THESIS

DESIGN AND CONSTRUCTION OF A REVERSE TORSION TESTING MACHINE

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Design and Construction of a Reverse Torsion

Testing Machine

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THESIS

---- Preface.----

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Historical. The following extract is taken from the eleventh edition of Merriman's "Mechanics of Materials".

"The first experiments on the strength of materials were made on the rupture of beams of timber. A picture in Galileo's Discorsi (Leiden, 1638), shows a cantilever beam projecting from a wall and loaded with a weight at the free end, and it was probably from experiments of this kind that Galileo was led to the conclusion that the strength of rectangular beams varies as the squares of their depths. During the eighteenth century experiments were made in France on timber in flexure and tension, only questions of ultimate strength being considered, while the elastic limit was unrecognized. Hooke's experiments on springs, from which he deduced the law of proportionality between stress and elongation had, indeed, been announced in 1678, but it was not until 1798 that Girard made the first series of experiments on the elastic properties of beams. Nearly a quarter of a century later Barlow, Tredgold, and Hodgkinson experimented on timber and cast iron, both in the form of beams and columns; their methods although now seemingly crude and defective, are deserving of praise as the first of real practical value.

"In 1849 was published in London the 'Report of the Commission on the application of Iron to Railway Structures',

which may be regarded as the landmark of the beginning of the modern methods of testing. The immediate result of this report was the decision of the English board of trade that the factor of safety for cast iron should be twice as great for rolling loads as for steady ones, while throughout Europe and the United States it aroused marked impetus in the subject of testing materials.

"The first testing machines in the United States were those built by Wade and Rodman in the period of 1850 to 1860 for testing gun metal. About this time the rapid introduction of iron bridges led to experiments by Plympton and Roebling. Prior to 1865 apparatus was built for his special work by each experimenter, but in that year Fairbanks put upon the market the first testing machines for commercial work. A little later the machines of Olsen and Riehle for tensile, compressive, and flexural tests were introduced, and have since been widely used. The machine devised by Emery, soon after 1875, is a very precise apparatus which is used in large laboratories. Large machines for testing eye bars have been built by bridge companies, and numerous testing laboratories now contain apparatus for every kind of work. *********

"Tensile tests are the most common, *****. Nearly all tensile machines may be also used for compressive tests, and also for the flexural testing of short beams. ****

"Commercial tests of materials are rarely made under shearing and torsional stresses. Thurston in 1870 devised a torsion

machine for small specimens, and the torsion machines of Olsen and Riehle, which are found in the laboratories of most engineering colleges, prove very serviceable for illustrating the phenomena of twisting. Impact machines have been built for special investigations, but the only one on the market is that of Keep, which is designed for test bars of cast iron. Fatigue or endurance tests, which subject the specimens to alternating stresses for long periods of time, are made on special machines."

It is for the investigation of these alternating or fatigue tests that the machine considered in this theses is designed. Since the publication of Merriman's text in 1914, or rather since the preparation of the material for this publication, more has been done in these directions than the extremely meager degree of progress cited above. It is becoming more and more recognized that, while the usual tensile and compressive tests are undoubtedly of great value in showing the behavior of materials under steadily applied loads, still, this does not establish an infallible criterion for the determination of relative merits. More should be known of the behavior of materials when subjected to intermittent, repeated, or reverse stresses, especially when these stresses are similar to those occurring in materials under actual working conditions. Since in many phases of practice stresses of this nature are the rule rather than the exception, it is very

desirable that data be secured bearing on the effect of such stresses on the materials used in construction.

Following are extracts taken from the American Machinist, and written by G.B.Upton, Assistant professor of experimental engineering, Cornell University, and G.W.Lewis, Assistant professor of experimental engineering, Swarthmore college.

"The Fatigue Failure of Metals"-

"Examination under the microscope of the internal structure of any of the engineering metals shows that the metal is an aggregate of an immense number of minute crystals. The separate crystals in a steel, for example, Vary from a maximum diameter of 0.01 in. or 0.02 in. down to around 0.0001 in. and even smaller, sometimes being too small for even the highest powered microscope to make them visible. The external shapes of these crystals are irregular, being determined simply by their interference with their neighbors. At their bounding surfaces the crystals are firmly and strongly interlocked with each other; not between them along the boundaries, merely pulling the crystals apart.

"The break of the metal, then whether it occurs at the end of a single loading, or at the end of many repeated loadings is a break through the crystals. The study of the nature of the break of the metal resolves itself into the study of the way in which the separate crystals may be broken. *******

"The way in which a crystal breakes is determined by the

nature of its component particles and the pattern in which they are put together. ***** No other breaks than these two (tension and shear) are possible. There is one essential difference between them. The tension break is a complete break from its beginning; the parts are separated. The shear may make one layer slip over the next a little way, and then cease. The crystal is not yet broken, but merely permanently distorted. The particles along one side of the shear slip plane still have hold of the particles on the other side of this plane, though not the same particles as before. The crystalline structure would not be hurt in the least by such a slip, were it not that in that in slipping past each other the layers interfere slightly, and a little debris or dust of particles torn out is left between the layers that have slipped. This amorphous material, formed in the slip planes, is peculiarly related to the chemically identical material from which it has been torn. In the engineering metals the amorphous material is somewhat (sometimes even decidedly) harder and stronger than the crystalline. This results in a complicated reaction of the shear crystals to shear loading.

"The crystalline structure starts to slip or yield along some plane of weakness lying near to the direction of the maximum intensity of shear stress. As the yield progresses, the formation of amorphous material makes it increasingly difficult for the slip to continue. It is easier to start a new slip along an adjacent parallel plane. This in turn increases its resistance; a third slip starts, and so on, until the crystal has yielded to the stress by slipping an infinitesimal amount on each of a great number of slip planes.

"These little slips are all permanent, persisting after the removal of the load which caused them. We say the crystal shows plastic yield to loading, and comment on the hardening and strengthening of the material by cold working. *****

"We may now study the loading of the metal as a whole, remembering that it is an aggregate of a great number of crystals. In the cast metals the crystals have about the same dimensions in all directions through the piece, save that at or near the surfaces of the casting they are longer and perpendicular to the piece. Away from the surface there has been no tendency for the crystals to grow in one direction more than another, or to arrange the alignment of their internal patterns in one direction more than another.

"In the worked metals the rolling or forging has tended to make the crystals somewhat longer parallel to the length and surface of the piece, giving a semi- fibrous structure; but again there is little else but chance in the alignment of the crystal patterns with regard to the dimensions of the metal as a whole.

"Consider now the ordinary tension test, where the loading process progresses till the rupture occurs in a single application. All the crystals are pulled out simultaneously and fairly equally.

Each crystal so yields as to become much longer in the direction of the pull, and thinner, perpendicular to the pull. With a material already semi- fibrous this results in the familiar "fibrous" or "silky" fracture.

"In the cast metals the stretching rarely goes so far as to give the 'fibrous appearance to the fracture, for it has to overcome too much of a handicap in the initial arrangement of the crystal dimensions. ******'

"What are the laws governing the number of applications of a given kind of loading and stress intensity required to produce failure? Frankly, we do not know. No one has been able as yet to tell us the relation between the results of the conventional tension test and the fatigue tests. There is no accepted and standard method of fatigue testing. Wohler's tests, over a generation old, are not yet explained. In this field of experimental engineering we are but on the threshold of knowledge.

"Something of the probable form of the laws of fatigue failure we may perhaps derive from the study more in detail of the way in which repeated loading breaks down, one after another, the crystals which compose the metal. We may best study the steels, both because of their importance, and because they are the most complex in structure of engineering metals; hence the study will be of the most general form.***

"When the load applied to a piece of metal is very small, the internal condition of the metal approximates most closely

to equality of deformation from crystal to crystal. Whether the stresses are similarly equal depends upon the stiffness of the different crystals. In the steels the stiffnesses are nearly equal for all crystals at first, and we have fair equality of stress as well as of deformation from crystal to crystal. In metals other than steels the stiffnesses frequently vary from crystal to crystal, so that with small loads the condition is one of equality of deformation with stresses varying from crystal to crystal in proportion to their stiffness.

"As the load increases, these conditions change radically. If no crystal broke before the rest and ceased to be useful in carrying the load we should approach at high loads a condition of equality of stress from crystal to crystal, with variable deformation. For at some stage of the loading the elastic limit of the ductile type of crystals in the structure is past, and they begin to give plastic yield, and subsequently must yield largely to increase their stresses. The soft ductile crystals cannot transmit to the hard crystals bedded among them any greater stresses than the soft crystals carry.

"Certain other factors must not be forgotten. To some extent the shape and size of the crystals and the way they come together at their boundaries influences the stress intensities that they can transmit to each other. The smaller the separate crystals in average size the more uniform are the stresses from crystal to crystal, and therefore the stronger the material in the engineering sense of the strength of the piece as a whole. It is perhaps true that surface tension effects make a very small crystal intrinsically stronger than a large crystal of the same material.

"There are residual stresses, from one crystal to another, or between different parts of the piece, coming over from the previous history of the piece. Such residual stresses come from cold working, heat treatment, etc. Cold working tends to elongate the outer parts of the cross- section of the bar more than the inner parts: consequently it leaves the outer parts with compression stresses, and the inner parts with tension stresses. *******

"There is a time factor between shear stress and shear deformation, or slip. This time factor depends partly on the fact that the slip itself is not quite instantaneous, and still more on the fact that after slip occurs in one crystal the distribution of stresses from that crystal to others is altered, which may cause slip somewhere else, and so on through an appreciable time interval of internal adjustment under load.

"Load suddenly applied, as by a blow, tends to equality of deformation from crystal to crystal, with stresses determined by the common deformation and the individual stiffness. Load slowly applied tends to equality of stresses, with adjust-

ment of deformations to an equilibrium depending on the load, but rarely attained with light loads.

"The above paragraphs on the distribution of stress and deformation from crystal to crystal, make it apparent that we do not mean what we say when we make the statement that the stress intensity is so many pounds per square inch. What we mean is that, if the material, instead of being a conglomerate of crystals, were a perfectly homogeneous substance; if our assumptions as to the elastic actions following a straight line law were true; if the load were applied as we assume it is, etc., the stress intensity would be what we say it is.

"When we calculate from our formulas for loading the result is at best only an average value of stress or deformation for the crystals of a given region of the piece, or perhaps for the whole section; at its worst a merely nominal quantity that is convenient to our calculations for design, because on reversing the calculation, perhaps with changed dimensions, we may come out again somewhere near to the external loading the piece will carry. Even in a case so simple as that of pure tension loading we calculate for engineering purposes "load per square inch of original area" beyond the yield point, forgetting that the original area ceased to exist when the yield point was passed.

"As for our "moduli of rupture" of the torsion and transverse loadings, we have recognized in the names used that

they are only nominal stress values. Yet how many engineers know just how far these "moduli of rupture" may be from the average real stress in the outer fiber at break, or the difference between 'maximum stress per square inch of original area' and the real value of (breaking load/area at break) of the same test?

"Keeping in mind the variation of stress and deformation from one crystal to another through the piece; the residual internal stresses; the effect of shape and size of the crystals; the variable orientation of crystal patterns, even if the crystals are all of one kind; and the inadequacy of the assumptions on which we calculate stresses, can we say that when the nominal stress from the external loading is 1000 # per sq. in., there is not at least one crystal in a million which has been pushed beyond 50000 # per sq. in., and perhaps ten crystals more in the million between 1000 and 50000, and hundreds between 1000 and 10000, and thousands that do not reach 1000; again others that have still, from residual internal stresses, even large stresses of the opposite sign from that which we impute? Remember that there are from 1,000,000 to more than 1,000,000,000 crystals per cubic inch of steel.

"We believe that the more careful the study given to this question, the more sure will be the conviction that a piece of metal, as a whole, has no "primitive elastic limit", because any loading, even no loading, finds some crystals

carrying high stresses and deformations; that the law of stress and deformation from crystal to crystal throughout the material is one of those "probability" laws rendered so familiar to mathematicians by the kinetic theory of gases.

"The exact form of these distribution laws for stress and deformation will vary with the material and with the history of the piece - whether it has been cold worked, heat treated, etc. It would follow from this that any stress, however small, if repeatedly applied, would finally cause failure. But it will be evident that from a study of the tests that have been made, a comfortably large stress from external loading may be applied so many times that the piece, and the machine of which it is a part, will surely wear out long before fatigue failure need be anticipated.

"The number of applications of a load to produce failure depends upon; (a), the number of shear slips caused at each application; (b), the number of crystals affected by these slips; (c) the ratio of the number of these crystals to the total number; (d) the percentage of crystals that must be put out of commission before the break occurs.

The factors (a) and (b) depend on the stress intensity, or deformation applied, the nature and size of the crystals, and the past history of the piece; (c) depends on the nature of the material; (d) depends on the kind of loading, being probably least for torsion, slightly larger for transverse,

and still larger for tensile loads.

"For any given material and method of loading, from the fact that we are dealing with probability functions, there is as the simplest possibility, a chance that the relation between stress intensity and number of applications required to break the so called endurance curve, is a logarithmic one- such a relation as that stress versus log (number) is a straight line, or log (stress) versus log (number) is a straight line. The real function between number and stress is probably a complicated exponential relation."

In a following article by the same authors there is a discussion of the design and operation of a reverse flexure fatigue machine. So far as the authors of this theses have been able to ascertain, this machine and the Landgraf-turner machine, handled by Queen & Company, Philadelphia, are the only ones of the kind that have been developed in a commercial way?

Below is a synopsis of the article mentioned, with extracts from it, together with editorial comment:-

"The design of a new machine for the fatigue tests of materials. From its use a number of important results have been obtained. A piece of ductile material broken with from one to ten applications of load has a fracture of the cup and cone type as in tensile tests. With applicatios greater than one hundred the breaks are square.

"The logarithmic curve giving the relation between maximum stress and number of cycles is found to be nearly a straight line. Further for similar materials the curves are found to be parallel. This foreshadows a radical change in fatigue testing, for the "figure of merit" can be obtained with a few cycles, say from 500 to 10,000. This would permit of the completion of a test in from five minutes to one hour, compared to half a day or several days as at present."

"For practical testing the same kind of materials for their comparative values, the parallelism of the curves opens the way to some radical changes from present methods. *****

"The testing time may be immensely shortened. On the Upton-Lewis machine runs ar 1, - 3/4, - and 1/2 inch crank radius would give from 500 to 10,000 cycles, and would be sufficient to place the material. A run of 500 cycles would take, complete, about five minutes; at 10,000 cycles, about one hour. This is a very pleasant contrast to the half day or days hitherto used for fatigue testing. ******* The real use of this straight line approximation to the curve, however, is that it enables the finding of a figure of merit for the material by which it may be compared with other materials of similar character. The equation of the straight line is:-

$$\log n = K - m \cdot \log p,$$

where n is the number of cycles to break, p, the stress, and k and m are constants. With p in pounds per square inch, m has the value $8.5 \pm$ for steels and wrought irons.

Assuming this, K becomes the figure of merit for the steel.

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The practical testing schedule for the test of steels on the Upton-Lewis machine would consist of runs of two or more pieces each, with 1, - 3/4, - and 1/2 inch crank throws. From the section dimensions of the piece, the value of pencil travel (indicating the stress), and the constants of the machine, the values of the nominal stress intensity for each run would then be found. Substituting in

$$\log n = K - 8.5 \log p$$

will give half a dozen or more values of K; the average of these is the figure of merit of the steel. The time for these tests would be about half a day. Different steels would be compared by comparing their figures of merit. This procedure breaks down when the materials to be compared are unlike; then the only way is to determine the endurance curve from $n = 100$ to 1000 up to $n = 10,000,000$ or more; plot the curves, and compare the plots.

Editorial comment on the Upton- Lewis articles and testing machine. From American Machinist.

****** Testing by applying repeated alternating stresses has not yet gained for itself a place in shop laboratories for two major reasons: There is some question as to its value in pointing out the property of endurance under load and vibration; and more important than this, a machine which

will permit of accomplishing such tests quickly has not been available.

"The types of apparatus used generally have been such that a long period of time was necessary to test a single specimen. This period may be from half a day to several days. This condition in itself has been sufficient to make this test practically unusable in commercial testing. The article in this issue is of importance in showing that a machine from which results can be obtained in a short time, say in from five minutes to half an hour, depending on the number of stress cycles required, has been made commercially practicable.

"Furthermore a connection between the tensile stress and the number of cycles to break is, we believe, for the first time pointed out. The stress- cycle curves, for metals with different tensile strengths, apparently belong to the same family, but occupy different positions with respect to the axis, these relative positions indicating their tensile strengths.

"Another worthy feature is the corroborative evidence that the older theory of crystallization under vibration is erroneous. This theory has been attacked many times, and is generally discredited. But in spite of that condition it is well to bring forward such evidence as the authors discovered, showing that the appearance of the fracture is dependant on the original crystallized condition of the material and the

manner of breaking. The tests illustrated from wrought iron specimens are particularly interesting in this particular; as this metal is usually looked upon as the best example of a fibrous metal. The tests show that under proper conditions of breaking, a fracture could be obtained from wrought iron equal in fineness to that of the highest grade, most carefully treated tool steel."

An article in the "Proceedings of the American Society for Testing Materials", volume fourteen, 1914, discusses a new vibratory testing machine designed to perform "oscillating and rotative vibratory tests," as well as tension tests. This is a stay bolt testing machine, and performs tests in flexure. No torsional stresses are developed in the specimen.

The material outlined above, with catalogs of the standard types of testing machines, constitutes all that we find available bearing on the subject of this work. On account of the scantiness of the supply of information along this line, the foregoing extracts and comments have beem presented somewhat in detail, in order to give something of an idea of the field for the application of this kind of testing of materials, particularly metals; to show the extent of what has been done in this direction; and to make clear that this work is one in which the investigators must proceed almost entirely on their own resources.

There is undoubtedly other material, but the idea in presenting this quite completely from are joint of view, rather than as a comparison of many theories is not to subscribe entirely to just the statements outlined, but were to show that the subject of fatigue testing is one in which little has been done, and that much of value may be accomplished along this line.

Nor do we claim for this study anything more than a step, tho truly hoping it to be a step forward, in the almost untouched field of fatigue testing. beginnings must be made slowly, especially along the lines of research, and this field is so broad and varied that only one small phase of it can be logically.studied at a time. The design, then of this machine, leading eventually to its construction, and operation in the testing laboratories of the Michigan Agricultural College, is submitted as the authors' mite toward the gaining of a more complete and detailed understanding of the phenomena of the fatigue failure of metals.

Design of the machine. The machine, which we have designated a "Reverse Torsion Testing Machine" is intended to investigate torsional fatigue failure of metals in form of test bars by subjecting them to repeated or reversed torsional stresses, these stresses not to exceed the nominal elastic limit of the material. This will be accomplished by having the specimen held at each end in chucks, one of which is to be provided with spring devices for determining the stresses, and the other having suitable mechanical connection with the driving mechanism so that it may be given an oscillatory

rotating motion with respect to the axis of the specimen.

The machine will embody the following general features:-

(1) A frame or bed of strong and rigid construction, suitable for maintaining the other parts in their proper relation to each other.

(2) Two chucks mounted in suitable bearings fitted in standards adapted to be securely fixed to the bed of the machine, and having for their object the gripping of the ends of the specimen as referred to in the previous paragraph. One of standards carrying its chuck will be permanently fixed to the bed of the machine, near one end, and will correspond to the head stock of a lathe. The other will be movable along the bed of the machine, and will correspond to the tail stock of a lathe. This movement is designed to accomodate specimens of varying length.

(3) Proper driving mechanism for operating the head chuck as described above, consisting of a crank of variable length (in order to vary the amplitude of the chuck motion, or angle of torsion), connected by a link to a rocker arm rigidly secured to the chuck spindle; the whole to be driven thru clutch and shafting by a direct current motor of the shunt type, adapted for operation on a 220- volt circuit.

(4) A spring device acting thru the tail chuck for determining the stress to which the specimen is subjected, together with a pencil recording attachment and counter for obtaining a graphical record of the test.

(5) All moving joints to be provided with means for taking up wear, and for adjustment.

All members of this machine will be designed for stiffness and rigidity, rather than strength so far as safe working stresses are concerned. The reasons for this are self-evident, so no explanation is necessary. Suffice it to say that we wish to determine the distortion of the test specimen, which at best is only a distortion relative to that of the machine itself, hence the necessity to keep the machine as a whole as rigid as possible.

In order to maintain a fairly uniform motion and load on the driving mechanism it will probably be necessary to mount a rather heavy fly wheel on the shaft of the driving crank.

The first step in the design of the machine is to determine the greatest angle of torsion that will ever be required, and to kinematically analyze the mechanism that will give this required motion. From the expression:-

$\phi = 57.3 \, P \, p \, l \, / \, J \, F$,where ϕ is the angle of torsion in degrees; P p. the moment giving this distortion; J, the polar moment of inertia of a section of the specimen at right angles to the axis of twist; and F, the shearing modulus of elasticity of the material of which the specimen is composed? It appears that the angle of twist for any given material increases with the length of the specimen, and decreases with its diameter. Then the greatest angle required will be in the case of the longest and slenderest specimen, as it is stresses up to its elastic limit. We assume that this specimem will be a cylinder of strong steel, one- half inch in diameter, and four feet long. The diameter of the largest specimen to be

tested in this machine will be one and one- half inches.

From Merriman, page 225, we have the relations:-

$M = P\,p = S\,J\,/\,c$, where S is the stress in the outer fibers of the material, and c is the distance from the neutral axis to that fiber, the other notation being the same as above, and

$$F = 57.3\ P\,p\,l\,/J\,\phi$$

From these two equations, we get :-

$$\phi = 57.3\ P\,p\,l/\,J\,F\ .$$

F, for steel is about 12,000,000 inch pounds per square inch. The elastic limit for strong steel is about 60,000 inch pounds per square inch. Using these values, then, for a specimen of this kind, 1/2" in diameter and 48" long, we get :-

$$\phi = 57.3\ (S\,J\,/c)\,l\,/\,J\,F = 57.3\ S\,l/c\,F$$

$$\phi = 57.3 \times 60{,}000 \times 48\ /(.25\ \times 12{,}000{,}000)$$

$\phi = 55^{\circ}$, the angle of torsion required to stress the material in the above specimen to its elastic limit.

Determination of maximum driving crank radius. For this purpose a graphical method is simplest and most convenient, and will therefore be used. In Plate 1, O A represents the lever arm actuating the head chuck, and is to be one foot in length. This length is arbitrarily chosen, the considerations being that too short a radius means heavy loads on the driving members, and too long a one involving higher velocities and greater wear on the moving parts, and more difficulty in balancing them.

Angle A O B is 55° . Point D is midway between C and E, and line D B' is perpendicular to line C O. B B' = A A' = two feet (tho its length is immaterial in the solution), A' and B' are located on line A B' by taking a 4" actual size radius (representing two feet to scale), and describing arcs intersecting line A B' , using points A and B as centers, respectively. Point O' then, midway between A' and B', is the center of the required driving crank circle. The radius O' A' by measurement is found to be 1.86" or representing an actual size of 0.465 feet.

Provision will be made for varying the crank radius from zero up to six inches. This will mean a maximum angle of torsion of sixty degrees, which will probably be sufficient for any test ever desired to be made. In order to stress the 1/2 inch by 48" inch specimen to its nominal elastic limit, on both sides of the zero point, an angle of 110° would be necessary, but it is not likely that the machine will ever be called on for such a test. In most tests a small angle of torsion, say from 10° to 30°, would probably be used, for it would not be advisable to stress the specimen much beyond half its elastic limit, for the idea of these tests is to approximate as closely as possible the conditions of actual practice. And furthermore, the test specimens will usually be much shorter than 48". Ten inches would be a convenient length to handle.

<u>Two kinds of tests</u>. As before stated, the tests that may be performed on this machine will be of two kinds: one involving repeated, and the other, reversed, torsional stresses.

They would both be performed in exactly the same manner, and with no change in the mechanism; the difference being merely in the position of the zero point with respect to the angle of torsion ; whether the angle falls entirely on one side of the zero, or partly on each side. Referring to Plate 1, if the rocker arm were in the position shown by line B O, one of the extreme positions it occupies during the cycle, when the specimen is clamped in the chucks, then the test would be one of repeated stresses only. The cycle would be: an increase in stress from zero to the maximum value, and a decrease to zero again, as the rocker arm actuating the head chuck moves from the position B O to position A O and back to B O again.

But if the rocker arm were in the position C O when the specimen is clamped in the chucks, then the cycle would be one of alternating, or reversed, stresses as follows: the stress increasing from zero to maximum value, then decreasing to zero, increasing to maximum in the opposite direction, and finally decreasing to zero again, as the rocker arm moves from the initial position C O and thru its cycle, back to C O again.

These two kinds of tests are, apparently, quite different, and due regard to the conditions mentioned above would of necessity have to be observed in the operation of the machine.

Determination of load on the driving members and power required to operate the machine.

For the kinematic outline of the driving mechanism see Plate I. It is desired to know the load on the link A A' which actuates the rocker arm O A.

First consider the 1/2" in diameter by 48" long specimen. We have, using the same notation as before,-

$$\phi = 57.3 \; P \, p \, l \; / \; (J \, F)$$

$$55^{\circ} = 57.3 \; P \times 12 \times 48/(.00614 \times 12{,}000{,}000)$$

$$P = 55 \times .00614 \times 12{,}000{,}000/(57.3 \times 12 \times 48)$$

$$P = 123 \text{ \#.}$$

Following a similar calculation for a specimen 1 1/2" in diameter and 48" long, we get a value of 3,330 #. This is a much greater value than computed above in the case of the smaller specimen, showing that the load, and consequently, the power required, will increase with the diameter of the test specimen - a conclusion that appears natural and reasonable in the light of other tests.

Taking the most extreme case, then, we would have the 1 1/2" diameter by 48" long specimen stresses to its elastic limit in one direction, returned to zero, stresses an equal amount in the opposite direction, and again returned to zero during one cycle. The distance covered by the force P would be twice as great as that allowed, but the average force acting would be only half of the maximum value.

Hence, the energy expended under the heaviest load would be 1100 foot pounds per stroke. This does not take into consideration the energy that would be returned to the driving members during the decrease of stress from its maximum to zero in the specimen, but for the sake of simplicity, and the uncertainty of the values

of these impulses, especially after the test specimen begins to be affected by fatigue, no attempt is made in this preliminary calculation to allow for the effects of this factor in computing the power required. On the basis of the assumption that power is supplied only to deform the specimen, and that it returns of itself to its initial condition when the stress is removed, and taking the R.P.M. as 500, we get:-

$$500 \times 1100/33000 = 16.6 \text{ H.P.}$$

But we know that there will be considerable power restored to the driving members, due to the elasticity of the test specimens. This is particularly true in the case of the larger ones before they have been weakened by fatigue, - the exact instance in which the greatest amount of power is required. As the specimen is weakened, the energy restored is lessened, but the total required is also reduced, so it seems reasonable to assume that the power required to operate the machine would come well below the value computed above. Ten horse power will probably be all that is necessary, especially if a heavy fly wheel is used.

Design of the Fly Wheel.

The conditions assumed are that the largest specimen is being tested, and no account is taken of the return part of the stroke when the stress is being removed. The force applied at the end of the rocker arm is a variable between values of zero and 3300 pounds. The excess energy over the mean, then, that must be supplied by the fly wheel will be represented by the expression:-

$$(3300/2)\ (1/2)(1/2) = 412.5 \text{ foot pounds.}$$

The mean speed of the machine has been taken as 500 R.P.M.

The upper limit of rotative speed may be assumed at 510, and the lower limit at 490 R.P.M. These values are equivalent to 8.50 and 8.17 R.P.S., respectively. The upper limit of speed will occur at the beginning of the distorting stroke, and the lower limit at the end of this stroke, so the decrease in speed of the fly wheel during this half of the stroke must supply the excess energy required.

In general, the kinetic energy of a moving body is given ny the expression:- $1/2\ M\ v^2$. The energy lost by reduction of velocity is equal to $1/2M\ (\ v_1^2 - v_2^2)$

For the fly wheel, then, -

$$412.5 = (W/64.4)\overline{(16\ pi/12 \times 8.5)^2 - (16\ pi/12 \times 8.17)^2}$$

$$412.5 = (W/64.4)(16\ pi/12)^2\ (8.5^2 - 8.17)^2$$

$$412.5 = (W/64.4)(17.5)(5.6)$$

$$W = (412.5)\ (64.4)/(17.5 \times 5.6) = 271\ \#$$

The driving crank is to be mounted in this wheel, of which the maximum diameter is 18". In the preceding calculations the radius of gyration has been assumed to be 8", which is not far from the correct value. Neglecting the effect of the weight not in the rim of the wheel, we find that the rim dimensions to fulfill the requirements are 5" face by 3 3/4" thick.

The variation of the crank radius is to be accomplished by fixing the crank in a block arranged to slide in a T-slot radial with respect to the wheel.

Drawings of the fly wheel are shown in Plate V.

Method of holding the test specimen.

The prime essential to be sought after in the means for securing the specimen is absolute rigidity. It is also very desirable make possible the use of a simple specimen, the ideal being merely a cylindrical rod. A careful consideration of various methods has led to the idea that this can not well be accomplished, so the specimen to be used will be cylindrical, with enlarged ends. This enlargement need not be very considerable, however. It would seem that an increase in diameter over that of the body of the specimen of one-fourth of an inch would be ample. These enlarged ends will be milled with four V-slots such that they will furnish a hold for the jaws of the chucks gripping the specimen.

The chucks to be used are of the universal type, with four jaws, and designed to be bolted on the face plates supporting them. These chucks are to be fixed in this manner to the spindles shown in Plate VIII. They are to be purchased, and the ones decided on as suitable for the purpose are listed in current circulars of the Skinner Chuck Company, of New Britian, Conn., as #406-C 4-jaw, universal type.

Method of lubricating moving parts.

Where practicable, sight feed drip cups will be used. For the moving bearings, such as at the ends of the rocker arms and driving crank, spring grease cups will be required for satisfactory lubrication. Since the machine is to operate continuously for periods of perhaps several hours, it is important that the matter of proper lubrication be given careful attention.

Roller bearings for the tail stock.

In order to reduce the friction error in the stress determining and indicating mechanism, it is deemed advisable to use roller bearings in the tail stock for the support of the main spindle. These bearings are to be purchased; the ones selected as suitable are Hyatt bearings of the heavy duty type, requiring a hardened spindle, since they have no inner race, short series. They are as advertised in current circulars under this head for a 3 1/4" shaft.

Design of Springs:-

Reference to Kimball & Barr, page 130. The springs are to be in tension only.

The maximum load on the end of the tail rocker arm is 3300 #. $3300 \times 12 = 39600"\#$.

It is desired to have 1" of deflection represent 20,000" #.

Then $\frac{39600}{20000} = 1.98"$, the deflection for 3300 #.

We have, $P = \frac{\pi p d^3}{16 r}$, where P is the load applied, p is allowable unit stress, 60,000#; d is the wire diameter; and r is the radius of coil to center of wire.

Then $3300 = \frac{\pi \times 60000 \times .75^3}{16 r}$, assuming that $d = \frac{3}{4}"$.

From this, $r = 1.5"$

We have, $n = \frac{S d^4 E^s}{64 P r^3}$, where S is the deflection; n, the number of coils; and E^s, the torsional modulus of elasticity 11,000,000#.

Then $n = \frac{1.98 \times .75^4 \times 11,000,000}{64 \times 3300 \times 1.5^3} = 10$

In assembling, stretch springs 3" so that they will always be in tension.

They are to be made with 14 coils in order to provide for end fastening.

References.

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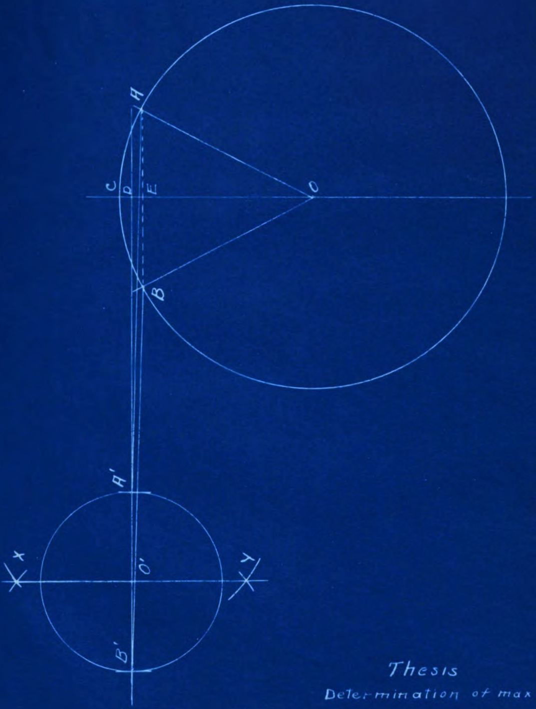
Merriman's "Mechanics of Materials", eleventh edition,- pages 470, 355, 225.

Church's "Mechanics",- Revised ediation of 1913,- page 233.

American Machinist, Volume 37,- pages 634, 678, 705.

Proceedings of the American Society for Testing Materials, Volume fourteen,- page 548.

Current catalogs of testing machines of the usual type.



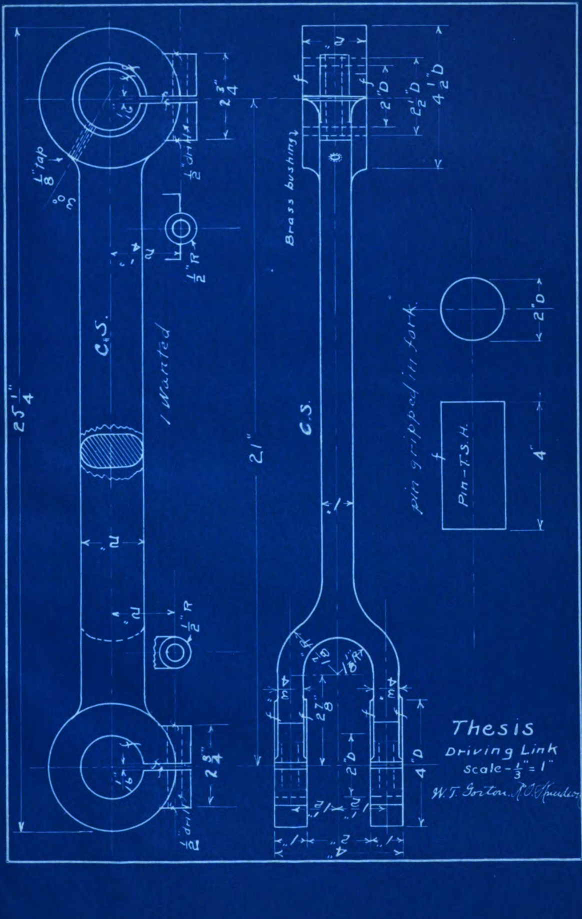
Thesis

Determination of max driving crank radius

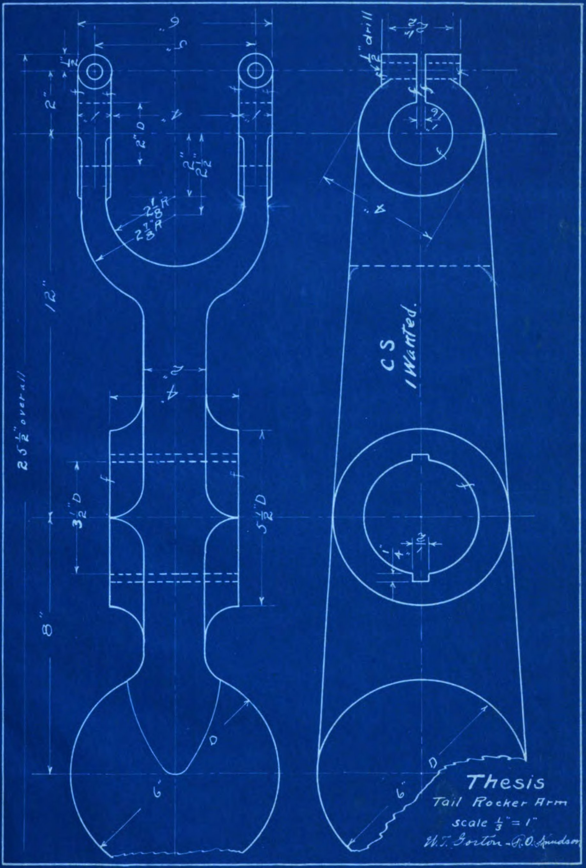
scale 1" = 6"

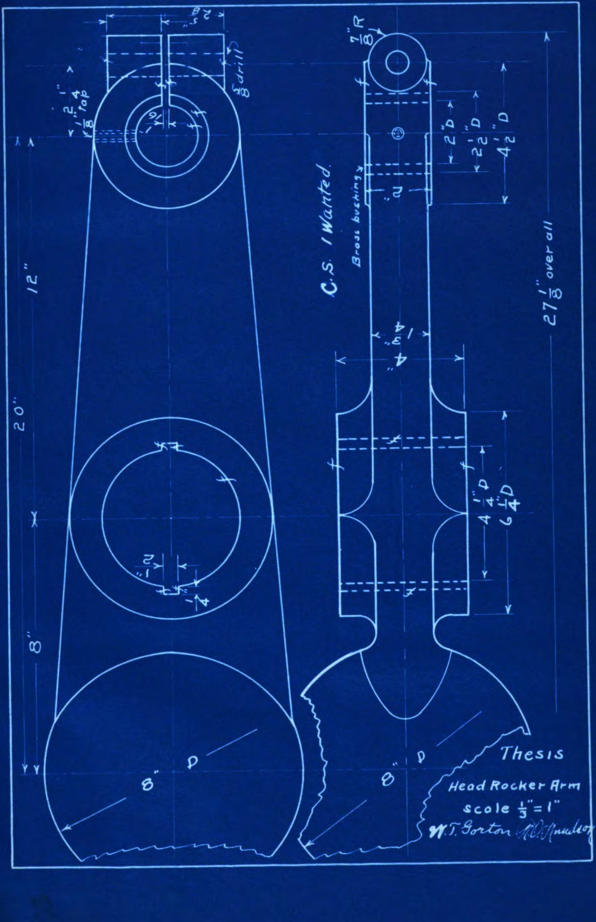
W. T. Gorton R. O. Knudson

II

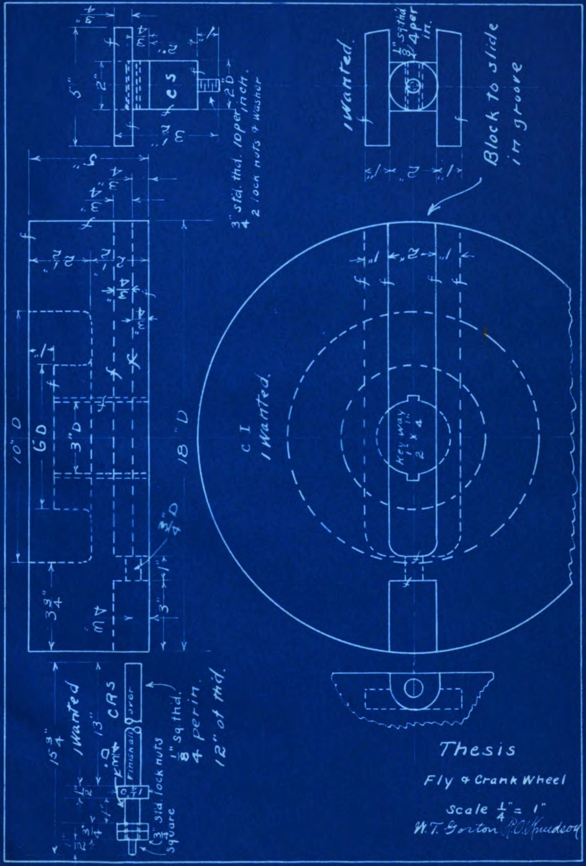


25 1/4"
C.S.
1 Wanted
1/8" Tap
1/2" drill
2 3/4"
1/2" R
2"
Brass bushing
21"
C.S.
1"
2" D
2 1/2" D
4 1/2" D
pin gripped in fork
Pin - T.S.H.
4"
2" D
4" D
2 7/8"
1/8" R
Thesis
Driving Link
Scale - 1/3" = 1"
W. T. Gorton

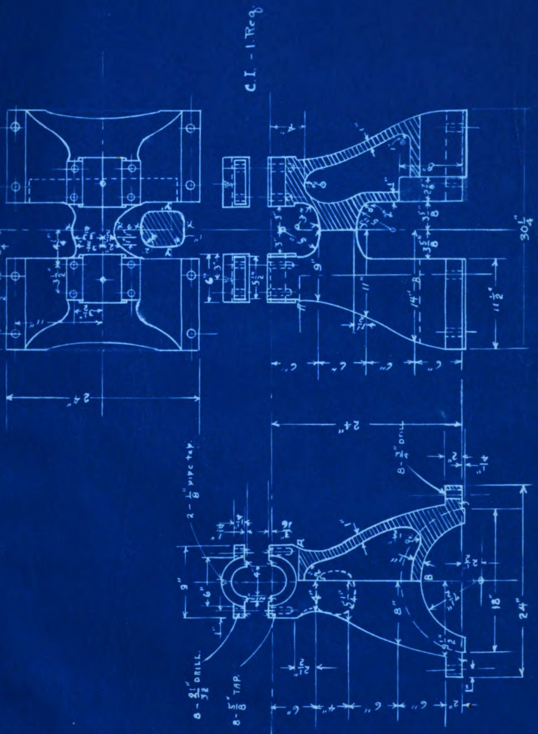

25 1/2" overall
12"
8"
2"
2" D
3 1/2" D
5 1/2" D
1/2" drill
CS
1 Wanted.
Thesis
Tail Rocker Arm
scale 1/3" = 1"


C.S. 1 Wanted
Brass bushing
27 1/8" over all
20"
12"
8"
8" D
Thesis
Head Rocker Arm
scale 1/3" = 1"
W. T. Gorton

V



1 Wanted.
Block to slide in groove
C I
1 Wanted.
Key way 1/2 x 1/4
10" D
6" D
3" D
18" D
3/4" D
3/4" std. thd. 10 per inch
2 lock nuts & washer
C S
2" D
1 Wanted
CRS
Finish all over
1/8" sq thd.
4 per in.
12" of thd.
3/4" Std. lock nuts
square
15 3/4"
13"
Thesis
Fly & Crank Wheel
Scale 1/4" = 1"
W. T. Gorton R. O. Knudson

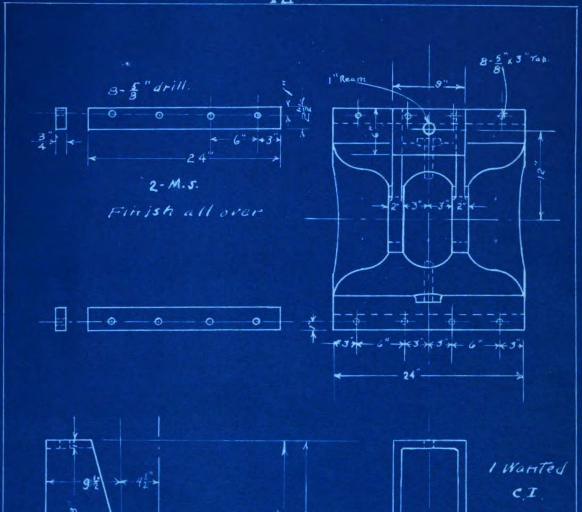


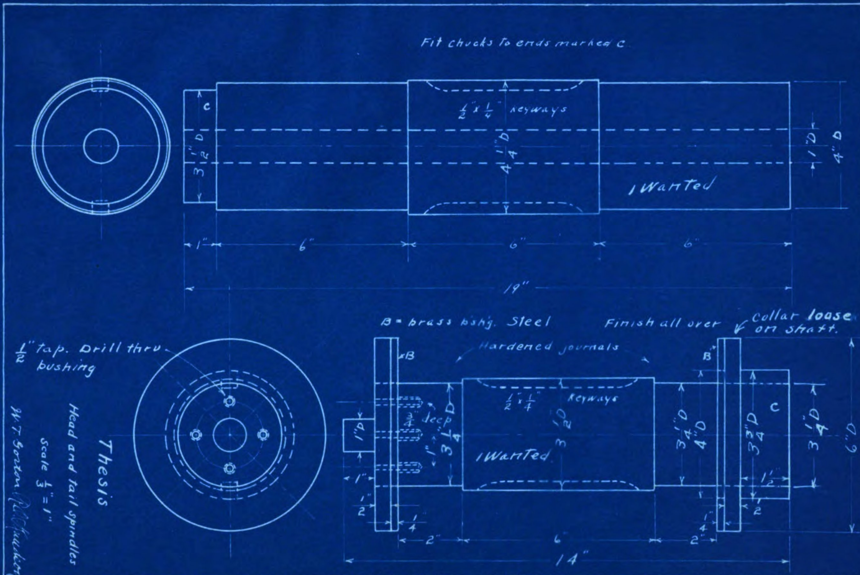
Thesis

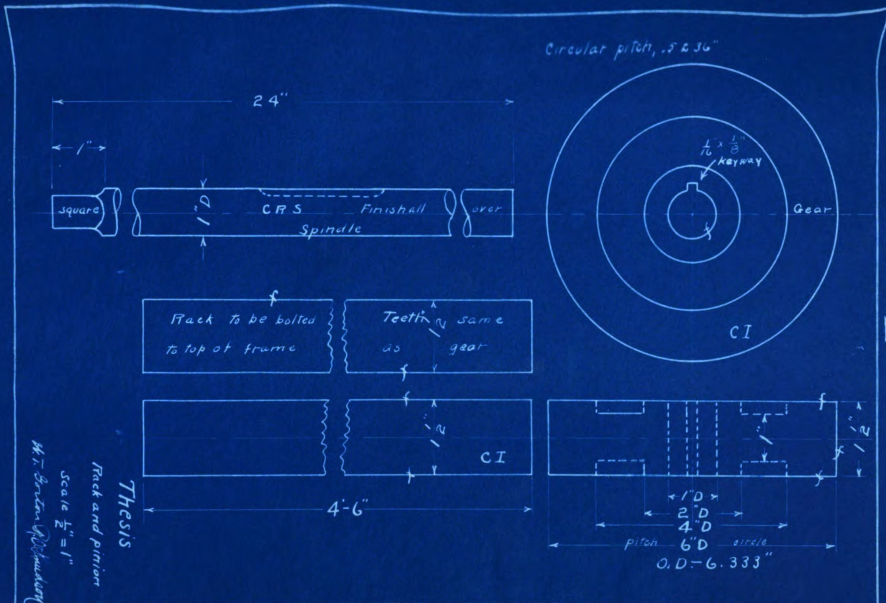
Head Stock

scale - 1/12" = 1"

W. T. Gorton R. O. Knudson

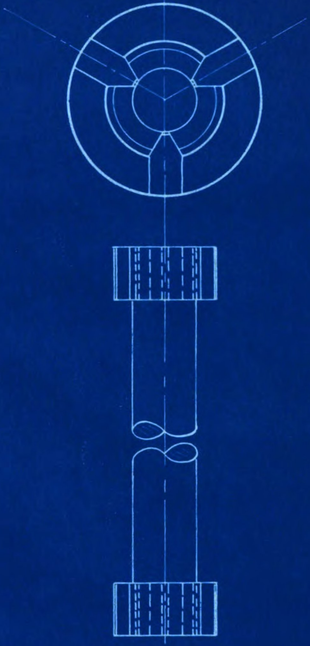

8-$\frac{5}{8}$" drill.
2-M.S.
Finish all over
1" Ream
8-$\frac{5}{8}$" x 3" Tap.
1" Ream.
1 Wanted
C.I.
Thesis.
Tail Stock.
Scale - $\frac{1}{12}$" = 1".
W. T. Gorton. R. O. Knudson


Fit chucks to ends marked C
C
$3\frac{1}{2}$"D
$\frac{1}{2}$" x $\frac{1}{4}$" keyways
$4\frac{1}{4}$"D
1"D
4"D
1 Wanted
1"
6"
6"
6"
19"
B = brass bshg. Steel
Finish all over
Collar loose on shaft.
Hardened journals
$\frac{1}{2}$" tap. Drill thru bushing
B
B
1"D
$\frac{3}{4}$" deep
1"
$3\frac{1}{4}$"D
$\frac{1}{2}$" x $\frac{1}{4}$" Keyways
$3\frac{1}{2}$"D
1 Wanted
$3\frac{1}{4}$"D
4"D
$3\frac{3}{4}$"D
C
$3\frac{1}{4}$"D
6"D
$1\frac{1}{2}$"
$\frac{1}{2}$"
$\frac{1}{4}$"
2"
6"
2"
14"
Thesis
Head and tail spindles
Scale $\frac{1}{3}$" = 1"
W. T. Gorton



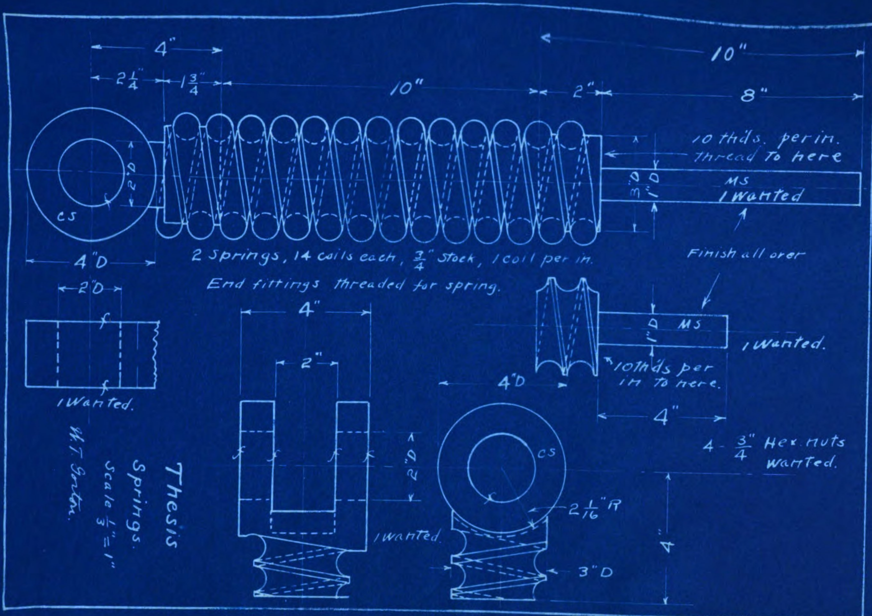
IX
Circular pitch, .5236"
24"
1"
square
1"D
CRS
Finish all over
Spindle
1/16" x 1/8" keyway
Gear
CI
Rack to be bolted to top of frame
Teeth same as gear
1/2"
CI
4'-6"
1/2"
1"D
2"D
4"D
pitch circle 6"D
O.D.-6.333"
Thesis
Rack and pinion
Scale 1/2" = 1"
H. T. Gorton R. O. Amundson

X

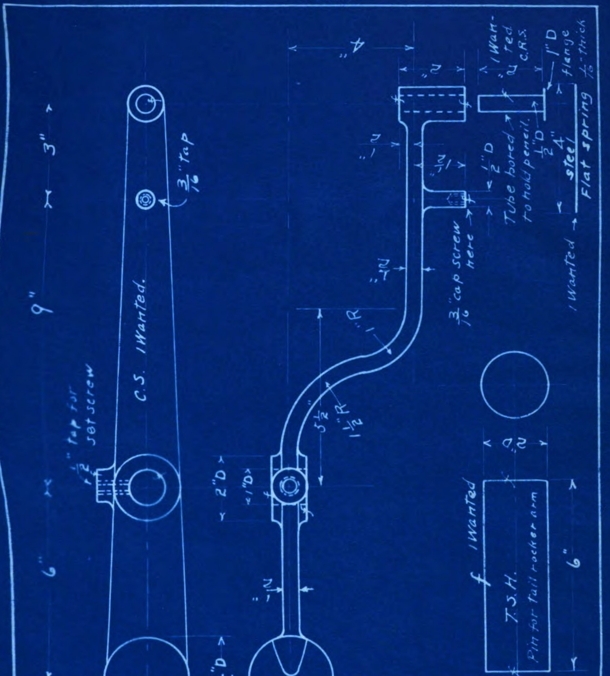


Thesis.
Method of Holding
Specimen

W. T. Gorton, R. O. Knudson



10"
4"
2 1/4"
1 3/4"
10"
2"
8"
10 thds. per in.
thread to here
3"D
1"D
MS
1 Wanted
CS
2"D
4"D
2 Springs, 14 coils each, 3/4" stock, 1 coil per in.
End fittings threaded for spring.
Finish all over
2"D
1 Wanted.
4"
2"
2"D
1 Wanted.
4"D
1"D
MS
1 Wanted.
10 thds per in to here.
4"
CS
4 - 3/4" Hex nuts wanted.
2 1/16"R
3"D
4"
Thesis
Springs.
Scale 1/3" = 1"
W. T. Gorton.

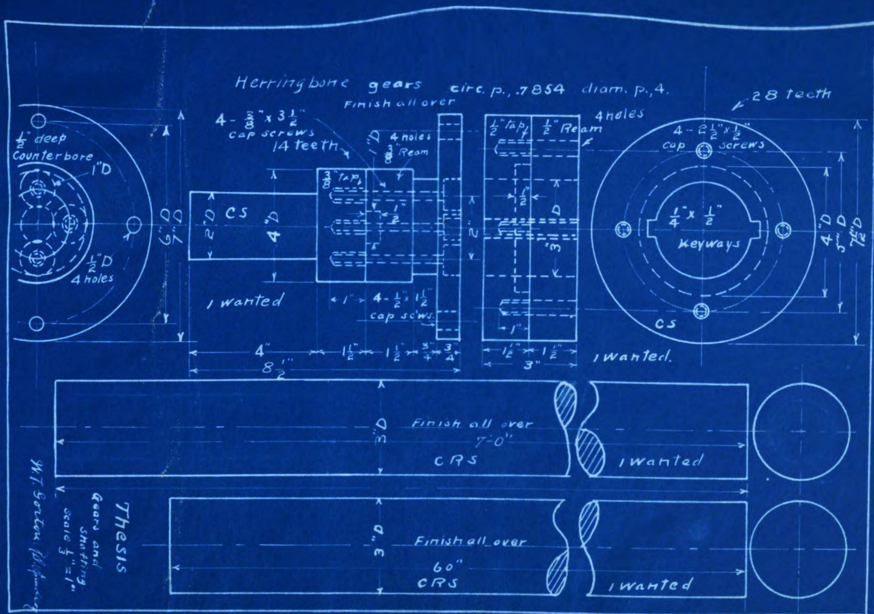


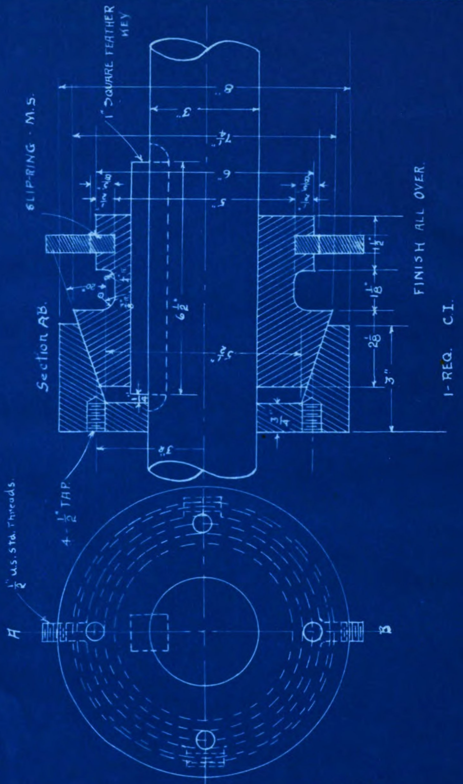
Thesis

Tracing Arm.

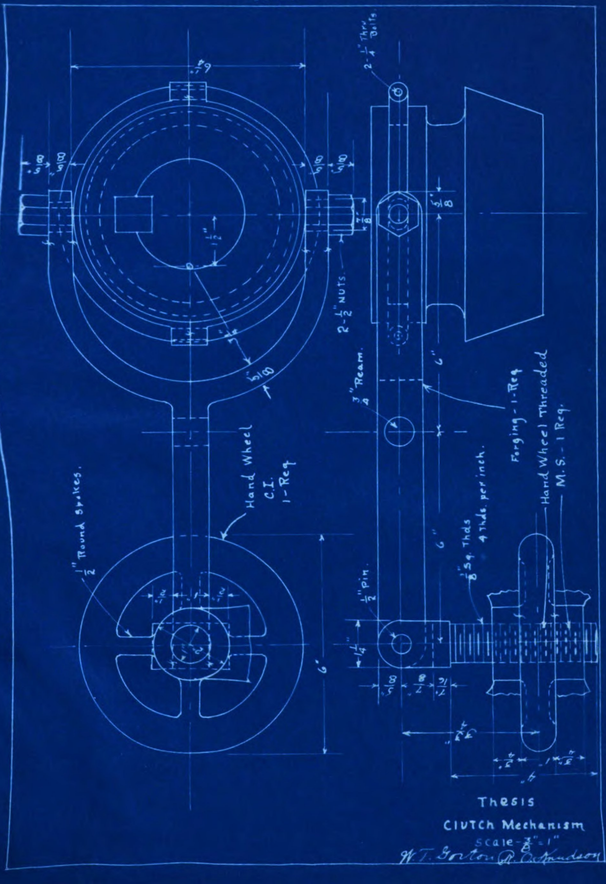
scale 1/3" = 1"

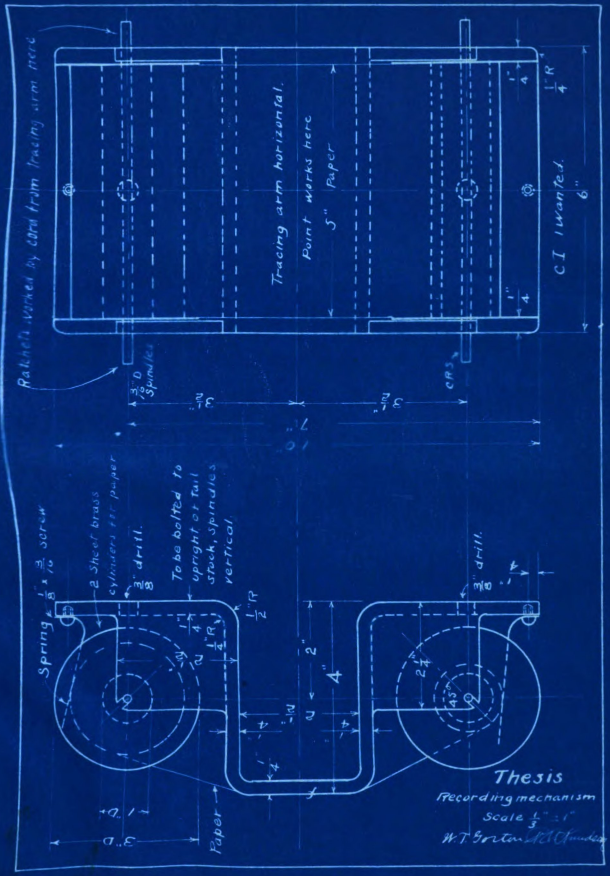
H. T. Gorton.

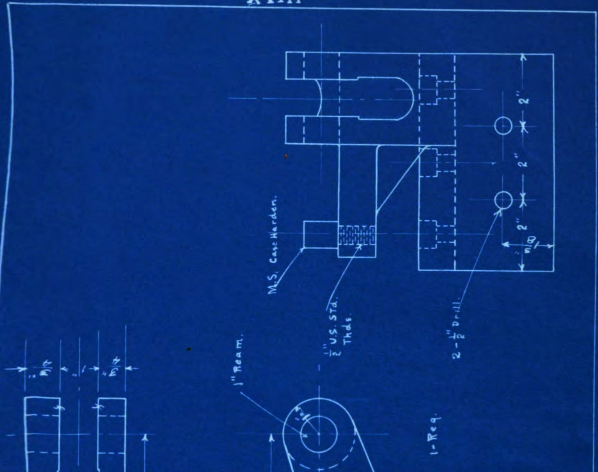

Herringbone gears circ. p., .7854 diam. p., 4.
Finish all over
4 - 3/8" x 3 1/2" cap screws
14 teeth
1"D 4 holes 3/8" Ream
3/8" tap
1/2" deep Counterbore
1"D
1/2"D 4 holes
6"D
7"D
2"D
CS
4"D
1 wanted
4 - 1/2" x 1 1/2" cap sc'ws.
4"
1 1/2"
3/4"
8 1/2"
4 holes
1/2" tap
1/2" Ream
2"
3"D
1"
3"
1 wanted.
28 teeth
4 - 2 1/2" x 1/2" cap screws
1/4" x 1/2" Keyways
5"D
7 1/2"D
Finish all over
7'-0"
CRS
3"D
60"
Thesis
Gears and Shafting
scale 1/3" = 1"
W. T. Gorton



THESIS.
FRICTION CLUTCH
SCALE 3/8" = 1"
W. T. Gorton. R. O. Knudson


2-1/4" Thru Bolts.
2-1/2" NUTS.
Hand Wheel
C.I.
1-Req
1/2" Round Spokes.
3/4" Ream.
Forging - 1-Req
1/2" pin.
5/8" Sq. Thds
4 Thds. per inch.
Hand Wheel Threaded
M.S. - 1 Req.
Thesis
Clutch Mechanism
Scale - 3/8" = 1"
W. T. Gorton R. O. Knudson


Ratchet worked by cord from tracing arm here
Tracing arm horizontal.
Point works here
5" Paper
C I 1 wanted.
6"
1/4"
1/4" R
3/16" D Spindles
CRS
3 1/2"
3 1/2"
7"
10"
Spring 1/8" x 3/16" screw
2 Sheet brass cylinders for paper
3/8" drill.
3/8" drill.
To be bolted to upright of tail stock, spindles vertical.
1/2" R
1/4" R
2"
4"
2 1/4"
45°
1" D
3" D
Paper
Thesis
Recording mechanism
Scale 1/3" = 1"
W. T. Gorton



Thesis.
Bracket for Clutch Mech.
Scale - 3/8" = 1"

W. T. Gorton   G. O. Knudson