THE BROADWAY SERIES OF ENGINEERING HANDBOOKS

THE TESTING OF MACHINE TOOLS

GEORGE W. BURLEY

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THE TESTING OF MACHINE TOOLS

BY

GEORGE W. BURLEY, Wh. Ex., A.M.I.Mech.E.

LECTURER IN ENGINEERING AND THEORY AND PRACTICE OF MACHINE TOOLS IN THE UNIVERSITY OF SHEFFIELD
AUTHOR OF "LATHES: THEIR CONSTRUCTION AND OPERATION"

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E V.

PREFACE

In this book an attempt has been made to deal with the various aspects of machine-tool testing in such a way as to make the book suitable for engineer apprentices and students as well as responsible engineers. Considerable space in the book has been devoted to the testing of machine tools and machine-tool elements for accuracy since the author believes that this subject is of prime importance at the present day with its increasing demands for accuracy and efficiency. Generally, more space has been given over to the description of methods and instruments than to the consideration of published results of tests, though the latter have by no means been neglected in view of their importance from the point of view of machine-tool design.

From the general subject of machine-tool testing it is impossible to separate the lesser subject of the testing of cutting tools; hence exclusive consideration has been given to this in a chapter towards the end of the book.

It is hoped that the book may prove useful to engineering teachers and others who have charge of classes in machine-tool testing.

GEORGE W. BURLEY.

Sheffield University,
July, 1915.

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CHAPTER I.

INTRODUCTION.

FROM the general point of view the common function of all machines may be regarded as the *performance of work* or the *production of output*, no special definition being given to the meaning of either work or output when used in this connection, since both vary considerably and depend in any particular case upon the kind of machine to which they have reference. In any case, however, we generally suppose that the work or output is (either immediately or ultimately) of a more or less useful character so far as man—individually or collectively—is concerned, though there are, undoubtedly, exceptions even to this excellent rule in regard to the utility of individual mechanical processes and machines.

As one example of the output of a machine we may take the case of any form of textile machine, say, that form of machine which is designed for weaving cloth. In this case, the ultimate work of the machine is the predetermined arrangement of threads of cotton or wool in the weft and wove to form the required cloth, and the output of the machine is, obviously, the quantity of cloth without blemish which the machine can produce in any given time.

Another example of the output of a machine is to be found in the case of self-propelled vehicles. Here the output cannot be stated in simple terms of a material quantity, since the work of such a machine is the transference of masses in the shape of the passengers or load from one place to another, the relation between the two places being either horizontal or vertical, or a combination of these. The output of such a machine may, therefore, be regarded simply as movement or motion, though it has another signification.

A further example is that of the class of machines known as machine-tools. The true function of such machines is the production of machined parts of required sizes, shapes, and degrees of finish, such work being effected by the actual removal or cutting away of the excess material from the stock or original material when this is in the cold state. In every case, such removal or cutting away of excess material is performed by means of one or more cutting tools, and the material which is removed is usually rejected as waste or scrap and consigned to the scrap heap. Obviously, the output of a machine-tool is related closely to the number of machined parts that such a machine can turn out or produce in a given time, and from the point of view of the practical man this is usually the only point of any importance, though from the economic point of view there are many other factors which demand consideration and which have a profound influence upon the output when defined as above. From the scientific point of view, machine-tool processes may be divided generally into two classes: roughing and finishing. In the case of the former, only an ordinary degree of accuracy is required, the chief object being the rapid removal of material from the stock, from which point of view the output of a machine-tool on which such work is done may be assumed to be the volume or weight of metal which can be removed in a given time. On the other hand, when the process is a finishing one, the depth of the cut is exceedingly small, so that the volumetric or gravimetric aspect of output does not hold here, a much more reasonable definition of output in such a case being based on the superficial area which can be machined or finished in a given time.

Now, in order to be able to perform work or to produce an output for which it is designed, a machine—without any exception whatever—must be provided with motion or movement, or, rather more strictly, certain principal and elemental parts of it must be so provided, though at the same time certain other principal parts must be maintained in a relative condition of rest, since the fundamental principle of operation or action of a machine is not the absolute motion of any part in it, but the motions of the various parts of the machine relative to one another.

In connection with this question of the relative motions or movements of the component parts of a machine, two points call for consideration. In the first place, except in the theoretically ideal machine—which is, of course, not realizable practically—wherever there is relative motion between machine parts or elements there is also a resistance to that motion, which we describe as friction. This resistance is of two kinds, or rather two forms: static friction—sometimes called sticktion—which is the resistance to motion which is exerted when the motion is just being commenced, and the friction of motion, or dynamic friction, which is the resistance to motion when there is actual relative motion between the parts or elements. In practically every case the former is considerably greater than the latter but, whatever may be their distinguishing features, in our study of this subject we may regard them as being merely two different phases of the same entity, and, therefore, treat them together. As a matter of fact, the former is only of any importance in connection with those parts of a machine which have relative reciprocating or to-and-fro motions and those parts which have relative circular or rotatory motions when these are just being started. Now, this frictional resistance, which always acts in a direction parallel to, or tangential to, the contact surfaces of the parts between which there is relative motion, has to be overcome before this motion can be imparted to, or maintained in, the parts of the machine. This requires always the application of a mechanical force or forces, the action or operation of which involves the expenditure or consumption of mechanical energy, since by the expression "mechanical energy" we mean the capability of doing mechanical work or overcoming mechanical resistance through space. This expenditure or consumption of mechanical energy *always* occurs whenever frictional resistance is overcome, though the energy, simply as energy, is not destroyed, but is converted into another form of energy, namely heat energy, by a process which is known as the "degradation of energy". Thus, to overcome the frictional resistance which occurs between bearings and journals and between machine slides and guides, mechanical force and mechanical energy are required, the amount of mechanical energy absorbed in such a process not being available for further use in connection with the overcoming or neutralization of frictional resistance, but being converted directly into heat, or heat energy. The heat which is thus produced is dissipated in several ways, such as the raising of the temperature of the parts between which there is relative motion, the heating of the atmosphere in the immediate neighbourhood of the parts by conduction, and the heating of remote objects by radiation and convection. In each and every case, however, the energy which is so utilized is lost to the machine and remains heat energy until by a process external to the machine it is re-converted into mechanical energy or converted into chemical or electrical energy.

We may associate air resistance—which is a form of molecular frictional resistance and an item of some considerable importance in connection with the operation of certain machines—with the ordinary frictional resistance to the relative movement of solid bodies, since its overcoming requires the expenditure of mechanical energy, which, in the process, is converted into heat energy as in the case of the above resistance.

The other point which calls for attention is that of the inertia of the moving parts of a machine. Owing to the existence of this property of all material bodies, whenever the motion of a part of a machine changes it is always accompanied by the application of a force, which may be regarded as either positive or negative according to whether the change in the motion of the part is in the nature of an increase or decrease. In other words, to accelerate the motion of a machine element it is necessary to apply a force the magnitude of which is greater than the magnitude of the force which is only required to overcome the frictional resistance to motion. This, of course, involves the expenditure of mechanical energy, though in this case the potential mechanical energy is not immediately converted into heat but into another form of mechanical energy, namely, kinetic energy, that is, the kinetic energy of the part is increased.

In any machine, it should be observed, since rates of motion or speeds do not increase indefinitely, but always ultimately become zero as at the start, for every acceleration there is a retardation or negative acceleration. In this case the above process is reversed, and the amount of kinetic energy in the moving part is reduced, since the rate of motion or speed is reduced. Thus, if E = the kinetic energy lost or given up by the moving part; W = the weight of the moving part, in lbs.; V = the velocity of the part, in feet per second, at the beginning of the period of retardation; V_1 = the corresponding velocity at the end of the period; and g = the gravitational co-efficient; then we know from the principles of mechanics that—

$$A = \frac{W}{2g}\left(V^2 - V_1^2\right) \quad . \quad . \quad . \quad . \quad (1)$$

Now this amount of kinetic energy which, it should be remembered, is one form of mechanical energy, has been given up by the moving part, and since energy is the capability of doing work or overcoming a resistance through space, it follows that this amount of kinetic energy is not destroyed, but is merely converted into another form of energy.

In this connection three cases present themselves. In the first case, all this kinetic energy is reconverted into potential mechanical energy, so that the accelerating and retarding processes just balance one another, and no energy from an external source is required to perform the accelerating process. This is the ideal case, and is never exactly attainable in practice. In the second case, all the lost kinetic energy is utilized in overcoming frictional resistances, as in the case of a train or automobile when the brakes are applied and the motive power cut off. Here all the lost kinetic energy is converted into heat energy and none is available for use in connection with the next accelerating process. In the third case, part of the lost kinetic energy is absorbed in overcoming frictional resistances and thus dissipated in the form of heat, whilst the remainder is made available in some way, either as potential energy or kinetic energy in some other part of the machine, for use in the next accelerating process. Thus, here, only a portion of the total accelerating energy has to be supplied during each cycle of movements from an external source, but this amount, whatever it is, represents the amount which was converted into heat during the previous cycle.

We thus see that all machines are consumers of mechanical energy. They can, however, be divided into two distinct classes, namely, those in which only a part of the energy which is supplied to the machine is actually consumed in it, so that a part of this energy is available for use in some other process or machine, and those in which all the energy which is supplied is consumed in the machine, either in the actual process for which the machine is designed or in overcoming frictional resistances in the internal mechanism of the machine.

To the first class all prime-movers, such as steam engines and turbines, internal-combustion or explosion engines, and electric motors and generators belong. In the case of any of these, the energy which is supplied to the machine is really converted into another form of energy which is more useful than the original. Thus, in the case of the steam engine the energy which is supplied is in the form of heat energy which is contained in the steam, and this, through the mechanism of the engine, is converted into mechanical energy, a part of which is available for use beyond the engine, the remainder being lost in the engine itself in overcoming internal resistances. In the case of the electric motor, it is electrical energy which is supplied and mechanical energy which is given out, whilst in the case of an electric generator the process is reversed, it being mechanical energy which is supplied and electrical energy which is yielded by the machine. In addition to the above there are machines in which mechanical energy is supplied and mechanical energy given out, the chief use of such machines being the alteration of the conditions of utilization of the energy, chiefly in regard to speed.

In the second class are to be found all machine-tools, textile machines, wood-working machines, presses, power and other hammers, and process machines generally. In the case of the machine-tool all the energy (which, incidentally it may be stated, is always mechanical energy and never energy in any other form) which is supplied to the machine at the first driving shaft or pulley (where there is only one driving point), or driving shafts or pulleys (where there are two or more driving points), is dissipated in the various parts of the machine and at the cutting edge or edges of the tool or tools. It is all, however, degraded and converted into heat, which passes into the work, tool or tools, machine, chips or turnings, atmosphere, and lubricating or cooling agent whenever any such is employed. The energy which is consumed or degraded in the actual metal-removing process at the cutting edge or edges of the tool or tools employed is consumed in overcoming the internal or molecular resistance of the metal to the action of the shearing, compressive, twisting, and bending forces which are brought to bear upon it in the removal of the chip or turning, and also in overcoming the frictional resistance which occurs between the turnings or chips and the cutting face or faces of the tool or tools, and which is due to the influence of the cutting force or forces which are brought to operate when the tool or tools enter the work. The remainder of the energy which is supplied to the machine is dissipated in overcoming the frictional resistances in the bearings, guides, and gear boxes.

From the foregoing it will be readily observed that the case of the second class of machines—of which the machine-tool is typical—is somewhat different from that of the first class. In the case of the latter, the output of any such machine can always be stated or measured in terms and quantities similar to those of the input or energy supplied to the machine, so that the economic value of the machine can easily be determined by means of experiment, this value being in the form of an efficiency. Another method of referring to the economic value of certain machines which belong to this class is to state it in terms of the quantity of the substance from which the total energy is derived (such as coal, steam, water, gas, oil, or petrol) per unit of output energy, and, though the conditions under which prime-movers generally work vary considerably, it has been found possible to evolve standards by means of which the economic values of machines belonging to this class can be determined and compared.

The case of the machine-tool, however, is somewhat different, since its input is in the form of energy, whilst the output may be any one of the following:—

(*a*) a volume or weight of metal;
(*b*) a superficial area;
(*c*) a number of machined pieces.

From one point of view all these outputs occur in every case. In certain cases, however, especially in connection with scientific investigations, the output (*a*) is much more suitable than either of the others. In some other cases, the output (*b*) is the quantity which is usually taken, these cases covering finishing cuts on large pieces of work. From the point of view of ordinary engineering-workshop practice, the output (*c*) is the only one that is of much significance, and it is, therefore, the one which is usually employed in this connection, so that the workshop efficiency or economic value of a machine-tool is determined largely by the total cost of machining each piece of work done on it. This method, however, and indeed any method, does not enable standards to be evolved whereby the performances of machine-tools which work under widely differing conditions and perform different kinds of work can be compared. In all cases where reliable comparative results are required, as far as possible the machines should work under identical conditions so that, even unwittingly, an advantage is not given to any one machine.

It may be stated that the output of a machine-tool and the machine cost of producing that output depend upon a number of factors, of which the following are the most important and influential: (1) the mechanical efficiency of the mechanism of the machine; (2) the accuracy of fitting of the elements of the mechanism; (3) the arrangement and number of tools in use; (4) the kind and form of the tool or tools; (5) the general design and condition of the machine and its component parts; (6) the kind of metal which is being machined; (7) the combination of cutting speed, feed of tool or work, and depth of cut; (8) the kind of machining operation which is being performed; (9) the degree of facility of operation of the various parts and controls of the machine; and (10) the use or absence of a lubricating or cooling agent. Among the other factors which are involved are the cost of labour and the proportionate rent and depreciation of the machine.

Now, practically all the improvements that have been made in workshop practice, both in regard to rate of production (or the rate of doing work) and accuracy of finish, with a view to the production of a cheaper and better article, have only been made possible as the more or less direct result of experimental investigations and general testing in both shops and laboratories, and even to-day, though a high degree of efficiency has been attained, it is found necessary to continue such investigations and tests in order, amongst other things, to prevent retrogression. From this, however, it must not be inferred that in every machine-tool manufacturing works there is installed a fully-equipped testing plant and laboratory, though this is probably true generally of the larger works, for there have been many machine-tool investigations made in university and technical-college laboratories and shops which have produced much information of distinct value to the designers, constructors, and operators of machine tools, and without such knowledge the unprecedented progress which has been made would not have been possible.

Machine-tool tests may be divided into the following classes:—

1. Those tests which have as their object the checking of machining operations and the testing of the accuracy of the component parts of machine-tools with a view to their correct fitting and operation.

2. Mechanical-efficiency tests, in which the energy losses and useful-energy values are determined.

3. Cutting-force tests, by means of which the forces which are brought to bear upon the cutting faces of the tools employed are determined.

4. Power-consumption and dynamometer tests.

5. Output tests, in which the output, according to either of three definitions which have already been given, is determined or calculated.

6. Capability or cutting-efficiency tests, in which the volume of metal removed is a determining factor.

7. Wear tests, in which the machine parts are tested for wear resulting from use.

8. Tests on cutting tools of different forms and sizes, with a view to the determination of the most efficient forms and the general conditions which are associated with the maximum cutting efficiency.

9. Tests to check designs, such as tests to determine what the speed arrangements of a machine-tool actually are.

10. Miscellaneous tests, such as the methods of testing metals for machining properties and hardness.

The above classes can be divided into three general sections, as follows:—

1. Shop tests, which are tests made to keep up the standard of the machines so far as the accuracy of parts and the materials of construction are concerned.

2. Commercial tests, which are more or less comparative, having as their immediate object the improvement of output and the general conditions of operation, without respect to any factor which is of a more or less scientific nature.

3. Scientific and laboratory tests, in which scientific and systematic attention is given to the many various aspects of machine-tool and cutting-tool operation and performance, both in regard to power and output as well as efficiency and the nature and magnitude of the forces which are associated with cutting action. The object of these tests and experiments is to acquire knowledge and information likely to be of use to the designer and user of machine-tools, so that improvements both in their design and use can be made.

CHAPTER II.

TESTS ON MACHINE-TOOL ELEMENTS FOR ACCURACY.

THE object of these tests is to determine whether the various parts or elements of machine-tools have been made accurately and so fit and work together as to produce satisfactory results in the operation of the machines. In other words, to determine whether they are made so that the machine as a whole is capable of turning out work which is accurate within certain specified limits. It may be safely assumed that no machine-tool—no matter what the nature of its particular work may be—is able to produce work which is more accurate than itself. Therefore, since the general accuracy of a machine-tool is dependent absolutely upon the accuracy of its component parts or elements, it follows that this latter accuracy is the one to be considered.

Now, these accuracy tests are of two kinds, namely, those in which the degree of inaccuracy cannot be stated as a quantity, and those in which the error or departure from the truth can be so stated. The first kind of test is purely a qualitative test, and is only generally useful when the adjustment of a part is possible. The other is strictly a quantitative test, and as such is far more valuable and useful than the former.

Of these two tests, the former is generally, though not exclusively, used in connection with the testing of machine-tool parts which are in course of construction, machining, or fitting; the latter is, in the majority of cases, the only test which is applied in the case of the finished machine with all the parts in their correct places.

ENGINE LATHE TESTS.

1. **Bed-levelling Tests.**—The function of such tests is to determine whether the machine as placed down on its foundation is in such a position that its main element, namely the bed, is in a perfectly horizontal position or not; and, if not, to check any adjustment of its position until the required position has been secured. In such tests it is the bed of the lathe which is worked on, and usually the top of this. The tests for straightness and levelness or planeness which are made when the bed is being scraped to shape also belong to this class, there being a close connection between the two.

Spirit Levels.—One method of performing this test—and the usual one—involves the employment of a spirit level. The essential feature of a spirit level is a hermetically sealed glass vial of circular section and a more or less curved interior. This vial contains spirit in which is trapped a bubble of air. This bubble, which has a density much less than that of the spirit, always tends to rise to the highest part of the vial, which is so arranged that when it is perfectly horizontal its highest point and middle point coincide. In this case the bubble occupies a position in the middle of the vial which is plainly indicated.

Now, as is perhaps not generally known, the internal surface of the vial is curved in the direction of the length of the vial and it is to the top of this surface that the bubble of air always tends. In ordinary practice this curvature—which is practically of a circular character — can be imparted to the internal surface of the vial in either of two ways, that is, either by bending the tube from which the vial is made, or by grinding the vial internally so as to make it barrel shaped therein. The two forms of vial, namely, the ground and the bent forms, are represented in Fig. 1, the ground form being shown at A, and the bent form at B.

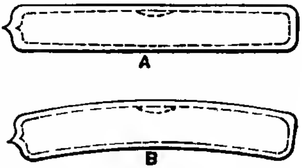


FIG. 1.—Spirit-level vials.

In all high-class spirit levels it is the ground form of vial which is used, since in the process of grinding it is possible to secure a higher degree of accuracy in regard to the curvature of the surface and also in regard to the regularity of the surface.

The vial is fixed in a frame—being firmly embedded in special cement or plaster. The nature of the frame varies considerably, such materials as cast-iron, brass, aluminium, boxwood, ebony, and mahogany being in considerable use. In any case, however, whatever the nature of the frame, the base of the frame should be made of a fairly hard metal, and the frame itself, when made of wood, firmly braced to prevent buckling or bending as the result of changes of temperature or other atmospheric conditions. The relation between the vial and the base of the instrument is such that the air bubble or index is in the middle of the vial about the zero mark when the base is quite horizontal. This condition is, of course, realized in the testing of the level during manufacture.

The overall lengths of these levels vary from about two inches to thirty inches, but for ordinary machine-tool bed levelling, both these lengths are unsatisfactory, a more suitable range extending from sixteen inches to twenty inches.

When spirit-levels have to be used on shafts or other parts of circular or curved section, it is better if their bases are provided with vee'd or curved grooves, so that the user of the instrument can place it on the shaft and be quite sure that the axis of the instrument is parallel to the axis of the shaft.

Some spirit-levels are equipped with cross vials for transverse testing, whilst others have additional vials for vertical testing. In other forms, an additional vial is mounted on a hinged or pivoted arm which works over a graduated scale, this being useful in cases wherein it is desired to set a part in a definite angular position or to test the accuracy of such a position.

Spirit-level Graduations.—The vials of spirit-levels can be marked or graduated in a variety of ways. The two commonest ways are to mark the vial or its frame at the middle of the air-bubble when the base of the instrument is quite horizontal, and to mark the vial or its frame at the two ends of the bubble when the same condition has been realized. On such vials there are no graduations, and the levels which contain them can only be used for qualitative work, though this is all that is required in the majority of cases.

Graduated vials can be marked out in at least four different ways. These are (1) in fractions of an inch, or millimetres; (2) in fractions of a degree, or minutes, or seconds; (3) in inches or fractions of an inch per foot; and (4) in binary or vulgar fractions with unity as the numerator in every case.

Vials which are graduated in the first way are generally provided with markings or graduations of either $\frac{1}{10}$ in. or $\frac{1}{20}$ in., or 1 millimetre, such a system of marking usually giving no indication of the value of each division so far as inclination of the level is concerned, though by means of a test it is always possible to determine this relationship.

Very sensitive vials when graduated in the second way read in units of two seconds or four seconds, whilst less sensitive ones read in one-minute or five-minute divisions. By means of this system of marking the actual inclination of a plane surface can be read off directly.

The third form of graduation also gives the actual inclination, but in this case it is represented by the rise of the inclined plane per foot length of the base of the plane. In other words, it gives a dimension which is proportional to the tangent of the angle of inclination. In the metric system of measurement, the corresponding form of graduation is millimetres per decimetre or per metre.

The binary or vulgar fraction which is the basis of the fourth method of graduation is really equal to the tangent of the angle of inclination with unity as the numerator.

Relations between Spirit-level Graduations.—Let l = the length of each graduation of the first form measured in, say, inches; R = the radius of curvature, in feet, of the upper internal surface of the vial (as shown in Fig. 2); and θ = the inclination of the base of the level required to cause the air-bubble or index to move through one division of the vial. Then, if we assume that the length l is measured on the circular surface, we have that—

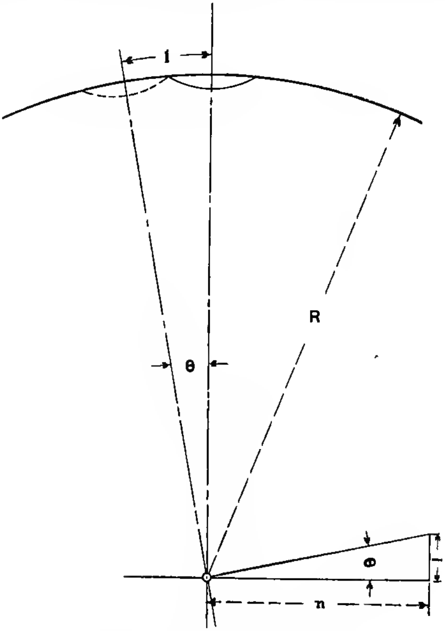


Fig. 2.—Spirit-level diagram.

$$\frac{l}{12R} = \theta \text{ (in radians)} \quad . \quad . \quad (2)$$

$$\frac{286{\cdot}5\,l}{R} = \theta \text{ (in minutes)} \quad . \quad . \quad (3)$$

$$\text{and } \frac{17{,}190\,l}{R} = \theta \text{ (in seconds)} \quad . \quad . \quad (4)$$

If we assume that the dimension l is a straight line quantity (either tangential or chordal), the above formulæ are still applicable since the value of θ is ordinarily exceedingly small.

Let i = the value of a graduation according to the third system of dividing, in inches or a fraction of an inch per foot. Then—

$$\frac{i}{12} = \tan\theta \quad . \quad . \quad . \quad (5)$$

$$= \theta \text{ (in radians) practically } (6)$$

Relating expressions (2) and (5) and (6), we find that:—

$$i = \frac{l}{R} \text{ practically} \quad . \quad . \quad (7)$$

$$\text{and that } \frac{l}{12R} = \tan\theta \quad . \quad . \quad (8)$$

This fraction $\frac{i}{12}$ or $\frac{l}{12R}$ is also equal to the binary or vulgar fraction which represents the fourth system of dividing and marking, so that if we let $\frac{1}{n}$ represent this fraction we have that—

$$\frac{1}{n} = \frac{l}{12R} \quad . \quad . \quad . \quad (9)$$

from which we derive this relationship:—

$$n = \frac{12R}{l} \quad . \quad . \quad . \quad (10)$$

n being invariably a large integer.

The angle θ (whatever unit it is measured in) may be defined as the angular value of each division or graduation of the level, the dimension i as the slope or inclination value of each division, and the fraction $\frac{1}{n}$ as the tangent value of the same.

In the following table are given the values of i, $\frac{1}{n}$, and R for various values of θ (the angular value of each division), two values of l (the length of a vial graduation) being taken in the case of the values of the radius of curvature.

TABLE I.

SPIRIT-LEVEL VIAL DATA.

Angular Value of each Division. (θ).	Slope or Inclination Value, per ft. (i.)	Tangent Value $\left(\frac{1}{n}\right)$.	Radius of Curvature of Vial Corresponding to Following Values of (l).	
			$l = \frac{1}{20}$ in.	$l = \frac{1}{10}$ in.
2 secs.	0·00012 in.	$\frac{1}{103{,}128}$	429·7 ft.	859·4 ft.
4 ,,	0·00023 ,,	$\frac{1}{51{,}564}$	214·9 ,,	429·7 ,,
5 ,,	0·00029 ,,	$\frac{1}{34{,}376}$	171·9 ,,	343·8 ,,
10 ,,	0·00058 ,,	$\frac{1}{17{,}188}$	85·9 ,,	171·9 ,,
15 ,,	0·00087 ,,	$\frac{1}{11{,}459}$	57·3 ,,	114·6 ,,
20 ,,	0·00116 ,,	$\frac{1}{8594}$	43·0 ,,	85·9 ,,
30 ,,	0·00175 ,,	$\frac{1}{5730}$	28·6 ,,	57·3 ,,
45 ,,	0·00262 ,,	$\frac{1}{3820}$	19·1 ,,	38·2 ,,
1 min.	0·00349 ,,	$\frac{1}{2865}$	14·3 ,,	28·6 ,,
2 mins.	0·00698 ,,	$\frac{1}{1432}$	7·1 ,,	14·3 ,,
1 deg.	0·20940 ,,	$\frac{1}{48}$	0·24 ,,	0·48 ,,

The above considerations apply strictly only to the tubular form of spirit level. There is, however, another form of spirit-level in which the glass or vial has an upper surface which is spherical or spheroidal. In this form, the motion of the air-bubble is not limited to one direction (as it is practically in the ordinary form of level), so that such an instrument can be used as a cross-test level as well as a straight-test level.

Hydrostatic Level.—In this form of levelling instrument the fact that water always tends to find its own level is made use of. It consists essentially of two exactly similar graduated cylinders with a flexible connecting tube between them. The two sets of cylinder graduations should be exactly alike and related in exactly the same way to the bases of the cylinders; otherwise the instrument is practically worthless. In any case, however, unless some form of magnifying apparatus (such as a microscope) is employed in connection with the readings, the results which are obtainable do not compare with those which are obtainable with the ordinary spirit-level.

With this instrument the difference in level between the two points or places tested equals the difference between the heights of water in the two cylinders, the higher cylinder showing the lower head of water.

Gravity or Pendulum Level.—The principle of action of this instrument is based on the fact that the bob of a heavy pendulum when left to itself always occupies its lowest position, so that the longitudinal axis of the pendulum is thus always perfectly vertical. If such a pendulum is supported in a case, obviously the position of it in the case will vary with any variation in the direction of the base with respect to the horizontal. In one form of this instrument, the motion of the pendulum is transmitted through toothed gear wheels to a pivoted needle or pointer which works over a graduated dial. The graduations of the dial can be in degrees of inclination, inches per foot, or fractions of an inch per inch.

Norton Pendulameter.—This is a precision level which works on the pendulum principle. It is an American instrument, and was originally designed for the testing of the beds and ways of grinding machines for straightness, evenness, and parallelism. It can, however, be used in ordinary levelling tests.

In this instrument the pendulum bob consists of a lead weight, weighing some 40 lb. avoirdupois, this being supported at the end of a piece of fine steel piano wire of a length of 10 ft. The movement of the bob is communicated to a long pivoted needle, which moves over a graduated scale, the total magnification (which consists of two distinct parts) being in the neighbourhood of 500 times. The degree of sensitiveness of this instrument is of the order of 0·00025 in., that is, dimensional and surface variations of this magnitude can be discovered by the use of this instrument. For the determination of transverse variations a second needle is provided, the magnifying power of this being of the order of 433 times, and the degree of sensitiveness equal to that of the other needle. The pendulum supporting wire is encased in a mast of seamless steel tubing, whilst the needles (which are made from thin aluminium sheet) are protected from the influence of air currents by being surrounded with light sheet-steel guards. The principle of action of this instrument is illustrated

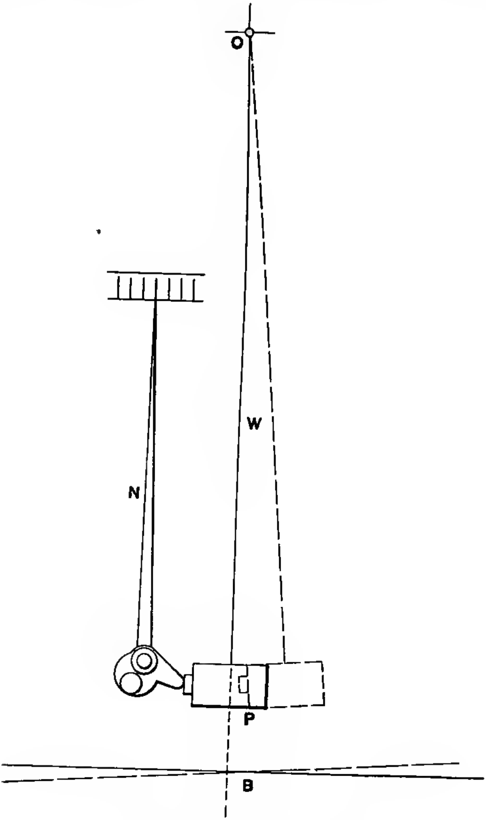


Fig. 3.—Pendulameter diagram.

diagrammatically in Fig. 3, P being the pendulum bob, W the supporting wire, O the point of suspension, N the needle of the instrument, and B the base.

It will be observed from this diagram that the angle through which the pendulum swings is exactly equal to the inclination of the base of the instrument, so that the movement of the bob of the pendulum will be to the rise of the high side or edge of the base in the ratio of the length of the pendulum to the length of the base. This ratio is 50 to 1 in one direction and 43·3 to 1 in the other. The average magnifying power of the needle alone is 10 to 1, the total magnifying power of the instrument being equal to the product of these ratios.

Tests.—These tests may be divided into two classes: general and local level tests. The former test has respect to the general horizontality of the ways of the bed from end to end and from front to back when it is placed on a solid foundation, as every lathe bed should be before the final scraping operations to remove the tool marks and to correct any machining inaccuracies are performed.

This test can be performed by means of any one of the instruments described above, though the commonest instrument which is employed is undoubtedly the spirit-level. The level used should be accurately made and marked and fairly large, and provided with a metal base or metal base-inserts so as to reduce the possibility of wear on the base. A level which is provided with longitudinal and cross vials can be employed at one setting to test the bed for horizontality in both the longitudinal and the transverse directions. Otherwise, of course, two different settings have to be made. In every case, however, the object of the test is merely to offer a guide which will indicate when the condition of general horizontality of the bed has been realized in each aforementioned direction. This test is thus a qualitative one, and does not admit of the making of measurements or of the obtainment of data of a quantitative nature.

In some cases it may be found necessary, owing, say, to the limited capacity of the spirit-level, to have recourse to the use of parallel strips or parallel straight-edges on which to place the level. Such pieces of apparatus, if used, should be accurately made and reliably parallel. Parallel strips are, generally, of two forms; these being distinguishable from one another by the difference in their sections. The first form is the block or prismoidal form, having a square or rectangular section. The other is usually of I section. Both are made of steel or seasoned cast iron. Parallel straight-edges are invariably longer than parallel blocks or strips. They are made of steel in the majority of cases, and exist in the several different sections shown in Fig. 4. For the majority of ordinary operations, the forms A and B are equally suitable, though when the straight-edge

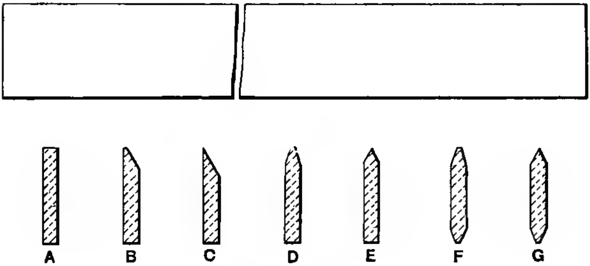


FIG. 4.—Sections of steel straight-edges.

has to be used as a parallel strip, preference should be given to the form A.

In regard to the transverse testing of beds provided with raised inverted vee ways, it should be noted that, unless there are two vees of precisely the same height or unless a compound level is employed, a difficulty may be experienced. This can, however, usually be overcome by making use of a packing-strip or small parallel strip of the same height as the vee to rest the level on.

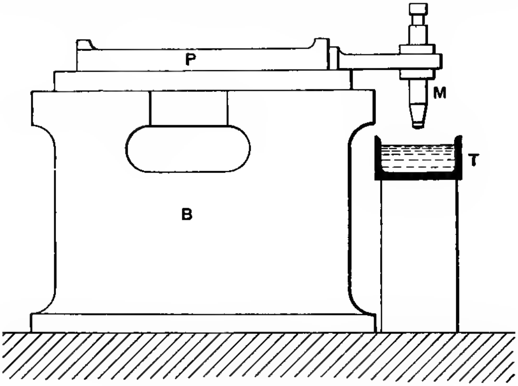


FIG. 5.—Optical bed-levelling test.

A rather unusual method of testing a lathe bed for general horizontality is indicated diagrammatically in Fig. 5. In this case use is made of the physical fact that the level of still water is always horizontal. The water is contained in a long, fairly shallow vessel T—such as, for example, a length of channel iron provided with ends. This vessel is supported in some convenient manner, though there is no need to set it in an exactly horizontal position. The position of the vessel with respect to the bed of the lathe is, as is indicated in the figure, approximately parallel. On the bed of the lathe (which is lettered B in the figure) a plane surface plate, P, is used, the plate being turned over so that it rests on the bed with its plane surface on the underside. A horizontal arm is attached to the plate, this arm carrying a simple microscope, M, of medium power. The axis of collimation of the microscope—that is, the axis of the tube of the microscope—should be arranged as nearly vertical as possible.

In the test, when the plate P is at one end of the bed, the microscope is adjusted until the surface of the water is brought into focus. The plate is then moved along the bed of the lathe to the other end. If the surface of the water is still in focus the surface of the bed is generally horizontal. If not, then the distance through which the collimator tube of the microscope has to be raised or lowered in order to bring the surface of the water into focus again represents the difference in level between the ends of the bed. Ordinarily, however, this dimension is of no value whatever, so that the setting of the microscope is not altered but, instead of this, the end of the bed is raised or lowered, whichever is required, until the surface of the water is again brought into focus. This method is a reliable one, but it is not as easy to apply as the spirit-level method.

Local bed-levelling tests are only made during the process of construction and erection of a lathe; they are practically never made when the lathe is fully equipped and ready for service. These tests can be divided into two classes, namely, those which are made during the process of finishing the bed either by scraping or grinding, and those which are made on the finished bed as a final check; and though both are desirable there are machine-tool manufacturing works in which the first at least is not applied, the scraping being done without the aid of any gauge and implicit confidence placed in the work of the planing or milling machine. This latter practice is not, however, very satisfactory, since it is practically impossible to machine on the planing or milling machine long parallel surfaces sufficiently accurate for fitting without any correction.

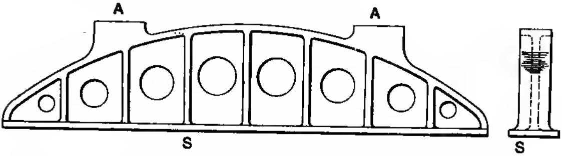


Fig. 6.—Cast-iron straight-edge.

Surfaces which have to be horizontal are tested for local truth by means of a plane surface plate, this being run over the surfaces more or less lightly, the location of high and hard places being indicated by the use of a marking of red lead or Venetian red between the two surfaces. In certain cases, also, the general flatness of the surface (apart entirely from general horizontality) can be tested by means of a straight-edge, one form of which has already been described. Another and superior form for this kind of work is represented in Fig. 6. This is a straight-edge, which is made of fine-grained cast iron, and has a test surface rather than a test edge. This surface is shown at S in the figure. The back of the instrument is curved more or less parabolically in order to prevent or resist the distorting influence of its weight on the test surface. On the back are cast two lugs A A with plane surfaces, these being used to support the straight-edge when it is not in use. On some edges these lugs are replaced by hard wood feet. The back of the straight-edge is in the form of a flange which is connected to the flange S by a web of a reasonable thickness, this web being stiffened by a number of cross ribs disposed uniformly from end to end of the straight-edge. Between consecutive ribs the web is centred to reduce its weight without impairing its resisting power.

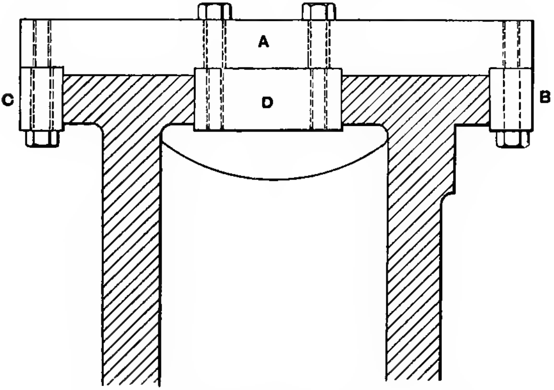


FIG. 7.—Testing plate for English lathe beds.

For the testing of vertical and inclined surfaces and combinations of these and horizontal surfaces it is usually found necessary to use a special testing jig or built-up or formed surface plate.

In Fig. 7 is shown the form of such a plate for testing and checking the truth of the four vertical guide faces of an ordinary English lathe bed, both with respect to each other for parallelism and with respect to the horizontal surfaces. It consists of a plate A, the length of which is approximately equal to the swing of the lathe or, say, twice the height of the centres of the lathe above the upper surface of the bed. To the underside of this plate are attached three blocks, B, C, and D, the surfaces of which are carefully machined and scraped so that the blocks are exact rectangular prisms, with faces parallel to and at right angles to one another. The block B is fixed permanently to the plate A in a recess at the front as shown, the under surface of the plate and the vertical face of the block being normal to one another. With this block only in position the truth of the front vertical face is tested and checked. The back block C is then attached to the plate, its position on the plate, and with respect to A, being exactly determined by the means shown. Finally, the block D is attached, and the inside faces of the shears of the bed scraped until this block can be inserted between the two shears and the whole plate moved along the bed with ease, but not accompanied by slackness at all. In place of one middle block, sometimes two are employed, the positions of these being adjusted one at a time.

When the plate can be easily and steadily pushed on the bed with all the blocks in their final positions, it may be rightly assumed that all the tested surfaces are parallel to one another.

A jig or gauge for checking the truth of underside horizontal surfaces is shown in Fig. 8, which is self-explanatory.

For the testing of surfaces which are compounded of horizontal and inclined faces, as in the case of the bed of the ordinary American lathe, a jig or test gauge of the form shown in Fig. 9 is employed.

Another method of testing the surface of the bed of a lathe for local accuracy in regard to flatness and alignment involves the use of a long wire (of uniform diameter) placed under tension. The wire used should be ordinary fine steel piano wire. This should be stretched above the top of the bed and between the two ends.

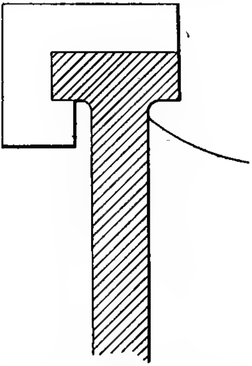


FIG. 8.—Lathe-bed testing gauge.

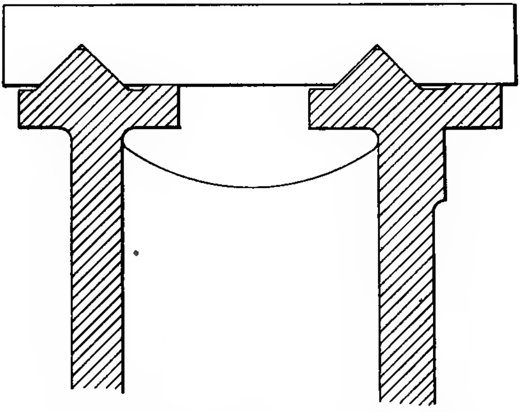


FIG. 9.—Testing plate for American lathe beds.

This can be done by securing a strong plate to each end of the bed, and fitting to each an adjustable threaded eye-bolt, in the eye of which the wire is fastened. The tension in the wire can then be adjusted quite easily by means of the nuts on the shanks of the bolts. It should be adjusted so that all the sag is taken out of the wire.

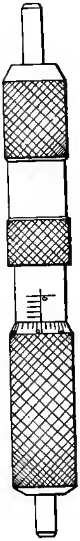


FIG. 10.—Inside micrometer screw gauge.

In the test, the position of the wire with respect to the top of the bed is initially adjusted until the wire is generally parallel to the top of the bed. This condition can be realized through the medium of some form of micrometer screw or vernier gauge (preferably the former). The inside form of micrometer screw gauge (represented in Fig. 10) is probably in this connection superior to any other form, though it is possible to obtain satisfactory results with the use of height and depth gauges of the micrometer screw type. This instrument is placed between the bed and the wire (as shown at A in Fig. 11) and adjusted until its ends are just in contact with these. It is then removed to the other end of the bed (as at B in the same figure) and its reading there noted. If the readings for the two ends coincide, the wire is generally parallel to the surface of the bed; if they are different the disposition of the wire is altered by adjustment at one end until they agree. For the testing of the bed for local flatness, the gauge is used at different places along the length of the bed, the differences between the readings for these positions and the end readings being noted. Wherever there are differences the bed is scraped down until the differences disappear. By this means the whole length of the bed can be tested.

A slight modification of the above arrangement—and one which makes the test easier of accomplishment—embodies the use of a telephone receiver, buzzer, or electrical bell or signalling apparatus. In this case the testing wire is insulated by means of

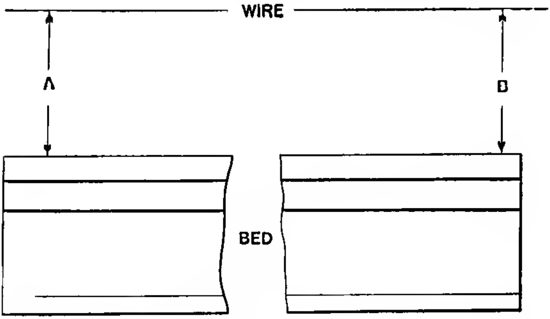


FIG. 11.—Bed levelling test involving use of fine wire.

vulcanized fibre or ebonite washers and bushes from the bed, but it is connected to one of the poles of a battery of dry or Leclanché cells. The other pole of the battery is connected to the receiver or buzzer, which in turn is electrically connected to the micrometer gauge. The micrometer gauge, which should preferably be provided with a small, perfectly flat base, is moved along between the bed and the wire, always in contact with the former. By setting the gauge initially to give the reading of the two ends, and reducing this by, say, 0·0001 in. so as to keep the receiver circuit open when the gauge is at either end, it is obvious that, as the gauge is moved along between the bed and the wire, the circuit will only be closed and the signal given when a high place in the surface of the bed is reached. This apparatus enables variations of the order of 0·0001 in. to be readily detected.

Local inaccuracies in the level of lathe beds can also be detected by means of the pendulameter which has been already described; in fact, it was for work of a kindred nature that this precision instrument was originally designed.

2. **Headstock Tests.**—The degree of accuracy with which stock is machined in the lathe depends to no slight extent upon the degree of exactitude which is embodied in the construction and erection of the two headstocks. Hence, it is necessary to test the various parts of these, both during the erection or fitting process and upon their completion.

Measuring and Indicating Instruments.—In connection with such tests recourse has to be had to indicating and measuring instruments, by means of which eccentricities, obliquities, and rectilineal differences can be determined, or at least discovered, though in connection with the final set of tests it is generally desirable to be able to state the observed errors as definite quantities. These instruments can be divided into three classes, these being as follows:—

(*a*) Measuring instruments working on the micrometer screw gauge or vernier principle;
(*b*) Non-measuring indicating instruments; and
(*c*) Measuring indicating instruments.

Micrometers.—The form of micrometer-screw gauge which is the most frequently employed is the plain micrometer head, as illustrated in Fig. 12. This consists of a barrel, B, on which a longitudinal scale is marked out to read, in the English system of measurement, in fortieths of an inch, or, in the metric system of measurement, in millimetres or half millimetres. On the barrel the thimble or sleeve, T, works, the connection between the barrel and the thimble being an accurately pitched screw and nut. In ordinary English micrometers the pitch of this screw and nut is a fortieth of an inch and coincides with the unit of the longitudinal scale on the barrel. Attached to the thimble is the spindle, S, the outward or inward movement of which is determined by the amount of rotation or revolution of the thimble round the barrel. Each complete turn of the thimble corresponds to a longitudinal or axial movement of the spindle of a fortieth of an inch or 0·025 in., so that each of the twenty-five circular divisions of the thimble corresponds to a thousandth of an inch movement of the outer end of the spindle.

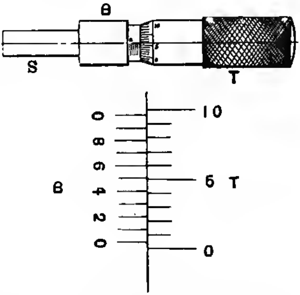


FIG. 12.—Micrometer screw-gauge head.

With some instruments it is possible to read or test to a ten thousandth part of an inch by means of a vernier scale which is provided on the barrel. The principle which underlies the use of this device is indicated in the figure below the drawing of the gauge. The vernier-scale lines are arranged parallel to the axis of the instrument, and are eleven in number, these enclosing ten equal spaces or divisions, which together are equal to nine divisions or graduations of the thimble scale. Each vernier division is thus equal in magnitude to nine-tenths of a thimble division, which represents 0·001 in. so that the difference between a thimble division and a vernier division represents a movement of the spindle of one-tenth of 0·001 in. or 0·0001 in. The number of ten-thousandths of an inch above the thimble reading is indicated by the number of the vernier mark which coincides with a thimble mark. In the case illustrated this number is 6.

The micrometer head is usually fitted in an arm or adjustable part when in actual use.

Other forms of the micrometer screw gauge are also used as, for example, the ordinary micrometer screw gauge, for external measurements, and the inside gauge, but, generally, these work on the principle described above.

Non-measuring Indicators.—These instruments (sometimes called "wigglers") do not admit of the actual measurement of errors, though they enable the presence of errors to be detected. They usually work on the principle of magnification, the operation of which causes the movement of the indicating element to be several—in some cases, many—times as great as the actual error, whether this be one of eccentricity or obliquity. As instruments for use in precision testing they are practically worthless except in those cases wherein only qualitative results for the purposes of correction are required.

One form of such an instrument is represented in Fig. 13. In this figure, T represents the test piece (supposed to be circular in section), E the holder of the instrument, A the indicator spindle, and P the indicating needle or pointer. The spindle A is secured in the part B, which in turn is mounted in the part C so that it can rotate freely on a horizontal axis. C is mounted on D, and has a vertical axis of rotation. The parts B, C, and D thus form a modified universal joint. The spindle A is so arranged in B that the combined length of the spindle and pointer on the one side is several times, say 20 or 30 times, as great as the length of the spindle on the other side. Any movement of the short part due to variations in the surface or motion of the test piece is therefore magnified 20 or 30 times during its transmission to the end of the pointer P.

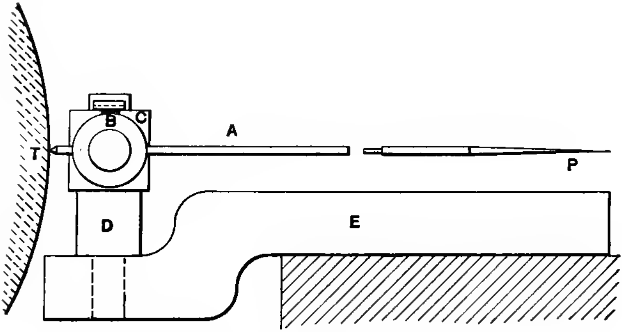


FIG. 13.—Simple form of non-measuring test indicator.

It should be observed that the pointer P is inserted in the spindle and removable therefrom when, for any reason whatever, it is found necessary to alter the degree of magnification.

In Fig. 14 is shown another form of non-measuring indicator. In this case a spring A is mounted on the base C, the end of this spring acting as the indicating needle and moving in front of an ungraduated face B. With this form of indicating mechanism it is not possible, however, to obtain even a moderate degree of magnification.

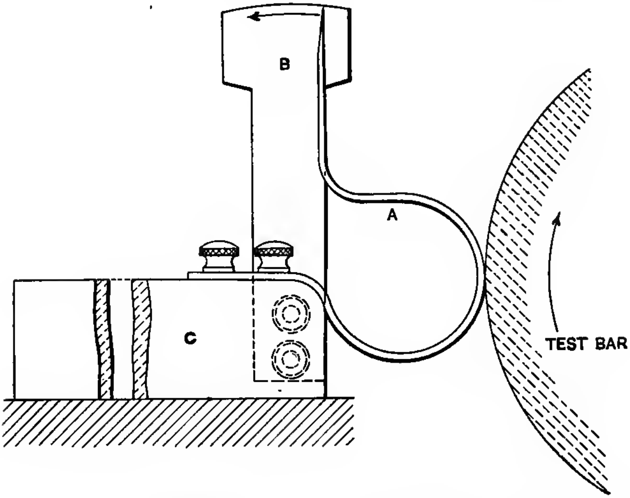


Fig. 14.—Another form of non-measuring test indicator.

An improvised form of non-measuring indicator is illustrated in Fig. 15. It consists of an ordinary unjointed 12-inch steel rule. This (represented at S) is employed in conjunction with a sharp-edged bar or turning tool, T. With the latter secured rigidly in the tool post or clamps, the rule is placed as shown in the figure, and the index finger is then pressed lightly on the rule at the place marked A. The pressure exerted by the finger keeps the rule down on the edge of the tool and up against the surface of the test bar. The line of contact between the bar or tool and the rule is the axis about which any movements of the rule due to the eccentricity or unevenness of the surface of the test bar occur.

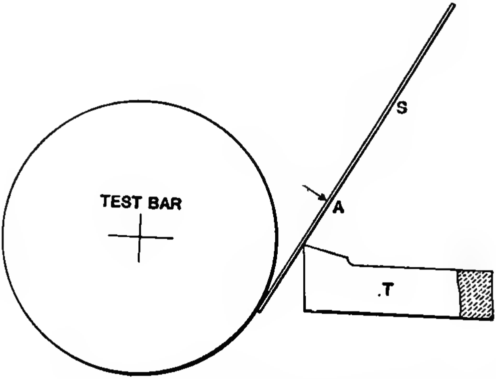


FIG. 15.—Improvised test indicator of the non-measuring type.

If the rule be placed in contact with the test bar at its extreme end, as shown, then the degree of magnification is simply the ratio of the two lengths of the rule.

One form of centre test indicator consists of a rod or spindle of steel provided with conical points at the two ends. One end is inserted for some distance in a metal ring and diametrically opposite to a needle or pointer. The inner end of the spindle is placed in a small centre hole in a strip of spring steel fixed in a holder.

Measuring Indicators.—These instruments, as well as those with which we have just dealt, usually go under the generic description of "test indicators". These can be used in connection with precision work of the highest type, though the degree of sensitiveness varies fairly considerably, ranging as it does from 0·0001 in. to 0·001 in.

The principles of operation which have been made use of in the design of these instruments are at least four in number. These are:—

1. The principle of the lever, both simple and compound.
2. The principle of the rack and pinion, or wheel and pinion.
3. The principle of the screw and nut provided with a very steep thread; and
4. The principle of the cylindrical grooved cam.

In some cases, two of these principles are compounded to obtain the desired result.

This type of indicating instrument is distinguished from the non-measuring type inasmuch as the indicating needle or pointer is so arranged that it moves over the face of a graduated sector or dial. Differences in dimensions can thus easily be read off the scale of the instrument or obtained by means of the differential principle.

A simple form of measuring test indicator is represented in Fig. 16. The principle of action is that of the simple cranked lever. One arm of the lever serves as the indicating pointer, whilst the other serves as the contact finger—that is, the element which is placed against the part whose truth is being checked. The axis or pivot of movement is in a small flat steel spring which is placed in the end of the instrument, and which can be adjusted readily by means of the set-screw shown whenever it is found necessary to adjust the zero position of the pointer.

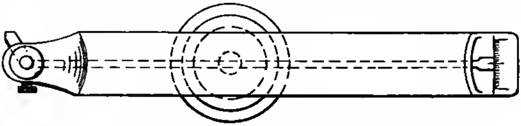


FIG. 16.—Test indicator of the measuring type.

The pointer is disposed inside the frame of the indicator to protect it from the influence of air currents when in use, and to protect it from rough usage in handling.

Another form of simple lever test indicator is shown in Fig. 17. The contact finger T (which is equipped with a conical point) and the indicating pointer P are the two parts of the lever which is pivoted in a vertical spindle A. Two flat springs B are used to bring the pointer back to its zero position when the instrument has been removed from the test bar or work.

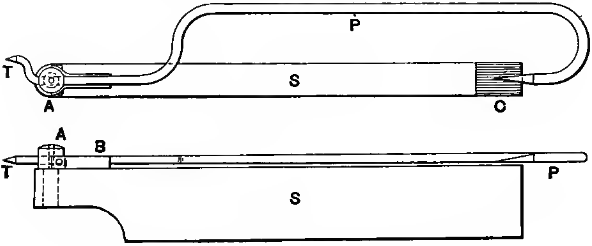


FIG. 17.—Another form of measuring test indicator.

C is the scale of the instrument, which, it will be noted, is so arranged that the pointer has to be used bent. S represents the holder of the instrument.

The principle of action of a modified form of simple lever indicator is represented in Fig. 18. In this case, however, the contact finger C is a distinct part of the instrument and quite separate from the lever. It will be seen that the contact finger works in a special guide and is prevented from dropping out of that by a cross pin. The lever consists of two parts, namely the arm L and the pointer P. The former is the part which is in communication with the inner end of the contact finger, whilst the latter moves over a graduated scale A. A curved leaf spring S is made use of in the manner shown in order to keep the arm L and the contact finger in contact and to throw the indicating pointer into its zero position when the instrument is not in use.

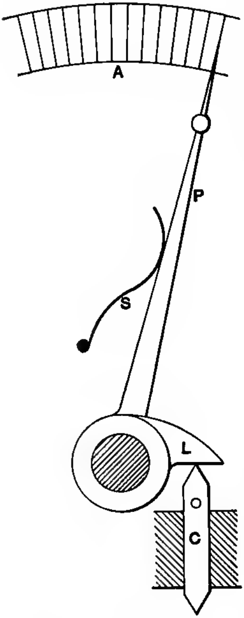


Fig. 18.—Test indicator.

A compound lever indicator is illustrated in Fig. 19. In this particular form of instrument two levers are employed, this arrangement enabling a higher degree of magnification or sensitiveness to be obtained without unduly lengthening the instrument. The principle of action of the instrument is clear from the diagram.

In some other forms of test indicator in which the compound lever principle is employed, the two levers are reversed and not continuous as in the case of the above instrument Straight and cranked levers in their several combinations are used, the first lever sometimes pressing on a pin secured in the body of the second lever or pointer. The contact fingers used in these instruments are of the free and lever-end types, as in the case of the single lever instruments.

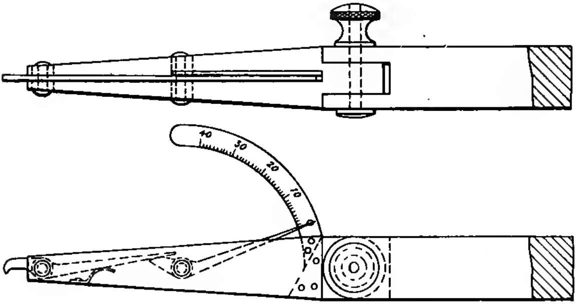


Fig. 19.—Compound-lever test indicator.

These instruments can also be divided into two classes, the differentiating factor in this case being the description of the scale over which the pointer moves. These two classes may be termed the sector class and the dial class. The pointer of an instrument of the former class has a scale which subtends but a comparatively small angle at the centre of motion of the pointer, the limiting positions of which form what is really the sector of a circle. All the instruments described above belong to this class. In the case of an instrument of the latter class, the pointer has a much more extensive range of movement, covering in many instances a complete revolution. These instruments have, generally, higher powers of magnification than have sector instruments.

The internal arrangement of one form of dial test indicator is represented in Fig. 20. With this type of instrument the axis of rotation of the pointer is usually situated in the middle of the case and serves as the centre of the scale on the dial. In this diagram, P represents the indicator pointer or needle, the arrow-head indicating the forward or positive direction of movement of the same. This needle is mounted on a drum spindle which is pivoted in fine hard conical or parallel bearings (sometimes jewelled). Round this drum passes a fine band or cord (preferably metallic and practically inextensible) which is connected at its other end to a pivoted sector or quadrant Q. This quadrant is swung on its pivot by a short band or cord B, which receives its motion directly from the contact finger T. The contact finger slides in the tube A, which is fixed to the case

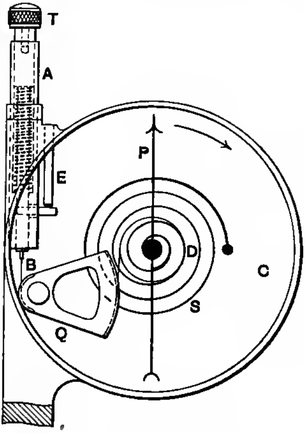


Fig. 20.—Dial test indicator.

of the instrument. The helical spring which is indicated in the figure provides the force of restoration when the indicator is removed from contact with the test bar or the element whose truth is being tested. The pin E serves as a control over the activity of the helical spring and enables the zero position of the pointer to be adjusted whenever, through any cause whatever, it has been altered. The spiral spring S is attached directly to the pointer drum D, its function being to provide the force of restoration for the pointer only.

It will be observed that the principle of action of the above design of indicator is essentially that of the compound or double lever.

In another form of dial test indicator a single straight or cranked lever is used in conjunction with a pivoted cylinder which is provided with a steep helical groove or screw thread. In this groove a die on the end of the long arm of the lever fits and is capable of motion. This motion in its translation from the lever to the cylinder becomes a rotatory one, and as such is communicated directly to a needle or pointer whose axis coincides with that of the grooved cylinder. The restoring spring may act directly on either the long arm of the lever or the pointer, though preferably the former; and the contact finger may be either free or formed on the short arm of the lever.

In Fig. 21 is illustrated a sector form of test indicator which, since it is enclosed in a cylindrical case not unlike that of a dial test indicator, may easily be taken as a form of the latter. In this instrument the compound lever principle is employed. The contact finger, A, is arranged in a radial guide in the interior of the case, and has secured to it a deflecting pin C. This pin C bears upon the short arm of the cranked lever B, and causes this to swing about its axis when the contact finger receives any motion. From B the motion is transmitted to the arm E and the pointer, and magnified by the latter. The spring D keeps the pin C and the lever arm in contact, whilst the small helical spring shown below E keeps the latter and the lever B always in contact.

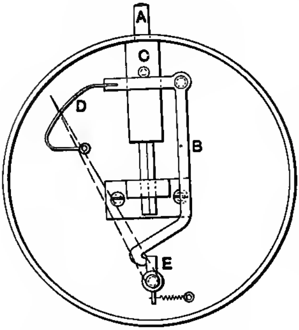


FIG. 21.—Interior of test indicator.

In another form of test indicator the scale is marked out on a fixed tube in an axial direction—not unlike a micrometer screw gauge scale—and the indicating mechanism consists of a sleeve which moves on the tube over the scale. The motion of the contact finger is magnified or increased by means of two levers and a toothed wheel device.

In some test indicators, chiefly though not entirely of the sector class, the scale divisions are either not exactly of equal value or of equal magnitude. This is due to a defect in the arrangement of the levers—in some cases ineradicable—whereby the leverage value varies with the positions of the levers. In dial test indicators this defect is not so prevalent as it is in the other class. In many cases wherein it exists, however, the error which it introduces into the readings made by means of the instruments is a negligible quantity.

The outer ends of contact fingers are spherical or spheroidal, knife-edged, sharp conical pointed, or blunt conical pointed. In some, accurately ground balls are actually used. For the greater part of machine-tool testing work the curved or spherical end is the most suitable.

In some cases it is found necessary to make use of external auxiliary cranked levers, as shown diagrammatically in Fig. 22, in which B represents the contact finger of the instrument and A the contact finger of the added lever. This lever, it should be understood, is mounted in a bracket which is secured to the case of the instrument. In order to maintain the value of each division of the scale the lengths of the two arms of the lever must be made equal. That is, the dimensions x and y must be equal. If it is desired to increase the sensitiveness of the instrument, it can readily be effected by making the dimension y an exact multiple of the dimension x. Thus, to halve the value of each division of the scale, y must be made equal to $2x$. This is, of course, tantamount to adding another lever movement to the instrument, though it is not a permanent element in the design of the instrument.

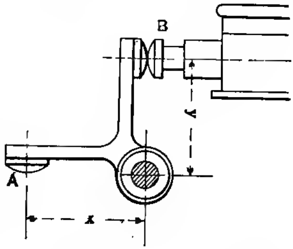


Fig. 22.—External test indicator lever.

A point which is of some material importance in connection with the use of test indicators is the magnitude of each division, since this determines largely the ease and exactitude with which the reading is taken. Generally, the dial form of test indicator, owing to its longer scale, has longer unit divisions than has the sector form. The length of a division on high-class dial indicators varies from $\frac{1}{32}$ in. to $\frac{1}{8}$ in., the value of each division being either 0·001 in. or 0·0005 in. The degree of magnification ranges from about 30 to 250 in ordinary dial indicators. In ordinary sector indicators it very rarely exceeds 100, whilst in the pendulameter (which is a form of sector test indicator) the degree of magnification is, in one case, 500, and in the other 433.

Choice of Test Indicator.—Concerning the choice of a test indicator for experimental work it may be pointed out that the important point is to select one upon the accuracy of whose calibration strict reliance can be placed, since the function of instruments of this type is the exact determination of errors. Micrometer screw gauges are, of course, reliable; but this cannot be said of all test indicators. Where it is required to know exactly the error in any part or parts of a machine-tool it is far better to trust to an instrument of the dial type than to one of the sector type, though it should not be understood thereby that all of the latter type are useless and all of the former type reliable. It is indisputably true, however, that, speaking generally, a dial test indicator lends itself more effectively to this kind of work than the other form. Improvised test indicators, whilst suitable in connection with certain kinds of machine-tool work wherein quantitative determinations have not to be made, are not suitable generally in connection with testing work on machine-tools.

3. **Driving Headstock Spindle Tests.**—The first essential of the driving headstock of an accurately built lathe is that the axis of the driving or main spindle shall be exactly parallel to the top of the bed in the one direction (that is, in a vertical plane) and to the guiding faces, strips, vees, or sides of the bed in the other (that is, in a horizontal plane).

The process of manufacture always aims at realizing this necessary condition, but it is found necessary in the process of erection of the lathe to test the headstock for this spindle accuracy.

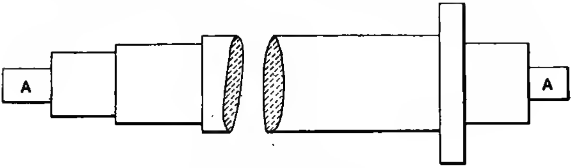


Fig. 23.—Spindle test bar.

In this test, the ordinary spindle is replaced by a special spindle or test bar, the diameter of which (at least at the journals) equals the journal diameter of the spindle, or the spindle is provided with a concentric cylindrical projection at each of its two ends, as is indicated at AA in Fig. 23. When the former is employed it should be accurately machined and finished with a smooth surface (preferably by grinding), so that, when the headstock bearing brasses have been secured in place, the test bar can be revolved by hand quite easily. There must, however, be no looseness in the bearings, the fit between the test bar and the bearing brasses or bushes being the ordinary spindle running fit. The length of the test bar should be such that a length of at least 12 in. of uniform diameter projects beyond the front bearing, as is shown in Fig. 24. The projections on the ends of the spindle need not be more than 2 in., and may be of any reasonable diameter above 1 in., but they must each be of the same diameter.

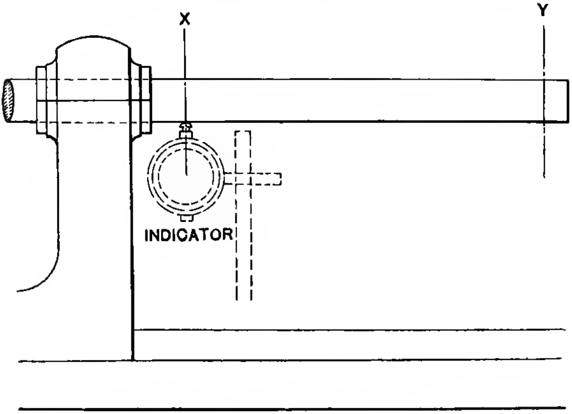


Fig. 24.—Test bar in lathe headstock.

When a test indicator or simple micrometer is used in the test, it should be mounted on a pillar fitted in a base in a manner somewhat similar to that of a scribing block or surface gauge. The under surface or surfaces should be carefully machined and scraped so as to ensure their being quite flat. For use on flat-topped beds the base should be provided with a lip having a vertical face to fit against a vertical face on the bed in the manner shown in Fig. 26. If it is an inclined face on the outside of the bed shears which is intended to act as the guide, then the base must be shaped accordingly. For use on vee-topped beds the underside of the base must be provided with a vee groove to fit over one of the vees on the bed, preferably a slide-rest vee. By such means as the above it is possible to move the indicator or micrometer parallel to the bed guides.

With the test bar of uniform diameter, readings should be taken with the indicator or micrometer at two places near the ends of the projecting length, such as, for example, those shown at X and Y in Fig. 24. Two sets of readings should be obtained at each end, one set being made with respect to the underside of the bar and the other with respect to the front. The two readings in each set should be secured from places which are diametrically opposite, as indicated in Fig. 25, in which H and H_1 represent the places on the surface of the bar for the front readings, and V and V_1, the corresponding places for the underside readings. To obtain these readings it is, of course, necessary to rotate the bar by hand through approximately half revolutions. The average of each set of readings at each end should be then taken, and the corresponding average results compared. In the practice of one firm of machine-tool manu-

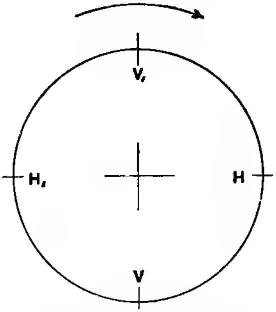
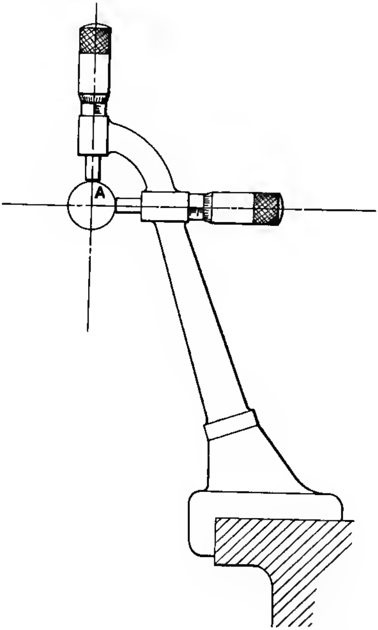


Fig. 25.—Diagram showing incidence of test indicator readings.

facturers, the test bar is allowed to be, on the average, 0·001 in. higher at the outer end and to have an offset of 0·001 in. towards the back also at the outer end. Each set of readings may comprise more than two readings if thought necessary or if found desirable, but ordinarily there is no need to make use of more than two readings in each set.



F.G. 26.—Test standard with two micrometers.

When the actual driving spindle is utilized in the test, the above method involving separate horizontal and vertical readings can also be applied, though in this case the indicator or micrometer has to be moved from one end of the headstock to the other, the headstock, of course, having to be moved slightly on the bed from the end to make room for the indicator.

An alternative method is indicated in Fig. 26. In this case a special fitting carrying two micrometer gauge heads is made use of, these having a horizontal axis and vertical axis respectively. The base of the fitting must, of course, be made to suit the shape of the ways or shears of the bed. When it is not required to take actual measurements the micrometer heads should be replaced by well-fitting screws provided with hardened ends made either perfectly flat or slightly convex.

Any correction of alignment of the headstock or headstock spindle, which the results of these tests may indicate, is usually effected before the test is finally completed. With flat-topped beds the correction generally can be made quite easily by making use of adjusting screws and studs in the headstock lugs or tongues which are placed between the shears of the bed. With vee-topped beds, however, it is essentially different, though with beds of this type there does not exist the same need for correction, since the vee strips on the bed and the vee grooves in the headstock determine the alignment of the latter with respect to the bed.

4. **Loose Headstock Mandrel Tests.**—The axis of the hollow spindle or mandrel of the loose headstock should be exactly in line with the axis of the driving headstock spindle (assuming, of course, that this has been correctly aligned). That is, this axis

should be parallel to the tops or ways of the bed in a vertical plane and to the guiding surfaces in a horizontal plane.

The tests which are necessary to check the alignment of this axis can be made in exactly the same way as the driving headstock spindle alignment tests are made; that is, a special test bar with a 12-in. projection or the actual mandrel with end projections of circular section like those shown in Fig. 23 may be employed in conjunction with a test indicator, micrometer screw gauge, or the special instrument, as described above, in which two micrometer heads are made use of.

The requirements of these tests are precisely the same as those of the driving headstock spindle tests, the two sets of results coinciding exactly.

Another test involves the use of the actual mandrel of the headstock. In this case, the ordinary surface of the mandrel is used, the contact finger of the indicator or the spindle of the micrometer gauge being placed in contact with it. Both horizontal and vertical indicator or gauge readings are taken at the outer end of the mandrel when the latter is first right in the barrel of the headstock and then as far out as it would be in an extreme case under ordinary working conditions. If the barrel has been accurately bored, and the fit between the barrel and mandrel is a good one, the two sets of readings will coincide with one another and with the corresponding readings taken in the driving spindle tests.

When the loose headstock is of the set-over type it is a comparatively simple matter to align the headstock mandrel horizontally by making use of the slipper or sole plate and adjusting the position of the body of the stock on it.

In the case of this type of headstock it is also desirable to know whether the cross or transverse guides are exactly at right angles to the mandrel axis and longitudinal bed guides. This information can be obtained in several ways. One way involves the employment of an accurate square and inside micrometer screw gauge or test indicator, this test being on the guides themselves. Another way is to test the cross or transverse movements of the extreme ends of the test bar or mandrel by means of a mounted test indicator or micrometer screw gauge. If the headstock guides are normal to the bed guides the differences between the two sets of end readings will be the same for any cross movement of the body of the headstock.

This is not, however, a point of much importance, the important point having reference to the position and relation of the mandrel axis when the body of the headstock is in its zero position.

5. **Slide Rest Tests.**—The cross or transverse slide of the slide rest of a lathe should be disposed exactly at right angles to the bed guides, to attain which it is necessary to test the cross slide in the course of erection and fitting so that any error can be corrected by means of further machining or scraping.

This test can be performed in at least two ways. In the first way a special testing slide (B, Fig. 27) is used. This is made to fit accurately on the bed of the lathe so that its motion is controlled directly by the guides or ways. The inside vertical face of the slide is planed and scraped to a dead square fit with respect to the bed guides. This is then used in conjunction with two test gauge blocks, GG, which are of the same length. In the figure, R represents the saddle of the slide rest.

In the actual test the test-gauge blocks are used as shown between the test surface of the slide and the nearer guide of the cross slide. When the correct relation between the bed guides and the saddle guide has been secured, the two gauge blocks, for any given force applied to the test slide and saddle, are gripped equally. The gauge blocks, when a quantitative determination is found necessary, can be replaced by an inside micrometer gauge or a dial test indicator, though preferably the former.

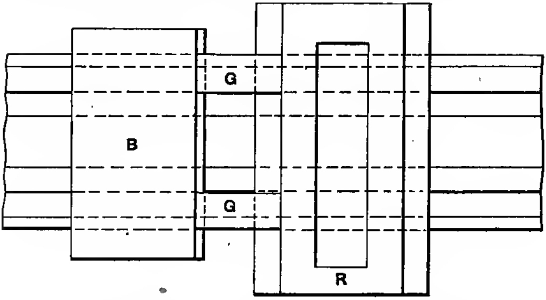


FIG. 27.—Method of testing slide-rest saddle guides.

The other method of making this test is indicated in Fig. 28. The apparatus required in this case consists of two round bars, B and B_1, together with an adjustable testing screw S. On the bar B are three collars. Two of these collars, CC, are carefully ground so as to be concentric and of the same diameter. The other collar, C_1, is made of a larger diameter. This bar is placed on the saddle of the slide rest, R, so that the two collars CC fit in the cross guide vee, each being in contact with two surfaces, the horizontal surface and the inclined surface of the guide. The larger collar is pressed up against the front end of the guide. The bar B_1 is secured at right angles to the bar B, and carries at its outer end the testing screw.

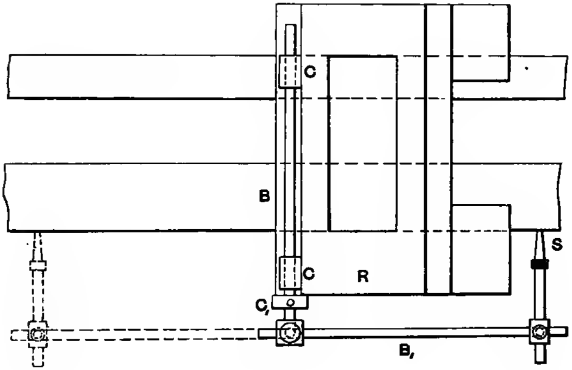


Fig. 28.—Alternative method of testing slide-rest saddle guides.

The cross slide is at right angles to the bed guide when the testing screw presses equally against the surface in the two positions shown for any one setting. If it is desired, for any purpose whatever, to throw the cross slide out slightly so that surfaced work will be slightly concave, the setting and testing can be effected by making use of a sheet of hard-sized writing paper about 0·002 or 0·003 in. in thickness between the bed and the end of the screw when the latter is on the left.

Modifications of the above involve the use of specially tested cylinders in place of the collars CC on the bar B, and also the substitution of a micrometer head for the testing screw S. In the latter case exact dimensional determinations can be made.

Another point which requires checking has reference to the horizontality of the cross guide in a transverse direction. This test is usually made by means of a spirit-level after the bed has been levelled transversely; it is, of course, of no use to test the saddle guides until one knows for a certainty that the bed is horizontal transversely.

6. **Finished Lathe Tests.**—When the various parts of a lathe have been assembled in their places, and the lathe is ready for service, it is the usual practice to put it under test so as to determine the inaccuracies (if any exist) in the various parts of the machine.

Before dealing with the actual tests which are carried out, it will be as well to consider the principal essential characteristics of a finished lathe which has some pretensions to accuracy.

These characteristics may be stated as follows:—

1. The axes of the driving spindle and loose headstock mandrel should be parallel to the top surface of the bed and to the ways or guides of the bed; if this condition is satisfied, they will be parallel to each other.

2. The axes of the centre sockets in the driving spindle and loose headstock mandrel should coincide with the spindle and mandrel axes.

3. The conical point of the centre in the driving

spindle should run true; that is, its axis should coincide with the axis of the spindle.

4. The axis of the point of the loose headstock should coincide with the mandrel axis.

5. The threaded nose of the driving spindle should run true; that is, its axis and the spindle axis should be coincident.

6. The cross slide of the slide rest should be exactly at right angles to the length of the bed, which, of course, should be parallel to the lathe axis, the latter being the line which joins the extreme points of the two centres when these are in their normal positions.

7. The screw thread of the leading or master guide screw should be uniform and accurate within certain well-defined limits.

If all these conditions are satisfied in a lathe, it is capable of turning out accurate work in the operations of turning (both sliding and surfacing), boring, and screw-thread cutting.

Some of the tests already described, which are made during the erection of the lathe, check several of the points enumerated above. There are, however, several tests which are necessary when the lathe is in a finished condition before an inspector or prospective buyer is justified in passing the machine, though in many cases reliance is placed on only one or two tests.

Centre Socket Tests.—In these tests it is necessary to use a special test bar. The form of this bar is shown in Fig. 29. It is a round solid bar or arbor, the greater portion of which is cylindrical or uniform in diameter. The remaining portion is tapered, the taper formed on this portion (which is the shank of the bar) being exactly the same as the taper of the centre socket or hole in the spindle or mandrel. The length of the cylindrical portion or body of the bar should be about 12 in., and its diameter from 1 to 2 in. The bar should be very carefully and accurately made, the axes of the two portions being coincident. Preferably, it should be finished by grinding on a reliable machine.

To test the accuracy of the tapered hole or centre socket in the driving spindle, the tapered shank of the test bar should be inserted in the socket, special care being taken to see that the surfaces of the two parts are quite clean and free from particles of dirt or metal. The contact finger or button of a test indicator or the spindle of a micrometer screw gauge head mounted on a pillar and resting on the lathe bed is placed against the surface of the bar towards its outer end (in fact, as near that end as possible). The contact may be on the side, bottom, or top of the bar as may be the most convenient. Indicator or micrometer readings are taken every quarter of a revolution of the bar, the spindle being, of course,

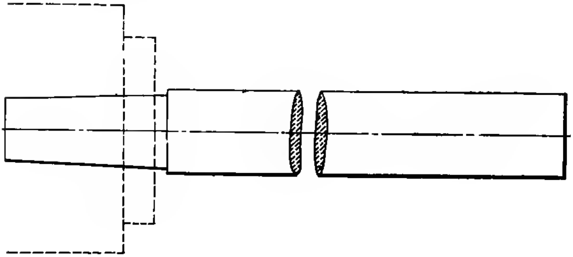


FIG. 29.—Centre test bar.

turned by hand. If there is no difference between the four readings, the centre hole or socket has been accurately formed.

Another method of carrying the micrometer screw gauge head is shown in Fig. 30. In this case, the head is secured to the outer end of a curved arm by means of a swivelling screw and locking nut. This arrangement admits of the head being placed in any convenient position irrespective of the position of the arm. This arm is secured at its other end to a shank by a bolt and nut, and is so connected that it can be readily swivelled about the stud whenever a change of position of the arm is required. The shank in this test is held rigidly in the slide-rest tool post. The spindle of the micrometer head is presented to the test bar or arbor in a horizontal position, as is indicated in Fig. 31, the *modus operandi* in this case being precisely that as described above, that is, a

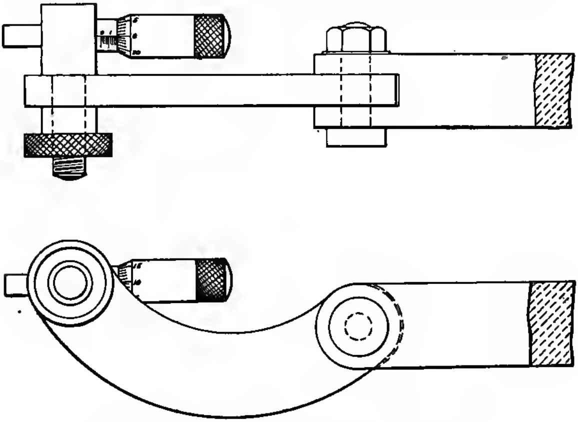


Fig. 30.—Adjustable micrometer test apparatus.

reading of the gauge is taken every quarter of a revolution of the test bar.

It should be observed that the possible centre socket error may be either one of two different kinds. In the first place, the axis of the socket may be parallel to the spindle axis, but not coincident with it, as is shown in Fig. 32. In the second place, the axis of the socket may be not even parallel to the spindle axis, but inclined at a small angle to it, as is indicated in Fig. 33. Whatever be the immediate cause of the inaccuracy, however, the above method will enable the magnitude of the error to be measured, though, to magnify it in the case of the oblique (that is, the second) form, it is necessary to employ a test bar of a length not less than 12 in.

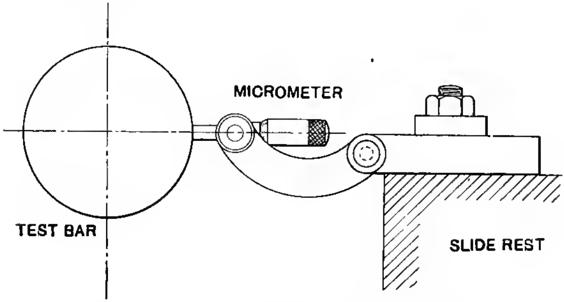


Fig. 31.—Method of testing lathe centre socket.

The loose headstock centre socket is tested in a manner somewhat similar to the above. The test bar is socketed in the mandrel in the ordinary way in place of the centre. The key which connects the mandrel to the barrel is then removed, thus admitting of the former being revolved in the latter. The accuracy of the socket is then tested as above, a reading being taken every quarter of a revolution of

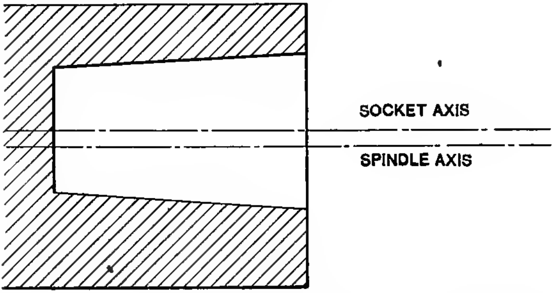


Fig. 32.—Diagram illustrating one form of centre-socket inaccuracy.

the mandrel. If they agree, the socket has been formed accurately.

A simpler test to determine the degree of parallelism (as distinct from obliquity) can be made by taking readings at two different places in the length of the test bar, the test indicator or micrometer pillar base

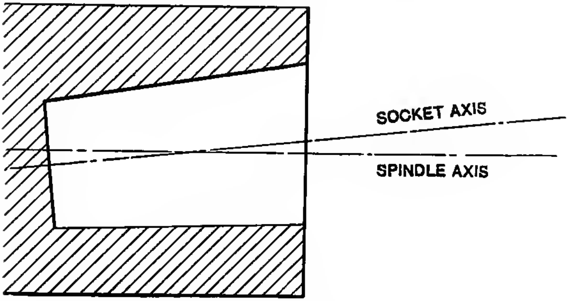


Fig. 33.—Diagram illustrating another form of centre-socket inaccuracy.

being moved along the bed guides in a direction parallel to the lathe axis. The first test is, however, generally the better of the two, since by means of it, the socket is tested for both kinds of inaccuracy.

Centre Tests.—These are readily made by means of a test indicator held in a fixture or the slide rest. The driving headstock centre is inserted in its socket in the driving spindle and the latter set in motion. The contact finger or button is then brought up and placed in contact with the conical point of the centre. If the point of the centre runs true, there will be no movement of the pointer or needle of the indicator. This test should be made for at least three positions of the centre in its socket, and in each test no eccentricity should be indicated.

The test on the loose headstock centre is performed in a similar manner, the mandrel being rotated by hand through the handwheel. To obtain satisfactory results in this case, however, considerable care has to be exercised to prevent any longitudinal or end movement of the mandrel in the barrel. This can be effected by locking the mandrel on the screw or to the handwheel in some way, such as, for example, by means of a nut. It will be obvious that in this test the key connection between the mandrel and the barrel must be removed.

To test the relation between the two headstock centres the test which is illustrated in Fig. 34 can be employed. In this two specially ground accurate centre plugs are used. These are exactly alike and fit respectively in the centre sockets of the two headstock spindles. The outer portions of these plugs are perfectly parallel and equal in diameter, the extreme ends being perfectly plane and recessed, as shown. The test consists in bringing the ends of the two plugs together. If they coincide, as in the figure, the centres and their sockets are passed as accurate. Any inaccuracy can be actually measured by means of a test indicator or a micrometer screw gauge, preferably the latter. If an ordinary outside micrometer gauge is used, the inaccuracy in any plane is determined as follows: Let D = the actual diameter of each of the test plugs; d = the reading of the gauge; and A = the inaccuracy. Then :—

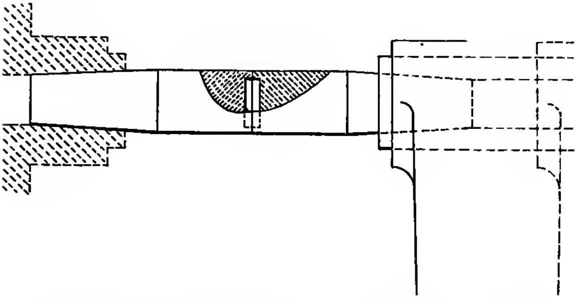


FIG. 34.—Testing relation between lathe headstock centres.

$$A = d - D \qquad . \qquad . \qquad . \qquad . \qquad . \qquad (11)$$

The inaccuracy in this case is the distance between the axes of the two centres in the plane chosen. It is indicated in Fig. 35, which represents this case.

To make this test complete the loose headstock mandrel should occupy each of its two extreme positions (as shown in Fig. 34), and the results of the two tests compared. Any difference between the results will indicate the obliquity of the axis of the headstock barrel, if the accuracy of the bed guides is beyond suspicion.

In Fig. 36 is shown another method of testing the relation of the loose headstock centre to the driving headstock centre. In this test the assumption is made that the centre socket in the loose headstock mandrel has been accurately located. The apparatus required consists of a short centred rod in which a bent arm carrying a micrometer head is secured.

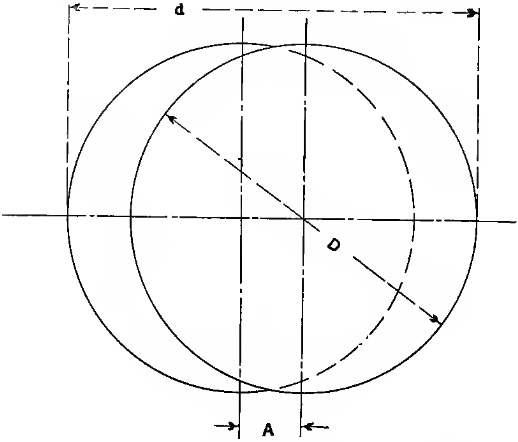


FIG. 35.—Diagram illustrating centre-test conditions.

The positions of the bent arm and micrometer head are adjusted until the end of the micrometer spindle just touches the upper surface of the loose headstock mandrel. The reading of the micrometer is noted, and the rod then turned through 180 degrees. The reading of the micrometer for this position is also noted. This process can also be applied to obtain side measurements. If the two centres are exactly in line there will be no difference between the two readings. If there is a difference, however, the alignment error for the given position of the loose headstock can be determined in the following way: Let R = the difference between opposite readings of the micrometer, and E = the alignment error at the extreme points of the centres. Then, practically :—

$$E = \frac{R \times X}{2(Y - X)}, \quad . \quad . \quad . \quad . \qquad (12)$$

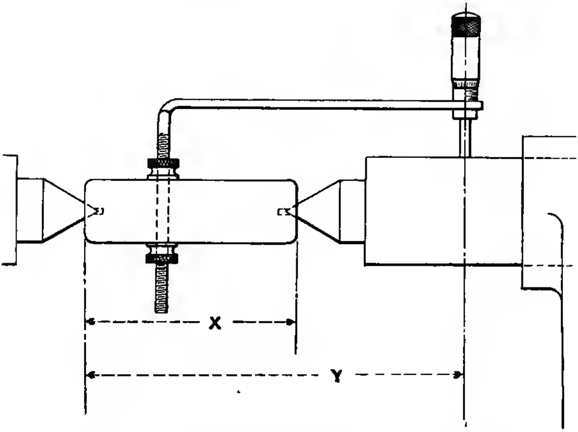


FIG. 36.—Alternative method of testing relation between lathe headstock centres.

X and Y being the dimensions indicated in the figure.

A method of testing the loose headstock centre with respect to the axis of the driving spindle or the axis of the lathe is indicated in Fig. 37. A short test bar or block, T, is secured in a chuck mounted on the driving spindle. The loose headstock centre is then brought up to T, and caused to penetrate it when this is stationary. The centre is then moved away from T to make room for the ground bar B, which is accurately pointed at one end and provided with an accurate centre hole at the hole at the other. When the lathe driving spindle is set in motion, if the loose headstock is correctly aligned the bar B from end to end will run true, and if a test indicator I is placed in contact with this bar as shown, its reading will not show any change. If, however, the centre is not correctly disposed, the indicator reading will vary, the magnitude of the variation depending upon the degree of error and the position of the indicator with respect to the end of the bar. When the indicator is placed at or near the pointed end of the bar, the alignment error is practically half of the difference between the maximum and minimum readings of the instrument. A micrometer screw gauge is not as useful as a test indicator in cases like this, since the action of a test indicator is

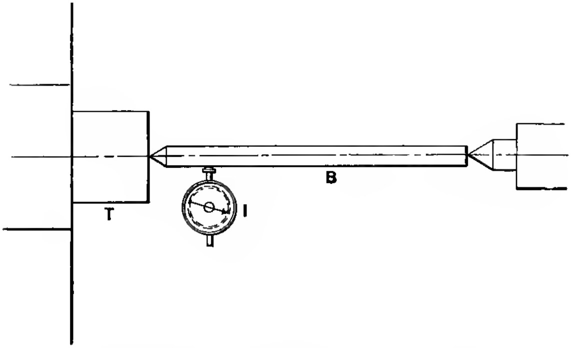


FIG. 37.—Method of testing loose headstock centre.

practically automatic, whereas adjustments of the micrometer gauge have always to be made by hand.

A simple test of the alignment of the two centres involves the use of a plain ground cylindrical test bar with a centre hole at each end accurately formed. This bar is placed between the centres, and a micrometer or test indicator is moved along the bed or in the slide rest in contact with the bar. Any difference between the end readings represents the alignment error of the centres. The length of the test bar should be equal to about six times the height of the centres. If the driving-spindle centre runs true the alignment-test result in this case will not be affected, but if there is any inaccuracy (however slight) in the running of the centre, it would be advisable to set the driving spindle in motion and take the average of the maximum and minimum readings of the indicator or gauge at each end. If the two averages are equal, it may be safely assumed that the average alignment of the centres is correct.

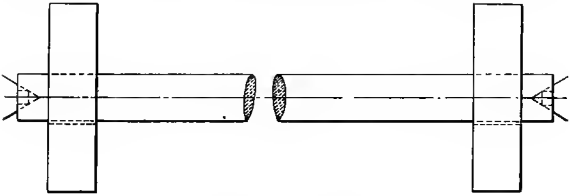


FIG. 38.—Test bar.

In place of a ground test bar of uniform diameter, sometimes a test bar as represented in Fig. 38 is employed. As will be seen, near each end of the bar a short collar is secured, each of these being hardened and ground exactly to the same diameter (between

3 and 6 inches). They are also concentric with one another, so that when the bar is rotated they both run quite true.

It should be observed that when the indicator or micrometer is secured to the slide rest, the alignment of the centres is with respect to the direction of motion of the slide rest; when, however, the motion of the indicator or micrometer gauge is controlled directly by the guides of the bed, the alignment is with respect to those guides. In an accurately built lathe, of course, these two alignments are identical.

Spindle Nose Test.—This test can be readily performed by means of a test indicator or micrometer screw gauge, preference being given to the former. This instrument is held rigidly, as, for example, in the tool post or clamps of the slide rest, and the end of the contact finger, button, or spindle is placed in contact with the outer surface of the nose or flange which is formed immediately behind the threaded part of the nose.

When a test indicator is used, it is possible to perform the test with the spindle in continuous rotation; when a micrometer gauge is in use, however, it is necessary to move the spindle round intermittently and to take readings only when the spindle is at rest. All the readings agree when the nose runs true.

Spindle Bearing Tests.—As a check upon the tests made on the driving headstock during construction, it is desirable to test the spindle bearings when the lathe is finished. The function of these tests is to determine whether the common axis of the two bearings is coincident with the lathe axis, that is, the imaginary line joining the two centre points.

The first test checks this relation so far as the vertical plane is concerned; the second test deals with the relation in connection with the horizontal plane.

The apparatus used is the same in each test. It is illustrated in Fig. 39. This apparatus consists of a long cylindrical test bar, centred at its two ends and mounted between the centres of the lathe. In this bar is fixed a compound bar or arm (similar to the one shown in Fig. 30), in the outer end of which is secured, by the method shown in Fig. 30, a micrometer head or test indicator. The method of gripping the compound arm in the bar should, if possible, include the use of one or two tapered wedges or a cone, so that the grip is obtained without damaging or distorting either the bar or the arm. On the nose of the driving spindle a face plate is mounted, and across the front face of this a light cut is taken. The

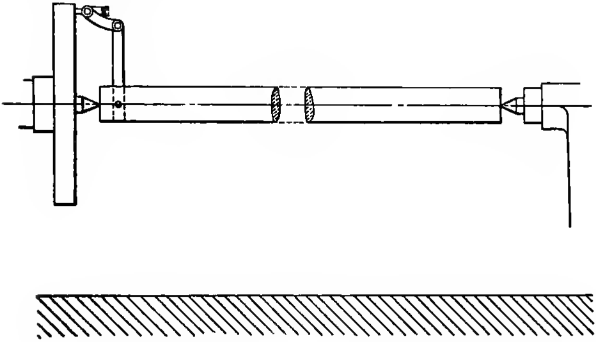


FIG. 39.—Micrometer test bar.

contact finger of the indicator or the spindle of the micrometer is placed in contact with the face plate at its highest point, and the reading noted. The central bar and indicator or micrometer are then swung through 180 degrees and another reading is taken, that is, when the indicator or micrometer occupies its lowest position. If the two readings agree, we know that the axes of the two bearings are at the same height above the shears of the bed. If there is a difference between them, however, the difference in height between the two axes can be determined as follows: Let x = the difference between the two readings; y = the difference between the heights of the two axes; l = the diametral distance between the two points on the face plate at which the readings are taken; and b = the horizontal distance between the vertical centre lines of the two bearings. Then, by applying the principle of the proportionality of similar triangles, we obtain that—

$$y = \frac{x \times b}{l} \quad . \quad . \quad . \quad (13)$$

This is the vertical error of alignment of the bearings.

The horizontal or transverse error of alignment of the bearings is obtained in precisely the same way. Readings of the indicator or micrometer are taken at the front and back of the test bar at points which are diametrically opposite, or approximately so. The actual determination of the error then proceeds on the above lines.

Slide Rest Movement Tests.—These tests have respect to the relation which exists between the movements of the slide rest and the axis of the lathe. Their chief function is to check the accuracy of these movements, though incidentally some of these tests check the accuracy of the dispositions and relations of other parts.

These tests in nearly every case involve the taking of a light cut in either a longitudinal or transverse direction, and may be divided into longitudinal and transverse tests.

The longitudinal tests deal with the relation between the axis of the lathe and the movement of the slide rest as a whole on the bed. This relation should, of course, be one of parallelism.

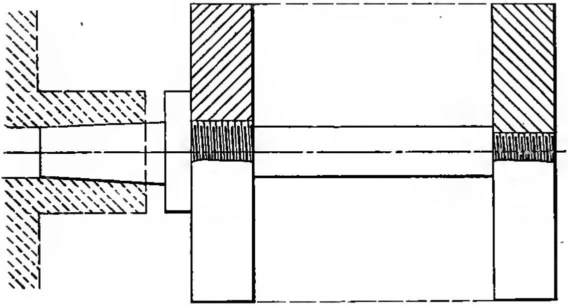


FIG. 40.—Lathe test mandrel.

In one form of this test, a test mandrel is used which is socketed in the driving headstock spindle, as is indicated in Fig. 40. On this test bar are screwed two cast iron or steel collars, these pressing against shoulders formed on the bar. The outer end of the bar is free. A very light cut is taken over the two collars with the same setting of the turning tool, the cut having to be a light one for two reasons: first, so that a smooth surface may be produced on each collar; and second, so that practically no wear shall occur on the cutting edge of the tool to affect the diameters of the collars. The diameters of the collars are measured by means of a micrometer screw gauge, and if they are found to be equal it is assumed that the slide rest travels in a direction parallel to the lathe axis, and that the lathe will turn parallel work and bore a parallel hole. In some cases three collars are mounted on the mandrel and treated exactly as the two above. This arrangement is regarded as being superior to the two-collar method, since, by means of it, it can be determined whether the lathe will bore a hole having a concave, convex, or parallel section, taken in a longitudinal direction.

Variations of the above consist in driving the test mandrel by means of a chuck or carrier and driving plate, and also in supporting the outer end of the mandrel on the loose-headstock centre.

Another method of testing this relation for accuracy is illustrated in Fig. 41. This method is an electro-micrometrical one, and involves the use of a micrometer screw head and an electric circuit. The electric circuit contains a battery of cells, B, and a telephone receiver, buzzer, or electric signalling bell, R. The micrometer, M, is insulated from the lathe, being held in the slide rest between vulcanized fibre washers. C is a rubbing contact or brush and is always in contact with the test bar, which is driven from the driving spindle in one of the ways detailed above and supported at its outer end as shown in the figure. When the micrometer comes into contact with the surface of the test bar, the circuit is completed and a sharp click is heard in the receiver.

In this test, the test bar is set to run true in the lathe, as shown in Fig. 41. The micrometer is then set to give the click in the receiver, and its reading is noted. After this it is moved along with the slide rest to the other end of the bar, and the same process is passed through again. If there is any difference between the two readings it is directly proportional to the error in the movement of the slide rest. By this method it is possible to obtain a reading at any point in the length of the test bar, and so determine whether there are any local errors or not.

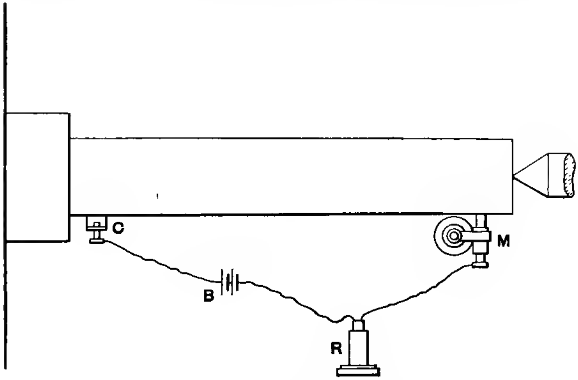


FIG. 41.—Electro-micrometrical accuracy test.

In one practice the method of testing the cross slide of the slide rest for rectangularity involves the use of a special test piece, as illustrated in Fig. 42. This test piece is provided with an annular channel which forms two annular rings in the piece, and it is mounted directly on the nose of the driving spindle.

A very light cut is taken over the ends of the rings. The surfaces formed are then tested in the manner indicated by means of a tested straight edge.

This straight edge is placed diametrically across the test piece, and if the cross movement of the slide rest is at right angles to the spindle and lathe axis, the straight edge will touch the annular surfaces at the points indicated by the arrowheads. If there is a space between the inner ring and the straight edge it is obvious that the lathe will face work hollow or concave, whereas if there is a space between the outer ring and the straight edge the work will be faced convex or round. The former condition is the better of the two and is, in some cases, actually aimed at, the variation, which amounts to about 0·002 in., being detected by the use of hard thin paper between the plate and the straight edge.

In a slightly different form of this test a test face plate is used. This is mounted directly on the driving spindle, as shown in Fig. 43 at P. A light cut is taken across the face of this, and the blade of a tested square, S, is then brought up against it. The stock of the square is placed on the bed of the lathe, B. This test checks the relation of the cross slide to the bed of the lathe, and it also checks the relation of the axes of the two bearings of the driving headstock to one another.

Neither of the above tests is a quantitative one. In such a test a micrometer gauge, or indicator, which will read to at least 0·001 in., must be used. In this case, also, a face plate which is lightly faced is made use of. After the facing operation the measuring instrument is secured to the slide rest, and two readings taken with respect to the surface of the plate, one at a point at or towards the front, such as A (Fig. 44, which is a plan view), and the other at a point A, diametrically opposite to it and at

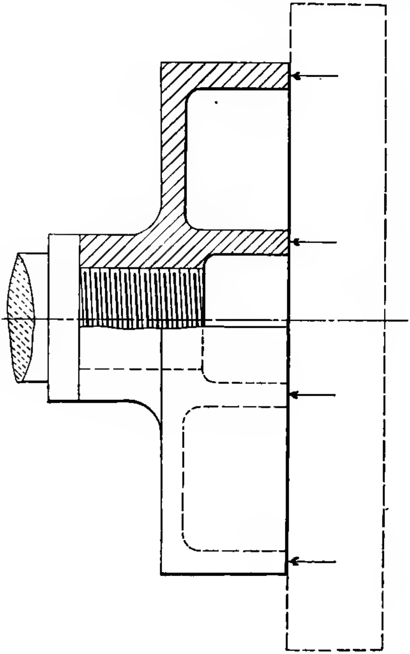


FIG. 42.—Test face plate.

the same distance from the centre of the plate. When the cross slide is being moved across the saddle of the slide rest, as shown by the arrow D, the latter must be gibbed securely to the bed so as to admit of no longitudinal movement. The two readings should either agree or differ only by about two or three thousandths of an inch, the back reading being the smaller of the two.

The explanation of the principle of this test is that the straight line joining two corresponding points in the surface of the plate, which are diametrically opposite and at the same distance from the centre of the plate, is a line which is exactly at right angles to the axis of the driving and lathe. Therefore, if the cross slide is disposed across the bed of the lathe at any angle other than a right angle, the direction of its movement will be slightly inclined to this line, the obliquity being indicated in the manner stated.

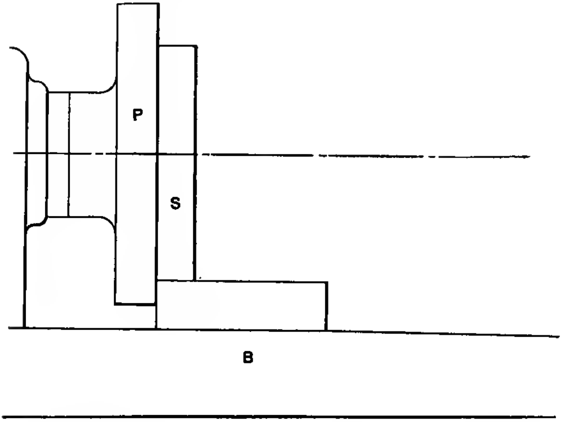


FIG. 43.—Method of testing transverse accuracy of slide rest.

If it is not convenient or desirable to take a light cut over the face plate in the above test, it is possible to make the tests with a face plate which may be assumed not to run true whether it does or not.

In this case the two readings are taken at exactly the same point on the face plate, the latter being swung through half a revolution before the second reading is taken. This method will give results equal to those obtainable in the above tests.

7. **Tests on the Bed for Wear.**—The wear which has occurred on a lathe bed may be determined qualitatively by means of a straight edge. To determine it quantitatively in both the horizontal and vertical planes, the apparatus shown in Fig. 45 can be used. This apparatus consists of a length of steel wire of a small uniform diameter, such as, for example, steel piano wire. This is attached at its two ends to the centres of the lathe in the manner indicated in the subjoined diagram in Fig. 45, and then stretched tight between them. Micrometer or indicator readings will show whether there is any wear at one place, such as A, with respect to any

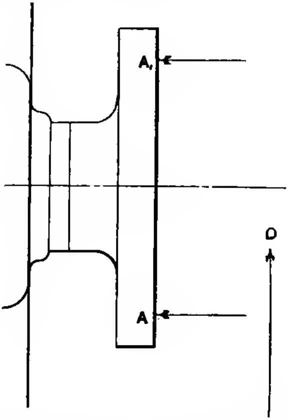


Fig. 44.—Diagram illustrating slide-rest test.

other place, such as B, whilst intermediate readings (both horizontal and vertical) will indicate the extent of the curvature (if any) of the bed and guides due to wear. In these tests, which are really old-lathe tests, the micrometer or indicator may be moved along the bed in a special holder, or it may be held in the slide rest and moved along therein.

Several of the tests which have been described in connection with the aligning of the centres and slide rests are also applicable in this case.

8. **Lead Screw Tests.**—The chief function of these tests is to check the accuracy with which the lead screw thread has been formed. In other words, in these tests the pitch of the lead screw, which is usually a single-threaded screw, is checked from point to point in the length of the screw.

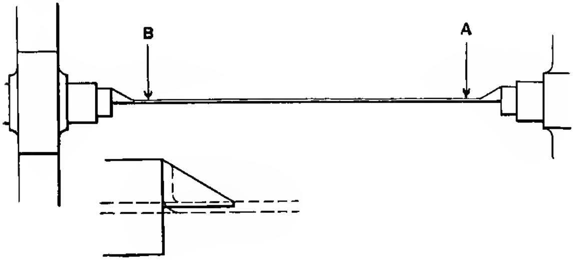


FIG. 45.—Wire test for wear of lathe bed.

These tests can be performed with the lead screw *in situ* in the lathe or in a special testing machine. In the majority of cases the former is adopted.

The requirements of these tests are means whereby the advance of the saddle of the slide rest for a definite revolution of the lead screw can be obtained. The measuring instrument which is generally employed in the lathe tests is a micrometer gauge of some form. In one form of the test a plain micrometer head is mounted in a fitting which is capable of sliding on, and being secured rigidly to, the bed of the lathe. The axis of the micrometer spindle is disposed parallel to the axis of the lathe, and the end of the spindle is arranged to work in contact with a specially planed and scraped surface on the side of the saddle. Any movement of the saddle with the range of the instrument can be measured, and as soon as the limit of the capacity of the instrument has been reached the micrometer fitting can be moved along the bed and the micrometer spindle set afresh. The definite rotation of the lead screw can be obtained by making use of the face or the surface of the rim of a face plate. On this face or surface a fine line is scribed by means of a surface gauge or scribing block resting on the bed of the lathe. If equal change gear wheels are employed between the driving spindle and the lead screw—then for each turn of the face plate a complete and exact revolution will be made by the lead screw. By this means variations in pitch per revolution can be determined. If a closer test is required, the surface of the face plate has to be divided into two or four equal parts on a dividing machine or universal milling machine. In each case, of course, the needle of the scribing block or surface gauge is made use of to determine the amount of revolution of the face plate and lead screw, that is, whether it is a whole or a fractional turn.

Another method of measuring the advance of the saddle is indicated in Fig. 46. In this case two fixtures or plates carrying vertical cylindrical pins are used. One of these is fixed rigidly and permanently (for the purposes of this test) to the saddle of the slide rest. The other is fitted on the bed and arranged so that it can be moved into any position on the bed in a longitudinal direction and secured rigidly therein. The axes of the two pins are arranged in such positions that their common plane is parallel to the axis of the lathe. Between these two pins an inside micrometer gauge or test indicator is used to determine the movements of the slide-rest saddle. Otherwise, the *modus operandi* is the same as in the above case.

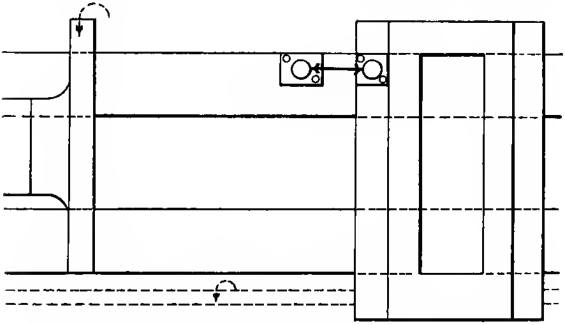


FIG. 46.—Diagram illustrating lathe lead-screw tests.

To obtain local inaccuracies in the pitch of the screw it is not necessary to make each setting of the micrometer bear a fixed and definite relation to the one immediately preceding it, but if it is required to obtain the general or cumulative inaccuracy in the pitch of the screw for its whole length, it is absolutely essential to move the micrometer or the gauge cylinder an exact amount each time. In this latter case the exact amount can always be obtained by means of the micrometer. For the determination of the general or cumulative inaccuracy a long adjustable inside micrometer gauge is far more suitable than a short one, since its use involves fewer movements of the movable gauge pin on the bed of the lathe.

In the practice of some works the lead screw of a lathe is tested for accuracy of pitch by placing it between the centres of another lathe (the accuracy of whose lead screw in regard to pitch is a matter beyond dispute) and connecting up the two as for the cutting of a screw with a thread having the same pitch as the one to be tested. A micrometer or test indicator is used to determine the pitch inaccuracy, if any exists. This instrument is secured in the tool post or clamps of the lathe, the spindle or contact finger of the micrometer or indicator being held against the edge or side of the screw thread. Assuming the pitch of the testing or master screw to be accurate from end to end of the screw, this test will show both local errors and the general or cumulative error in the pitch of the thread.

An improved method is to place the screw in a special lead-screw testing machine, one form of which works on the above principle, but which is generally more accurately built than an ordinary lathe. Moreover, in such a machine are usually incorporated refined methods of measuring small distances, these being chiefly microscopic in character. In another form standard end or length gauges are employed to obtain the required measurements.

Concerning the degree of inaccuracy of pitch which is permissible, it may be stated that this varies considerably. Thus, in the practice of one firm of lathe builders a cumulative error of 0·01 in. in a length of 8 ft. is allowed, whilst in the practice of another firm it is much less, being only 0·005 in. in 6 ft.

9. **Slide Screw Tests.**—In these tests the accuracy of the threads of the cross and top slide screws is checked. They are, however, frequently omitted from works' schemes of tests. They can be performed in any of the ways indicated above, the performance of the tests being facilitated when the screws are equipped with micrometrically graduated dials.

10. **Test Report on Lathe.**—In an up-to-date machine-tool manufacturing works, a report is given by the inspector or tester on every lathe which passes through his hands. This report is arranged so that reference to it is an easy matter at any time, and so that an examination of it by a prospective buyer will reveal the more salient features of the lathe in regard to this question of the accuracy of parts.

The form of the report varies considerably with different firms. The outline of one such form, which combines the best features of reports of this kind, is given in the Appendix. It should be understood, however, that the particular form of the report in each case depends largely upon the nature of the tests which are applied and upon the way in which they are carried out. Therefore, since the test methods in different works vary, it is only to be expected that inspection test reports vary also in their contents and their arrangement.

TURRET LATHE TESTS.

The principal special test for accuracy in this case is that in which the position of each of the holes in the turret head is tested with respect to the position or positions of the driving spindle or spindles of the machine. Practically all the other tests are similar to those which have been dealt with in connection with the testing of the engine lathe.

In one form of this test it is necessary to use a ground hardened steel test plug which just fits in each of the turret holes. A simple micrometer or test indicator is fitted at the end of a bent arm (as indicated in Fig. 47), this arm being carried in an arbor which is secured to the driving spindle in some way as, for example, through the medium of a lathe chuck. The spindle is rotated by hand and carries the micrometer or indicator round with it, enabling a number of readings of the instrument to be taken

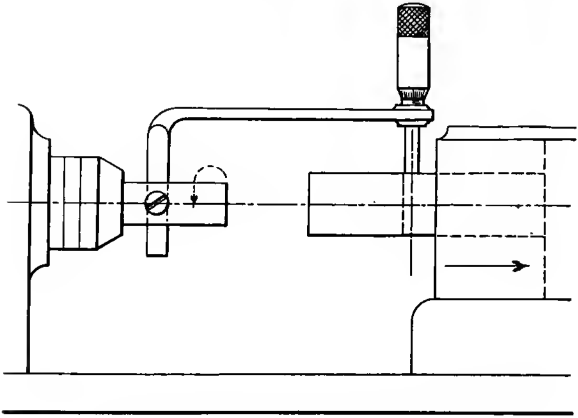


FIG. 47.—Turret lathe accuracy test.

in the revolution. Four places are usually selected, these being situated at the top, bottom, front, and back of the plug. The vertical error is one-half of the difference between the first two readings and the horizontal error one-half of that between the others.

To test the accuracy of disposition of the turret-head slide, corresponding readings are taken with the turret head in another position, as indicated by the dotted lines in the figure, the movement of the head being in the direction of the arrow-head.

MILLING MACHINE TESTS.

The nature and function of milling machine tests vary somewhat with the type of the machine, but, speaking generally, the same fundamental principle underlies them all.

1. **Levelling Tests.**—These tests apply chiefly to the tables of these machines, and are usually performed by means of one or more spirit levels, the method of application being the same as in the case of lathe-bed levelling tests, though any of the other methods applicable in the latter case are also applicable in this. In each case, the table should be tested for both longitudinal and transverse horizontality, with the table in its middle position on the saddle or clamp bed, and the latter in its middle position on the top of the knee or bed, whichever form of support is used.

2. **Driving Spindle Tests.**—The first of these tests is that of coincidence of the axes of the driving spindle and the cutter-arbor socket which it contains. This is performed by means of a special mandrel or proof bar and test indicator or micrometer screw

gauge, the mandrel or proof bar being socketed in the spindle and the indicator or micrometer being mounted on the table of the machine and kept in one place thereon. The test is similar to the corresponding lathe test.

The second test has reference to the relation between the direction of the driving spindle axis and the top of the table. One method of performing this test is indicated in Fig. 48. In this test, if the spindle is horizontal and the cutter arbor socket has, by the previous test, been shown to be correctly disposed in the end of the spindle, a test or proof bar with or without an outboard support in the overhanging arm is used. This bar should be of uniform diameter and admit of the use of a spirit level on it, as shown in the figure. The comparison is obtained by placing the spirit level also on the table transversely and

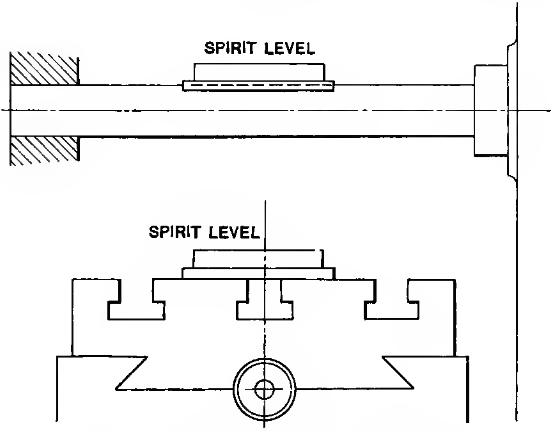


FIG. 48.—Milling machine test for parallelism.

noting the two results. In the second form of this test, a test indicator or micrometer is used, this being moved across the table between the latter and the test bar, either with or without parallel test strips underneath it. In ordinary cases the test bar and the table top should be nearer to one another at the outer end of the bar than at the other end by, say, one or two thousandths of an inch.

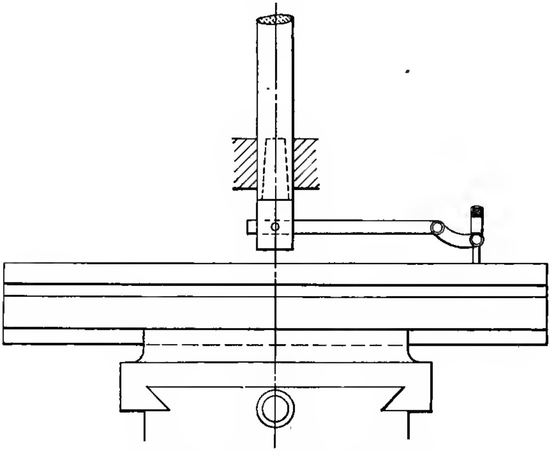


FIG. 49.—Milling machine spindle test.

When the driving spindle is vertical (as in the case of vertical milling machines), a test bar which carries a micrometer head or test indicator is used as indicated in Fig. 49. It will be noticed that the test bar is socketed in the spindle, so that, when the spindle is rotated, the micrometer or indicator is carried round with it. Readings taken at points which are diametrically opposite will indicate the degree of error (if any exists) in the alignment of the spindle.

Another test which is sometimes performed checks the alignment of the outboard bearing support in the overhanging arm with respect to the driving spindle. In this case, the method adopted is similar to that which is shown in Fig. 47, the turret head being replaced by the end of the overhanging arm of the milling machine.

3. **Cutter Arbor Tests.**—The first of these tests checks the truth of running of the arbor when it does not carry either washers, nuts, or cutter. A test indicator used as in the corresponding spindle test is all that is required.

The second test is performed when the washers and nuts are in their respective places, the nuts being tightened up. If there is any inaccuracy in the thickness or fitting of the washers, the effect of tightening up the nuts on the arbor will be to bend the arbor more or less slightly, and so throw it out of truth when running. Readings should be taken at the ends of the arbor. A slight modification of this test is to mount the arbor between fixed centres and then test for running truth by the application of a test indicator or micrometer, the difference between the maximum and minimum readings at any one place being equal to twice the eccentricity or error, as has been already explained in connection with the running-truth tests on lathes.

4. **Knee Tests.**—The knee of a correctly built milling machine possesses three features. First, the knee guide on the column of the machine should be exactly vertical so that the movements of the knee

are really vertical movements; second, the upper surface of the knee should be horizontal in every direction so that the saddle is always moved on the knee horizontally; and, third, the actual guiding faces on the top of the knee are parallel to the axis of the driving spindle.

The first point is tested by means of a cross or transverse spirit level, a try square, or a dead-square triangular frame carrying a spirit level vial is one of its limbs. This is placed against both the front face of the column and the dovetailed guide faces inclined to the front face. A necessary preliminary to this test is, of course, the levelling of the table and spindle.

Another method of testing this point is to use a try square against the front face of the column and a test indicator or micrometer between the horizontal limb of this and a special arbor in the spindle of the machine.

The second point is tested by means of an ordinary spirit level placed transversely on the top of the knee when the saddle is close in and right out. If this test does not show that this guiding surface is horizontal then, though the top of the table may be horizontal transversely, the machine will not be capable of turning out accurate work under all ordinary working conditions. To secure this end this guiding surface must be parallel in a vertical plane to the common axis of the driving spindle and cutter arbor, which is assumed in this test to have been checked for horizontality.

The third point can be tested by means of a test indicator or micrometer mounted in a fitting which is designed so as to be capable of being moved along the saddle guides. This instrument is used in conjunction with an accurately ground test bar of uniform diameter which is socketed in the driving spindle and supported in the overhanging arm.

5. **Saddle and Clamp Bed Tests.**—The table guides in the saddle of a plain machine, or the clamp bed of a universal machine, should be horizontal in the direction of the length of the table and situated exactly at right angles to the axis of the spindle.

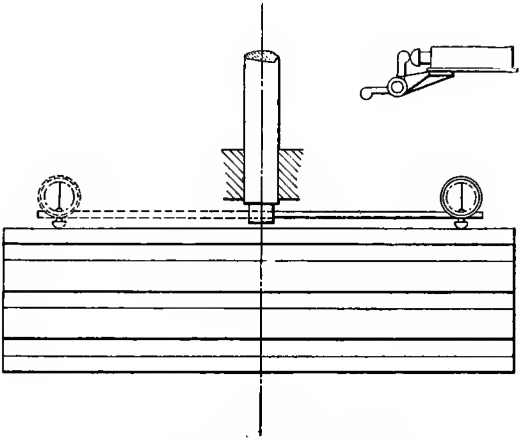


Fig. 50.—Milling machine table test.

The first point is tested with a spirit level in the usual way.

The second point is dealt with in a way similar to either of those shown in Figs. 49 and 50. A test indicator or micrometer is secured to the end of an angle bar socketed in the end of the driving spindle. The spindle is moved through a part of a revolution, and a reading on each side of it is taken against the guides. If the rectangular condition is satisfied, the two readings will agree.

6. **Table Tests.**—The relation of the edge of the platen or table to the axis of the spindle can also be tested in the manner indicated in Fig. 50, as can the relation between the longitudinal slots in the table and the spindle axis. In these latter tests recourse may have to be made to the use of an external contact lever on the indicator (as shown in the small view) in order to enable contact to be made with the sides of the slots, though the use of a specially planed and scraped strip in the slot will obviate the necessity for this. In all these cases the readings taken on the opposite sides should agree.

There is another method of performing this test. It is, however, only applicable in those cases wherein the machine table is provided with a clamp or swivel bed. In such a case, the bed and table are swung through 90° from their normal position and clamped there, use being made of the scale on the bed to determine this position. A test indicator is then used in conjunction with a parallel arbor in the spindle to determine any inaccuracy which may exist.

In the case of universal milling machines the position of the slot which guides the dividing head and foot stock is of considerable importance. In practically every instance the tongues or wards on the undersides of the head and stock are disposed immediately below the centres, so that it is desirable to have one slot in the middle of the table to take these tongues. Furthermore, it is desirable to have the vertical axis of swivel of the trunnion or clamp bed, the common axis of the dividing head and footstock centres, and the common axis of the cutter arbor and

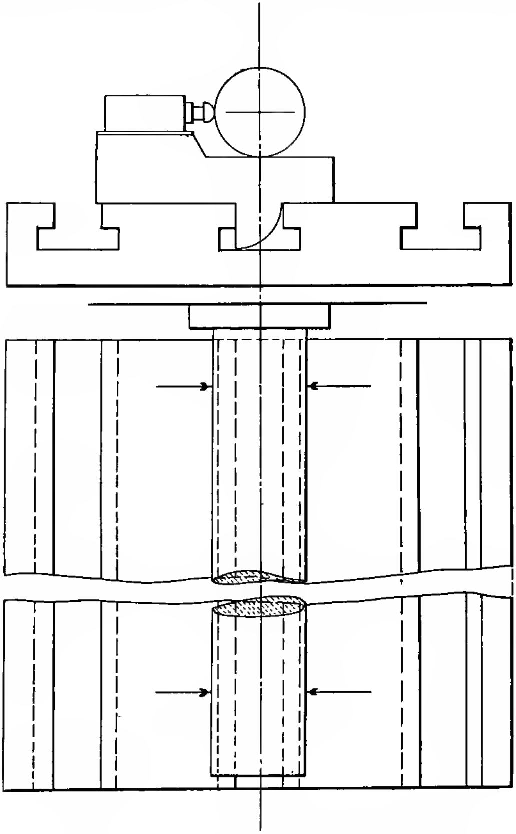


FIG. 51.—Test on milling machine table slots.

driving spindle so disposed that the first and second and the first and third intersect.

The absence of this condition in a universal milling machine results in restricted usefulness, since such a condition is one of the essential conditions for satisfactory, straight-forward, spiral, or, more correctly, helical cutting.

The alignment of the central slot can be tested by the method indicated in Fig. 51. In this method the table has to be swung through 90 degrees from its normal position. On the table is mounted a test indicator, to the base of which a hook projection is provided, this projection fitting against one of the sides of the slot, and so controlling the motion of the indicator along the table. In the driving spindle a test bar of uniform diameter is inserted, its outer end, where possible, being supported. Readings of the indicator are taken towards the two ends of the test bar, first on the one side and then on the other, as indicated in the figure. If the slot is in correct alignment all the readings will be alike. If not, the error of alignment at either end will be given by the difference between the two readings taken at that end, and it will be on the side of the greater reading.

The relation of the three axes of the clamp bed, spindle, and centres is indicated in the oblique diagram in Fig. 52 in which the line AB represents the vertical axis of the clamp bed, CD the horizontal axis of the driving spindle, and EF the horizontal axis of the table centres when the correct condition obtains. This is also represented in rectangular or orthogonal projection in the upper view of Fig. 53, the letters used in the two diagrams being identical. When, however, the axis of the table centres is offset the axis AB does not pass through the horizontal intersection of the axes CD and EF. In such a case the chain-dotted line E_1 F_1 would represent the table-centre axis in Fig. 52 whilst, as shown in the lower view in Fig 53, the vertical axis would pass through AB at a distance from the horizontal intersection of the other axes. The dimension x is the same in each diagram, and is the offset of the table-centre axis with respect to the vertical axis of the clamp bed, or vice versa.

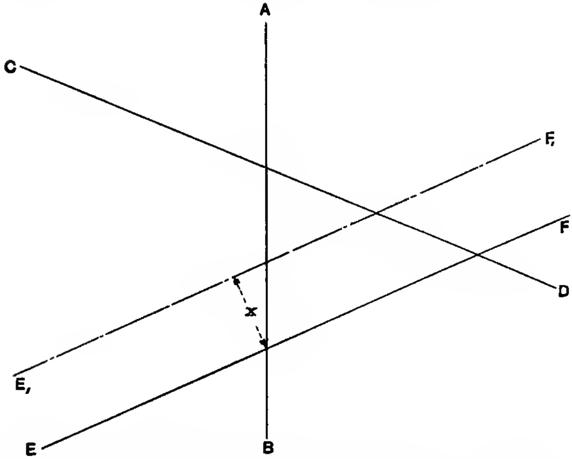


FIG. 52.—Oblique diagram showing relation between principal axes of milling machine.

The determination of this dimension can be affected by means of a sharp marking tool gripped between washers on the cutter arbor and arranged with its point just in the upper surface of the table. If the table is now swung round, a circular arc will be described on the table surface, the centre of which can be readily obtained by geometrical methods. The offset (that is, x) equals the perpendicular distance between this point and the centre line of the slot. Another method involves the use of a test indicator and cylindrical gauges which can be mounted on the table.

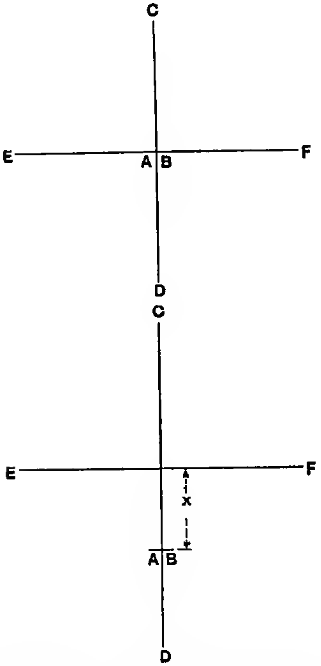


FIG. 53.—Orthogonal diagram showing relation between principal axes of milling machine.

If an odd number of slots is provided the offset will be nothing. If, however, the number is even, there may or may not be some offset—more usually there is. To give some indication of the magnitude of this dimension, it may be stated that in one case an offset of 1½ in. was found, this spoiling an otherwise good machine.

7. **Feed Screw Tests.**—These tests check the accuracy with which the screw threads were formed, and are of some importance since milling machines are generally equipped with micrometrically graduated screw dials to read to a thousandth of an inch, direct measurement being dispensed with. The reliability of such dials depend largely upon the accuracy of the pitching of the screws.

The methods of carrying out these tests which are applicable are practically those which are available in the case of the lathe, with, of course, modifications to suit the special circumstances of their application. All the three feed screws are important elements in the design of a milling machine, but the most important for general work is the vertical or elevating screw. Hence, special attention should be given to the testing of this screw.

8. **Feed Rack Tests.**—The test to determine the accuracy of the pitching of the feed rack and pinion (where such a combination is embodied in the design of the machine in place of the feed screw) is conducted on the same lines as are the screw tests, the revolutions of the feed handle or hand wheel being related to and compared with the rectilinear movements of the table. To obtain the latter, cylindrical pins and plug gauges or inside micrometer or micrometer head are used.

9. **Dividing Head Tests.**—The dividing head of a universal milling machine is probably the most important element in its design. Its function is two-

fold, being the support and the dividing or indexing of work. The tests which are made to determine the accuracy (relative or absolute) of the spindle, spindle socket, etc., of the head are not very different from those which are applied in similar cases in connection with the lathe and other machines.

Indexing tests are, however, different. They aim at testing the accuracy with which the dividing or indexing is performed, and check errors which occur in the indexing plates and in the gearing of the head. In all well-made dividing heads these errors are exceedingly small, since the indexing plates are engine divided and the gearing is of the worm type with the worm hardened and ground to shape and size.

One method of testing the worm for accuracy is to give the indexing handle a number of equal movements each covering an exact number of turns and to scribe a series of fine lines on the rim of a special testing plate mounted on the spindle of the head. When the complete forty turns of the indexing handle have been made, the first and the last scribed lines can be compared. If they coincide, there is practically no general error in the worm gearing. To check for local errors the testing plate has to be removed and the graduations tested in a special dividing machine.

As a variation of the above, a specially graduated testing plate and microscope, containing a graduated micrometer screen to determine the values of the local errors, can be used. This latter method also lends itself admirably to the testing of the indexing plates, though this work is better done on a special machine.

Another method is to use a special test plate on the spindle, this having 40 notches in its periphery. In conjunction with this a notch finger mounted in a fixture works. This finger is capable of being moved in a direction tangential to the plate periphery, its movements in this direction being measured by means of a dial gauge. The local error permissible is a movement of the finger of 0·0005 in., whilst the cumulative error in 40 turns must not exceed 0·002 in. on a 12-in. plate.

DRILLING MACHINE TESTS.

Many of the tests which have been already described are also capable of application in the case of the drilling machine, since many of the movements and the relations of the various elements of the machine are of the same general types as in the cases of the lathe and milling machine. An instrument for checking the disposition of a drilling machine table with respect to the driving spindle when no actual measurement is required is shown in Fig. 54. This instrument consists of a tapered body, S, which contains a small coiled spring. This spring presses on a concave die, which in turn presses on the spherical end of a scribing needle or awl, N. The end of the needle is prevented from falling out of the body by the screw cap C. The spring provides just sufficient frictional resistance at the joint to hold the needle in any position in which it is placed. The manner in which this tool is used is also indicated in this figure, T representing the machine table, and S the machine spindle.

A measuring instrument for this test involves the use of a micrometer head or test indicator in the end of an arm secured to an arbor socketed in the driving spindle of the machine.

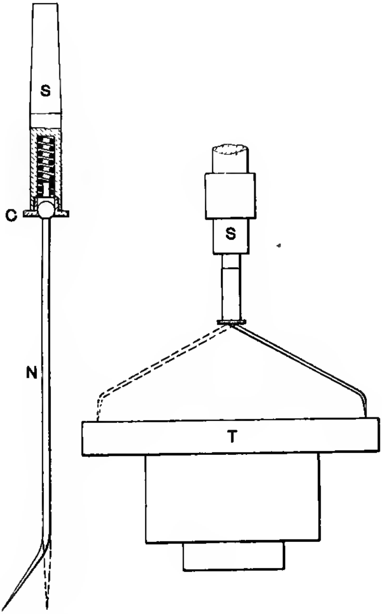


FIG. 54.—Drilling machine test indicator.

PLANING MACHINE TESTS.

1. **Table and Bed Tests.**—The principal tests which are made on planing machines are tests of the horizontality of the ways of the bed (whether square or vee), table or platen, and cross-rail. These tests cover both longitudinal and transverse horizontality.

One method of making the longitudinal bed-level-ing test is indicated in Fig. 55. In this figure W represents one of the ways of the bed, S a parallel test block, and L a spirit level. The parallel test block (which is dispensed with in the rather rare cases of square ways) is placed in the way as indicated, so that its top face is practically horizontal transversely. The spirit level then enables longitudinal horizontality to be obtained for this way, or the levelling error to be calculated or otherwise determined if the vial of the level is graduated. The other way is treated in exactly the same manner.

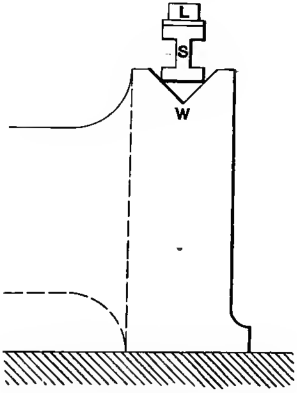


Fig. 55.—Longitudinal test on planing-machine table.

One method of making the transverse level test is represented in Fig. 56. In this figure C C represents two test cylinders or standard plug gauges of the same diameter, S represents a parallel test block, L represents a spirit level, and W W represent the ways of the bed. This test—as well as the above—

should be made at several places in the length of the bed.

In Fig. 57 is indicated a method of combining the longitudinal and transverse levelling tests. The view shown is a plan of the apparatus required in its position on the bed. This apparatus consists of six similar test cylinders, C, four or five parallel test blocks, S, and a spirit level, L. With the arrangement shown, the level can be used in the two directions. This enables any twist or wind in the bed to be detected readily. In removing this twist, thin tissue paper is sometimes placed immediately underneath the upper blocks and the bed raised locally until this is just gripped.

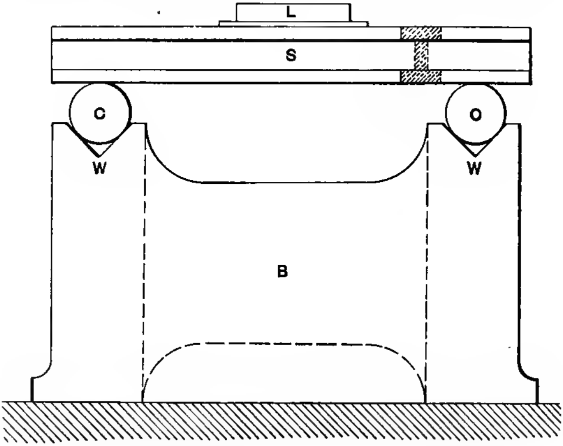


FIG. 56.—Transverse test on planing-machine table.

2. **Housing Tests.**—The testing of the faces of the housings for verticality and transverse alignment can be performed by means of a special square sup-

ported on a special block on the ways of the machine and a dial test indicator secured to a special block capable of being moved on the housing faces.

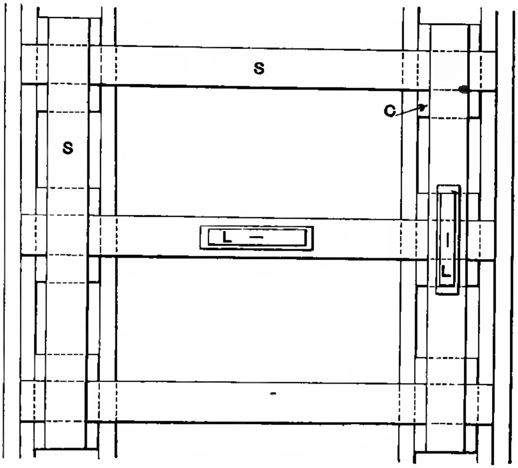


FIG. 57.—Combined longitudinal and transverse tests on planing-machine table.

TESTS ON OTHER MACHINE-TOOLS.

In the foregoing all the most important aspects of machine-tool accuracy testing have been dealt with. In the considerations, however, several machine tools, such as grinding machines, boring machines, slotting machines, and shaping machines have not directly received attention. This is because practically all the accuracy tests required on these machines are required, either in the same form or only very slightly modified, on the machines which have received attention. Hence, many of the tests which have been described under other headings can be made to apply under this.

General Observations.

It should be borne in mind that the primary function of all machine-tool accuracy testing is the checking of the workmanship incorporated in the construction and erection of the machines so that these latter will be capable of turning out work which is also accurate, that is, accurate within the limits of error which are permissible in each case. Hence, it is often advisable to test certain machine-tool elements under conditions which are somewhat similar to those which are likely to obtain in the actual use of the machine. Thus, for example, it may be advisable to test overhanging machine tool tables and radial arms for accuracy when these are more or less loaded, since it is known that, owing to their cantilever form, they are liable to deflection under the influence of the cutting force or forces which are created in cutting operations.

A secondary function of machine-tool accuracy testing is the maintenance of a high degree of fitting between engaging parts, and the reduction of the frictional resistances encountered by the moving parts. This, of course, bears a direct relation to the question of the mechanical efficiency of machine-tools.

CHAPTER III.

MACHINE-TOOL SPEED AND FEED TESTS.

THE variations which occur in the general design of machine-tools are due largely to the following causes, taken either singly or collectively:—

1. Differences in the forms of the cutting tools;
2. Differences in the kinds of cutting operation; and
3. Variations in the dimensions and shapes of the work pieces which have to be machined.

According to these variations in design, machine-tools can be classified in quite a number of ways. Probably the principal classification, however, is based upon the differentiation between the forms of motion from which the cutting speed is directly derived. In some machine-tool types this motion is imparted to the work under operation, such as the lathe, planing machine, and boring mill; in others, to the cutting tool, such as the milling machine, the drilling machine, and the shaping machine; and in another though rather restricted type, to both the work and the cutting tool, such as certain forms of grinding machine and gear-cutting machine.

Whether this motion, however, is given to the work or to the cutting tool, it occurs in ordinary cases in only two forms, namely, circular or rotatory motion, and rectilinear or reciprocatory motion. The basis of the former is an angular speed which is measured usually in revolutions per minute, whilst that of the latter is a linear speed, which is usually stated in feet per minute (English system of measurement) or metres per minute (metric system of measurement).

ROTATIONAL SPEED TESTS.

The chief machine-tools in which the speed is a rotational one are the various forms of lathe, the drilling machine, the milling machine, and the grinding machine. In connection with the efficient operation of any one of these it can be readily shown that the various speeds incorporated in the design of the machine should be in a geometrical progression, or as nearly in such a progression as possible, though there are machine-tool designers who are still of the opinion that these speeds should be in arithmetical or harmonical progression, whilst in some few cases the progression of the speeds is very irregular and follows no definite law.

1. **Revolution or Speed Counting.**—The exact determination of the speed of a rotating shaft or spindle is a matter of some little skill and practice. If the speed is very low, its magnitude may be obtained by actually counting the number of times a marked point or projection on the shaft or spindle passes a fixed point in a given period of time. The results which are derivable from such a method are not very satisfactory, since to eliminate errors of observation, and to make allowance for a fractional turn as far as possible, the time-period has to be

prolonged excessively. For high-speed determinations it is absolutely worthless.

A superior method—and indeed the only one which is suitable for general application—is to use a counting or indicating instrument. With a counting instrument—portable or otherwise—the instrument has to be used in each determination for an exact period of time, such as a minute or two minutes, or the time observed, by means of a chronometer stop-watch (which will read to one-fifth of a second), in which the spindle or shaft makes an exact number of revolutions.

Fundamental Formula.—In each case, the speed in revolutions per minute (generally denoted by the letters R.P.M.) is calculated by means of the following expression :—

$$\text{N (R.P.M.)} = \frac{\text{No. of revolutions counted}}{\text{Period of counting}} \qquad (14)$$

In certain cases it is not possible to count directly the number of revolutions made by the shaft or spindle owing to its position. When such a condition exists it is necessary to determine the surface or circumferential speed of a part of the shaft or spindle which has a known diameter, and from this calculate the angular speed. The formula which is employed in these cases is as follows :—

$$\text{N (R.P.M.)} = \frac{\text{Surface speed, in feet per minute}}{\text{Circumference, in feet}}$$

$$= \frac{12\text{S}}{\pi\text{D}} \quad . \quad . \quad . \qquad (15)$$

where S = the surface speed, in feet per minute, and D = the diameter of the part, in inches.

This may also be written :—

$$N\ (\text{R.P.M.}) = y \times S \quad . \quad . \qquad (16)$$

where y equals $\frac{12}{\pi D}$ and is a constant for any given value of D. To facilitate the use of this formula the curve given in Fig. 58 has been plotted, this curve being the graphical form of the relationship between the value of y (ordinate) and the diameter (abscissa).

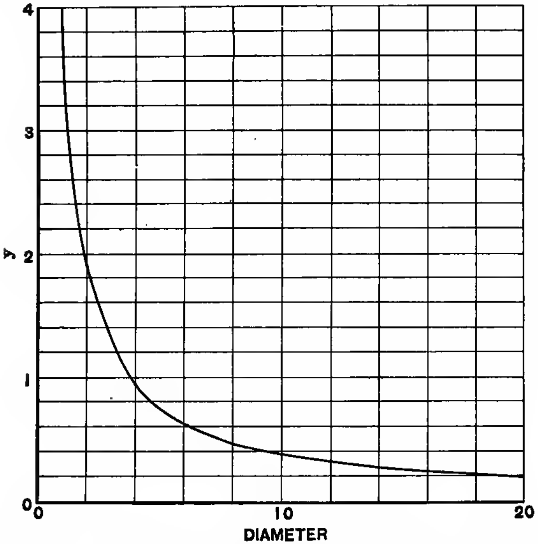


FIG. 58.—Curve showing relation between diameter and speed coefficient of a rotating bar.

In connection with the use of this curve it should be noted that the diameter of the shaft should be in inches and *not* feet.

2. **Revolution Counters and Indicators.**—The simplest type of hand revolution counter works on the worm-gear reduction principle. One form of it is represented in Fig. 59. In this instrument the

ratio of reduction is $\frac{1}{100}$, so that for each revolution of the graduated dial 100 revolutions of the spindle are required. In its earliest forms this instrument merely enabled numbers between the hundreds to be read off directly, the hundreds having to be counted separately by noting the number of times a small button or pin passed a given point. With the form shown, however, it is also possible to count the hundreds, as well as the tens and units, directly. The end of the spindle is in the form of a triangular pyramid, this being placed in a countersunk centre hole in the end of the shaft or spindle whose speed is required. An improved method is, however, to use a rubber tip at the end, this having greater adhesive power than even sharp metal edges. In the figure are shown two forms of rubber tip which are in use, another form being the hemispherical form. Of the three, the conical form is the best and easiest to use.

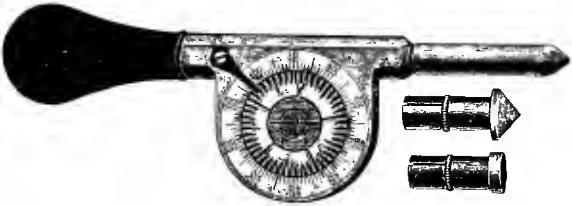


FIG. 59.—Revolution or speed counter.

Since this form of speed counter is never used for more than, say, five minutes at a time, there does not appear to be any need for many of the long ranges with which some are provided. Any instrument, however, with which it is not possible to count the hundreds quite easily and without any uncertainty is in the majority of cases practically useless.

Another form of revolution counter is the Harding counter. In this instrument there is no dial or scale of the ordinary type. In place of this a row of rectangular or circular openings (usually the former) are provided in the face of the instrument, as shown in Fig. 60. Behind each of these openings, a wheel, which carries a figured rim, rotates (more or less intermittently) on a horizontal shaft. Each wheel is toothed, and is moved, one figure at a time, by the one preceding it, with the exception of the wheel which carries the unit figures. Thus, the addition of only one opening with its corresponding wheel apparatus multiplies the range of the instrument by ten. The case of the instrument is either round or square, and the motion applied may be either rotatory or oscillatory, according to the design of the mechanism of the instrument. The design of the form for use in connection with an oscillatory motion is a little more complicated than that of the other form.

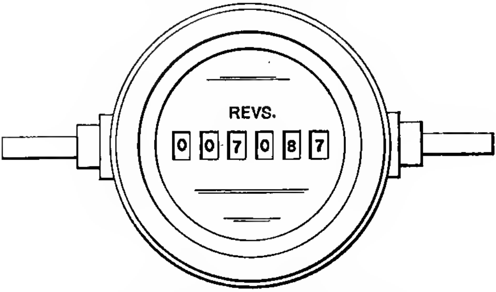


FIG. 60.—Machine revolution counter.

Such a form of counter is employed chiefly as a fixture on a machine tool, it being continuously driven from the spindle or shaft whose speed is required. When the rotary form is used, some form of non-slip or positive drive, such as a chain, must be employed; when the oscillatory form is used, the usual practice is to drive by means of two cranks and a connecting rod, with, if necessary, a chain and sprockets and intermediate shafts. The hand form works on the rotatory principle, and is equipped, like the rotating disk form, with a pointed or rubber-tipped spindle for direct connection with the shaft or spindle to be tested for speed. As a machine counter, a capacity of 10,000,000 revolutions is not unusual; as a hand counter, however, it is not usual to provide for more than 10,000 revolutions.

In using this form of speed counter it is necessary to note the readings of the instrument at the beginning and at the end of a definite period of time, and then by means of expression (14) calculate the speed in revolutions per minute.

With the above instruments a watch (preferably a chronometer stop watch) has to be used. Some of the disk or dial counters are combined with watches, so that it is possible to observe the two together, though whether such an arrangement enables more accurate results to be obtained, or the work of taking the readings to be more expeditiously performed, is a moot point except in those cases wherein the insertion of the end of the spindle into the centre hole in the shaft starts the watch and counter simultaneously, and the reverse operation stops them.

3. **Tachometers.**—Speed indicators, gyrometers,

or tachometers show at a glance, and without any calculation, the angular speeds of rotating shafts and spindles. They may be divided into three classes, as under:—

(1) Centrifugal instruments,
(2) Electro-magnetic instruments, and
(3) Aero-dynamic instruments.

Centrifugal Tachometers. — Tachometers which

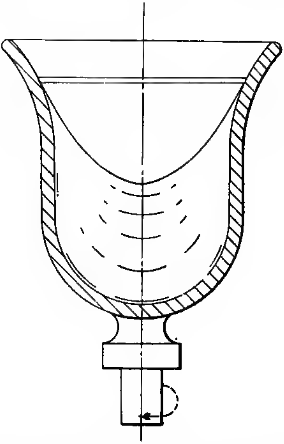


FIG. 61.—Centrifugal tachometer or speed indicator.

work on the centrifugal-force principle are of two kinds. In the first kind, the centrifugal-force element consists of a very sensitive governor with metal balls or disks which are deflected by the action of centrifugal force. This deflection is converted through the medium of suitable gearing into motion of a needle or pointer working over a graduated scale, the scale divisions being either units or tens of revolutions per minute. In the second kind, the centrifugal-force element is a liquid. This is contained in a specially shaped vessel (such as is shown in Fig. 61), to which a rotatory motion is given, this motion being derived from the motion of the spindle or shaft under test. The immediate effect of the centrifugal force on the liquid is, of course, to tend to throw it outwards, but, since no liquid is perfectly mobile and the containing vessel is of a real height, the ultimate effect is to cause the surface of the liquid to assume a concave (parabolic) form, as is shown in the figure. For each speed of rotation there is a definite position of the lowest point in the curve. The liquid must be enclosed in a hermetically sealed containing vessel so as to prevent its evaporation, since the position of the index for any speed will depend upon the volume of liquid in the whirling vessel. On this form of tachometer the scale divisions are practically equal throughout the range of the scale.

Electro-magnetic Tachometers.—Electro-magnetic tachometers are also of two kinds. In the first kind use is made of the principles of electro-magnetism directly. A permanent magnet of a suitable form is mounted directly on the rotating shaft of the instrument, and rotates in the neighbourhood of a pivoted metal disk (either steel or copper), with the result that Foucault or eddy currents are set up in the latter and interaction takes place. This interaction, according to the principles of electro-magnetism, causes the disk to tend to travel round with the magnet. It is, however, restrained from doing this completely by a control spring, and is only allowed to travel through a small angle. The magnitude of this angle depends *inter alia* upon the pole strength of the deflecting magnet, the strength of the control spring, and the speed of rotation of the magnet. The movement of the disk is transmitted directly or through gearing to the needle of the instrument which works over a graduated scale.

In the other form of electro-magnetic tachometer a magneto-electric machine or permanent magnet electrical generator is used, as indicated in Fig. 62.

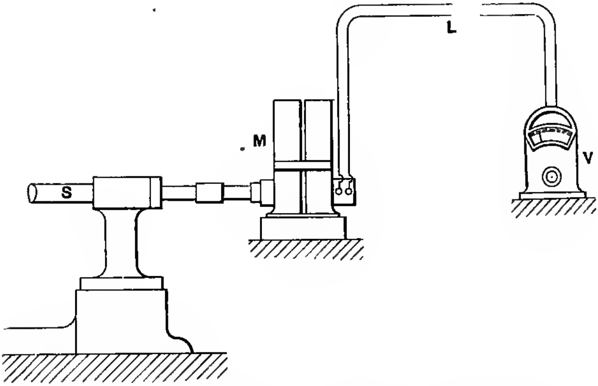


FIG. 62.—Electro-magnetic tachometer or speed indicator.

The magneto, M, is secured directly and positively to the shaft or spindle, S, whose speed is required. The terminals of the magneto are connected to those of a voltmeter, V, by line wires, L. The voltmeter should be either of the electrostatic type or of the high-resistance electro-magnetic type, so that the drop in pressure in the line wires is a negligible quantity. The principle of this method is the generation of an electro-motive-force (E.M.F.) in the armature of the magneto, the magnitude of this E.M.F. being directly proportional to the speed of rotation. This E.M.F. in volts is indicated practically on the scale of the voltmeter, so that, under any given set of circumstances, the voltmeter reading is a direct measure of the speed. To make use of such a reading it is necessary to possess a " calibration curve " (Fig. 63), which shows the relation between the voltmeter reading and the speed, and from which the value of the speed can be taken off for any voltmeter reading. This calibration curve can only be obtained, of course, as the result of special speed tests.

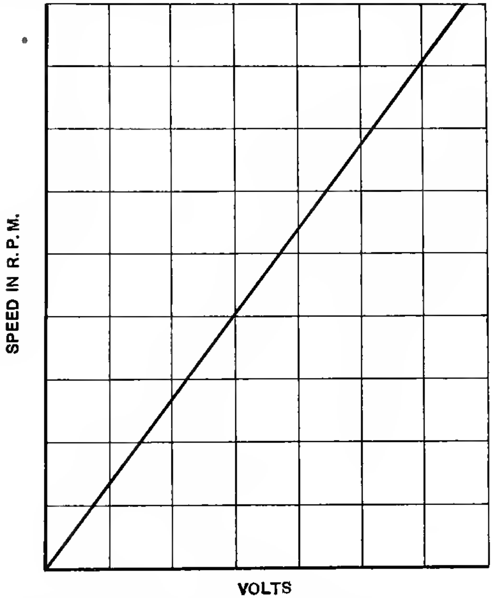


FIG. 63.—Calibration curve for electro-magnetic tachometer.

An improvement upon the above method is the graduation of the voltmeter scale to read in R.P.M. and not volts. In such a case, the instrument requires special calibration.

Aero-dynamic Tachometers.—Aero-dynamic tachometers are also of two kinds. The first works on the "fan or vortex-current principle". The fan is enclosed in a hermetically sealed case and mounted on the spindle of the instrument. Its rotation causes a current of air to be created and to be impinged directly on a rotatable vane, the freedom of motion of which is limited by the influence of a hair-spring, this latter being the controlling or restraining element. The extent of the movement of the vane is not proportional directly to the speed of the fan but to some power of it. The scale of the instrument, therefore, has to be specially graduated. In some cases, to increase the speed above that of the shaft or spindle to be tested, recourse has to be had to the use of bevel, spur, or spiral gearing in the instrument itself.

The other form of aero-dynamic tachometer is represented in Fig. 64. This instrument consists of a tube, C, which is connected to the spindle by the bevel gearing, H. This tube has two tee connections, A, at its lower end, and is rotated at a high speed. The upper end of C is immersed in mercury which is contained in a vessel, O. Over the upper end of C and also in the mercury is a tube, E, which is connected to a gauge pipe, G, by a union joint, F, the gauge pipe running to a sensitive vacuum gauge. B is the base of the instrument. It also serves as a shield to the tubes A, and protects these from the influence of external air currents. The centrifugal action on the air in the tubes A creates a greater or less vacuum in the pipe E, this being indicated on the dial of the vacuum gauge which may be graduated to read in R.P.M. in preference to "inches of mercury" or "pounds per square inch". In this case also the scale divisions throughout the range of the scale are not of equal lengths.

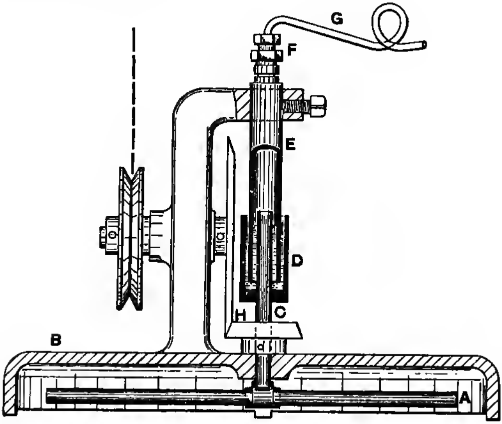


FIG. 64.—Aero-dynamic tachometer.

Tachoscope.—An instrument which produces a graphical chart from which the average speed, in R.P.M., during any period of time can be obtained, is called a tachoscope, though sometimes, but wrongly, it is referred to as a tachometer. The principle of action is that of the governor, centrifugal force being employed to move the style or pencil of the instrument on a moving sheet of paper or specially prepared plate. The general form of the diagram is indicated in Fig. 65. Its average height, which is proportional to the average R.P.M., is obtained by any one of the several methods of averaging a diagram, such as the planimeter method, the mean ordinate method, or Simpson's rule.

In regard to the method of driving fixed counters and tachometers, it should be pointed out, however small the amount of power is which is required to drive them, there will be some slip unless the drive is a positive, non-flexible one. For this reason belt, strap, and rope drives should only be tolerated when it is not possible to use any other.

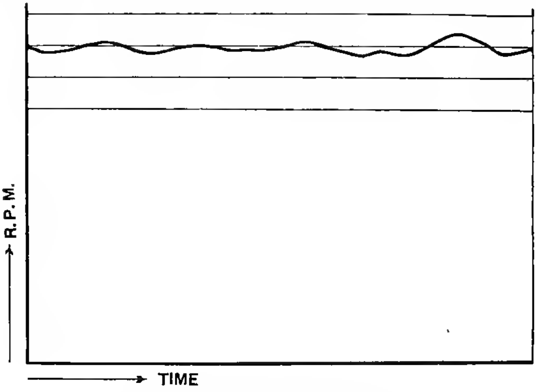


FIG. 65.—Tachoscope curve.

4. **Surface Speed Counters and Indicators.**—These, generally, can be designed to work on any one of the above principles, though they differ from the above inasmuch as a roller or wheel of a definite diameter is embodied in their design in place of a centre or direct driving mechanism.

In Fig. 66 is shown one form of surface speed counter of the worm-gear reduction type. A rubber-tyred wheel W is mounted directly on the worm shaft. The worm drives the graduated wheel, A, the ratio of reduction being $\frac{1}{100}$. A small pinion which is secured to A in turn drives another graduated wheel, B, the ratio of reduction in this case being $\frac{1}{10}$. It is thus possible to count up to 1000 revolutions of the wheel. Now, the circumference of the wheel is just 6 in. Hence, each revolution of it corresponds to 6 in., and the range of the instrument is 500 ft., and the complete turn of the wheel A corresponds to 50 ft. The scales on the two wheels A and B are very easily set to zero, and the instrument can be readily converted into a revolution counter.

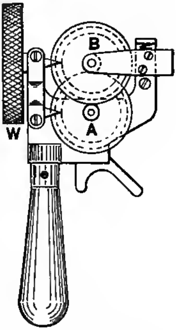


Fig. 66.—Surface speed counter.

A superior form of surface-speed counter wheel which has been designed by the author, is the milled or knurled steel wheel. The rubber tyre tends to alter its shape and size with time and use, and hence cannot be relied on always; with the steel wheel there is no such trouble, though on highly finished surfaces this form of wheel may not be suitable.

Ordinary revolution counters are readily converted into surface speed counters by the provision of a surfacing wheel of a diameter of 1·91 in. This diameter corresponds to a circumference of 6 in., so that 1 ft.

is represented by two divisions on the scale of the counter.

Cut Meter.—A surface speed indicator is represented in Figs. 67 and 68. It is an electro-magnetic instrument, and is known generally as the cut meter since it was originally designed for use in connection with mechanical cutting operations. The wheel, W, drives a split permanent magnet, the external lines of force of which are deflected by a soft steel ring through an aluminium disk, the motion of this disk being controlled by a fine hairspring. On the outside surface of the disk are engraved the figures which indicate the speed in feet per minute. These figures appear behind a lined glass screen or window which is fitted in the case of the instrument, as is shown in Fig. 67. The wheel is fitted with a hard rubber tyre of circular section, a form which is vastly superior to the rectangular section found on many surface speedometers. The bearings are all

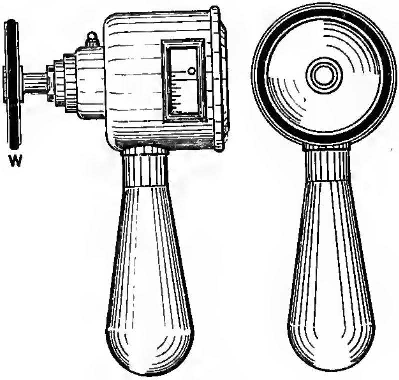


FIG. 67.—Cut meter.

made as nearly frictionless as possible by the use of either balls or sapphire jewels.

Rotameter.—In Fig. 69 is represented a simple form of surface speed counter which goes under the name of rotameter. It is a pocket watch-like instrument and works on the toothed wheel principle. The measuring wheel W is a small smooth-edged disk.

The range of the instrument shown is only 25 ft., but in larger sizes the range goes up to 100 ft.

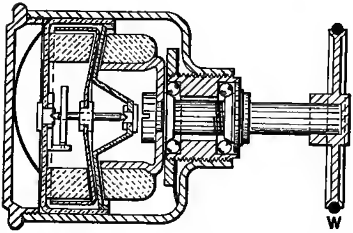


FIG. 68.—Sectional view of cut meter.

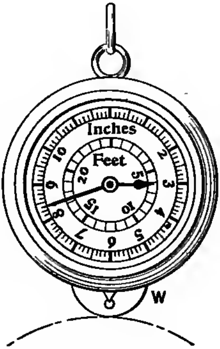


FIG. 69.—Rotameter.

This type of instrument is not, however, as suitable for general surface speed testing as are those which are described above; for very high speeds they are practically useless.

5. **Tests.**—In these tests the successive speeds of the driving spindle or shaft are determined. These are the speeds which are given directly to the work (as in the case of the lathe or boring mill) or cutter (as in the case of the drilling machine and milling machine).

The method of making the changes of speed in any particular case will depend, of course, upon the design of the machine. The principal methods in use involve the use of stepped speed cones, collapsible cones, toothed gearing, and two or more countershaft speeds.

The machine placed under test should be tested preferably unloaded, because it is not possible, except by means of a brake, to keep the load even fairly constant throughout such a test. As a matter of fact, the speeds of machine-tools which are generally given are no-load speeds, and these do not differ greatly from the corresponding speeds obtained with a cut. Of course, this point is related directly to the question of belt-slip, but, except in those cases of notorious overloading, this is a matter of but slight importance.

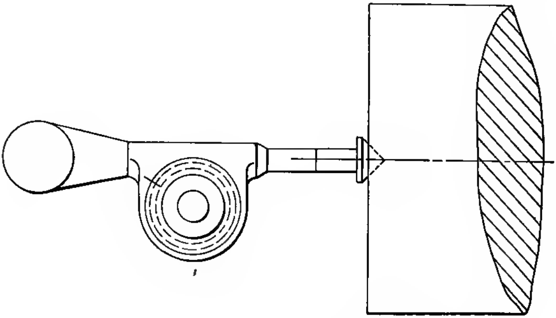


Fig. 70.—Method of using speed counter.

The method adopted to measure the speeds will depend entirely upon circumstances. In Fig. 70 is shown the method of using an ordinary revolution counter, the use of which must be combined with that of a stop watch.

The following table gives three typical sets of spindle speeds obtained in tests on a lathe, drilling machine, and milling machine.

TABLE II.

SPEED TEST DATA.

Type of Machine Tool.	Lathe.	Drilling Machine.	Milling Machine.
No. of countershaft speeds	2	2	2
No. of cone steps	4	4	4
No. of back gears	1	1	1
No. of spindle speeds	16	16	16
High countershaft speed	250·5 R.P.M.	150·9 R.P.M.	190·9 R.P.M.
Low ,, ,,	200·4 ,,	90·6 ,,	160·8 ,,
Spindle speeds: 1	16·5 ,,	12·5 ,,	16·0 ,,
,, ,, 2	21·0 ,,	17·1 ,,	19·1 ,,
,, ,, 3	26·6 ,,	21·1 ,,	24·1 ,,
,, ,, 4	33·1 ,,	24·1 ,,	29·1 ,,
,, ,, 5	42·0 ,,	28·7 ,,	35·2 ,,
,, ,, 6	52·1 ,,	34·7 ,,	43·2 ,,
,, ,, 7	70·9 ,,	40·3 ,,	53·3 ,,
,, ,, 8	88·0 ,,	57·8 ,,	64·3 ,,
,, ,, 9	140·0 ,,	76·4 ,,	79·4 ,,
,, ,, 10	178·1 ,,	106·7 ,,	96·4 ,,
,, ,, 11	225·1 ,,	128·4 ,,	118·5 ,,
,, ,, 12	280·9 ,,	149·0 ,,	144·6 ,,
,, ,, 13	360·0 ,,	177·1 ,,	175·7 ,,
,, ,, 14	450·2 ,,	213·4 ,,	213·0 ,,
,, ,, 15	600·3 ,,	247·5 ,,	263·3 ,,
,, ,, 16	750·0 ,,	356·0 ,,	319·6 ,,

Having obtained the speed values throughout the entire range, and observed how the changes are effected, it is necessary now to analyse the set to determine the relationship which exists between the individual speeds, and also to determine the nature of the speed progression. If the progression is a regular one—which is not invariably the case—it is usually of the simple arithmetical, geometrical, or harmonical series type.

6. Arithmetical Speed Progression.—The characteristic of this progression is that the difference between any two consecutive speeds is a constant quantity throughout the entire range of speeds obtained by means of any one form of mechanism. Thus, if N_1 = the first spindle speed in revolutions per minute; N_2 = the second; N_n = the last; and n = the total number of speeds in the range, we have that—

$$(N_2 - N_1) = \text{the common difference} \qquad (17)$$

Also—

$$\frac{(N_n - N_1)}{(n - 1)} = \text{the common difference} \qquad (18)$$

If we represent the common difference by d, then—

$$N_n = N_1 + (n - 1)\,d \qquad (19)$$

These are the three chief statements relating to arithmetical progressions which are associated with speed changes effected by only one form of change-speed mechanism such as a speed cone or toothed gearing. The case which involves a combination of speed-change methods is essentially different.

The graphical form of the relationship between the value of the speed (N_1, N_2, . . . N_n) and its number (1, 2, . . . n) is indicated by the straight line A in Fig. 71, in which a maximum of 10 speeds has been adopted, it being understood that these speeds are all obtained by means of one method of driving.

To determine whether the series in any case is in arithmetical progression, it is only necessary to calculate the differences between consecutive speeds. If they are all equal, or practically so, it may be assumed that the design of the speed arrangements of the machine was based upon this type of progression.

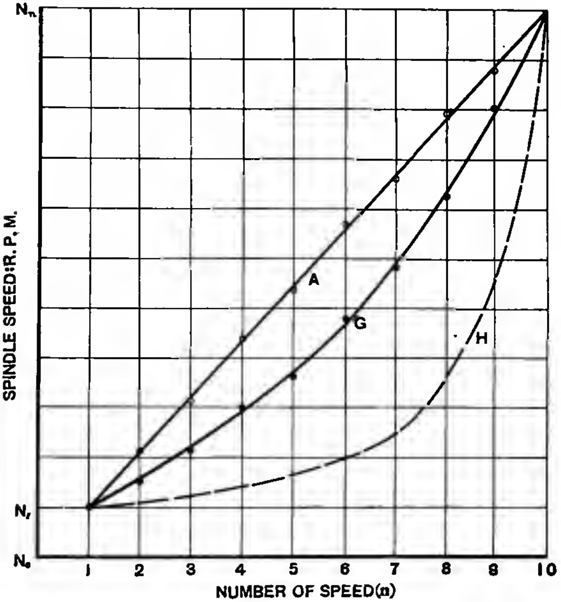


FIG. 71.—Diagram of spindle speeds.

An examination of the three series of speeds given in Table II will show that not one of the three is an arithmetical series of this type. This, of course, follows from the fact that the speed changes are affected in three different ways, since it is not possible to arrange speeds obtained in such a manner in a simple arithmetical progression, though it is possible to arrange them in a series of arithmetical progressions.

7. Geometrical Speed Progression.—In this progression the fundamental principle is that of a continuous increase by multiplication, as distinct from the addition principle of the preceding progression. The ratio of any two consecutive speeds is a constant quantity throughout the entire range of speeds. Thus, if N_1 = the first spindle speed, in revolutions per minute; N_2 = the second; N_n = the last; and n = the number of speeds in the range, we have that $\frac{N_2}{N_1}$ = the ratio of consecutive or successive speeds . . . . (20)

If we represent this common ratio by r, we then have the following relationships

$$\frac{N_n}{N_1} = r^{n-1} \qquad (21)$$

and

$$N_n = N_1 \times r^{n-1} \qquad (22)$$

These are the three principal statements relating to geometrical progressions which have any bearing in this case.

The graphical form of the relationship between the value of the speed (N_1, N_2 . . . N_n) and the number of the speed (1, 2, . . . n) is indicated in Fig. 71 by the curve G. It should be observed, however, that the precise shape of the curve will depend, *ceteris paribus*, upon the value of the common ratio (r).

The determination as to whether the series in any case is a geometrical one or not can be effected in at least two ways. The first method involves the determination of the actual ratios between consecutive speeds and their comparison. If they are all equal, or practically so, it may be safely assumed that the series is a regular geometric one upon which the design of the speed-change mechanism was based.

The analysis of the three series of speeds given in Table II, according to this method, gives the following results :—

TABLE III.

CONSECUTIVE SPINDLE SPEED RATIOS.

Deduced from Table II.

Type of Machine Tool.			Lathe.	Drilling Machine.	Milling Machine.
Spindle	speeds :	1—2	1·27	1·37	1·20
,,	,,	2—3	1·26	1·24	1·26
,,	,,	3—4	1·25	1·14	1·21
,,	,,	4—5	1·27	1·20	1·21
,,	,,	5—6	1·24	1·21	1·23
,,	,,	6—7	1·36	1·14	1·23
,,	,,	7—8	1·24	1·43	1·21
,,	,,	8—9	1·59 (*a*)	1·32 (*a*)	1·24 (*a*)
,,	,,	9—10	1·27	1·39	1·22
,,	,,	10—11	1·26	1·20	1·23
,,	,,	11—12	1·25	1·16	1·22
,,	,,	12—13	1·28	1·19	1·21
,,	,,	13—14	1·25	1·20	1·22
,,	,,	14—15	1·33	1·16	1·23
,,	,,	15—16	1·25	1·44	1·21
Average ratio			1·29	1·25	1·22

These results show plainly that the milling machine series is a very regular geometrical progression. The other two series may be described as irregular geometrical progressions, since they follow the general law of the geometrical series but contain one or two abnormalities.

In Fig. 72 is shown the form of the variations which occur in practice in the value of the consecutive ratio in a range of speeds. The case represented is that of ten speeds (a five-stepped cone pulley with double back gear). This method of representing the changes shows them up distinctly and enables a designer to pick out the weak spots in his design. On this diagram the straight line AB represents the ideal case wherein the value of *r* is absolutely constant for the same maximum and minimum speeds.

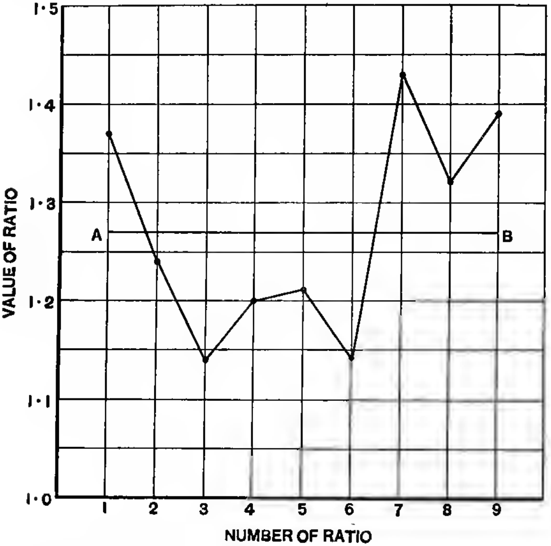


FIG. 72.—Diagram of consecutive spindle speed ratios.

A word of explanation may be necessary in regard to the ratios which are lettered (*a*) in Table III. These ratios are those between the last speed obtainable with the back gearing in and the first speed obtainable with it out. It will be observed that in the case of the lathe the back-gear ratio does not agree with the other elements in the design of the speed-change arrangements. The back-gear ratio in any case depends upon the sequence of the speed changes. In each of the above cases, however, it may be expressed in terms of the average ratios of consecutive speeds as follows:—

$$R = \frac{1}{r^{\frac{n}{2}}} \quad . \quad . \quad . \qquad (23)$$

where R = the ratios of the product of the numbers of teeth in the driving wheels in the back-gear train to the product of the number of teeth in the driven wheels in the same.

Regarding the value of r, it should be pointed out that this can be obtained either by averaging the individual consecutive speed ratios, as is shown in Table III, or by taking the highest and lowest speeds in the series and using the expression

$$r = \sqrt[n-1]{\frac{N_n}{N_1}}. \quad . \quad . \quad . \quad . \quad . \qquad (24)$$

In all ordinary cases the two values so obtained are practically identical.

8. **Logarithmic Analysis.**—The second method of analysing a series of machine-tool spindle speeds to determine whether it is a geometrical one or not involves the use of ordinary logarithms.

If we apply common logarithms to expression (21) we obtain the following equation :—

$$\log\left(\frac{N_n}{N_1}\right) = (n - 1) \times \log r \quad . \quad . \quad . \qquad (25)$$

By applying logarithms to expression (22) we obtain that—

$$\log N_n = \log N_1 + (n - 1) \times \log r. \quad . \qquad (26)$$

Each of these two expressions represents a straight-line law. In the first expression, the two variable quantities are $\log\left(\frac{N_n}{N_1}\right)$ and $(n - 1)$, the value of $\log r$ for any given geometrical progression being constant. We can, therefore, plot on rectangular co-ordinate axes the values of $\log\left(\frac{N_n}{N_1}\right)$ and $(n - 1)$, where n has any integral value from 1 upwards. The expression $\left(\frac{N_n}{N_1}\right)$ may de defined as the spindle-speed ratio, by which is to be understood the ratio between any spindle speed and the lowest one, and not the consecutive-speed ratio or the ratio between consecutive speeds. Hence, $\log\left(\frac{N_n}{N_1}\right)$ is the logarithm of the spindle-speed ratio. A representative case is shown in Fig. 73, ten spindle speeds forming the basis of this diagram. The line drawn is a fair or average graph through the plotted points.

If logarithmic or logarithmically graduated paper (for the ordinates only) is used, reference to a table of logarithms is rendered unnecessary; the result is, however, precisely the same as when ordinary squared or sectional paper is employed.

If preferred, we can use the expression (25) in the place of expression (24). In such a case, the curve, which we shall be able to plot, will represent the relation between log N_n and $(n - 1)$. Since, however, N_1 and r are assumed to be constant quantities in any given case, the only difference between this result and the above will be in the scale of units adopted along the ordinate or y axis. In Fig. 74 the above case of ten spindle speeds is represented in this manner, semi-logarithmic sectional paper being used.

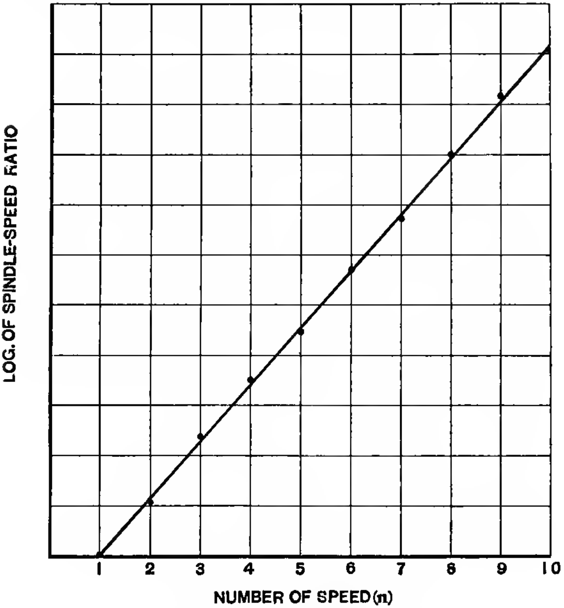


FIG. 73.—Logarithmic speed diagram.

From either of these curves the value of r which operates throughout the entire series can be determined. The logarithm of this ratio is equal to the tangent of the angle of inclination of the graph to the line of abscissae or the x axis. This angle is indicated in Figs. 75 and 76, which represent the two above cases respectively; it is lettered θ. The explanation of this may be given as follows :—

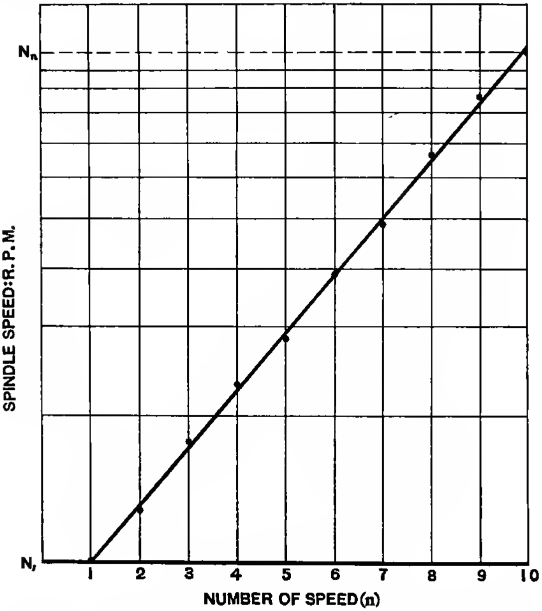


FIG. 74.—Alternative form of logarithmic speed diagram.

Consider the two expressions (24) and (25). By transposition, these become respectively—

$$\log r = \frac{\log \frac{N_n}{N_1}}{(n-1)}, \quad . \quad . \quad . \quad . \qquad (27)$$

and $\log r = \dfrac{(\log N_n - \log N_1)}{(n - 1)}$ . . . . (28)

An examination of each of the diagrams in Figs. 75 and 76 will show that, in the case of the first—

$$\frac{\log \dfrac{N_n}{N_1}}{(n-1)} = \tan \theta \quad . \quad . \quad . \quad . \qquad (29)$$

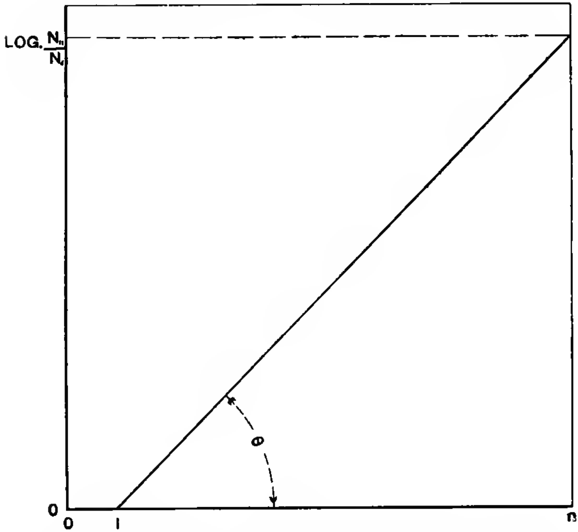


FIG. 75.—Diagram illustrating logarithmic speed computation.

and in the case of the second—

$$\frac{(\log N_n - \log N_1)}{(n - 1)} = \tan \theta. \quad . \quad . \quad . \quad . \qquad (30)$$

Therefore, in each case, we have that—

$$\log r = \tan \theta. \quad . \quad . \quad . \quad . \qquad (31)$$

The relative magnitudes of the local variations in the whole series from a true geometrical progression can be readily determined by noting the extent to which the various points plotted on the diagram lie off the average line which is drawn through the points. As far as possible this line should pass through the two terminal points, as is indicated in Figs. 73 and 74.

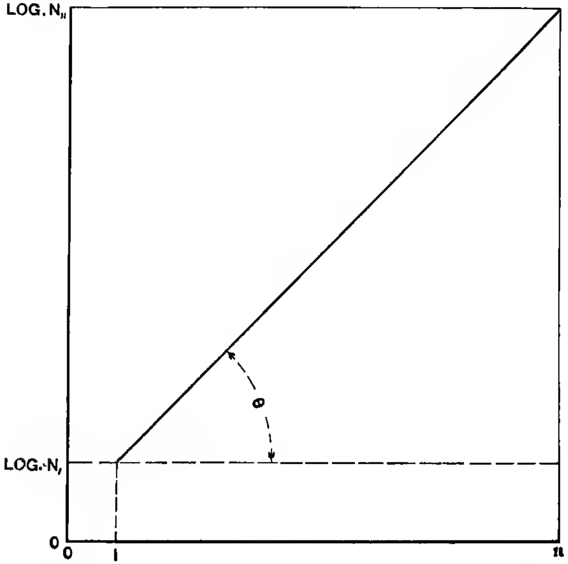


FIG. 76.—Diagram illustrating logarithmic speed computation.

9. **Harmonical Speed Progression.**—This form of progression is generally regarded as being the inversion of the arithmetical form of progression, since the difference between the reciprocals of consecutive speeds in the series is a constant quantity. Thus, using the symbols adopted in connection with the

discussion of arithmetical speed progressions, we have that $\frac{1}{N_1} - \frac{1}{N_2} = h$ . . . . (32)

The quantity h may be defined as the common reciprocal speed difference of the series, and if we take the sum of all these common reciprocal speed differences throughout the whole series we obtain this relationship—

$$\frac{1}{N_1} - \frac{1}{N_n} = (n - 1) \times h \qquad . \quad . \quad . \qquad (33)$$

from which, by suitable transposition, are derived the following expressions :—

$$N_n = \frac{N_1}{1 - (n - 1)\, h\, N_1} \qquad . \quad . \quad . \qquad (34)$$

$$\text{and } h = \frac{N_n - N_1}{N_1 N_n (n - 1)} \qquad . \quad . \quad . \quad . \qquad (35)$$

These are the four principal statements relating to harmonical speed progressions. The value of h is invariably a comparatively small fraction, whereas the arithmetical common difference is invariably a number much greater than unity, whilst the consecutive speed ratio of the geometrical progression usually lies between 1·1 and 2·0, though in the majority of cases the range is much less, being only from 1·20 to 1·50.

The graphical form of the relationship between the value of the speed (N_1, N_2, . . . N_n) and the number of the speed (1, 2, . . . n) is represented in Fig. 71 by the dotted curve H. It should be observed here that the exact shape of the curve is very definite, given fixed maximum and minimum values of the speed and a fixed number of speeds. Even if the latter number is changed only, the shape of the curve is altered.

This figure illustrates in a very interesting manner the way in which the three regular speed progressions differ.

To determine whether the speed series in any case is a harmonical one or not, it is necessary either to calculate the individual values of *h* (the reciprocal difference) and compare these with one another, or to calculate the value of *h* for the whole series by means of expression (35) and use this in expression (34) to calculate the intermediate speed values which would occur if the series were a harmonical one, comparing these with the actual speed values which obtain.

10. **General Note.**—In the majority of cases where any attempt is made to base the design of the speed arrangements upon a scientific basis, the progression selected is of the geometrical form, though in one or two cases the arithmetical or harmonical form is adopted.

Their chief comparative feature is this: in an arithmetical speed progression of the simple form the arithmetical differences are equal; in a geometrical speed progression they are gradually and regularly increased as the speed is raised; whilst in a simple harmonical speed progression they are increased but much more rapidly than in the second case. In other words, the ratio between the last and first arithmetical differences is greatest in a simple harmonical speed progression and least in a simple arithmetical speed progression, assuming these to operate between the same limiting speeds and with the same number of speeds. The corresponding ratio in a geometrical speed progression is intermediate to the above.

When the speed changes are effected by a combination of methods it is, however, not possible to make use of a simple arithmetical or harmonical progression; but a simple geometrical progression can be used in such a case.

RECTILINEAR SPEED TESTS.

1. **Speed Counters and Indicators.**—Any of the above forms of surface speed counters and indicators can be used to determine the speeds which are connected with rectilinear reciprocating motions, such as the motion of a planing machine table or that of a shaping machine ram. When an ordinary counter is used, it is the average speed spread over a certain (usually short) period of time which is obtained. With an ordinary indicator such as a speedometer or cut meter, it is the instantaneous speed which is given. Where, however, very rapid changes of speed occur, as they do at the beginning and end of the stroke of the moving element of a planing or shaping machine, it is practically impossible to determine the manner and extent of such changes by means of any of these instruments. This is due, principally, to the fact that these changes occur in a very short space of time (in some cases, considerably less than a second), and, secondarily, to the power of the eye to retain visual impressions for short periods of time, the effect of this being to render it impossible to relate the changes of position of a moving needle or pointer to a duration of time.

When it is desired to know how the speed in such a case changes, it is necessary to have recourse to some form of autographic or self-recording instrument, suitably desigued for this kind of work, so that the need for instantaneous observation is dispensed with, a curve or graph being obtained from which both the instantaneous and average speeds can be readily computed.

There are several ways in which such an instrument can be designed. In one design, due to the author, the motion of the indicating pencil or style is obtained by means of positive gearing from the reciprocating element, the motion of the pencil being a reduced facsimile of that of the table or ram under test. In one form the motion of the pencil is controlled by a square threaded screw and nut, the pencil being attached to the latter and constrained by a guide to move in a straight line. The pencil works on a sheet or ribbon of paper, the motion of this being at right angles to the motion of the pencil and controlled by a clockwork mechanism which maintains the motion uniform. The screw receives its motion from the reciprocating table or ram through the medium of toothed gearing, as is shown in Fig. 77. The first element in this gearing is a specially made rack which is secured directly to the table or ram in a position parallel to the direction of motion. This rack drives a pinion and a train of spur gear wheels which terminates in the wheel on the screw. In this instrument all the moving parts are hardened and corrected for distortion in the hardening process so as to be able to resist wear and eliminate back lash as much as possible.

In another form of indicator for this kind of work, the motion of the pencil or style over the paper is controlled by an inextensible cord and helical or coiled spring. The cord receives its motion directly from the table through the medium of a number of reducing pulley-combinations, as shown in Fig. 78. These combinations are connected together by inextensible cords, the large pulley of the first combination being driven from the table, P. The number of combinations required is determined by the reducing power of each and the total reduction required. It will be seen that the small pulley of the last combination carries the pencil cord, P. To keep the cords always taut, the pulleys are provided with internal springs.

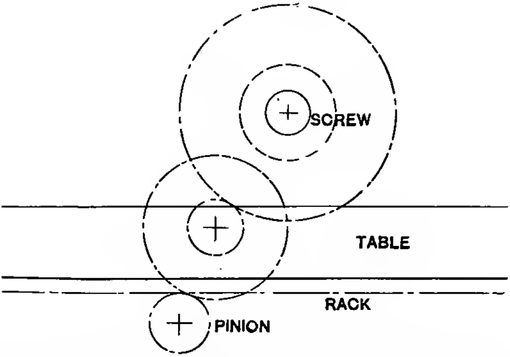


FIG. 77.—Diagram of gearing of instrument for obtaining travel-time curves from rectilinear reciprocating motions.

The form of the graph obtained with this instrument is precisely the same as that which is obtained with the other form of indicator.

The electro-magnetic principle is also capable of application in connection with the design of rectilinear tachographs, but up to the present time no instrument embodying this principle has, to the knowledge of the author, been constructed. The graph which would be obtained from such an instrument would be one involving the instantaneous linear speed and the time, and as such would be much easier to analyse than the curves which are obtained from instruments of the first variety, these involving the travel of the table or ram and the time.

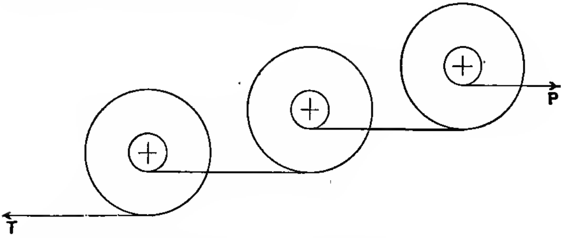


Fig. 78.—Diagram of speed-reducing gear.

2. **Tests.**—The form of the curve which the above instruments produce is indicated in Fig. 79. The point O represents the beginning of the power or cutting stroke of the table or ram of the machine. The distance that the table or ram travels from this point is set off automatically on the line of ordinates, the point T representing the end of the stroke. The duration of the stroke is set off automatically on the line of abscissæ, the distance OP on that line representing the time occupied by the table in moving through this forward stroke. The distance PR represents the time taken by the table to make the return stroke, the point R coinciding with the point O of the curve representing the next cycle of strokes.

In all machine-tools in which reciprocating motion is employed, the return stroke (unless a double cutting mechanism is embodied in the design of the machine—which is rare) is a non-productive stroke. Hence, to reduce its non-productiveness, the average speed during this stroke is higher than that of the forward stroke, the ratio of average speeds varying from 1·5 to 6. Consequently, the time occupied in making the return stroke is less than that occupied in making the forward stroke in exactly the same ratio. In the case represented in the figure, the value of this ratio is 3; that is, OP = 3 × PR.

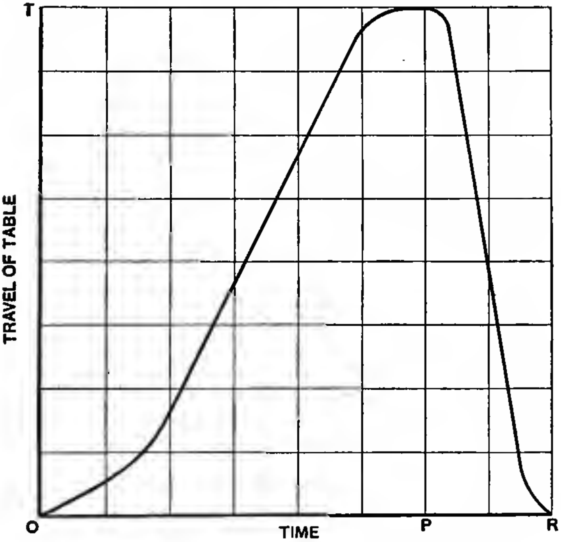


FIG. 79.—Travel-time curve for planing machine.

An examination of this curve will show that the distances moved through in equal periods of time are not equal, either during the time OP or during the time PR. This, of course, indicates roughly that the speed is not a constant quantity during either the forward or the return stroke. This is a result which can be obtained by direct observation, since at the beginning of each stroke the speed has to be increased from zero to a maximum, and at the end it has to be reduced from a maximum to zero, these being operations which cannot be performed without varying the speed by accelerating or retarding the motion.

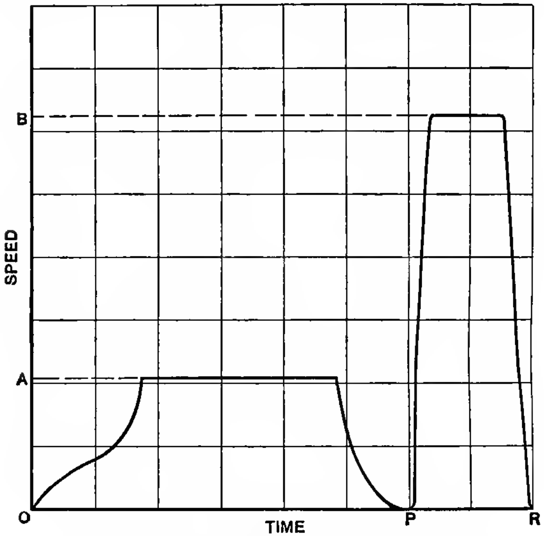


FIG. 80.—Speed-time curve for planing machine.

3. **Speed Determinations.**—The relationship between the instantaneous speed and the time for the complete cycle is indicated graphically in Fig. 80. This diagram is derived from the preceding travel-

time curve. This derivation can be accomplished by either of two methods. In the first method, the time base is divided into a number of equal parts representing equal times. The distance travelled during each of these times is obtained by setting up ordinates and drawing horizontal lines from the points of intersection of these ordinates and the curve, as is shown in Fig. 81. Each distance is proportional to the

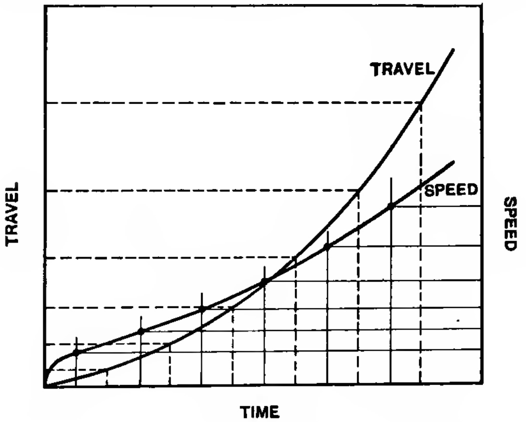


FIG. 81.—Fundamental travel-time curve and derived speed-time curve.

average speed during its interval of time, the actual value of this speed being the ratio of these quantities. It is assumed that this average speed occurs at least one instant in the interval of time, namely the middle instant, so that between the bounding ordinates of the time interval another ordinate—termed a mid-ordinate—is set up, and on this line the corresponding average speed value is set off. The curve drawn through the points so obtained is a speed-time curve, and it indicates graphically the relation between the instantaneous speed and the time. This process which is illustrated in Fig. 81 is only approximate from the theoretical point of view, though practically in all ordinary cases it gives results which are both reliable and satisfactory.

The fundamental principle of the second method is indicated in Fig. 82. In this figure the curve shown is

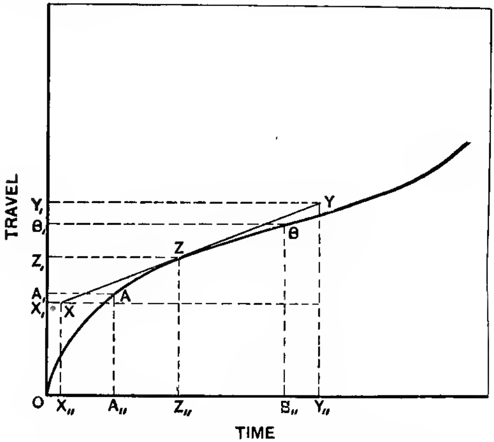


Fig. 82. Fundamental travel-time curve.

a travel-time curve. We will suppose that we wish to know the instantaneous speed which corresponds to the point Z on the curve, and therefore to the time OZ_{11} from the starting point. The corresponding travel from the starting point is OZ_1. Let us take any two points A and B on the two sides of Z on the curve as shown. Then the average speed is equal to $\frac{A_1\,B_1}{A_{11}\,B_{11}}$, but we do not know exactly where this occurs, though, if the curve were a straight line, it would occur at every point in it. Imagine the points A and B brought closer and closer together on the curve until they are both in contact with the point Z, one on each side. Then, since points are regarded as being of no finite dimensions, we may assume that the three points are in the same straight line. This straight line is called the tangent to the curve at the point Z. This is a geometrical concept. In the figure it is the line XZY, in which X and Y are any points whatsoever. The instantaneous speed at the point Z is the average speed which would occur if the curve were identical with the tangent XZY. But the value of this latter speed is determined by the tangent (trigonometrical ratio) of the angle which the geometrical tangent XZY makes with the horizontal. This equals $\frac{X_1 \, Y_1}{X_{11} \, Y_{11}}$. Therefore, the instantaneous speed at Z, Z_1, and Z_{11} also equals this. If we make $X_{11} \, Y_{11}$ equal to unity, then the instantaneous speed equals $X_1 \, Y_1$. Hence, the position of the point representing the speed at Z is obtained by setting off the distance $X_1 \, Y_1$ on the ordinate Z_{11} Z from Z_{11}.

This indicates the principle of this method, which involves the selecting of a series of points in the travel-time curve, the drawing of the geometrical tangents to the curves at these points, the construction of right-angled triangles about these tangents with unit bases, and the setting off of lengths equal to the heights of the triangles on the ordinates which pass through the selected points from the line of abscissæ.

Referring again to Fig. 80, it will be seen that the maximum forward speed is much less than the maximum return speed, the ratio being $\frac{OA}{OB}$ and its value about 1 to 3. Furthermore, it will be noticed the time occupied in accelerating the motion of the table in the forward stroke is much greater than the corresponding period of retardation, whilst in the case of the return stroke the two corresponding periods are of about the same duration.

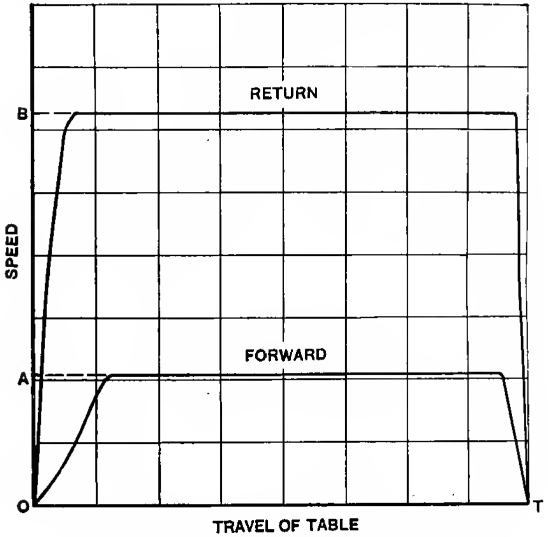


Fig. 83.—Speed-travel curves for planing machine.

In Fig. 83 is shown the graphical relationship between the instantaneous speed of the table or ram and its travel from the starting point. It is derived in exactly the same way as is the curve given in Fig. 80, though in this case the travel of the table is taken as the other co-ordinate instead of the time. That is, the ordinates of the fundamental curve in Fig. 79 are taken in place of the abscissæ in the formation of the new curve. The speeds OA and OB are identical with the speeds OA and OB in Fig. 80.

In this curve are shown the distances of acceleration and retardation in the travel of the table as distinct from acceleration and retardation periods as given in Fig. 80.

The average speed for each of the strokes, and also for the cycle, can be readily obtained by dividing the travel of the table or ram by the duration of the stroke, or cycle. These quantities can also be obtained by suitably averaging (by any one of the several methods available) the whole diagram or the parts thereof given in Fig. 80.

POWER FEED TESTS.

The feed (whether power or hand-actuated) of a machine-tool is the movement of the cutting tool with respect to the work, or vice versa, which enables the former to work over the surface of the latter. It is irrespective of the depth of the cut, and it is also distinct from the speed, its value being, in all cases, much less than that of the speed.

In the majority of machine-tools (such as lathes and planing and drilling machines) the direction of the feed movement is normal to that of the speed movement, and in only one or two cases, such as certain forms of milling and grinding, are the two motions even approximately parallel.

1. **Definitions of Feed.**—*Lathe.*—There are two ways of stating the feed of a lathe tool with respect

to the work. The first way is to state it in terms of the travel of the tool per revolution of the spindle; this may be called the linear feed. It is always a distance, and in the vast majority of cases it is less than one inch or one centimetre per revolution of the spindle. The second way is to state it in terms of the number of revolutions of the spindle required for each inch traverse of the tool; this may be termed the rotational feed. It is always a number, and in the greater number of cases this is greater than unity. These two feeds are reciprocals, as the following considerations will show. Let F_e and F_r be the two feeds respectively; T the traverse advance of the tool in inches in any given time; and N the number of revolutions made by the spindle and work in the same time. Then—

$$F_e = \frac{T}{N} \quad \ldots \quad (36)$$

$$= \frac{1}{\frac{N}{T}}.$$

But, $$F_r = \frac{N}{T}. \quad \ldots \quad (37)$$

Therefore, $$F_e = \frac{1}{F_r} \quad \ldots \quad (38)$$

and $$F_e \times F_r = 1. \quad \ldots \quad (39)$$

Drilling Machine.—The ordinary feed of a drilling machine can also be stated in two ways these being identical with the above. The linear feed is the axial or longitudinal movement of the drill spindle per revolution of itself; the rotational feed is the number of revolutions that the spindle makes per inch of longitudinal movement. They are reciprocals, the first being a distance and the second a number.

In addition to the above, there is the feed which is related directly to the unit of time. It is defined as the longitudinal or axial movement or traverse of the drill per minute, and as such is dependent upon the ordinary feed and the speed of rotation of the drill, as the following considerations will indicate. Let F_e and F_r be the linear and rotational feeds respectively; F_m the feed in inches per minute—this may be termed the unit-time feed; N the number of revolutions made by the drill spindle in any given time; T the total axial movement or traverse of the drill in inches in the same time; and M the given time in minutes. Then—

$$F_m = \frac{T}{M}. \qquad . \quad . \quad . \quad . \qquad (40)$$

But, $T = N \times F_e$ and $\frac{N}{F_r}$ (from expressions (36) and (37)):

therefore :—

$$F_m = \frac{N \times F_e}{M} = n \times F_e, \qquad . \quad . \quad . \qquad (41)$$

$$\text{and also } F_m = \frac{N}{M \times F_r} = \frac{n}{F_r}, \qquad . \quad . \qquad (42)$$

the symbol n in each of these expressions representing the rotational speed of the drill spindle in revolutions per minute.

Milling Machine.—In connection with the operations of this machine there are three different feeds. The first is the feed movement per cutter tooth and may be termed the linear tooth feed; this is a very small distance and varies is magnitude from 0·0003 in. to 0·0400 in. The second is the feed movement per revolution of the cutter and may be termed the linear cutter feed; this also is a small distance.

The third is the feed movement per minute and may be described as the unit-time feed; this has the largest value of the three. The relations between these are given in the following considerations of the case. Let F_t be the linear tooth feed in inches per tooth; F_c the linear cutter feed in inches per revolution; F_m the feed in inches per minute; n the rotational speed of the cutter in revolutions per minute; and C the number of teeth in the cutter, these being assumed to be evenly spaced. Then :—

$$F_m = F_c \times n \qquad (43)$$

$$F_c = F_t \times C \qquad (44)$$

and

$$F_m = F_t \times n \times C \qquad (45)$$

In some cases, instead of the linear cutter feed in inches per revolution, it is the rotational cutter feed in revolutions per inch of travel which is given. Obviously, these are reciprocals. It is much simpler, however, to have to deal with three feeds, as above, which are of the same general type. In using this rotational cutter feed in connection with expressions (43) and (44), it is necessary to invert it in order to obtain its reciprocal F_c.

Grinding Machines.—The feeds in this case are (1) the feed movement per revolution of the grinding wheel; (2) the feed movement per revolution of the work (where any exists as in the case of circular or cylindrical grinding); and (3) the feed movement per minute. The relation between feeds (1) and (3) has already been given in expression (43). A similar expression represents the relation between feeds (2) and (3). Between feeds (1) and (2) the relationship is a variable one, the ratio of these feeds being equal to the inverse ratio of the rotational speeds of the two elements.

Planing and Shaping Machines.—There are only two feeds in this case. They are (1) the normal feed in inches per cutting stroke, and (2) the unit-time feed in inches per minute. The relation between the two is a simple one, as is indicated in the following considerations. Let F_s = the feed per cutting stroke in inches; F_m the feed in inches per minute; and S the number of cutting strokes made per minute. Then we have that—

$$F_m = F_s \times S. \qquad (46)$$

Gear Cutting and Hobbing Machines.—The feeds which are operative on machines of this type depend upon the details of their design and the principle which underlies their operation. In some cases of spur-gear cutting, the feeds are identical with those which exist in the case of milling machines. In other cases, the three principal feeds are: (1) the feed per revolution of the gear-wheel blank, in inches; (2) the feed per tooth formed in the gear-wheel blank, also in inches; and (3) the feed in inches per minute. The relations between these three feeds are as follows:—

$$F_g = F_t \times G \qquad (47)$$

$$F_m = F_g \times n \qquad (48)$$

$$\text{and } F_m = F_t \times G \times n \qquad (49)$$

where F_t represents the tooth feed, in inches per tooth; F_g the blank feed, in inches per revolution of the blank; F_m the feed in inches per minute; G the number of teeth to be cut in the blank; and

n the rotational speed of the blank, in revolutions per minute. The values of F_t, F_g, and F_m are all comparatively small in such cases, and the value of n is related directly to the number of teeth to be cut in the blank, the number of threads on the hobbing cutter, and the rotational speed of the cutter. The relationship between these four latter quantities can be expressed in the following equation:—

$$n = \frac{n_h \times H}{G} \quad . \quad . \quad (50)$$

where n_h represents the rotational speed of the hobbing cutter, in revolutions per minute; H the number of separate threads on the cutter; and n and G have the meanings given to them above.

2. **Tests.**—In the experimental determination of machine-tool feeds—whatever their definition—it is necessary to know at least two quantities: a distance and a number of revolutions, a rotational speed, or an interval of time.

In such tests distances can be obtained by taking measurements between scribed marks which indicate the two positions of a fixed point on the sliding element corresponding to the initial and final test positions of the latter. Or they can be obtained by making use of fixed test pins and an outside micrometer gauge, as in the case of the tests on lead and feed screws, one test pin being secured to a fixed part of the machine, such as the bed or knee, and the other to the sliding element, such as the slide rest, table, or spindle. The latter method is capable of giving the more accurate results, though the percentage error which naturally occurs in the case of the first method can be substantially reduced by prolonging the test and making the distance greater.

On machines, such as upright or pillar drilling machines with graduated spindles and horizontal-spindle milling machines with micrometer-dial feed screws, it is, of course, a comparatively easy matter to measure feed distances with a reasonable degree of accuracy.

3. **Test Results.**—There appears to be abundant reason for the arrangement of machine-tool power feeds (of the linear type, and not the rotational) in geometrical progression, and this irrespective of the number of feeds in the range. In practice, however, machine-tool power feeds are arranged in regular arithmetical, geometrical, and harmonical progressions as well as in series which are more or less irregular and follow no fixed law.

Lathes.—In Table IV is given a series of eight feeds taken from an experimental lathe:—

TABLE IV.

LATHE FEED SERIES.

No. of Feed.	Feed, in Inches per Revolution of Spindle.	Ratio of Consecutive Feeds.
1	0·017	1·47
2	0·025	1·36
3	0·034	1·47
4	0·050	1·60
5	0·080	1·50
6	0·120	1·40
7	0·167	1·50
8	0·250	

From the list of ratios of consecutive feeds it will be seen that this series is what may be termed, for want of a better description, an irregular geometrical progression. The average ratio of progression is 1·47, and the extreme values are 1·36 and 1·60, these showing percentage differences of 7·5 and 8·8 per cent respectively.

On a smaller lathe with only four-power feeds it was found that these were as follows: (1) $\frac{1}{12}$ in.; (2) $\frac{1}{24}$ in.; (3) $\frac{1}{36}$ in.; and (4) $\frac{1}{48}$ in. per revolution of the spindle. An examination of this series will show that its terms are in harmonical progression, since 12, 24, 36, and 48 are in arithmetical progression.

In another case, the four feeds of a small lathe were found to be 0·0556, 0·0375, 0·0250, and 0·0167 in. per revolution of the spindle. If these values are analysed it will be seen that they are practically in a regular geometrical progression, the average ratio of progression being 1·50.

Drilling Machine.—In Table V are given the eight power-feed values of a high-speed experimental drilling machine.

A perusal of this list of feeds will show that this series also is a slightly irregular geometric one, the average ratio of progression being 1·48 as against 1·47 for the above case. In this case the two extreme percentage differences compared with this value are 9·5 and 12·8 per cent respectively. These figures show that this series is more irregular than the eight-feed lathe series dealt with above.

(3) *Milling Machine.*—The feed motions of lathes and drilling machines are almost invariably derived from the work or drill driving spindle, so that to

alter the feed it is necessary in all such cases to make the change in the feed-change mechanism, since any change in the spindle speed affects equally the unit-time feed (that is, the traverse advance per minute). In other words, there is a definite, fixed relation between the spindle motion and the feed motion (if all question of belt-slip, where a feed-drive belt is used, is neglected), and the linear feed cannot be changed by altering the speed.

TABLE V.

DRILLING MACHINE FEED SERIES.

No. of Feed.	Feed, in Inches per Revolution of Spindle.	Ratio of Consecutive Feeds.
1	0·0033	
		1·51
2	0·0050	
		1·34
3	0·0067	
		1·49
4	0·0100	
		1·67
5	0·0167	
		1·50
6	0·0250	
		1·34
7	0·0334	
		1·50
8	0·0500	

On some designs of milling machine this condition holds also, and the only effect on the feed when the spindle speed is changed is to alter the unit-time feed, the linear feed per revolution and the linear feed per tooth remaining unchanged. In such cases, the number of unit-time feeds possible is equal to the product of the number of available spindle speeds and the number of available linear cutter feeds; or

$$x = y \times z \quad . \quad . \quad . \quad (51)$$

where x = the number of unit-time feeds; y the number of spindle speeds available; and z the number of available linear cutter feeds.

In contra-distinction to the above, some milling machines are so designed that the feed motion is derived quite independently of the speed motion, either from the same countershaft or from an independent countershaft or motor. In such cases, the number of unit-time feeds available is limited, this depending upon the design of the feed-change mechanism, and the number of linear cutter feeds possible is equal to the product of the number of available spindle speeds and the number of available unit-time feeds; or

$$z = x \times y \quad . \quad . \quad . \quad (52)$$

where x, y, and z have the meanings ascribed to them in connection with expression (51).

In the case of the first type of milling machine the feeds which are experimentally determined are the linear cutter feeds, that is, the feeds in inches per revolution of the cutter spindle. In the case of the second type the feeds dealt with experimentally are the unit-time feeds, that is, the feeds in inches per minute.

In Table VI are given the twelve unit-time power feeds of a modern horizontal-spindle milling machine of the second type.

TABLE VI.

MILLING MACHINE FEED SERIES.

Independent Feed Drive.

No. of Feed	Feed, in Inches per Minute.	Ratio of Consecutive Feeds.
1	0·50	1·30
2	0·65	1·29
3	0·84	1·37
4	1·15	1·31
5	1·51	1·35
6	2·03	1·26
7	2·56	1·31
8	3·35	1·35
9	4·52	1·30
10	5·90	1·31
11	7·72	1·35
12	10 40	

These feeds have been plotted against their respective numbers to form the curve given in Fig. 84. The curve is a fair one through the points as actually plotted, and it will be seen that all the points lie fairly evenly on the curve. This indicates that, whatever the nature of the law of the progression of the feeds, the progression is a fairly regular one.

By obtaining the individual values of the ratio of consecutive feeds as given in the table we can see whether the series is of the geometric type or not. It will be noticed that these values are fairly regular, varying only from 1·26 to 1·37, and giving an average value of 1·32. We may, therefore, assume that the series is of this type. That it is not either an arithmetical progression or a harmonical one will be plainly evident from even only a cursory examination of the series. The extent of the variations in the

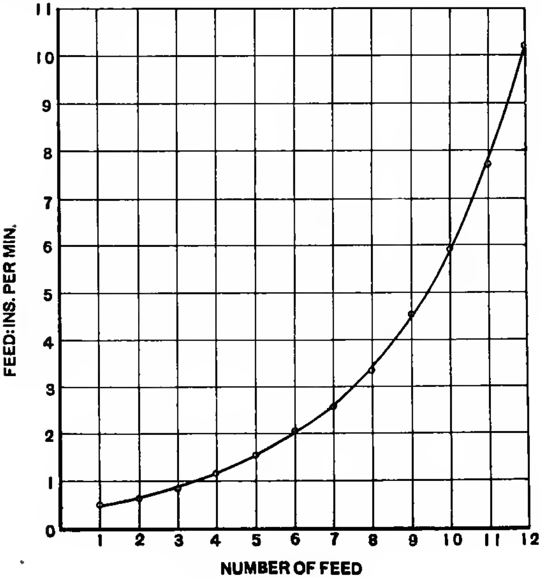


FIG. 84.—Feed curve.

value of the ratio of consecutive speeds can be shown graphically by means of a diagram of the form of Fig. 72.

On the particular machine in question it was found that the spindle speeds ranged from 13 to 467 revolutions per minute, the number of speeds being 16 and the average ratio of consecutive speeds 1·27. By means of these figures we can obtain the maximum and minimum linear cutter feeds. These are 0·80 and 0·00107 in. per revolution of the cutter respectively.

Concerning the manner in which the linear cutter feed changes in such cases as this, it can be readily demonstrated that, if the unit-time feed and the spindle-speed progressions are regular geometrical ones with the same ratio of progression or succession, the linear cutter feed series is also a geometrical one with the same succession ratio.

Thus, we have in this case that—

$$F_m = F \times r^{l-1} \quad . \quad . \quad (53)$$

where F_m is lth unit-time feed, F = the lowest unit-time feed, and r = the ratio of progression or succession. Also—

$$N = \frac{N_g}{r^{g-n}} \quad . \quad . \quad . \quad (54)$$

where N_n is the nth speed, N_g is the highest speed, and g the number of terms in the speed series. Then—

$$F_c = \frac{F_m}{N_n} = \frac{F \times r^{l-1}}{N_g \times r^{n-g}} = F_1 \times r^{l-n+g} \quad (55)$$

where F_1 = the lowest linear cutter feed. Since l and n may have any integral values within their respective ranges, and g has a fixed value in any one case, it follows that the linear-cutter feed series is a geometrical progression. When the two ratios are nearly equal, the departure from a true geometrical progression is very slight indeed.

CHAPTER IV.

MACHINE-TOOL MECHANICAL EFFICIENCY TESTS.

By the expression "mechanical efficiency" is meant the ratio between the amount of mechanical energy which is usefully employed in a machine in a given time and that amount of the energy which is supplied to it in the same time. In a machine-tool the two essential movements are those which provide the cutting speed and the feed, and the energy which is usefully employed in such a case is absorbed in overcoming the resistances to the cutting-speed movement and those to the feed movement, irrespective of the conditions which obtain in regard to the identity of the elements of the machine to which these two movements are individually or collectively given.

1. Input Determinations.—The determination of the amount of input energy communicated to a machine-tool (or one part of it only), or the input power, can be made in at least three different ways. These are (1) the electric motor method, (2) the transmission dynamometer method, and (3) the cradle dynamometer.

Electric Motor Method.—In this method an electric motor is the power-supplying element. It is necessary in every case to know the efficiency of this machine, since the output of this is the input of the machine-tool. In determining this efficiency the input must be given by means of a wattmeter in the case of an alternating current machine in order to eliminate the influence of the power-factor; whilst in the case of a direct-current machine either a wattmeter or a voltmeter and ammeter may be used.

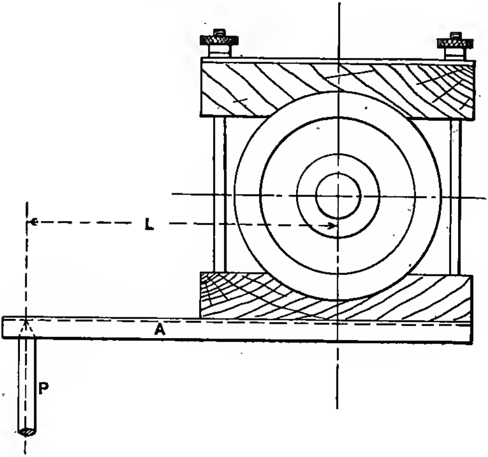


FIG. 85.—Simple Prony brake.

This input is given in watts of electrical power. It is converted into horse-power by means of the following expression :—

$$\text{Input Horse Power} = \frac{\text{Watts}}{746} \qquad (56)$$

When, with direct currents, a voltmeter and ammeter are used together, the wattage equals the product of the readings of the two instruments.

The output of the motor, or the motor B.H.P., or

O.H.P., is usually experimentally determined by means of a friction brake, though there are also special quasi-electrical methods in use in connection with such work.

A simple form of this type of brake, usually known as the Prony brake, is represented in Fig. 85. The frictional resistance is set up at the rim of the driving pulley of the motor, or a specially fitted brake drum (preferably water-cooled), through the medium of two blocks of hard wood which are pressed on the pulley or drum by means of the nuts shown. The influence of the frictional resistance is resisted by a pillar, P, upon which rests the arm, A, of the brake. This pillar rests on the platform of a weighing machine, on which the downward force (sometimes called the "load") is measured. The B.H.P. or O.H.P. is calculated by means of the following formula:—

$$\text{B.H.P. or O.H.P.} = \frac{L \times N \times W}{5252} \quad . \quad . \quad (57)$$

where L = the length of the arm, as indicated in the figure and measured in feet; N = the number of revolutions of the armature shaft per minute; and W = the net load registered on the weighing machine.

In regard to the obtainment of W, it should be observed that when there is no rotation and the brake is free on the pulley or drum there is a component of the weight of the brake exerted on P and the platform of the weighing machine. This should always be allowed for by subtraction from the gross value of W, though it is a point which in many cases is overlooked.

This is practically the simplest form of brake or absorption dynamometer. It is possible, however, to apply in this case any other form, such as the rope brake, and obtain practically the same results.

In designing a Prony brake drum, the diameter and width of the rim should be such that at least 0·12 square foot of rim surface is allowed per horse power absorbed.

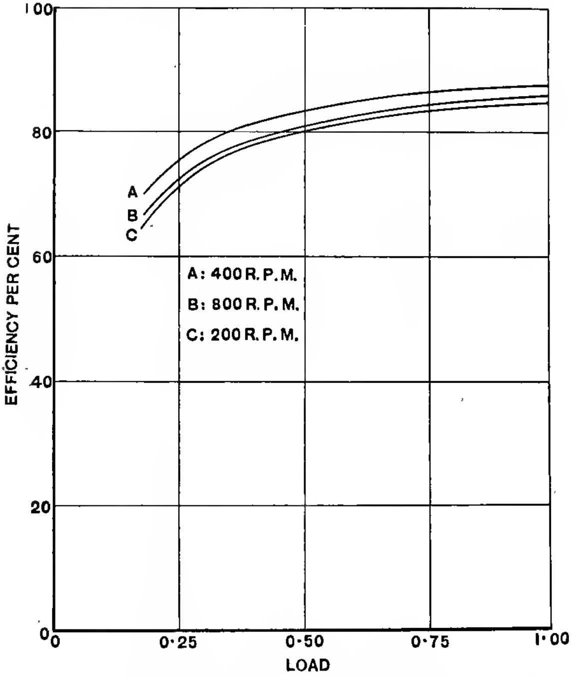


FIG. 86.—Motor efficiency curves.

In making the test the electrical meter readings should be taken at the same time as the brake load and speed determinations.

The efficiency of the motor in any given case is determined as follows:—

$$\text{Motor Efficiency} = \frac{\text{Brake Horse Power}}{\text{Input Horse Power}} \quad . \quad (58)$$

$$\text{or } \eta = \frac{\text{B.H.P.}}{\text{I.H.P.}} \quad . \quad . \quad . \quad (59)$$

The fundamental principle of this method is the determination of the machine-tool input power by multiplying the motor input power by the efficiency of the motor. To be able to do this for a range of powers it is necessary to have either a table of motor efficiencies or, better still, a curve of motor efficiencies with either the input power values or the output or load power values as abscissæ. If the motor is a variable speed motor, then this should be done for a number of speeds, though it will be generally found that the speed in the case of a variable speed continuous-current motor has not a very marked influence on the overall motor efficiency. This is shown in Fig. 86, which represents the case of a 40 B.H.P. shunt wound motor provided with series inter-poles, the speed variation being effected by means of a rheostat inserted in the field-winding circuit.

Knowing the motor efficiency for any given set of conditions, we obtain the machine-tool input power by using expression (59) in the following form:—

$$\text{Machine-tool I.H.P.} = \text{Motor I.H.P.} \times \eta \quad . \quad (60)$$

The value of η is, of course, taken from the table or curve of motor efficiencies.

Transmission Dynamometer Method.—A transmission dynamometer is a mechanical device for the measurement of mechanical power without consuming it, that is, without causing its conversion into heat, which is the characteristic of the absorption or brake dynamometer.

There are at least four different types of transmission dynamometer which are capable of application in connection with the testing of machine-tools. These are :—

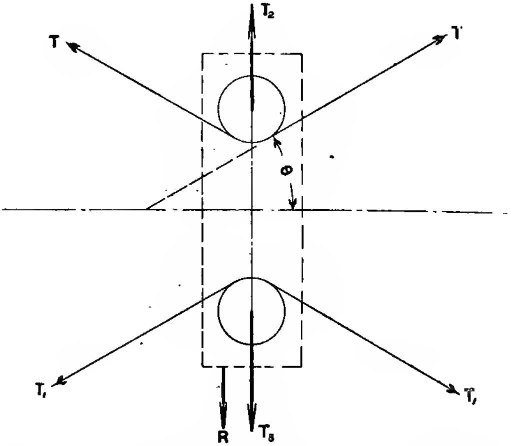


FIG. 87.—Force diagram for transmission dynamometer.

1. Dynamometers in which use is made of belt tension ;
2. Dynamometers in which use is made of gear-tooth loading ;
3. Dynamometers which work upon the spring torsion, tension, or compression principle ; and
4. Dynamometers which work on the thrust principle.

The principle underlying the action of the first is indicated graphically in Fig. 87. Two pulleys of equal diameters are mounted in a sliding or swivelling frame, the two sides of the belt from the driving to the driven pulley passing on the inside of these pulleys. Let T_1 be the driving-side tension and T that of the driven side. Then, if θ is one-half of the angle between the two sides of the belt as shown, we have that—

$$T_3 = 2T_1 \sin\theta \quad . \quad . \quad (61)$$

$$\text{and } T_2 = 2T \sin\theta \quad . \quad . \quad (62)$$

Let R be the resultant of these two forces; then—

$$\begin{aligned} R &= T_3 - T_2 \\ &= 2(T_1 - T) \sin\theta \quad . \quad . \quad (63) \end{aligned}$$

from which we get that—

$$(T_1 - T) = \frac{R}{2\sin\theta} \quad . \quad . \quad (64)$$

Let D = the diameter of the driven pulley (machine-tool pulley) in inches, and N = the speed of this pulley, in R.P.M. Then, we have that—

$$\text{Machine-tool I.H.P.} = \frac{D \times N \times R}{252{,}108 \times \sin\theta} \quad (65)$$

The measurement of the force R, which is the force required to keep the pulley frame in one position and the value of θ constant, can be effected either mechanically or hydraulically.

In Fig. 88 a mechanical method is indicated. It will be seen that the two jockey pulleys are mounted in a pivoted three-bar frame. The weight W acting on this frame resists the upward turning moment of R on it. A plumb pointer P is employed to assist in the maintenance of the normal position of the frame. The value of R is determined as follows:—

$$R = \frac{W \times L}{L_1} \quad . \quad . \quad (66)$$

L and L_1 representing the lengths of the two arms, as indicated in the figure.

Another mechanical method is to weigh the force R directly; it is not, however, quite as good as the method of moments since it does not admit of the same delicacy of balance.

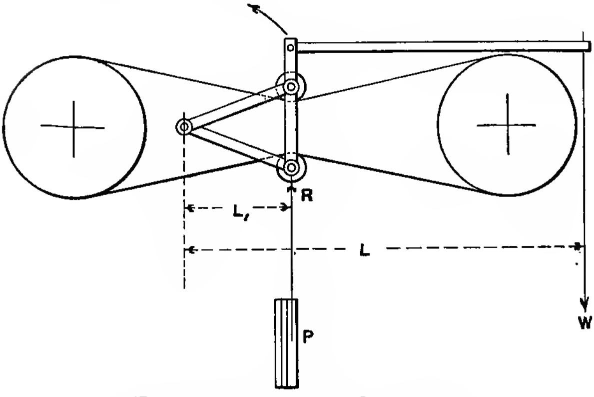


FIG. 88.—Transmission dynamometer.

The hydraulic method is represented in Fig. 89. The driving and driven pulleys are indicated by the letters PP. BB are two pedestals. F is the jockey pulley frame, and S is a horizontal bar, by means of which the condition of verticality of the frame F is maintained. The frame is supported on a plunger A which fits in a cylinder C, this latter being filled with oil and communicating to either a pressure gauge or an engine indicator. In the case of the former the pressure of the oil is read off directly on the scale of the gauge (the average of a number of readings being taken); in the latter case, the average pressure has to be obtained from a diagram similar to the one shown in Fig. 90 by finding the mean height of the ordinates. In this figure the line AB is the no-load line; the line CD is the load line, the mean ordinate height being taken between these two lines.

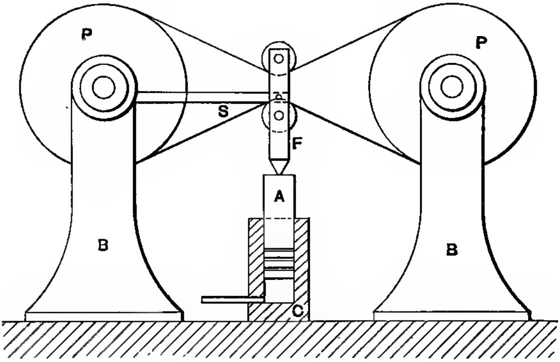


FIG. 89.—Transmission dynamometer.

The value of R in this case is determined as follows:—

$$R = P \times A \quad . \quad . \quad . \quad (67)$$

where P = the specific pressure, or pressure intensity, in lb. per square inch, as obtained, and A = the cross-sectional area of the plunger A in square inches.

The form of transmission dynamometer in which toothed gearing is employed is represented in Fig. 91. The gearing employed is of the bevel differential type, B. P is the driving pulley; P_1 is the driven pulley;

S is the driven shaft or spindle, say, of the machine-tool; and A is an arm attached to the middle gear wheel, its motion being limited in the manner shown. The arrow R indicates the direction that the arm A would take if it were allowed to move through any appreciable angle. W represents the weight which is required to keep the arm A floating under any given set of conditions. In this case, we have that—

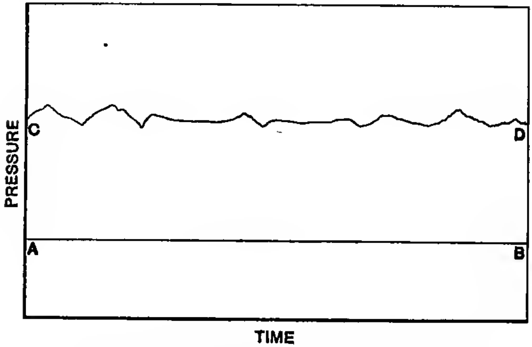


Fig. 90.—Transmission dynamometer curve.

$$\text{Machine-tool I.H.P.} = \frac{L \times N \times W}{2{,}626} \qquad (68)$$

W and L having the meanings given to them on the diagram (L being in feet), and N being the speed of the driven shaft S in revolutions per minute.

In Fig. 92 is shown one form of torsion transmission dynamometer. P_1 is the driving pulley and P_{11} is the driven pulley. The element which connects these two is the helical spring, S, and the power which is transmitted at any instant is directly proportional to the relative angular movement or angle of torsion or twist between the two ends of the spring. This angle is determined by means of an eccentric, E, which is secured rigidly to the driven pulley, P_{11}, and a vibrating slide, V. This slide carries a pencil, M, at its outer end, the pencil working, in conjunction, an indicator drum, I. The rotatory movement of the eccentric is converted into a quasi-simple harmonic motion of the vibrator and pencil, with the result that when the indicator is rotated through positive gearing from the driving pulley (the gearing having unit ratio) the pencil

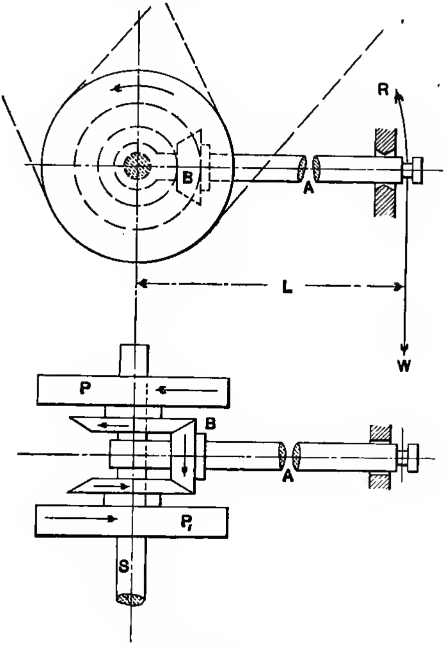


FIG. 91.—Transmission dynamometer.

traces a quasi-sine curve on the indicator card or paper. Such a curve is shown in Fig. 93, in which are represented a no-load curve and a load curve. The distance D represents one complete revolution of the driving pulley, and D_1 the angle of torsion or twist.

This form of transmission dynamometer requires calibration, so that for any given value of D_1—which may be termed the curve displacement—the power transmitted can be obtained. Otherwise, the instrument is useless. Calibration is effected by means of Prony brake tests. The power transmitted is computed by means of the formula—

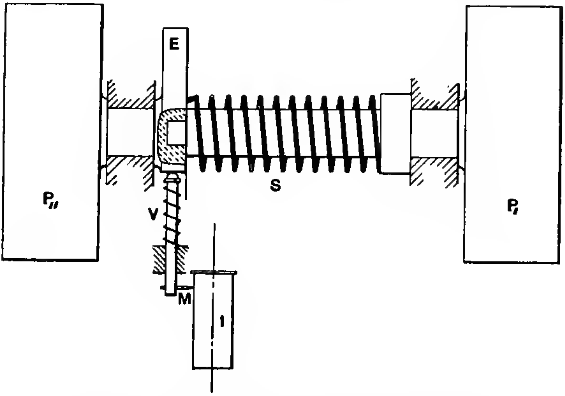


FIG. 92.—Torsion transmission dynamometer.

$$\text{Machine-tool I.H.P.} = \frac{D_1 \times N \times \text{constant}}{5{,}252} \qquad (69)$$

the value of the constant being determined from the brake test-results.

Where the load is variable, it is better to work out the power transmitted by integrating or planimeter-ing the area which is enclosed within the two curves. In this case, we have that:—

Machine-tool I.H.P.

$$= \frac{\text{Area of diagram} \times N \times \text{constant}}{33{,}000} \quad . \quad (70)$$

the value of this constant being related to that of the above.

In another form of torsion transmission dynamometer the torsion is measured by means of a series of pulleys and cords, the cords passing round the two parts of the shaft and controlling the movements of a scale and pointer. The scale is graduated, so that the constant for any given set of conditions is read off directly.

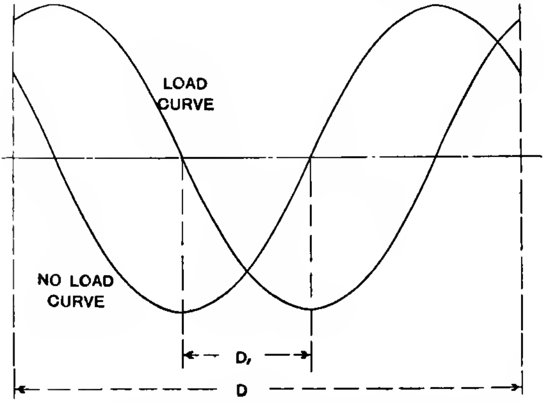


FIG. 93.—Torsion transmission dynamometer curve.

In the spring tension form of transmission dynamometer, helical springs are used between coupling flanges, the amount of their extension being measured through the medium of differential gearing. In the spring compression form, a spiral spring is employed, the relative movement of the outer end being communicated through a cam and shaft to an indicator pencil.

In the thrust form of transmission dynamometer the tangenital or twisting force is resolved into an axial force or thrust by means of a link or inclined plane device. This axial force is communicated through diaphragms to oil which is thus put under pressure. The characteristic formula is:—

$$\text{Machine-tool I.H.P.} = N \times P \times \text{Constant} \quad . \quad (71)$$

where N is the speed in revolutions per minute, and P is the oil pressure, in lbs. per square inch.

Cradle Dynamometer Method.—In the application of this method an electric motor is used. The carcase of the machine, carrying with it the field-magnet system, is arranged in a swivel or trunnion frame, and an arm attached to it is supported at its outer end on the pillar of a weighing machine. The torque on the armature which is exerted by the field magnets reacts on the latter. This reaction is weighed by means of the weighing machine, and is assumed to be a measure of the output of the motor. The formula to be employed in such a case is as follows:—

$$\text{Machine-tool I.H.P.} = \frac{L \times W \times N}{5{,}252} \qquad (72)$$

where L = the horizontal distance between the axis of the pillar of the weighing machine and the motor-armature axis (in feet); W = the net load as given by the weighing machine readings; and N = the speed of the armature, in revolutions per minute.

When the power transmitted is a rapidly varying quantity, as it is in the case of shaping machines, it is desirable to have some autographic method. One such method involves the tracing of a graph or curve on an indicator card, the horizontal axis representing time or distance and the vertical axis the movement of the end of an arm attached to the motor frame, this movement being proportional to the torque exerted on the frame with a fixed load on it.

Another method is to use a hydraulic piston and cylinder filled with oil or glycerine, and use in conjunction with these either an ordinary pressure or an engine indicator, preferably the latter in the majority of cases.

Strictly speaking, this method does not yield the true output of the motor since it is based on the pull between the motor armature and the motor field magnets, and does not cover the mechanical losses in the bearings at all. In fact, the output as determined by this method is equal to the product of the electrical input and the electrical efficiency. In many cases, however, the results obtainable by this method are sufficiently close for all practical purposes.

2. **Output Determinations.**—*Lathe.*—For the determination of the energy output of a lathe headstock a Prony brake may be used. The use of an absorption dynamometer under such circumstances is represented in Fig. 94, which illustrates a brake in the Machine-tool Laboratory of the University of Sheffield. A is a I girder section; B an oak block; D the brake drum (water cooled); S the headstock spindle; L the lathe bed; K a knife-edged pillar; W a weighing machine; and P a support for the

weighing machine. By means of such a brake it is possible to determine the headstock energy output for each energy input.

The computing formula is :—

$$\text{Headstock O.H.P.} = \frac{L \times N \times W}{5{,}252} \quad . \qquad (73)$$

where L = the length of the arm, and N and W

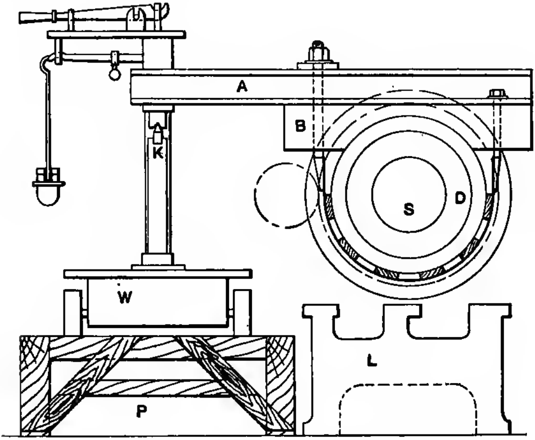


FIG. 94.—Strap brake as applied to lathe headstock.

have the meanings already given to them. In the actual case of the brake referred to, this became :—

$$\text{Headstock O.H.P.} = \frac{WN}{1{,}400} \quad . \qquad (74)$$

With this form of brake the load on the weighing machine is maintained as nearly constant as possible by the adjustment of the brake nut shown. In another form of friction test brake, the load on the arm is absolutely constant, being a dead weight, and any variation in the frictional resistance between the brake drum and the brake blocks is allowed for by a movement of the weighted arm and a consequent alteration in the real or effective length of the arm. The changes in this length can easily be recorded autographically on a chart such as the one shown in Fig. 95. In this figure H represents the instantaneous effective length of the arm and T the time. In this case, we have that:—

$$\text{Headstock O.H.P.} = \frac{H \times N \times W}{5{,}252} \quad . \qquad (75)$$

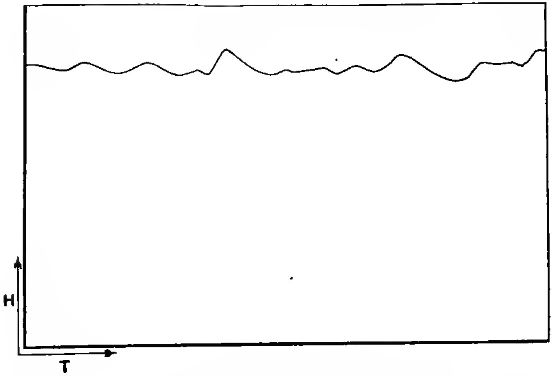


Fig. 95.—Brake diagram.

where H = the average length of the arm (obtained by averaging the diagram), and N and W have the meanings already ascribed to them.

The determination of the energy output of the slide-rest of a lathe can be determined by means of a dead weight (or a number of such) suspended at the end of a wire or cord and attached to the tool post after being passed over a deflecting pulley mounted on ball bearings, together with a dead weight on the slide-rest itself. The dead weight on the slide-rest in each case should be four times the suspended weight in order to approximate to the conditions of actual cutting. The output computation is effected as follows:—

$$\text{Slide-rest O.H.P.} = \frac{F_e \times N \times W}{396{,}000} \quad . \quad (76)$$

where F_e = the feed, in inches per revolution of the spindle, and N and W are as above.

Drilling Machine.—The spindle or twisting O.H.P. can be obtained in the same way as can the lathe headstock O.H.P., that is, by means of a brake test.

The feed O.H.P. can be readily obtained by suspending a dead weight from the spindle and raising this at a definite feed rate; the determination formula is as follows:—

$$\text{Feed O.H.P.} = \frac{F_m \times W}{396{,}000} \quad . \quad . \quad (77)$$

where F_m = the feed, in inches per minute.

Milling Machine.—The headstock O.H.P. is determined in precisely the same way as is the lathe headstock O.H.P.

The feed or table O.H.P. can be determined in the same way as is the slide-rest or feed O H.P. of the lathe.

Planing Machine.—The O.H.P. of the table or platen can be obtained by using a suspended weight attached to the table and raising it. Since, however, a planing-machine stroke is considerable, a number of pulleys (all with ball bearings) or a deep pit is required. The latter is to be preferred, though usually it is the less easy of the two to secure. When the required weight becomes considerable a ball-bearing pulley block with known efficiency can be employed. By this means the actual value of the weight required can be easily reduced to one-tenth, or even less than this.

Shaping Machine.—The O.H.P. of the ram of such a machine is a rather difficult datum to obtain owing chiefly to the rapidity with which the strokes follow one another. The dead-weight method is not applicable here because of this rapidity of changes. The only possible method is to work on some fluid which can either be compressed or raised a definite amount.

In one method the plunger of a hydraulic cylinder is secured to the tool post of the ram, the cylinder being connected to two water tanks, one a delivery tank and the other a suction or supply tank. In the case of the former the upper part of the tank is filled with air which is kept under constant pressure by means of a throttle or relief valve which is fitted to the tank below the water level and which allows water to pass from this tank to the suction or supply tank, the head of water in this over the cylinder or pump being just sufficient to overcome the frictional resistance in the suction valve and pipe. The average pressure during the forward or power stroke is obtained by means of an indicator diagram taken from the cylinder. During the return or suction stroke no work is actually done since the water just runs into the cylinder from the suction or delivery tank under the influence of the head which is maintained therein. This condition, of course, involves the use of special valves in the delivery and suction pipes. The cylinder I.H.P. is practically equal to the ram

O.H.P., the difference being an exceedingly slight one due to the frictional resistance between the plunger and stuffing-box packing. The determining formula is—

$$\text{Ram O.H.P.} = \frac{P \times L \times A \times N}{33{,}000} \quad . \qquad (78)$$

the symbols having the usual significance.

Another method of determining the ram O.H.P. is to use the ram to drive the plunger of a hydraulic cylinder as above, but to raise the water from one fixed level to another. The ram O.H.P. is then related to the external work done in one minute, a quantity which can be obtained as follows:—

$$\text{External Power} = \frac{Q \times H}{33{,}000} \quad . \qquad (79)$$

where Q = lb. of water raised per minute, and H = the sum of the suction and delivery heads. The ram O.H.P. equals the product of this quantity and the cylinder efficiency.

It is sometimes assumed that the cylinder I.H.P. is less than the ram or tool O.H.P. owing to the inertia of the plunger. This, however, is of no moment whatever for two reasons: first, the inertia or massiveness of the plunger is considerably less than that of the ram; and, second, the acceleration effect (at the beginning) and retardation effect (at the end) tend to balance one another.

3. **Tests and Test Results.**—In making machine-tool mechanical efficiency tests it is necessary to reproduce actual working conditions as far as possible, so that, for any given load on the machine under

test, the frictional losses in the various working parts of the machine are as nearly as possible equal to those which obtain when the machine is actually working under the same load.

In every case the useful power is the power which is consumed at the cutting edge or edges of the tool or cutter, and this includes both the power required to drive and the power to feed. All the remaining power is, practically speaking, consumed in overcoming the internal machine resistances, and it is these which should be reproduced exactly as far as possible in the brake and dead-weight tests. It is impossible, however, to accomplish this exactly in the majority of cases, though the degree of inaccuracy which is involved in the test results is usually inappreciable.

Lathe.—In the case of the lathe, the application of a friction brake on the driving spindle has practically the same effect on the frictional resistances as has the cutting of a tool. The difference between the two is in the thrust, which is created by the cutting tool, but which is practically non-existent in the case of the brake test. When it is desired or found necessary to make allowance for this thrust resistance, the thrust horse-power for various thrusts and spindle speeds can be reproduced experimentally by exerting an experimental axial thrust on the spindle through a large thrust ball bearing and dead weight or force applied through the medium of levers. The dead weight method can only be applied in the case of hollow spindle lathes. To approximate to actual cutting conditions, this applied axial thrust should in value be equal to one-quarter of the tangential force, measured at a distance from the spindle axis equal to, say, one-half of the height of the lathe centres, which corresponds to the load on the brake. This, stated algebraically, is :—

$$T = \frac{W \times L}{2H} \quad . \quad . \quad (80)$$

where T = the axial thrust required; W = the brake load; L the length of the brake arm; and H the height of the lathe centres above the top of the bed.

The measurement of the headstock O.H.P. remains the same, however, whether this refinement is adopted or not, and the efficiency of the headstock or the efficiency of the drive is determined as follows :—

$$\eta = \frac{\text{Headstock O.H.P.}}{\text{Headstock I.H.P.}} \quad . \quad . \quad (81)$$

The following table gives the results of headstock mechanical efficiency tests which were made by the author on an electrically driven 18-in. centre lathe, which is installed in the University of Sheffield. The motor I.H.P. and the headstock O.H.P. were obtained from meter and brake observations respectively. The motor O.H.P. (headstock I.H.P.) was obtained by making use of the efficiency curves illustrated in Fig. 86.

TABLE VII.

LATHE EFFICIENCY TEST RESULTS.

Driving Headstock.

Motor I.H.P.	Motor O.H.P.	Headstock O.H.P.	Efficiency, Per Cent.		
			Overall.	Motor.	Headstock.
6·81	3·82	1·70	25·0	56·0	44·6
7·59	4·43	2·30	30·3	58·3	52·0
7·70	4·50	2·38	31·0	58·6	53·0
8·49	5·09	3·10	36·5	60·0	60·9
9·95	6·57	4·38	44·0	66·0	66·7
10·84	7·49	5·20	47·8	69·0	69·4
12·33	8·90	6·40	52·0	72·3	72·0
12·78	9·26	6·80	53·1	72·8	73·2
16·20	12·31	9·38	58·0	76·0	76·0
18·35	14·11	11·00	60·0	78·0	76·8
21·80	17·38	13·26	60·8	79·5	76·9
25·70	20·51	15·80	61·6	80·0	77·0
35·75	30·01	29·32	82·0	84·0	97·5
42·00	35·50	33·73	80·4	84·5	95·2

From this table it will be seen that the three efficiency values rise with the load or headstock O.H.P. There appears, however, to be a limit to each value, and it is hardly probable that the ordinary working efficiencies are even nearly equal to the maximum. The following values may be taken as ordinary working efficiencies, these being the respective averages of the above values :—

Overall efficiency	50 per cent.
Motor ,, ...	71 ,,
Headstock ,, ...	70 ,,

The separate efficiency of motion of the slide-rest can only be determined exactly if the motion is obtained from a separate motor, or a transmission dynamometer is used. The following table gives the results of tests in such a case. The motor experimented with was a 5 B.H.P. variable-speed direct current motor.

TABLE VIII.

LATHE EFFICIENCY TEST RESULTS.

Slide Rest.

Motor I.H.P.	Motor O.H.P.	Slide Rest O.H.P.	Efficiency, Per Cent.		
			Overall.	Motor.	Slide Rest.
1·38	0·62	0·02	1·5	45·0	3·3
1·40	0·64	0·03	2·1	46·0	4·6
1·43	0·67	0·04	2·7	47·0	5·8
1·51	0·72	0·05	3·3	48·0	6·9
1·42	0·65	0·06	4·2	46·0	9·2
1·45	0·68	0·08	5·5	47·0	11·7
1·49	0·71	0·10	6·7	48·0	14·0
1·58	0·77	0·12	7·6	49·0	15·5
1·46	0·69	0·10	6·9	47·0	14·7
1·50	0·72	0·13	8·7	48·0	18·2
1·56	0·76	0·17	10·9	49·0	22·0
1·66	0·83	0·20	12·1	50·0	24·2

This table shows how comparatively small the slide-rest or feed efficiency of a lathe really is. The average slide-rest efficiency is only about 13 per cent.

On ordinary lathes the feed motion is derived from the driving headstock in which case the feed efficiency cannot be determined as a separate item without rather elaborate apparatus. It can, however, be incorporated with the headstock efficiency to give the overall efficiency of the lathe. In such cases, of course, the total O.H.P. is the sum of the two horse powers usefully employed in overcoming the tangential and longitudinal resistances, as actually determined by experiment.

When the two motions are separate, either for experimental or operation purposes, the overall efficiency of the lathe can be obtained by working on the two motor O.H.P.'s and the headstock and slide-rest O.H.P.'s. The ratio of the two sums is the efficiency, the formula which applies in this instance being—

$$\eta = \frac{\text{Headstock O.H.P.} + \text{Slide-rest O.H.P.}}{\text{Driving Motor O.H.P.} + \text{Feed Motor O.H.P.}} \quad (82)$$

In the following table are given the calculated results of such a process, the results given in Tables VII and VIII being employed for this purpose.

TABLE IX.

LATHE EFFICIENCY TEST RESULTS.

Complete Machine.

Motor O.H.P.	Lathe O.H.P.	Overall Lathe Efficiency, Per Cent.
4·44	1·72	38·8
5·08	2·36	46·5
5·14	2·41	46·8
5·78	3·20	55·4
7·25	4·46	61·7
8·16	5·24	64·2
9·62	6·45	67·0
9·98	6·93	69·4
13·02	9·48	72·9
14·88	11·12	74·8
18·14	13·43	74·3
21·34	16·00	75·0

This table shows that the limiting overall efficiency of a lathe is the neighbourhood of 75 per cent. The average overall efficiency of a lathe is about 62 per cent, this being the mean of the above values.

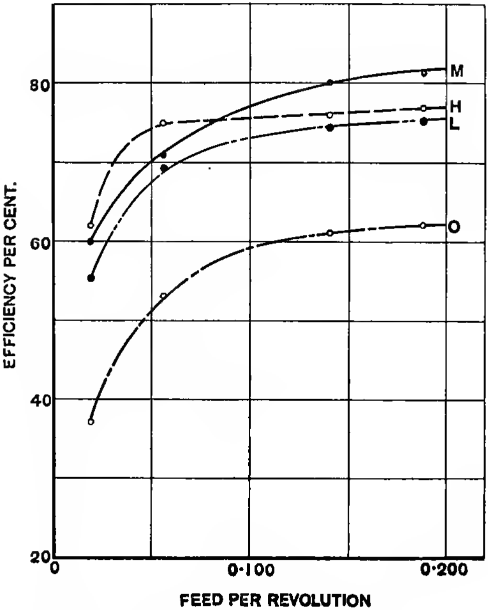


Fig. 96.—Lathe-headstock efficiency curves.

It will be observed that each value is less than its correspondent in the headstock efficiency tests.

When, with a single source of power, the slide-rest is loaded, as already described, to enable the feed O.H.P. or net power to be obtained, the pull on the slide should be one-half of the tangential force on the headstock brake drum measured at a radial distance from the axis equal to the height of the centres. The dead load on the slide-rest should be one-quarter of this. The test conditions will then approximate closely to the ordinary conditions of cutting.

Another method of determining either headstock, slide-rest, or overall lathe efficiency involves the taking of a cut, the no-load and load lathe I.H.P.'s being obtained according to the kind of drive and measuring methods employed. In the no-load test the slide-rest should be in motion at the feed of the cutting test. The efficiency is then determined as follows:—

$$\eta = \frac{\text{Load I.H.P.} - \text{No-load I.H.P.}}{\text{Load I.H.P.}} \qquad (83)$$

This method, it will be seen, is based upon the assumption that the frictional losses during the load test are the same as those which occur during the no-load test. Such an assumption is obviously not quite true, but in many cases the error which is involved when this assumption is made may be ignored.

In addition to relating the overall efficiency to the lathe I.H.P. or O.H.P., it is also possible to relate it to, say, the feed, provided that this is the only variable in the conditions of cutting. In such a case it is necessary to select a definite spindle speed, and then to make the preceding tests at this speed, obtaining data similar to those recorded in Tables VII, VIII, and IX. The lathe is then furnished with a test bar of a suitable length and diameter, and at the same angular speed as the above cuts of a fixed depth, but with a variable feed, are taken, observations being made to obtain the required results.

In Figs. 96 and 97 are shown graphically the results of a series of such tests, in which the cutting speed was 50 ft. per minute, the depth of cut was $\frac{3}{8}$ in., and the feeds worked at were 0·0185 in., 0·0551 in., 0·1417 in., and 0·1880 in. per revolution of the test bar. The curves in Fig. 96 are headstock curves; those in Fig. 97 are slide-rest curves. The curves M are motor efficiency curves; those lettered H and S are respectively headstock and slide-rest efficiency curves; whilst the O curves are overall efficiency curves. The two most important curves are the H and S curves. On Fig. 96 has also been

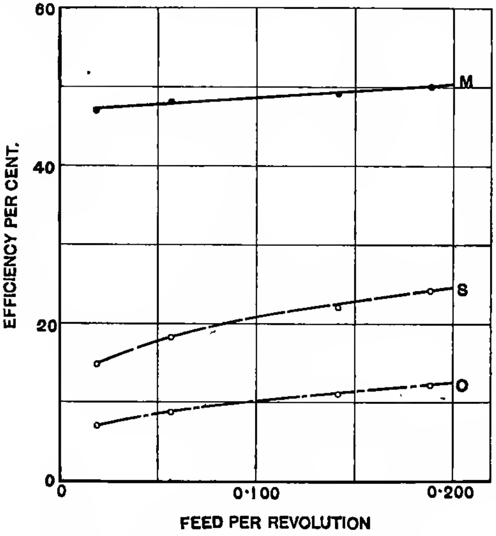


Fig. 97.—Lathe-slide-rest efficiency curves.

drawn the overall lathe efficiency curve; this is lettered L. A comparison of this with H will show that they are both of the same type of curve. It will also show that the disparity between them grows less as the feed (and, therefore, the load) gets greater. This is due to the rapid improvement in the slide-rest efficiency with feed or load increases.

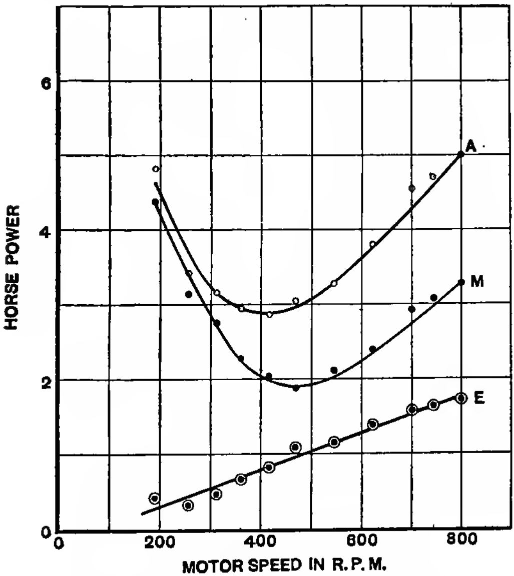


FIG. 98.—Lathe and motor power curves.

In Figs. 98 and 99 are represented graphically the amounts of power required to drive the various elements of the motor and headstock of this lathe. M is the no-load motor curve; A is the no-load headstock and motor curve without face-plate gearing and spindle in; E is the corresponding no-load headstock curve; B is the motor, headstock, faceplate, and test bar curve; F is the test bar curve.

In Fig. 100 are shown the headstock horse-power curves for three different depths of cut, namely, $\frac{1}{8}$, $\frac{1}{4}$, and $\frac{3}{8}$ in., and a cutting speed of 50 ft. per minute.

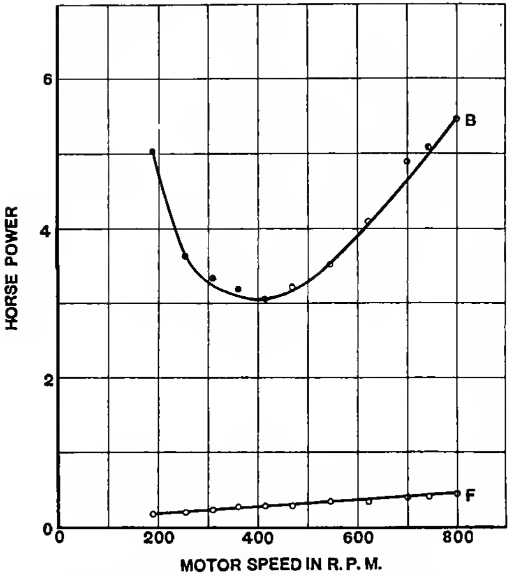


Fig. 99.—Lathe and motor power curves.

The full lines represent I.H.P., and the dotted lines O.H.P.

The corresponding curves for the slide-rest are given in Fig. 101.

Drilling Machine.—In the drilling machine the application of a friction brake has practically the same effect on the drill spindle as drilling has, so far as the turning moment or torque is concerned. In addition to the torque horse-power, and the power which is required to overcome the radial bearing resistances, there is a power required to overcome the axial resistances which are set up by the thrust. This latter power can be practically allowed for by suspending from the driving spindle a known dead weight through a ball bearing, the numerical value of the weight in lbs. being equal to the value of the

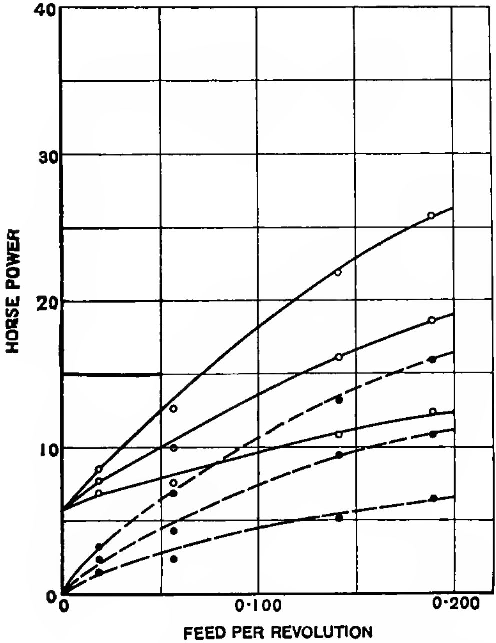


Fig. 100.—Curves showing relation between lathe headstock power and feed of tool.

torque (product of the length of the brake arm and the load on the brake) in lb.-ins. This relationship between the thrust and the torque is an average one, the extreme values of it being about 1·5 and 0·5. In this case the brake drum has to be supported on a large ball bearing, and connected to the driving spindle by means of a feathered shaft, so as to admit of the vertical movement of the latter as it raises the weight representing the thrust. As drilling-machine feeds are usually derived directly or indirectly from the driving spindle, it is not possible ordinarily to

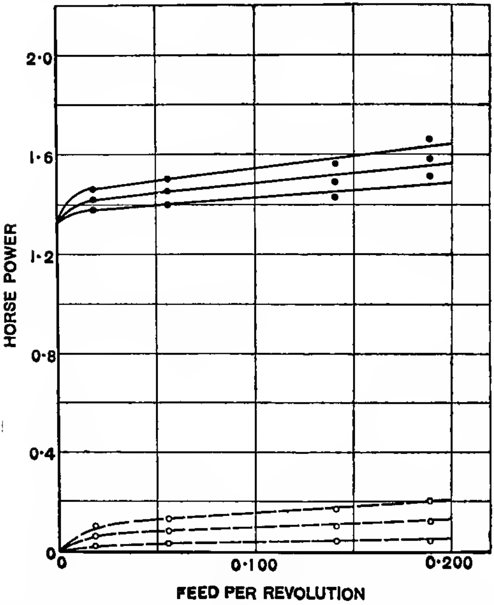


Fig. 101.—Curves showing relation between lathe slide-rest power and feed of tool.

separate the feed and the drive, though, of course, in special cases it can be done. In either case, the feed test can be made at the time of the drive test by causing the feed motion to raise the dead thrust weight.

Generally, it is the overall machine efficiency which is required, and only in special cases is it desired to know the values of the two separate efficiencies.

The quantities required are the load on the brake, the length of the arm of the brake, the thrust weight, the spindle speed in revolutions per minute, the unit-time feed (in inches per minute), and the machine I.H.P. or the O.H.P. of the power source.

Another method of determining the mechanical efficiency of a drilling machine is to obtain the machine I.H.P. when it is running light with the feed motion in action, and also when it is loaded with a cut. The difference between the two I.H.P.'s is taken to be the O.H.P. of the machine, and the efficiency is obtained by means of expression (82). The basic assumption of this method has already been discussed in connection with the efficiency testing of lathes.

In the following table are given the results of some mechanical efficiency tests on an electrically-driven high-speed drilling machine. In these tests the second method was adopted, the machine I.H.P. being determined from the motor I.H.P. by making use of the motor-efficiency curve.

The average value of the efficiency is 77 per cent, which may be taken to be of fairly general application in practice.

TABLE X.

DRILLING MACHINE EFFICIENCY TEST RESULTS.

Complete Machine.

Motor O.H.P.	Machine O.H.P.	Overall Machine Efficiency, Per Cent.
1·04	0·70	67·3
1·18	0·87	73·7
1·49	1·18	79·2
2·16	0·96	44·4
2·57	2·15	83·6
3·40	2·40	70·6
4·19	3·33	79·8
4·53	3·53	77·9
4·58	4·28	93·5
5·13	3·80	74·1
5·75	5·00	87·0
6·11	5·71	93·3

Milling Machine.—In milling machines in which the power feed is derived from the driving spindle it is not feasible to make separate tests for the headstock and table or feed efficiencies. With a separate drive for the feed it is possible, however, to do this.

The driving spindle of a milling machine is subjected chiefly to torsion, there being very little thrust effect in the majority of cases, and what there is in the others may be neglected. Therefore, the simple brake test will cover all the headstock losses and enable the headstock efficiency to be obtained in the usual way.

The table of a milling machine is subjected to a pull against the resistance of the cut and a vertical force. The latter, of course, sets up increased frictional resistance in the table guides, and should, therefore, in the tests be allowed for. This can be done by means of a dead weight surmounting the table. The value of this weight, for an average, should be one quarter of the suspended weight. The efficiency determined in such a case will be as nearly correct as it is possible to determine it, experimentally or otherwise.

The overall machine efficiency can be obtained by combining the two separate efficiencies. In such a case, the weight attached to the table must bear some reasonable relation to the load on the brake. The following formula will give it for all ordinary cases :—

$$W_1 = \frac{W \times L}{2} \quad . \quad . \quad (84)$$

where W_1 = the suspended weight, in lb. ; W = the brake load, in lb. ; and L = the length of the brake arm, in inches. In this formula an average cutter diameter of 4 in. is assumed. In these tests it is, of course, possible to ring the changes on the spindle speed and the feed.

If the two drives are separate, expression (82) or one of the same form, applies. If they are interconnected, the overall machine efficiency is determined by means of the following expression :—

$$\eta = \frac{\text{Headstock O.H.P.} + \text{Table O.H.P.}}{\text{Machine I.H.P.}} \quad (85)$$

The differential test is also applicable in this case. The machine I.H.P. is obtained without a load, both the spindle and the table being set in motion at the working speed and feed respectively. A load is then put on the machine in the nature of a cut, and the machine I.H.P. determined for this condition. The overall machine efficiency is calculated by means of expression (83).

In the following table are given the results of some differential tests on a high-power horizontal spindle milling machine.

TABLE XI.

MILLING MACHINE EFFICIENCY TESTS.

Complete Machine.

Machine I.H.P.	Machine O.H.P.	Overall Machine Efficiency, Per Cent.
3·10	2·58	84·6
3·70	3·20	86·5
4·25	3·64	85·9
5·58	4·87	87·1
6·50	5·70	89 0
7·62	6·53	85·8
8·25	7·14	86·6

The average efficiency is 86·6 per cent, which is fairly representative of modern practice. It will be noticed that, in common with other machine-tools, the efficiency value rises somewhat with the load, though not in the same ratio.

Shaping Machine.—With this type of machine-tool the ram efficiency is the machine efficiency since the cross feed of the work is effected either during, or at the end of, the return stroke and has no cutting resistance to overcome. Furthermore, since the feed motion is invariably derived from the driving motion, it is not possible without special apparatus to determine the power which is required to feed the work across the motion of the tool. In any case, however, this power is lost in overcoming frictional resistances and, hence, cannot appear in the O.H.P. of the machine.

There are at least three ways of determining the mechanical efficiency of a shaping machine. The first method covers just the power or cutting stroke. The input per stroke is determined by means of the cradle dynamometer or other recording dynamometer, the motion of whose diagram card is controlled by the ram. The output per stroke is determined by means of the hydraulic method described, the output being obtained from the cylinder indicator card, whose motion also is controlled by the ram. The efficiency during that stroke is obtained by taking the ratio of the output to the input. To obtain a fair average it is necessary to take a number of simultaneous cards. In the following table are summarized the results of a number of such tests on a 16-in. shaping machine:—

TABLE XII.

SHAPING MACHINE EFFICIENCY TESTS.

Power Stroke Only.

Length of Stroke, in In.	Average Load on Tool, in Lb.	Average Speed of Ram, Strokes Per Minute.	Efficiency, Per Cent.
2	761	27·1	34·0
4	445	34·5	37·7
6	622	27·1	42·3
8	698	29·6	41·7
10	725	27·1	51·0
12	448	32·1	39·3
14	761	21·3	43·7
16	553	25·3	45·0

The average value of the efficiency is 41·8 per cent, which is considerably less than the corresponding value for any machine-tool whose main motion is uni-directional.

In the second method either an electric motor (of known or determinable efficiency) with an integrating or recording wattmeter or a continuously recording transmission dynamometer is employed to give the average machine I.H.P. over a given period of time. This includes the average machine or ram O.H.P., the driving losses during both power and return strokes, and the power required to give the cross feed to the table or knee of the machine. The average ram O.H.P. during the same time can be determined by either of the two hydraulic methods already described. In the case of the former either a continuous indicator diagram must be obtained, or representative diagrams taken and used in conjunction with the number of power strokes made per minute. In the latter case, the total quantity of water pumped up in the given time is measured and the quantity pumped per minute calculated from this; it is then used in expression (79). The efficiency is determined in the ordinary way, and is generally slightly less than the above.

In these tests, in which no cutting is involved, the table should be loaded to the extent of about one-quarter of the average load on the tool in order to approximate to actual cutting conditions.

The third method is the differential method. Owing, however, to the extent of the changes which occur in the power demand on the part of the machine an integrating or recording wattmeter has to be used in the motor circuit and the test run, first, with no load, and afterwards with a cut, for some considerable time, in order to admit of fair averages being obtained. The efficiency is obtained from the average I.H.P.'s in the usual way by means of expression (83).

Planing Machine.—The case of this machine is, speaking generally, not very different from that of the shaping machine; therefore, the above remarks with the modifications implied in the previous paragraphs on planing machines apply here.

CHAPTER V.

CUTTING FORCE TESTS.

THE employment of high-speed tool steel in metal-cutting tools of all descriptions, with its concomitant high cutting speeds and heavy cuts, has rendered the question of the forces which occur during cutting operations of prime importance, since the ability of a machine-tool to work at high speeds with heavy cuts depends, *inter alia,* upon the power possessed by it to resist the influence of the cutting forces. This power is, obviously, related to the mass and distribution of the metal in the machine; and the scientific solution of this problem is based upon a knowledge of the magnitude and influence of the forces which occur during the cutting operation or operations.

Cutting or cutting-action forces can, in general, be determined in three different ways, in each of which actual cutting is involved. These are:—

1. The direct weighing or dynamometer method;
2. The brake and feed-test method; and
3. The differential-reading method.

1. **Direct Weighing or Dynamometer Method.**—In this method the operative cutting force or forces are balanced by a suspended weight or weights according to the principle of moments, or by a hy-

draulic pressure or pressures either according to the same principle or directly.

Lathe.—Without dealing with any theory of cutting-action, it may be stated that the total force which acts on a lathe tool during cutting is the resultant of a large number of small forces which are not all of the same magnitude, neither do they all act in the same direction. This force can, however, be resolved into components acting in three rectangular co-ordinate planes, these being a vertical plane normal to the direction of cutting and a vertical and horizontal plane each parallel to the direction of cutting. Two of the components are thus horizontal and one is vertical. These are represented in their correct relationship in Fig. 102. The vertical force, V, is usually regarded as a pressure and the most important force of the three; N is called the normal horizontal force; and F the feeding or parallel horizontal force.

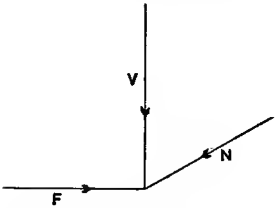


FIG. 102.—Lathe cutting forces.

The force V is related directly to the headstock O.H.P., whilst the force F is related directly to the feed or slide-rest O.H.P. The force N sets up increased frictional resistance both to the rotation of the spindle and the rectilinear motion of the slide-rest, and is therefore involved in both O.H.P.'s.

The principle of a direct weighing method devised by the author for experimentally determining the values of the three forces V, N, and F is indicated in Fig. 103. In this method the tool, T, is secured in a gimbal tool holder, G, which consists of a hollow block as shown, the tool being pivoted in this with its axis horizontal as near to its cutting end as possible. The connection between the two is either a pair of adjustable conical-ended screws which pass through the sides of the block and form pivot bearings in the two sides of the shank of the tool or two adjustable thrust ball bearings, though in the latter case side plates have to be fitted specially to the shank of the tool. This block is fitted as near to the nose of the tool as possible, and is pivoted in another block, B, which is secured to a free horizontal cross slide, S, on the slide-rest, the axis of pivot in this case being vertical and the pivot in the form of either conical points or thrust ball bearings.

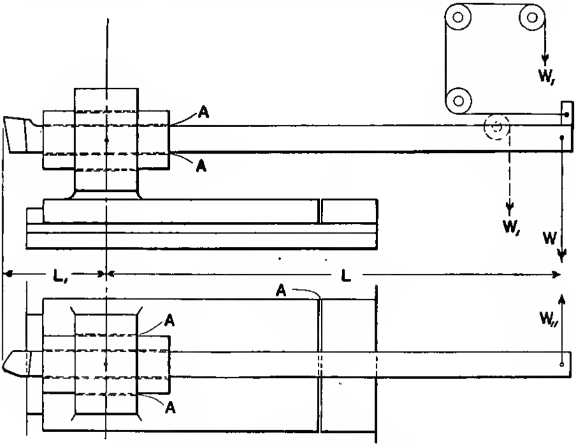


Fig. 103.—Lathe dynamometer.

With such an arrangement the nose of the tool is capable of more or less restricted motion in the three co-ordinate directions indicated in Fig. 102, there being, however, sufficient freedom of movement in each direction to admit of the respective force to be balanced. To obtain the position of balance in each direction three feelers or thickness gauges are used at the places lettered A, the spaces within which they fit having widths equal to the thicknesses of the respective feelers correct to within 0·001 in. The contact faces are all accurately machined and finished by scraping.

The vertical cutting force is balanced by the weight W, its computation being effected by means of the following formula :—

$$V = \frac{W \times L}{L_1 - d} \quad . \quad . \quad (86)$$

where W, L, have the meanings given to them in the figure, and d = the distance between the extreme point in the cutting edge of the tool and the position of the centre of pressure measured in a direction normal to the axis of the work or test bar. The magnitude of d with respect to the depth of the cut varies somewhat, depending, as it does, chiefly upon the kind and physical characteristics of the metal in the test bar, the relation between the depth of the cut and the feed, the cutting speed, aud the shape of the nose of the tool. As an average result, the value of d may be taken to be equal to the depth of the cut.

The normal horizontal force is balanced by the weight W_1, which is attached to the bar through a cord or other flexible connector which has to be passed over one or three ball-bearing pulleys as shown. In this case, the value of W_1 equals the force measured, that is, N.

The parallel horizontal force is balanced by the weight W_{11}, which acts through a cord or other flexible connector which passes over a ball-bearing pulley. The calculation of the force value is effected by means of the following expression :—

$$F = \frac{W_{11} \times L}{L_1 - d} \qquad . \qquad . \qquad (87)$$

L, L_1, and d having the above meanings.

The weight W can also be measured by means of a weighing machine directly; W_1 through a bell-crank lever (ratio of arms unity or any other definite value); and N also through a bell-crank lever.

In the Nicolson lathe-tool dynamometer,[1] a somewhat similar method of supporting the tool is employed. A gimbal tool holder is used, but in this case it grips the tool at its non-cutting end. The vertical cutting pressure is resisted by a strut which is placed directly under the nose of the tool and connected, either directly or by means of a lever and knife-edge bearings, to an oil diaphragm pressure vessel and pressure gauge. The normal horizontal cutting force is measured by direct transmission through a strut and diaphragm pressure vessel to a pressure gauge. The parallel horizontal cutting force is measured by means of two struts, diaphragm pressure vessels, and pressure gauges, one set on each side of the tool near to its rutting end. The force in each case is obtained from the pressure-gauge readings, the value used in connection with the de-

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[1] "Proc. Inst. Mech. Engs.," 1905.

termination of the parallel horizontal force being the difference between the readings of the two gauges, which have initial readings given to them.

The results which have been obtained by means of this method appear to indicate that the magnitude of the specific cutting pressure or vertical cutting force is influenced by (1) the cutting speed; (2) the depth of cut; (3) the feed; (4) the nature of the material being machined; and (5) the general shape of the tool.

By the application of autographic or recording indicators to the above dynamometer it should be possible to obtain a pressure curve from which the average and instantaneous values of the vertical cutting forces would be obtainable.

In a slightly different form of lathe-tool dynamometer a thin steel diaphragm is used as the tool-holder pivot in place of gimbals, the tool in such a case having only a slight freedom of movement, but one quite sufficient under the circumstances.

Drilling Machine.—In this case, generally, the total cutting force can only be resolved into two components: a tangential force and an axial force or thrust. With an improperly ground drill or a test piece of non-homogeneous material there is a third force, namely, a normal force but, theoretically, in a drilling operation this does not exist.

The tangential force (which is the main cutting force) sets up a torque or turning moment, and either the value of this or the value of the force itself referred to, say, the circumference of the drill, can be determined by setting up a counter resisting moment. The thrust can be measured by setting up a measured or measurable resistance.

With a rotating drill both sets of measurements are obtained from the work; with rotating work they are obtained from the drill. In the case of the first, it is necessary to have recourse to the use of a floating work table and piston. The piston must fit in a large vertical cylinder. When the thrust is actually balanced by a dead counterweight or a weighing machine resistance, the cylinder acts as a bearing, and it should then preferably be equipped with ball bearings to reduce the frictional resistance, and the connection between the chuck and piston should be of the gimbal or universal jointed type.

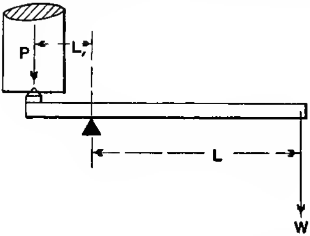

Fig. 104.—Drilling machine dynamometer.

To measure the thrust in this case a lever has to be used, one end of which is fitted with a vertical cone which fits in a central conical recess in the lower end of the piston. This lever is balanced and weighted as shown in Fig. 104, in which P represents the thrust and W the applied net weight. The value of P is obtained as follows:—

$$P = \frac{W \times L}{L_1} \quad . \quad . \quad (88)$$

When a weighing machine is used, the extra lever shown in Fig. 105 has also to be used. The value of T in this case is determined as follows:—

$$P = \frac{W_1 \times L \times L_{111}}{L_1 \times L_{11}} \quad . \quad . \quad (89)$$

The torque or turning moment is obtained by attaching a strong connector to the rim of the table and taking a strong wire or cord over a pulley and attaching the weights to the end of this wire or cord. The torque is then determined by means of the following formula :—

$$T = W \times R . \quad . \quad . \quad (90)$$

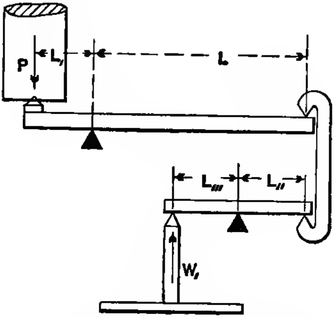

FIG. 105.—Drilling machine dynamometer.

where T = the torque in, say, lb.-in. ; W = the suspended weight, in lb. ; and R = the rim radius, in in.

By the use of a rigid connecting lever and a bell-crank lever fitted with knife edges, it is possible to measure the tangential force by means of a weighing machine, though in this case the arm of the moment will be the distance between the axis of rotation and the point of contact between the straight lever and the knife edge of the bell-crank lever. If the bell-crank lever ratio is 1 : 1, the weighing machine reading will give the force applied at the end of the table lever.

The hydraulic method is also capable of application in connection with the determination of both the thrust and the torque. In the case of the former, the cylinder is filled with oil or glycerine, and the thrust is transmitted through this to either a pressure gauge (recording or indicating) or an engine indicator. In the case of the latter, the table lever presses on a piston in an oil cylinder or on a diaphragm pressure vessel, the pressure so created being transmitted to either a pressure gauge (recording or indicating) or an engine indicator. When an engine indicator or recording pressure gauge is used, the movement of the card or chart should be controlled from the spindle or spindle saddle of the drilling machine.

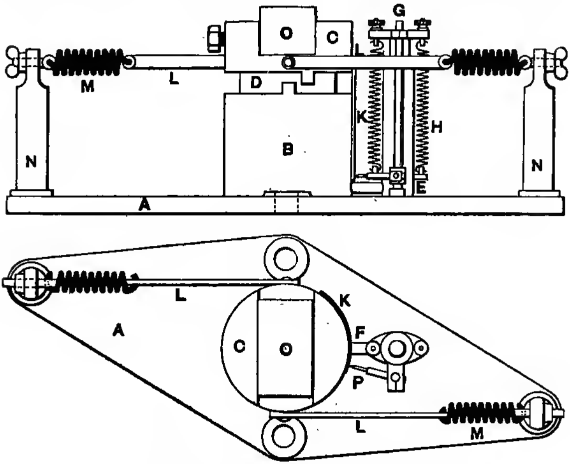

Fig. 106.—Twist-drill dynamometer.

In Fig. 106 is illustrated a form of autographic twist-drill dynamometer by means of which the relation between the torque and thrust can be determined. The springs, MM, resist the turning moment and allow the chuck, C, which carries the chart or card, K, to move through an angle which is proportional to the torque. The springs, H, resist the thrust, the movement of the pencil P being proportional to the thrust.

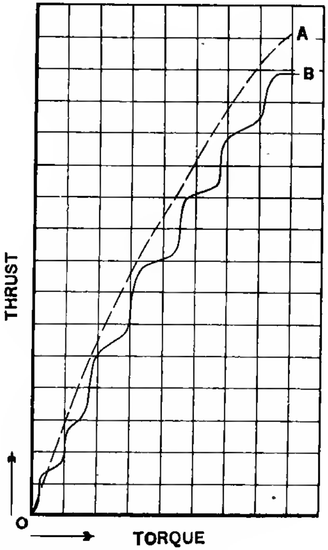

FIG. 107.—Twist-drill dynamometer curves.

In Fig. 107 are shown two representative graphs, the graph OA representing a more homogeneous metal than does the graph OB.

Milling Machine.—In Fig. 108 is represented one form of milling cutter dynamometer, as devised by Prof. Poliäkoff for determining the tangential or horizontal and vertical forces which operate when a straight or axial-tooth milling cutter is working. The test piece is secured to a sliding sub-table, D, which is mounted on a hinged fixture, C. This fixture is supported by the hinge bracket, B, and the point, H. This point, H, is connected to a diaphragm pressure vessel F, by means of which and a pressure gauge the downward cutting force is determined. The front end of the table D is in contact with a point G, which is connected to a diaphragm pressure vessel, this being in communication with a pressure gauge or indicator by means of which the tangential or horizontal cutting force is determined.

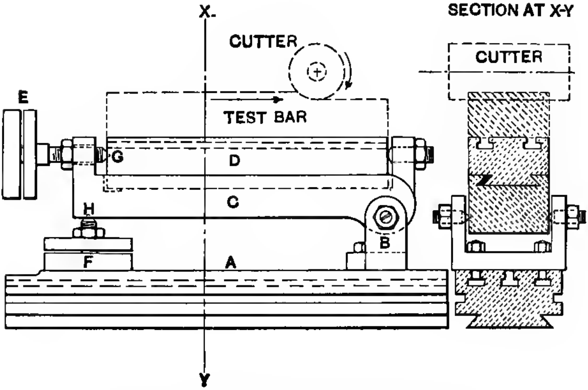

FIG. 108.—Milling-machine dynamometer.

Planing Machine.—The cutting force in this case can be treated in exactly the same way as the cutting force in the case of the lathe tool; that is, it can be resolved into three rectangular components. Each of these can be weighed or measured in each of the ways specified in the case of the latter; that is, by direct weighing, by pressure dynamometer, or by indicator.

With one form of dynamograph or autographic dynamometer the type of curve which is traced is indicated in Fig. 109, which represents a pressure-stroke diagram. It will be seen from this that the pressure is by no means constant—a fact which is true in regard to other tools and machines as well as this. In Fig. 110 are shown typical curves for different metals. The index to this diagram is as follows: A, brass; B, soft cast iron; C, hard cast iron; D, wrought iron; E, mild steel; and F, annealed tool steel. Taking soft cast iron as the basis and representing it by unity we have the following comparative pressure values for the different metals:—

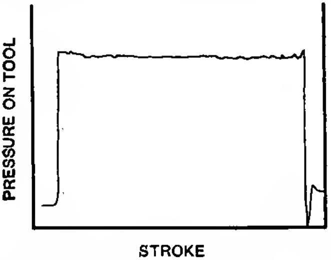

Fig. 109.—Planing-machine dynamometer curve.

Brass	1·1
Soft cast iron	1·0
Hard cast iron	1·8
Wrought iron	2·5
Mild steel	2·7
Hard steel	2·8

2. **Brake and Feed Test Method.**—By means of this method it is only possible to obtain the values of the main cutting force and the feeding force at the nose of the tool, since it involves the determination of the powers which are absorbed at the nose or edge of the tool or cutter in driving the work (or cutter) and in feeding the cutter (or work). It is related directly to the mechanical efficiency tests in which actual loads and weights are used, and involves the use of efficiency or input-output curves.

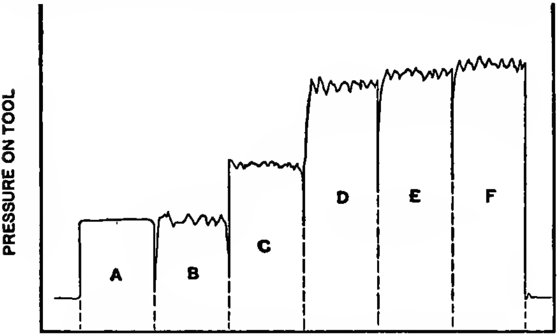

FIG. 110.—Planing-machine dynamometer curves.

Knowing in any case the machine input (either for the drive or for the feed) we can readily determine the output by using the efficiency value or the determined relation between the machine I.H.P. and O.H.P. or referring to the efficiency or input-output curve. The main cutting force is calculated by means of the following expression:—

$$V = \frac{\text{O.H.P.} \times 33{,}000}{S} \quad . \quad . \quad (91)$$

where V is the force, in lb., and S is the cutting speed referred to the outside circumference of the work or test bar in the ordinary case. The cutting pressure intensity is obtained as follows:—

$$P = \frac{V}{A} \quad . \quad . \quad . \quad (92$$

where A = the area of cut, in square inches.

The feeding or traversing force is computed as follows:—

$$F = \frac{\text{O.H.P.} \times 396{,}000}{F_m} \quad . \quad . \quad (93)$$

where F = the force, in lb., and F_m the feed, in inches per minute. The intensity of pressure is calculated as in the above case.

3. **The Differential Method.**—This method only differs from the above in regard to the means adopted to obtain the O.H.P. of the machine. In this case the difference between the load and no-load I.H.P.'s is assumed to be equal to the O.H.P., the validity of which assumption has already been referred to.

Test Results.—The published results of cutting-force tests show exceedingly wide variations, both in regard to the values of the cutting-forces and in regard to the relations that these bear to other cutting-operation data.

In the following table are given the values of the specific vertical cutting-force or cutting-pressure intensity which may be taken as the general average values which are applicable in ordinary lathe-design practice.

TABLE XIII.

LATHE-TOOL TESTS.

Specific Vertical Force, in Tons per Square Inch.

Material.	Soft.	Medium.	Hard.
Steel . . .	100	110	130
Cast Iron . .	50	60	70

In regard to the values of the horizontal forces there is much disagreement. The late Dr. Nicolson stated that the results of his Manchester tests indicated that the normal horizontal force varied from 18 per cent to 78 per cent of the vertical cutting force, with an average of 50 per cent, and that the parallel horizontal force (which he called the traversing force) was about 25 per cent of this latter force. Mr. Taylor's American results[1] indicate, on the other hand, that these percentages are low, the operative values being not less than 100 per cent in each case.

Further, the first-named did not find that the depth of cut or the thickness of the chip had any marked effect on the specific cutting pressure, whereas the last-named stated that the actual pressure acting on a lathe tool varied as indicated by the formula :—

$$V = CD^{\frac{14}{15}}F^{\frac{3}{4}} \quad . \quad . \quad . \qquad (94)$$

in which C is a constant of an average value of 57,000 for cast iron, and a maximum value of 69,000 ; D is the depth of cut, in inches ; and F is

---

[1] "Proc., A.S.M.E.," 1906.

the linear feed per revolution. The pressure intensity is obtained by means of the following formula:—

$$P = \frac{C}{D^{\frac{1}{15}} F^{\frac{1}{4}}} \quad . \quad . \quad . \quad (95)$$

The most exhaustive cutting force tests on twist drills are those which were made by Messrs. Smith & Poliakoff at Manchester. The following formulæ represent the most important results of these tests, these formulæ, according to the experimenters, being simple approximate formulæ of general application:—

For cast iron—

$$\text{Thrust} = 35{,}500 d^{0·7} F_e^{0·75} \quad . \quad (96)$$

$$\text{Torque} = 740 d^{1·8} F_e^{0·7} \quad . \quad . \quad (97)$$

For steel—

$$\text{Thrust} = 35{,}500 d^{0·7} F_e^{0·6} \quad . \quad (98)$$

$$\text{Torque} = 1{,}640 d^{1·8} F_e^{0·7}. \quad . \quad (99)$$

In these formulæ d represents the diameter of the drill, in inches, and F_e the feed in inches per revolution. The thrust is in lb. and the torque in lb.-ft.

In some of the earlier American tests the results suggested that the torque varied as the 0·6 power of the feed, a result which does not differ greatly from the above.

The use of a lubricant has been shown to reduce the torque by 18 per cent and the thrust by 25 per cent on the average.

CHAPTER VI.

OUTPUT AND POWER CONSUMPTION TESTS.

By the "output" of a machine-tool is meant the maximum quantity (volume or weight) of metal which can be removed from the stock in the unit of time, say one hour or one minute. This depends, in any case, upon many factors, some of which are actually extraneous to the machine itself, such as the general form of the cutting tool or tools, the tool angles, the kind of tool steel of which the tools are made, the relation between speed and area of cut, and the nature of the material which is being machined. Given constancy in these conditions, however, the output of a machine-tool is a measure of its capability, whilst its cutting efficiency is related to quantity of metal which is removed per unit of energy, such as the H.P. minute or the H.P. hour.

There are machine-tool operations in which the immediate object is the machining of surfaces, and the removal of a volume or weight of metal is an incidental operation. In such cases, of course, output may be regarded as connoting superficial area, though, from the general point of view, even here it is possible to determine a volumetric or gravimetric output.

1. **Determination of Volumetric Output.**—In any given time, say, T minutes, at an average cir-

cumferential speed of S ft. per minute with an area of cut of A square inches, we have that:—

Volume removed = $12S \times A \times T$ cubic in. . (100)

and cubic in. per minute = $12SA$ . . (101)

If we have determined the angular speed instead of S, then we have that:—

Cubic in. per minute = πDNA . . (102)

where D = the mean diameter, in inches, and N the mean angular speed in revolutions per minute.

The above expressions apply generally to the lathe and boring mill, the mean diameter being taken as that midway between the outside and bottom diameters. In some cases, however, the outside diameter is taken to simplify matters, though obviously this is a slight approximation.

Expressions (100) and (101) can be made to apply to the planing machine and shaping machine by substituting average cutting speed for average circumferential speed.

In the case of the drilling machine, we have that:—

Cubic in. removed per minute = $0{\cdot}7854\, D^2F_m$. (103)

where D = the drill diameter, in inches, and F_m the feed, in inches per minute. In another form this is:—

Cubic in. removed per minute = $0{\cdot}7854\, D^2F_eN$. (104)

where F_e = the linear feed, in inches per revolution, and N the angular speed of the drill, in revolutions per minute.

With a milling cutter on plain work, we have that:—

Cubic in. removed per minute = $F_m dw$ (105)

where d = the depth of cut, in inches, and w the width of the cut, also in inches.

2. **Determination of Power Consumption.**—This can be made in the several ways already described in connection with the experimental determination of mechanical efficiency. In the case of an electrically driven machine-tool there are three powers, these being the motor I.H.P., the machine I.H.P. or motor O.H.P., and the machine-tool O.H.P. The relations between the volumetric output and these quantities are, in any given case, different, since the relations between these quantities themselves involve the motor and machine-tool efficiencies.

3. **Test Results.**—In the case of the lathe, the volumetric efficiency varies from 80 to 120 cubic in. per machine per O.H.P. hour, the lower values being associated with the harder metals. There is, of course, a connection between this and the magnitudes of the cutting forces. From this we get that the machine O.H.P. required to remove 1 lb. of metal per minute varies from 1·8 to 2·7, the higher value being associated with the harder metals. These values are for steel. For cast iron, the corresponding values are 150 to 220 cubic in. per machine, O.H.P. hour, and 1·0 to 1·6 machine O.H.P., per lb. of metal per minute.

In the case of the drilling machine, the volumetric efficiency varies from 24 to 60 cubic in. per machine O.H.P. hour for steel, and from 55 to 140 for cast iron; whilst the machine O.H.P. per lb. of metal removed per minute varies from 3·6 to 9·0 for steel and from 1·6 to 4·2 for cast iron.

With milling machines the volumetric efficiency varies from 30 to 60 cubic in. per machine O.H.P.

hour for steel and from 60 to 120 for cast iron; whilst the machine O.H.P. per lb. of metal removed per minute varies from 3·6 to 7·2 for steel and from 1·8 to 3·6 for cast iron.

From the practical point of view it is usually the volumetric efficiency based upon the machine I.H.P. which is the quantity of importance. Obviously, this includes the mechanical efficiency, which, as has been indicated, is a somewhat variable quantity. The following table gives the average values of this efficiency for the several different types of machine-tool. It must, however, be borne in mind in considering these values that the efficiency of cutting of the cutter or tools is an important element in the machine volumetric efficiency, and that improvements in this efficiency can be effected either by improving the design of a machine in regard to both stability and speed and feed arrangements or by improving the design of the cutter or tool, or both.

TABLE XIV.

TABLE OF VOLUMETRIC EFFICIENCIES.

Type of Machine-tool.	Cubic Inches per Machine I.H.P. Hour.		Machine I.H.P. per Lb. of Metal per Minute.	
	Steel.	Cast Iron.	Steel.	Cast Iron.
Lathe . . .	60-100	100-180	2·0-3·5	1·2-2·0
Drilling Machine .	20- 50	50-120	4·0-10·0	1·8-4·0
Milling Machine .	25- 50	50-100	4·0- 8·0	2·0-4·0

This table shows that the lathe is the most efficient of machine-tools.

CHAPTER VII.

COMPARATIVE TOOL TESTING.

In these days of economic production it is desirable for works to use the tool steels which are most suitable for the different kinds of machine work done in them. Hence, the necessity for the comparative testing of tool steel from the point of view of the user.

Without entering deeply into this subject, it may be said that there are several methods of making comparative tool tests extant, though the conditions which exist in many works when this kind of work is being performed are by no means the best, and the results are not always as reliable as is generally imagined.

These several methods may be classified as follows:—

1. The constant-speed method;
2. The constant-duration method;
3. The increasing-speed or speed-increment method; and
4. The increasing-feed or feed-increment method.

The first is the one which is most commonly employed in works, the life of the tool or duration of the test being taken as the criterion of the cutting ability of the tool.

The second method (which was originally devised by Mr. F. W. Taylor) involves the use of a batch of similar tools up to the number of eight, the object of the test being to determine a cutting speed by the cut-and-try method which will give a tool-life or test-duration of twenty minutes.

The third method (which was devised and worked out by Dr. Ripper and the author at Sheffield University) is based upon the tensile-testing method of gradually increasing the load on a specimen until rupture occurs. In this case, the cutting speed takes the place of the load, and this is increased by uniform increments until the breakdown of the cutting edge or edges occurs. The volume of metal removed in the process is taken as the criterion value of the tool. In the initial stages of the development of this method, an initial cutting speed of 20 ft. per minute was adopted in conjunction with a speed increment of 5 ft. per minute made every three minutes. In the present form of the test, the starting speed is 30 ft. per minute and the increment of speed 1 ft. per minute made first at the end of the first half minute and then at the end of every succeeding minute. The application of this method involves the use of a variable speed electric motor, so as to give speed changes which are practically continuous.

In the fourth method (which is of German origin and later than the third method) the feed of the tool is gradually and regularly increased, the cutting speed being maintained constant. In this case also the volume of metal removed by the tool up to the point of breakdown is the best measure of the cutting capabilities of the tool, as this gives the best tool credit for working at the coarsest feeds.

In making these tests, certain precautions must be taken, the principal ones being as follows:—

1. All the tools should be of precisely the same shape, both in regard to plan and in regard to the various angles of the tools.

2. They should all be tested on the same test bar, and as far as possible on the same diameter, though a practical scheme where there are several tools to test is to test each tool on a number of diameters and average the results.

3. The extreme cutting points or edges of all the tools should be arranged at the same height with respect to the centres of the lathe; preferably they should be disposed in the horizontal plane which passes through the lathe axis.

4. Each tool should be disposed in relatively the same position to the test bar so that the operative tool angles are the same in every case.

CHAPTER VIII.

COMMERCIAL MACHINE-TOOL TESTING.

There does not appear to be any uniform practice in regard to this matter, though the object of many of these tests is the testing of the pulling or driving power of the machine. Thus, if a lathe, for instance, has sufficient pulling or driving power to take a specified cut at a specified cutting speed on a specified material and to do it for a specified period of time without undue heating of the bearings and slippage of the belt (where one is employed) it is passed. Or, again, a drilling machine is commercially tested for its penetrating power. If it will drill, say, holes 1 in. in diameter, at a given rate (such as 10 or 12 in. per minute) in cast iron or steel, it is passed.

Of course, much depends upon the quality of the steel in the tool or cutter which is used, in the tests, and it not infrequently happens that the modern machine-tool is superior to the cutting tool. In fact, it is the claim of at least one manufacturer of machine-tools that he cannot find a tool steel which does not break down in his lathes when they are being commercially tested, so severe are the conditions of the test, to the credit, of course, of the machine. Whether this is of general application or not is a douhtful point; but it is undoubtedly true that, since the introduction of high-speed tool steel, machine-tool manufacturers have put a large amount of metal into their machines so as to successfully cope with the increased forces which operate in cutting operations. The direct result of this is that machine-tools can be worked with heavier speed and feed combinations than suitably dimensioned tools can work with. The limit of a machine is, however, the limit of its weakest element, and it thus appears that at the present time this is the cutting tool. Probably it will be always so, since the tool is the element which is, generally, the most severely stressed under the influence of the cutting forces.

No reliable data concerning these tests are available since the conditions under which such tests are made are so very variable and, generally, are such that the test results are of little comparative value.

THE END

APPENDIX.

INSPECTOR'S REPORT OF LATHE TESTS.

Size of Lathe . . . centres.

. . . long.

1. Longitudinal level.
2. Transverse level.
3. Local bed inaccuracies.
4. Kind of fit between driving headstock and bed.
5. ,, ,, ,, ,, loose ,, ,, ,,
6. ,, ,, ,, ,, slide-rest saddle ,, ,,
7. ,, ,, ,, ,, saddle and cross slide of slide rest.
8. ,, ,, ,, ,, clamp bed and top slide of slide rest.
9. Driving headstock spindle: high or low at front.
10. ,, ,, ,, transverse alignment.
11. ,, ,, centre: runs true or not.
12. Centre socket in driving spindle: runs true or not.
13. Driving spindle nose; runs true or not.
14. Boring test: large at outer or inner end.
15. Facing test: plane, convex, or concave.
16. Loose headstock mandrel: high or low at front.
17. ,, ,, ,, transverse alignment.
18. ,, ,, centre socket: true or not.
19. Alignment of centres: longitudinally.
20. ,, ,, ,, transversely.
21. Running of back gearing.
22. ,, ,, driving headstock reverse gearing.
23. ,, ,, gearing in feed apron.
24. ,, ,, ,, ,, change feed gear box.
25. Fit between rack and pinion.
26. Lost motion in saddle gearing.
27. Error in lead screw in length of . . . feet.
28. Finish of scraped surfaces.
29. Back lash in cross slide screw.
30. ,, ,, in top slide screw.

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