOAD : W. KILOGRAMS
(LIMIT)
LEVER RATIOS = 1:10 & 1:8
LEVER RATIO = 1:12
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FIG. 7.

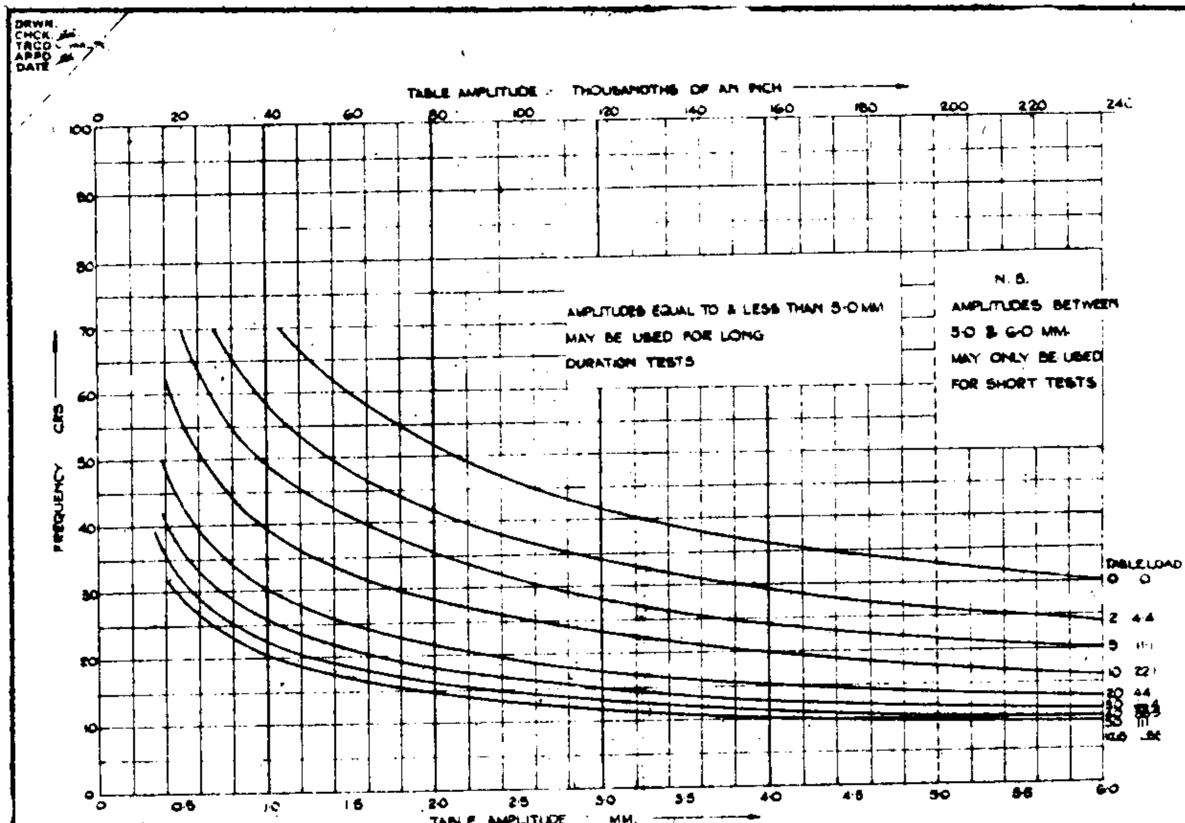

FIG 8

THE RELATION BETWEEN PERMISSIBLE TABLE· FREQUENCY, AMPLITUDE & LOAD FOR THE LEVER DISPLACEMENT RATIO 1:2

(CURVES TRACED FROM C SHENCK DRG. No 84926 DATED 11th DEC 1944)

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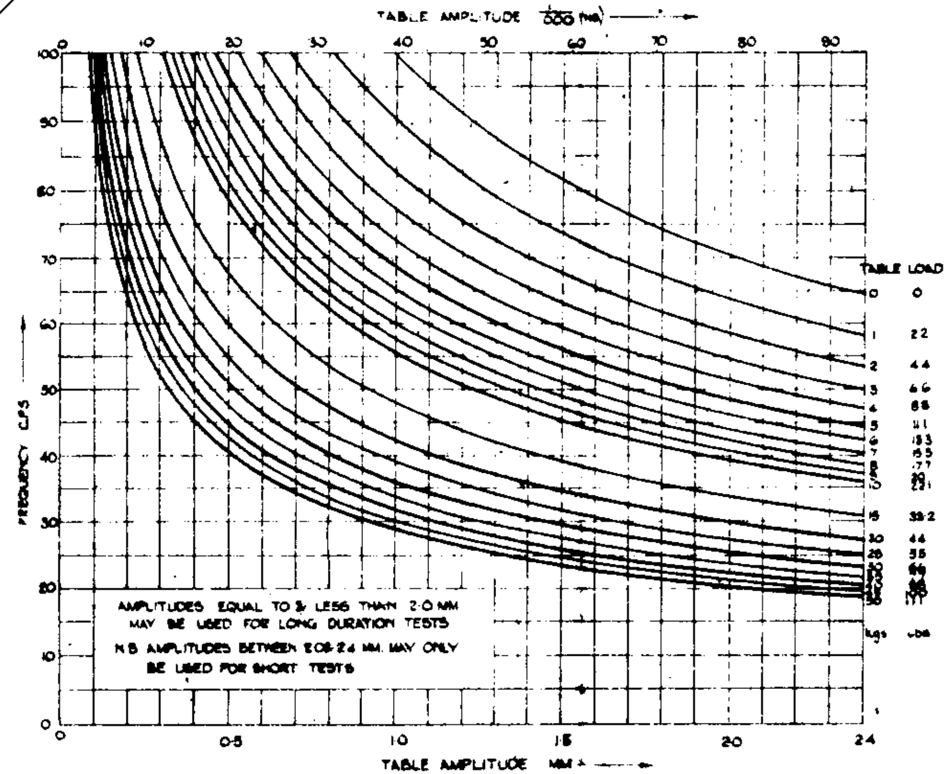

FIG. 9.

THE RELATION BETWEEN PERMISSIBLE TABLE FREQUENCY, AMPLITUDE & LOAD VALUES FOR THE LEVER DISPLACEMENT RATIO 1:5

(CURVES TRACED FROM C. SCHENCK DRG. No 85490 a DATED 7th DEC 1944)

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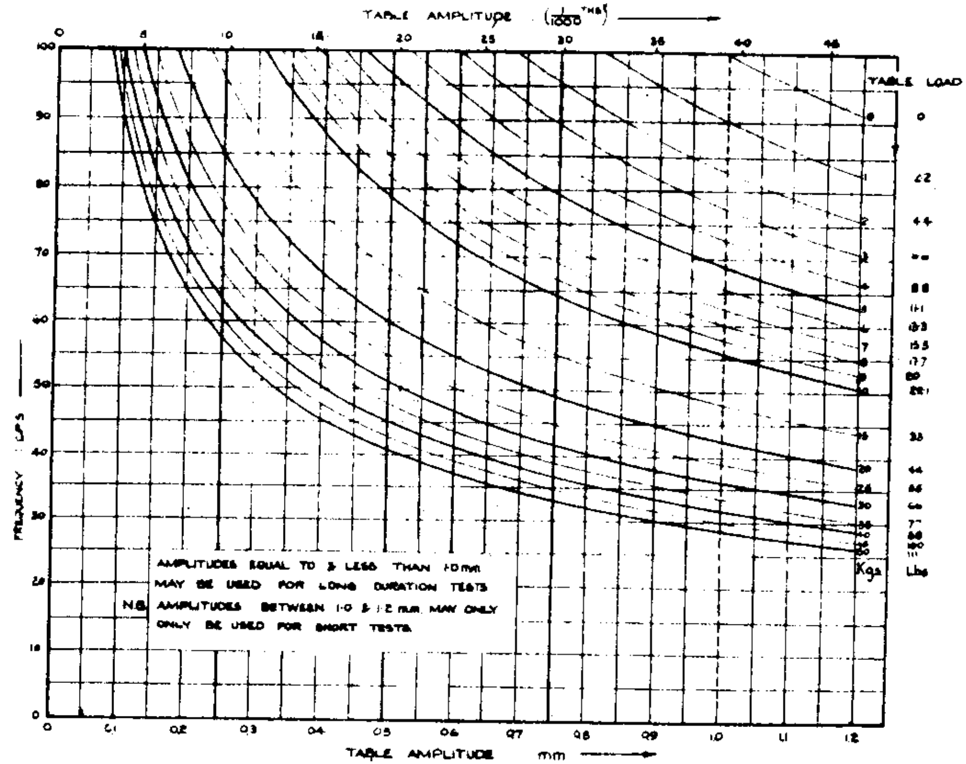

FIG. 10.

THE RELATION BETWEEN PERMISSIBLE TABLE FREQUENCY, AMPLITUDE & LOAD VALUES FOR THE LEVER DISPLACEMENT RATIO 1:10

(CURVES TRACED FROM C. SCHENCK DRG No. B5491a DATED 7TH DEC 1944)

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1 7·10·46	DWG. NO. R4 10/1282

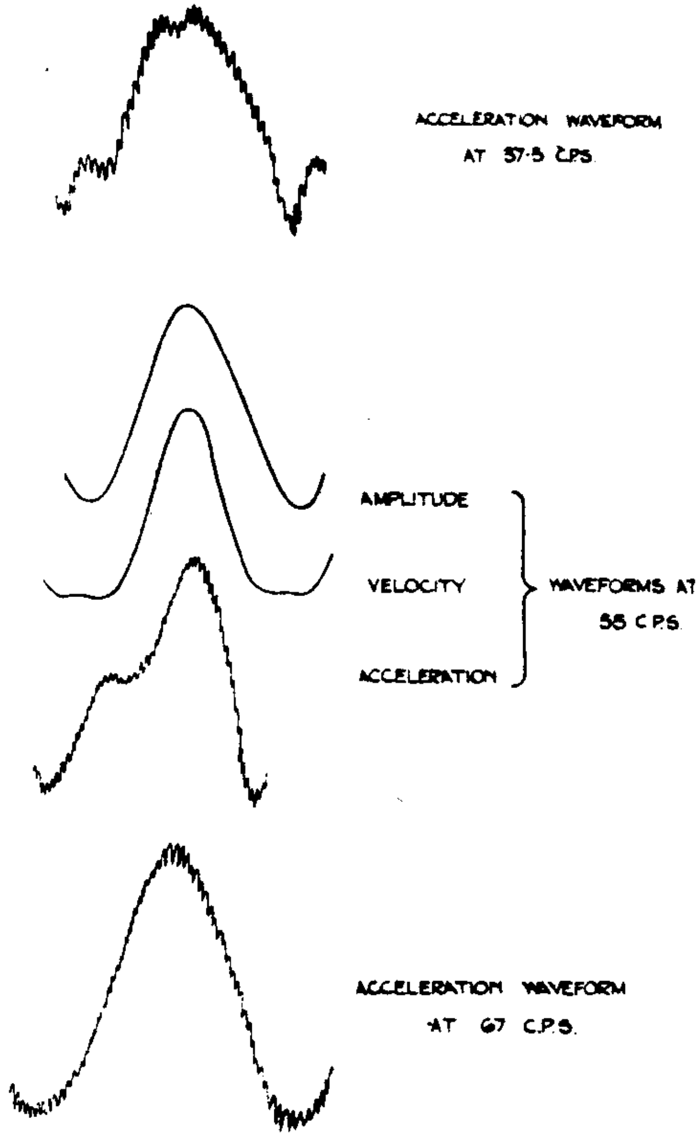

FIG.11

VIBRATION TABLE WAVEFORMS FOR LEVER RATIO = 2/1

LOAD 2Kgs ECCENTRIC SETTING = 0 24MM

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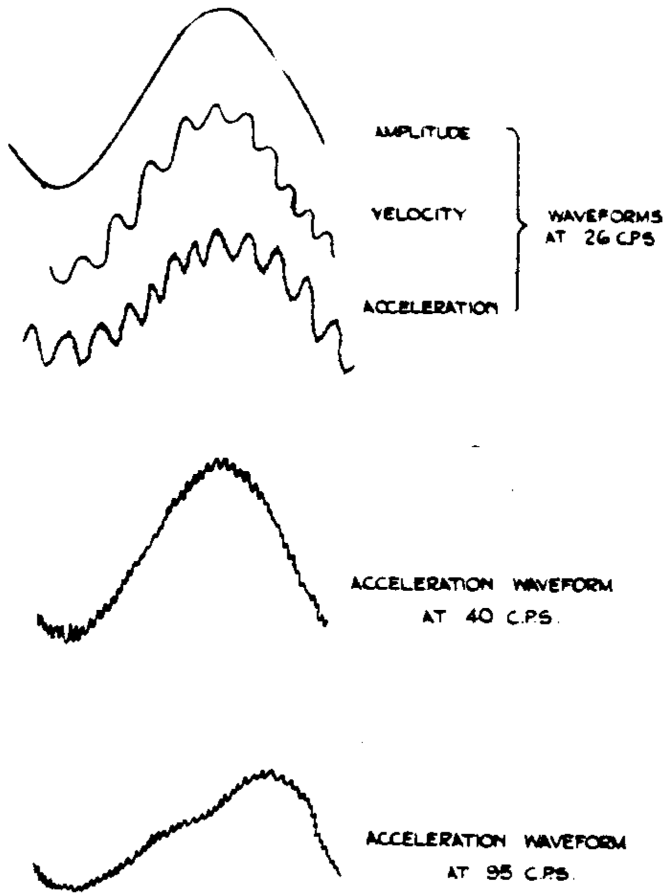

FIG. 12

VIBRATION TABLE WAVEFORMS FOR LEVER RATIO = 10/1

(LOAD 2Kgs ECCENTRIC SETTING = 1·2MM)

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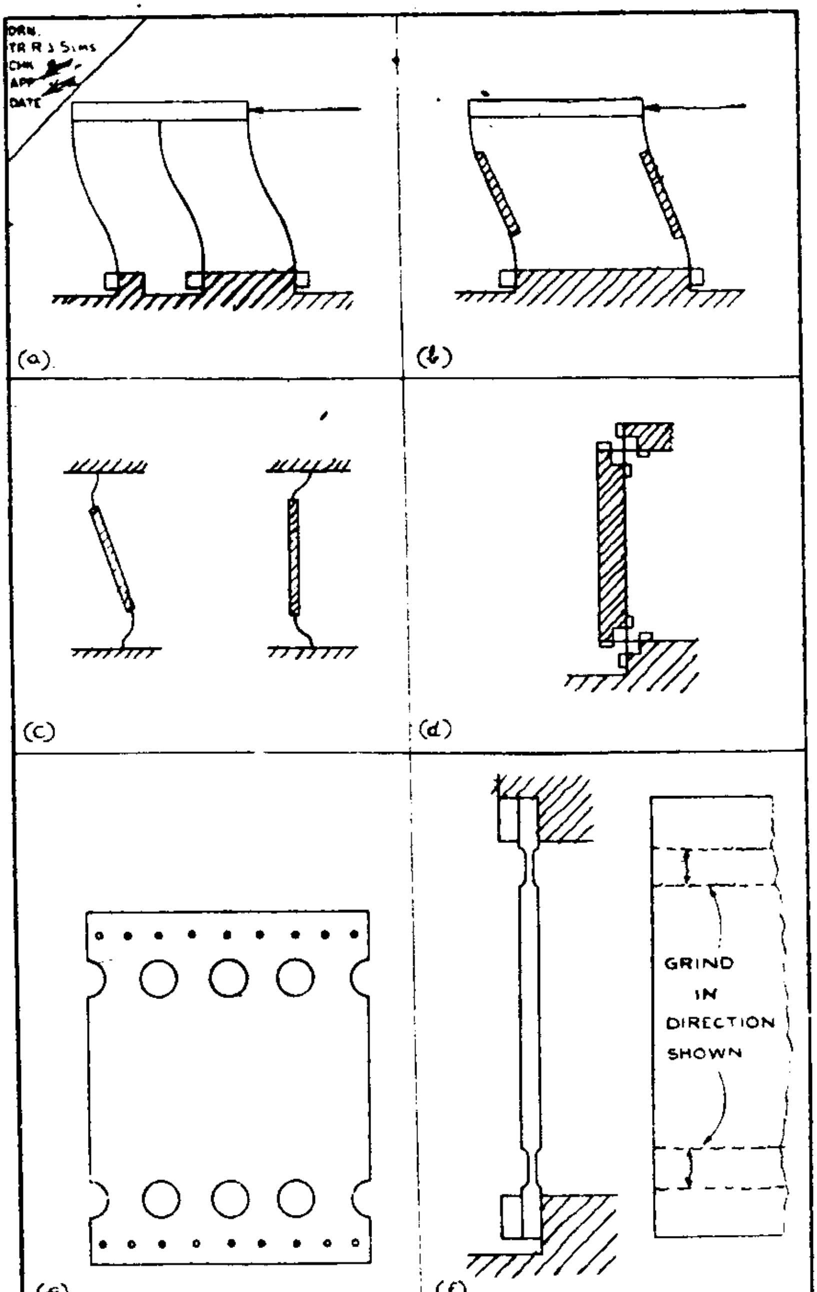

FIG 13.
METHODS OF TABLE SUSPENSION.

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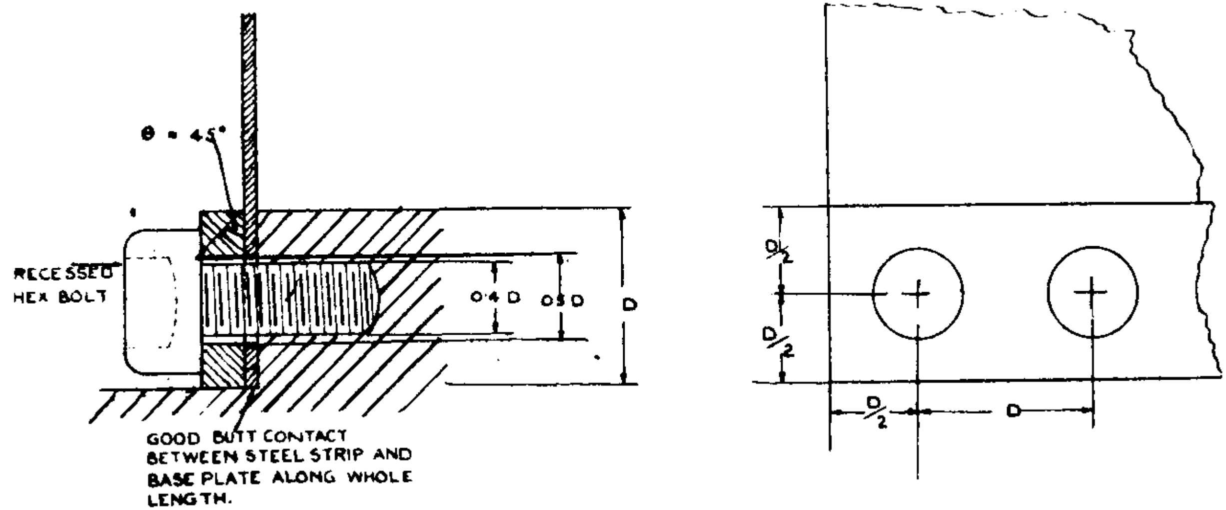

FIG. 14.

DETAILS OF CLAMPING PLATE FOR STEEL STRIP TABLE SUSPENSION.

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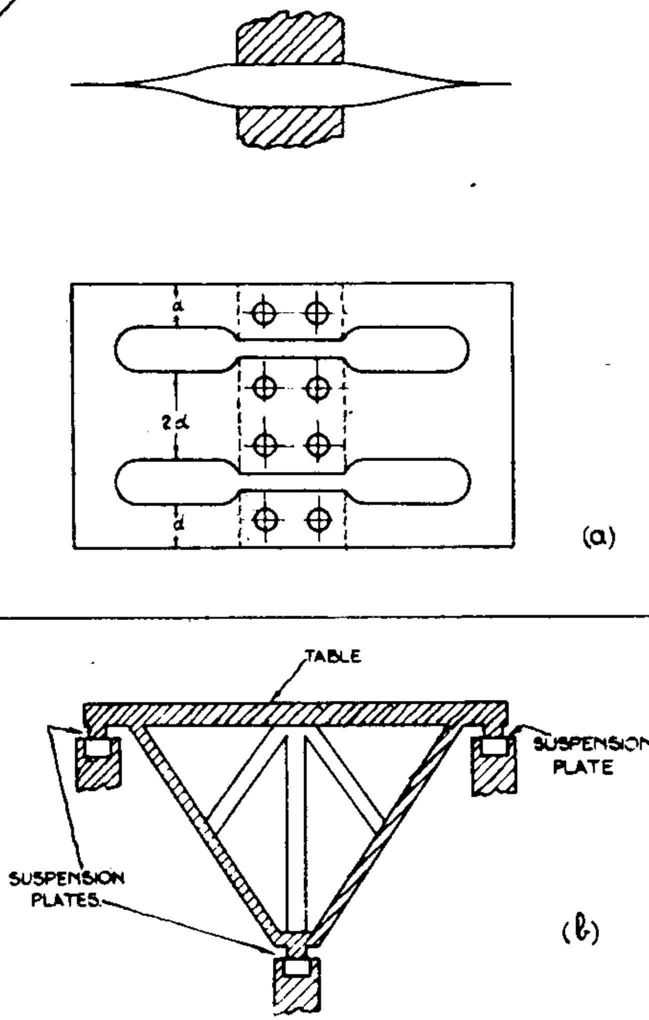

FIG. 15.

STEEL PLATE SUSPENSION FOR VERTICAL VIBRATION

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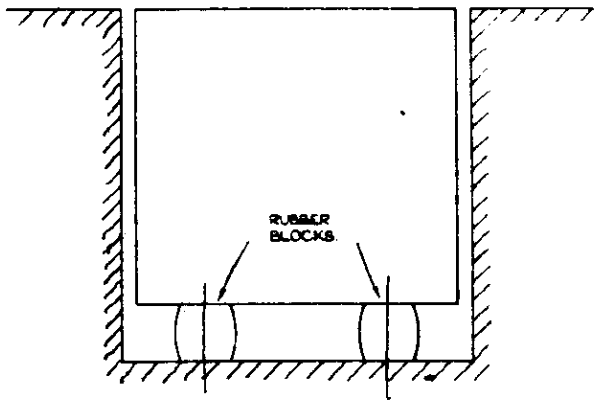

FIG. 16.
FOUNDATION FOR VIBRATION MACHINE ARRANGED FOR FLOOR LEVEL MOUNTING.

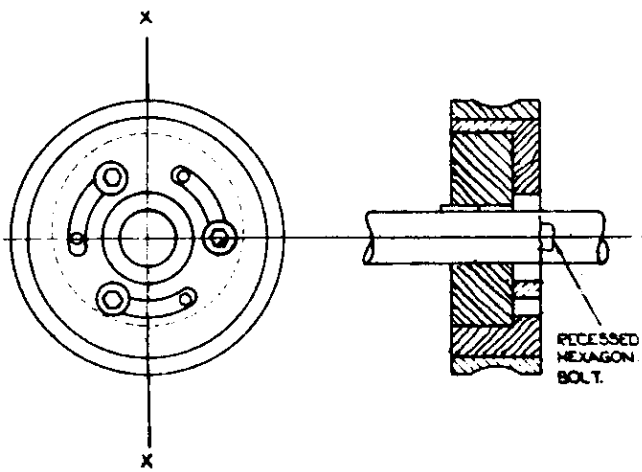

FIG. 17.
DOUBLE ECCENTRIC FOR USE BETWEEN TWO BEARINGS.

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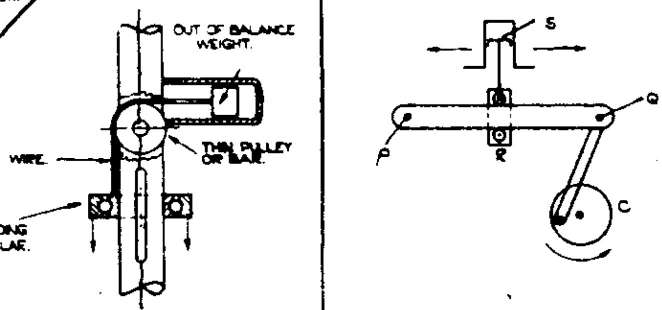

FIG. 18.
TWO DEVICES FOR VARIATION OF TABLE AMPLITUDE WHILST OPERATING.
(DUE TO DR. PETERSEN.)

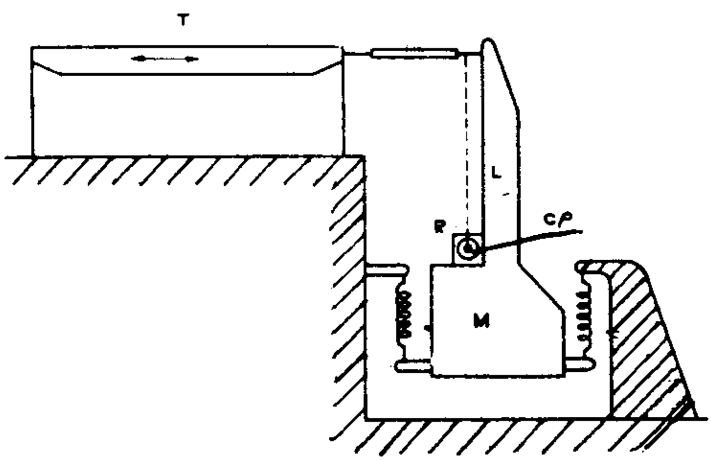

FIG. 19.
METHOD OF VARYING TABLE AMPLITUDE WHILST OPERATING.
(DUE TO DR. FEDERN.)

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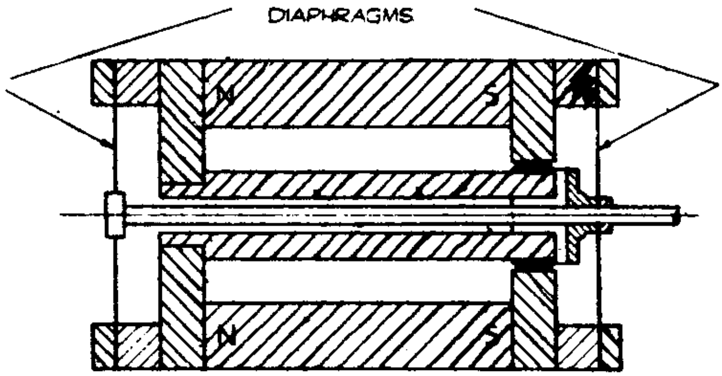

DETAIL
OF
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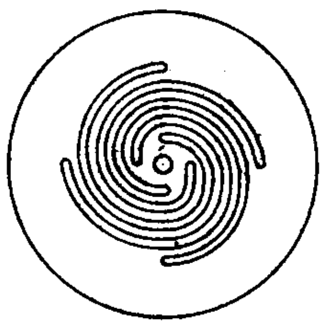

FIGURE 20
DIAGRAM OF MOVING COIL
VIBRATION PICK-UP.

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