DYNAMIC BALANCING MACHINES FOR LARGE ROTORS.

In this age of high-speed rotating machinery the necessity for dynamical balancing is becoming ever more insistent, not only in the interest of smooth running, but also because vibration is inimical to the safety and life of such machinery.

Notwithstanding the care exercised in the selection of materials and in manufacturing processes, rotors occasionally suffer disastrous "failures," and it is a possibility that such failures may partly be due to stresses arising from out-of-balance.

It is interesting to reflect on the magnitude of the centrifugal forces which result from small amounts of out-of-balance.

An out-of-balance of only 4 ozs. at a radius of 3 feet, in a rotor revolving at 3,000 r.p.m., produces a centrifugal force of over 1 ton, and this force is operating radially at every point round the circle 3,000 times a minute. (Fig. 1.)

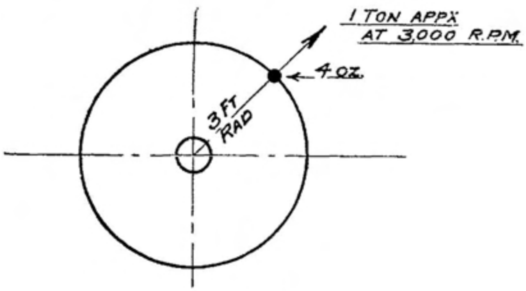

FIG. 1.

Imagine that a second out-of-balance of the same amount exists in a rotor, and that they are disposed at the two ends of the rotor as shown in Fig. 2, *i.e.*, at opposite corners of an axial plane.

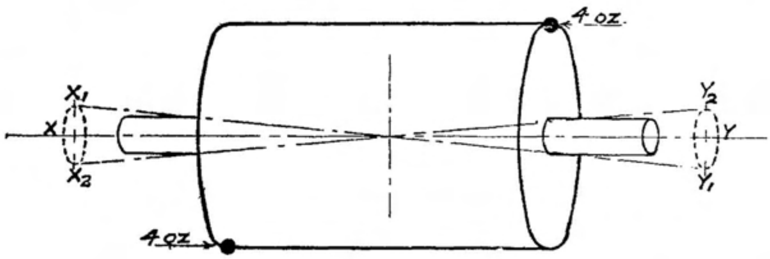

FIG. 2.

Another centrifugal force of 1 ton is now operative, and the two together form a couple which is constantly tending to swirl the axis of the rotor out of its normal position XY into positions X_1, Y_1, X_2, Y_2.

Incidentally the diagram (Fig. 2) may be used to illustrate in a simple manner the well-known fact that a rotor in *static* balance is not necessarily in *dynamic* balance also.

Assuming that the rotor, exclusive of the two 4 oz. weights shown, is in perfect balance in every respect, these weights affixed at positions indicated would not affect the static balance at all, but dynamically they would subject the rotor to a couple of 1 ton multiplied by the length of rotor between the weights.

For high-speed rotors of any considerable axial length, in relation to their diameter dynamic balancing is essential, and particularly so where the rotor is not composed of separate wheels or sections which can be tested separately before assembly.

In the course of Admiralty investigations into the practice of leading British engineering firms as regards the dynamic balancing of large turbines and other rotors, some points have been noted which may be of general interest.

There are extraordinarily few Dynamic Balancing Machines on the market which are capable of dealing with rotors up to say 15 tons or over, *i.e.*, machines which have definitely passed their experimental stage and have been standardised for manufacture and sale like any other machine. These machines are :—

> The Lawaczeck-Heymann Patent Dynamic Balancing Machine made by Carl Schenck Ltd., Darmstadt ;
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> The Akimoff Patent Dynamic Balancing Machine made by the Vibration Specialty Co., of Philadelphia, U.S.A.; and
>
> The Westinghouse Patent Dynamic Balancing Machine made by the Westinghouse Electric and Manufacturing Co., Philadelphia.

Some British firms, and individuals, have carried out, and are carrying out, research work with the object of evolving a satisfactory means of dynamically balancing large rotors. Messrs. Laurence Scott of Norwich have made a few machines for rotors up to 3 tons—one of which was ordered by the Admiralty in 1920 for Portsmouth Dockyard.

The Dynamic Balancing Machine devised by Mr. J. J. King-Salter, R.C.N.C., the late Deputy Director of Dockyards, in which the rotors of the Australian Cruiser "Adelaide" were balanced at the Cockatoo Island Naval Dockyard in 1920, also an improved type of this machine now being manufactured for Portsmouth Dockyard are in mind, but the fact remains that commercially the field is at present occupied by the three machines mentioned above.

The Schenck machine is gradually being adopted in this country, in the U.S.A., and in Japan. In Germany it is installed in many of the more important Engineering Works.

The Dynamic Balancing Machines for the heavier class of rotors being more particularly under review, these notes are not intended to deal in detail with the smaller Dynamic Balancing Machines of which there are a few on the market, but it may be of interest to mention them, if only to emphasise the limited choice available even in the smaller machines, which comprise :—

The " Norton " made by the Norton Emery Wheel Co., Worcester, U.S.A.

The " Gisholt " made by the Gisholt Machine Co., Wisconsin, U.S.A.; and

The " Olsen-Carwen " made by the Lippencott-Carwen Corporation, and factored in this country by Messrs. Edward G. Herbert of Levenshulme, Manchester.

Several of the " Norton " machines have been installed in H.M. Dockyards for many years.

Certain *Static* Balancing Machines have been devised, and particular mention is made of that associated with the name of Mr. W. H. Martin and manufactured by Messrs. Joshua Buckton & Co., but this type of machine does not strictly come within the scope of the subject now under consideration, as it is not a *Dynamic* Balancing Machine.

Further, there are " Over-speed " Testing Machines for turbine rotors in use by certain firms, but these machines are not for testing dynamical balance, but for subjecting the material of the rotors to stresses in excess of those they will experience in service conditions, in order to detect possible weakness in the material of the rotor. The latter is speeded up from 20 to 25 per cent. above maximum service speed in a machine placed in a dug-out or isolated building, suitably surmounted by sandbags and ballast, &c. This test is carried out before rotors are bladed.

The question naturally arises as to why there are so few types of Dynamic Balancing Machines obtainable.

Small Dynamic Balancing Machines have been in use for many years, and were developed to deal with small rotors, armatures, and other high-speed rotating parts. The problem of securing proper balance first became acute in this field, and particularly with high speed rotating bodies which worked with their axes vertical, such as centrifugal milk separators, oil-separators and purifiers, and vertical motors.

The provision of a Heavy Dynamic Balancing Machine, capable of dealing with large rotors, such as are in common use to-day, is a very costly proposition, taking into consideration not only the machine itself, but also the powerful motor and control arrangements required, the substantial foundations, and floor-space served by a suitable overhead traveller.

In these circumstances it is to be expected that on the one hand the turbine manufacturer requires very definite conviction as to the indispensability of these machines before sinking the necessary capital, and on the other hand the would-be inventor and machine tool manufacturer have little hope of early return of capital expended in costly research and experimental work.

Many firms of the highest standing in this and other countries have hesitated to adopt dynamic balancing machines because of their confidence in the quality and homogeneity of the metals employed, the high-class workmanship, the care exercised in the blading of the rotor, and the static balancing of the rotor, followed by the testing of the turbines under steam in the shop, during which the behaviour of the turbine as regards vibration is carefully watched.

More or less improvised means have been adopted in the past in many large Engineering Works by which any material dynamic out-of-balance could be detected, whilst the turbines were being run under steam, and it was possible, by trial-and-error methods to eliminate the out-of-balance so as to eventually arrive at a condition where various rough and ready tests demonstrated the absence of any material vibratory forces. When this stage was reached it was considered there could not be anything radically wrong with the balance, and on the whole subsequent experience confirmed the verdict. It must be admitted, however, that these methods were too crude to be regarded as a really satisfactory engineering proposition.

With the advent of Curtiss, and other impulse type turbines, the dynamic balancing machine cult received a further set back, because each wheel could be statically balanced before being threaded on the shaft. The manufacturer then had some justification for assuming that the assembled rotor could not reasonably be materially out of balance, and in any case the steam test in the shop was still applied to determine whether any noticeable dynamic out-of-balance existed or not.

Other reasons of a practical nature have also operated to arrest the adoption of elaborate Balancing Machinery, chief of which were the expansion and distortion of parts which take place whilst the turbine is being heated up, particularly when highly superheated steam is employed.

It was therefore regarded as futile to dynamically balance rotors when cold, to fine degree of accuracy, in view of such distortion of parts, together with the slight blade displacement which invariably occurs when first run up to speed.

There is some truth in these contentions, but the need for dynamic balancing still remains. Some makers urge that turbine rotors should be dynamically balanced both before and after the turbine has been run at its maximum speed, under steam.

In any case it may be stated with certainty that the dynamic balancing machine is rapidly gaining in favour in the large Engineering and Electrical Firms of Great Britain, Germany, and U.S.A.

Germany has made greatest progress to date in the development of the large type machines on a commercial basis, and the particular type which has met with greatest success so far is that based on the original design of Dr. Lawaczeck in 1908, as simplified and commercialised by Dr. Heymann of Darmstadt University about 1917. This machine is that manufactured and sold by the Firm of Carl Schenck of Darmstadt.

There are over 20 of this design of machine in this country at the present time, and over 200 in Germany, U.S.A., Japan and Italy. No apology is therefore tendered for proceeding to describe this particular machine in some detail.

The Schenck Machine is of very practical design. The rotor to be tested is supported in bearings which are mounted on two vertical "leaf springs," one at each end, as shown at (*A*) in Fig. 3.

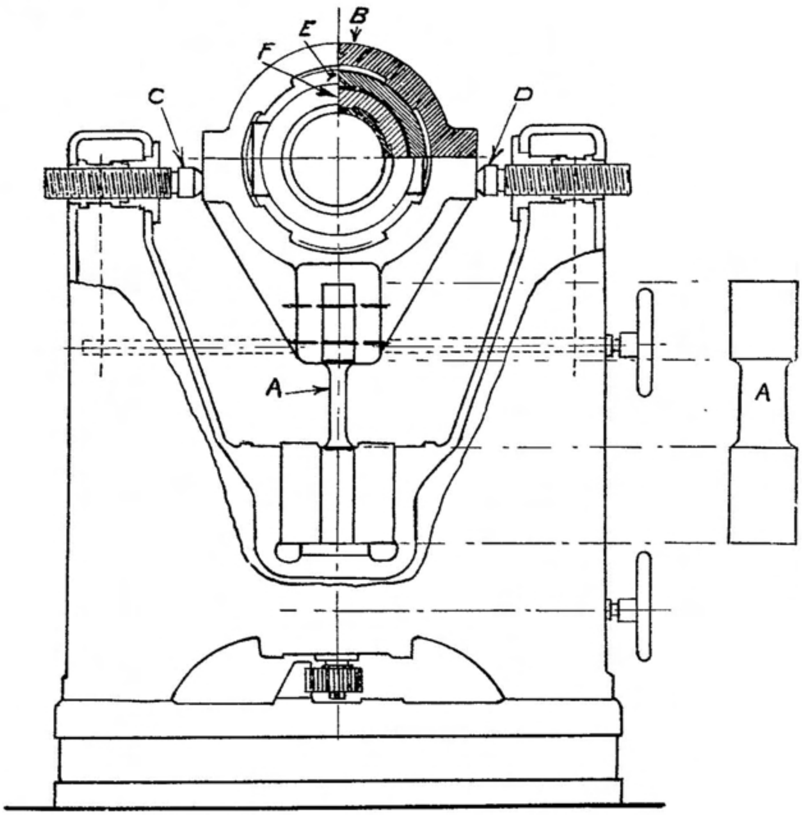

Fig. 3.—Schenck Machine—Diagram illustrating Leaf Spring at *A*.

These "leaf-springs" are of very special steel accurately machined and rigidly secured at the top to the bearings taking the rotor spindle, and at the bottom to the base of the machine.

They are of very substantial rectangular section as may be gathered from the diagram, and are removable, several pairs of varying section being provided with each machine to suit different weight rotors.

The tendency of any out-of-balance of the rotor when revolved is to sway the top end of the leaf-springs to right and left alternatively. If the rate of revolution can be arranged to cause these lateral impulses to correspond with the natural period of vibration of the spring, very considerable oscillation of the top of the spring to right and left will occur.

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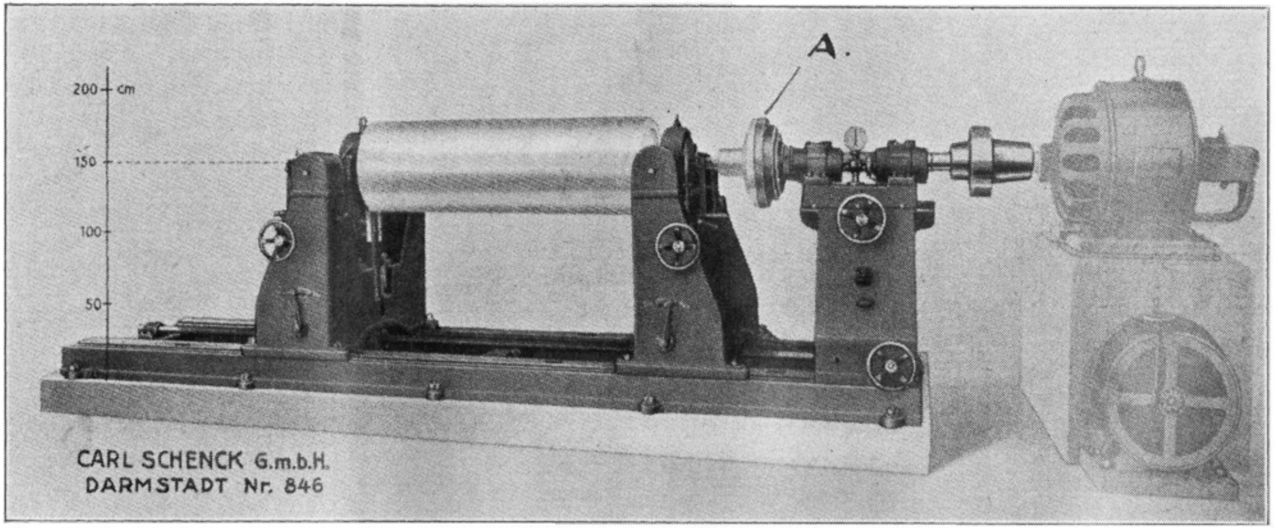

FIG. 4.

This is the basic idea on which the Schenck Machine is designed. The principle is old and was enunciated by Dr. Stodola in connection with steam turbine rotors many years ago.

The rotor under test is gradually speeded up to about 700 or 800 revs. per minute. During this period "stops" (C) and (D) are engaged which prevent any movement of the springs at all, *see* Fig. 3.

Then the motor is entirely disconnected from the rotor under test, the drive being transmitted through a special magnetic clutch (A) to enable this to be readily effected, *see* Fig. 4.

The stops are now disengaged, a limited extent, but sufficiently to allow a reasonable amount of free vibration of the spring under bearing (X) leaving the other one (Y) still restricted, observing that the bearings on which the spindle is carried have spherical external seatings, hence either end of the rotor can oscillate laterally, whilst the other end is restricted, *see* Figs. 5 and 6.

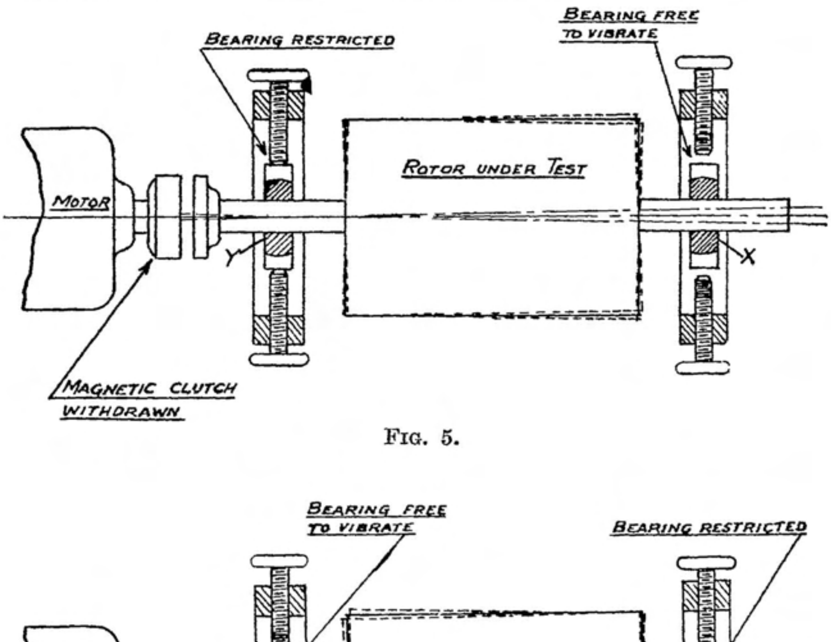

Fig. 5.

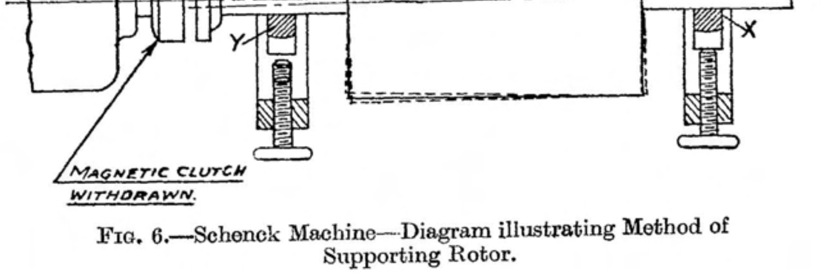

Fig. 6.—Schenck Machine—Diagram illustrating Method of Supporting Rotor.

As soon as the motor is disconnected the rotor runs entirely free, and its speed gradually coasts down till it reaches a point when it momentarily causes the spring to "shudder." Both before and after this critical period there is usually very little vibration of the leaf springs.

The amplitude of the vibration or "shudder" is recorded as an arc on a flat piece of paper held in the frame shown at (*A*), Fig. 7. The movement of the pencil can be readily adjusted to equal the amount of the springs or to multiply such movement, and it is commonly about 2 inches overall to start with and finishes up with a mark of less than a $\frac{1}{16}$ inch amplitude if desired.

The amplitude of vibration as recorded on the diagram enables the operator to determine the weight required at a given radius to eliminate this vibration, but a little experience is necessary before the correct "counterweight" can be arrived at without a succession of trial runs.

The position around the circumference at which the "counterweight" has to be fixed is found by running the rotor whilst a special scribing instrument as illustrated at (*B*), Fig. 7, and in detail in Fig. 8, marks the shaft both before and up to the maximum vibration of the spring. The instrument is set with the parallelogram frame in the horizontal position, and means are provided so that frictional resistance to movement of the pencil away from the rotor can be adjusted.

When the scriber is first applied the rotor shaft may be running with so little oscillation that a mark is then made nearly all round the shaft, but as the rotor speed reaches that at which the spring commences to vibrate, the shaft in rotating and oscillating, gradually pushes the scriber away from its original setting, and at the same time causes the point to move slightly in the axial direction of the rotor due to the parallelogram frame construction, *see* Fig. 8.

Consequently a series of ever-shortening arcs are made on the rotor spindle, the last mark made being that when the shaft is at its maximum oscillation.

In order to eliminate the effect of "lag" the rotor is now revolved in the opposite direction and another series of curves is recorded by the scriber. From these two markings the position at which the counterweight requires to be affixed can be determined.

The first stage of the operation is now finished, the counterweight is temporarily affixed, and the same procedure is carried out with the other end of the rotor, the completed end now being "restricted," in its bearing (*see* Fig. 6). It will be noted that the rotor does not need to be lifted out and turned round, but remains in its original position in the bearings. A repeat run of the first end is then made to eliminate the effects of any alteration made to the second end, before finally trying the rotor with both springs unrestricted, when it will usually be found

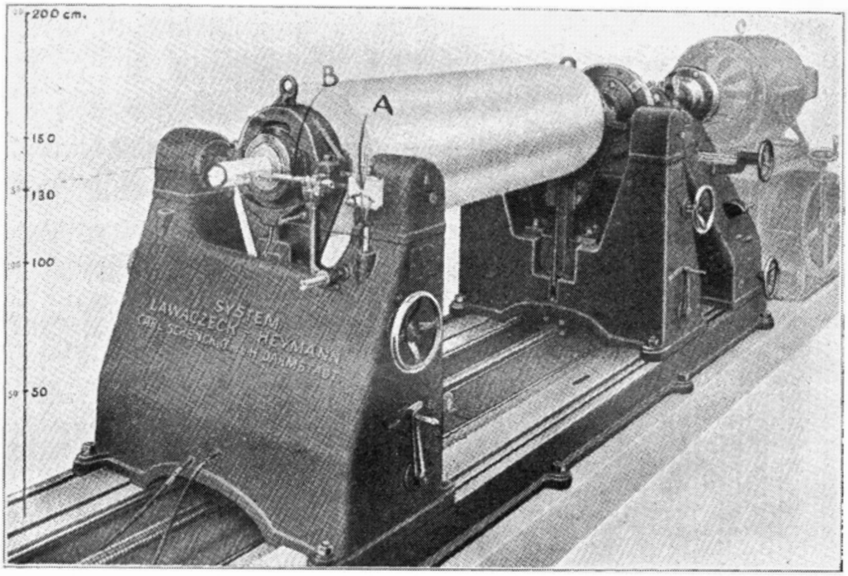
200 cm.
150
130
100
50
B
A
SYSTEM
LAWACZECK - HEYMANN

FIG. 7.

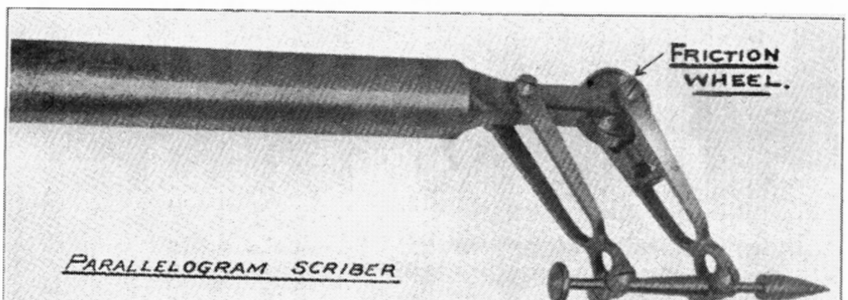
FRICTION
WHEEL.
PARALLELOGRAM SCRIBER

FIG. 8.

that the rotor will coast down through the resonance period without appreciable tremor. It is very rare that more than three temporary "counterweights" are required to render the rotor satisfactory.

The small "counterweights" are only used for convenience to arrive at a state of balance. Their amount and position being known the balance is effected in practice by removing equivalent metal from the rotor, or adding thereto if the former alternative is impracticable.

Foundations.—The foundations of this and other large dynamic balancing machines require to be very substantial. Four or five feet depth of concrete is usual, depending on the nature of ground and the size of the machine.

Magnetic Clutch.—The means for giving the rotor the "spin" to get it above the resonance speed requires to be such that the driving gear is entirely disconnected from the rotor under test. In the Schenck Machine this is effected by a magnetic clutch between the motor and the rotor.

All parts engaged in speeding up the rotor, including the driving motor itself and the magnetic clutch, are themselves dynamically balanced to avoid any possibility of these parts affecting the rotor under test.

Lubrication.—The proper lubrication of the two bearings in which the rotor under test revolves is important, and an auxiliary motor-driven pump is fitted in the bed of the machine to supply forced lubrication. In the large machine this motor can also be used for traversing the heads along the bed to set them at any particular position.

Bearings.—These are composed of three main parts (*see* Fig. 3), viz., the outer housing (B), the spherical-seated intermediate portion (E), and the inner shell (F), with its white metal lining. The reliability of the readings is enhanced if the working clearance of these parts is kept at lowest practicable limits.

Windage.—The power absorbed in windage when revolving large rotors with the concave side of the blades against the air is very considerable, and for machines of 15 ton capacity 250 to 300 horse power is required.

Such a rotor under test creates no inconsiderable breeze, and for this reason it is advisable to site the machine where this will not cause inconvenience or disturbance of work, and if necessary to erect a screen.

It is therefore evident that large machines require to be equipped with very powerful motors. The controllers also require to be of suitable type for frequent stopping and starting.

After experience it is possible to dispense with the rotation of the rotor in the astern direction as well as in the ahead direction in order to locate the circumferential position of unbalance, *so that it is not absolutely essential to run the* *rotor in both directions.* It greatly facilitates finding the required position, however, and in most cases is of great assistance.

Large Propellers.—The comparatively high speed at which the Schenck Machine is designed to operate is an advantage in one respect, as it raises the order of the centrifugal forces due to out-of-balance and renders their calibration easier, but high speed of revolution is not favourable when dealing with the larger diameter propellers, as the power absorbed in windage and the air disturbance caused are so considerable.

Degree of Sensitiveness.—The Schenck Machine for rotors of 1 to 15 tons is guaranteed by the makers to detect out-of-balance amounts at either bearing as follows :—

Rotors of 1 to 3 tons	- -	6 grammes	at 1 metre	radius.
,, 3 to 6 ,,	- -	10	,,	,,
,, 6 to 9 ,,	- -	15	,,	,,
,, 9 to 12 ,,	- -	20	,,	,,
,, 12 to 15,,	- -	30	,,	,,

(30 grammes = 1 oz. approx.)

Rotor Spindles.—It is essential that the spindles of rotors under test shall be within fine limits of accuracy, to avoid false vibratory readings, and to obtain satisfactory results in this, or any similar type of dynamic balancing machine, a tolerance from true cylinder of not more than half a thousandth of an inch on any diameter between 6 inches and 10 inches is advisable.

Method of Supporting Rotor.—In the Schenck Machine the "leaf-springs" supporting the rotor are in compression. In the Westinghouse Dynamic Balancing Machine these springs are arranged to be in tension, thus enabling springs of less cross sectional area to be employed. The general construction, however, is somewhat complicated thereby.

In consequence of the much lighter springs employed in the Westinghouse Machine the designed resonance speed averages about 200 r.p.m., as against the average resonance speed of about 600 r.p.m. which obtains in the Schenck machine.

The lower speed is advantageous as it entails less driving power, and is helpful when dealing with propellers, but on the other hand it is not so sensitive in detecting small out-of-balance amounts. This type of machine with springs in tension has advantages in dealing with rotors of exceptional size and weight.

Axial thrust of rotors under test.—Special means are provided in some machines to take the axial thrust of bladed turbines, due to the windage when under test.

Testing large fans.—Large fans such as those used for ventilating coal and other mines have been dynamically tested with the eye of the fan and the peripheral opening securely covered in with thin material such as canvas to avoid "windage."

Exact points of out-of-balance not located.—It is necessary to realise that dynamic balancing machines do not pretend to locate the precise point, or points, at which there is an out-of-balance in a rotor.

They deal only with the resultant centrifugal forces operating at the bearings, and afford the means of ascertaining the magnitude and position of counterweights necessary to neutralise such forces.

Trial and error methods.—Although the Schenck Machine is based on sound principles it is open to the criticism that the *modus operandi* of arriving at a state of dynamic balance entails a considerable number of runs, and depends upon trial-and-error methods to determine the exact " counterweights " required.

It is therefore interesting to know that the original Lawaczeck Machines embodied devices of a highly scientific character with the object of avoiding trial-and-error methods, and that these have gradually been abandoned in favour of making the machine simpler, less expensive, and more suitable for usage in a workshop.

The present design is the outcome of many years experience, and in actual practice it is found that the operator soon learns to interpret the requirements with a close degree of accuracy.

At a well-known firm of centrifugal pump makers in this country the average time required to completely balance a 36 inch pump and its shaft averages $1\frac{1}{2}$ hours from start to finish.

Turbine rotors, up to about 5 tons, are normally completely balanced in 3 hours, and rotors up to 15 tons in 5 or 6 hours, depending on the amount of out-of-balance.

It is therefore evident that, trial-and-error methods notwithstanding, the state of balance is arrived at expeditiously.

Olsen-Carwen Machine.—One of the class of smaller dynamic balancing machines, viz., the Olsen-Carwen, made in the U.S.A., claims to dispense with the necessity for any calculation whatever.

The rotor under test is connected to rotating parts in a headstock and a compensator, and the whole is supported on a vibrating frame which can be supported on a longitudinal or a transverse axis as desired.

Weights in the headstock can be adjusted by external hand-wheels to put the whole system into balance, and dials register the amount and position of the weights. The makers state that the charts supplied with the machines then enable the counterweights to be determined without necessity of any calculations.

The machine is made in several sizes, the smallest dealing with work between 2 and 8 lbs. in weight, and the largest with rotors up to 7 tons.

In the larger sizes the cost is exceedingly heavy, which may be readily understood seeing that the " vibrating frame " has to support the whole system of rotor under test, driving gear, compensating apparatus, and headstock.

There are some good features in this machine, particularly for light work, but in the heavier machines there seems to be a possibility of false readings being registered due to so many other revolving parts being connected up to the rotor under test, particularly after the inevitable wear-and-tear develops in the moving parts.

References.—The following references may be found of interest in connection with the subject of dynamical balancing of rotors :—

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