A

E X P E R I M E N T A L   S T U D I E S

on

C R A N K S H A F T   S T I F F N E S S

and

F O R C E   F I T   G R I P

by

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(d)/

PART I. (continued)

INTRODUCTION

INTRODUCTION

The phenomena of torsional vibrations in engine shafts have created engineering problems of the first importance. While possibly the conditions have been demonstrated with all engines they are most conspicuous in the internal combustion types. In particular, the rapid progress of the Diesel engine has given rise to extreme difficulties of the kind, and the large scale on which such engines are built makes the probability of failure a serious matter that compels the most careful attention to all the factors of the problem.

The fundamentals of vibration theory are quite sufficient for an understanding of the phenomenon. The synchronisation of a periodic engine impulse with the natural period of the shaft system and the severe cyclical stresses that may thereby be created are simple conditions than can be easily appreciated. Material fatigue, due to repeated actions, will account for any failure that may take place. But these are qualitative conceptions and are wholly inadequate in a design process that attempts to escape the difficulties. Design requires precise quantitative information on a number of points that are quite reasonably ignored in general theory but are important in practical construction.

Chief amongst these essential factors are the elastic and damping properties of the construction. In both, theory works mainly with overall representations. Both, however, require detail specification for accurate estimates of critical states. While the importance of these is now well/

well understood and investigations of various kind are fairly numerous, their full significance have only been appreciated after many failures. In fact, the seriousness of critical conditions and the necessity for close investigation of what appears to be trifling detail, are lessons that were learned at excessive cost.

We are here concerned with the problems of stiffness. Damping problems are in a different category, and have to be tested independently. The stiffness characteristics of a cranked shaft raise many points, obvious or unexpected. Analytical treatments have been frequently made and empirical expressions are quite well known. The former necessitates simplifying assumptions for such conditions as web stiffness and are, to a large extent, incapable of representing the influences of form changes or shaft and pin connections. The latter incorporate observed results, generally in approximate forms and for limited series, and do not serve to indicate possible effects due to form changes while they wholly ignore the consequences of assembly method.

The more general analytical studies apply to solid shaft types, while the better empirical rules apply to the smaller high speed engines which use this construction. Neither class of investigation can, therefore, be held to apply directly or conclusively to the heavier arrangement with assembled elements. These introduce most notably the web influences and the force fit problem thereby providing variables that are not amenable to theoretical treatment. In a theoretical sense they are too complex; and in a practical sense have been imperfectly understood.

(a) Analytical Formulae for Crankshaft Stiffness.

The single-throw-two-bearing crankshaft has been treated analytically for different conditions of bearing restraint by Timoshenko[1] and Holzer[2]. It is assumed, in their treatment of the problem, that the throw is built up of component parts and that the strain induced in each part when transmitting torque is unaffected by any other part. On the assumption that simple elastic theory can be applied the deformation of each component part is found from known elementary formulae for bending and torsion.
By the principle of superposition of small deformations the overall deformation or twist is found by summing up the separate effects of all the parts.

In their analysis certain web form conditions and dimensions are assumed as best meeting the requirements for calculating the flexural and torsional rigidities of the web. With some such simplifying assumptions a theoretical analysis of the problem may lead to results in close agreement with experiment or, at least, sufficiently accurate for practical purposes.

It is important to note that neither Timoshenko nor Holzer considers the effect of web form changes beyond the crank-pin and journal centre lines. More important still, both writers base their analysis of the problem on a crankshaft of solid form. No attempt is made by either to deal with the built-up crankshaft and the effect of web fit.

As the discussion of the experimental resulst that are to be presented involves a critical comparison with the expressions and ideas of theory, it is considered advisable to give/

give here an outline of the accepted theoretical method. This serves to establish notations and define the approximations and assumptions underlying the general treatment. In the main the development is according to Timoshenko. It has been extended to cover the case when the bearings are not fitted close to the crank webs, as in the experiments carried out.

If a pure torque M is applied to the ends of a single-throw crankshaft out of bearings, the straining actions induced will result in a lateral displacement of the journals. In the journals, crankpin and webs shown in Fig.1 . there will be pure torsion about the axes OO, $O_1 O_1$, and $O_2 O_2$. There will be bending in the webs in addition. On the assumption that simple elastic theory can be applied the relative angular displacement of the overall gauge points E-F is given by:

$$\theta_{EF} = M \left(\frac{2b}{C_1} + \frac{a}{C_2} + \frac{2h}{C_3} + \frac{2r}{B_3} \right) \qquad \ldots\ldots\ldots\ldots (1)$$

where C_1 = Torsional rigidity of journal

C_2 = Torsional rigidity of crankpin

C_3 = Torsional rigidity of web

B_3 = Flexural rigidity of web with respect to bending in the plane pp perpendicular to the plane of the figure.

The actual twist may differ from that given in (1) due to the fact that the expression is based on the assumption that the strain in each component part is unaffected by any other part. Also, the cross-section of the web, which is not clearly defined, is assumed to be that of a rectangle with sides r and C, and in addition, the effect of web form beyond the pin and journal centre lines is not considered.

When/

FIG. 1. SINGLE THROW CRANKSHAFT.

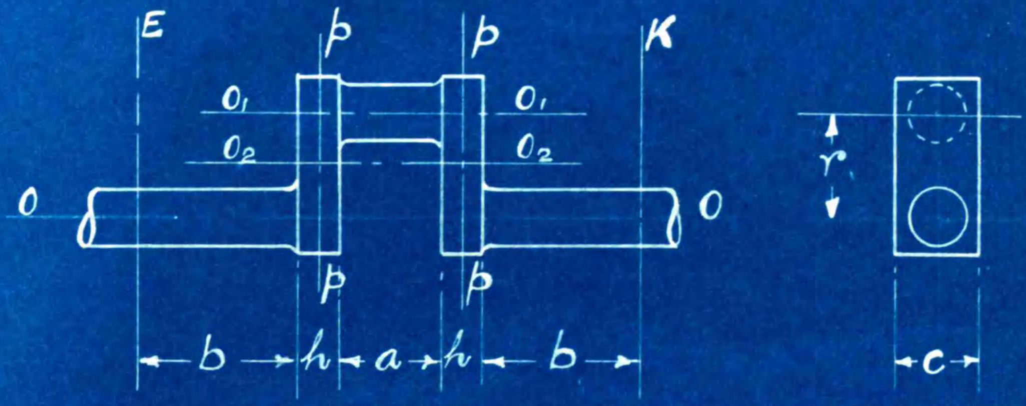


FIG. 1(a) DIAGRAM OF DEFORMED CRANK

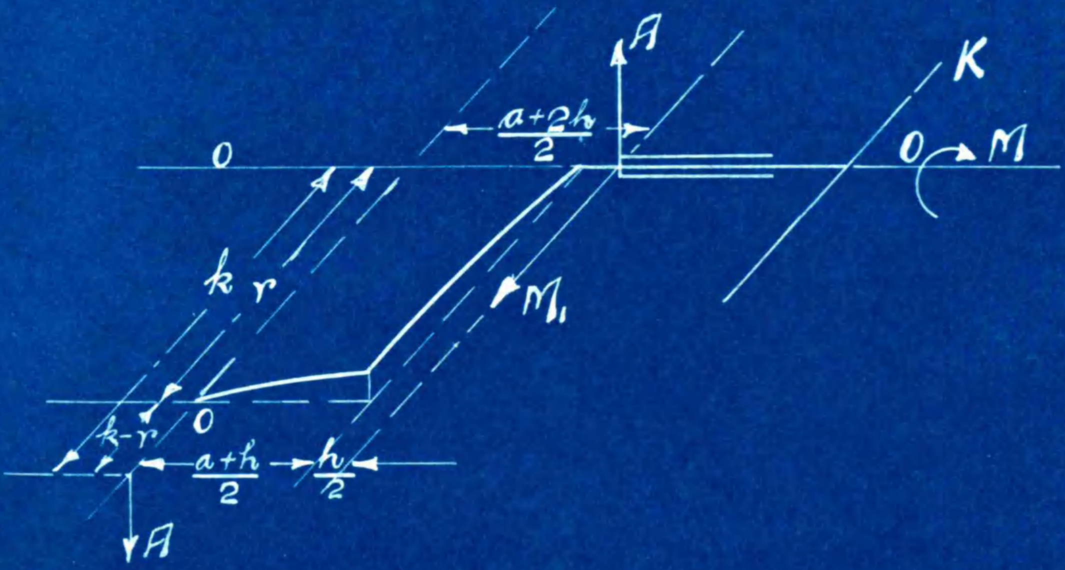
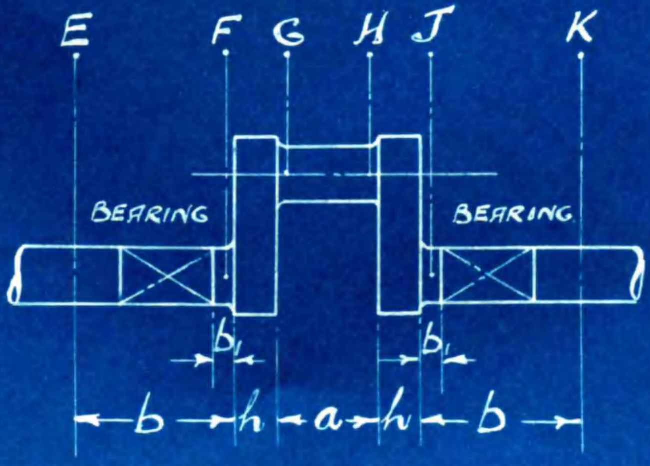


FIG. 1(b). POSITION OF BEARINGS RELATIVE TO CRANK WEBS.



When subjected to an in-bearings test, as in experiment, with bearings preventing any movement of the journals other than one of rotation about the axis of the shaft, additional forces and couples are introduced. The constraint gives rise to a force A and a moment M_1 at each journal acting in a plane through the axis of the journal perpendicular to the plane of the throw. The straining action induced in the shaft is such that the effect of one half of the shaft on the other half can be established by introducing an equal and opposite force to the journal force A acting at a distance k from the axis 00, such that

$$M_1 = \frac{A(a + 2h)}{2} \quad \text{and } Ak = M.$$

The relative angular displacement of the overall gauge points E-F is then given by:-

$$\theta_{EF} = \frac{2Mb}{C_1} + \frac{A(R-r)a}{C_2} + \frac{2A(R-\frac{r}{2})h}{C_3} + \frac{2Ar(R-\frac{r}{2})}{B_3}$$

$$\text{and since } A = \frac{M}{k}$$

$$\theta_{EF} = \frac{2Mb}{C_1} + \frac{M(R-r)a}{kC_2} + \frac{2M(R-\frac{r}{2})h}{kC_3} + \frac{2Mr(R-\frac{r}{2})}{kB_3} \text{-----} (2)$$

The in-bearings overall angle of twist for any given twisting moment M is thus readily found if the bearing reaction A is known.

This is obtained from the calculated value of k which may be determined as follows:-

Assume one half of the crank, before straining, to be rotated about the journal axis an amount equal to one half the angle θ_{EF} given by (2). The mid point of the crank-pin will thus move forward a distance $\frac{r\,\theta_{EF}}{2}$. If the journal is now fixed at the gauge point K, distant b from the web, and/

and a force A and couple M applied, the mid point of the crank-pin will be displaced back a distance $\frac{r\theta_{EF}}{2}$, and will thus have no displacement perpendicular to the initial plane of the crank. The deformed crank is shown in Fig. 1(a) The displacement of the point O may now be equated to the sum of the displacements of the point O resulting from the straining actions induced in the separate parts.

The separate displacements will be due to

(a) Flexure of Crankpin $= \dfrac{Aa^3}{24 B_2}$ ....(3)

where B_2 = Flexural rigidity of crankpin.

(b) Shear in Crankpin $= \dfrac{yAa}{2F_2G}$ ....(4)

where y = Constant taken to be 1.2.

(c) Twisting of web about axis pp $= \dfrac{A(a+h)^2r}{4C_3'}$.(5)

where C_3' = Torsional rigidity of web considered rectangular cross-section with sides h and c.

(d) Bending of web $= \dfrac{Ar^2(3R-r)}{6 B_3}$ ....(6)

(e) Shear in web $= \dfrac{yAr}{F_3'G}$ ....(7)

where F_3' = Cross-section of web taken on OO.

(f) Shear in web $= \dfrac{yAh}{F_3G}$ ....(8)

where F_3 = Cross-section of web on line pp which, not being clearly defined, is taken as rectangular with sides r and c.

(g) Twist of web about O_2O_2 $= \dfrac{A(R-r/2)hr}{2C_3}$ ....(9)

(h) Twist in journal b $= \dfrac{Mbr}{C_1}$ ....(10)

by/

by substituting the expression for θ_{EF} given in (2) and putting $\frac{M}{k}$ for A we obtain:-

$$k=\left\{\frac{(a+h)^2 r}{4C'_3}+\frac{ar^2}{2C_2}+\frac{a^3}{24B_2}+\frac{r^3}{3B_3}+\frac{hr^2}{4C_3}+\frac{\gamma a}{2F_2G}+\frac{\gamma r}{F'_3G}+\frac{\gamma h}{F_3G}\right\}$$

$$\div\left(\frac{ra}{2C_2}+\frac{r^2}{2B_3}+\frac{hr}{2C_3}\right) \qquad \text{...........(11)}$$

This is the form of the expression for k given by Timoshenko when the bearings are fitted against the crank webs. The bearing reaction A can be obtained from the condition $Ak = M$, where M is the known applied torque.

Due to the bearings being at a distance b_1 from the webs in the experiments carried out the mid point of the crank-pin will have a displacement due to the flexure of part b_1 of the journal together with a displacement due to shear of the journal, Fig. 1(b).

(i) Flexure of journal =

$$\frac{ab_1^3}{3B_1}+\frac{Ab_1^2(a+2h)}{2B_1}+\frac{Ab_1(a+2h)^2}{4B_1} \qquad \text{.........(12)}$$

where B_1 = Flexural rigidity of journal.

(j) Shear of journal $=\frac{\gamma Ab_1}{F_1G}$ ....(13)

where F_1 = Cross-section of journal.

By substituting $\frac{M}{k}$ for A the following terms must be included:-

$$\frac{b_1^3}{3B_1}+\frac{b_1^2(a+2h)}{2B_1}+\frac{b_1(a+2h)^2}{4B_1}+\frac{\gamma b_1}{F_1G}$$

and the value of k is given by:-

$$k=\frac{\left\{\frac{(a+h)^2 r}{4C'_3}+\frac{ar^2}{2C_2}+\frac{a^3}{24B_2}+\frac{r^3}{3B_3}+\frac{hr^2}{4C_3}+\frac{\gamma a}{2F_2G}+\frac{\gamma r}{F'_3G}+\frac{\gamma h}{F_3G}\right.}{\frac{ra}{2C_2}+\frac{r^2}{2B_3}+\frac{hr}{2C_3}}$$

$$+\frac{b_1^3}{3B_1}+\frac{b_1^2(a+2h)}{2B_1}+\frac{b_1(a+2h)^2}{4B_1}+\frac{\gamma b_1}{F_1G} \qquad \text{..................(14)}$$

the bearing reaction A is thus influenced by the position of the bearing relative to the web.

(b) Empirical Formulae for Crankshaft Stiffness

There is associated with any theoretical analysis of the static stiffness of a crankshaft the uncertainty of the error involved consequent to the simplifying conditions assumed. There is a reasonable probability of the webs effecting a reduction in the twist induced in the crank-pin and journals. There is, on the other hand, a possibility of the twist in the pin and journals persisting some distance into the webs.

Efforts have not been lacking in an attempt to develop formulae based on the results of static torsion tests. A semi-rational formula by Seelman[3] assumes that the influence of each crank web is equivalent to a lengthening of the crank-pin by 0.45 web thickness. This value was determined by comparison with test results of torsional experiments.

B.C. Carter[4] has put forward an empirical formula evolved from the results of stiffness tests on a number of marine, aircraft, and motor car crankshafts. This formula expresses the crankshaft stiffness in terms of the equivalent length of journal shafting.

It comprises three terms and is given by:-

$$\ell = (2b + 0{\cdot}8h) + \frac{3}{4}\frac{(d_1^4 - \delta_1^4)}{(d_2^4 - \delta_2^4)}\,a + \frac{3r}{2}\frac{(d_1^4 - \delta_1^4)}{h w^3}$$

where ℓ = equivalent length of journal shaft.
d_1 = outside diameter of journal.
δ_1 = inside diameter of journal.
b = length of journal.
d_2 = outside diameter of crank-pin.
δ_2 = inside diameter of crank-pin.
a = length of crank-pin.
w = width of web.
h = Thickness of web
r = length of throw.

The numerical coefficients of the three terms were determined by trial and error and the test readings taken were for overall distortion.
The formulae presented by Seelman and Carter do not include factors to cover the effect of web form variations beyond the crank-pin and journal centre lines. They are conditioned also by factors associated with crankshafts of a solid form only and thus fail to deal with the effect of web fit which is an inherent feature of the heavier shaft systems in practice.

(c) <u>The problems Associated with Crankshaft Stiffness</u>

If the crank throw were reduced to zero then the effect of the webs would be to increase the stiffness of a shaft of uniform diameter, if the crank-pin and journal diameters were equal. When thus located the webs would provide a maximum stiffening influence. The stiffening influence of any particular web form, however, would decrease as the throw increased and for some particular length of throw the stiffening influence would be reduced to zero.
The angle of twist would then be equal to that obtained over a corresponding length of the journal shaft.
The crank webs would then be subjected to complex straining actions due to conditions induced by the crank throw.

A long and a short throw crank, a solid and a fitted web crank, a crank with bevelled, curved, and square webs, cannot all equally justify the assumption of uniformity in web cross-section between the crank-pin and journal centre lines.

The/

The influence of web elasticity and form together with the influence of web fit are the main problems associated with crankshaft stiffness. The first has been analysed by making simplifying assumptions in theory and by trial and error rules based on test results. The latter has not yet been critically examined.

It would appear, therefore, as Tollé states that "the problem as to how a crankshaft acts during the transmission of torque has, as yet, not been satisfactorily explained; the most practical way is to determine the amount of torsional deflection due to several moments by experimental results."

Such tests are necessary to provide proper guidance for full theory, if they support the theory, but still more necessary if they discover effects that are ignored by theory. It may be contended that there is no valid theoretical attack on the built-up shaft schemes and that experiment alone can be relied upon for proper views regarding these.

The difficulties associated with an experimental study of large size elements may be reduced somewhat by the use of small scale forms. Investigations on models play an increasingly important part in modern technical research not only on the score of convenience but also because of a better appreciation of the powers and limitations of the method.

Correct testing by models is not merely a matter of satisfying geometrical similarity. The conditions of dynamical similarity have to be kept in view. In the case/

case of elastic structures it is easy to meet these, if there is continuity and similarity of structure, and if distortions due to static external loads only are to be considered. But, where special conditions arise, such as occur at structural joints, bearing surfaces, or fitted parts, there is confusion of the factors, and the various conditions of similarity necessary to attain a completely valid model cannot be satisfied together.

Except in connection with bearing effects these difficulties should not arise for solid shaft types. Hence model tests for this class should give fair guidance, on the whole, and the study of web effects, in particular, by the use of such shaft models, is legitimate. The same cannot be said, however, for shafts with fitted webs involving elastic grip and contact surface conditions of variable degrees. These factors cannot be properly incorporated in shaft models that are geometrical copies of a full scale arrangement. It is therefore necessary to examine their effects independently. This examination may be on small size parts but the essential factors are therein isolated, and hence the research is not essentially by models.

The slipping of pins and journals in crank webs of built-up crankshafts is a form of crankshaft failure which cannot be said to have been thoroughly investigated. The result is that workshop practice in the making of these important shrink fit assemblies varies greatly and suggests that differences exist in the minds of engineers regarding factors which control the quality of the grip in such assemblies.

The/

The nature of the finish of the mating surfaces differs widely. These may be machined but unpolished and not very smooth; they may be turned or ground, or they may be dry-scraped or wet-scraped and given a high polish.
The webs may be water cooled or cooled out in air.
The shrink fit allowance would appear to vary within wide limits. The allowance adopted by marine crankshaft builders ranges between 1 and 1.75 thousandths of an inch per inch diameter for the same quality of material.
The results obtained, in many cases, would appear to be equally successful despite the fact that at the higher fit allowances the stress induced in the bore layers is well above the elastic limit of the material.

It is the practice, in some places, to lubricate the pin and journals in making such assemblies in the hope of avoiding a preseizure of the elements during assembly.
Is it possible that a surface contact film may thus separate the elements on assembly and control in a large measure the quality of the grip established? Failure by slipping of the webs on journals of crankshafts assembled under such conditions, and fitted with a high shrinkage allowance, has taken place even during trial runs.

Failure by slipping is one associated with the factors affecting the grip in an elastic grip assembly. Such an assembly, making use of the elastic properties of the material, would require the free diameter of the solid element to be slightly bigger than the free diameter of the hollow element. When such conditions obtain the assembly may be made by heating the outer or hollow element as a shrink/

shrink fit. It may be made by pressing the hollow element over the solid element as in force fit practice. It may also be made by cooling the inner or solid element to an extremely low temperature, by immersion in a refrigerant, and then allowing it to grip the hollow element as an expansion fit.

The effect of the assembly method on the quality of the grip established in an elastic grip assembly has, as yet, not been satisfactorily analysed on the basis of comparative test results. Although many important built-up units in Railroad practice have for long been effected by force and shrink fit assemblies of the mating elements, there is evidence that there are factors controlling the grip in these assemblies which are not yet clearly defined.

An examination of 2,100 locomotive tyres on the German State Railways by L. Koch[5] showed that 11.16 per cent. had become loose. From experiments made upon the shrinkage of tyres he suggests that thin tyres are more prone to work loose, and that in all cases, accuracy of machining, precise determination of temperature, definite shrinkage allowance and uniform heating of the tyre for the shrinking on process should be assured. It is important to note that the effect of a surface contact film, which may separate the tyre and the wheel centre on assembly, has not been considered as a factor affecting the quality of the grip in this analysis.

Many important elastic grip assemblies are made by force fits. The use of a lubricant is a characteristic feature and/

and an essential requirement to a successful mating of the elements by this method of assembly. It is surprising, therefore, that, in spite of the importance of such assemblies in engineering practice, little or no attention has been given to the possible influence of this factor on the axial resistance of the elements to slip. If the surface contact film should control, in a great measure, the quality of the grip established then the empirical formulae presented by various investigators cannot be regarded as providing a satisfactory basis on which to fix a force fit allowance.

(d) Objects of the Present Investigations

The investigations undertaken by the author commenced with a desire to examine the influence of web form. This was the outcome of a feeling that the assumptions behind the theory of the subject were rather drastic in regard to this factor. Owing to the obvious convenience of model sizes small scale shafts of solid type were proposed and an examination of a geometrical series of these for scale effect was first undertaken. This was followed by an investigation of the effects of progressive changes in web form.

It was intended to include fitted webs in this series but these at once brought in new and unexpected factors that compelled a special study of force fits. This is carried out, first of all, on small size fittings but because of the significant results, it is extended to include practical assemblies.

The/

The original idea of the research was to study the features of crank shaft elasticity. The inherent characteristics of the fitted element were unexpected but considered of sufficient importance to justify the later concentration on this factor.

It is, therefore, thought that the presentation of the work should follow the natural order of development and so the following pages deal successively with the author's experimental work on a series of similar single-throw crankshaft models; on the influence of web form on stiffness of these; on the factors affecting the grip in small size elastic grip assemblies; and on the examination of corresponding influences in full scale wheel assemblies.

PART I.

INVESTIGATIONS ON A SERIES OF GEOMETRICALLY SIMILAR SINGLE-THROW CRANKSHAFT MODELS

(a) Object of Tests.

An experimental investigation of the angular distortion of crankshafts by small scale models requires (a) geometrical similarity between the model and the full size crankshaft, and (b) the use of similar or identical materials if the model is to give results comparable to those of the full size body.

If a single model is used in such an investigation it is reasonable to suggest that it should be made to a relatively large scale so that the condition of geometrical similarity, which must obtain, might be easily and faithfully reproduced. Even then, it is doubtful whether definite conclusions could be drawn with assurance.

A more critical analysis of the problem would indicate the necessity of demonstrating a close similarity of effects when a number of geometrically similar models made to widely different scales were subjected to torsion. This method of analysis would provide a basis of comparison and a means of determining whether inaccuracies due to workmanship, which become more serious as the scale is reduced, had any appreciable influence on the torsional characteristics of the shaft. On the basis of such comparative tests it would be possible to study and examine, with greater confidence, the effect of varying conditions on any one model.

The latter method of investigation was adopted. In order to assist in the analysis of the torsional effects, it was decided to measure not only the overall distortion, but also the distortion of the separate elements, namely webs, journals, and crank-pin of each model.

(b) Experimental Crankshafts./

(b) Experimental Crankshafts.

Four single-throw-two-bearing crankshaft models were made to different scales. The shafts were machined out of short lengths of mild steel shafting supplied by the Govan Shafting and Engineering Co. Each length was roughly shaped to crankshaft form, first of all, in a shaping machine. Subsequent machining operations were carried out in a centre lathe and the final metal removal operations were carried out by wet-scraping the surfaces. All the machining operations on crankshafts A, B, and C, shown in Fig. 2. were carried out by the Author in the workshops of the Civil and Mechanical Engineering Department of the Royal Technical College, Glasgow. The crank proportions and dimensions of shaft D were taken from a four cylinder petrol engine crankshaft, of an old type, stored in the College workshops. This model was made with cylindrical webs having a diameter equal to that of the web length of the four cylinder crankshaft.

This type of web form gave a maximum stiffness to the experimental shafts and was the most suitable one for carrying out experiments to investigate the effect of web form on crankshaft stiffness. The change in web form was effected by a reduction in the web width between the crank-pin and journal centre lines and beyond these centres by a reduction in the web thickness by bevelling the webs.

All four shafts were machined from the shafting as supplied by the makers and when completed were not subjected to any special treatment.
The four crankshaft models, as machined, are shown in Fig. 2
The crank proportions and dimensions of the experimental shafts, reduced to the same journal diameter, are shown in Fig. 3 .

(I)/

Fig. 2 . Crankshaft Models used in Static Torsion Tests.

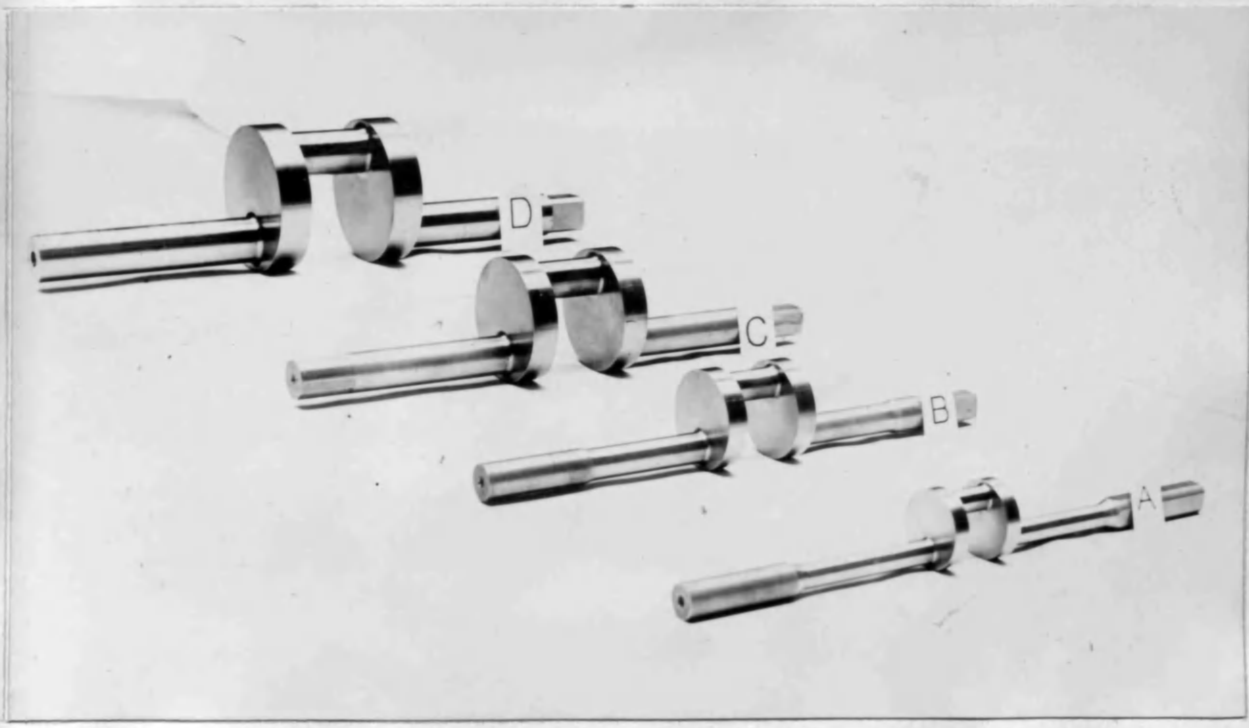
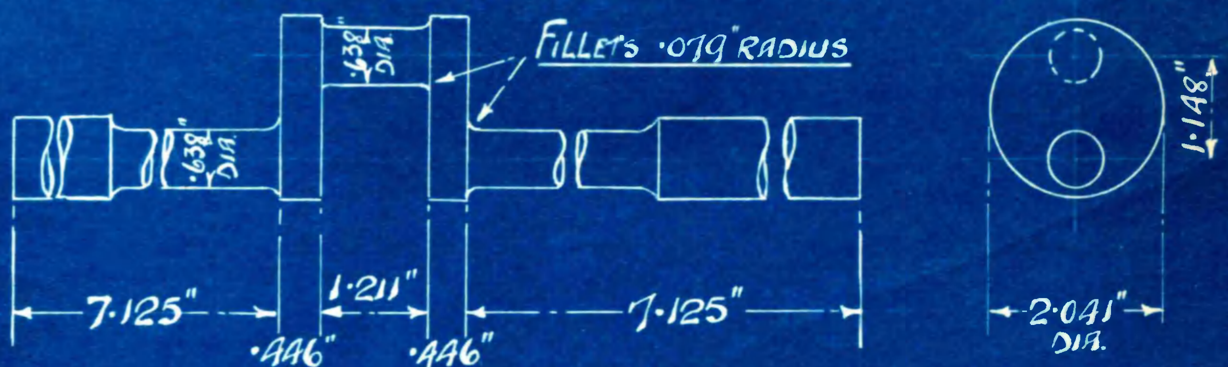


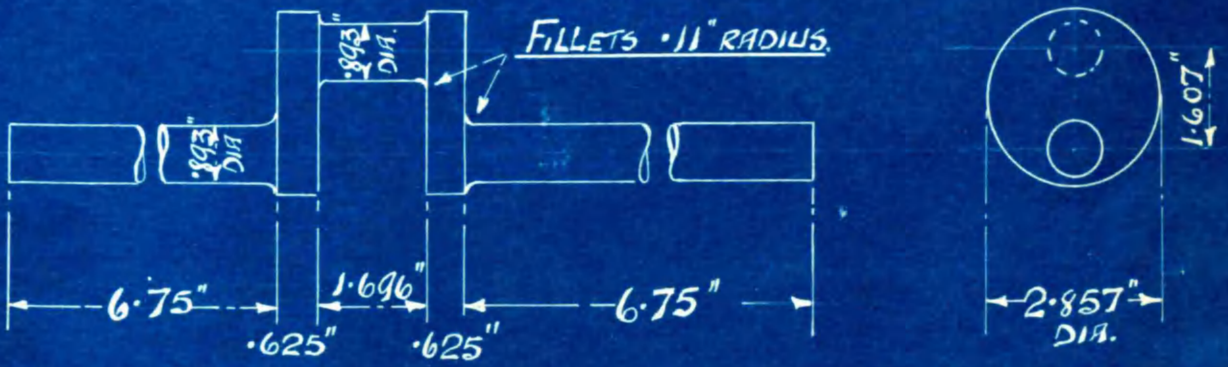
Fig. 3 Crankshaft Models for Static Torsion Tests

Reduced to same Journal Diameter.

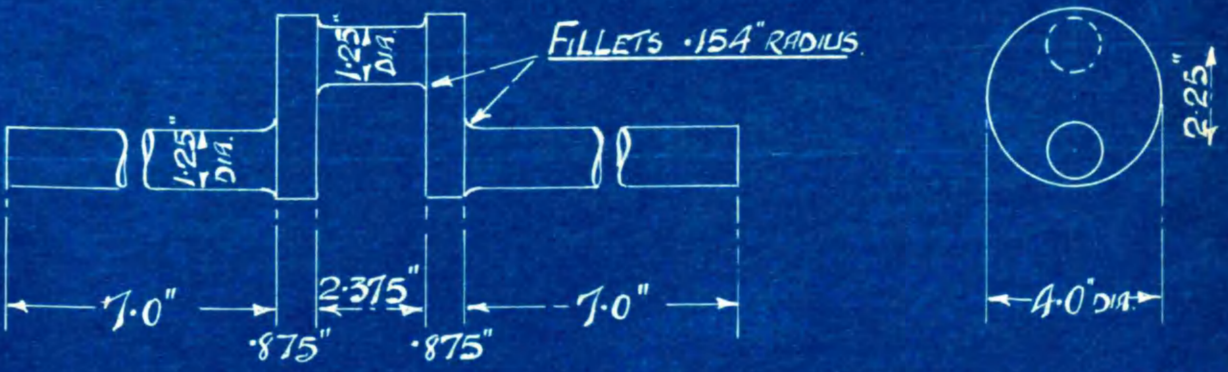
Crankshaft Model A



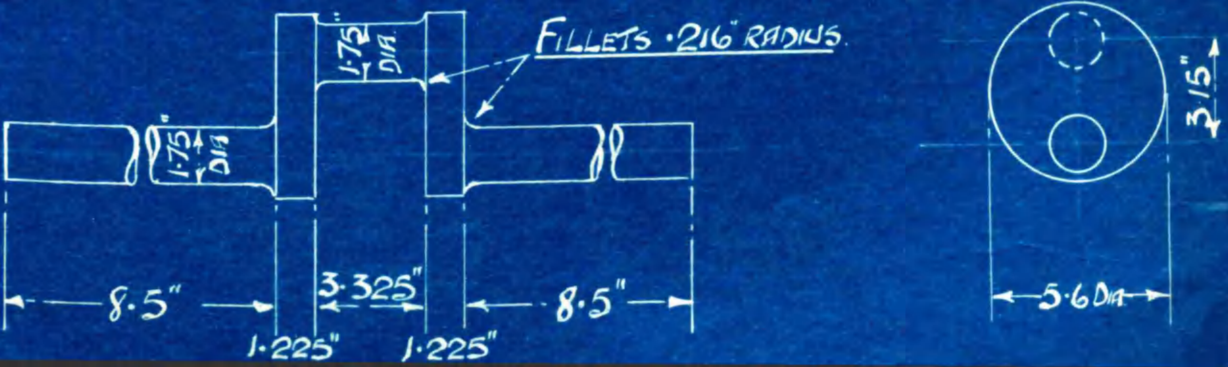
Crankshaft Model B



Crankshaft Model C.



Crankshaft Model D.



(I) <u>Crankshaft Notation and Dimensions</u>: Fig. 4. comprises a diagram showing the notation adopted and a table giving the dimensions of the various models and the torsional gauge lengths of the separate elements of each crankshaft. Two light centre punch marks, 180 degrees apart, engage the points of the clamping screws at the gauge points indicated. The gauge points F, G, H, and J were at the ends of the journal and crankpin fillets.

Although crankshaft stiffness is usually measured over a length between the journal centres, the gauge points E and K may be any points on the journals. These points were fixed, in the tests which follow, so as to be clear of the bearings when in-bearings static torsional tests were carried out.

The crankshaft dimensions and the torsional gauge lengths tabulated for each scale model satisfies the condition of geometrical similarity. The crank scantlings and torsional gauge lengths are shown tabulated in terms of the journal diameter d_1. The average values of the scantling journal diameter ratios of a number of marine, aircraft, and car cranks examined by Carter are given to serve as a standard of comparison in considering crank proportions.

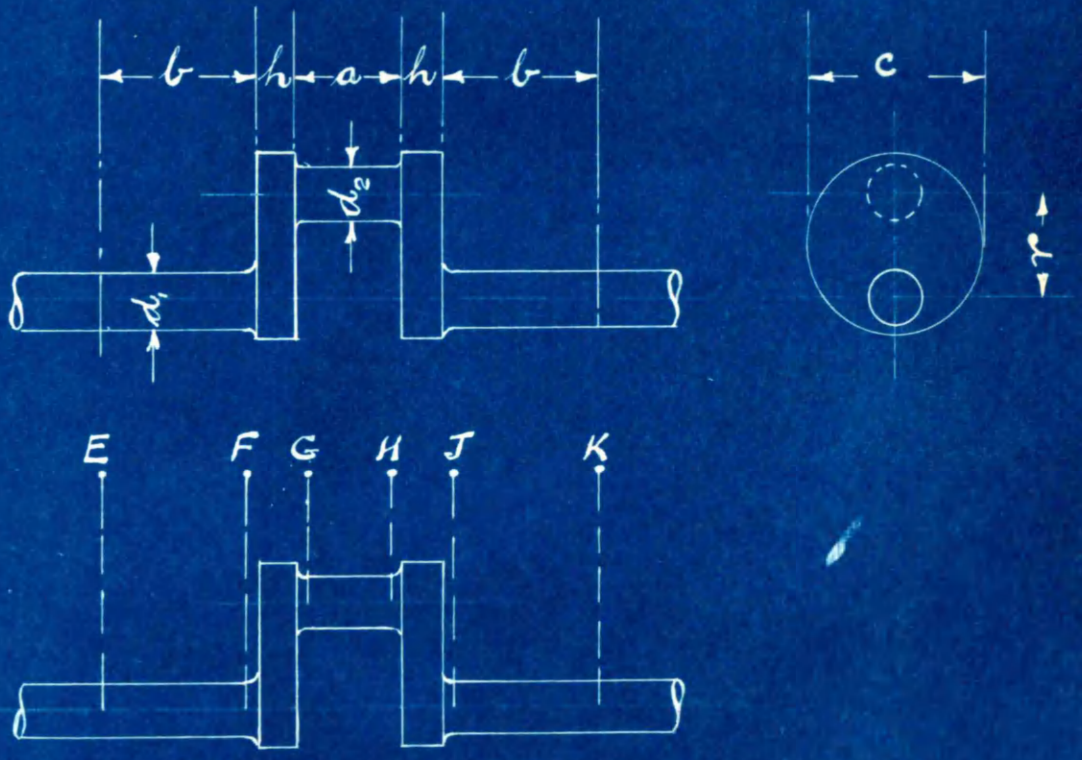
"Average Shaft" Scantling Journal Diameter Ratios.

d_2/d_1	h/d_1	a/d_1	r/d_1	c/d_1
0.9	0.42	0.927	1.06	1.29

Compared with the "average shaft", the crankshaft models are characterized by webs which are cylindrical and very thick, and by a crankpin length and throw which are exceedingly long.

(c)/

Fig. 4 Diagrams showing the Notation used and the Proportions Adopted in Tests of Geometrical Similar Crankshafts.



Shaft	d_1 INS.	d_2 INS.	h INS.	a INS.	c INS.	r INS.	E-F INS.	F-G INS.	G-H INS.	H-J INS.	J-K INS.
A	·638	·638	·446	1·211	2·041	1·148	1·625	·604	1·053	·604	1·625
B	·893	·893	·625	1·696	2·857	1·607	2·275	·845	1·476	·845	2·275
C	1·25	1·25	·875	2·375	4·00	2·25	3·185	1·183	2·067	1·183	3·185
D	1·75	1·75	1·225	3·325	5·60	3·15	4·459	1·657	2·893	1·657	4·459

Non-dimensional Relationship.

d_2/d_1	h/d_1	a/d_1	c/d_1	r/d_1	EF/d_1	FG/d_1	GH/d_1	HJ/d_1	JK/d_1
1·0	0·699	1·90	3·2	1·80	2·548	0·946	1·654	0·946	2·548

(c) Properties of Crankshaft Material.

Two test pieces were cut from each of the four short lengths of mild steel shafting supplied. Tensile and static torsion tests and chemical analyses were made of the specimens prepared and the physical properties and chemical composition of each shaft material determined. Youngs modulus and the rigidity modulus of elasticity enter as the principal physical constants in most elastic equations. Although these values are practically the same for most steels, and may be assumed with a fair degree of accuracy, the experimental values obtained allow of exact values being substituted in subsequent equations. The experimental values obtained of the tension and torsional limits of proportionality of each crankshaft steel, which in general may vary, clearly defined the safe stress conditions for the static torsion tests of the shafts, in and out of bearings.

(I) Tensile Tests: These tests were carried out on a 100-ton Avery horizontal testing machine. The extensions of the test piece, 0.564 in. diameter, were read on a 2 in. gauge length by means of a Marten's mirror extensometer.

TABLE I. - Tensile Test Results.

Test Piece Reference	Shaft	Limit of Proportionality tons per sq.in.	Ultimate Tensile Strength tons per sq.in.	Youngs Modulus tons per sq.in.	Percentage Elongation	Percentage Reduction in area.
a	A	10.8	27.87	11,800	41	60
b	B	10.8	30.0	11,260	35	48
c	C	8.0	31.58	11,150	38	57
d	D	5.6	26.1	10,840	44	63

(II) Static Torsion Tests: These tests were carried out on a 15,000 in.lb. Avery reverse torsion testing machine. The free axial/

axial movement of the chuck at the weighing end of this machine ensures that the specimen is subjected to pure torsion.

To determine the limit of proportionality and modulus of elasticity two small saddles were attached to 0.6 in. diameter test-piece. Each saddle had two pointed clamping screws which engaged with light centre punch marks 180° apart. The final adjustment of each saddle on planes 4 in. apart was made by a small set-screw suitably positioned on each saddle. An adjustable mirror of a Marten's mirror extensometer was mounted on each saddle. The rotational movement at each plane was observed by means of two telescopes and two scales. Each mirror reflects a view of the curved illuminated scale with the cross-hair of the telescope lying across it.

The twist on the gauge length, for each small increment of torque, was measured by observing the relative movement of the two mirrors.

TABLE 2. - Static Torsion Test Results.

Test-piece Reference	Shaft	Limit of Proportionality tons per sq.in.	Modulus of Rigidity lb.per sq.in.
a	A	7.26	11.880×10^6
b	B	7.89	11.440×10^6
c	C	5.7	11.640×10^6
d	D	4.37	11.490×10^6

(III) <u>Chemical Tests:</u> The results of analysis of the material for crankshafts A, B, C, and D were obtained from cuttings of two test-pieces taken from each short length of shafting supplied.

TABLE 3. - Chemical Composition of Material

Steel Cuttings Reference	Shaft	C %	Mn %	Si %	S %	Ph %
a	A	0.21	0.63	0.014	0.020	0.043
b	B	0.20	0.52	0.004	0.06	0.04
c	C	0.21	0.65	0.016	0.035	0.047
d	D	0.23	0.62	0.008	0.036	0.02

(d) Experimental Procedure

Each crankshaft model was subjected to static torsion tests, out-of-bearings and measurements were taken of the angular distortion of each individual element in addition to readings of the overall shaft distortion.

(I) Plant Arrangement: If a pure torque is applied to the ends of a single throw crankshaft, which is out of bearings, the straining actions induced will result in an axial displacement as well as a more pronounced lateral displacement of the journals. Complete freedom to any scuh axial and lateral movements of the journals is an essential condition which must be satisfied in any such tests.

It was thought that this condition would be best met by twisting the shafts in a 15,000 lb.in. reverse torsion testing machine and adopting the lay-out as shown in Fig.5 .
One journal, fixed in the chuck at the weighing end of the machine, was constrained directionally by a bearing providing free rotational movement and complete lateral constraint.
The bearing rested on a thick sole-plate supported by a rigid H beam which was bedded and firmly bolted to the bed of the machine.
The other end of the shaft, machined to receive one end of an assembly made up of two universal joints, was, by this arrangement, securely fixed to the chuck carried on a sliding carriage. This carriage, by means of a screw in the bed of the machine, could be moved to suit the test length requirements of the various shafts tested.

This method of fixing the shaft, although accompanied by a bearing constraint which was complete, provided for the free lateral movement of the journals due to any straining action induced in the webs and crankpin when subjected to torque.
By/

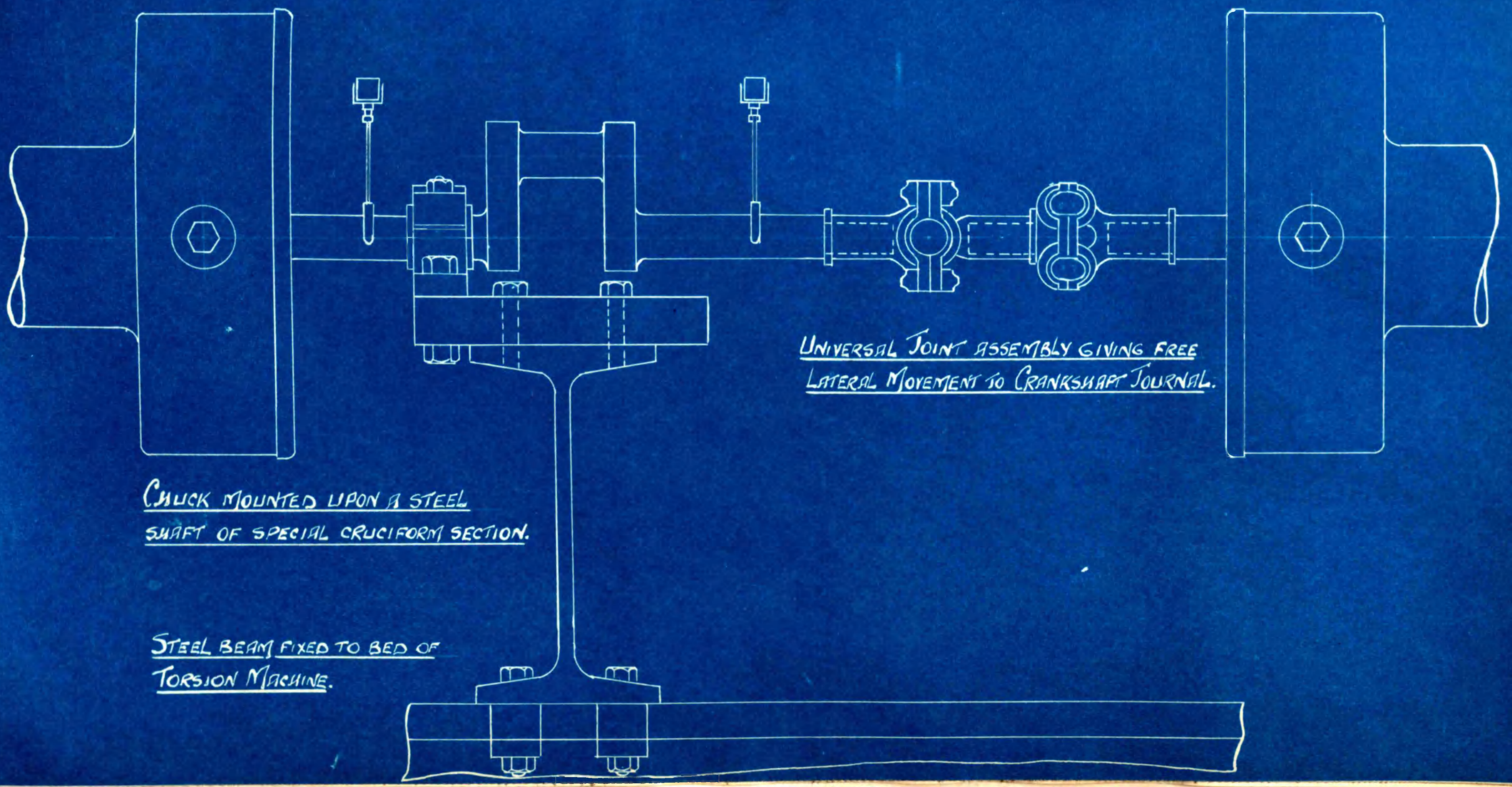
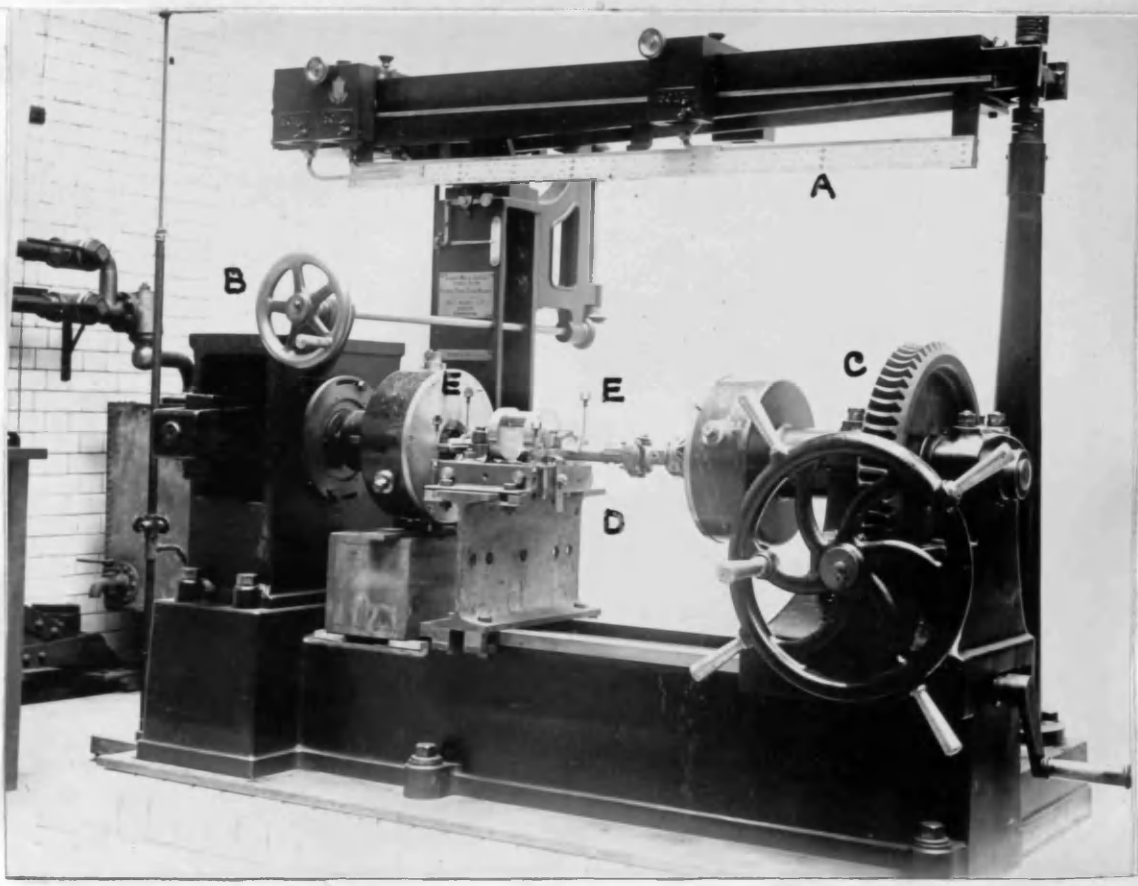


Fig. 5 Crankshaft Lay-out for Out-of Bearings Static Torsion Tests.

Fig. 6. Torsion Machine and Plant Lay-out for Static Torsion Tests of Crankshaft Models.



A. Three graduated scales to measure torques of ½ in.-lb., 1 in.-lb., and 5 in.-lb., respectively.

B. Hand-wheel to operate poises.

C. Carriage with worm gear by which the torque is applied.

D. Steel beam with cast iron sole-plate rigidly fixed to bed of machine.

E. Mirrors for measuring the angular deflections of shafts.

By this means experimental measurements of the free lateral movement of the journals was made possible.

The chuck at the weighing end of this machine is mounted upon a steel shaft of special cruciform section which is supported by hardened steel rollers within a sleeve. This sleeve, in turn, is carried by ball bearings fitted into a short column. This construction permits of complete axial freedom of the shaft. A pure torsion test is thus ensured. A general view of the torsion machine and plant lay-out is given in Fig.6.

(II) Measurement of Angular Distortion: Adjustable mirrors of a Marten's mirror extensometer were attached at any two gauge points so that they were in the vertical plane passing through the axis of the crankshaft. The rotational movement at each plane was observed by means of two telescopes and two curved scales.

The torque applied was increased by regular increments to a maximum value of 520 lb.in., 1800 lb.in., 3400 lb.in., and 4800 lb.in., for shafts A, B, C, and D, respectively.

Torque-deflection readings were taken with the mirrors clamped, in turn, at the gauge points E-K, E-J, E-H, E-G, and E-F. By this means the angular distortion of each component part was found by the method of difference.

The readings were repeated and checked in each case. The average overall angle of twist and the average angle of twist induced in each component part, for any particular torque, was determined from a series of torque-deflection diagrams.

Typical examples of such diagrams for shafts A, B, C, and D are shown in Figs.7, 8, 9, and 10 respectively.

The angular distortion in radians, for any particular torque, is given by the difference in the scale readings divided by twice the distance from the mirror to the scale.

(e)/

FIG. 7 TORQUE - DEFLECTION DIAGRAMS FOR CRANKSHAFT MODEL A.

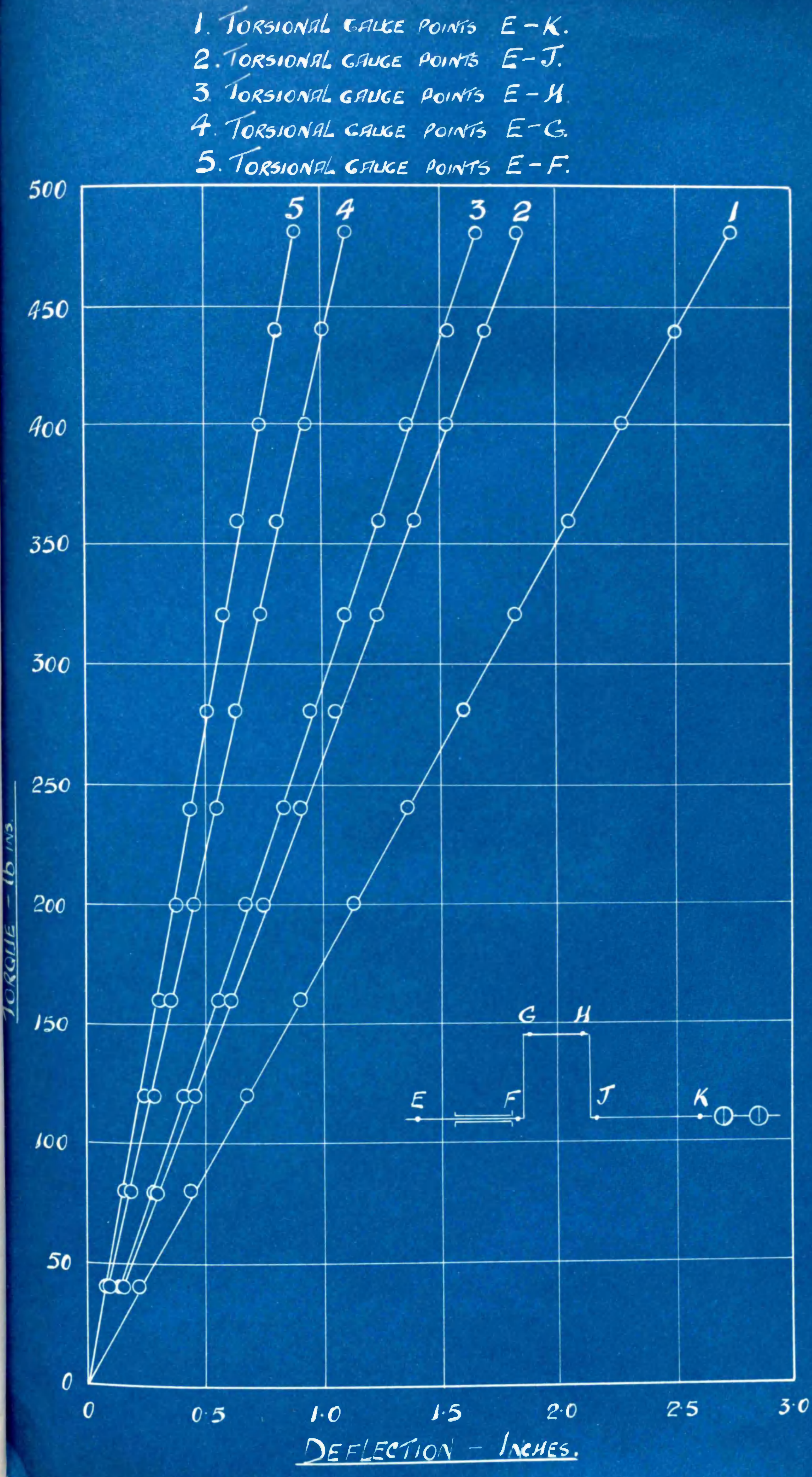


FIG. 8 TORQUE - DEFLECTION DIAGRAMS FOR CRANKSHAFT MODEL B.

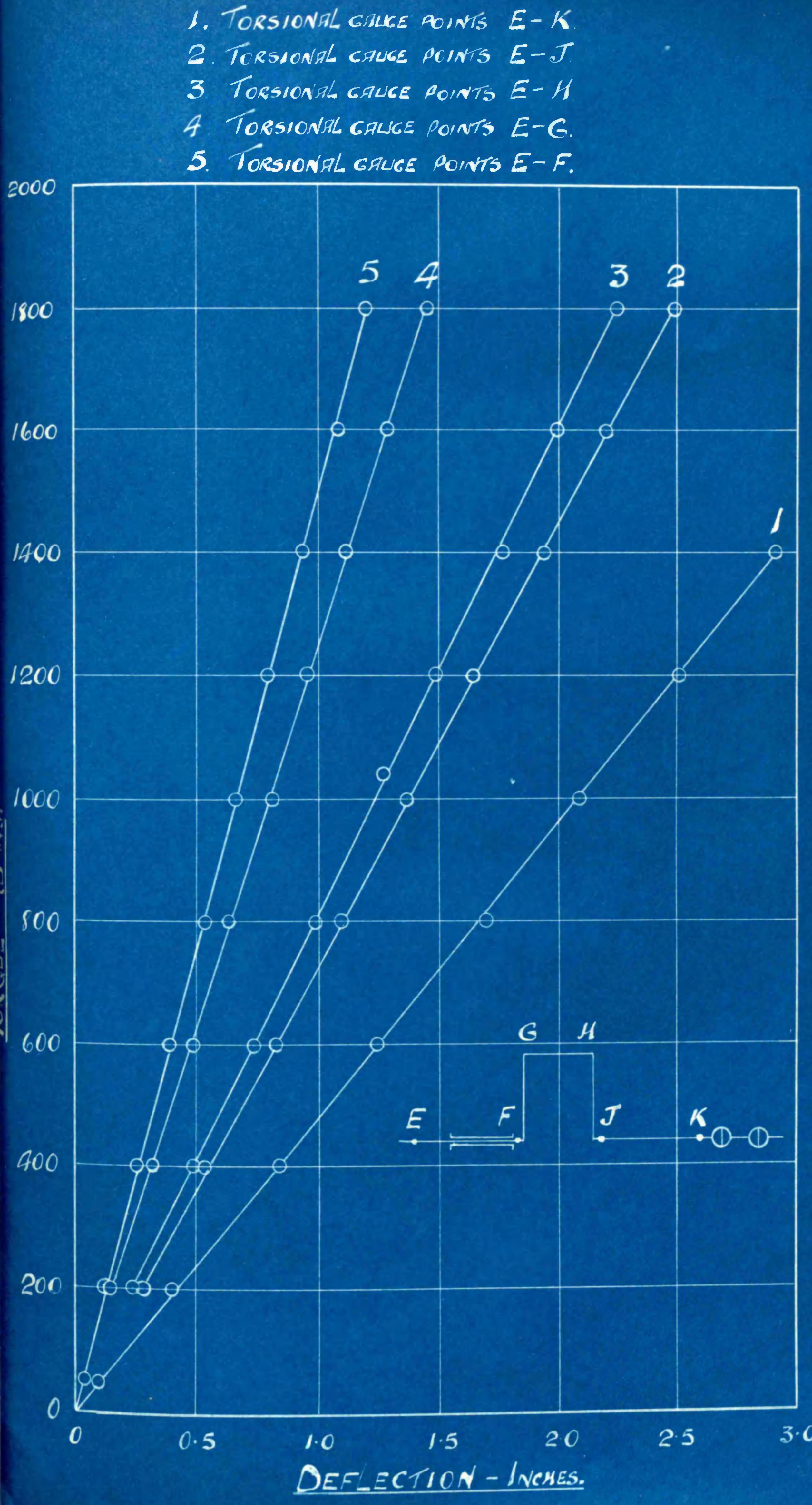


FIG. 9 TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT MODEL C.

1. Torsional gauge points E–K.
2. Torsional gauge points E–J
3. Torsional gauge points E–H
4. Torsional gauge points E–G
5. Torsional gauge points E–F.

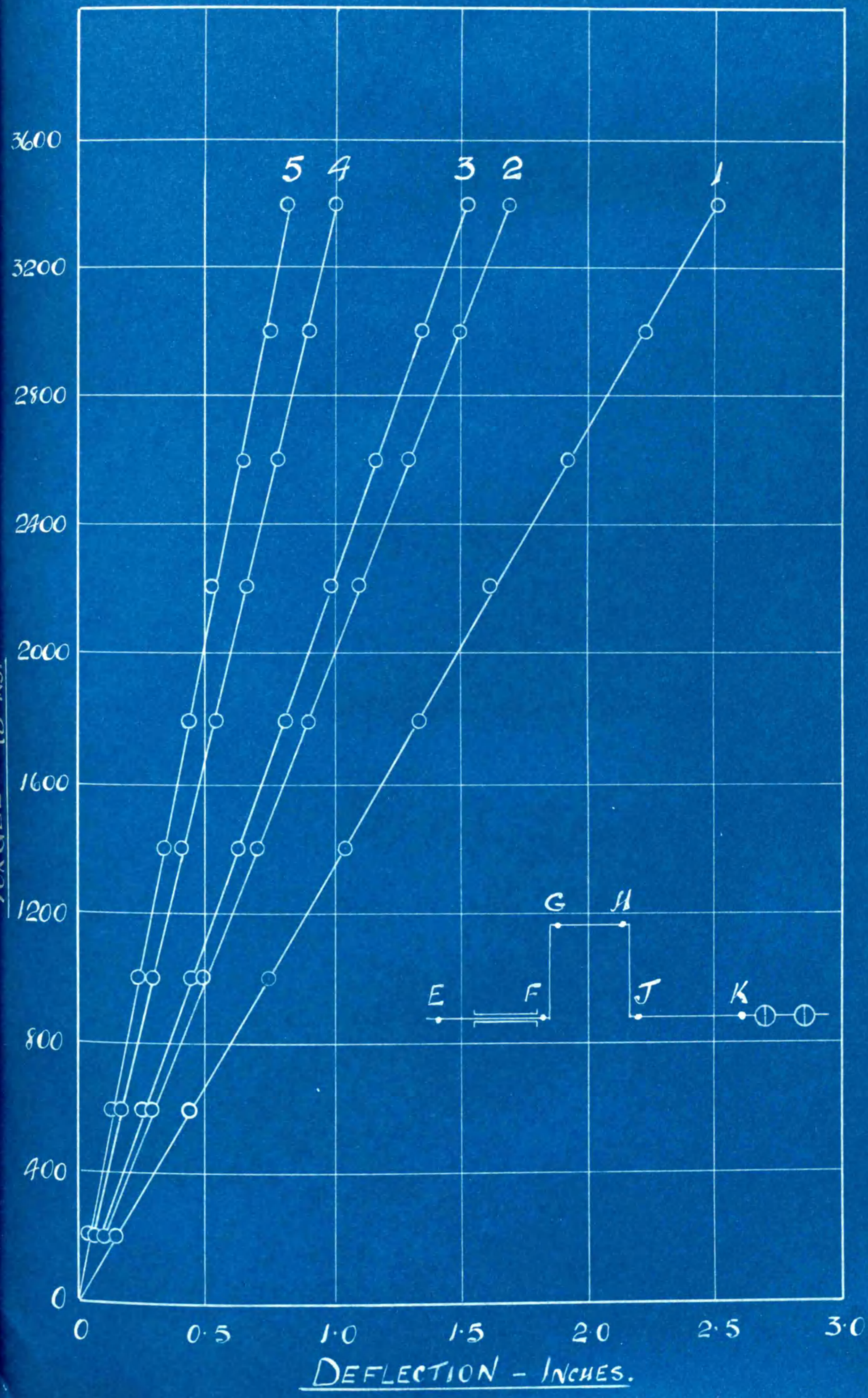
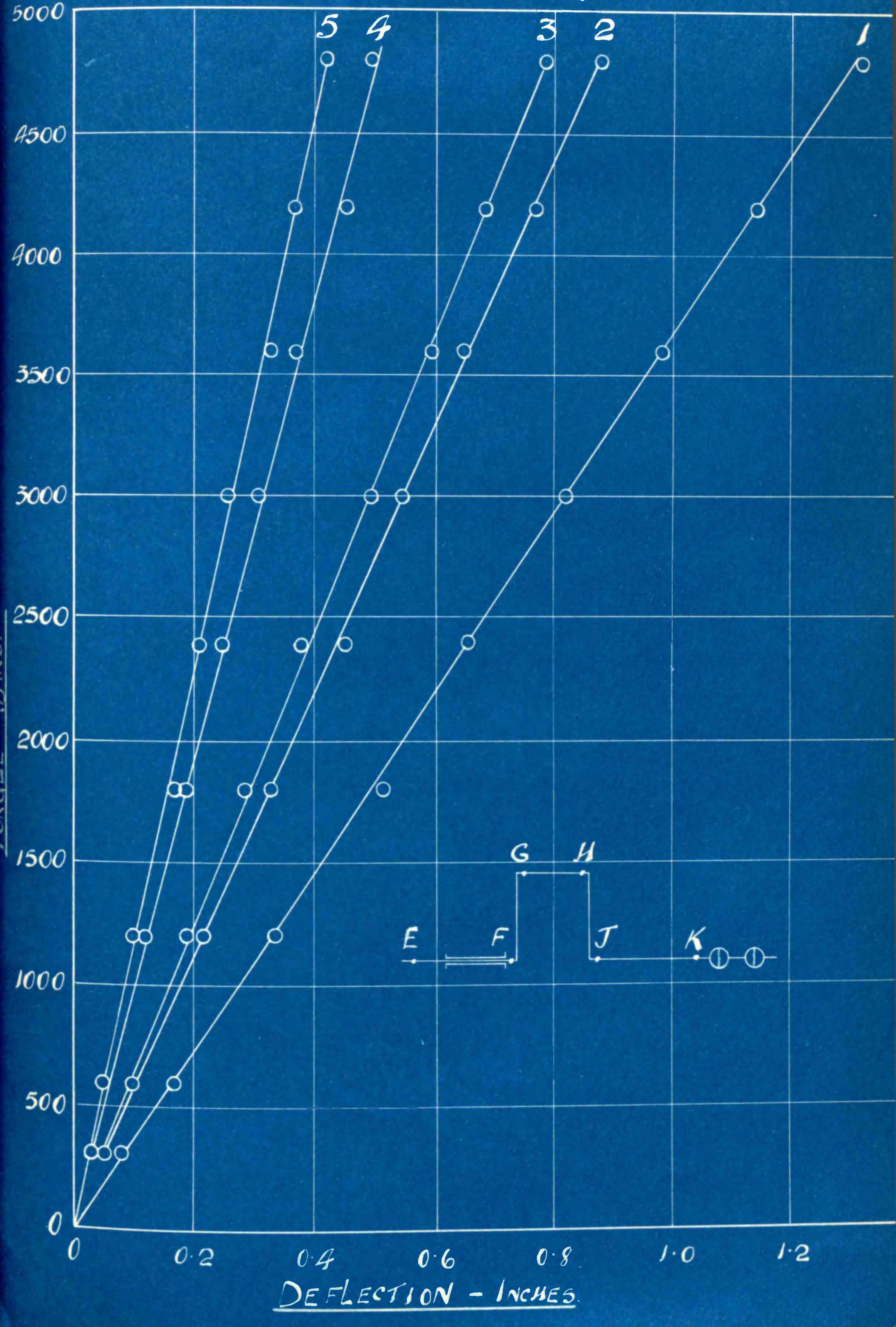


FIG. 10 TORQUE - DEFLECTION DIAGRAMS FOR CRANKSHAFT MODEL D.

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS E-J.
3. TORSIONAL GAUGE POINTS E-H.
4. TORSIONAL GAUGE POINTS E-G.
5. TORSIONAL GAUGE POINTS E-F.



(e) Experimental Results

The torsional strains induced in the shafts A, B, C, and D due to an applied torque of 250 lb.in., 1000 lb.in., 1000 lb.in. and 3000 lb.in., respectively, are given in Table 4 . The values tabulated are in radians multiplied by 10^3.

TABLE 4. - Angular Distortion of Crankshaft Models.

Shaft	Torque lb.in.	Twist in radians x 10^3					
		E-F	F-G	G-H	H-J	J-K	E-K
A	250	2.164	0.4397	1.389	0.4397	2.164	6.597
B	1000	3.171	0.6134	2.06	0.6134	3.171	9.629
C	1000	1.134	0.2199	0.7406	0.2199	1.134	3.449
D	3000	1.25	0.2431	0.81	0.2431	1.25	3.796

TABLE 5. - Calculated Angular Distortion of Journals and Pin.

Shaft	Torque lb.in.	Twist in Radians x 10^3	
		Journals	Crankpin
A	250	2.101	1.361
B	1000	3.185	2.065
C	1000	1.141	0.7408
D	3000	1.264	0.8202

The calculated value of the angle of twist of those parts subjected to pure torsion only is given by:-

$$= \frac{M}{G\,J}$$

where M = Torque applied.
= Distance between specified gauge points on component part.
G = Modulus of rigidity
J = Polar moment of inertia of the cross-section.

The angular distortion of the crankpin and the journals is given in Table 5. The value of the modulus of rigidity of each shaft material, as determined by experiment, was used in each case.

The angle of twist of each component part is given as a percentage of the overall angle of twist in Table .

TABLE 6 . Percentage Angular Distortion of each Component Part.

Shaft	Percentage of Total Angular Distortion				
	E-F	F-G	G-H	H-J	J-K
A	32.81	6.665	21.06	6.665	32.81
B	32.94	6.369	21.39	6.369	32.94
C	32.88	6.375	21.47	6.375	32.88
D	32.94	6.403	21.34	6.403	32.94

CONCLUSIONS

These tests were carried out to examine and compare the effect of scale. As the scale is reduced inaccuracies become more serious. Any errors in the model forms will be associated with the results of machine work and also with the fixing of torsional gauge lengths in an attempt to establish geometrical similarity. Although it was possible to obtain a very close approach to geometrical similarity in most parts of the models there were certain measurements which could not be determined with a high degree of accuracy. It was difficult, for instance, to obtain a high degree of geometrical similarity in the lengths of the crank throws by machining operations carried out in a general purpose lathe. There was a tendency also, due to "spring" in the job, of out-of-roundness conditions being generated during certain machining operations.

The/

The angular distortion of each crankshaft element, given in Table 4 . enables the effect of any such inaccuracies to be compared and assessed. The experimental and calculated angular distortion of journals and pin are in close agreement.
The angular distortion of each part relative to the overall shaft distortion is given in Table 6 .
A very close similarity of effect is demonstrated in each shaft model. It is evident that, with average engineering skill, models of continuous structural form may be made to any scale with success, and that by such means the torsional characteristics of a crankshaft could be examined, being both economical and reliable.

The test results indicate that any one model could thus be taken to study and evaluate the effect of web form on crankshaft stiffness.

PART II.

INVESTIGATIONS ON THE INFLUENCE
OF
WEB FORM

(a) Selection of Web Forms

Crankshaft model C. was selected to carry out a separate investigation on the effect of crank webs form on the torsional rigidity of a single-throw-two-bearing crankshaft. A change in crank web form between the pin and journal centre lines is effected, in general, by varying the width of the web. Outside these centres a reduction in the width of the web is obtained by chamfering the web ends. The web form in many cases, however, is characterized by a reduction in the web thickness at the journal and pin ends by different degrees of bevelling. Seven different types of crank web forms, as shown in Fig.11 were examined.

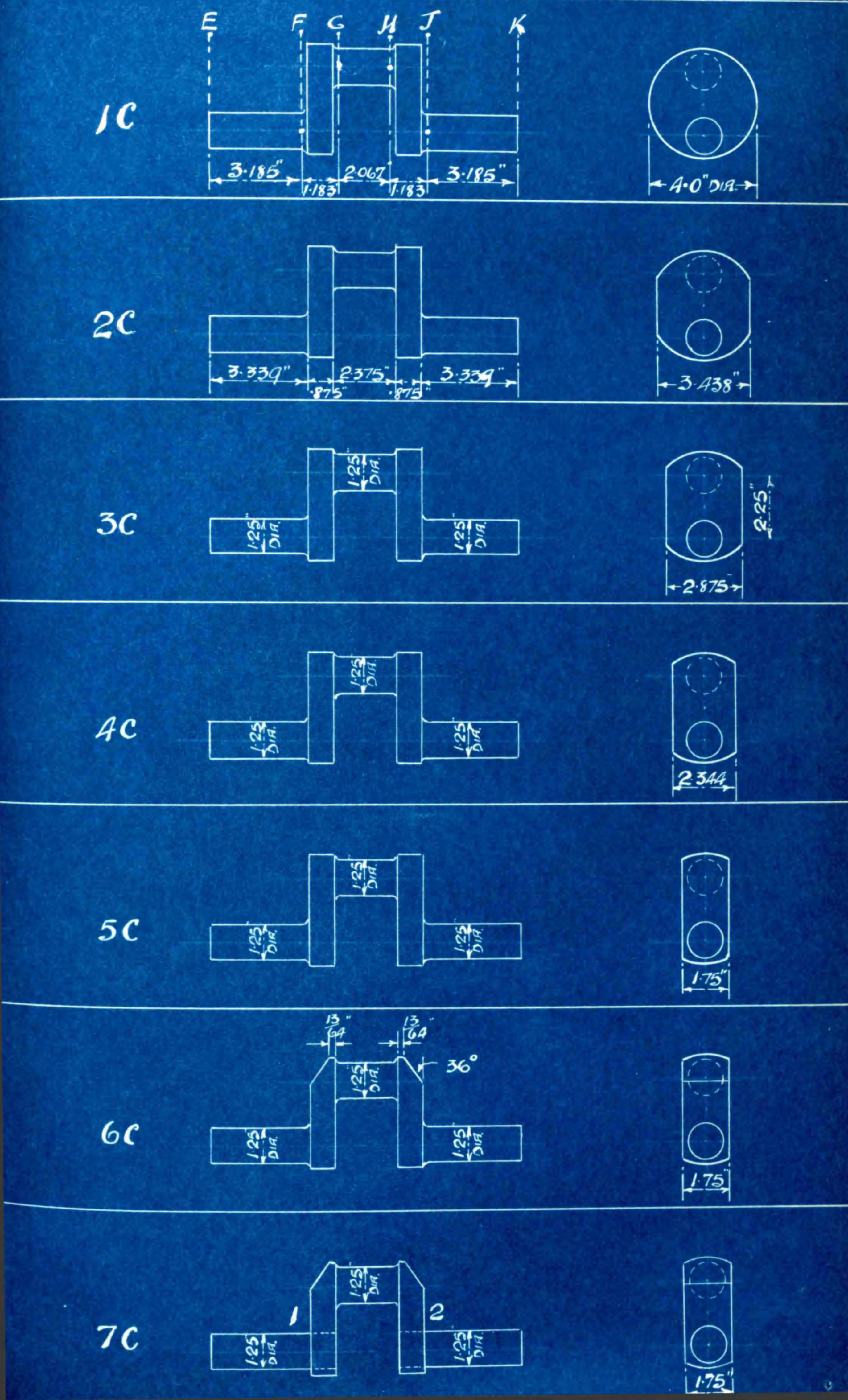
The disc webs of crankshaft model C., designated shaft 1C, were successively planed down in width to dimensions shown in shafts 2C, 3C, 4C, and 5C. The webs at the crank pin end of shaft 5C were afterwards reduced in thickness, by bevelling, to the form shown on shaft 6C.
Shaft 7C took the form of partly built-up crankshaft, the webs being fitted on the journal shafts. The journals of shaft 6C were machined off at the crank webs. The journal end of the webs were then bored to dimensions giving a difference in the free diameters of the mating elements, web and journal shaft, suitable for making a shrink fit assembly. The webs were heated by a gas burner.

The assembly of web and journal shaft 1 was made with the mating surfaces of the elements perfectly dry and free from film. The assembly of web and journal shaft 2 on the other hand was made with a film of rape oil on the surface of the solid element. This assembly was afterwards fitted with a security taper pin passing through the web and journal shaft.

(b) Experimental Procedure.
The shaft was subjected, in each case, to static torsional tests in and out of bearings.

FIG. 11 CRANK WEB FORMS OF SHAFT SUBJECTED TO STATIC TORSION TESTS.

OUT OF BEARINGS AND IN BEARINGS.



(I) Crankshaft Lay-out and Measurement of Angular Distortion: The crankshaft lay-out adopted for carrying out these tests is shown in Fig. 12. A general view of the actual set-up in the machine with the necessary mirror attachments for an in-bearings static torsion test is given in Fig. 13. The positional setting of the mirrors for measuring the overall angular distortion of the crank is clearly shown. The roller optical gauges N, fixed to rigid angle-iron supports, were fitted to measure the degree of bearing constraint or amount of journal clearance.

An attempt was made in the in-bearing tests to obtain complete lateral constraint of the journals. The brasses were carefully bedded and the bearings fixed to rigid supports so as to prevent any movement of the journals other than one of rotation about the axis of the shaft.

The torque applied was increased by regular increments of 400 lb.ins. to a maximum value of 3,400 lb.ins. in each case. Mirror readings were taken at each of the gauge points when the shaft was tested in bearings. By this means the angular distortion of each component part was determined. When tested out of bearings readings of the overall distortion of the shaft and the overall distortion of the crank only were taken. These were obtained by taking mirror readings over the torsional gauge points E-K and F-J respectively. To assist, however, in the subsequent analysis, readings were taken at each of the gauge points on shaft 1C when twisted out of bearings.

Measurements of the lateral displacement of the journals, when out of bearings and in bearings, were taken by two optical gauges and, as a check, by readings from two Ame's dials. In one case an attempt was made to measure the bearing reactions induced when a pure torque of increasing magnitude was applied to the shaft.

FIG. 12 CRANKSHAFT LAY-OUT FOR IN-BEARINGS STATIC TORSION TESTS.

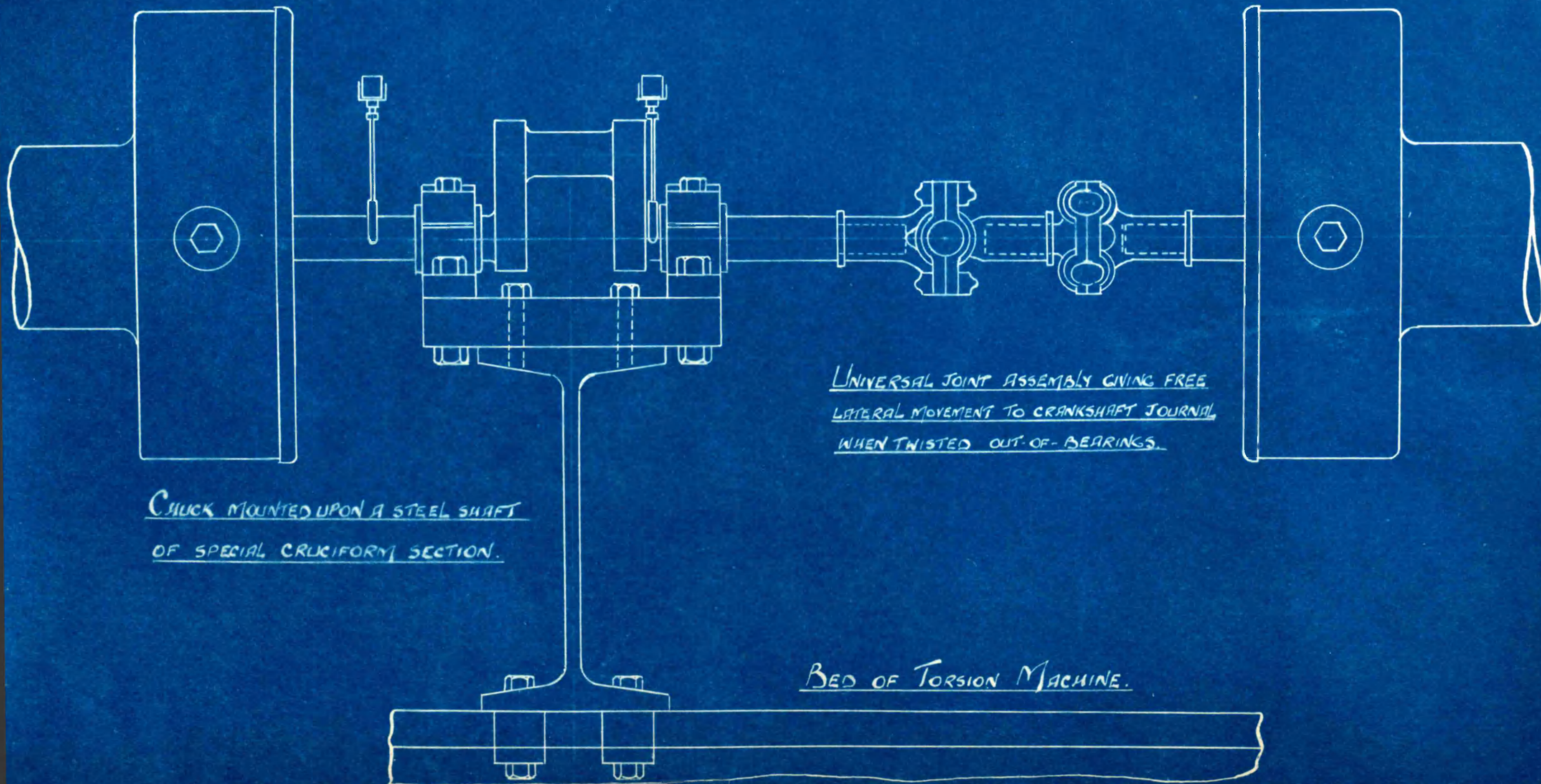
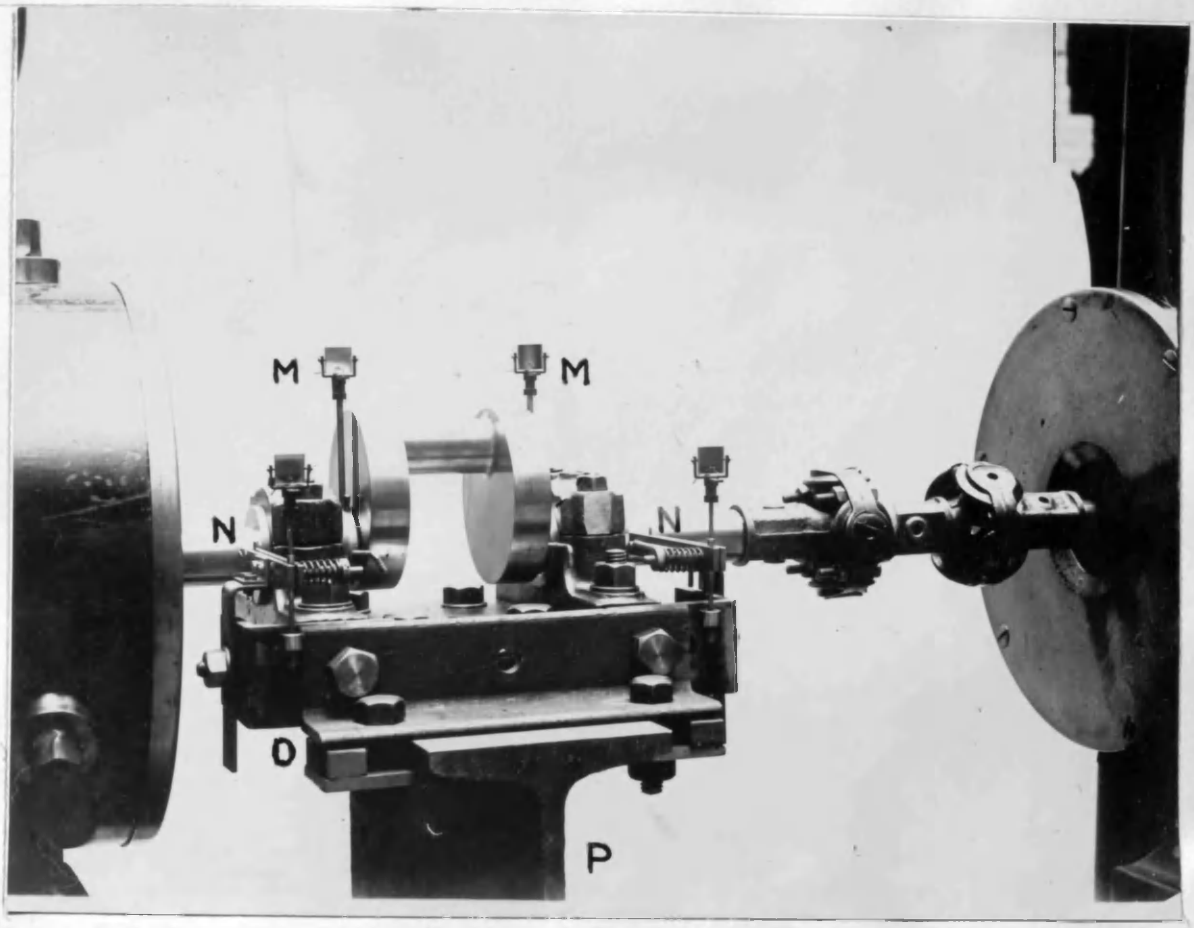


Fig.13. General View of Shaft Lay-out with Mirror Attachments for In-Bearings Static Torsion Tests.



M M. Positional setting of mirrors for measuring overall distortion of crank.

N N. Roller optical gauges for measuring the degree of bearing constraint or journal clearance.

O. Rigid angle-iron supports for optical gauges.

P. Steel beam fixed to bed of torsion machine.

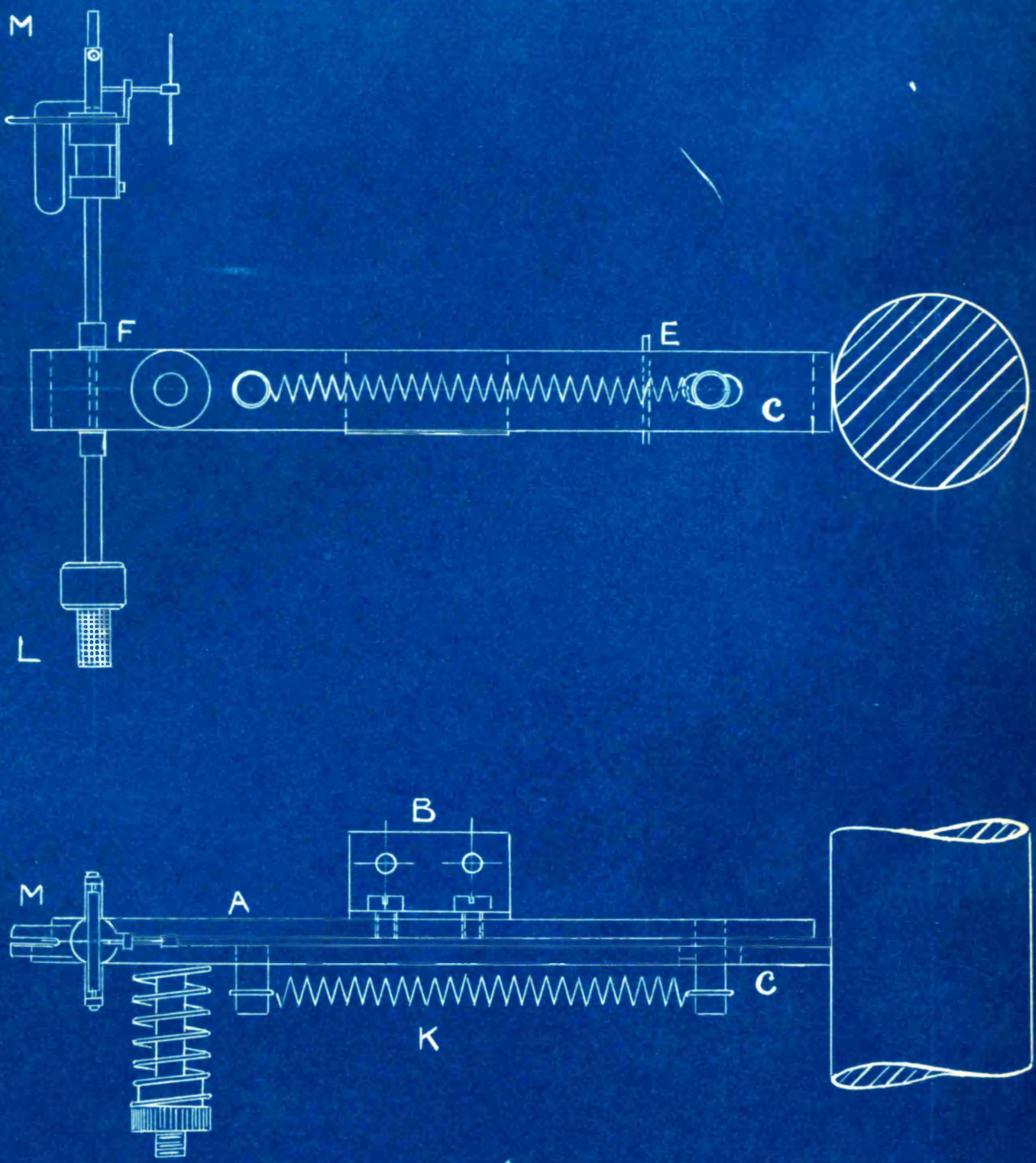
(II) Measurement of Crankshaft Journal Displacement: Measurement of the crankshaft journal displacement when subjected to out-of-bearings and in-bearings static torsion tests was made possible by the shaft lay-out adopted in carrying out the tests. The universal joint assembly, introduced between the shaft and the straining chuck of the torsion machine, gave complete freedom to any lateral movement of the journals induced by the applied torque.

Several optical devices were constructed by the Author for making these measurements. The design of each consisted essentially of the elements of a roller extensometer. The earlier models, consisting of a pair of brass strips of rectangular section separated by two small rollers, were designed to operate between the crank webs. Each strip had a projecting cylindrical piece near one end fitted with a screwed element with a conical point. These engaged with light centre punch marks on the inner faces of the crank webs concentric with the shaft axis.

The results of preliminary tests gave evidence that this method of fixing the optical gauge was unsatisfactory.
The deflection of the adjustable elements centred in the punch marks introduced errors into the readings observed.
It was also found that, within reasonable limits, the degree of tightness in centring the instrument influenced the readings obtained. There was evidence, also, that any side movement of the free end of the brass strips, which rested on rollers, gave a rotational movement to the mirror which was not related in any way to a journal displacement.
To overcome this, an improvement in design was effected by guiding the free end of the gauge between steel ball guides.
The results of further tests showed, however, that the readings obtained by fixing the gauge between the crank webs on point supports were unreliable.

It/

FIG. 14 OPTICAL ARRANGEMENT FOR MEASURING JOURNAL LATERAL DISPLACEMENT.



It was decided, therefore, to develop a method of measurement in which the relative movement of the parallel strips would be controlled by the direct end contact of one of the strips with the journal.

Tests carried out revealed that a direct line contact of journal, with movable strip, together with the fixing of the stationary strip to rigid supports gave displacement readings of the journals which were unaffected by any slight axial movement of the shaft or deflection of the measuring elements. This method, however, necessitated the use of two optical gauges the principal features of which are shown in Fig. .

Two brass strips, $\frac{1}{2}$in. deep by $\frac{1}{8}$in. thick, separated by two small rollers ·06166 in. dia., are lightly clamped by spring pressure. Strip A is secured by the angle section B to a rigid angle iron support fixed to the steel beam. One end of the strip C bears against the journal making a line contact under pressure induced by the spring K.

Any lateral movement of the journal is transmitted to the strip C which rotates the rollers E and F. An adjustable mirror M of a Marten's mirror extensometer is suitably fitted to one end of roller F which has its axis extended in both directions. A small knurled head L, at the other end, may be used for making any adjustments.

The rotational movements of the mirrors were observed by means of two telescopes and two curved horizontal scales.

The relative displacement of the journals, for any particular torque, is given by:

$$e = \frac{dx}{2D}$$

where d = diameter of roller in inches.

x = difference in scale readings.

D = distance from mirror to scale.

(III)/

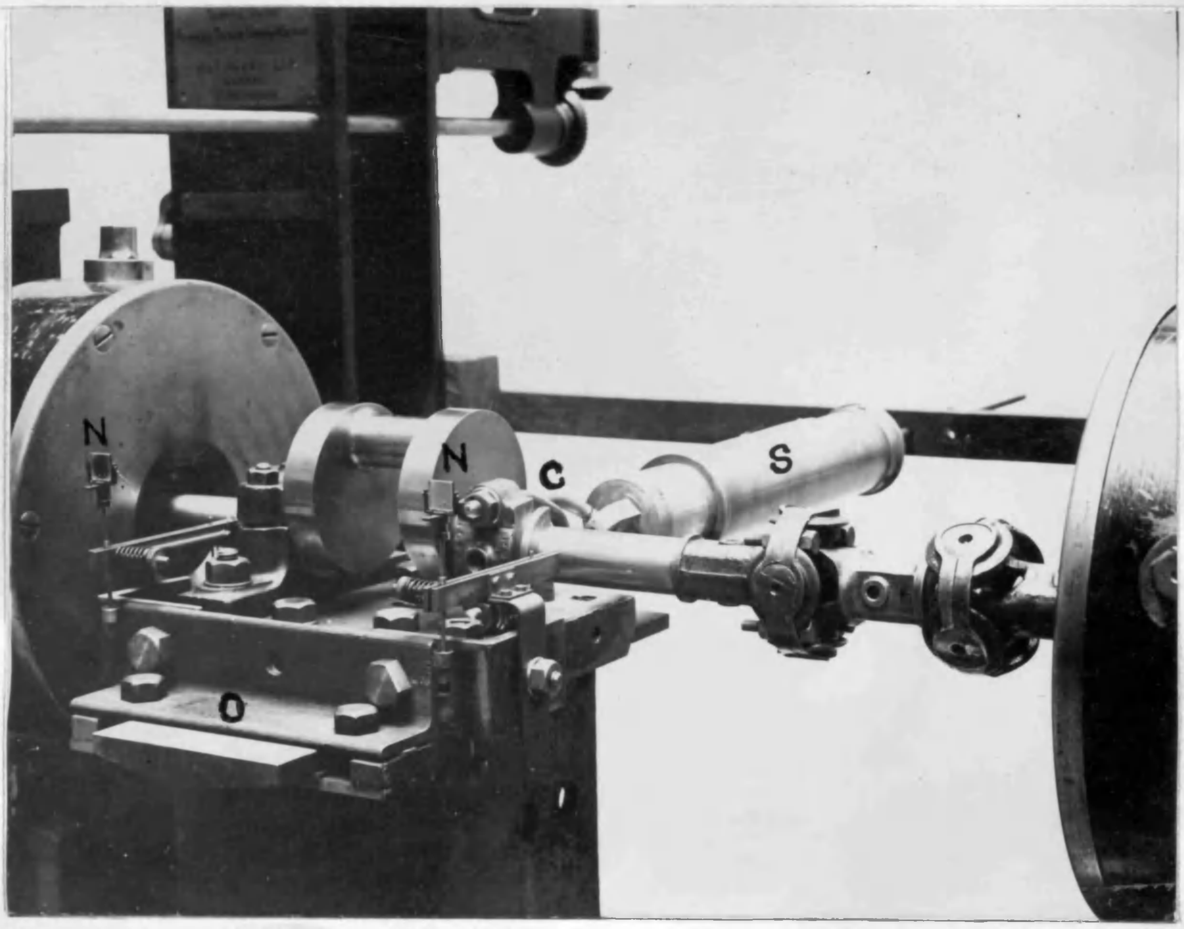
(III) Measurement of Bearing Reactions: When a pure torque of increasing magnitude is applied to a crankshaft, in bearings, the crank webs tend to deflect out of the original crank plane. This tendency can only be partly prevented by the bearings even when the journal constraint is complete. The lateral displacement of the journals, however, may be either partly or wholly prevented by the bearings depending on the clearance and the degree of rigidity of the bearing supports.

Except for the case, therefore, of bearings which have so much play that no bearings reaction arises, restraint of the journal movements will induce a bearing pressure and a bearing straining moment at each journal.
The torsional strains induced in the shaft will then be due to the applied torque and to the bearings pressures and moments which act through the axis of the journals perpendicular to the plane of the throw. The torsional effects caused by the bearings reactions will be opposite to those induced by the applied torque.

If these reactions are known, then, the total in-bearings angle of twist may be readily determined on the basis of the out-of-bearings strains induced in the separate elements of the crankshaft.
An expression for calculating the value of the bearings pressures, however, involves a lengthy analysis of all the straining actions induced in the shaft. Moreover, since some simplifying assumptions are made in this analysis it was thought that an attempt to determine the bearing reactions experimentally might provide data which would justify the assumptions made and the method of analysis.

The journal, fixed in the chuck at the weighing end of the machine, shown in Fig. 15, was constrained directionally by a/

Fig. 15. Lay-out adopted for Measuring the Bearing Reactions of Single Throw Two Bearing Crankshaft.



N N. Roller optical gauges for measuring lateral displacement of journals.

O. Rigid angle-iron supports for optical gauges.

S. Spring balance for measuring magnitude of bearing reaction.

C. Bearing cover and brass fitted with stirrup for transmitting journal restraining force.

a bearing fitted a little clear of the crank web.
This bearing, although exercising a complete lateral constraint, permitted a free rotational movement of the journal.
An optical gauge N was fitted just clear of the outer face of the bearing to measure any possible journal displacement.
The other journal was fitted with the top half of the bearing only and had a stirrup attachment for transmitting the necessary journal restraining force.
Zero mirror readings were taken before applying the shaft torque. The tendency towards a lateral displacement of the "free" journal was corrected by applying the necessary journal restraining force to bring the optical gauge reading back to zero. By this means the journal pressure readings for different torques were obtained.

(c) Experimental Results.

Typical in-bearings and out-of-bearings torque deflection diagrams for shafts 1C to 7C are given in Figs. 16 to 29
The angular distortion, in radians, for any particular torque is given by the difference in the scale readings divided by twice the distance from the mirror to the scale.
The angular deflections of the separate elements of each model form, in bearings, due to an applied torque of 3.600 lb.in. are given in Table 7. These are average values taken from a large number of torque-deflection curves.
The overall angle of twist for each crank and shaft, in-bearings and out-of-bearings, are also given. The values tabulated are the angles of twist, in radians, multiplied by 10^3.

The test stiffness of each crankshaft, out-of-bearings and in-bearings, is given in Table 8. The ratio of the stiffness/

FIG. 16 OUT-OF BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 1C.

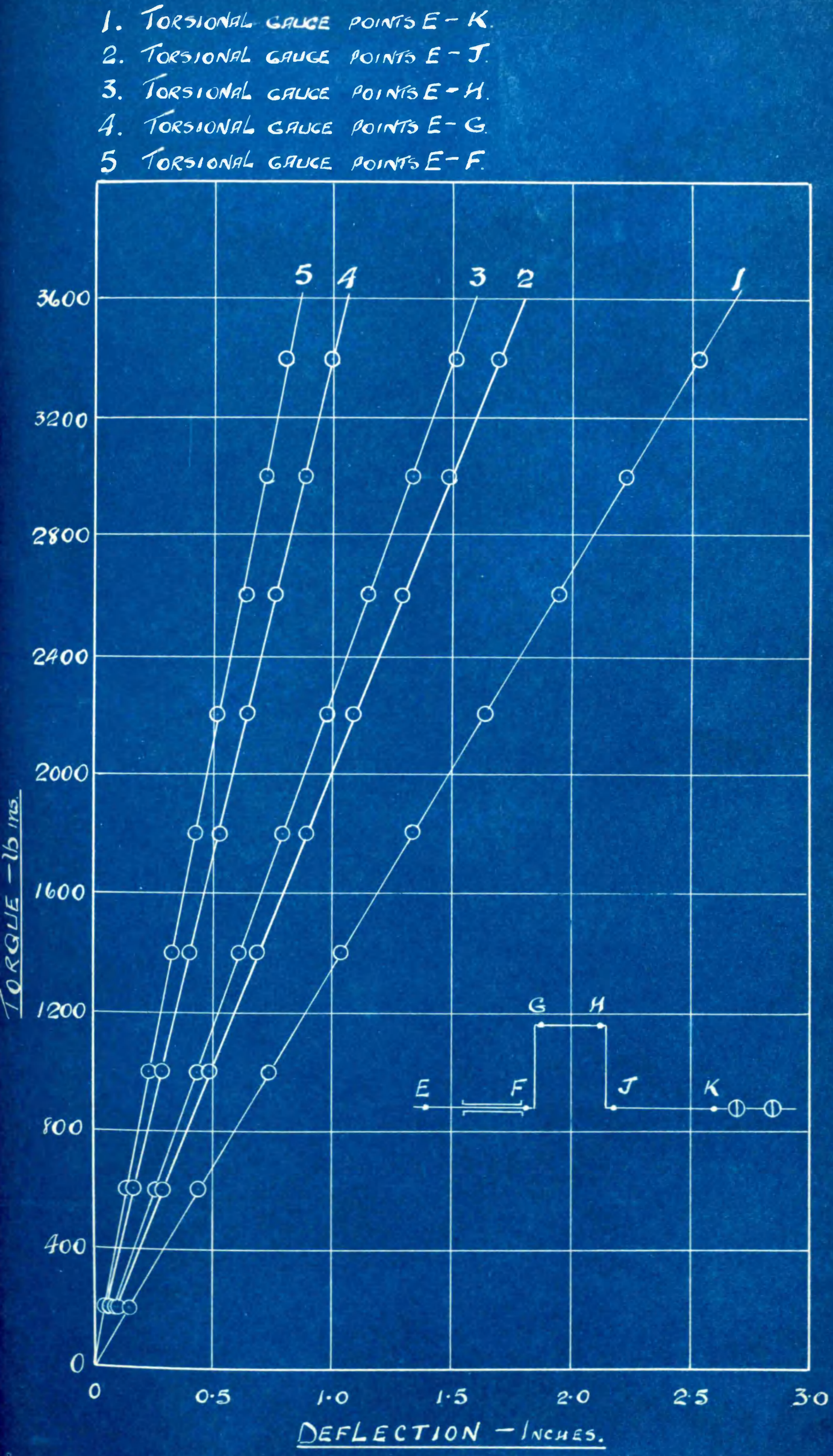


FIG. 17 IN-BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT IC.

1. TORSIONAL GAUGE POINTS E-K.
2 TORSIONAL GAUGE POINTS E-J
3 TORSIONAL GAUGE POINTS E-H
4 TORSIONAL GAUGE POINTS E-G.
5 TORSIONAL GAUGE POINTS E-F

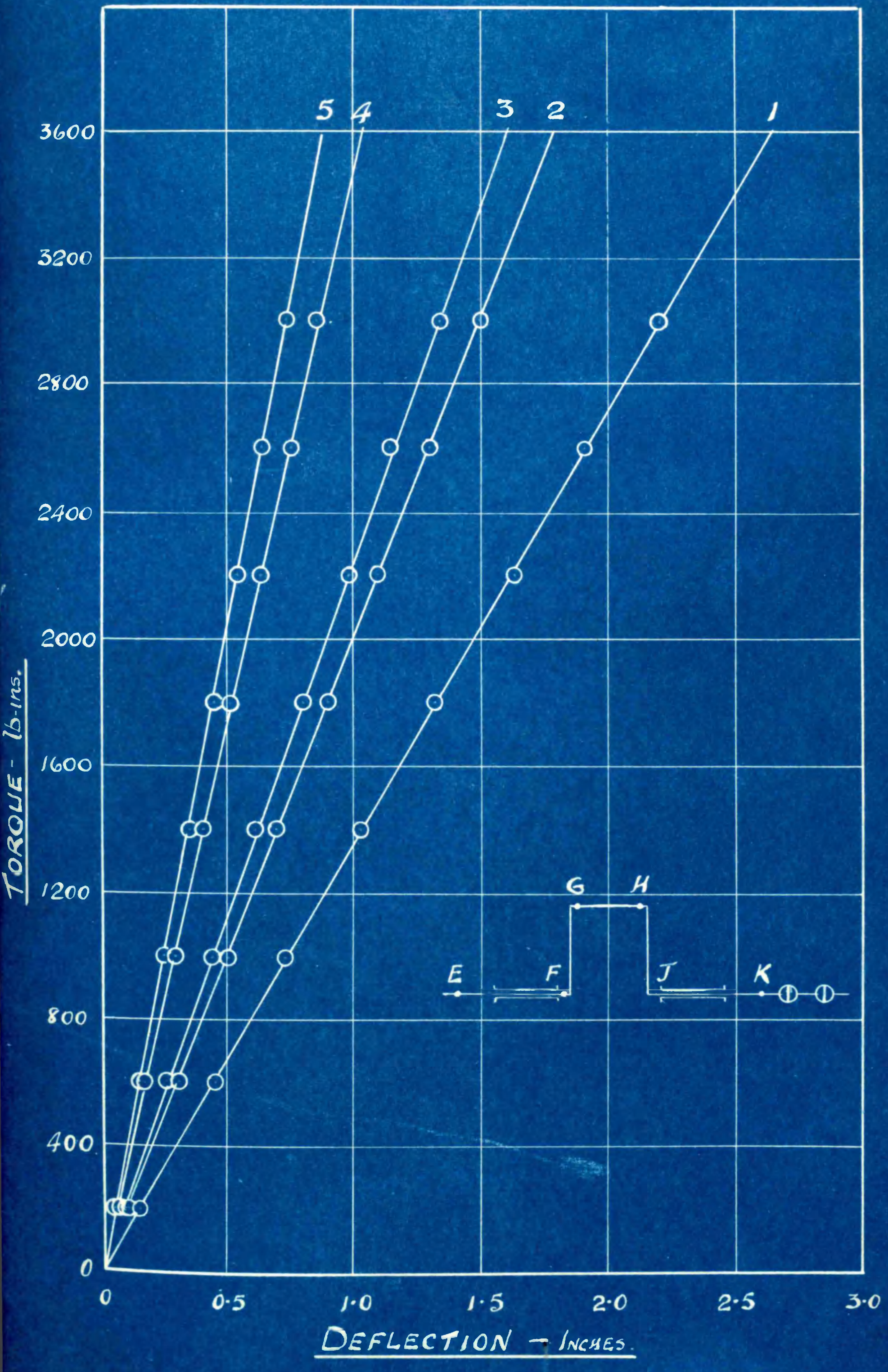


FIG. 18 OUT-OF BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 2C.

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS F-J.

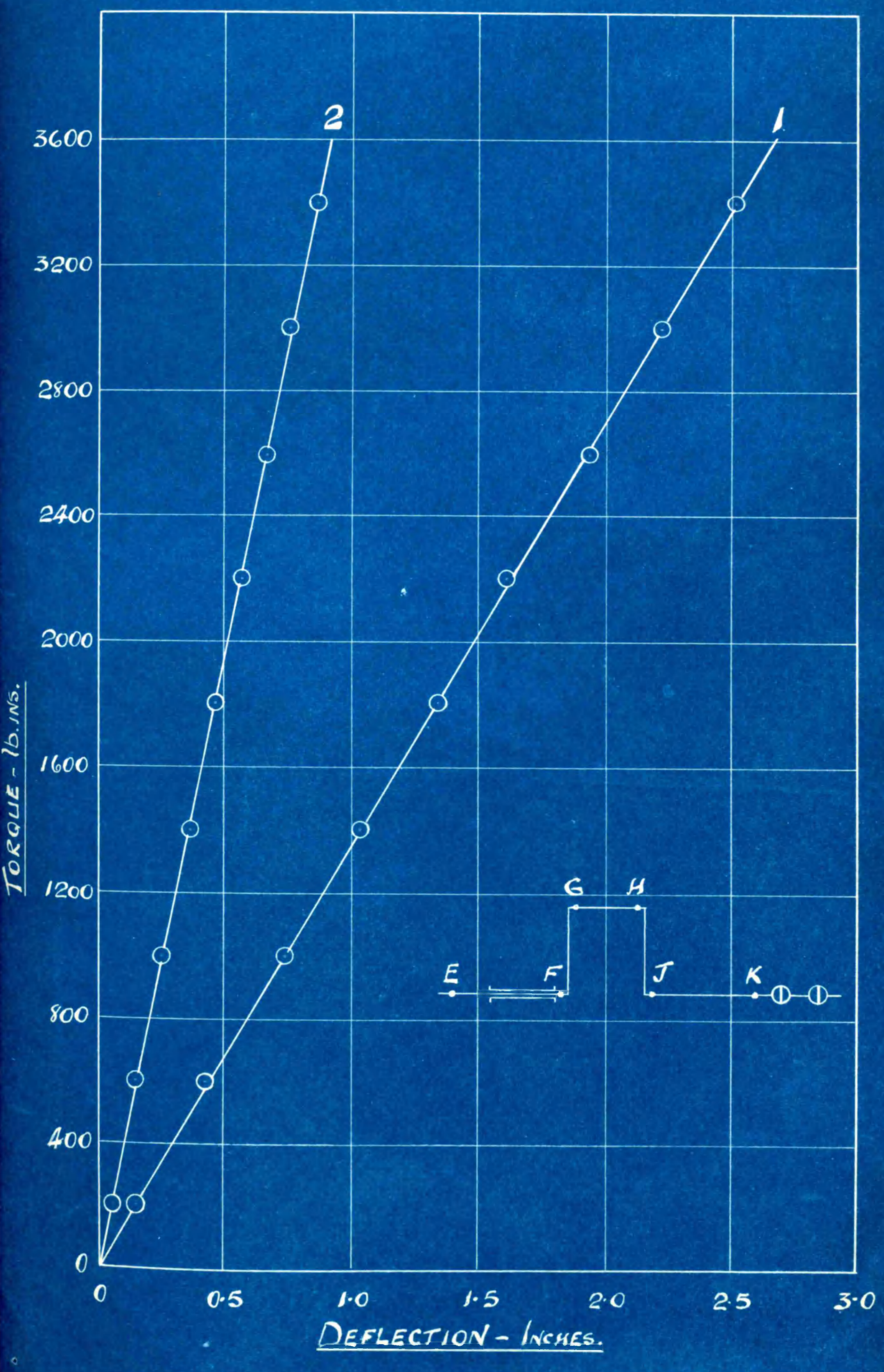


Fig 19 In-Bearings Torque-Deflection Diagrams for Crankshaft 2C.

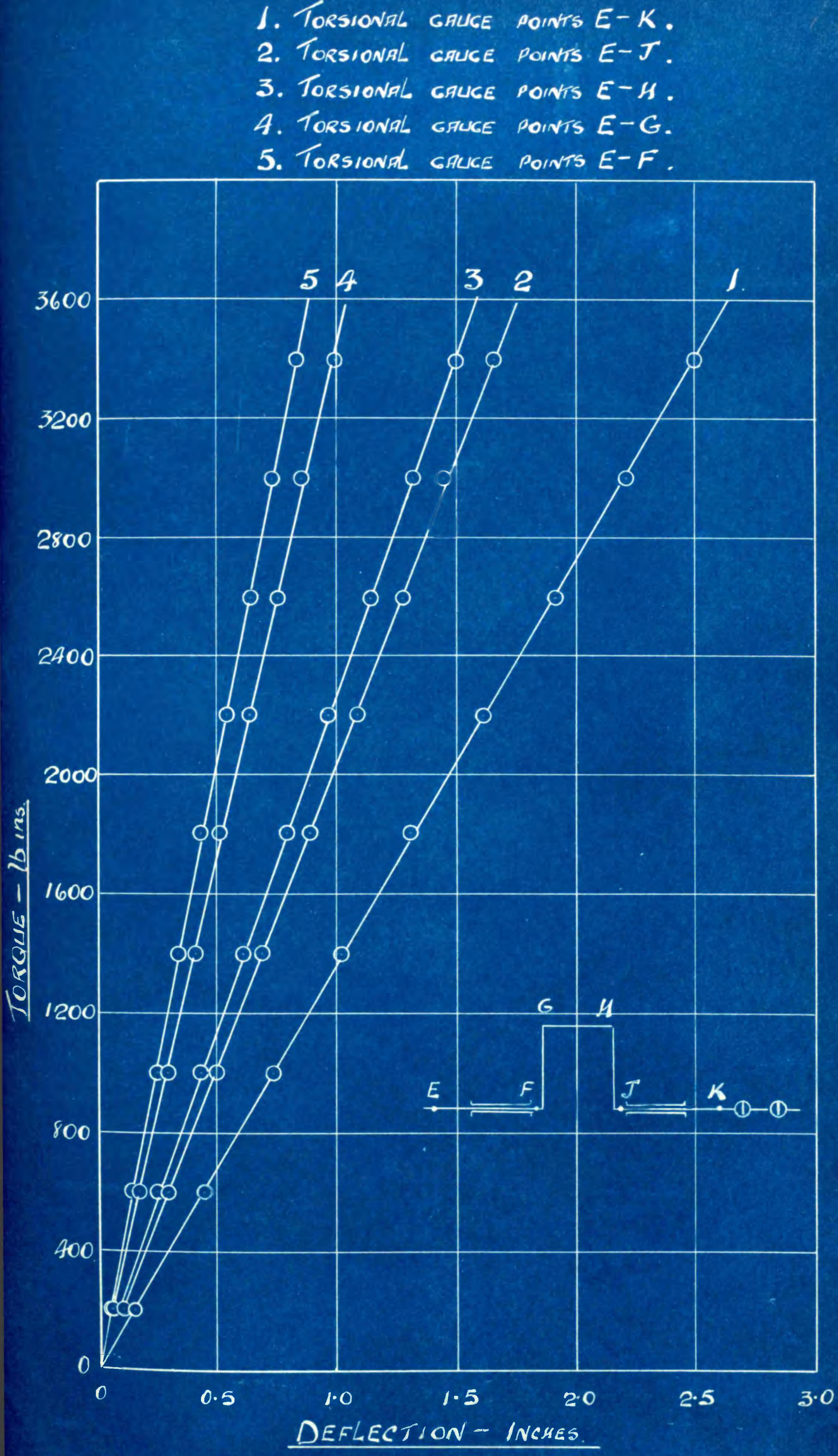


FIG. 20 OUT-OF-BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 3C.

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS F-J.

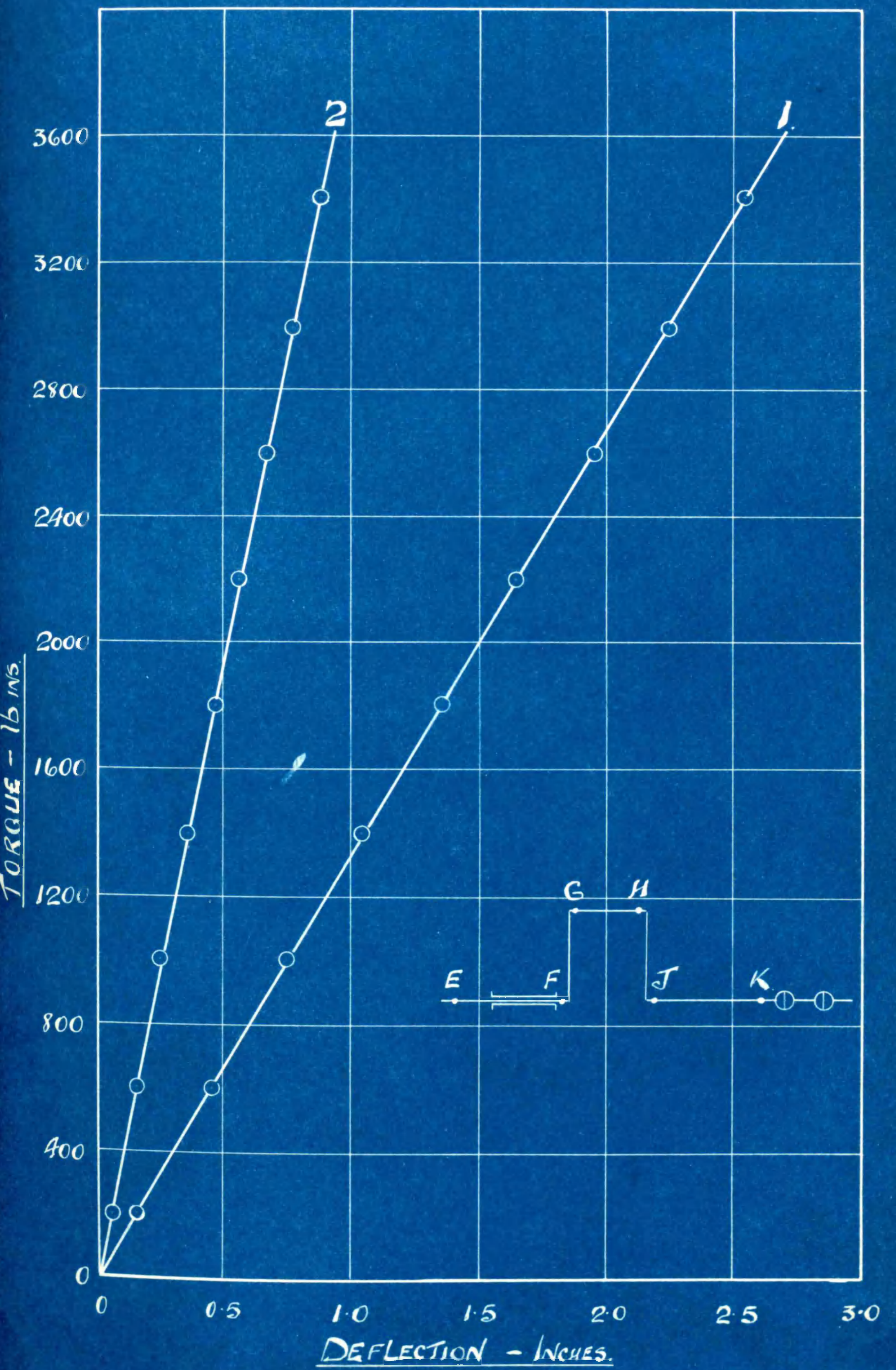


FIG. 21 IN-BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 3C

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS E-J.
3. TORSIONAL GAUGE POINTS E-H.
4. TORSIONAL GAUGE POINTS E-G.
5. TORSIONAL GAUGE POINTS E-F.

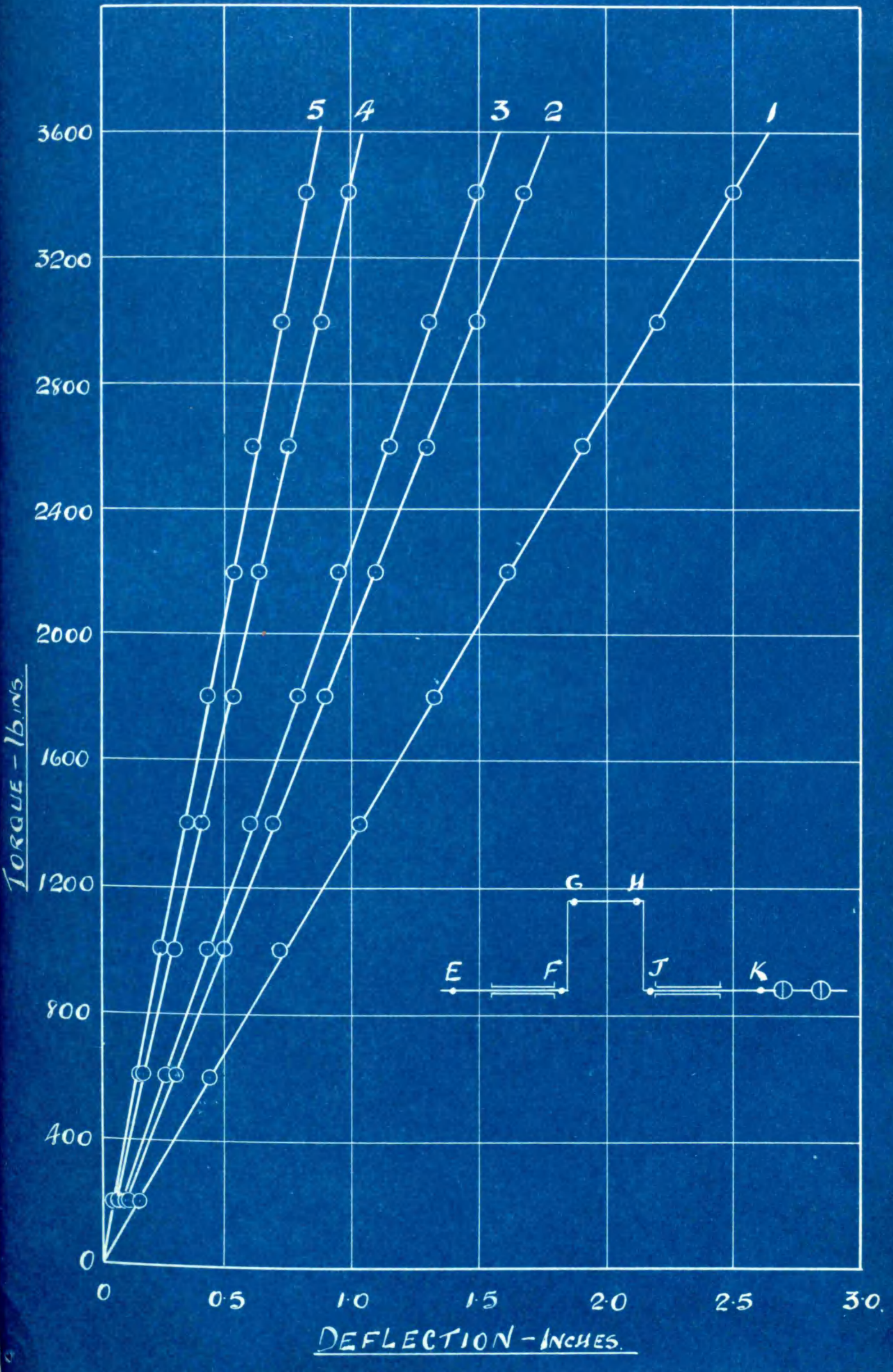


FIG 22 OUT-OF BEARINGS TORQUE DEFLECTION DIAGRAMS FOR CRANKSHAFT 4C

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS F-J.

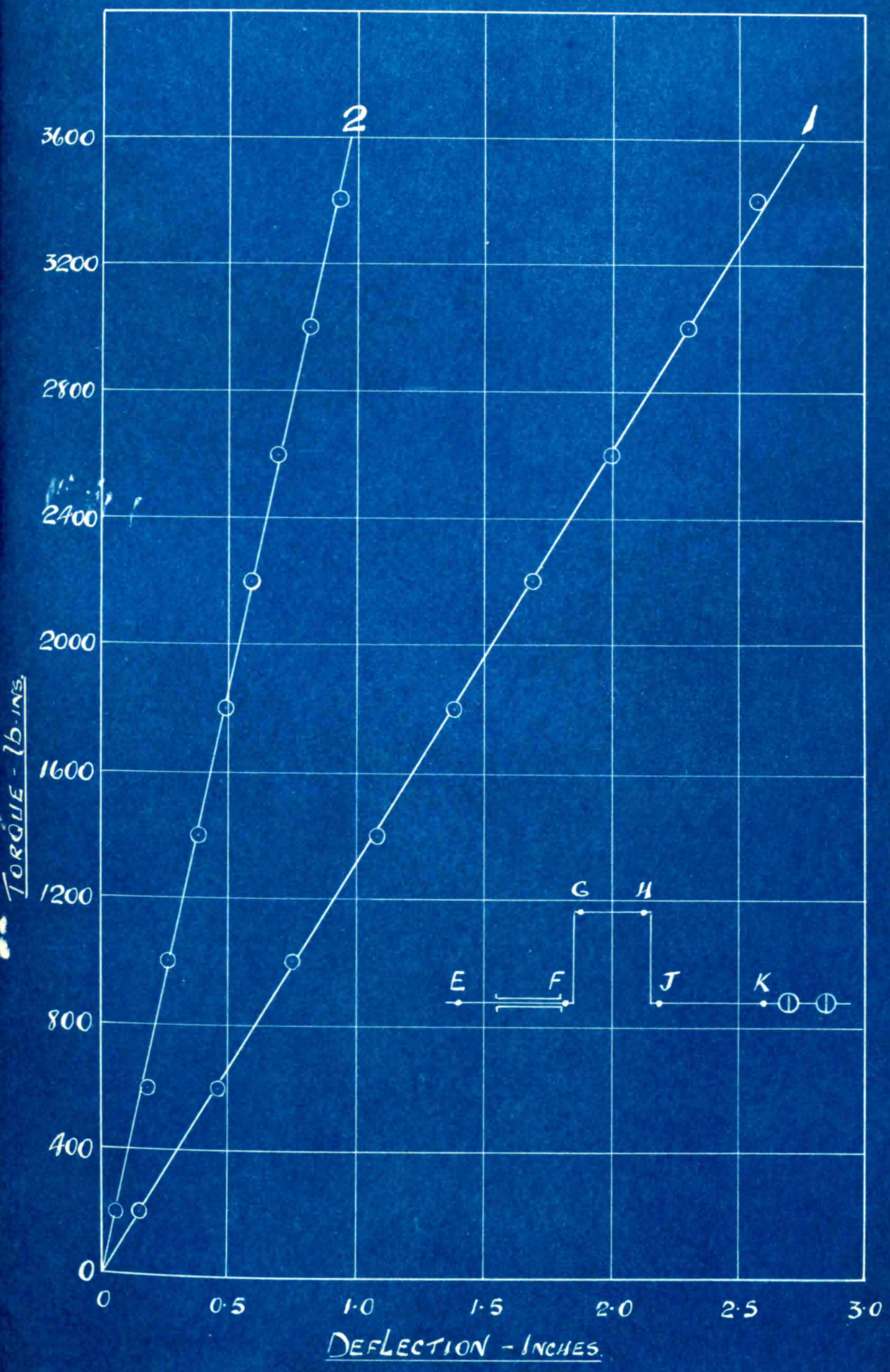


FIG 23 IN-BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 4C

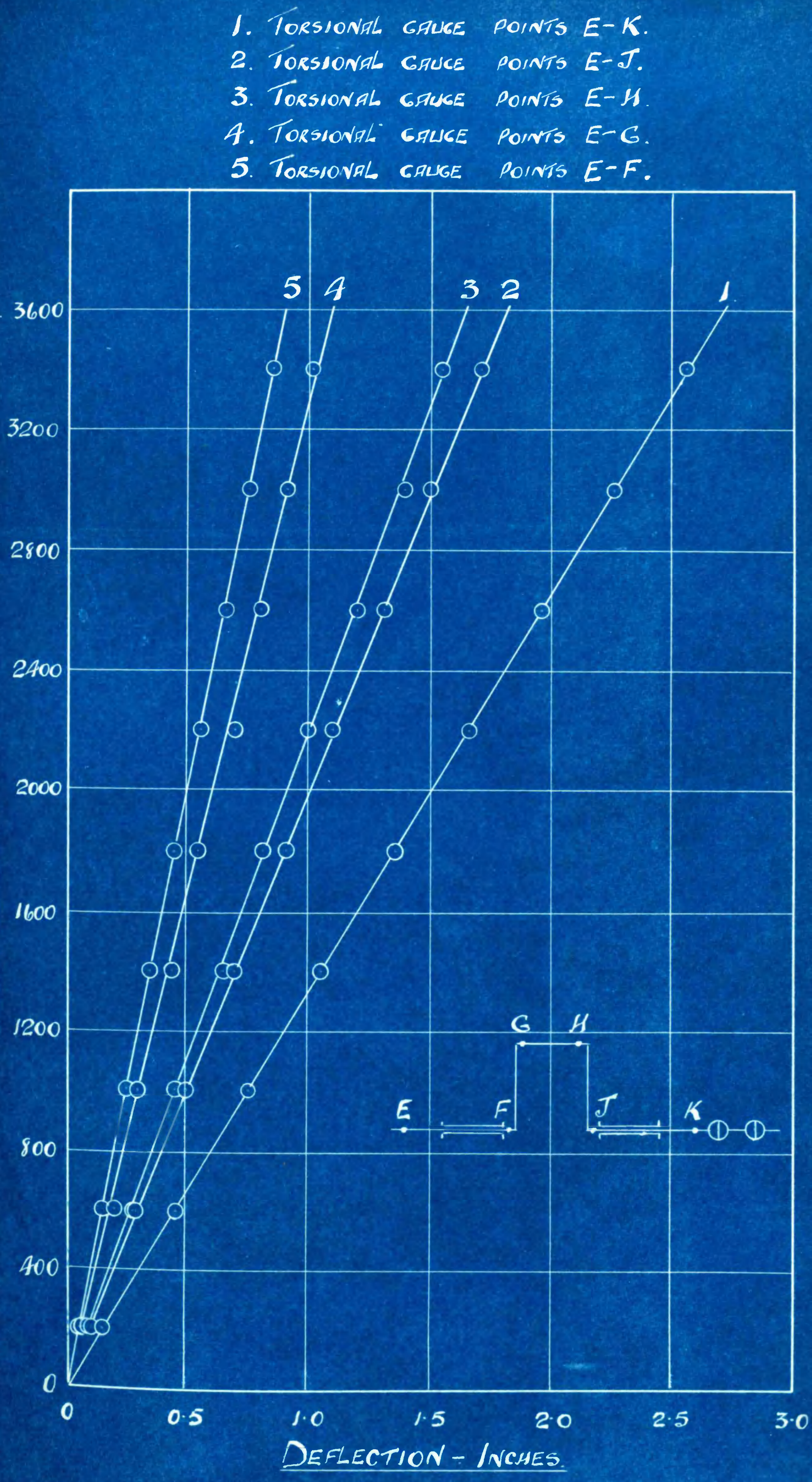


Fig. 24 Out-of-Bearings Torque-Deflection Diagrams for Crankshaft 5C.

1. Torsional gauge points E-K.
2. Torsional gauge points F-J.

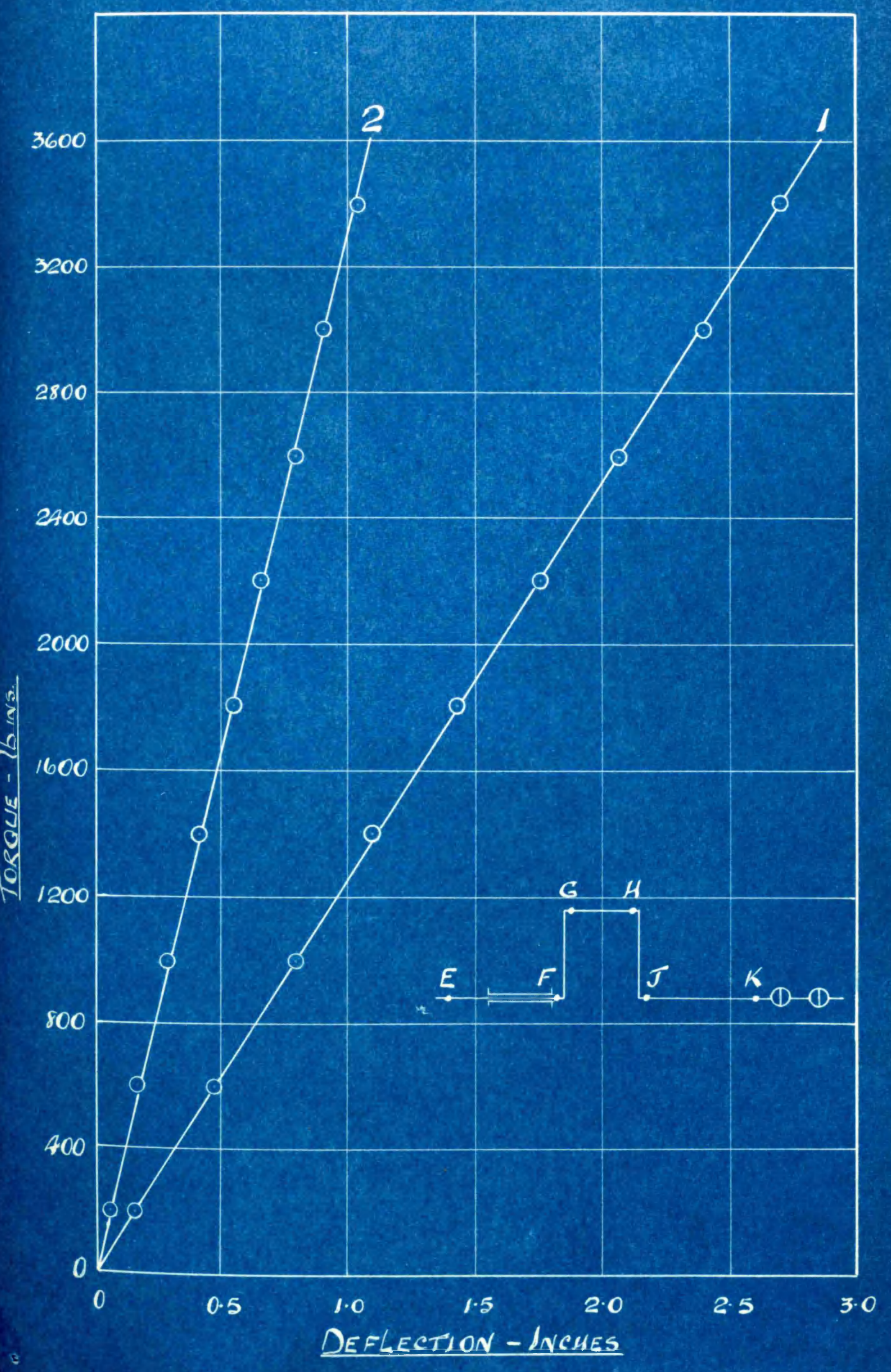


FIG. 25 IN-BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 5C

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS E-J.
3. TORSIONAL GAUGE POINTS E-H.
4. TORSIONAL GAUGE POINTS E-G.
5. TORSIONAL GAUGE POINTS E-F.

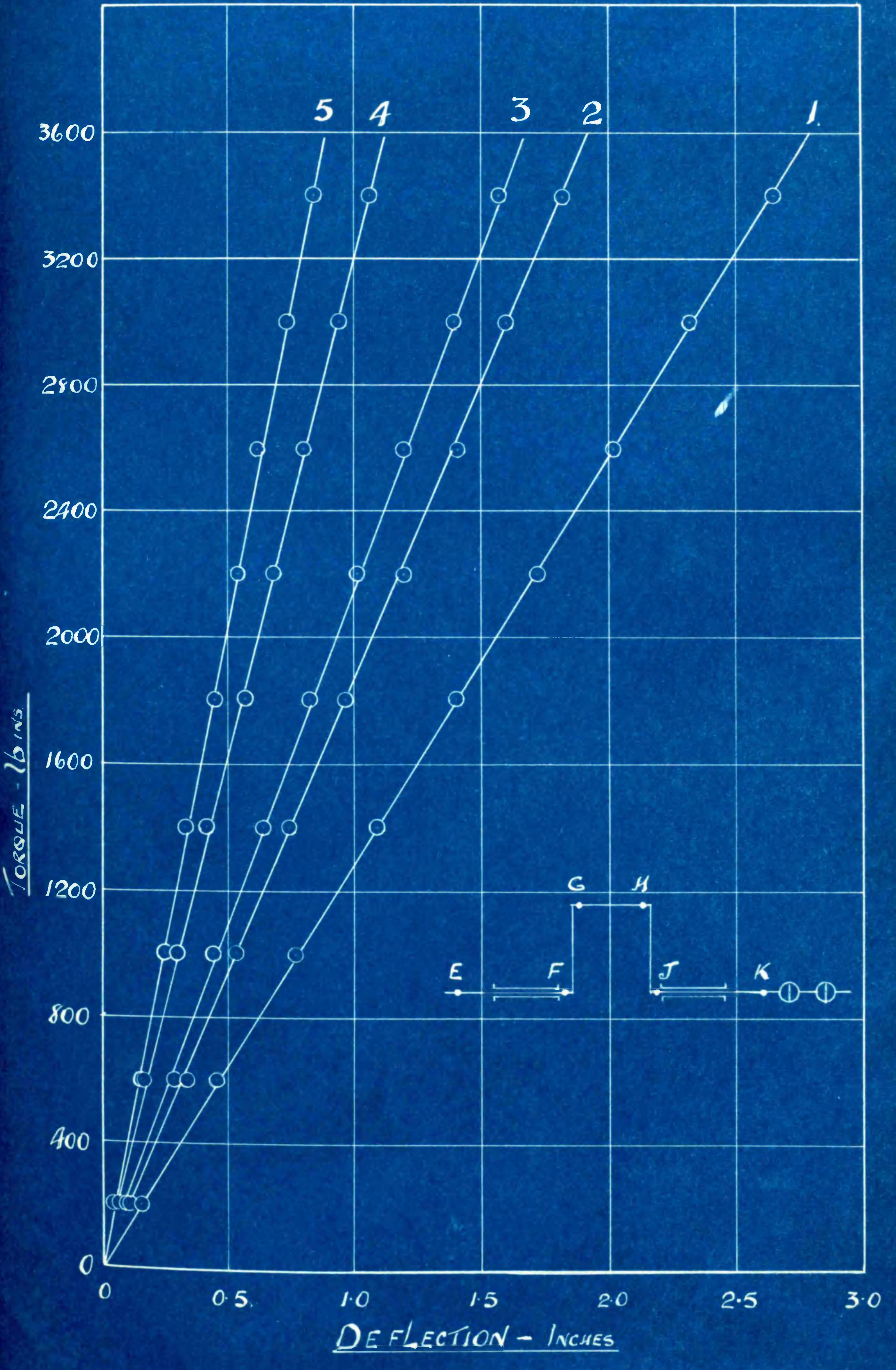


FIG. 26 OUT-OF BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 6C

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS F-J.

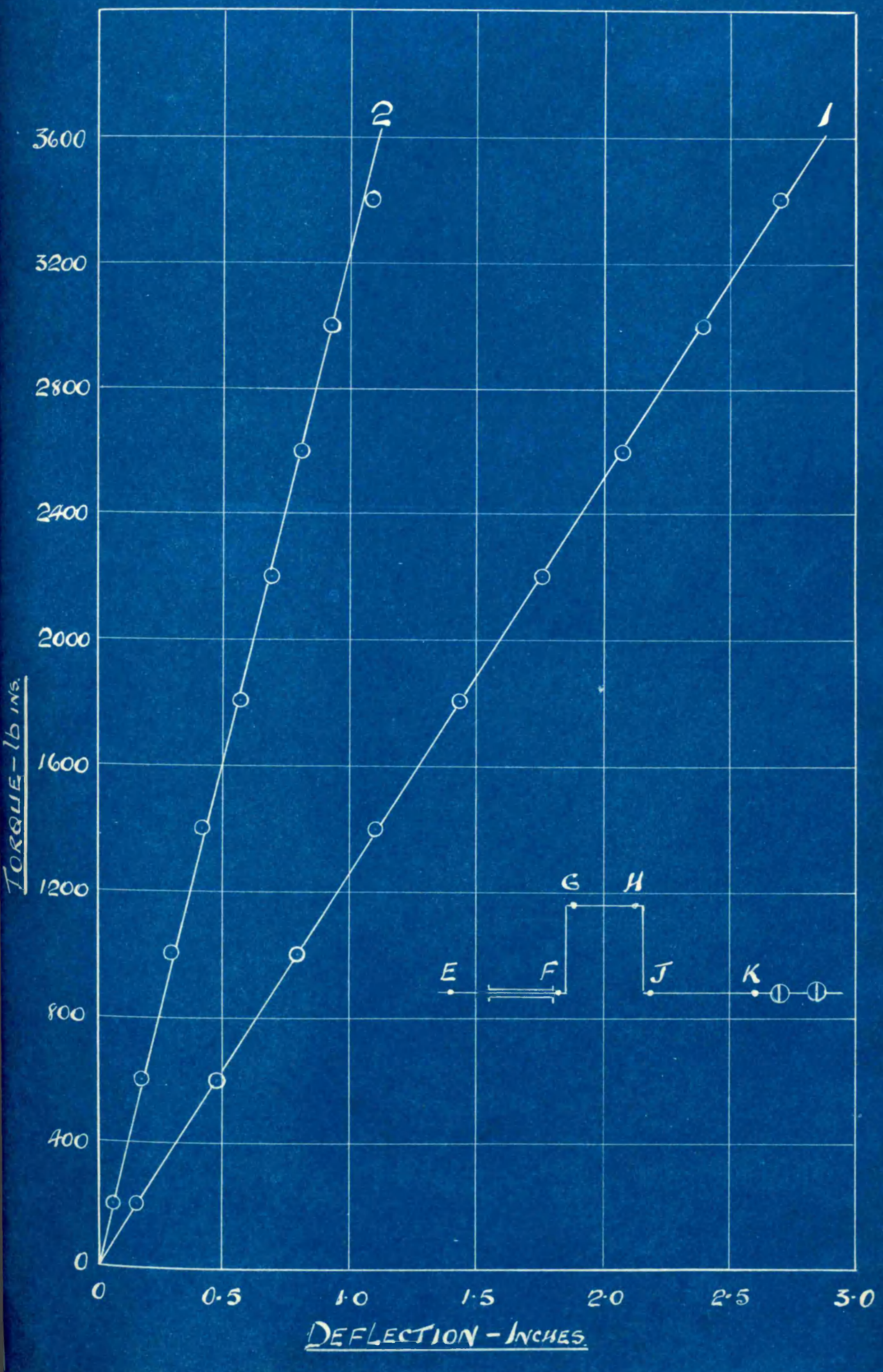


FIG. 27 IN-BEARINGS TORQUE-DEFLECTION DIAGRAMS FOR CRANKSHAFT 6 C

1. TORSIONAL GAUGE POINTS E-K.
2. TORSIONAL GAUGE POINTS E-J.
3. TORSIONAL GAUGE POINTS E-H.
4. TORSIONAL GAUGE POINTS E-G.
5. TORSIONAL GAUGE POINTS E-F.

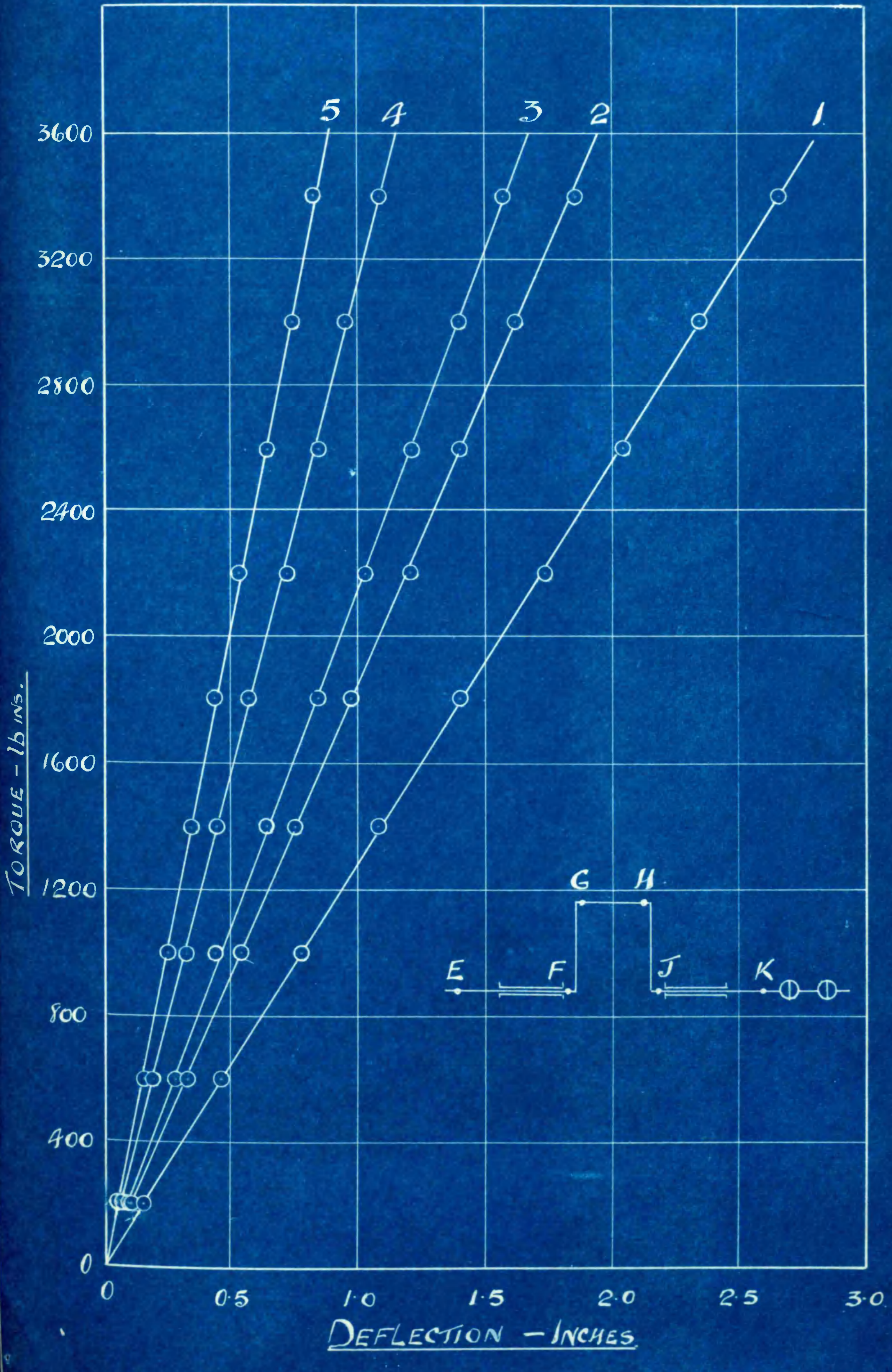


Fig. 28 Out-of Bearings Torque-Deflection Diagrams for Crankshaft 7C.

1. Torsional gauge points E-K.
2. Torsional gauge points F-J.

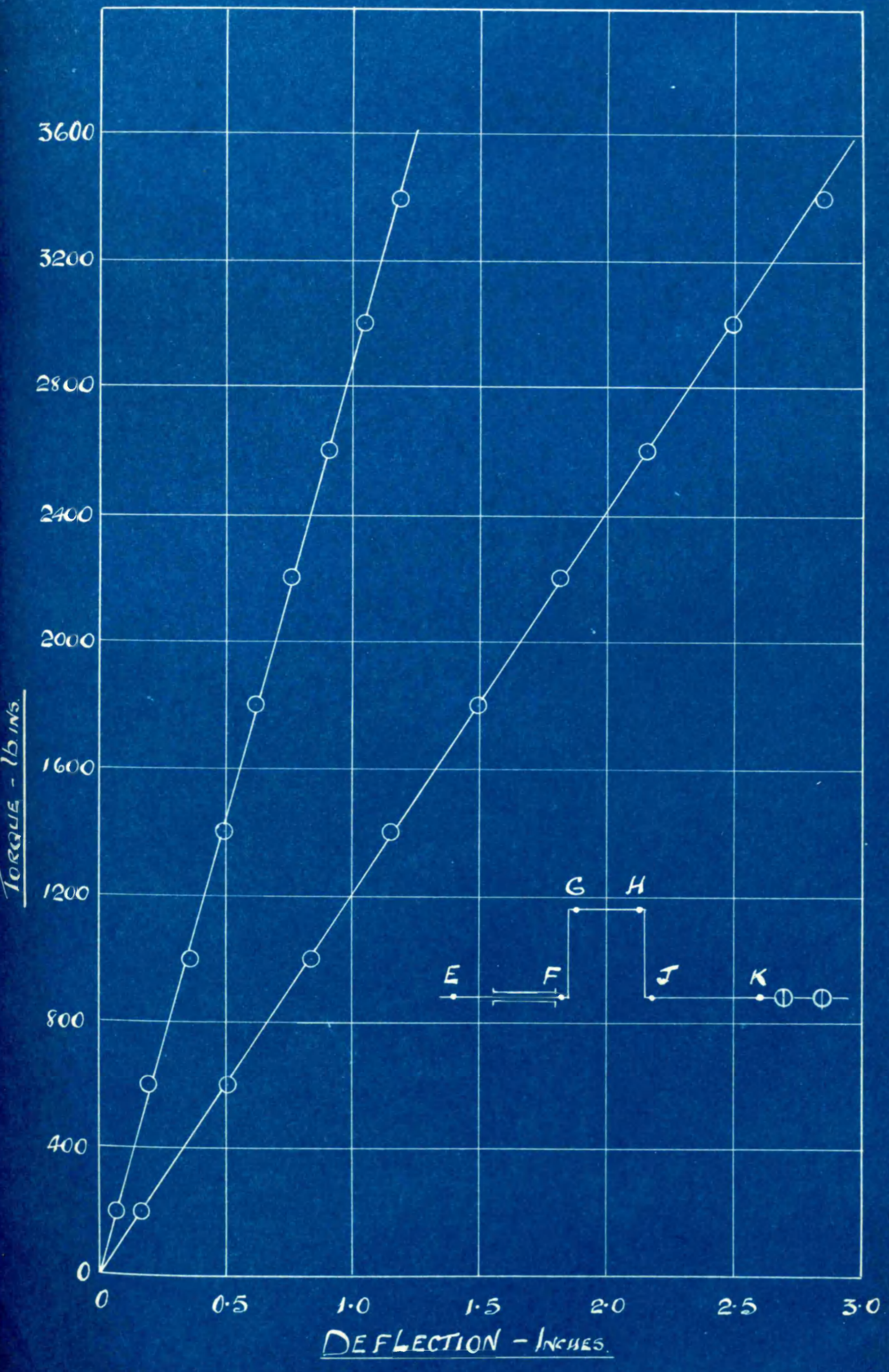


Fig. 29 In-Bearings Torque-Deflection Diagrams for Crankshaft 7C

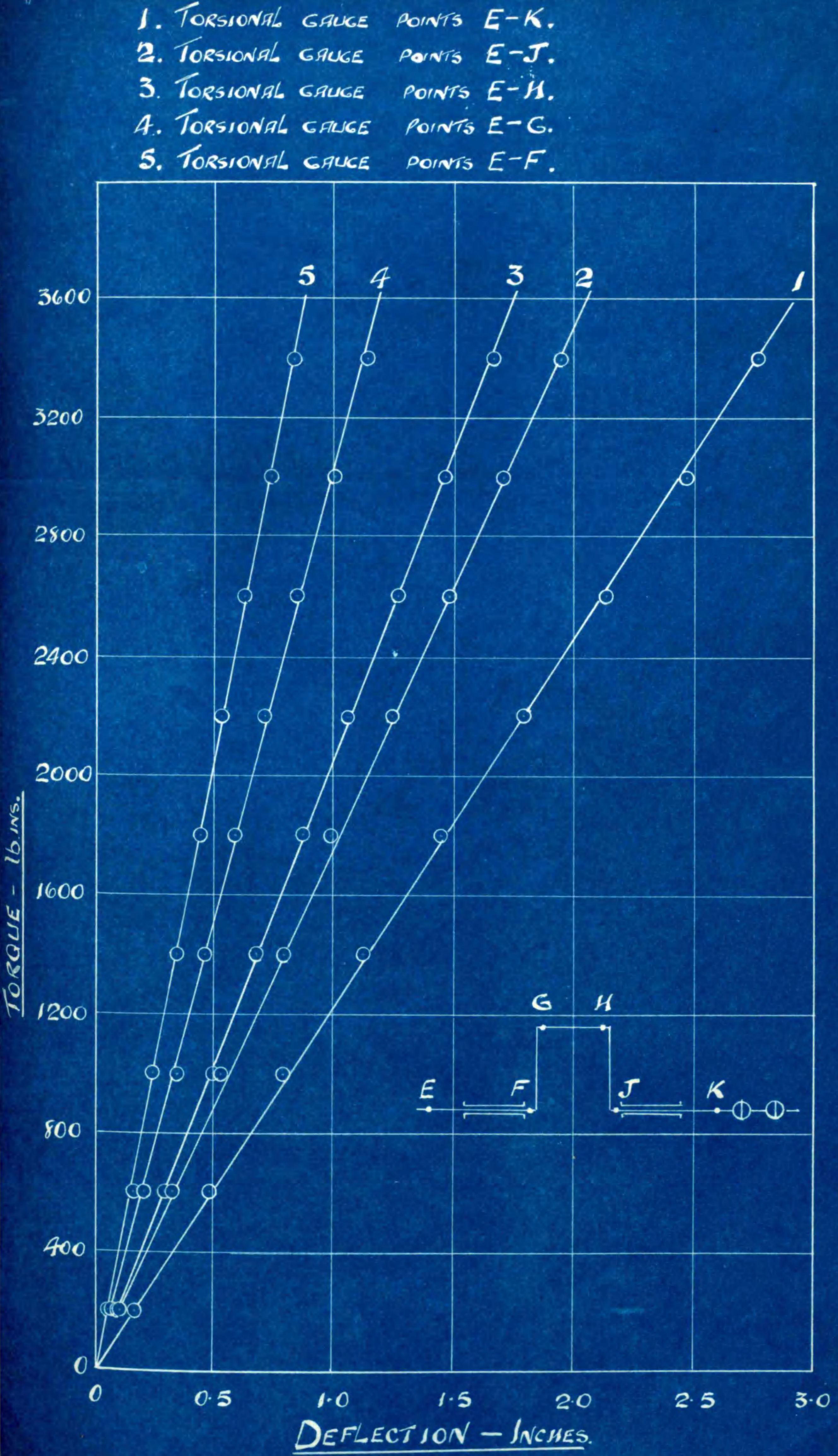


TABLE 7. - Angular Distortion of Crankshaft Model with Various Web Forms.

Shaft	Torque lb.in.	Twist in Radians x 10^3						
		In-Bearings (By Difference)					In-Bearings (Direct)	Out-of-Bearings (Direct)
		E-F	F-G	G-H	H-J	J-K	E-K	E-K
1C	3600	4.074	0.7754	2.523	0.7754	4.074	12.22	12.31
2C	3600	4.074	0.7868	2.523	0.7868	4.074	12.25	12.38
3C	3600	4.074	0.8217	2.523	0.8217	4.074	12.31	12.50
4C	3600	4.074	0.8796	2.523	0.8796	4.074	12.43	12.71
5C	3600	4.074	1.123	2.523	1.123	4.074	12.91	13.17
6C	3600	4.074	1.215	2.523	1.215	4.074	13.10	13.24
7C	3600	4.074	1.481	2.523	1.481	4.074	13.66	13.89

TABLE 8 . - Crankshaft Stiffness.

(Out-of-Bearings and In-Bearings)

Shaft	Test Stiffness, lb.in./radian		Stiffening Ratio per cent.
	Out-of-Bearings	In-Bearings	
1C	2.925×10^5	2.945×10^5	100.7
2C	2.908×10^5	2.939×10^5	101.0
3C	2.880×10^5	2.925×10^5	101.6
4C	2.832×10^5	2.897×10^5	102.3
5C	2.733×10^5	2.789×10^5	102.1
6C	2.719×10^5	2.748×10^5	101.1
7C	2.599×10^5	2.636×10^5	101.4

TABLE 9 . - Crankshaft Stiffness.

(Ratio of Reduced Web Form to Disc Web Form)

Shaft Ratio $\frac{\text{Reduced Web Form}}{\text{Disc Web Form}}$	Effect of Web Form on Shaft Stiffness (Ratio per cent)	
	Out-of-Bearings	In-Bearings
C_1/C	99.42	99.82
C_2/C	98.47	99.33
C_3/C	96.83	98.4
C_4/C	93.46	94.71
C_5/C	92.96	93.35
C_6/C	88.86	89.17

stiffness in-bearings to the stiffness out-of-bearings, stiffening ratio, is given as a percentage in the last column of this table.

The values given in Table 9 give the ratios of the stiffness of model with reduced web form to the stiffness of model with cylindrical web form, which may be regarded as the form giving maximum stiffness.

(d) Comparison with Analytical and Empirical Formulae for Crankshaft Stiffness.

The calculated overall angle of twist by different shaft formulae, for out-of-bearings and in-bearings conditions, together with the corresponding experimental values are given in Tables 10 and 11 . It should be noted that, due to web form changes not included in these rules, the calculated angle of twist for shafts 5C, 6C, and 7C remains the same.

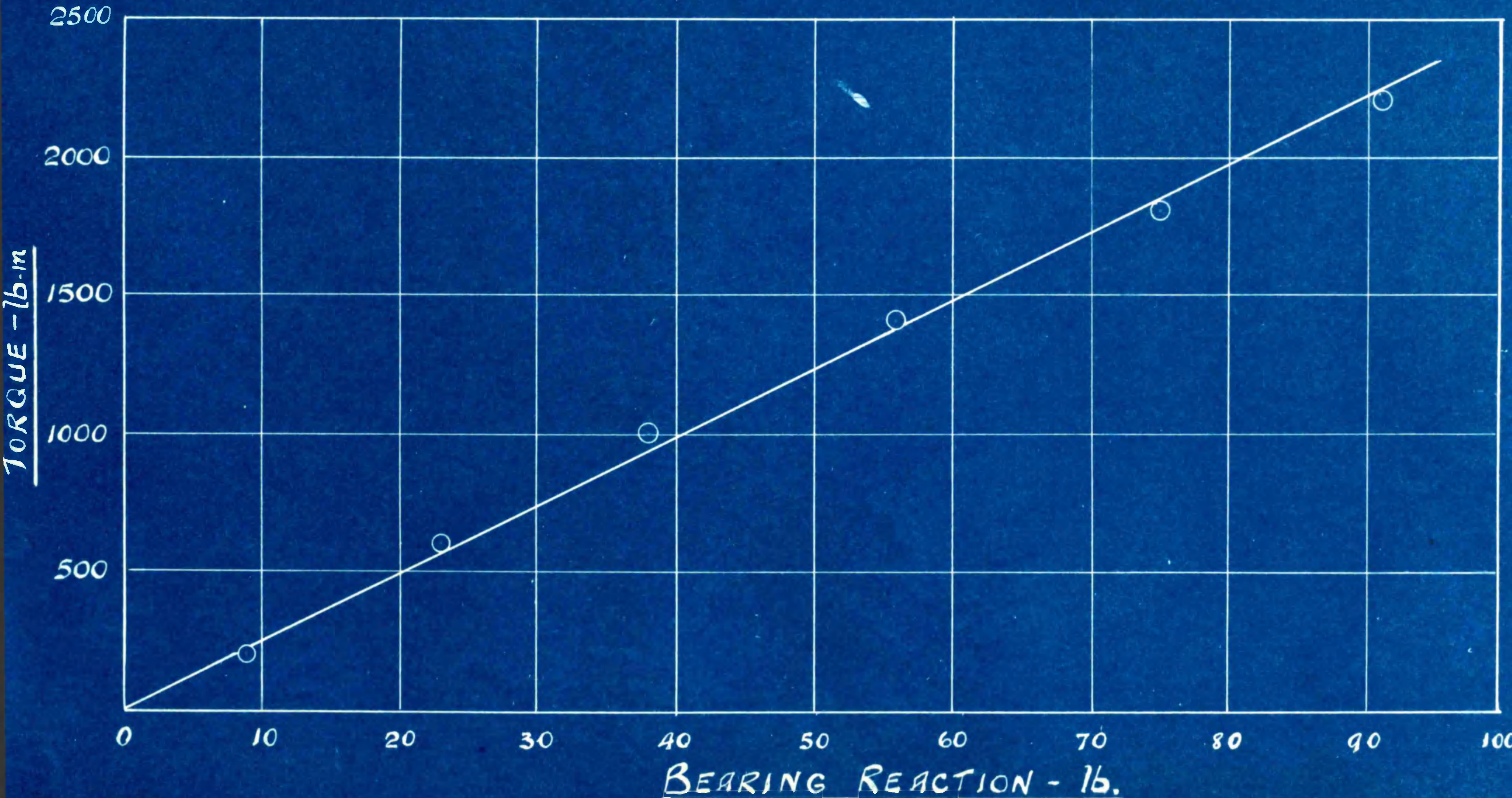
TABLE 10. - Out-of-Bearings Angular Distortion.

Shaft	Torque lb.in.	Twist in Radians x 10^3			
		Experiment	Timoshenko	Holzer	Seelman
1C	3600	12.31	11.87	11.83	13.86
2C	3600	12.38	11.96	11.92	13.93
3C	3600	12.50	12.14	12.09	14.09
4C	3600	12.71	12.51	12.42	14.41
5C	3600	13.24	13.59	13.45	15.37
6C	3600	13.24	13.59	13.45	15.37
7C	3600	13.89	13.59	13.45	15.37

TABLE 11. - In-Bearings Angular Distortion.

Shaft	Torque lb.in.	Twist in Radians x 10^3		
		Experiment	Carter	Timoshenko
1C	3600	12.22	12.00	9.731
2C	3600	12.25	12.12	9.868
3C	3600	12.31	12.32	10.08
4C	3600	12.43	12.76	10.42
5C	3600	12.91	14.08	11.41
6C	3600	13.10	14.08	11.41
7C	3600	13.66	14.08	11.41

FIG. 30 TORQUE - BEARING REACTION CURVE FOR CRANKSHAFT 1C.



CONCLUSIONS

The values given in Tables 7 and 8 indicate that the in-bearings shaft stiffness in each case differs very little from the out-of-bearings shaft stiffness. Since the stiffening ratio of the crank is determined by the amount of journal constraint induced, it will be influenced by such factors as the crank proportions or design, the bearing clearance, and the degree of rigidity of the fixture which supports or forms the bearings. Due to the absence of reliable data on bearing clearances it has, so far, been impossible to estimate the importance of this factor. In the crankshaft stiffness tests conducted by Carter the shafts were twisted in bearings with clearances approximating to those used in ordinary working conditions.

Although great care was taken in fitting the bearings of this crankshaft model so as to prevent, as far as possible, any lateral movement of the journals, provision was made for measuring any such lateral movements by fitting very sensitive optical gauges. The test results obtained gave evidence of complete lateral constraint of the journals. The supporting fixture, made of heavy sections with metal to metal contacts, provided very rigid bearing connections. Conditions were thus established for obtaining maximum journal constraint.

There is experimental evidence, nevertheless, that the journal constraint had very little effect on the shaft stiffness. This is contrary to theory. The torque-bearing reaction curve for shaft 1C, shown in Fig 30. gave bearing pressure values very much lower than those obtained analytically, using equation (14).
Since the bearings must have a certain amount of clearance for/

for successful running, the experimental results indicate that under such conditions no appreciable error would arise by taking the in-bearings stiffness, in each case, as being equal to that for out-of-bearings.

The test results for shafts 5C and 6C, Tables 7 and 8, show that a reduction in the web thickness, by bevelling, has a very pronounced effect on the shaft stiffness. if the webs, in addition, are bevelled at the journal ends, giving a web from very common in practice, the effect would be to reduce still further the torsional rigidity of the shaft. It is reasonable to expect, therefore, although no test results were obtained, that a reduction in the width of the crank webs at the pin and journal ends, by chamfering, would also appreciably affect the shaft stiffness.

It is important to note that, although crank webs are characterized, in general, by forms produced by bevelling and chamfering the webs beyond the pin and journal centre lines, the effect of such changes on the shaft stiffness is not included in the analytical treatment presented by Timoshenko or Holzer, nor provided for in the empirical formulae given by Seelman and Carter.

It is shown that the effect of fitted webs is to increase considerably the torsional flexibility of the shaft. This is brought out very clearly on comparing the test results for shafts 6C and 7C. Many readings were taken to check the very pronounced increase in the angular distortion over the gauge points F-G and H-J. It suggested a certain amount of slip between the webs and the journals. The zero reading at the beginning of each series of readings, however, gave evidence that the increase in the angular distortion was not associated with a condition of slip.

One/

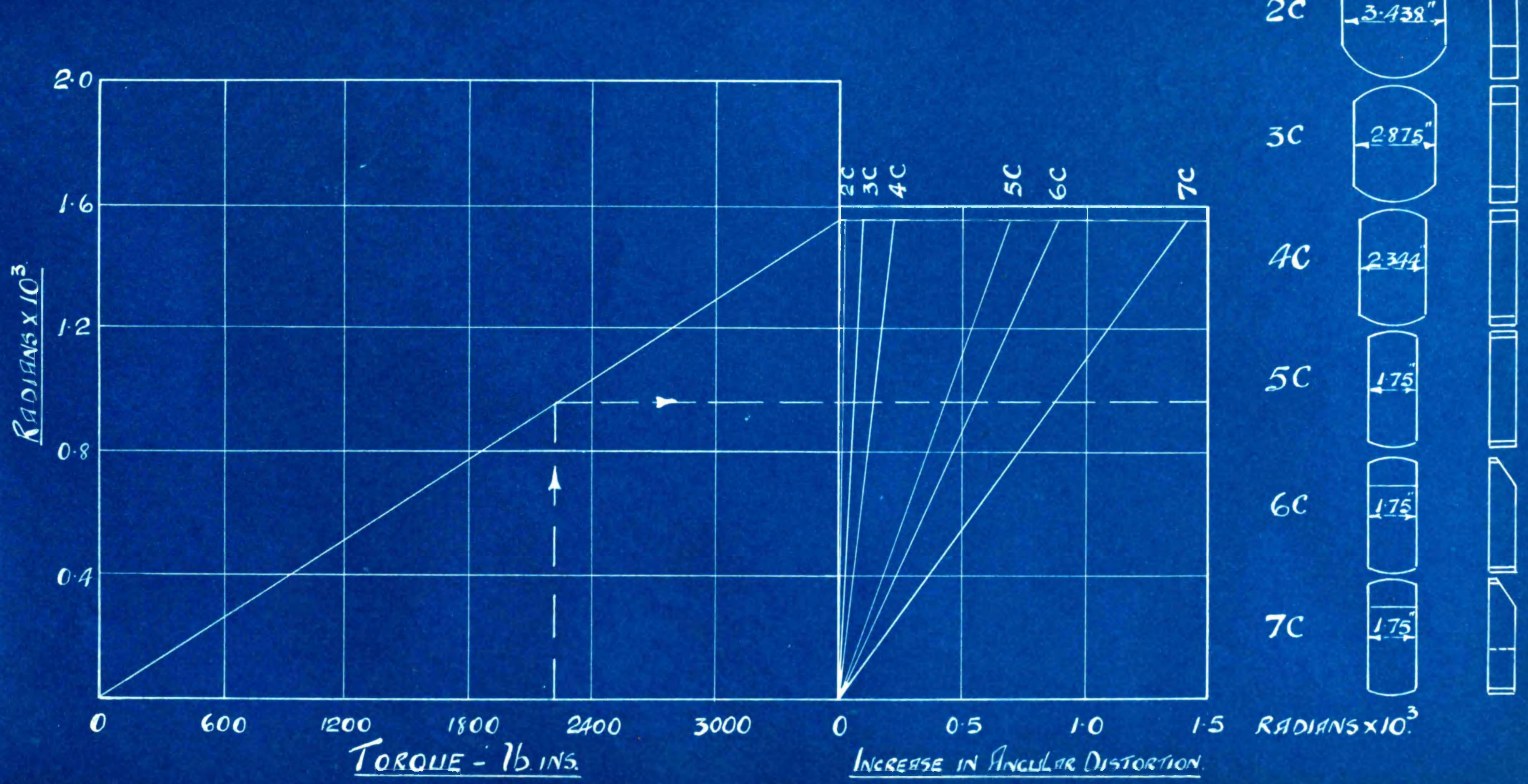
One of the assumptions made in equation (1) is that the cross-section of the web between the pin and journal centre lines is uniform. The junction, however, of solid journals and pin to the webs and the junction of hollow journals and pin to the webs bring about appreciable changes in the cross-section of the webs. It is evident, from test results, that journal holes through the webs, for a shrinkage assembly, create conditions which are not operative in a solid journal-web element.

A fitted web, by shrinkage, is a method of assembly adopted by many builders of crankshafts. It is important to note that it represents a condition not analysed or included in any of the formulae relating to crankshaft stiffness.

Due to the fact that the distortion of each individual element was measured during the tests, it is possible to make a comparison of the variation in web stiffness due to web form.
Fig.31 gives the angular distortion of the cylindrical webs of shaft 1C which may be regarded as providing maximum stiffness. Readings were taken over the gauge points G-F and H-J. Curves giving the increase in the angular distortion, over these points, for other web forms, are shown. These curves indicate an increase of 11.87 per cent. in the torsional flexibility of the webs due to bevelling and also an increase of 34.28 per cent. in the angular distortion of the webs when fitted by shrinkage. The cylindrical web form was taken as the basis for comparison.

The values given in Table 10 show that the experimental results for the out-of-bearings condition are in close agreement with the calculated values obtained by the formulae/

FIG. 31 VARIATION OF WEB STIFFNESS WITH WEB FORM.



formulae given by Timoshenko and Holzer. The values given in Table II show that Carter's formula gives values in close agreement with experiment results for shafts 1C to 4C. This is remarkable in view of the fact that certain proportions of the experimental shaft model were far removed from those of the marine, aircraft and motor car types from which the formula was evolved.
Carter has shown, however, that the formula gives values from 6.4 per cent. low to 8.2 per cent. high for the marine shafts, 10.2 per cent. low to 4.8 per cent. high for the aircraft shafts, and 3.8 per cent. high to 13 per cent. high for the car shafts.

It is shown that the in-bearings angle of twist by Timoshenko's formula, which takes the form given in equation (2) gives values which are not in close agreement with experimental values. It was found that Dr. Seelman's formula, which assumes the distortion of the pin and journals penetrating a distance 0.45 into the webs gave better results.

The contrasts indicate that experimental studies by means of small scale models should form the basis of an analysis when the structural member or element is of a complex form requiring many simplifying assumptions to permit of a theoretical treatment.

P A R T I I I .

SMALL SCALE INVESTIGATIONS

ON

ELASTIC ASSEMBLY GRIPS.

(a) Tests with Fitted Webs on Crankshaft

Certain features of the test results for crankshaft 7 C suggested the possibility of a certain amount of slipping. It was decided, therefore, to investigate by carrying out a torsion test of the crankshaft to destruction. The crank webs of this shaft were fitted under different conditions of assembly. One web was shrunk on the journal shaft with the mating surfaces perfectly dry and free from film. The other web was assembled with a film of rape oil on the mating surface of the journal shaft and fitted with a security pin pressing through the web and shaft.

The Author's workshop experience of force fit assemblies suggested that the surface contact film which may separate the elements in an elastic grip assembly might prove to be an important factor in controlling the quality of the grip.

The test resulted in a failure by slipping of the assembly made with rape oil film on the surface of the solid element. The assembly made with surfaces perfectly dry and free from film, although without a security pin, showed no signs of failure. It was at once evident that the shrinkage allowance, even if correct, was not sufficient of itself to ensure a successful assembly of shrink fit elements.

The results of this test suggested the need and great importance of carrying out research work so as to define more clearly the factors affecting the grip in an elastic grip assembly.

It was decided to carry out a separate investigation with small scale elements using steel pins and collars.

(b)/

(b) Divisions of the Investigation.

The experiments carried out were arranged so that the effects of the following conditions on the grip of an elastic grip assembly might be measured and compared

(I) Effect of a reduction in the surface contact area.

(II) Effect of the nature of the lubricant used in a force fit assembly.

(III) Effect of "skin" or film on the mating surfaces before applying the lubricant.

(IV) Effect of the assembly method.

(V) Torsional and axial resistance to slip.

(c) Failure of Material under Combined Stress

It is usual, in the case of an assembly having cylindrical mating surfaces and dependent upon the elastic properties of the materials for its grip, to employ Lame's thick cylinder theory to determine the magnitude of the stresses induced in the hollow and solid elements. The theory is so well known in connexion with force and shrink fits that it need only be briefly referred to.

The magnitude of the radial interface pressure p_r and hence of the hoop stress f_r induced in the bore layers of the hollow element will depend upon the difference in the free diameters of the mating elements. If S is the fit allowance inducing p_r, R and r the outer and inner radii, and E Young's Modulus for the material, then

$$\frac{S}{2} = p_r \frac{r}{E} \left(\frac{R^2 + r^2}{R^2 - r^2} + 1 \right) \quad \ldots\ldots\ldots\ldots\ldots \quad (1)$$

The radial and hoop stresses are a maximum at the mating surface and vary throughout the wall thickness of the element, reaching minimum values at the outer surface. The maximum value of the hoop stress at the mating surface is given by

$$f_r = p_r \left(\frac{R^2 + r^2}{R^2 - r^2} \right) \quad \ldots\ldots\ldots\ldots\ldots \quad (II)$$

The/

The maximum value of the interface radial pressure, however, will be limited by the failure condition of the layers at the bore surface under combined stress. Many theories have been advanced for ascertaining the criterion of failure of materials under such conditions.

A study of load-extension diagrams would suggest that the maximum radial interface pressure between the elements would be attained with a fit allowance that resulted in yield conditions at the inner bore layers. This has been illustrated in the case of force fits by Russell and Shannon[6]. It is well known, however, that in an endeavour to improve the grip between the elements excessive fit allowances are often used in important assembly units, and that under such conditions a plastic or semi-plastic range may penetrate well into the wall thickness of the hollow element and result in the compressive elastic properties of the material being almost destroyed. The layers within the permanently deformed range will be subject to a compressive stress exerted by the outer layers which are in tension within the elastic range of the wall thickness. The magnitude of the radial interface pressure cannot be accurately determined if the elements are mated under such conditions. The material suffers a change in state when stressed beyond the elastic limit, and load-extension diagrams reveal a casual stress-strain relationship between the elastic and yield point conditions of the material. There is also the uncertainty as to the depth of the plastic range and the effect of the elastic layers in such a case. Overstraining in this manner, if followed by a mild heat treatment, will improve the elastic properties of the hollow element and result in a more uniform distribution of stress throughout the wall thickness. This will allow of a greater interface pressure between the elements of an assembly unit or an increase/

increase in the permissible bore pressure in gun construction. Since the equations giving the hoop and radial stresses at the mating surface of the hollow element are based on proportionality between stress and strain, it is proposed to consider failure as having taken place when the elastic limit condition is reached at the inner bore layers.

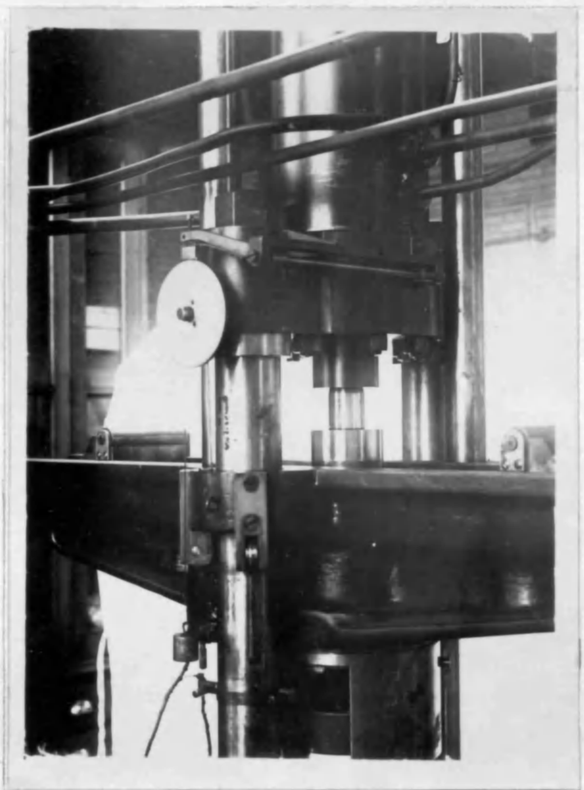
(d) Effect of Surface Contact.

It was decided to carry out tests to determine the effect of a surface out-of-straightness on the pressure necessary to induce axial slipping in a force fit assembly.

(I) Material and Method of Measurement. A pair of hollow and solid elements were machined out of a 28 to 32-ton steel which had been supplied for boiler-stay purposes. The collar or ring was 1¼ inches deep with inside and outside diameters of 1½ inches and 3 inches respectively. The pin was 2 inches long, having a parallel length of 1½ inches with a slight taper at each end to ensure good entry conditions. The mating surfaces of the elements had a high degree of finish, and evidence of a complete surface contact was established after an assembly of the units had been made and broken. Measurements of the pin diameter were taken by a Newall measuring machine and the bore diameter of the collar by the pin-gauge-travel method with a pointer magnification of 7·9. This enabled a force-fit allowance of 0·1 thousandth of an inch to be accurately determined. Readings of both elements were taken at right-angles on three different planes and an average of the six readings was taken. These measurements were made before and after each test to check any possible change in the dimensions of the mating elements.

(II)/

Fig. 32 Force Fit Assembly of Steel Pin and Collar.



(II) Pressing Operation. The pressing operations were carried out on a 20-ton Amsler testing machine operated under oil pressure. The pin was accurately inserted in the bore and the ring placed centrally on a steel collar which gave a maximum bearing surface support to the ring by having a hole clearance of $1/32$ inch for the pin on penetration. Bayonne oil was used as a lubricant and pressing-on and back pressure loads were measured at pin displacements of 0·1 inch. The machine was regulated so that the time taken to effect a complete assembly was approximately twenty-five minutes. Great care was taken before each assembly to have clean mating surface conditions. This was done by washing the pin and the collar three times with soap and hot water and then rinsing well with hot water. The surfaces were finally dried with sheets of clean blotting paper and coated immediately with a film of Bayonne oil. The no-load position of the crosshead with the mating elements suitably arranged for a pressing-on operation is shown in Fig.32. All subsequent pressing operations were carried out in a similar manner.

(III) Tests to investigate Surface Contact. Five tests were carried out with a pin and collar, as shown in Fig.33, having a force fit allowance of 0·6 thousandth of an inch. Substituting for R and r in equation (1) it can be shown that the interface radial pressure intensity is given by

$$p_r = 5056\frac{S}{d}$$

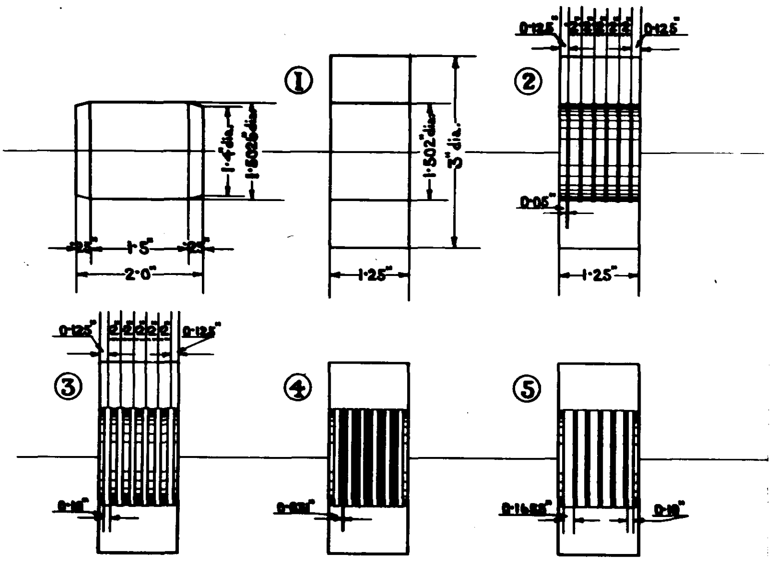
and the hoop stress at the bore layer is given by

$$f_r = 1.67\,p_r$$

Hence a fit allowance of 0·6 thousandth of an inch will induce a radial pressure intensity at the mating surfaces of 2 tons per sq.in., and a hoop stress intensity at the bore layer of 3·3 tons per sq.in.

Test 1./

Fig. 33 Tests on a Pin and Collar.



	Bearing Length	Bearing Surface	Maximum Loads	
			Pressing-on	Pressing-off
1	1·25 ins.	5·9 sq. ins.	9·5 Tons	10·4 Tons
2	0·95 "	4·484 "	8·025 "	9·55 "
3	0·65 "	3·068 "	6·575 "	8·00 "
4	0·495 "	2·336 "	6·15 "	7·45 "
5	0·3225 "	1·522 "	4·875 "	5·15 "

Test 1. - The length of the bearing surface was $1\frac{1}{4}$ inches, giving a surface area of 5·9 sq.in. The maximum pressing-on and back pressure readings were 9·5 tons and 10·4 tons respectively.

Test 2. - Six grooves, each 0·05 inch broad, were cut in the ring bore at centres 0·125 inch from each face, and 0·2 inch between adjacent grooves. This gave an effective bearing length of 0·95 inch and an effective bearing surface of 4·484 sq.in. The maximum pressing-on and back pressure readings were 8·025 tons and 9·55 tons respectively.

Test 3. - Six grooves, each 0·10 inch broad, were cut at the same centres as in Test 2, giving an effective bearing length of 0·65 inch and an effective bearing surface of 3·068 sq.in. The maximum pressing-on and back pressure readings were 6·575 tons and 8·00 tons respectively.

Test 4. - A groove, 0·031 inch broad, was cut in the middle of each of the five inside bearing bands or strips. This left an effective bearing length of 0·495 inch and an effective bearing surface of 2·336 sq.in. The maximum pressing-on and back pressure readings were 6·15 tons and 7·45 tons respectively.

Test 5. - Five of the narrow bearing strips, each 0·0345 inch wide, were machined off, leaving a bearing strip 0·075 inch wide at each face and five inside bearing strips each 0·0345 inch wide. The effective bearing length was now reduced to 0·3225 inch and the bearing surface to 1·522 sq.in. The maximum pressing-on and back pressure readings were 4·875 tons and 5·15 tons respectively.

The/

The load diagrams on a base of bore length are given in Fig.34, and indicate the axial force required at different points of the bore depth to press the pin in and out. The curves clearly indicate that a reduction of area created by bearing-band surface conditions on the surface of one of the mating elements will result in a big reduction in the maximum load at which axial slip will take place in assembly units.

An out-of-straightness condition of the cylindrical surface of an element may result in a series of such bearing bands approaching any one of the conditions illustrated in the tests. This will depend upon the degree of accuracy obtained in the generation of the cylindrical mating surface. If the mating surface of the solid element is also not straight then the back pressure necessary to bring about initial slip may be much lower still. It is important to note that this pressure may be lower than the final pressing-on load, and that the load required to bring about further slipping is greatly in excess of the initial value.

The Author has encountered this condition in trying to dismantle an important assembly unit at a colliery where all efforts to remove a hollow element failed, with tools at hand, after having shifted it 1/64 inch. It is also worth noting that the maximum back pressure is also much in excess of the maximum pressing-on load. These points, clearly illustrated in the load diagrams shown in Fig.34, have been brought out in the many tests on force fits carried out in the Mechanical Engineering Laboratories of the Royal Technical College, Glasgow.
Fig.35 shows the actual ring or collar on which these tests were carried out. The dimensions of the mating surfaces measured/

Fig.34 Load Diagrams for Pin and Collar Tests.

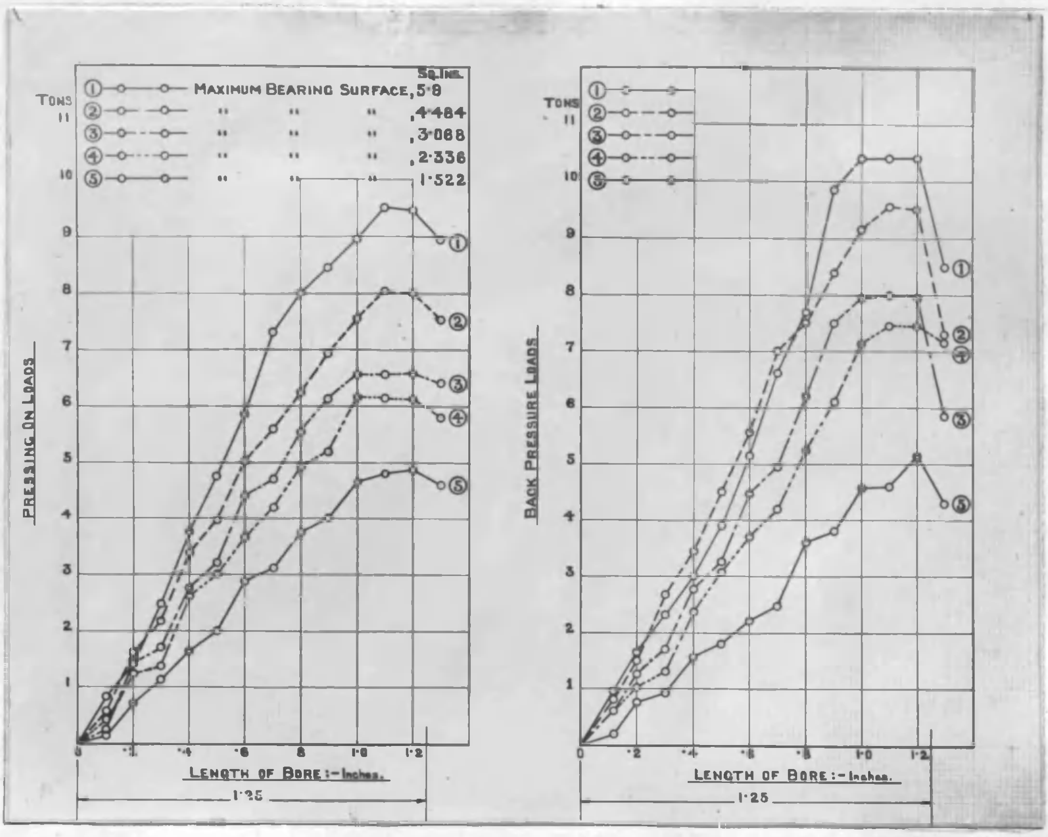


Fig.35 Photograph of Collar Tested.

measured throughout the tests showed no signs of change. This was remarkable in view of the fact that the elements were assembled and dismantled five times. Inspection of the narrow bearing strips gave evidence of good bearing conditions with no signs of damage due to shear in the assembly operation despite the fact that the force-fit allowance of 0·6 thousandth of an inch induced an interface radial pressure intensity of 2 tons per sq.in.

(e) Effect of Lubricant.

Workshop experience of the influence of a lubricant in preventing the mating surfaces of a force-fit assembly from being scored or destroyed in making or breaking the fit, together with the evidence, presented by R.M. Delly, M.I. Mech.E., that the static coefficient of friction between the surfaces of different metals when lubricated depends upon the oiliness of the lubricant, suggested that in a force-fit assembly the lubricant employed might have an appreciable influence on the pressing-on load. There appears to be a very general impression that, due to high radial pressure intensity, the oil would be shorn off the surfaces of the mating elements in the assembly process, and in such a case the nature of the lubricant could have little or no influence on the pressure required to make the assembly. In a force-fit assembly lubrication of the mating surfaces, which are in contact except for a thin film which may be only one molecule thick, is termed solid film or boundary lubrication. It has been suggested by Langmuir and others that the adsorbed film will give a static coefficient of friction depending not only upon the chemical nature of the lubricant but also upon the chemical nature of the mating elements to which it adheres chemically. If there is relative motion between the mating elements, however, the mating surfaces will be completely/

completely separated by a relatively thick film of oil, and friction will depend upon the viscosity of the oil.

It was decided to carry out tests on force-fit assemblies, with some of the earlier vegetable and animal oils used as lubricants by the engineer, together with the later types of mineral and blended oils.

The general method of procedure was similar to that carried out in the surface contact series of tests. Great care was taken to establish constant conditions of cleanness in the mating surfaces, the importance of which has been demonstrated by Hardy. The following may be quoted from the work of J.M. Macaulay:[8] "Using two glass plates of good optical polish, coefficients of limiting friction as high as 0·8, 0·9, over 1·0 were obtained when the surfaces had been very thoroughly cleaned by prolonged washing with sodium oleate in running hot water, then in hot distilled water, drying being done rapidly before an electric radiator and the surfaces placed together while hot. Slipping, when it did occur, was with a jerk and a scratching of the glass. If the surfaces, after washing, were rubbed with 'clean' dry linen, coefficients of limiting and of kinetic friction as low as 0·2, 0·1, 0·08 were observed, and slipping was then steady and slow (say 1 cm. per minute), suggesting the viscous flow of a fluid."

In the experiments about to be described constant mating surface conditions, with respect to cleanness, were obtained by washing the pin and collar three times with soap and hot water and rinsing well with hot water from the tap.
The surfaces were then dried with sheets of clean blotting-paper. They were then wiped three times with a clean cloth moistened by petroleum, and again wiped three times with a clean cloth moistened by methylated ether. The surfaces were/

were then immediately coated with a film of oil which was allowed to drain off partially before the elements were assembled. After the washing operations the mating surfaces were not touched with the fingers.

(I) Tests to investigate Effect of Lubricant. Six tests were carried out with the same pin and collar having mating dimensions of 1·5038 inches and 1·5030 inches respectively, giving a difference in the free diameters of the mating elements of 0·8 thousandth of an inch. It can be shown that a force-fit allowance of this order will induce a radial pressure intensity at the mating surfaces of 2·7 tons per sq.in. and a hoop stress intensity at the bore layer of 4·51 tons per sq.in.

Test 1. - Elements assembled and dismantled with film of Bayonne oil on mating surfaces.

Test 2. - Elements assembled and dismantled with film of Sperm oil on mating surfaces.

Test 3. - Elements assembled and dismantled with film of "Texaco" motor oil (heavy) on mating surfaces.

Test 4. - Elements assembled and dismantled with film of rape oil on mating surfaces.

Test 5. - Elements assembled and dismantled with film of graphite and engine-bearing oil on mating surfaces.

Test 6. - Elements assembled and dismantled with film of cutting lubricant on mating surfaces.

The lubricant used in Test 6. was an emulsion, suitable for general lathe work, made by mixing a cutting compound with water. Cutting oils and cutting compounds are used in metal machining operations not only to dissipate the heat generated and wash away the cuttings but also to give a smooth finish and prevent rusting. The dimensions of the mating surfaces were measured before and after each test and/

and these showed no signs of change. This is an essential condition, of course, to any successful investigation of the influence of the lubricant on the force-fit tonnage. It was made possible by the use of elements having good mating surfaces and a relatively small force-fit allowance. It is surprising that after assembling the elements six times there was no evidence of a tendency for the bore dimensions to become bigger or the pin dimensions to become smaller. A critical examination of the mating surfaces gave no evidence of improved bearing contact, a condition which would have influenced the pressure necessary to make the assembly.

N.N. Sawin[9] observed that if the elastic limit was not exceeded and the assembly repeated several times an increase in pressure, the rate of which decreased with each assembly, was necessary to make the fit.

Load diagrams on a base of bore length are given in Fig.36 and indicate clearly the remarkable influence of the lubricant on the force necessary to make an assembly of the elements. The values range from 5·1 tons with rape oil as a lubricant to 13·55 tons with Bayonne oil as a lubricant. This represents an increase of 166 per cent. in the pressing-on load due to a difference in the nature of the lubricant.

The coefficient of friction for each assembly was obtained by the equation

$$F = \mu p_r \cdot 2\pi r \cdot l \; .$$

where F is the maximum pressing-on tonnage, l the length of the hollow element, and μ the coefficient of friction. The values obtained by this method of approach to boundary lubrication, given in Table 12, are of a very high order and greatly in excess of those obtained by the Deeley machine using the same lubricants but different mating materials.

It/

Fig.36 Load Diagrams for Tests on Effect of Lubricant.

TEST 1. Bayonne oil.

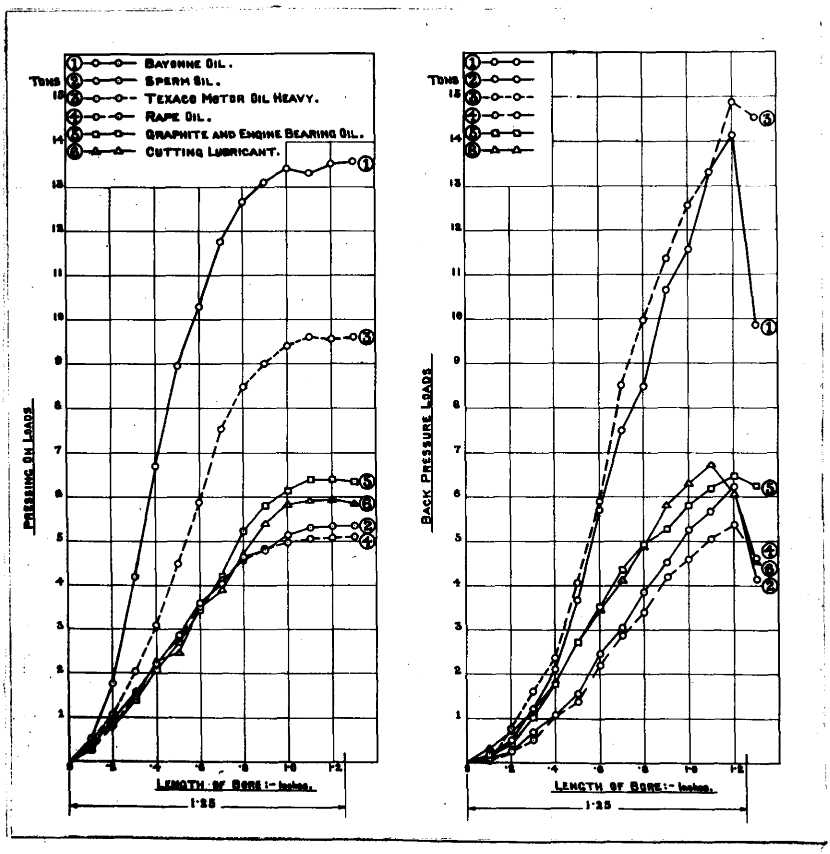
TEST 2. Sperm oil.

TEST 3. "Texaco" motor oil (heavy)

TEST 4. Rape oil

TEST 5. Graphite and engine bearing oil

TEST 6. Cutting Lubricant.



It was found, however, that the ratios of the maximum load obtained with the various lubricants to the maximum load obtained with rape oil were of the same order as the ratios of the coefficients of friction obtained by the Deeley machine.

TABLE 12.

Description of Lubricant.	Force Fit.			Deeley Machine. *					
	Max. Pressing-on Tonnage	Mild Steel on Mild Steel		Mild Steel on Cast Iron.		Mild Steel on Lead Bronze.		Cast Iron on Cast Iron.	
		μ	Ratio of Pressing Loads	μ	Ratio of Friction Coeffs.	μ	Ratio of Friction Coeffs.	μ	Ratio of Friction Coeffs.
Bayonne Oil	13.55	0.852	2.657	0.213	1.79	0.234	1.721	0.236	2.458
Sperm Oil	5.35	0.336	1.049	0.127	1.067	0.180	1.32	0.122	1.271
"Texaco" Oil	9.60	0.604	1.882	-	-	-	-	0.186	1.938
Rape Oil	5.10	0.321	-	0.119	-	0.136	-	0.096	-
Graphite & Engine-bearing Oil	6.40	0.402	1.255	-	-	-	-	-	-
Cutting Lubricant	5.94	0.374	1.165	-	-	-	-	-	-

* Jl. Royal Technical College, Glasgow, 1931, vol.2, p.495.

It has come to be recognized that these lubricating films are really in chemical union with the mating surfaces of the elements. The molecules orient themselves on the solid surfaces and form an adsorbed primary film. This orientation may extend and result in a film of this nature several molecules thick. There can be no doubt that under the high pressure intensity with which the elements are assembled, the static coefficient of friction between the surfaces is that of two primary films in contact. The results obtained would suggest that these films are in the nature of soft solids which require a definite force to produce shear.

Deeley/

Deeley[10] suggests that in the case of hydrocarbon oils the ends of the chains of the molecules forming the adsorbed film must be regarded as asperities which are elastic and give rise to friction. The actual value of the coefficient of friction he regards as depending upon the form and stiffness of the asperities which are displaced when relative motion is about to take place and spring back into position after slip has occurred. Observations made during the assembling and dismantling of these elements would suggest that some such conditions hold. In no case was an assembly made by a gradual slipping of the mating elements. The load gradually rose until slip took place, when there was a sudden fall in the oil pressure. This gradually rose again until a further slip and another drop in the oil pressure occurred. With those lubricants having a pronounced oiliness the slips took place without any noise until the assembly was half complete, after which slipping was accompanied by a light bumping on the sudden fall in pressure, which became more pronounced towards the completion of the fit. Periodic slips of much greater magnitude and accompanied by heavy bumping was noticed throughout the complete assembly of the elements when a lubricant not having an oiliness property was employed.

(f) Effect of "Skin".

It was thought that the "skin" or thin layer of grease that tends to gather on the surface of an element during or after machining operations might in some way influence the effect of the lubricant in an assembly process. After completing the back pressure test in which the cutting compound was used as a lubricant, the surfaces of the pin and collar were wiped only with a clean cloth and then re-assembled with rape oil as the lubricant. But the closeness of the readings/

readings obtained with these two lubricants and perfectly clean surfaces prevented any definite conclusions being reached. It was therefore decided to use two lubricants giving a high and a low load respectively with perfectly clean surfaces, and Bayonne oil and rape oil were selected as providing the best means of analysing this "skin" effect.

(I) Tests to investigate Effect of "Skin". Five tests were carried out with the same pin and collar having a force fit allowance of 0.6 thousandth of an inch. It can be shown, as in the series of tests to investigate surface contact, that a fit allowance of this order will induce a radial pressure intensity at the mating surfaces of 2 tons per sq.in., and a hoop stress intensity at the bore layer of 3.3 tons per sq.in.

Test 1. - The mating surfaces were carefully cleaned as previously described. Rape oil was used as the lubricant. The maximum pressing-on and back pressure readings were 3.65 tons and 4.10 tons respectively.

Test 2. - The rape oil which remained on the mating surfaces after dismantling was carefully wiped off with a clean cloth. The surfaces were then finished off by rubbing thoroughly with a clean cloth until the surfaces were polished. Bayonne oil was used as the lubricant. The maximum pressing-on and back pressure readings were 8.50 tons and 9.05 tons respectively.

Test 3. - The surfaces were treated as in Test 1. Bayonne oil was used as the lubricant. The maximum pressing-on and back pressure readings were 8.40 tons and 10.40 tons respectively.

Test 4. - The surfaces were treated as in Test 2. Rape oil was used as the lubricant. The maximum pressing-on and back pressure readings were 3.95 tons and 4.18 tons respectively.

Test 5./

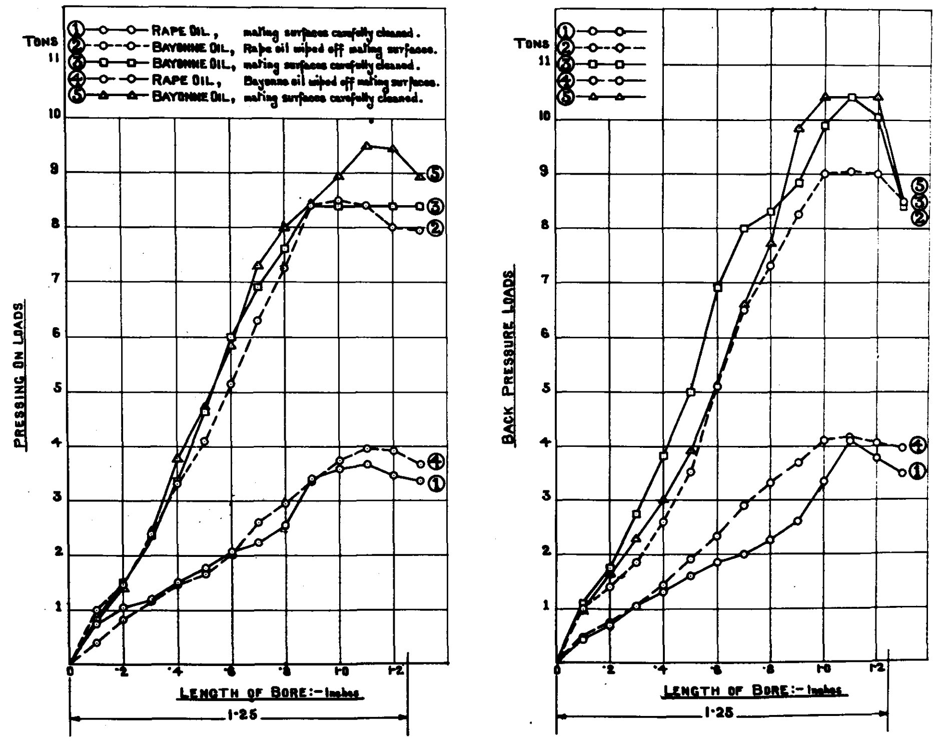
Test 5. - Pin and collar were washed three times with soap and hot water and rinsed well with hot water from the tap. The surfaces were then dried with sheets of clean blotting paper and coated immediately with a film of oil which was allowed to drain off partially before the elements were assembled. Bayonne oil was used as the lubricant. The maximum pressing-on and back pressure readings were 9.50 tons and 10.40 tons respectively.

(II) TEST RESULTS:- The load diagrams on a base of bore length are shown in Fig.37. They represent two well-defined types, the characteristic feature of each being clearly determined by the nature of the lubricant employed. The latter is also responsible for a difference in the load necessary to make the assembly of the two elements of the same order as in a previous test. It would appear, therefore, that the "skin" or invisible primary film which may have been formed on the mating elements which were reassembled after having been treated with a clean cloth only, had no effect on the characteristic influence of the lubricant on the force-fit load. Otherwise it would appear that polishing off as much of the lubricant as possible with a clean cloth has produced a surface condition of cleanness similar to that obtained by the treatment described above for "clean" surfaces. This would account for the load diagrams obtained with the respective lubricants being practically the same.

It will be noticed in Fig. 37 that towards the completion of the assembly in Test 5. there is certainly an appreciable increase in pressure compared with Tests 3. and 2. In this particular assembly the elements were not treated with petroleum or methylated ether. No great importance is attached to this, however, since it is clear that the same characteristic/

Fig.37 Load Diagrams for Tests on "Skin" Effect.

TEST 1. Rape oil. Mating surfaces carefully cleaned.
TEST 2. Bayonne oil. Rape oil wiped off mating surfaces.
TEST 3. Bayonne oil. Mating surfaces carefully cleaned.
TEST 4. Rape oil. Bayonne oil wiped off mating surfaces.
TEST 5. Bayonne oil. Mating surfaces carefully cleaned.



characteristic load curve is clearly defined in each case when Bayonne oil is employed as the lubricant. A consideration of the load curves for Tests 1. and 4. suggests that this applies also in the case when rape oil is used as the lubricant.

Sir William Hardy and Miss Doubleday[11] carried out experiments on the effect of polishing off as much of the lubricant as possible with perfectly clean linen. The experimental apparatus employed consisted of a slider with a highly polished spherical face applied to a plane surface which was also highly polished. They found that the lubricant, if a fluid, was completely removed from the surface by polishing, and the friction between the elements rose to "clean" surface value. If the lubricant was a solid, however, polishing produced an invisible primary film which gave a very low frictional value. The assumption made therefore that the mating surfaces in the force-fit assembly tests were "clean" after treatment with a clean cloth would appear to be well founded. The situation is an interesting one. There is evidence that by gentle treatment with a clean cloth the lubricant may be polished off the surfaces of the mating elements and its influence completely removed. There is also evidence that this cannot be done by a force-fit assembly process, and that the primary films, even under high radial pressure intensities, exercise an enormous influence on the tonnage to produce slip.

(g) Effect of Assembly Method.

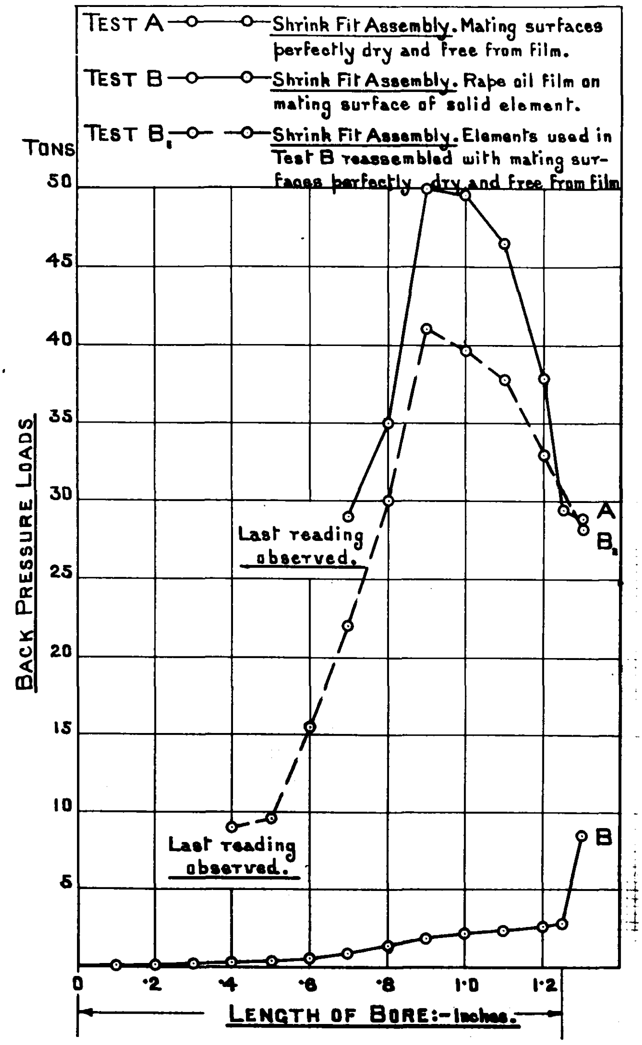
It was decided to carry out tests on the quality of the grip established by each assembly method. The load required to produce axial slip of the elements when the mating/

mating surfaces were perfectly dry and free from film was compared with that required for an assembly in which the mating surfaces were separated by an adsorbed lubricating film. Six pins and collars of the same material and dimensions as those previously described were obtained. The mating elements in each case had a fit allowance of 1/1,000 inch. A fit allowance of this order will induce a radial pressure intensity at the mating surfaces of 3.37 tons per sq.in., and a hoop stress intensity at the bore layer of 5.63 tons per sq.in. The elements before each assembly were washed three times with soap and hot water, rinsed well with hot water from the tap and the mating surfaces dried immediately with sheets of clean blotting paper. A period of two days elapsed after each assembly was made before the back pressure test was applied.

(I) Shrink Fit Assembly Tests:- Experiments were carried out with two pins and two collars having a fit allowance of 1/1,000 inch. The collars were heated by a gas-burner until a hole clearance was obtained, as measured by a gauge, which would just allow the collar to slip over the pin.

Test A. - The elements were assembled with the mating surfaces perfectly dry and free from film. The mating surface of the hollow element had just the slightest touch of colouring due to the heat applied.
The 20-ton Amsler testing machine failed to separate the elements and the back pressure test was completed by a 100-ton Avery horizontal testing machine. An initial load of 28.8 tons was required to produce slip but a much greater axial load was required to maintain this condition. The load diagram on a base of bore length for the partial separation of the elements is given in Fig.38.
The elements of this assembly are shown in Fig.39. The clean/

Fig.38 Load Diagrams for Shrink Fit Assembly Tests.



clean mating assembly had allowed seizure to occur so strongly that when slip took place the surface particles of each solid were pulled out of position. The mating surfaces were torn to such a depth that no further use could be made of the elements. The quality of the grip established was such that the taper part of the solid element was shorn off, as shown in Fig.39, and left firmly embedded in the surface of the hollow element.

Test B. - The hollow element was prepared as in Test A. The solid element or pin had a thin film of rape oil on the surface.

The back pressure test was carried out on the Amsler machine and a tonnage of 8.5 tons was recorded when slip occured. This pressure dropped immediately to a value of 2.8 tons and then gradually fell off as the separation of the elements proceeded. The load diagram on a base of bore length for the complete withdrawal of the solid element is given in Fig.38.

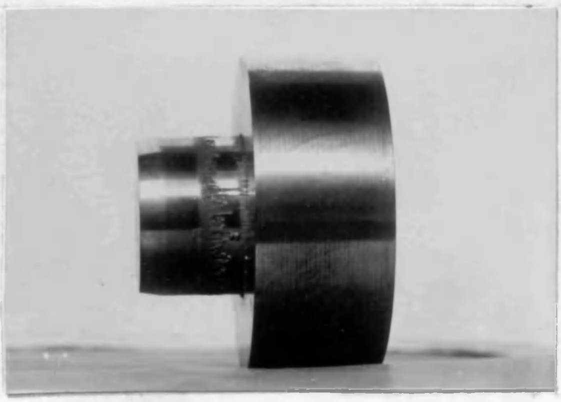
Very good surface conditions obtained after dismantling the elements, as is shown in Fig.39. There was no sign of even the tiniest hair-line scratch but there was evidence that the mating surfaces had been separated by a lubricating film. The darkish colour in the bore surface could be lifted off with the lightest touch of the finger tip, revealing the bright steel mating surface underneath.

It was decided, since the surfaces were in perfect condition and there was no change in the mating dimensions, to re-assemble these elements under different conditions.

Test B_1. - The elements were re-assembled with the mating surfaces perfectly dry and free from film as in Test A. There was a slight creaking when the load applied reached 8.5 tons, but there was no measurable sign of slip within the load capacity of the Amsler machine. Slip took place when a load of 28.2 tons was reached on the 100-ton testing machine/

Fig. 39 Shrink Fit Elements.

TEST A: Partial separation of elements mated with surfaces perfectly dry and free from film.



TEST B: Elements after back pressure test. Assembled with rape oil film on solid element.

TEST B_1: Elements after back pressure test. Assembled with surfaces perfectly dry and free from film.

machine, and an increase in load was required, as in Test A, to produce further slipping between the elements. The load diagram for the complete separation of the elements is given in Fig.38.

The elements of this assembly are shown in Fig.39 .

The mating surfaces were very badly torn, large pieces of each mating element being pulled out of position. The grip was so well established that, as in Test A, part of the solid element was shown off during withdrawal and left firmly fixed to the surface of the hollow element.

These experiments demonstrate that a primary film may separate the elements in a shrink fit assembly even when mated under a high radial pressure intensity. The results indicate that the presence of such a film will result in an enormous reduction in the axial resistance of the elements to slip.

(II) Crank Web Shrinkage Allowances:- It is important to note that many builders of crankshafts make adsorbed film conditions possible by lubricating the pin and journals in the hope of avoiding preseizure of the elements on assembly. The detrimental effects of this cannot be fully met by increased shrinkage. It is possible that, in some cases, high shrinkage allowances have been adopted to prevent failures not associated with a shrinkage allowance but entirely due to a surface film condition. Moreover, these excessive shrinkage allowances cannot be regarded as representing the best conditions of assembly in a unit which is dependent upon the elastic properties of the mating materials for its grip.

It may be advisable, at this stage, to examine the fit allowances employed and the magnitude of the stresses induced in a built-up crankshaft assembly.

Consider/

Consider the web having dimensions wuch that $R/r = 1.876$. Substituting for R and r in equation (1), the interface radial pressure intensity is given by

$$p_r = 4785\frac{S}{d}$$

and the hoop stress at the bore layer by

$$f_r = 1.8\ p_r$$

The elastic failure condition under the combined stress due to p_r and f_r is given by

$$1.8\ p_r = f_L \text{(Maximum principal stress theory)}$$
$$2.1\ p_r = f_L \text{(Maximum principal strain theory)}$$

$$2.8\ p_r = f_L \text{(Maximum shear stress theory)}$$
$$2.3\ p_r = f_L \text{(Maximum strain energy theory)}$$

TABLE 13. - Stresses Induced by Various Shrinkage Allowances.

Shrinkage Allowance.	Radial Pressure.	Hoop Stress.	Principal Stress Theory.	Principal Strain Theory.	Shear Stress Theory.	Strain Energy Theory.
S/d thousandths of an inch per inch dia.	p_r tons per sq.in.	f_r tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.
1.0	4.785	8.613	8.61	10.05	13.40	11.01
1.25	5.981	10.766	10.77	12.56	16.75	13.76
1.50	7.178	12.92	12.92	15.07	20.10	16.51
1.75	8.374	15.073	15.07	17.59	23.45	19.26
2.00	9.57	17.226	17.23	20.10	26.80	22.01

The webs in such an assembly are generally made from a 28- to 32-ton rolled slab steel and the shrinkage allowance by crankshaft makers is between 1 and 1.75 thousandths of an inch per inch diameter. It will be observed from Table 13 that a shrinkage allowance of 1.75 thousandths of an inch per/

per inch diameter would require the elastic limit to be not less than 19.26 tons per sq.in. An elastic condition of this order obtained in the case of the force-fit assembly having a fit allowance of 1.75 thousandths of an inch per inch diameter.

This is a figure associated with steel of high quality and is much in excess of that obtained from a 28- to 32-ton steel. Load-extension curves obtained from test pieces taken from a steel bloom before being rolled into slabs for crank webs gave an average elastic limit value of 9.5 tons per sq.in. This material may be regarded as representative of the quality of the steel in the centre part of the crank web which has had comparatively little work done on it during rolling. The high shrinkage allowance, therefore, adopted by some crankshaft makers is bound to introduce plastic or semi-plastic conditions well into the wall thickness of the web material. It is suggested that a successful shrinkage assembly may be effected with the wall thickness of the hollow element entirely within the elastic range of the material.

(III) Assembly by Expansion Fit. By this method the solid element is cooled to an extremely low temperature by immersing it in a refrigerant. This brings about a reduction in the diameter which will allow of it being slipped easily into the hole of the hollow element. The solid element, in returning to normal temperature conditions, expands and the elastic strains and stresses induced in the elements result in an effective grip being established. Liquid oxygen, whose boiling point is at the extremely low temperature of–190 deg. C.–(310 deg. F.), was used as a cooling agent. The amount of contraction possible, which sets a limit to the maximum possible fit allowance by this method of assembly, is determined by the temperature at which the/

the cooling agent evaporates. The solid element should just be a push fit when fully cooled. Although the shape of the part may influence the time necessary to bring about the lowering of the temperature, the liquid requirements are determined by the material and weight of the part immersed. These requirements can be reduced if the refrigerating capacity of the vapour which comes off the oxygen as it evaporates is usefully employed in precooling the parts before immersion.

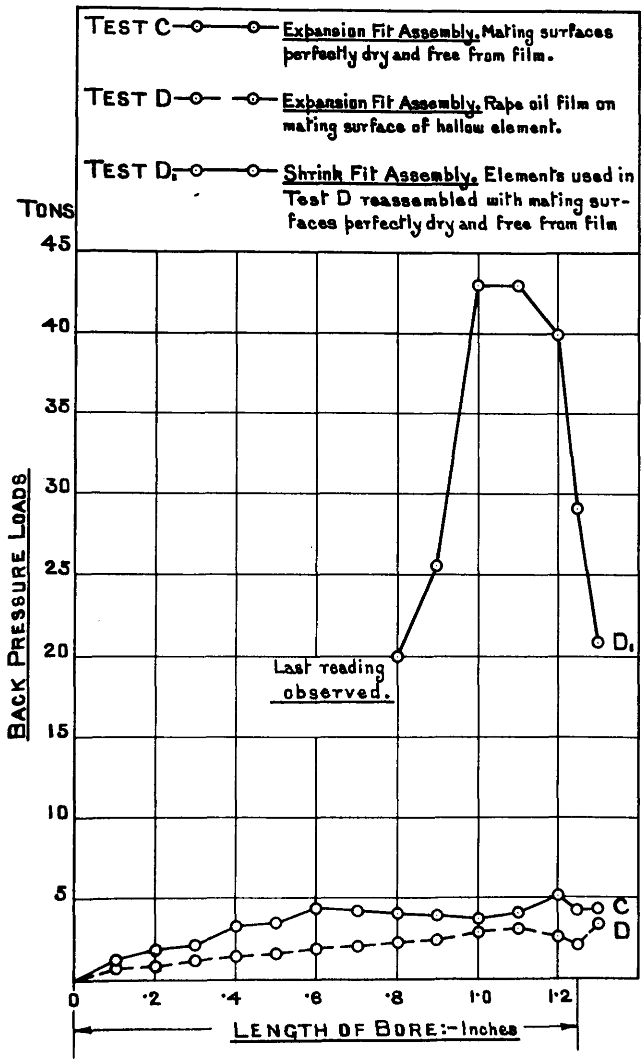
(IV) Expansion Fit Assembly Tests: Experiments were carried out with two pins and two collars having a fit allowance of 1/1,000 inch. The solid elements were cooled by liquid oxygen to a temperature of approximately - 310 deg.F. which allowed of the pin, in each case, being easily slipped into the hole of the collar.

Test C. - The elements were assembled with the mating surfaces perfectly dry and free from film. The pin, after removal from the refrigerant, was rubbed hurriedly with a clean cloth before making the assembly.
The back pressure test was carried out on the Amsler machine, and a load of 4.35 tons was recorded when slip occurred. This pressure rose to a maximum of 5.05 tons and an average pressure of a little over 4 tons was required to maintain a condition of slip over the first half of the operation. The load diagram on a base of bore length for the complete withdrawal of the solid element is shown in Fig.40.

The elements of this assembly are shown in Fig.41 .
The mating surfaces were in perfect condition. A few extremely fine hair-lines could be seen, but not felt, on the bore surface, and these, if rubbed lightly with the finger-tips, could be removed.

Test D/

Fig. 40 Load Diagrams for Expansion Fit Assembly Tests.



Test D. - The solid element was prepared as in Test C. The hollow element had a thin film of rape oil on the bore surface.

The back pressure test gave a load of 3.425 tons when slip occurred. The pressure then dropped suddenly, rising afterwards to a value of 3.15 tons, after which there was a gradual drop. The load diagram on a base of bore length for the complete separation of the elements is shown in Fig. 40.

The elements of this assembly are shown in Fig. 41 and give evidence of the excellent surface conditions that obtained after breaking the fit. There were a few fine hair-lines on the mating surface of the hollow element similar in character to those described in Test C. It was decided, since there was no change in the mating dimensions, to reassemble these elements under different conditions.

Test D_1. - The elements were reassembled by shrinkage with the mating surfaces perfectly dry and free from film.

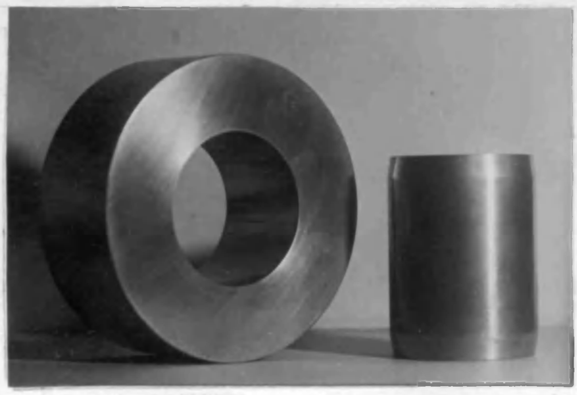
Slip took place when a load of 20.86 tons was reached on the 100-ton testing machine. A much greater axial thrust was required, however, to effect further separation of the elements. Part of the load diagram on a base of bore length is shown in Fig. 40.

The elements of this assembly are shown in Fig. 41. Seizure at the points of contact was so strong that slip caused the surface particles of each element to be pulled out of position and these were rolled and dragged during further separation of the elements so that the mating surfaces were very badly torn.

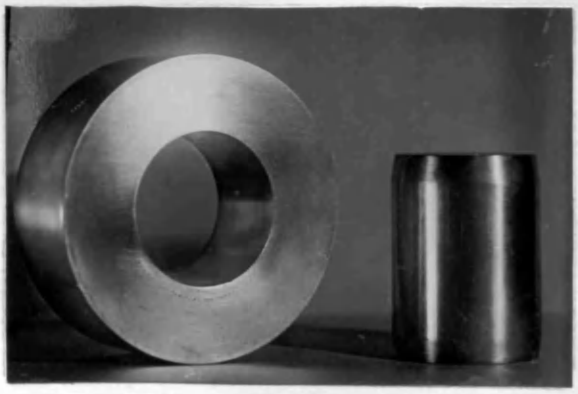
The low tonnage obtained in Test C, together with the excellent condition of the mating surfaces after dismantling, would suggest that the solid elements were separated by a very/

Fig. 41. Expansion Fit Elements.

TEST C: Elements after back pressure test. Assembled with surfaces perfectly dry and free from film.



TEST D: Elements after back pressure test. Assembled with rape oil film on bore surface.



TEST D: Elements after back pressure test. Shrink fit assembly with surfaces perfectly dry and free from film.

very thin film and failed to make actual contact. Although an endeavour was made to have the surfaces perfectly dry and free from film at the instant at which the assembly took place, there can be no doubt but that a water film formed on the pin by frosting during the expansion period. This film would act as a lubricant and the friction during slip of the elements would be that due to two primary water films,, one on each mating surface. As already pointed out, the temperature of the refrigerant sets a limit to the maximum possible fit allowance by this method of assembly, and it would appear that a limit is also set to the quality of the grip by the water film produced during frosting. The Author knows of no effective means, unless the operation is carried out under ideal dry air conditions, of getting rid of the film produced during the period of expansion. This would indicate that this method is limited to an assembly of units in which the hollow element takes only a small proportion of the maximum thrust or torque transmitted by the solid element.

(V) Force Fit Assembly Tests: Experiments were carried out with two pins and two collars having a fit allowance of 1/1,000 inch. The pressing operations were carried out on a 20-ton Amsler testing machine operated under oil pressure. The machine was regulated to effect a complete assembly or dismantle the elements in approximately twenty-five minutes. Two different lubricants were used.

Test E. - Elements assembled with rape oil film on mating surfaces.

Test F. - Elements assembled with Bayonne oil film on mating surfaces.

The load diagrams on a base of bore length for the complete separation of the elements are shown in Fig.42. The back pressure/

Fig. 42 Load Diagrams for Force Fit Assembly Tests.

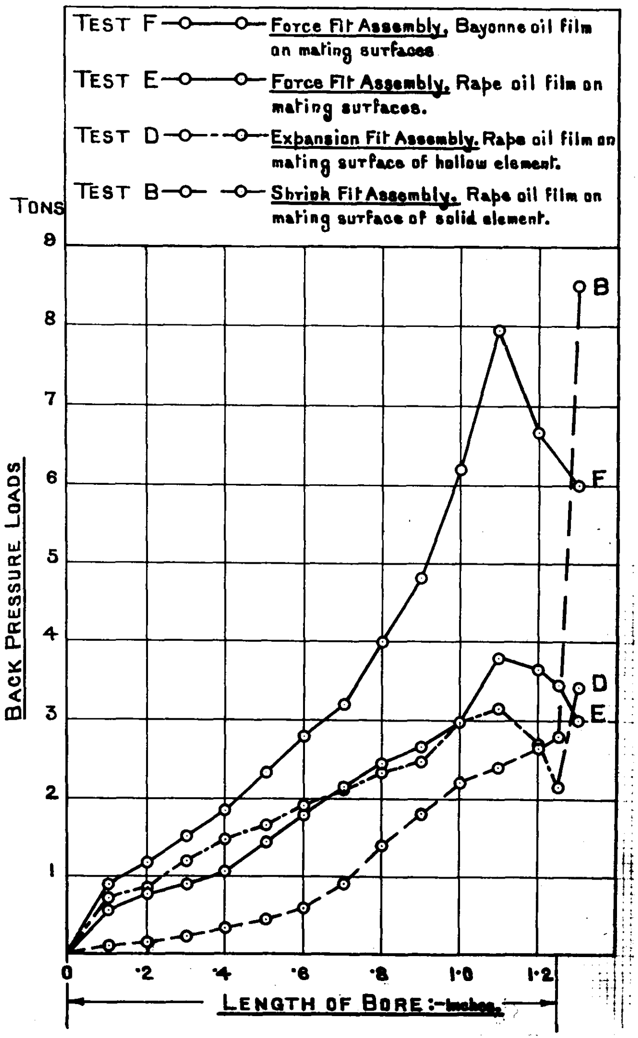
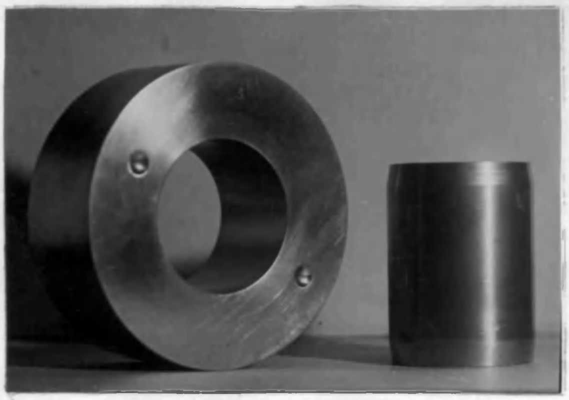
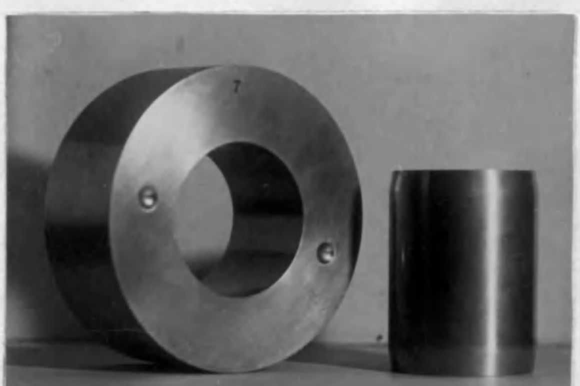


Fig. 43 Force Fit Elements.

TEST E: Elements after back pressure test. Assembled with rape oil film on mating surfaces.



TEST F: Elements after back pressure test. Assembled with Bayonne oil film on mating surfaces.



pressure loads for the same film condition by the other two assembly methods are also shown and provide a basis of comparison.

The elements of these assemblies are shown in Fig.43.

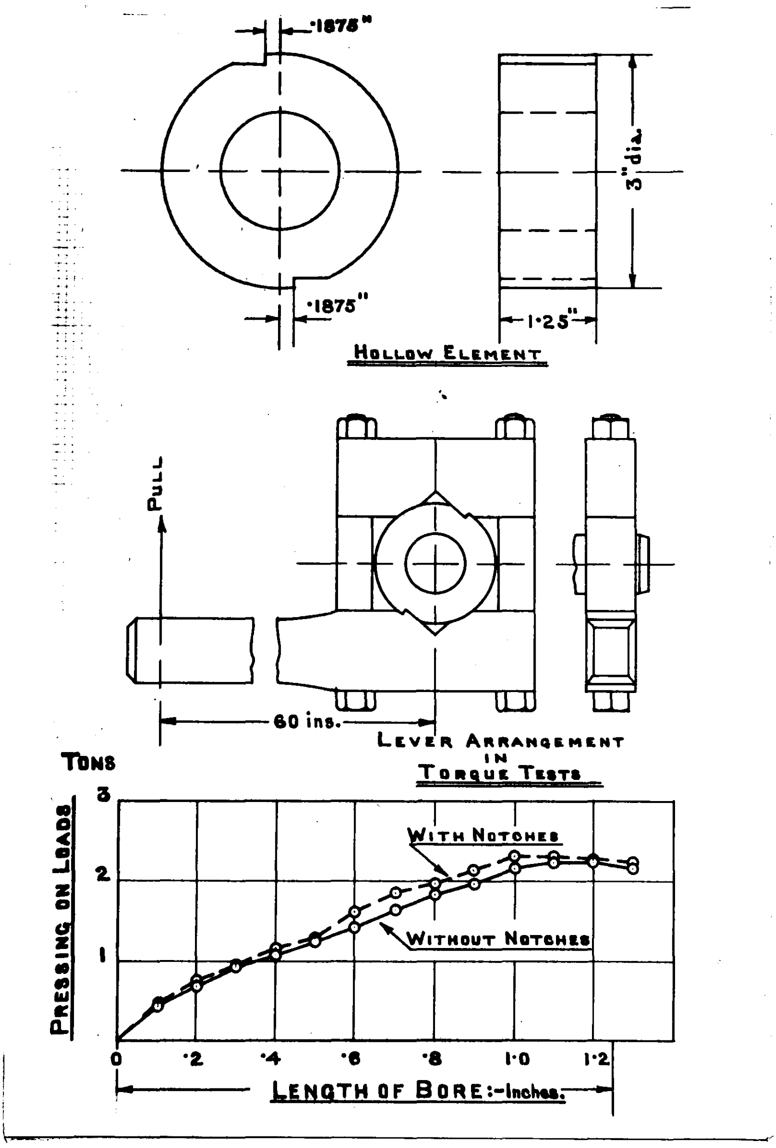
The bearing surfaces of the hollow and solid elements in both bases were in an excellent condition, not even the tiniest hair-line mark being visible.

The load diagrams shown in Fig.42 would suggest that a force fit assembly of elements using as a lubricant Bayonne oil, a mineral oil, may afford a quality of grip far superior to that established by even a shrink fit or expansion fit assembly if the mating surfaces are separated by a film with a distinct oiliness characteristic.

(h) Torsional Resistance to Slip.

The quality of the grip established between elements under the different conditions of assembly considered has been measured by comparing the axial resistance of the elements to slip. These assemblies are usually made, however, for the transmission of torque, and although in some cases they may only transmit a small proportion of the shaft torque they may in other cases be called upon to transmit the full torque capacity of the shaft. It was thought advisable, therefore, to make a few assemblies under different conditions and to compare the grip by the torsional resistance of the elements to slip. It was hoped that this would also provide sufficient data for comparing the axial and torsional coefficients of friction for the different assembly conditions. The solid element, in this case, was supported horizontally between centres and firmly fixed at one end. The hollow element had two notches cut on the outer surface as shown in Fig.44. In order to test if this would in any way influence the grip, a force fit/

Fig. 44 Lever Arrangement in Torque Tests.



fit assembly with rape oil as a lubricant was made with the same hollow element before and after notches had been cut on the outer surface. The load diagrams plotted on a base of bore length, shown in Fig.44, suggest that the gripping power of the assembly was unaffected. The lever arrangement, shown in the same Figure, was made to engage with the notches cut in the hollow element without transmitting any appreciable load to the outer surface. The pull at the free end was applied by pulley blocks and the value, when slip occurred, was accurately determined by the sudden drop in the spring balance reading. Clean mating surface conditions were obtained in each case by the methods described in previous tests.

(I) Tests to investigate Torsional Slip: Three tests were carried out with the same pin and collar having a fit allowance of 0·5 thousandth inch. A fit allowance of this order will induce a radial pressure intensity at the mating surfaces of 1·68 tons per sq.in.

Test G. - The force fit assembly was made with a rape oil film on the mating surfaces.
The pull was applied at 60 inches from the axis of rotation and initial slip took place when a pull of 90 lb. was recorded on the spring balance. The pull was increased to 95 lb. before further slip occurred.
The back pressure test was carried out on the Amsler machine and a load of 2·6 tons was recorded when slip occurred.
The load diagram on a base of bore length for the complete withdrawal of the solid element is given in Fig.45.
There was no change in the mating dimensions, after separation, and the surfaces were found to be in a very good condition. This allowed of a further assembly of the elements.

Test H. - For the shrink fit assembly the elements were re-assembled with a rape oil film on the mating surface of the solid element.

Initial/

Fig.45 Back Pressure after Slipping by Torque.

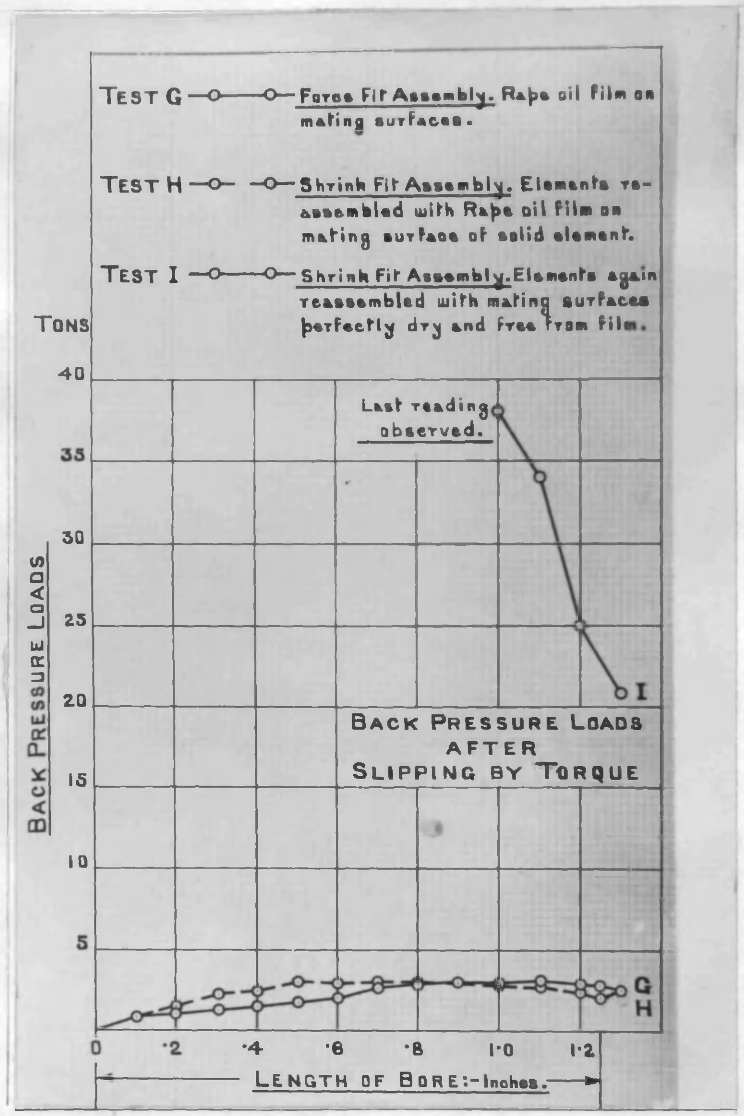


Fig.46 Torsional Slip Elements.

Initial slip took place when a pull of 100 lb. was recorded on the spring balance. Further slip was possible, however, with a pull of 80 lb.

A back pressure of 2.6 tons was required to induce initial slip. The load diagram for the complete separation of the elements is given in Fig.45.

No change was found in the mating dimensions after separation but there were a few thin grooves along the surface of the hollow element. The solid element had still a very good mating surface. An examination of the hollow element indicated that the surface, although bearing a few score lines, would allow a further assembly to be successfully made.

Test I. - The elements were again re-assembled as a shrink fit with the mating surfaces perfectly dry and free from film.

Initial slip took place when a pull of 260 lb. was recorded on the spring balance.

The back pressure test was carried out on the Amsler machine, and although a slight slip in the pendulum was noticed at 12·8 tons there was no measurable slip between the elements within the capacity of the machine. The back pressure test was completed on the 100-ton testing machine and axial slip took place at a load of 20·8 tons.

The elements of this assembly, after separation, are shown in Fig.46. The mating surfaces were very badly torn, large pieces of each mating element being pulled out of position.

The torque that an assembly fit is capable of transmitting is based, by designers, on an assumed coefficient of friction. This value, unfortunately, varies over a wide range and may lead, if a low or "safe" coefficient is assumed, to excessive fit allowances and possible failure. Researches on force fits by C.F. MacGill[12] gave coefficients of friction for steel shafts

and/

and steel hubs which ranged from 0·077 to 0·33. In a series of tests by J.W. Baugher[13] in which the mating surfaces of steel elements had different degrees of finish, values of the friction coefficient varied from 0.03 to 0.25. He attributed the lower values to "yielding of the soft steel around the bore because of too large fit allowances". The same author found that the coefficients of friction for shrinkage fits were 25 per cent greater than the corresponding values for force fits made with the same shaft and hubs. The values given in Table 14 confirm a previous finding that the quality of the grip, and hence the coefficient of friction, is greatly dependent on a mating surface film condition. The coefficient of friction in Test I was based on the load which caused a slight slip in the pendulum of the testing machine, namely, 12.8 tons.

TABLE 14.

Test.	Assembly Method.	Surface Condition	Torsional Slip spring balance reading.	Axial Slip tons.	Coefficient of friction.	
					Torsional	Axial.
G	Force Fit	Rape oil film	90 lb. at 1st slip 95 lb. at 2nd slip.	2.6	0.3	0.263
H	Shrink Fit	Rape oil film	100 lb. at 1st slip. 80 lb. at 2nd slip.	2.6	0.29	0.263
I	Shrink Fit	Perfectly dry and free from film.	260 lb. at 1st slip. 260 lb. at 2nd slip.	12.8 (slight slip of pendulum) 20.8	0.906	1.295

CONCLUSIONS/

(a) The effects of an out-of-straightness or bearing-band surface condition suggest that a high degree of accuracy in the generation of cylindrical mating surfaces is more important than the nature of the surface finish.

(b) The load to produce slip in a force fit assembly of elements is greatly influenced by the nature of the lubricant. Results give evidence of an increase in the load of 166 per cent and suggest that the resistance to axial slip is not altogether a question of the fit allowance.

(c) The "skin" or film, if a fluid, which may gather on the mating surfaces before assembly, may be removed without the use of solvents, and the resistance to slip on assembly is controlled by the nature of the lubricant.

(d) The assembly method may not determine the quality of the grip. Results show that this is greatly dependent on the surface film condition which may separate the mating surfaces after assembly. Such surface films, although more likely to be present under certain conditions of assembly than in others, may be equally operative under all three.

(e) A successful shrinkage assembly may be established with a fit allowance producing stresses well within the elastic range of the material, if the mating surfaces are perfectly dry and free from film.

(f) There is no evidence that the "torsional coefficient of friction is greatly in excess of the axial coefficient of friction" as sometimes quoted.

PART IV.

FORCE FIT PRACTICE AND LARGE SCALE INVESTIGATIONS

1 WHEEL ASSEMBLY PRACTICE

(a) General Specification and Assembly of Railway Wheel.

As an illustration of force fit practice, the assembly of wheel centres on axles for the standard 12-ton mineral wagon will be considered. The axles are made either of acid open-hearth or basic open-hearth steel with an ultimate tensile stress of 35 - 40 tons per sq.in. The wheel seat, according to specification, must be turned with a smooth finish, and be either parallel or having a taper of not more than 3/1,000 inch. The boss of the wheel centre must be bored with a smooth finish, and the hole and wheel seat coated with a suitable lubricant, preferably pure rape oil. The wheel centre, when untyred, is forced on to its seat by hydraulic power with a pressure of not less than 60 tons and not more than 80 tons. For tyred wheel centres it is assumed that a load is imparted to the wheel boss by the shrinkage of the tyre on the rim and an extra 10 tons in each case is allowed. It is important to note that in this assembly the Railway Clearing House specify a minimum and maximum axial tonnage, but make no reference whatever to a force fit allowance for the mating elements.

The axle suitably supported for entering and placing the wheel centres in position is shown in Fig47. When the two wheel centres are pressed "home" or into position, as shown in Fig.48, special regard must be paid to correct distance between the wheel flanges and also to the setting relative to the journals. Gauges are supplied for checking the distances.

A back pressure test is seldom asked for by home railways. It is, however, specified by some foreign railways, in which case a definite back pressure, as a rule less than that required/

Fig.47 Axle for Standard 12- ton Mineral Wagon.

Fig.48 Wheel Centres assembled on Axle.

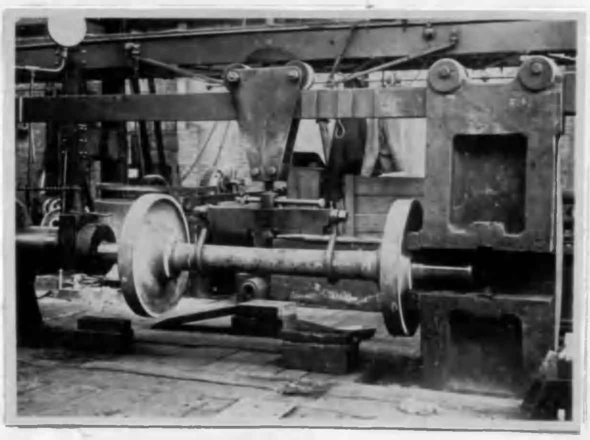
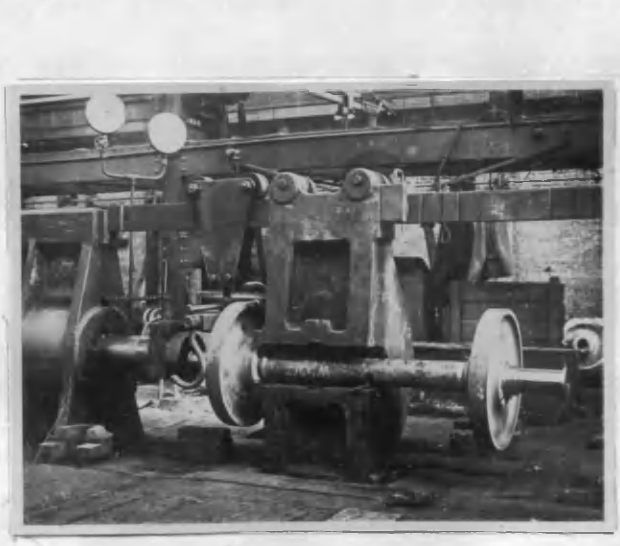


Fig.49 Back Pressure Test of Force Fit Assembly.



required during pressing on, is applied. Such back pressure values provide evidence of the success or failure of the force fit assembly but they may fail as a check on the pressing-on tonnage. On the other hand, the specified pressing-on tonnage can be obtained by those who operate the press by methods which may result in a reduction in the grip between the mating elements as demonstrated by low back pressures. The wheel shown in Fig.49 was subjected to an instantaneous back-pressure of 67 tons, equal to the maximum pressing-on tonnage, and gave no signs of slip.

An automatic recorder, Fig.50, gives a continuous record of the load obtained during the pressing-on operation. If there is an appreciable amount of friction between the pen point and the paper the values obtained may be a little below or a little above those recorded on the pressure gauge during rising and falling pressures respectively. This difference may amount to as much as 2 or 3 tons and hence a very fine adjustment of the recording pen is most important. The pressure records for each axle are kept and submitted for inspection.

The diagram to the left, shown in Fig.51, is the continuous record of the tonnage necessary to press home a wheel centre. The diagram to the right is a record of the tonnage during the pressing-off operation. A maximum value of 74 tons was recorded in making the assembly and a pressure of 77 tons was required to bring about initial slip when subjected to back-pressure.

(b) <u>Tram Car Wheel Assembly, General Specification</u>.

Glasgow Corporation Transport Department, which has had a great measure of success in operating a tramcar service, specify that the axles shall be made from Siemens acid steel with/

Fig.50 Automatic Pressure Recorder.

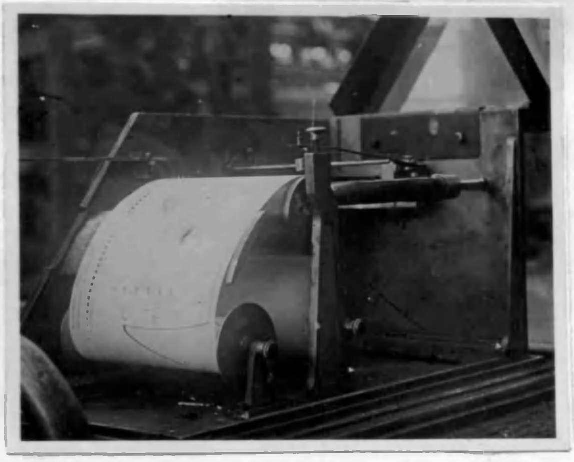
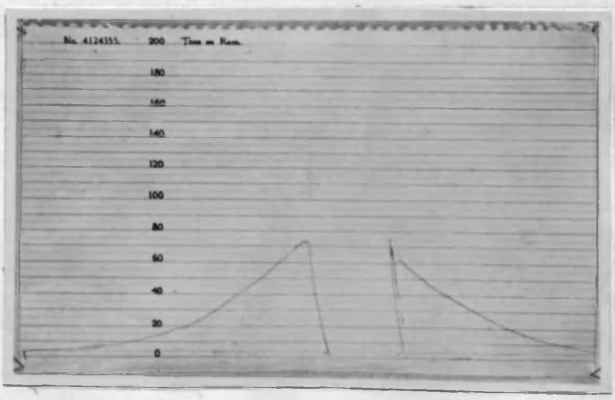


Fig.51 Assembling and Dismantling Pressure Diagrams.



with an ultimate tensile stress of 40 - 50 tons per sq.in. The axles are to be finished to size by grinding. The wheels must be pressed on the axles by hydraulic power after the tyre is fitted to the centre. The pressure used must be 35/40 tons for the wheels and gear wheels, and the tonnage recorded on the press must be stamped on the bosses of the disk centres and gear wheels.

It will be noticed that a minimum and maximum axial tonnage is specified. There is no reference, however, either to the fit allowance or to the nature of the lubricant that may be used in making the assembly.

The assembly method is similar to that adopted by the railways.

2. LARGE SCALE INVESTIGATIONS

The back pressure required to cause axial slipping of the wheel on its seat is directly proportional to:-

(1) The radial pressure intensity induced at the mating surfaces.

(2) The coefficient of friction between the surfaces in contact.

(3) The area of the surfaces in contact.

The back pressure may be controlled, therefore, by:-

(a) An out-of-roundness and an out-of-straightness of the wheel bore and wheel seat.

(b) The nature of the surfaces in contact.

(c) The quality of the wheel material as represented by its elastic properties.

(d) The nature of the surface contact film which separates the mating surfaces.

(e) The period of time which may elapse before dismantling.

The extent to which each of these may affect the back pressure may be gathered from an examination of railway wheel and axle assemblies, made to contract, and also of tramcar wheel and axle assemblies specially prepared for research work.

(a)/

(a) Effect of Machine Work on Grip.

Fig. 52 gives the principal dimensions of the assembly elements, the disk wheel centre and axle of the standard 12-ton mineral wagon. The dimensions of importance in connexion with the force fit are: length of wheel centre boss, 7 inches; inside and outside diameters of boss, $5\frac{1}{2}$ inches and $9\frac{1}{4}$ inches respectively. The wheel seat, it will be noticed, is machined over a length of $8\frac{1}{2}$ inches. It may be mentioned in passing that, according to specification, the journals of the axle must be truly circular and parallel, and afterwards burnished with a revolving tool or by some other approved method.

(I) Out-of-Roundness and Out-of-Straightness of Mating Surfaces. A mild steel wheel centre, Fig. 53 was one of many which had failed to satisfy a back pressure test and was removed from its seat. The dimensions of the wheel bore and wheel seat were then measured at right-angles, to the nearest half-thousandth of an inch, by inside and outside micrometers on planes $\frac{1}{2}$-inch apart. This was done in order to determine the degree of roundness and straightness of the wheel bore and wheel seat, and to obtain the residual force-fit allowance between the mating elements. An examination of the dimensions given in the first two lines of the tabulated readings indicates that the wheel seat was not truly circular at the planes considered. There is a maximum out-of-roundness of 3·5 thousandths of an inch at planes 9 and 13. The dimensions given in the two lines below indicate an out-of-roundness condition also in the wheel bore, reaching a maximum of 2·5 thousandths of an inch at planes 1 and 14. This results in a variation in the interface radial pressure intensity on assembly. The pressing-on load will thus be influenced by the chance setting of the wheel centre in assembly/

Fig.52 Disk Wheel Centre and Axle for Standard 12- ton Mineral Wagon.

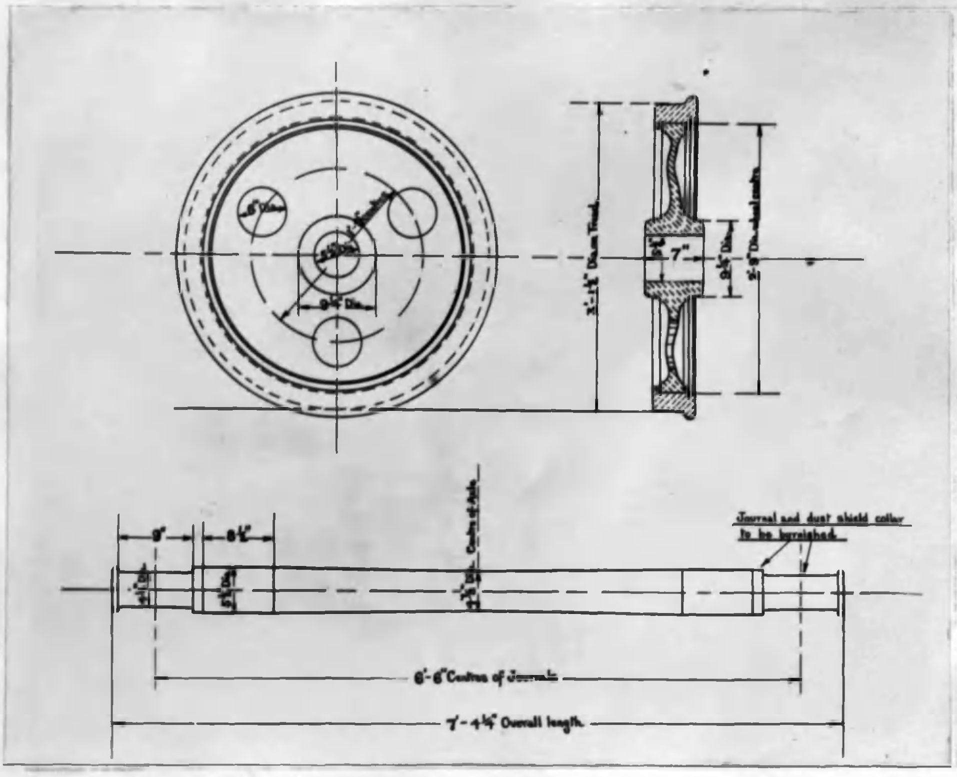
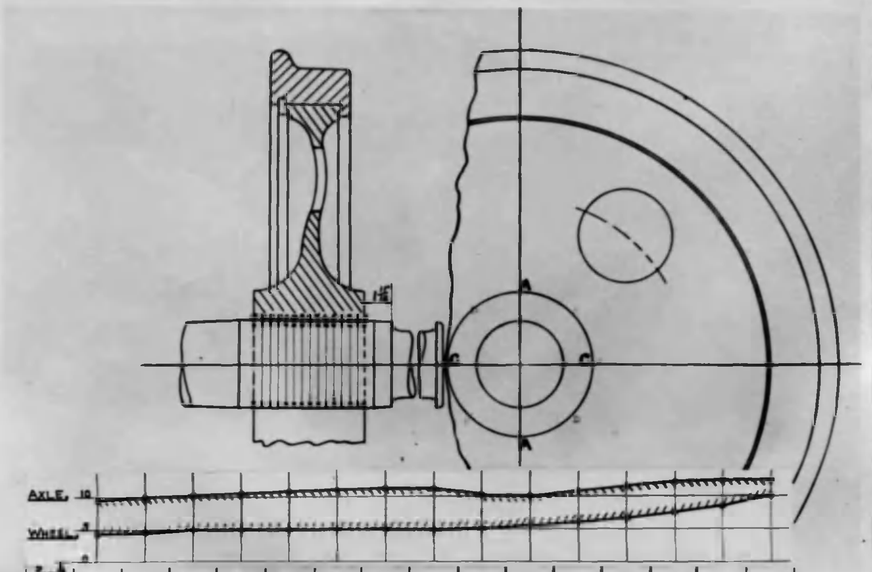


Fig.53 Mild Steel Wheel Centre Assembly which failed to satisfy Back Pressure Test.



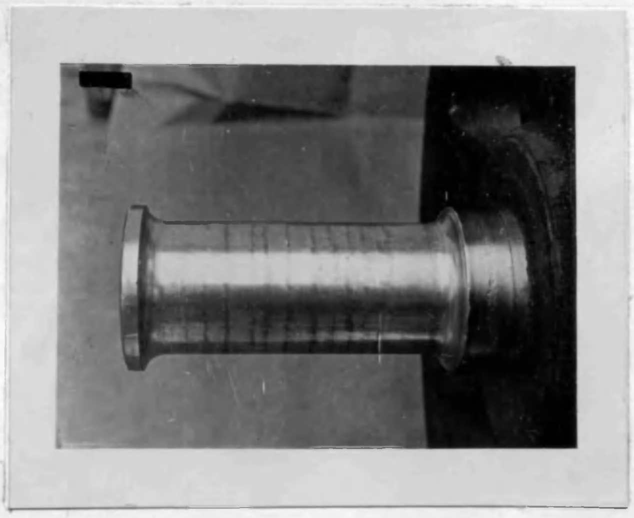
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	
A A	5·5110	·5115	·5115	·5120	·5120	·5120	·5125	·5125	·5120	·5115	·5120	·5130	·5140	·5140	·5140	AXLE
C C	5·5080	·5085	·5090	·5090	·5095	·5100	·5100	·5100	·5085	·5085	·5095	·5100	·5105	·5110	·5110	AXLE
A A	5·5055	·5055	·5060	·5060	·5060	·5060	·5060	·5055	·5060	·5060	·5065	·5070	·5085	·5095	·5110	WHEEL
C C	5·5030	·5035	·5040	·5040	·5040	·5040	·5040	·5040	·5040	·5050	·5055	·5060	·5065	·5070	·5090	WHEEL
MEAN DIFF.	·0052	·0055	·0053	·0055	·0058	·0060	·0063	·0065	·0053	·0045	·0048	·0050	·0048	·0042	·0025	

assembly. The larger diameters of the mating elements may coincide, they may be at right-angles, or at any intermediate setting. This may partly or wholly account for the appreciable variation in tonnage often recorded when elements of the "same" force-fit allowance are assembled. N. N. Sawin[9] found that the load was greatest when the larger diameters were at right-angles.

The out-of-straightness of the wheel seat and the wheel bore are shown graphically by plotting from a horizontal base line the average value of the readings taken at right-angles on planes ½-inch apart. The upper curve shows that the wheel seat has a pronounced hollow between planes 8 and 13, resulting in an out-of-straightness extending over a length of 2½ inches. The wheel bore, as regards straightness, would appear to have been well machined. There is certainly an out-of-straightness beginning at plane 9, but it rather indicates that the boss had received a permanent set. Assuming, therefore, the wheel bore to have been straight before assembly, and neglecting any possible loss of surface contact by overstraining the leading part of the boss, actual contact between the mating surfaces will have been less than the possible by about 36 per cent. The out-of-straightness of mating surfaces may be characterized by a series of hollow parts separated by ridges or bearing bands of varying width. The axle journal shown in Fig. 54 illustrates such a surface condition. This journal, after being turned and burnished, was lightly lapped.

(II) <u>Nature of Surface Finish</u>. The surface generated during the boring of the hollow element or the turning of the solid element is always a helical groove. The depth of the groove depends on the condition of the cutting edge and the feed of the tool. There are differences of opinion regarding/

Fig. 54 Axle Journal with Out-of-Straightness Surface.



regarding the effect of the finish of the mating surfaces. Ideal conditions would appear to be afforded by smooth surfaces, which would make possible the greatest amount of bearing contact between the mating elements. Surfaces that are not too smooth, however, are recommended by some as providing the best protection against axial slip. Others suggest that the tops of the little ridges of such surfaces will be crushed during shrinkage, or sheared off in a force fit operation, and that the back pressure would be less than for elements with smooth surfaces. There is evidence to show that the degree of smoothness of the surfaces resulting from different machining operations has practically no effect on the coefficient of friction and the force necessary to bring about slip.

Good modern machine-shop practice gives a degree of finish to mating surfaces which is more or less free from tool ridges. Many such surfaces, however, have a pronounced out-of-straightness. It is suggested, therefore, that for elastic grip assemblies greater attention should be directed to the need of a higher degree of accuracy in the generation of the cylindrical mating surfaces, a condition more difficult to attain than a high surface finish. This would be facilitated by carrying out the final metal removal operation by grinding.

(b) Effect of Wheel Material on Grip.

As has been mentioned, the Railway Clearing House[14] specify a tonnage instead of a fit allowance in pressing wheels on their seats. This tonnage is obtained in practice by a method of trial and error. Wheels are bored with a given fit allowance, based on experience of similar work, and the fit which gives the specified tonnage determines the dimensions for the mating assembly. The specification and the/

the method of conforming to it are both open to serious objection. If no consideration is given to the physical properties of the material the fit allowance may result in an overstressed boss and a diminished grip between the mating elements. Test pieces taken from the web of wheel centres must, according to the specification, show a tensile breaking strength of not less than 28 tons per sq.in. No reference is made to the elastic limit or yield point of the material. Yet the stress-strain proportional limit is the factor which should determine the fit allowance and the suitability of the material for an elastic grip assembly.

(I) Tensile Tests of Specimens from Bosses of Wheel Centres: In order to get some idea of the quality of the material surrounding the bore, four tensile test pieces were machined from each of the bosses of wheel centres made from eight different casts. Test pieces 1 and 3, Fig. 55 were taken circumferentially from the wheel boss and at right-angles to one another. Test pieces 2 and 4 were taken from the boss parallel to the wheel bore and at right-angles to one another. The average values obtained from the four tests of each cast show a great variation in the quality of the steel. Values as low as 24·38 tons per sq.in. were obtained, and in three casts only did the material give a minimum strength in excess of 28 tons per sq.in.

TEST RESULTS/

TEST RESULTS

TABLE 15 - Test Pieces from Bosses of Wheel Centres.

(Tensile Breaking Strength in tons per sq.in.)

Cast Number	Test Piece				
	1	2	3	4	Average
1	29.6	28.52	28.67	29.0	28.95
2	25.72	25.60	25.5	26.0	25.71
3	25.86	26.26	26.16	26.02	26.08
4	27.68	26.93	27.52	27.20	27.33
5	28.27	28.80	29.0	28.77	28.71
6	24.55	24.29	24.75	23.91	24.38
7	29.33	28.31	29.39	29.18	29.05
8	27.10	24.47	27.00	26.61	26.30
Average	27.26	26.65	27.25	27.09	

(Percentage Elongation on 2 inches)

Cast Number	Test Piece				
	1	2	3	4	Average
1	34.0	20.00	29.25	21.0	26.06
2	40.0	32.00	41.00	28.5	35.40
3	32.0	30.50	39.50	34.25	34.06
4	34.5	15.00	27.0	22.25	24.69
5	17.0	22.50	31.5	21.80	23.20
6	30.0	18.0	38.0	21.50	26.9
7	29.5	15.75	35.0	24.25	26.12
8	33.5	10.00	29.0	16.50	22.25

The typical load-extension diagrams in Fig.55 indicate that the yield point of the material was not clearly defined. The similarity in nature and shape of the curves for test pieces 3 and 4 taken from casts Nos. 5 and 4 respectively should be noted. It provides evidence that the material at certain points of the boss has been influenced by the mechanical processes in manufacture. It is apparent that the material had a low elastic limit and yield point. Test readings gave elastic limit values of 6.4 to 9 tons per sq.in. and yield point values of 11 to 16.04 tons per sq.in.

(II)/

Fig.55 Load-Extension Diagrams for Specimens from Bosses of Wheel Centres.

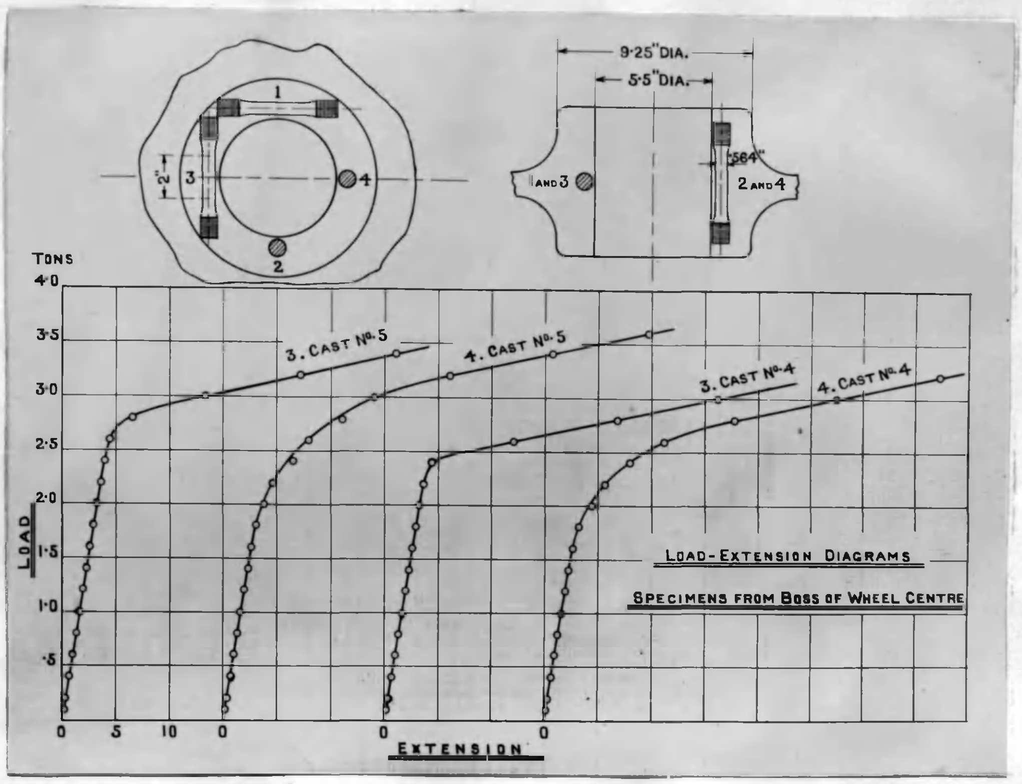
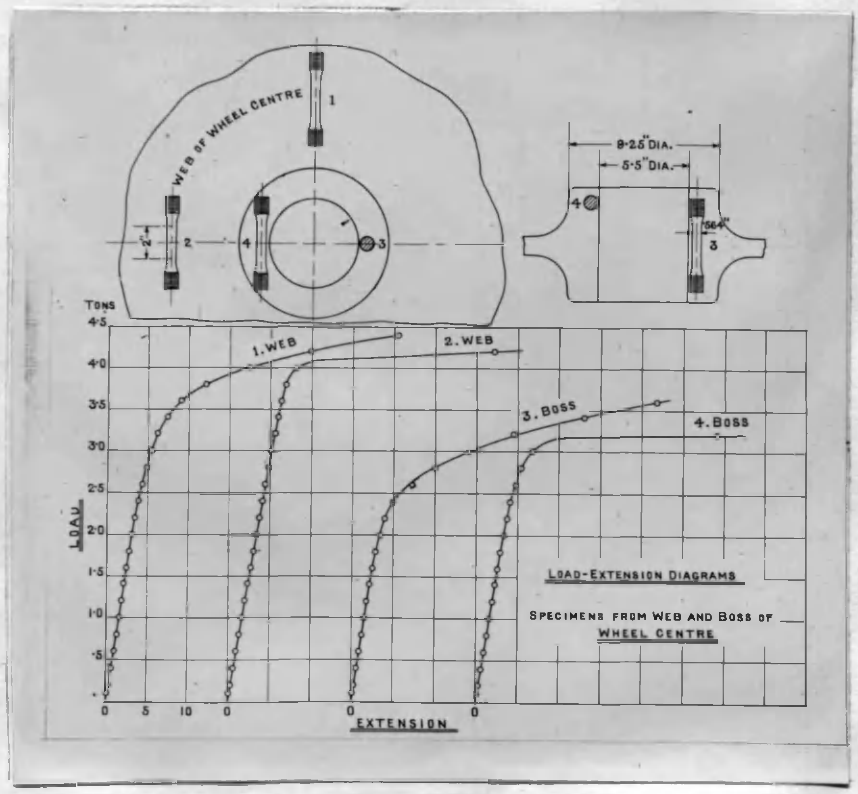


Fig.56 Load-Extension Diagrams for Specimens from Web and Boss of Wheel Centre.



(II) Tensile Tests of Specimens from Web and Boss of Wheel Centre. The author suggests that in specifying that the test piece is to be taken from the web of the wheel centre a part has been selected which will give higher elastic limit and yield point values than those obtained from the wheel boss. Doubt may be expressed as to which of these two parts is more severely stressed under running conditions. There can be none, however, regarding the part which is weakest due to manufacturing processes. In order to get some idea of the extent by which the elastic limit and yield point of the web material had been raised by rolling, tensile tests were carried out on test pieces taken from the web and boss of a wheel centre. This centre was not from any of the casts previously considered.
The positions from which the test pieces were taken, together with the load-extension diagrams of the material are shown in Fig.56. Values of the tensile breaking strength, elastic limit, and yield point are given in Table 16. It is evident that a force fit allowance based on the results obtained from test pieces taken from the web will lead to an excessive over-strain of the wheel boss and a consequent diminished grip between the elements on assembly.

TEST RESULTS.

TABLE 16 - Test Pieces from web and boss of wheel centre.

Test Piece.	Tensile Breaking Strength, tons per sq.in.	Elastic Limit, tons per sq.in.	Yield Point tons per sq.in.
1 Web	31.6	9.2	Not clearly defined 16.8
2 Web	30.9	11.8	16.8
3 Boss	29.6	6.0	Not clearly defined 12.8
4 Boss	29.5	7.2	12.8

The/

The Glasgow Corporation Transport Department specify that the test piece be taken from the rim of the wheel centre. It is suggested that test pieces taken from the rim will give higher elastic limit and yield point values than those obtained from specimens taken from the wheel boss.

(c) Effect of Fit Allowance on Grip.

(I) Railway Wheel Assembly. This particular assembly was made with a force-fit allowance of from 10/1,000 to 12/1,000 inch, and to give some idea of the magnitude of the stresses induced in the boss by different fit allowances, Table has been compiled. Substituting for R and r in equation (1), the interface radial pressure intensity is given by

$$p_r = 4322\frac{S}{d}$$

and the hoop stress at the bore layer by

$$f_r = 2.09\,p_r$$

Elastic failure condition under the combined stress due to p_r and f_r is given by

$$2.09\,p_r = f_L \text{ (Maximum principal Stress theory)}$$

$$2.39\,p_r = f_L \text{ (Maximum principal strain theory)}$$

$$3.09\,p_r = f_L \text{ (Maximum shear stress theory)}$$

$$2.57\,p_r = f_L \text{ (Maximum strain energy theory)}$$

TABLE 17 - Stresses Induced by Various Fit Allowances.

Force-fit Allowance.		Radial Pressure.	Hoop Stress.	Principal Stress Theory.	Principal Strain Theory.	Shear Stress Theory.	Strain Energy Theory.
S/d thousandths of an inch per inch dia.	S thousandths of an inch.	p_r tons per sq.in.	f_r tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.
0.75	4.13	3.242	6.78	6.78	7.75	10.02	8.33
1.00	5.5	4.322	9.03	9.03	10.33	13.36	11.11
1.25	6.88	5.403	11.29	11.29	12.91	16.69	13.89
1.50	8.25	6.483	13.55	13.55	15.50	20.04	16.66
1.75	9.63	7.563	15.81	15.81	18.07	23.37	19.44
2.00	11.0	8.644	18.06	18.06	20.65	26.71	22.21
2.25	12.38	9.725	20.32	20.32	23.25	30.05	25.00

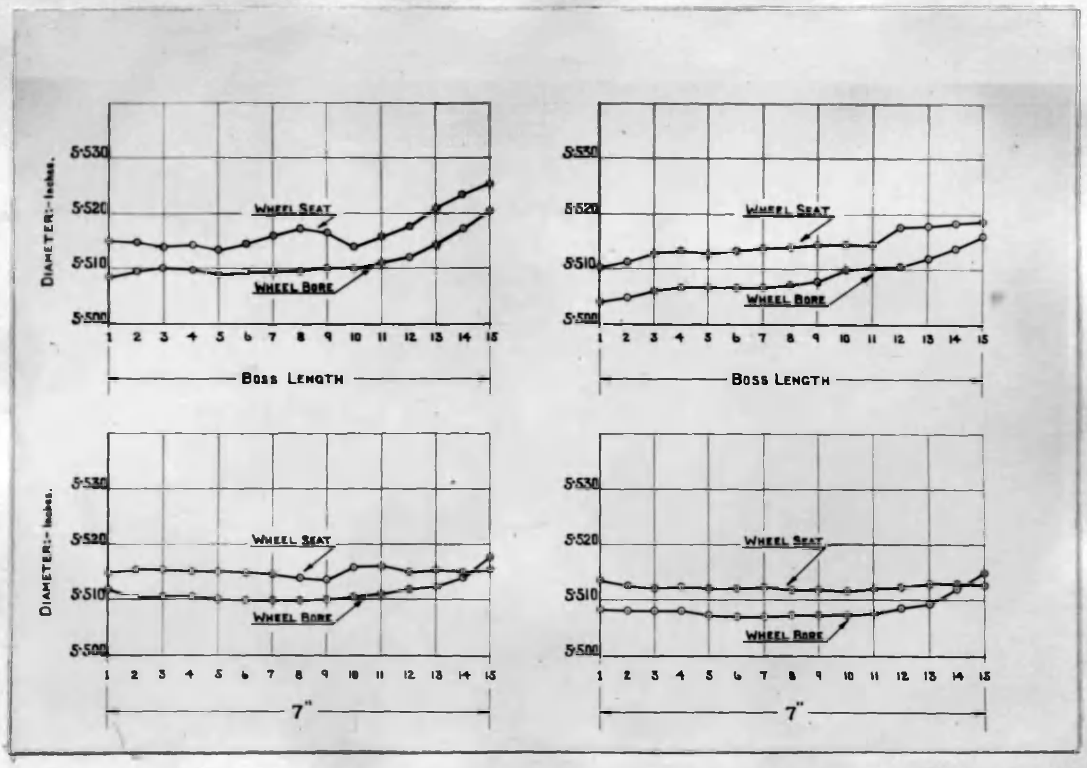
It will be observed that with a force-fit allowance of 1.75 thousandths of an inch per inch diameter, which gives approximately the minimum difference in the free diameters of the mating elements in this assembly, the elastic limit of the material has been greatly exceeded. On a strain energy basis, which has been found by Cook and Robertson[15] to meet failure conditions most closely for ductile materials, the elastic limit would require to be not less than 19.44 tons per sq.in. This is a figure associated with a steel of high quality and is, indeed, even much in excess of the yield point of the material, which had an average value of 13.27 tons per sq.in. There can be no doubt that the assembly has been made under plastic flow conditions which may penetrate well into the wall thickness of the wheel boss and lead to a condition of permanent set. If so it would be reasonable to expect a low back-pressure tonnage and evidence of an overstressed boss on dismantling. A number of the wheels were put under test and many low back-pressures were recorded. After the wheels had been removed from their seats an examination of the surfaces of many mating elements revealed conditions such as those illustrated in Fig.57.

Careful measurements, made as already described, gave an average difference in the free diameters of the mating elements, or residual force-fit allowance, of about 5/1,000 inch. This is approximately one-half the assembly fit allowance of from 10/1,000 to 12/1,000 inch. A similarity of the surface form of the hollow element to that of the solid element gave, in many cases, further evidence of an over-stressed boss condition.

The overstrain of this element in some cases resulted in the bore diameter at the leading face of the boss being bigger than the wheel seat by 1/1,000 to 2/1,000 inch. A pronounced out-of-roundness and out-of-straightness of the solid elements were also observed.

(II)/

Fig. 57 Conditions after Removing Wheels from their Seats.



(II) Tramcar Wheel Assembly. An overstressed condition of the wheel boss would appear to be a feature of all wheel assemblies used in rail transport equipment. Further evidence of this is provided by a consideration of the fit allowances which may be adopted in making the disk tyred wheel and axle assembly as fitted to the Glasgow Corporation tramcar.

In this assembly the outside and inside diameters of the wheel boss are 8 inches and $4\frac{1}{2}$ inches respectively. Substituting for R and r in equation (1), the interface radial pressure intensity is given by

$$p_r = 4588\frac{S}{d}$$

and the hoop stress at the bore layer by

$$f_r = 1.93\ p_r$$

Elastic failure condition under the combined stress due to p_r and f_r is given by

$$1.93\ p_r = f_L \text{ (Maximum principal Stress theory)}$$

$$2.23\ p_r = f_L \text{ (Maximum principal Strain theory)}$$

$$2.93\ p_r = f_L \text{ (Maximum Shear Stress theory)}$$

$$2.425\ p_r = f_L \text{ (Maximum Strain Energy theory)}$$

TABLE 18 - Stresses Induced by Various Fit Allowances.

Force Fit Allowance.		Radial Pressure.	Hoop Stress.	Principal Stress Theory.	Principal Strain Theory.	Shear Stress Theory.	Strain Energy Theory.
S/d thousandths of an inch per inch dia.	S thousandths of an inch.	p_r tons per sq.in.	f_r tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.	f_L tons per sq.in.
0.75	3.38	3.442	6.64	6.64	7.68	10.09	8.35
1.00	4.5	4.588	8.86	8.86	10.23	13.45	11.13
1.25	5.63	5.736	11.08	11.08	12.79	16.81	13.91
1.50	6.75	6.884	13.28	13.28	15.35	20.17	16.69
1.75	7.88	8.03	15.50	15.50	17.91	23.53	19.47
2.00	9.00	9.176	17.71	17.71	20.46	26.89	22.25
2.25	10.13	10.32	19.93	19.93	23.02	30.26	25.03

This particular assembly, using neatsfoot oil as a lubricant, required a force fit allowance of 9/1,000 inch to obtain the desired tonnage of 45 to 50 tons. It will be observed from Table 18 that a force fit allowance of 2 thousandths of an inch per inch diameter would result in the elastic limit of the material being greatly exceeded. On a strain energy basis the elastic limit of the material would require to be not less than 22.25 tons per sq.in. This value is even much in excess of the yield point of the wheel material which could be regarded as being in the region of 14 to 17 tons per sq.in. The repair assembly tonnage at Coplawhill Works is 10 tons greater than specified tonnage.

(d) Effect of Lubricant on Grip.

The use of a lubricant is an essential requirement to a successful mating of the elements in a force fit assembly. It is surprising, therefore, that, in spite of the importance of such assemblies in engineering practice, little or no attention has been given to the influence of this factor on the axial resistance of the elements to slip, or to the quality of the grip established. Any reference to the use of a "suitable lubricant", in a railway specification, is of little value without a knowledge of what constitutes a suitable lubricant, either in terms of the film resistance during assembly for a given fit allowance, or in terms of its influence, if any, on the resistance of the elements to slip when operating under running conditions after a period of years. Glasgow Corporation Transport Department make no reference to the use of a lubricant in their specification for wheel and axle assemblies.

The range investigated and the magnitude of the film resistances experienced in making the force fit assembly of small elements suggested the importance of carrying out further tests to investigate whether the contact film resistance/

resistance could be regarded as an important factor on the tonnage necessary to make an assembly of large size elements. The Author would acknowledge his great indebtedness to the General Manager of the Glasgow Corporation Transport Department for granting facilities to carry out such a series of tests at the Corporation's Coplawhill Works. The elements selected for assembly consisted of the 2'-3" diameter disk tyred wheel and axle as fitted to the Corporation tramcars.

Figure 58 gives the principal dimensions of the hollow element. Neglecting the possible stiffening effect of the web, which may be regarded as small, the dimensions of importance in connection with the force fit assembly are: length of wheel centre boss, $4\frac{1}{2}$ inches; inside and outside diameters of boss, $4\frac{1}{4}$ inches and 8 inches respectively. This inside diameter was adopted so that wheels used in carrying out tests could subsequently be rebored to a diameter of $4\frac{1}{2}$ inches for actual assembly purposes.

It was decided to assemble and dismantle the elements using a wide range of lubricants which would include those commonly used in railroad assembly practice and turbine work.

(I) Conditions Observed in Tests. A successful investigation of the film resistance in a series of force fit assemblies is dependent on two main conditions:

(a) a constant difference in the free diameters of mating elements.

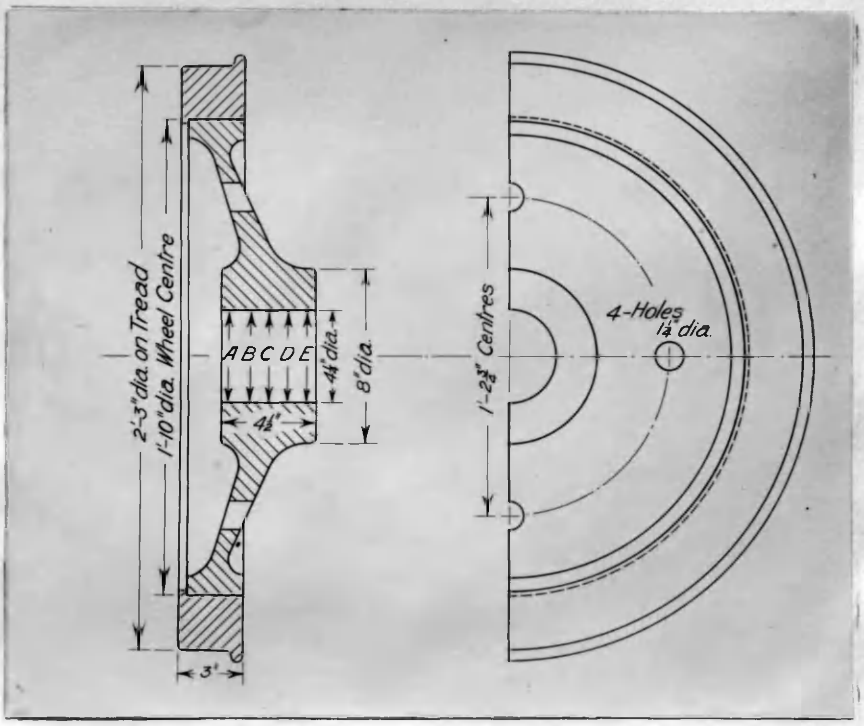
(b) similar surface conditions giving equal bearing contact.

An attempt to meet these conditions was made by using the same two assembly elements throughout the complete series of tests. This in turn necessitated the following requirements:

(1) a high degree of accuracy in machining the mating surfaces resulting in a good surface finish and a near approach to perfect roundness and straightness of the wheel bore and wheel seat.

(2)

Fig. 58 2'-3" Dia. Tramcar Disk Tyred Wheel.



(2) a relatively small difference in the free diameters of the mating elements.

(3) the making of each assembly with the same setting of the wheel boss relative to the wheel seat position.

The dimensions of the wheel bore and wheel seat were measured at right angles on planes A, B, C, D, and E, as indicated in Fig.58, ¼ inch in from each face and 1 inch apart. This was done to determine the degree of roundness and straightness of the wheel bore and wheel seat and to obtain the mean difference in the free diameters of the mating elements. These measurements were made before and after each test to check any possible change in the dimensions of the mating elements.

(II) Pressing Operation and Procedure: The wheel centre was forced on to its seat by hydraulic power and the press was regulated to operate at the same rate during each assembly. The mating surfaces were carefully cleaned with the use of solvents before each assembly of the elements. The surfaces, after drying, were immediately coated with a film of oil which was allowed to drain off partially before the elements were assembled. This was unnecessary in some cases due to the viscous nature of the lubricant used. Great care was taken to establish good entry conditions and to have, in each case, the same setting of the wheel centre, relative to the wheel seat. The pressing-on and back-pressure loads were read off a pressure gauge at wheel displacements of 0.5 inch.

(III) Tests to Investigate Effect of Lubricant. Nine tests were carried out with the same wheel and axle with a difference in the free diameters of the mating surfaces/

surfaces of 1.1 thousandth of an inch.

Test 1. Neatsfoot oil
Test 2. Bayonne oil
Test 3. Pure tallow
Test 4. Tallow and 10% white lead.
Test 5. Rape oil
Test 6. Rape oil plus 2% oildag.
Test 7. Graphited spindle oil plus 1% oildag.
Test 8. Petroleum jelly plus 1% oildag
Test 9. Mercurial ointment.

The fact that the primary conditions remained unchanged throughout the tests completely justified the method adopted of measuring the contact film resistance of different lubricants by the repeated assembly of the same two mating elements. The dimensions of the mating surfaces, measured before and after each assembly at right angles on five different planes, showed no signs of change. The mating surfaces, after each assembly, were carefully examined and gave no evidence of an improved bearing contact or even signs of the slightest hairline marking. The constant mating conditions, which were thus established, provide evidence of a high standard of machine work and of a successful fit allowance which gave a suitable degree of polish to the mating surfaces.

(IV) TEST RESULTS. Pressing-on load diagrams on a base of bore length are given in Fig.59. They are of a form and type similar to those obtained from laboratory experiments on small size elements and indicate, clearly, the remarkable influence of the surface contact film on the tonnage necessary to make a rail wheel force fit assembly. The values range from 6.0 tons with tallow and 10% white lead as a lubricant to 25 tons with Bayonne engine oil as a lubricant. This represents an increase of approximately 300 per cent. in the pressing-on tonnage due to a difference in the nature of the film which separates the surfaces during assembly.

The/

Fig. 59 Pressing-on Load Diagrams for Tests on Effect of Surface Contact Film.

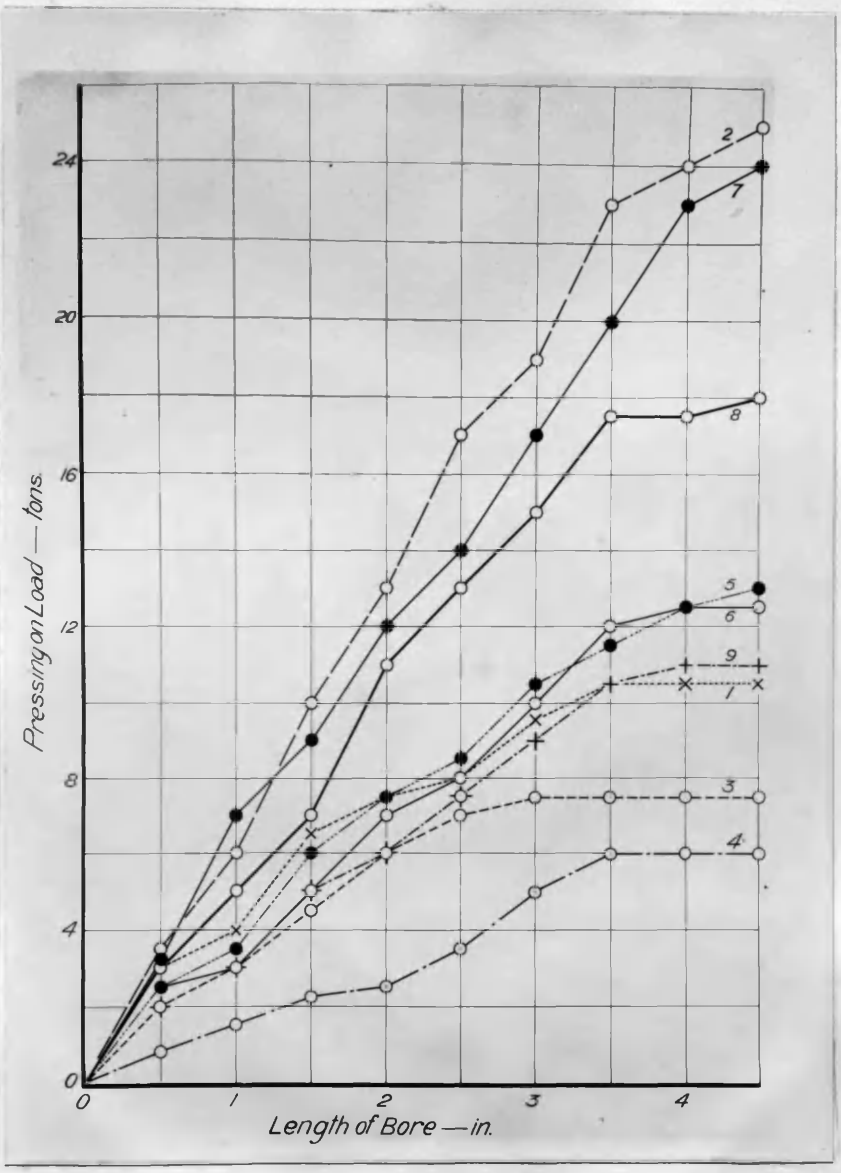
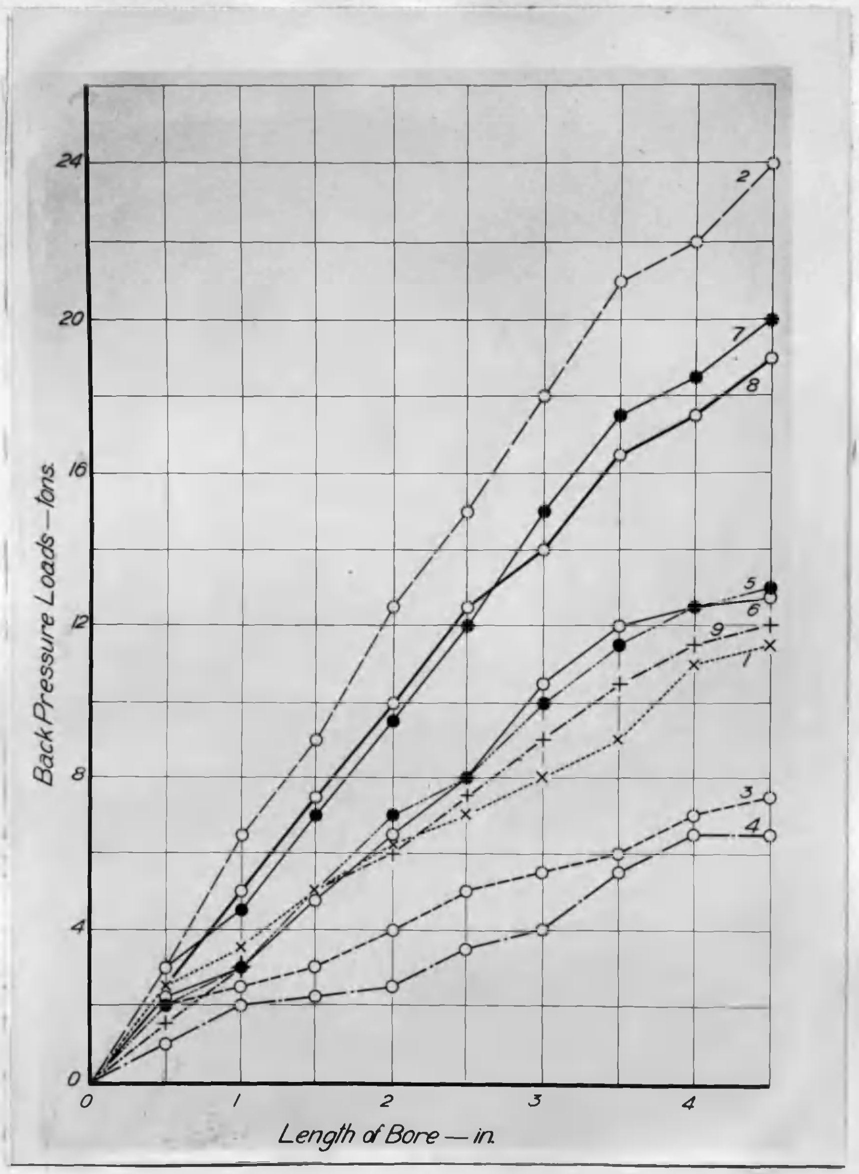


Fig. 60 Back Pressure Load Diagrams for Tests on Effect of Surface Contact Film.



The back-pressure loads on a base of bore length are shown in Fig.60. It will be seen that, in general, the characteristic influence of the film on the axial resistance of the elements to slip during assembly is equally operative throughout the dismantling process. Periodic slipping accompanied by a bumping noise characterized the mating of the elements when a lubricant not having an oiliness property was used. When mated with a lubricant of pronounced oiliness a gradual slipping of the elements without noise marked the assembly.

It follows that, for a given specified axial tonnage, an assembly in which a lubricant of pronounced oiliness is used will require the elements to be mated with a greater radial pressure intensity than when assembled with a lubricant which has not an oiliness characteristic. It is important to note that in railroad practice a force fit assembly is invariably made with a lubricant having a pronounced oiliness.

It is not surprising to find, therefore, that in all such cases a large force fit allowance is necessary to obtain the specified pressing-on tonnage. Under such conditions of assembly some qualities of steel may become overstrained. A plastic or rather semi-plastic range may penetrate well into the wall thickness of the hollow element, and thus impair the quality of the grip.

(e) Effect of Time on Grip.

Since the surface contact film had been found to be a factor which controlled, in a great measure, the axial resistance of elements to slip, it was considered desirable to examine its influence after a period of time. The problem of the back-pressure/

back-pressure tonnage required to dismantle elements a few years after having been assembled has, in general, necessitated the installation of larger and more powerful presses.
Time would appear to influence considerably the action of the film on the resistance of the elements to slip.
A lubricant which would establish surface contact film conditions over a long period of time would, in many cases, be most desirable.
It was decided to make four tramcar wheel and axle assemblies using four different lubricants, with a fit allowance in each case of approximately one half the fit allowance adopted in practice.

(I) Assembly Conditions and Test Results.

TABLE 19 - Time Effect on Grip of Force Fit Assembly.

Assembly No.	Fit Allowance Thousandths of an inch.	Lubricant	Max. Pressing-on Tonnage.	Back-pressure tonnage	
				After Assembly	After 8 months.
1	4.5	Neatsfoot oil.	32	38	41
2	4.7	Mercurial Oint.	21	23	49
3	5.0	Bayonne oil	46	48	52
4	4.0	Tallow & 10% White Lead.	16	17.5	35

The fit allowances shown in Table 19. were based on measurements taken at right angles on five different planes of the mating elements. The back-pressure value in each case was taken as the tonnage necessary to produce initial slip. This resulted in an axial displacement of the wheel on its seat of about 3/32 inch. Provision was made for a series of such displacements, in a time effect test, by having the wheel seat projecting 1 inch beyond the outer face of the wheel boss.
It/

It is shown that the tonnage to produce slip, after a period of 8 months, has increased considerably in assemblies 1 and 2 compared with 3 and 4. These tests, together with many on small size steel elements, are still continuing.

CONCLUSIONS:

(a) The degree of accuracy obtained in machining the mating surfaces of an elastic grip assembly is of much greater importance than the nature of the machine finish of the surfaces.

(b) The materials used in such assemblies should be specified by their elastic properties. The test pieces used for this purpose should be taken from that part of the element where the material, due to the particular method of manufacture, is weakest. Railway practice in selecting the web of a wheel centre is shown to be unsatisfactory as failing to provide reliable data on which the force fit assembly of wheel centres may be made.

(c) Excessive fit allowances producing plastic flow conditions in the wheel centre or hollow element are shown to produce a condition of overstrain leading to failure of the grip.

(d) The axial tonnage required to make a force fit assembly cannot be expressed in terms of the fit allowance alone.

(e) The empirical formulae presented by various writers cannot be regarded as providing a satisfactory basis on which to fix a force fit allowance. The effects of the lubricant, the use of which is an essential requirement in the making of such assemblies, is not examined in terms of the quality of the grip established when such allowances are employed.

(f)/

(f) Current railway practice, which fixes the difference in the free diameters of the mating elements of a force fit assembly so that a load of 10 tons per inch diameter is required to press the elements into position, is open to criticism. It is conditioned by workshop methods based on past experience and custom and not on the results of a complete investigation of all the factors which are operative in the making of such assemblies. An over-stressed condition of the wheel boss, consequent to the use of a lubricant having a very pronounced oiliness which, in general, necessitates a large fit allowance, cannot be regarded as the ideal condition of material in an elastic grip assembly.

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