Some Factors Influencing the Life of Marine Crankshafts

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The author outlines the principal causes of reduced life in marine oil engine crankshafts and cites typical illustrative examples from service. Reported defects for the various types of crankshaft construction, taken over a ten-year period from January, 1953, are classified and certain conclusions drawn as to relative reliability. Some metallurgical factors are discussed and their influence illustrated.

Computer methods are used for evaluating crankshaft torque variations at any section along the shaft from gas pressure diagrams, including the effect of added torsional vibration stresses. A programme is applied to a large modern six-throw two-stroke cycle crankshaft of Rule size with a view to assessing the factors of safety against fatigue failure under combined bending and twisting with and without torsional vibration, taking into account stress concentration and notch sensitivity, together with an estimate of the effects of misalignment and axial vibration.

INTRODUCTION AND HISTORICAL

The idea of the crank is probably about as old as Christianity and originated, like many other benefits of our Western civilization, in China, where it was first used for such devices as water-powered bellows and rotary fans for winnowing husked rice.(1)

Development of cranked mechanisms was extremely slow and the idea lay dormant for many centuries. So far as Europe is concerned its probable earliest appearance is illustrated in the Utrecht psalter (9th century) (see Fig. 1), as a means of turning a grinding wheel for sword-sharpening purposes in the form of an overhung "simple" crank!

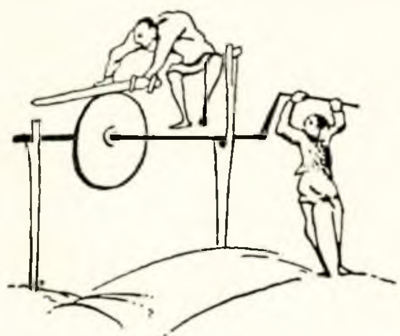

FIG. 1—*Sharpening a sword on a grindstone revolved by a crank handle. Reproduced from the Utrecht psalter (9th century)*

It was not, however, until the Middle Ages that the "compound" crank emerged in the form of two simple cranks "back to back", firstly in the carpenter's brace, possibly as early as the 13th century, and later for mill drives and treadle mechanisms.

The earliest examples of combined crank/connecting rod mechanisms are from the early 15th century. Fig. 2 (c. 1430 A.D.) shows a mill operated by a double-throw crank through a gear transmission . The two connecting rods are supposed to be linked by ropes to a pair of treadles (not clearly shown).

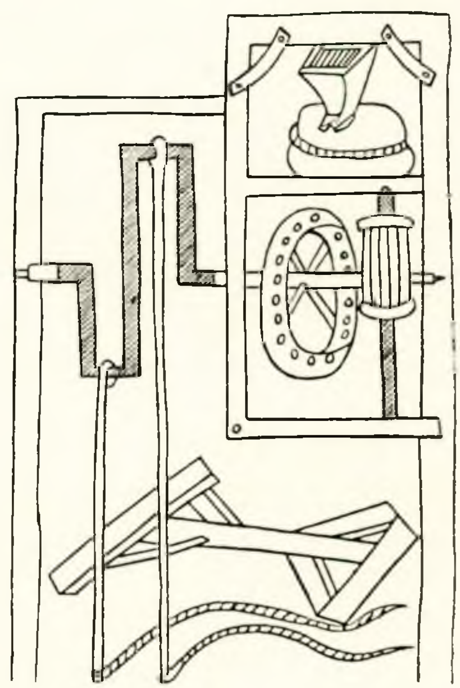

FIG. 2—*Mill operated by a double-throw crank through gear drive. From a manuscript of c. 1430*

The concept of the crankshaft for marine propulsion emerges, astoundingly, as early as the 14th century. Guido da Vigevano, an Italian physician, published in 1335 a treatise on military machines aimed mainly at the pagan foes of the

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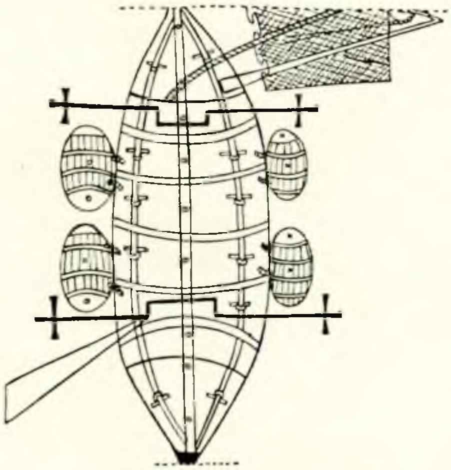


FIG. 3—*Crank-propelled paddle boat. From a manuscript of 1335*

Crusaders. Fig. 3, taken from this book, illustrates a two-shaft crank-propelled paddle boat (complete with buoyancy tanks, incidentally!). Whether it was ever built is doubtful, but the design shows amazing anticipation.

Although Leonardo da Vinci in the late 15th century contributed ideas for the use of the crank in various forms, including the drive for a screw-cutting lathe, progress was slow until the early nineteenth century with the dawn of the steam age and the advent of mechanical pioneers, such as Watt, Murdoch, Newcomen, Trevithick, Maudslay, Stephenson etc.

From the relatively crude cast or wrought-iron, single or double-throw cranks of those early days to the multi-throw, semi-built or fully built steel mammoths of today, weighing up to 150 tons and 30in. diameter or more, is indeed a far cry.

We have little idea of the service reliability of those pioneering crankshafts, but doubtless the wrought iron shafts, at least, gave a good account of themselves having regard to the excellent toughness and ductility of their material and the generally low stress levels under which they operated.

The era of the compound and triple expansion steam reciprocator from 30 to 100 years ago brought the marine crankshaft to a generally high level of reliability, mostly in the fully built "dumbell" form.

With the advent of the marine Diesel engine some 50 years ago, however, crankshaft troubles became increasingly prevalent and the bogey of torsional vibration reared its ugly head. These have been sufficiently described in the literature e.g. Porter, 1928,[2] Dorey, 1935 [3] and 1939.[4] With improved understanding of this phenomenon and moulded by the sharpened demands of the internal combustion cycle, marine crankshaft design has developed characteristically, but on the whole conservatively, until today its products can fairly be claimed for the most part to have reached a high level of reliability with due regard to efficiency of utilization of material.

In this progress, designers have benefited by advances in metallurgy, Dorey (1955),[5] and manufacturing methods, and reliability has also been served by the impressive modern developments in non-destructive testing.

PART I—SERVICE PERFORMANCE

STATISTICAL

In an attempt to examine the operational reliability of post-war built main propulsion heavy oil engine crankshafts in ships classed with Lloyd's Register*, the Society's records of reported defects for a ten-year period, 1953 to 1962 inclusive, have been scrutinized and analysed in respect of engines of over 1,000 b.h.p. The analysis covers the incidence of cracked, broken or "slipped" shafts (but does not include cracks in couplings) classified in relation to ranges of horsepower, types of crankshaft construction, position of machinery in the ship and location of fracture, and has been given for both two-stroke

* The term " classed " throughout the paper refers to vessels classed with Lloyd's Register of Shipping.

TABLE I.—NUMBERS OF TWO-STROKE AND FOUR-STROKE SINGLE-ACTING MAIN ENGINES BUILT TO LLOYDS' REGISTER CLASS 1947 TO 1962 INCLUSIVE (16 YEARS).

Type of construction		Cycle	B.h.p. per shaft 1,001–2,000	2,001–5,000	5,001–8,000	8,001–12,000	12,001–20,000	Total shafts
Forged	Solid	2-SC	413	121	7	—	—	541
		4-SC	516	76	—	—	—	592
	Semi-built	2-SC	77	254	301	90	19	741
		4-SC	12	1	—	—	—	13
	Fully built	2-SC	1	28	23	23	—	75
		4-SC	8	45	—	—	—	53
Cast	Solid (S.G.I.)	2-SC	56	4	—	—	—	60
	Semi-built (forged journals)	2-SC	44	176	378	122	20	740
		4-SC	—	1	1	—	—	2
	Fully built (forged journals and pins)	2-SC	25	231	287	173	10	726
		4-SC	8	47	—	—	—	55
Triple-crank units (combination type)		2-SC	9	416	406	35	—	866
Totals		2-SC	625	1,230	1,402	443	49	3,749
		4-SC	544	170	1	—	—	715
		All	1,169	1,400	1,403	443	49	4,464

TABLE II.—YEARS OF SERVICE WITH LLOYD'S REGISTER CLASS BETWEEN 1.1.53 AND 31.12.62 (10 YEARS) OF TWO-STROKE AND FOUR-STROKE SINGLE ACTING MAIN ENGINES BUILT 1947 TO 1962 INCLUSIVE (16 YEARS).

Type of construction		Cycle	B h.p. per shaft					Total service years
			1,001–2,000	2,001–5,000	5,001–8,000	8,001–12,000	12,001–20,000	
Forged	Solid	2-SC 4-SC	2,697 2,007	663 261	27 —	— —	— —	3,387 2,268
	Semi-built	2-SC 4-SC	457 90	1,828 4	1,641 —	402 —	22 —	4,350 94
	Fully built	2-SC 4-SC	4 70	162 420	119 —	99 —	— —	384 490
Cast	Solid (S.G.I.)	2-SC	191	24	—	—	—	215
	Semi-built (forged journals)	2-SC 4-SC	285 —	999 8	2,279 4	592 —	42 —	4,197 12
	Fully-built (forged journals and pins)	2-SC 4-SC	200 75	1,540 440	1,969 —	951 —	22 —	4,682 515
Triple crank units (combination type)		2-SC	80	3,060	3,147	187	—	6,474
Totals		2-SC 4-SC	3,914 2,242	8,276 1,133	9,182 4	2,231 —	86 —	23,689 3,379
		All	6,156	9,409	9,186	2,231	86	27,068

cycle single-acting and four-stroke cycle single-acting engines.

The rates of incidence have been expressed in relation to the total service years in class of the various categories of crankshaft over the ten-year period in question as follows:—

Rate of incidence = number of incidents per 100 shaft-years of classed service.

In assessing the relative performance of the various categories of crankshaft, as discussed hereafter, due regard should be had to the small numbers of defects in certain classes, especially when associated with a very low number of shaft-years. This can, of course, bring about large fluctuations in rates of incidence, a well known difficulty in statistical presentation. Nevertheless, it is hoped that at least certain broad trends will be discernible from the data given.

Table I classifies the total numbers of 2-S.C. and 4-S.C. single-acting main engines built to class over the 16-year period 1947 to 1962 inclusive. It will be noted that of the 2-S.C. engines about 70 per cent are included in the horse-power range, 2,000 to 8,000 b.h.p., whilst over 75 per cent of the 4-S.C. engines are in the power bracket, 1,000 to 2,000 b.h.p., a remarkable demonstration of the waning popularity and competitive power of the 4-S.C. engine for main propulsion purposes above about 2,000 b.h.p.

For 2-S.C.S.A. engine shafts the principal categories are solid forged, semi-built forged, semi-built cast, fully built cast and triple-crank (combination type) for all of which the totals are not widely dissimilar. The number of fully built forged shafts, so popular in the early long-stroke engines, is now very small.

Fig. 4 (a) to (n) illustrates some typical large modern crankshaft designs, the majority of which are of the semi-built type.

4-S.C.S.A. engine shafts are mostly solid forged and included in the horsepower range, 1,001-2,000 b.h.p.

Table II is arranged similarly to Table I, but gives the total service years in Lloyd's Register class of the various categories of crankshaft over the ten-year period, Jan. 1953 to Dec. 1962, for post-war built engines as indicated. The totals are broadly in much the same proportions as the corresponding numbers of crankshafts given in Table I, i.e. the average service years for this ten-year period will be similar for each of the main categories, say from about $5\frac{1}{2}$ to $7\frac{1}{2}$ years.

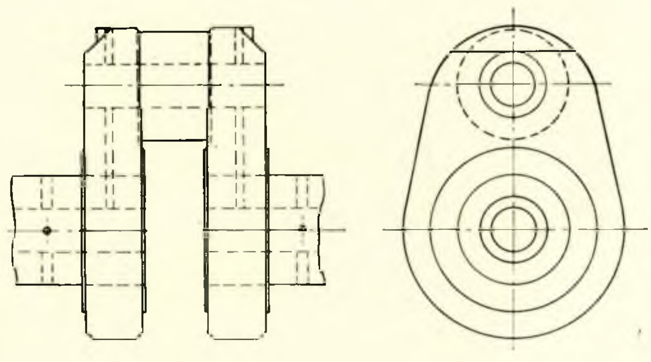
Table III is a "break-down" of the crankshaft records into types of defects with respect to 2-S.C. and 4-S.C., b.h.p. range, and types of construction. Of the main types the semi-built forged shafts show the best overall performance with 0·23 defects per 100 shaft-years service. There is very little variation between the rates for solid forged, semi-built cast and fully built cast webs, i.e. 0·36, 0·33 and 0·28, respectively, but the triple-crank, combination type of construction shows a higher rate at 0·74 defects per 100 shaft-years service. In any case, the overall average incidence, at 0·43 per 100 shaft-years, could justifiably be claimed as remarkably low, being equivalent to no more than one major defect every 25 years for a fleet of ten single-shaft ships.

The figures for the triple-crank (combination type) shafts are clearly heavily swollen by the incidence rate in the 8,001-12,000 b.h.p. range and partly also by that in the 5,001-8,000 b.h.p. range, since these embrace the defects in the six-cylinder, 750 mm. bore engines described by Atkinson and Jackson [25] in their 1960 paper before this Institute (*loc cit*). Omitting these engines, the resulting figures for the triple crank, combination type, shafts are shown in Table IV.

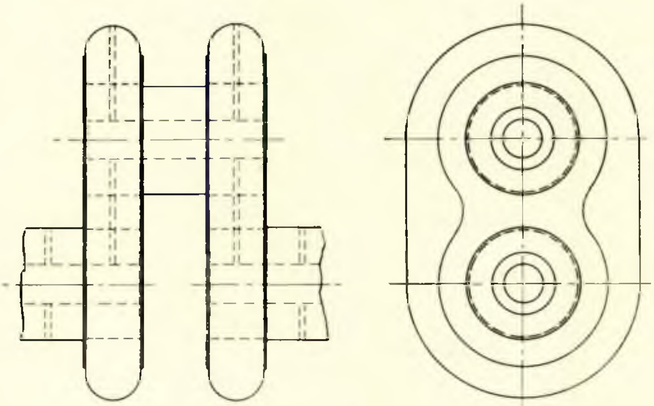
It will be seen that on this reckoning the performance of the triple-crank shafts is little different from that of other types so far as cracked and broken shafts are concerned. The main point of difference is the unusually high incidence of slipped shrinks in the 5,001 to 8,000 b.h.p. range, a good many of which resulted from leakage of cooling water into the cylinder during "standby", possibly influenced by inadequate maintenance in some cases. In the latest designs of these engines, the cooling water system has been so arranged as to eliminate the possibility of water leakage into the cylinders.[29]

A study of Table III for 2-S.C. engines would suggest that for cracked and broken shafts there may well be a trend towards increase of incidence with horsepower especially in the range from 8,001-12,000 b.h.p. This rising trend is particularly marked in the case of semi-built forged shafts. However, in considering the tabulated statistics it should be borne in mind that the really large-bore engines have so far logged only a comparatively small number of shaft-years, so that it would be premature to infer too much concerning these larger shafts from the figures presented.

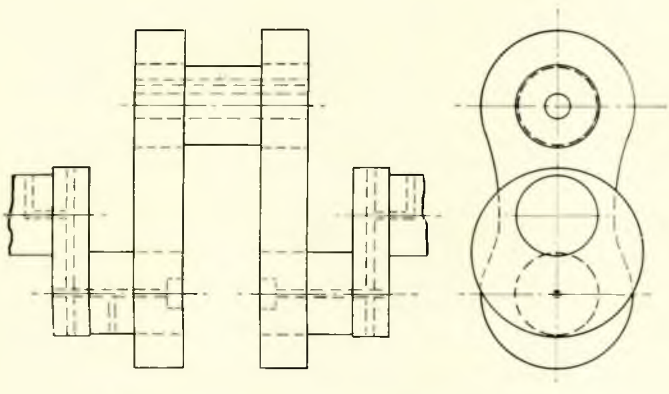
Of the total defects, about 80 per cent were cracked or broken, leaving slipped shrinks to account for the balance of some 20 per cent, mostly in the 2,001-8,000 b.h.p. range.



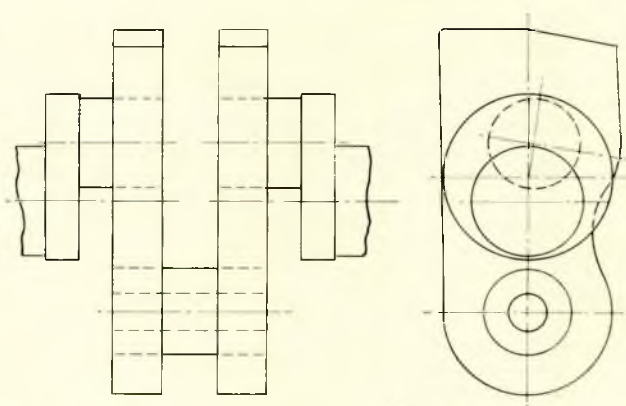
(a) Burmeister and Wain. Semi-built, cast throw type



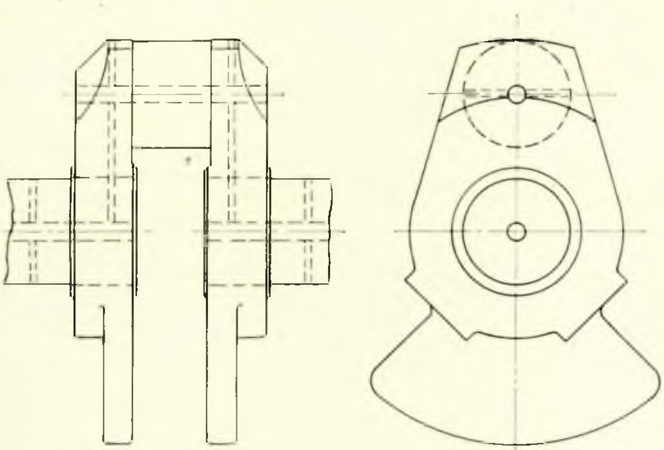
(b) Burmeister and Wain. Fully built, cast webs type



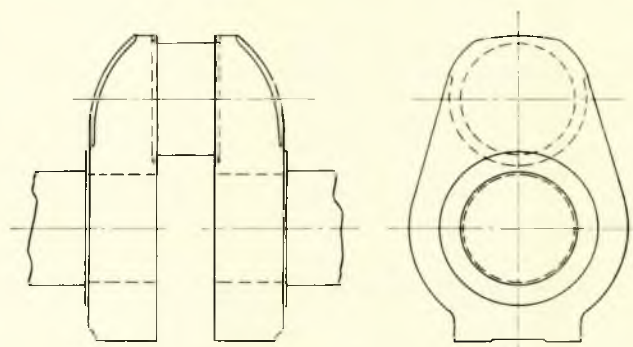
(c) Doxford. Combined fully built, solid forged type (67 LB)



(d) Doxford. Combined fully built, solid forged type (67 PT)

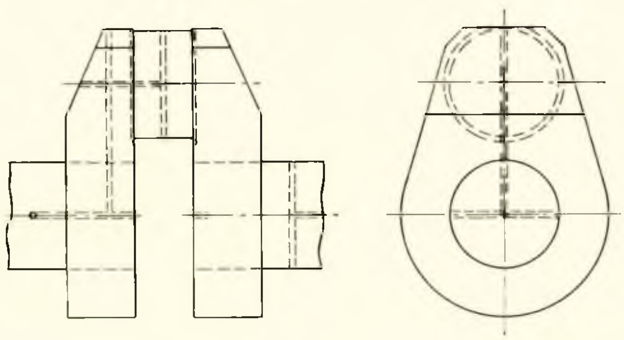


(e) Götaverken. Semi-built, cast throw type, with integral balance weights

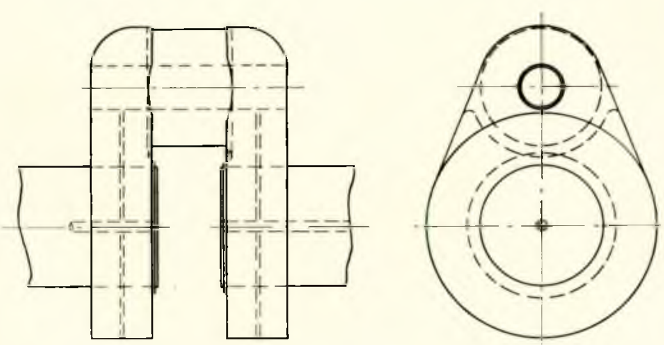


(f) Fiat. Semi-built, cast throw type

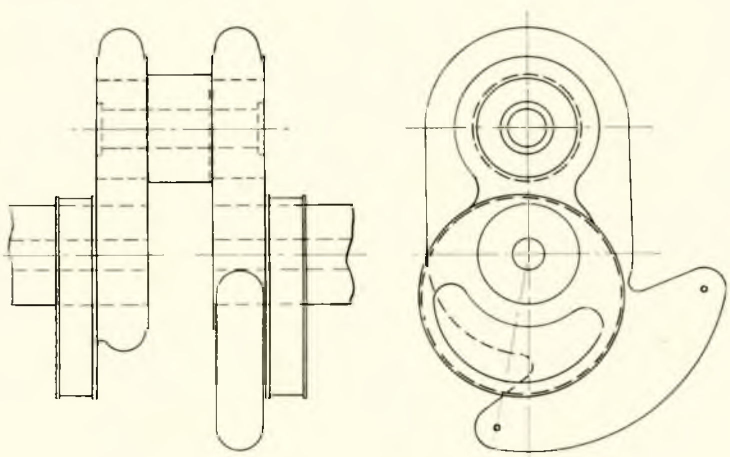
FIG. 4—*Some typical large modern crankshaft designs*



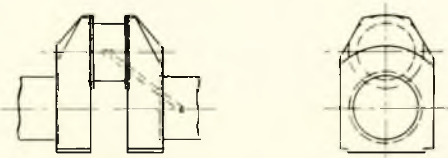
(g) Mitsubishi UEC. Semi-built, forged type



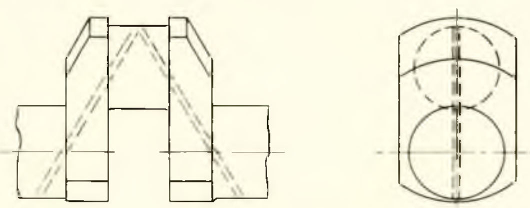
(h) M.A.N. Semi-built, cast throw type



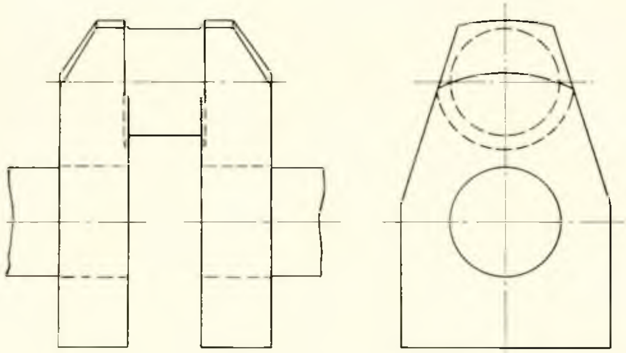
(j) B and W—Harland and Wolff. Fully built, cast webs type with eccentrics and integral balance weights



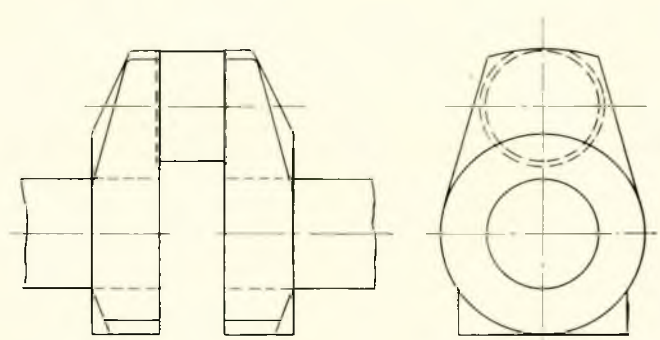
(k) Mirrlees-National. Solid forged type



(l) Nordberg. Solid forged type



(m) Stork. Semi-built forged type



(n) Sulzer. Semi-built forged type

FIG. 4—*Some typical large modern crankshaft designs*

TABLE III.—CRANKSHAFT DEFECTS INCIDENCE IN TWO-STROKE SINGLE-ACTING MAIN OIL ENGINES BUILT 1947 TO 1962 INCLUSIVE (16 YEARS) [(DEFECTS RECORDED 1953 TO 1962 INCLUSIVE (10 YEARS)].

	B.h.p. per shaft															Total 1,001–20,000		
	1,001–2,000			2,001–5,000			5,001–8,000			8,001–12,000			12,001–20,000					
Type of defect	No. of defects	Years of service	Incidence per 100 years service															
	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C
	Solid forged																	
Cracked	6	2,697	0.22	2	663	0·3	0	27	0	—	—	—	—	—	—	8	3,387	0·24
Broken	4	2,697	0·15	0	663	0	0	27	0	—	—	—	—	—	—	4	3,387	0·12
Total	10	2,697	0·37	2	663	0·3	0	27	0	—	—	—	—	—	—	12	3,387	0·36
	Semi-built forged																	
Cracked	1	457	0·22	2	1,828	0·11	2	1,641	0·12	1	402	0·25	0	22	0	6	4,350	0·14
Broken	0	457	0	0	1,828	0	1	1,641	0·06	0	402	0	0	22	0	1	4,350	0·02
Slipped shrinks	1	457	0·22	0	1,828	0	1	1,641	0·06	1	402	0·25	0	22	0	3	4,350	0·07
Total	2	457	0·44	2	1,828	0·11	4	1,641	0·24	2	402	0·5	0	22	0	10	4,350	0·23
	Fully-built forged (no defects)																	
	0	4	0	0	162	0	0	119	0	0	99	0	—	—	—	0	384	0
	Solid cast (S.G.I.) (no defects)																	
	0	191	0	0	24	0	—	—	—	—	—	—	—	—	—	0	215	0
	Semi-built cast throws																	
Cracked	0	285	0	1	999	0·1	2	2,279	0·09	6	592	1·01	0	42	0	9	4,197	0·21
Broken	1	285	0·35	2	999	0·2	2	2,279	0·09	0	592	0	0	42	0	5	4,197	0·12
Slipped shrinks	0	285	0	0	999	0	0	2,279	0	0	592	0	0	42	0	0	4,197	0
Total	1	285	0·35	3	999	0·3	4	2,279	0·18	6	592	1·01	0	42	0	14	4,197	0·33
	Fully built cast webs																	
Cracked	0	200	0	1	1,540	0·07	5	1,969	0·25	1	951	0·1	0	22	0	7	4,682	0·15
Broken	0	200	0	0	1,540	0	0	1,969	0	1	951	0·1	0	22	0	1	4,682	0·02
Slipped shrinks	0	200	0	3	1,540	0·19	2	1,969	0·10	0	951	0	0	22	0	5	4,682	0·11
Total	0	200	0	4	1,540	0·26	7	1,969	0·35	2	951	0·21	0	22	0	13	4,682	0·28
	Triple crank units (combination type)																	
Cracked	0	80	0	8	3,060	0·26	9	3,147	0·28	5	187	2·67	—	—	—	22	6,474	0·34
Broken	0	80	0	7	3,060	0·23	4	3,147	0·13	2	187	1·07	—	—	—	13	6,474	0·20
Slipped shrinks	0	80	0	2	3,060	0·07	10	3,147	0·32	1	187	0·53	—	—	—	13	6,474	0·20
Total	0	80	0	17	3,060	0·56	23	3,147	0·73	8	187	4·27	—	—	—	48	6,474	0·74
	Summary—all types																	
Cracked	7	3,914	0·18	14	8,276	0·17	18	9,182	0·19	13	2,231	0·58	0	86	0	52	23,689	0·22
Broken	5	3,914	0·13	9	8,276	0·11	7	9,182	0·08	3	2,231	0·14	0	86	0	24	23,689	0·10
Slipped shrinks	1	1,217	0·08	5	7,613	0·13	13	9,155	0·14	2	2,231	0·09	0	86	0	21	20,302	0·11
Total	13	3,914	0·41*	28	8,276	0·41*	38	9,182	0·41*	18	2,231	0·81	0	86	0	97	23,689	0·43*
B.h.p. per shaft	1,001–2,000			2,001–5,000			5,001–8,000			8 001–12,000			12,001–20,000			1,001–20,000		

*These incidence figures have been adjusted so as to exclude the years of service of solid forged crankshafts in relation to the slipped shrink defects.

TABLE IV—TRIPLE-CRANK UNITS
(excluding 750 × 2,500 mm. engine)

B.h.p. per shaft	1,001-2,000			2,001-5,000			5,001-8,000			8,001-12,000			12,001-20,000			Total 1,001-20,000		
Type of Defect	No. of defects A	Years of service B	Incidence per 100 yrs. C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C
Cracked	0	80	0	8	3,060	0·26	6	3,030	0·2	0	78	0	—	—	—	14	6,248	0·22
Broken	0	80	0	7	3,060	0·23	1	3,030	0·03	0	78	0	—	—	—	8	6,248	0·13
Slipped shrinks	0	80	0	2	3,060	0·07	9	3,030	0·3	0	78	0	—	—	—	11	6,248	0·18
Total	0	80	0	17	3,060	0·56	16	3,030	0·53	0	78	0	—	—	—	33	6,248	0·53

TABLE V.—CRANKSHAFT DEFECTS IN FOUR-STROKE SINGLE ACTING MAIN ENGINES BUILT 1947 TO 1962 INCLUSIVE (16 YEARS) [DEFECTS RECORDED 1953 TO 1962 INCLUSIVE (10 YEARS)]

B.h.p. per shaft	1,001–2,000			2,001–5,000			5,001–8,000			8,001–12,000			12,001–20,000			Total 1,001–20,000		
Type of defect	No. of defects	Years of service	Incidence per 100 years service															
	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C
	Solid forged																	
Cracked	1	2,007	0·05	2	261	0·77	—	—	—	—	—	—	—	—	—	3	2,268	0·13
Broken	2	2,007	0·10	2	261	0·77	—	—	—	—	—	—	—	—	—	4	2,268	0·18
Total	3	2,007	0·15	4	261	1·54	—	—	—	—	—	—	—	—	—	7	2,268	0·31
	Semi-built forged (no defects)																	
	0	90	0	0	4	0	—	—	—	—	—	—	—	—	—	0	94	0
	Fully built forged																	
Cracked	0	70	0	1	420	0·24	—	—	—	—	—	—	—	—	—	1	490	0·2
	Semi-built cast throws (no defects)																	
	—	—	—	0	8	0	0	4	0	—	—	—	—	—	—	0	12	0
	Fully built cast webs																	
Slipped shrinks	0	75	0	1	440	0·23	—	—	—	—	—	—	—	—	—	1	515	0·19
	Summary—all types																	
Cracked	1	2,242	0·04	3	1,133	0·26	0	4	0	—	—	—	—	—	—	4	3,379	0·12
Broken	2	2,242	0·09	2	1,133	0·18	0	4	0	—	—	—	—	—	—	4	3,379	0·12
Slipped shrinks	0	235	0	1	872	0·11	0	4	0	—	—	—	—	—	—	1	1,111	0·09
Total	3	2,242	0·13*	6	1,133	0·55*	0	4	0	—	—	—	—	—	—	9	3,379	0·33*
B.h.p. per shaft	1,001–2,000			2,001–5,000			5,001–8,000			8,001–12,000			12,001–20,000			1,001–20,000		

*These incidence figures have been adjusted so as to exclude the years of service of solid forged crankshafts in relation to the slipped shrink defects.

Table V gives similar information to Table III, but for 4-S.C. engine shafts. In general, their performance compares favourably with that of 2-S.C. engine shafts. It will be seen that two-thirds of the casualties occurred in the 2,001-5,000 b.h.p. range and that the smaller solid forged shafts have given a good account of themselves.

Table VI shows a summarized addition of Tables III and V which requires no further comment.

In Table VII the defects have been summed for all horsepower ranges and cracked or broken shafts have been lumped together. Differentiation is between the various types of construction, without regard to whether forged or cast. It will be noted that the overall liability to failure from slipped shrinks is only about 32 per cent of that from cracking or breaking.

Tables VIII and IX, in conjunction with Figs. 5 and 5(a), are of particular interest in that the location of the failures,

TABLE VI.—SUMMARY OF DEFECTS IN TWO- AND FOUR-STROKE SINGLE-ACTING ENGINES, DIFFERENTIATING THE VARIOUS TYPES OF CRANKSHAFT CONSTRUCTION.

Crankshaft types		Number at risk	Years of service	Number of defects	Incidence per 100 years' service
Solid	Forged	1,133	5,655	19	0·34
	Cast	60	215	0	0
	Total	1,193	5,870	19	0·32
Semi-built	Forged	754	4,444	10	0·22
	Cast throws	742	4,209	14	0·33
	Total	1,496	8,653	24	0·28
Fully built	Forged	128	874	1	0·11
	Cast webs	781	5,197	14	0·27
	Total	909	6,071	15	0·25
Triple crank units (combination type)		866	6,474	48	0·74
Totals (all types)		4,464	27,068	106	0·41*

* These incidence figures have been adjusted so as to exclude the years of service of solid forged crankshafts in relation to the slipped shrink defects.

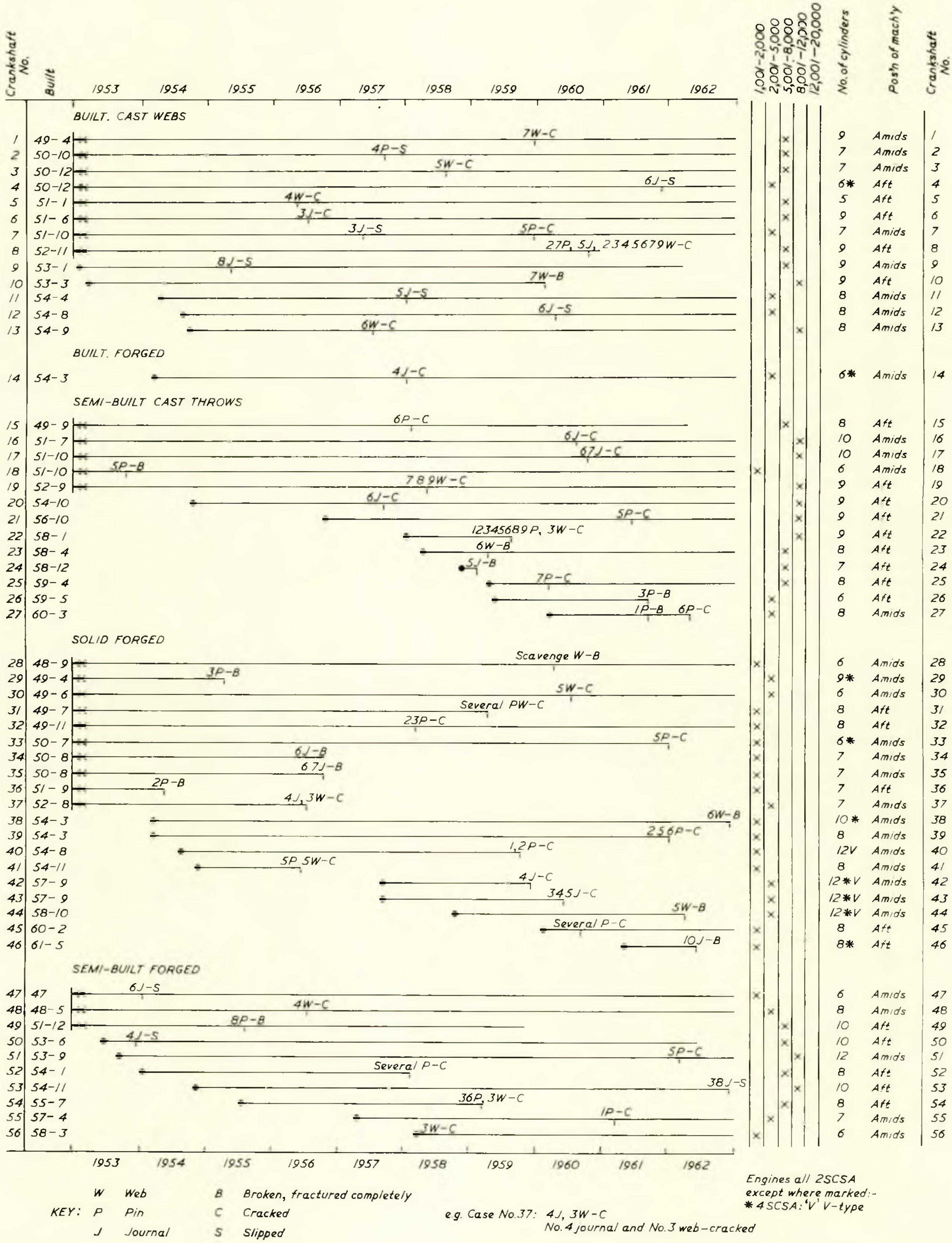


FIG. 5—*Crankshaft failures*

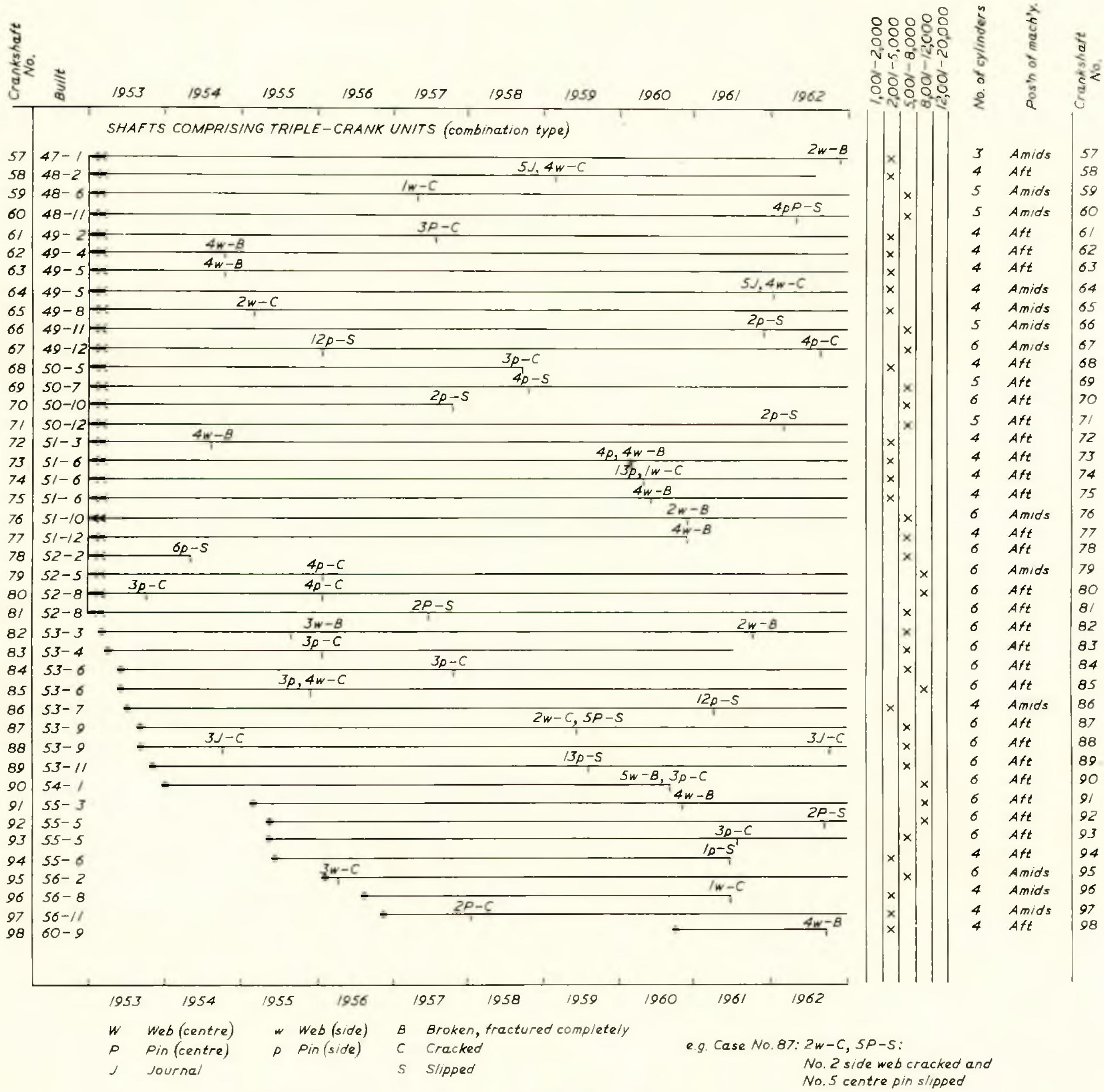


FIG. 5a—*Crankshaft failures*

TABLE VII.—SUMMARY OF TYPES OF DEFECTS IN CRANKSHAFTS IN TWO-STROKE AND FOUR-STROKE SINGLE-ACTING MAIN ENGINES.

	Solid forged		Semi-built		Fully built		Triple crank (combination type)		All types	
Type of defect	No. of defects	Incidence per 100 yrs. service	No.	Incidence	No.	Incidence	No.	Incidence	No.	Incidence
Cracked or broken	19	0·34	21	0·24	9	0·15	35	0·54	84	0·31
Slipped shrinks	0	0	3	0·04	6	0·10	13	0·2	22	0·10
Total	19	0·34	24	0·28	15	0·25	48	0·74	106	0·41 *
No. of shafts at risk	1,193		1,496		909		866		4,464	
Total years of service	5,870		8,653		6,071		6,474		27,068	
Average years of service	4·92		5·78		6·68		7·48		6·08	

*These incidence figures have been adjusted to exclude the years of service of solid forged crankshafts in relation to slipped shrink defects.

also that of the machinery (amidships or aft end), together with other information such as number of cylinders in the engines affected, are presented. Examination of Fig. 5 shows that over 85 per cent of the defects occurred within the middle two-thirds of the crankshaft length, which is not unexpected having regard to torque variations, torsional vibration, extra bending stresses due to differential wear-down of main bearings, flexing of bed plates, etc.

TABLE VIII.—LOCATION OF CRACKS AND BREAKS IN CRANKSHAFTS.

Type of construction		Location of cracks and breaks			
		Pins	Webs	Journals	Totals
Forged	Solid	9	7	6	22
	Semi-built	5	3	0	8
	Fully built	0	0	1	1
	Total	14	10	7	31
Cast	Semi-built	8	3	4	15
	Fully built	2	6	2	10
	Total	10	9	6	25
Triple crank units (combination type)		14	22	4	40
Totals (all types)		38	41	17	96

TABLE IX.—LOCATION OF CRACKS AND BREAKS IN CRANKSHAFTS.

Type of construction		Location of cracks and breaks			
		Pins	Webs	Journals	Totals
Solid	Forged	9	7	6	22
	Cast	0	0	0	0
	Total	9	7	6	22
Semi-built	Forged	5	3	0	8
	Cast	8	3	4	15
	Total	13	6	4	23
Fully built	Forged	0	0	1	1
	Cast	2	6	2	10
	Total	2	6	3	11
Triple crank units (combination type)		14	22	4	40
Totals (all types)		38	41	17	96

It will be noticed that the totals in Tables VIII and IX do not agree with those in Tables III, V and VII because in the latter tables some incidents affect adjoining elements.

It will be seen that 41 per cent of failures occurred in aft end and 59 per cent in amidships installations.

Fig. 5(a), covering crankshafts of the triple-crank, combination type of construction, gives the corresponding picture to Fig. 5, except that 71 per cent of the shafts affected were in aft end and 29 per cent in amidships installations, this being largely influenced by the 750 mm. bore engines most of which were fitted aft.

The analyses of the location of cracks and breaks given in Tables VIII and IX reveal some interesting trends. Failures in solid forged shafts tend to be rather evenly divided among pins, webs and journals with, however, somewhat greater incidence in crankpins. With semi-built shafts there is a pronounced increase in the proportion of shafts with failures in the crankpins. The fully built shafts show a quite different pattern in that 90 per cent of all failures in these shafts occur in cast web constructions, the failures predominating in the crankwebs, a probable reflection of casting difficulties.

EXAMPLES OF SERVICE FAILURES

In general, marine service experience, supported by "post mortem" examination, suggests that crankshaft failures can rarely be attributed solely to specific metallurgical defects associated with normal manufacturing processes. The quality of the material, however, particularly in relation to its fatigue resistance, can and does influence the margin of safety available to withstand those additional and often unpredictable stresses which are sometimes encountered under the arduous conditions of sea service.

In the final section of this paper an attempt is made to assess the order of magnitude of this safety margin for a six-throw crankshaft of Rule size for a large modern 2-S.C. engine.

Some examples of typical oil engine crankshaft failures will now be described and illustrated with suggestions as to their probable cause (or causes).

Case I

This twin-screw ship was electrically propelled, the a.c. power being supplied by four nine-cylinder, 4-S.C.S.A. Diesel engines developing a total of 14,400 b.h.p. at 240 r.p.m. and each fitted with an hydraulic damper at the free end of the crankshaft. On account of early difficulties with the crankshaft main bearings, the engine builders decided to fit balance weights to the crankshaft and also to replace the cast iron pistons with light alloy ones in an attempt to reduce the main bearing loading. The net effect on the torsional vibration characteristics was not great, the position of the 1-node $4\frac{1}{2}$ order critical being only raised about 5 r.p.m., the calculated vibration stress remaining substantially unaltered.

It was later measured that with the torsional vibration damper working efficiently, the critical speed is very close to the normal service speed of 240 r.p.m. with a torsional vibration stress of about ± 3,300 lb./sq. in. in No. 8 crankpin. However, owing to persistent failures of the mechanical parts of the damper, the crankshaft had been subjected to considerably higher vibration stresses for prolonged periods, and failures commenced as follows:

No. 2 Generator (November, 1952)
Crack in fillet of No. 8 crankpin. Shaft replaced by previously used spare.

No. 4 Generator (February, 1953)
Crack in fillet of No. 8 crankpin. This section of shaft renewed.

No. 3 *Generator (October,* 1954)
Crack in fillet of No. 7 crankpin. This section of the shaft was renewed and at the same time the hydraulic dampers on all engines were replaced by others of the silicone viscous fluid type having considerably greater effective moment of inertia, whereby the major critical was lowered to about 195 r.p.m. Unfortunately, it would appear that the damage had already been done, as further crankshaft failures occurred.

No. 2 *Generator (March,* 1955)
Crack in fillet of No. 7 crankpin extending into web. New shaft fitted December, 1956, after temporary repair.

No. 4 *Generator (March,* 1956)
Cracked at after web of No. 5 crank (vibration stress about 90 per cent of that in No. 8) through the coupling bolt holes connecting the two half-shafts. Fractured shaft replaced with a spare used half-shaft already containing two small cracks on the coupling face. These latter drilled at ends.

Fig. 6 shows a typical failure of torsional fatigue type.

FIG. 6—*Torsional fatigue fracture in No. 7 crankpin* (Case I)

February-September, 1957
Machinery converted to single screw driven by a direct-coupled, nine-cylinder, 2-S.C.S.A. Diesel engine developing 8,100 b.h.p. at 115 r.p.m., without further crankshaft incident.

These failures would appear to have been mainly influenced by torsional vibration stresses consequent upon malfunctioning of the hydraulic dampers, although from the location and direction of some of the fractures additional bending stresses cannot be ruled out. These could have arisen from the still persistent main bearing troubles referred to and/or cracking could have been initiated by the frequent piston seizures which resulted from the fitting of the light alloy pistons in June, 1951. It may be significant, however, that No. 1 unit, which sustained the greatest number of piston seizures, suffered no damage to the crankshaft, which consequently turned the greatest number of revolutions.

This example well illustrates the dangers which may attend the use of vibration dampers for continuous control of a major critical speed, particularly where, as in the case described, the damper incorporates mechanical parts subject to wear and tear and fatigue.

It is to guard against such troubles that the Society's *Guidance Notes on Torsional Vibration* recommend that dampers should preferably not be used for this purpose.(6)

Case II *

This failure relates to a twin-screw geared installation in which one of four engines suffered a fatigue fracture through No. 1 forward and No. 2 forward crank webs of a six-throw crankshaft some 900 running hours after entering service. Fig. 7 shows the appearances of the fracture surface in No. 2 forward web, from which it is clear that failure commenced in the fillet radius between pin and web. It is probable that the break in No. 1 forward web was secondary to that in No. 2 forward, i.e. consequential.

The shaft material was 3 per cent CrMo to B.S. En40B(U) nitrided all over to a surface hardness of approximately 900 V.D.H. (10 kg.), the depth of case being specified at 0·012-0·016 in.

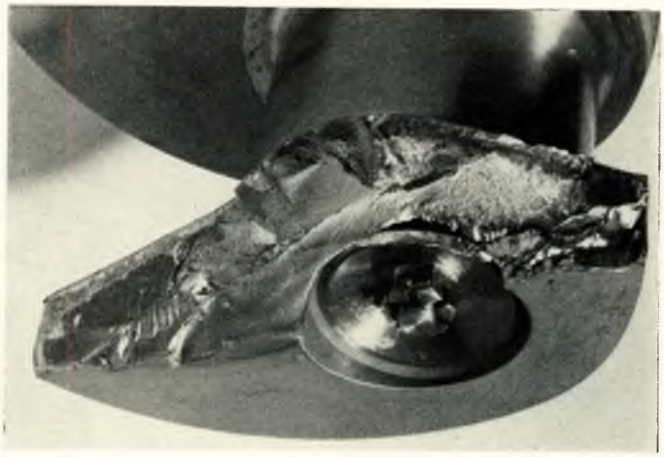


FIG. 7—*Primary fatigue fracture at junction of No. 2 crankpin and forward web* (Case II)

It appears that in consequence of abrasive material in the lubricating oil, all four engines had to have their crankshafts re-ground after only 800 hours in service, which was an unprecedented occurrence. Unfortunately this grinding operation, in which 0·005 in. was specified to be removed from pins and journals, just broke through the case thickness in the crankpin fillet radius (as shown in Fig. 8) to constitute a serious weakening of the shaft. The examination also showed up the presence of micro-grinding cracks (subsequently detected only with the greatest difficulty) on the crankpin thrust faces where the surface had been "kissed" by the grinding wheel (see Fig. 9). In this particular case it was reported that fluorescent dye-penetrant methods proved more sensitive than magnaflux in showing up these extremely fine cracks. It is sufficient to state that this particular crankshaft failed some 80 hours only after re-grinding and that against a background of some 370 engines of the same type with a then total running time of well over 150,000 (now over 500,000) hours without crankshaft failures or need to re-grind.

There was no evidence of bearing misalignment or excessive wear-down to account for this breakdown which must therefore be ascribed entirely to the re-grinding process which

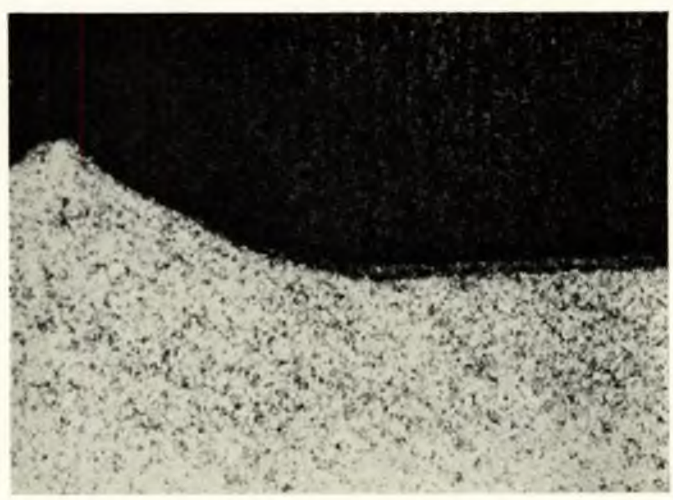


FIG. 8—*Showing run-out of nitrided case in vicinity of radius* (Case II)

* This vessel not classed with Lloyd's Register.

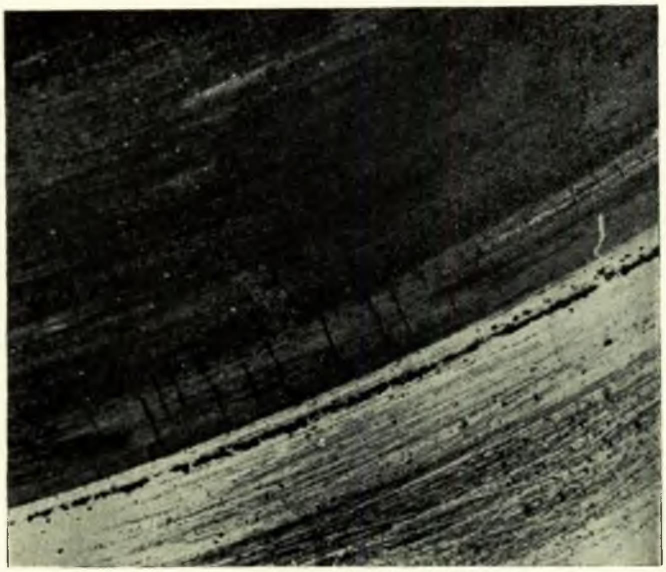


FIG. 9—*Radial cracks on thrust face of No. 4 crankpin after crack detection* (Case II)

was caused solely by running on contaminated oil. The incident highlights the dangers of such re-grinding operations on the relatively thin case material typical of the nitriding process.

Case III

This case brings out the potential dangers of pushing design to the limit in order to economize in axial length of crankshaft. The shaft concerned was of the solid forged type fitted in an eight-cylinder, 2-S.C.S.A. engine and failed after some three years' service. The design was such as to embody a recess in the oversize after web of No. 4 crank to take a bolted flange coupling forward of No. 5 journal, thus securing the advantage of separate half-shafts without the penalty of an additional centre bearing.

The first intimation of trouble was when a number of studs belonging to the caps of the main bearings on either side of No. 4 crank, as well as studs from the aforesaid centre coupling, were found fractured at sea. After temporary repairs to the main bearings and two hours' emergency running (owing to the ship being in dangerous waters) re-examination revealed a fracture through No. 4 after crankweb below the crankpin and originating at the sharp edges of the three holes for the coupling studs (see Figs. 10 and 11). The fracture surface is clearly typical of a fatigue failure, as were those of the broken bearing and coupling studs. There was also plain evidence of fretting on the surface of the recess in the crankweb around the stud holes.

Metallurgical examination at the Society's Research Laboratory showed no deficiency or inherent metallurgical faults in the steels of the parts which failed and their finish also appeared to be good, apart from the lips of the stud holes, aforesaid, which were sharp. Since the remaining studs and bolts in the centre coupling were found to be tight and a good fit and there was no apparent sign of gross misalignment of the shaft, it would seem likely that the breakdown was a result of the normal engine operating forces in conjunction with inadequate design, further undermined by the stress concentration associated with the sharp edges of the stud holes in the region of the crankweb/pin fillet. There is also little doubt that the shaft was in fact already fractured before the two-hour emergency operation mentioned and that this therefore probably preceded the failure of the bearing and coupling studs.

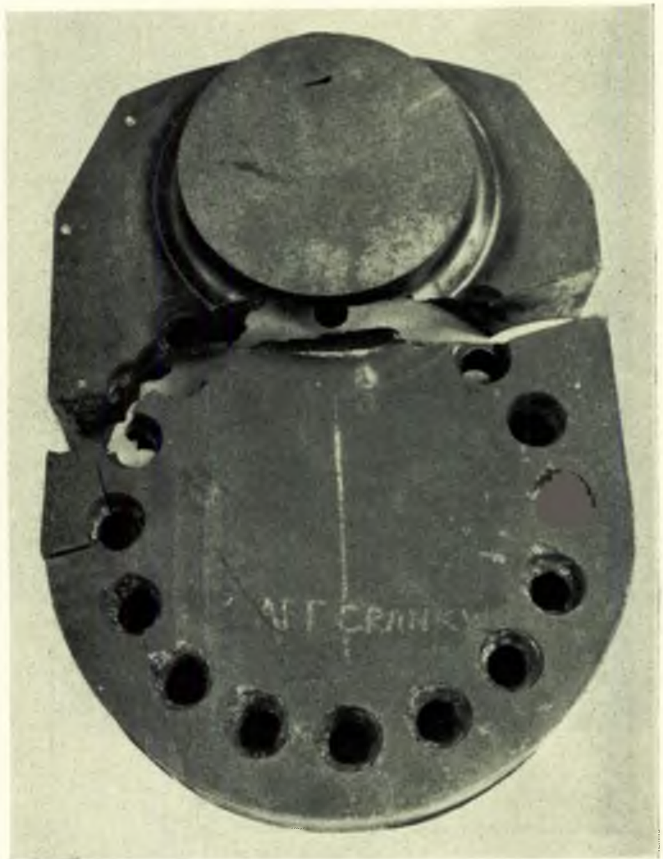


FIG. 10—*Fracture through No. 4 aft centre coupling/web* (Case III)

FIG. 11—*Fracture section through No. 4 aft centre coupling/web* (Case III)

Case IV *

The main machinery of this ship consists of an eight-cylinder, 4-S.C.S.A. direct-coupled engine and after some 13 years' service the first fracture to be brought to the notice of the Society occurred in the No. 5 crankweb assembly, which, as in Case III, embodied a recess for a bolted flange. After repairs involving the renewal of No. 5 crank, the vessel re-entered service, only to sustain a similar fracture in No. 5 crank, no more than four months later.

As the vessel was unclassed, the early history of this ship is unknown, but following the failure of the renewed assembly, a different type of main engine was fitted.

As will be seen from Fig. 12, the fracture was located in the crankpin fillet and, as shown in Fig. 13, appears to be of bending fatigue type and had propagated almost completely across the section. It originated at a sharp fillet of a recess

* This vessel not classed with Lloyd's Register.

FIG. 12—*Crack in No. 5 crankpin/forward web in way centre coupling* (Case IV)

which had been machined in the fillet radius between pin and web to provide clearance for the coupling bolts.

In this case a contributory cause of failure was undoubtedly metallurgical in that the steel, when examined at the Society's Research Laboratory, was dirty and contained numerous bands of sulphide segregation. The origin of the fracture was located in one of the most pronounced of these bands (Fig. 14).

It should be noted that in the Society's experience, the practice (fortunately less common today) of recessing crankwebs to form a bolted joint between half-shafts has been responsible for many failures in the past and, in the author's opinion, is most undesirable.

Case V

This ship was not built under the Society's survey but was accepted for class as a trawler three years later. The

FIG. 13—*Fracture section through No. 5 crankpin* (Case IV)

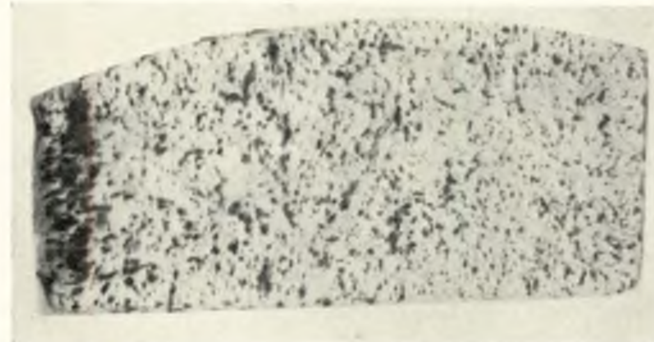


FIG. 14—*Sulphur print from origin of fracture* (Case IV)

main machinery consisted of a three-cylinder, 4-S.C.S.A. engine geared to a single screw.

The crankshaft was of unusual design, being a fully built forged type, but having the pins and journals of appreciably larger diameter than the shrinkage bores in the crankwebs (see Figs. 15 and 16), the transition being by means of recessed fillets. The strength of pins and journals in the shrinkage fit was about 16 per cent below normal Rule requirements for a ship built to class.

After four years' service a fracture occurred in No. 3 crankpin at the forward end extending right through the reduced portion of the pin. As shown in Fig. 17, the failure appears to be of the bending fatigue type.

The damage was stated to have been due to the propeller having been fouled by the net and fishing gear. However, from the progressive fatigue appearance of the fracture surface and the fact that the flywheel and thrust ball bearing races had to be renewed, it would appear possible that extra bending stresses associated with misalignment at the aft end of the crankshaft may have played a part.

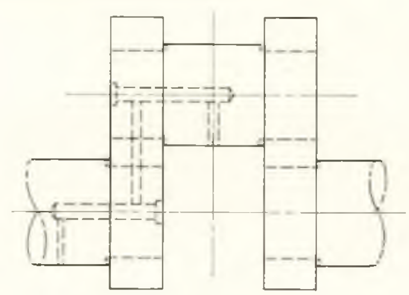


FIG. 15—*Fully built crankshaft design* (Case V)

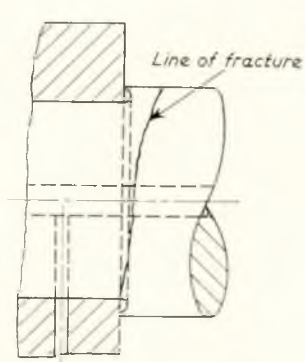


FIG. 16—*Detail of No. 3 crankpin/forward web shrunk connexion showing course of fatigue fracture* (Case V)

FIG. 17—*Fatigue fracture section through No. 3 crankpin (forward)* (Case V)

The case is a good example of poor design in the concealment of a weak point of stress concentration, in such a way that the onset of failure could not possibly have been detected by normal means. In the author's opinion, such a design is unacceptable for marine service.

Case VI

Some four years after entering service, whilst undergoing special survey, the solid forged crankshaft of the seven-cylinder 2-S.C.S.A. main engine was found fractured in way of No. 4 journal and on investigation the crack was seen to extend right through the after web of No. 3 throw.

Fig. 18 shows the course of the crack in No. 4 journal from which it would seem to be typical of combined torsion and bending.

The torsional vibration characteristics were, however, satisfactory and there was no suggestion of misalignment.

The cause of failure was therefore mysterious until the journal surface was etched with Adler's reagent, when just at the transition from journal to fillet radius an elliptical-shaped zone about 30 mm. long in the axial direction was revealed. This is typical of a welded repair and would be practically impossible to detect by means other than chemical etching. It would, therefore, be reasonable to conclude that the defect occurred in the original forging and was surreptitiously repaired without the surveyor's knowledge or approval.

The range of torque variation in a seven-cylinder 2-S.C. engine is a maximum near the centre of the crankshaft which would account for the largely torsional appearance of the break.

This case is an excellent illustration of the dangers of indescriminate, clandestine welding-up of surface defects, equally reprehensible in forgings as in castings, but more particularly so when the weld area is close to a point of high stress concentration as in this example.

Welded repairs of shafting are not generally acceptable to Lloyd's Register, although, with the owner's consent, and where it is practicable to effect full pre- and post-weld heat treatment, welding of minor blemishes well removed from discontinuities and points of high stress such as fillets and oil holes, may be specially submitted for the surveyor's approval. In the author's opinion, however, such surface flaws should rather be smoothed off by hand-dressing, unless they are too gross, when in all probability it would be better to scrap the part concerned in any case.

The Society's records contain other examples of failures in important forgings and castings directly attributable to this undesirable practice, and where found to have been done without the surveyor's knowledge, a particularly serious view is taken, since such incidents inevitably disturb the mutual confidence which should exist between surveyor and builder or manufacturer.

Case VII

A number of cases of cracking of cast steel webs in the fully built type of crankshaft has been reported and almost invariably these have been traced either to indescriminate and inefficient welded repairs located in regions subjected to high stress variation or to the presence in such regions of hot tears, shrinkage cavities, or damaged surface material resulting from

FIG. 18—*Combined torsion/bending fatigue crack in No. 4 journal/forward web originating from welded repair* (Case VI)

FIG. 19—*Fracture section through cast steel fully built crankweb* (Case VII)

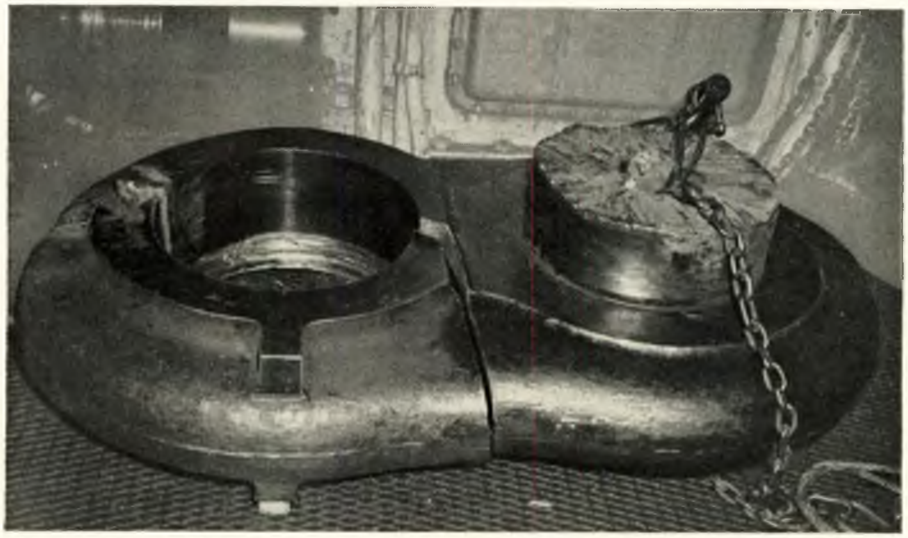


FIG. 20—*Fractured cast steel fully built crankweb* (Case VII)

in-gates or cast-on test coupons. In some cases, the riser or head has been located immediately above the spectacle piece at mid-throw, thus giving possibilities for unsoundness etc. at this critical position. Taken all round, it would appear preferable to cast the web with its plane almost horizontal and with the riser head above the bore for the journal piece which, when machined out, should effectually remove any shrinkage cavities, segregation etc. A well known manufacturer and licensor claims that using such casting methods, they have experienced no grave casting defects or fractures in crankwebs over a period of more than 30 years.

Fig. 19 shows a typical example* of such a fractured web, after 4½ years' service, the crack emanating from a near-surface flaw about mid-throw. It is clearly of a fatigue character and propagated gradually from the surface inwards to the highly hoop-stressed region of the journal bore.

Fig. 20 shows a similar web from another engine which failed from a hot tear after nearly ten years' service. The castings were made by the same method as that shown in Fig. 19 and metallurgical examination at the Society's Research Laboratory revealed that the surface material around the origin of the fracture was heavily de-carburized.

It is understood the casting method adopted in these cases was with the web in the horizontal flat position with a number of small risers arranged around the outer edges of the web. This method of casting was later changed to the one described above with the same good result.

It was largely to avoid the possibility of such defective webs entering service that the Society recently introduced requirements for the magnetic crack detection of cast steel crankwebs. (Chap. P. 515)

The following three examples of crankshaft failures in auxiliary engines are considered to be of sufficient interest to warrant inclusion: —

Case VIII

This was a six-cylinder, 4-S.C.S.A. generator engine developing 250 b.h.p. at 500 r.p.m., which sustained a crankshaft failure some 20 months after entering service. The shaft fractured through the free-end web of No. 4 crank and from the appearance of the break (Fig. 21) it is clearly a bending fatigue failure, the origin being located in the recessed fillet between No. 4 journal and the adjacent web.

The engine was returned to the engine builders for a complete investigation and some of the findings illustrate the dangers of neglect and poor maintenance.

The material of the shaft was specified to conform to B.S.S. En 8, with a tensile strength of about 40 tons/sq. in. and subsequent check tests confirmed this and showed the steel to be in the normalized condition, clean and of good quality.

Careful alignment checks were made and Fig. 22 shows the results, graph a being the line of main bearing housing bores and graphs b and c the bed face alignments, front and back of engine, respectively. It will be seen that the bore alignment is generally similar to that of the front bed face which deviates some 0·007in. from a true plane in way of the fractured No. 4 crank. This is clearly consequential distortion since the back bed face shows no such local deviation. It may here be stated that main bearing caps Nos. 2, 3, 4, 5 and 6 were all found fractured but not their studs, presumably consequent upon the engine not having been shut down immediately. It might therefore reasonably be inferred that, prior to the accident, the bore alignments were within 0·002-0·003in.

The main bearing shells were of the modern thin steel-backed variety manufactured to an accuracy of 0·0003in. Examination revealed that the shells of Nos. 3, 6 and 7 housings

FIG. 21—*Bending fatigue in auxiliary engine crankweb* (Case VIII)

* This vessel not classed with Lloyd's Register.

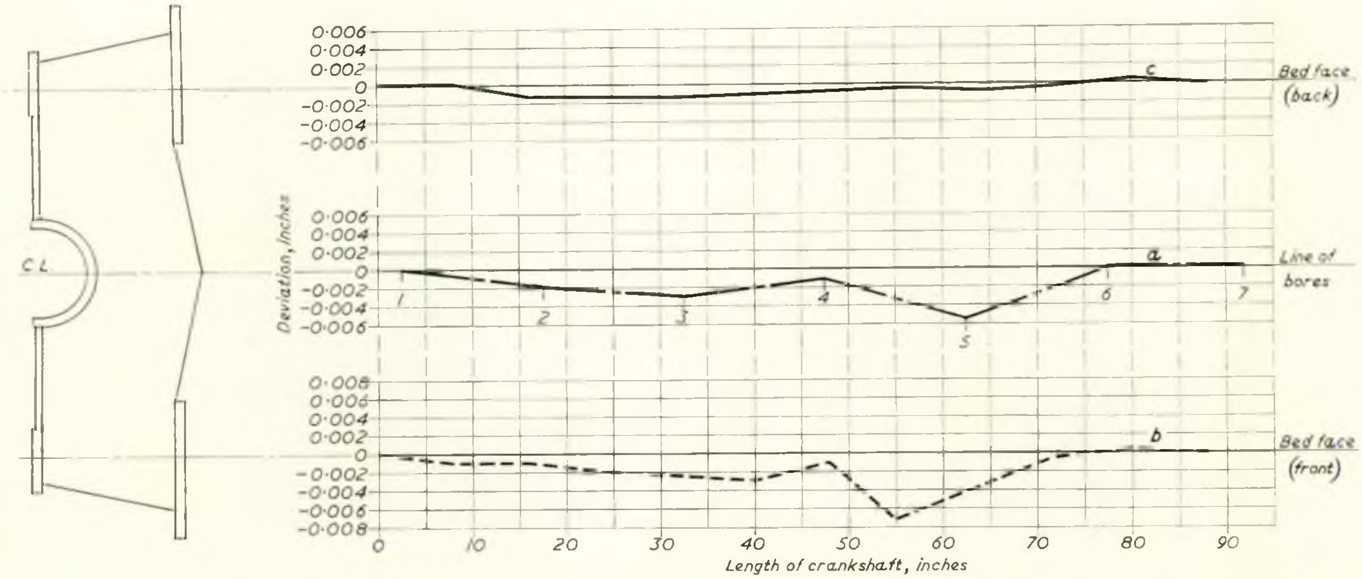


FIG. 22—*Alignment checks of main bearing bores and bed faces subsequent to crankshaft failure* (Case VIII)

were not those originally fitted by the engine builders because: —

i) The shells were not numbered.

ii) The backs of the shells had been heavily filed (see Fig. 23).

iii) Attempts had been made to stretch the shells by hammering (see Fig. 23).

FIG. 23—*Back of No. 7 bottom main bearing shell showing filing and hammering marks* (Case VIII)

These three shells had all been loose as evidenced by the fretting of the mating surfaces, whereas all the remaining shells were an excellent fit in their bores.

The condition of the running surfaces of the bearing shells Nos. 1, 3, 6 and 7 was reasonably good with little reduction of thickness, whereas the white metal in Nos. 2, 4 and 5 was extensively damaged, hammered and/or wiped and generally had lost thickness.

The inference is that bearings Nos. 3 and 6 were not taking their share of the load and thus caused overloading of bearings Nos. 2, 4 and 5. Nos. 4 and 5, being the centre bearings of a six-cylinder, 4-S.C.S.A. engine crankshaft, would of course in any case tend to be rather more heavily loaded than the remaining bearings due to the internal couple in the engine set up by the inertia forces. (Dorey and Forsyth, I.N.A. 1942).[7]

Probably, during operation the bearings had "run", filling the central oil grooves with white metal, thus further aggravating the condition, causing a collapse of the white metal with resultant high bending moments on the crankwebs leading to fatigue failure.

Unfortunately, no measurements of crankweb deflexion were available, either immediately before or after the replacement of bearing shells Nos. 3, 6 and 7, but the probability is that such deflexions were not maintained within the limit of 0·00008 to 0·00013in. per inch of crank throw recommended by the engine builders.

Excessive clearances measured in the big end bearings may also have contributed to failure by hammering action on the crankpins.

The condition of the lubricating oil from tests on samples drawn from crankcase and sump indicated between 4 and 7 per cent solids, mostly carbon and iron oxides, and in the sump various "foreign bodies" were found, such as set screws, washers, brass shims, wire, wood etc.

The general picture is therefore one of neglect and inadequate and unskilled maintenance, which it is hoped will help to avoid similar malpractice elsewhere.

Case IX

Two ships sustained auxiliary engine crankshaft failures as follows: —

Ship A

Three sets of five-cylinder, 4-S.C.S.A. generator engines each developing 195 b.h.p. at 500 r.p.m. were carried, two of which also drove main air compressors.

After a running life of about 25,000 hours in each case, first one crankshaft driving a main compressor broke and three days later one of the remaining two crankshafts fractured in an almost identical manner, i.e. in the journal between No. 5 crank and the flywheel, the only difference being that in one set the crack, which was of the 45 deg. torsional type passed through the journal oil hole and in the other case did not, being displaced some 60 deg. from it (see Fig. 24). The ship proceeded on the remaining intact set, which fortunately also drove the remaining main compressor.

Ship B

One of three sets similar to those in Ship A broke at an identical position after some 2½ years' running life. In this

FIG. 24—*Torsional fatigue crack clear of oil hole in No. 6 journal of five-throw auxiliary crankshaft* (Case IX)

case the fracture passed through the journal oil hole at 45 deg., but in addition cracks were visible on the shaft surface extending about 1¼in. from the oil hole in a plane at 90 deg. from the main fracture (see Fig. 25) and, further, branched 45 deg. cracks were seen at mid-length of journal surface but displaced 135 deg. from the oil hole.

The evidence of these fractures pointed strongly to torsional vibration and accordingly torsiograph records were taken from one of the sets on Ship B. These confirmed the builders'

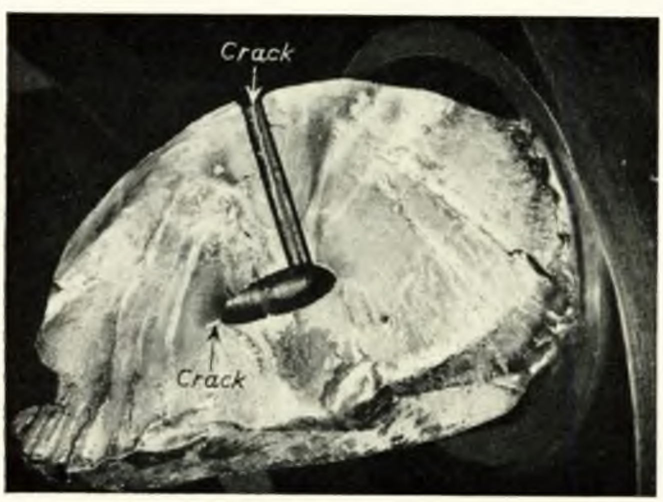


FIG. 25—*Torsional fatigue fracture through oil hole in No. 6 journal of five-throw auxiliary crankshaft* (Case IX)

original calculated 1-node natural frequency at about 3,100 v.p.m. giving the peak of the major 1-node 5th order critical at 620 r.p.m. and of the minor 5½th order at about 560 r.p.m. Although the combined additional flank stresses due to these two criticals at the normal service speed of 500 r.p.m. only amounted to about ± 2,500lb./sq. in. in No. 6 journal, any increase of speed above the 500 r.p.m. normal setting would considerably increase this stress condition due to dynamic magnification, especially up to 560 r.p.m., the resonant speed for the 5½th order critical, at which the combined stress reached a value of about ± 4,600lb./sq. in. It should be noted that no vibration damper was fitted to the crankshafts.

It was therefore concluded that the generator sets had been operating at speeds appreciably above 500 r.p.m., either inadvertently owing to tachometer error, or deliberately in order to carry a higher load. In any case, some errors in tachometer readings were found and corrected.

The dilemma, in such cases as this, lies in the fact that the fitting of a heavy free-end damper, to deal with the flank effects of the major critical, is likely to defeat its own purpose in that the additional inertia will lower the critical speed still closer to the service speed, with resulting greater energy input to the vibration and, in any case, continuous damper control of critical speeds is undesirable practice.

An interesting deduction from these failures is that the torsional fatigue strength reduction factors for the journal oil hole and the journal/crankweb fillet were clearly about equal. The shafts were of solid forged construction and the proportional dimensions of oil hole diameter and fillet radius were 0·06 and 0·08 respectively, in terms of crank journal diameter. The lip radius of the oil hole was 60 per cent of the oil hole bore.

Case X *

Fig. 26 shows an exceptionally good example of torsional fatigue failure in a crankpin of the semi-built crankshaft of a large generator engine. The point of initiation of the frac-

FIG. 26—*Torsional vibration fatigue failure in crankpin of semi-built generator crankshaft* (Case X)

ture is clearly in the fillet radius almost on the centre line of the web and this is of course a region subject also to high tensile hoop stress from the shrinkage of the journal. From the angle of about 90 deg. between the two intersecting planes of the fatigue fracture surfaces near the origin, it is reasonable to conclude that the failure was caused by a torsional critical speed giving a relatively high torsional vibration stress in comparison with the mean transmission stress.

It is quite remarkable how small a residual area of cross-section was finally left to carry the torsional load.

* This vessel not classed with Lloyd's Register.

Unfortunately, no other details of the case are available to the author.

SOME SPECIAL FACTORS AFFECTING CRANKSHAFT LIFE

It would be impossible within the confines of a single paper to go fully into all aspects of crankshaft performance as affected by the many design, manufacturing and metallurgical variations which arise in practice.

It is thought well, however, to mention a few points, additional to those arising from the examples of service failures cited, and which the author believes are of special importance.

Materials

Shafts of Forged or Rolled Steel

If webs are flame-cut, this should preferably be done hot, or if cold, a more generous machining allowance provided, say ½in. to 1in., depending upon shaft size. In any case crack detection before and after shrinkage would seem well worthwhile. Edges of crankwebs should be well rounded off. Rolled or forged slabs for fully built shafts should not be planed transverse to the line of centres and the practice of stepping webs excessively on the inside surface, in order to obtain a greater shrinkage area for the journal bores whilst maintaining an adequate bottom end bearing surface, is to be deprecated. The resulting notch effects transverse to the plane of bending have produced some serious failures in good quality material.

With the increasing size of shafts today the forgemaster's problems are not becoming easier. To enable the material to be worked sufficiently, the required size of ingot for the really large shafts is approaching the limit of present capacity for some firms. Forgings, unlike castings, exhibit directional characteristics, or "grain flow", the properties transverse to the "grain flow" being noticeably inferior to those in the principal (longitudinal) direction of forging.[3,4] This is particularly so in the case of ductility and impact values, but is also reflected in a pronounced drop in fatigue strength, in some cases as much as 30 per cent. For this reason forging methods in which the principal direction of forging is parallel to the major direct stresses coming on the crank (e.g. due to bending) are to be preferred. The ideal method for a large crankshaft of semi-built type is probably ring-forging [3] in which the ingot is first forged into cylindrical form, the centre containing pipe, segregates etc. removed and the hollow cylinder then expanded on a mandrel and finally flattened at diametrically opposite positions to a shape resembling a chain link from which two separate throws can readily be forged. For large shafts, of course, this method is expensive and today a modified method is by folding a straight cylindrical forging, the ends of which are flattened and bent round to form the webs, care being necessary, of course, to avoid wrinkling and crushing the material at the junctions of pin and webs. Another method of ensuring a favourable amount and distribution of forging work is by the continuous grain flow process [8] in which the webs are die-pressed to oval shape from cylindrical forged bobbin pieces and simultaneously laterally displaced from the axis of the pin to form the throw. The only drawbacks here are limitations of size due to power of press required and difficulty of machining webs all over, which, however, for medium and low speed shafts is no real disadvantage.

The process most commonly used in this country for large semi-built shafts is block forging, in which the ingot is first forged down in diameter to a long cylinder which is then flattened and tapered to a sectional form approximating to the web shape of the finished throw. In this method the axis of the pin is parallel with the axis of the ingot and the central, relatively unsound material can usually be arranged to be removed when machining the web bores for the shrunk-in journals. In common with all block forging methods, the amount of forging work given to the pin is somewhat limited, especially on its underside, which is unfortunate since this is the most vulnerable part of the shaft. The excess material between the webs has, of course, finally to be removed by machining or other means.

A variation of this method has recently been introduced for very large throws in which the axis of the pins instead of being parallel to that of the ingot is at right angles to it, the final forged section of the ingot being roughly rectangular. Whilst there may well be something to be said for this method as a means of avoiding segregates etc. in the critical shaft section, there is no doubt that the amount of forge work given to the pin is even less than in the first method described and the fatigue properties, in particular, are likely to be inferior to those obtainable by other forging methods. The properties could doubtless be somewhat improved if the ingot were sufficiently upset during the forging-down process, but preferably a larger ingot should be employed and forged down by the first method.

As mentioned previously, it is perhaps early days yet to judge the performance of the really large crankshafts, but what evidence there is, does suggest an increasing incidence of cracked and broken shafts as size goes up, especially in the semi-built forged type, although it is only fair to note that, comparatively speaking, this type of shaft has so far recorded one of the best performances from this point of view.

In all the circumstances, in the author's opinion, every possible care should be taken to ensure that the material of these very large semi-built forged shafts is in the optimum condition both as regards soundness and fatigue strength, for which reason only the best forging methods should be adopted.[9]

The influence of heat treatment and cleanliness of plain carbon steels emerges from some results reported by Takeo Yokobori,[10] who found that the fatigue limit of these steels increases inversely with ferrite grain size but is nearly independent of pearlite content (or carbon content) and of inclusion rating, thus pointing to the desirability of efficient normalizing practice.

From other work in Germany [11,12] and elsewhere there is considerable evidence to support the last finding, namely that geometrical discontinuities such as fillets, oil holes, machining grooves etc. are much more dangerous than inclusions in the steel provided these latter are of small size. This tends to line up with findings of Frost, Dugdale and others.[13,14] The first has found that for a given steel and type of notch a relationship of the form:

$$\sigma^3 l = \text{constant}$$

seemed to apply, where σ is the limiting fatigue stress to produce a propagating crack and l is the size of the notch. This suggests that the smaller the absolute notch size, the greater is the limiting fatigue strength.

Among other precautions meriting the consideration of the forgemaster who values his reputation for reliable products could well be the use of vacuum-melted steel and vacuum degassing of ingots, beneficial modern developments aimed at reducing the danger of hydrogen hair cracks in large forgings.[15]

For the smaller shafts, surface-hardening by the nitriding process, preferably on steel of the 3 per cent CrMo (En40) type, undoubtedly confers very large gains so far as fatigue strength is concerned, particularly so where notches are present whether by design or mischance.[16, 17] At the same time, of course, the hardness induced in the pin and journal surfaces greatly enhances resistance to wear. In the author's opinion, other methods of surface hardening are less successful for crankshafts owing to the danger of excessive sub-surface residual tensile stresses, even though the depth of case achieved is greater. For similar reasons the use of nickel and chromium plating of crankshafts is not viewed with favour.[18]

Corrosion Fatigue

One of the most insidious causes of reduced fatigue endurance is corrosion fatigue.[19,20] For a marine crankshaft this can and does occur through excessive acidity in the lubricating oil, often a result of the waste products of combustion entering the crankcase from the cylinders, especially of course in the case of trunk piston designs or crosshead engines in which no sealing diaphragms are fitted. In the presence of water, fresh or salt, from piston cooling or other system leakage, crankshafts can be very severely and rapidly corroded [21,22] with consequent adverse effect on fatigue life. The necessary precautionary measures should be sufficiently obvious.

Design

Web Scantlings [23,24,25,26,27]

In the constant quest for reduced axial length, designers tend to reduce web thickness to the minimum and endeavour to compensate by increasing transverse breadth in order to maintain an equivalent section modulus. Unfortunately, a crank throw is not a simple beam of high span/depth ratio and, further, the stress distribution is complicated by the presence of pin and journal, with positive or negative overlap. Furthermore, it is doubtful if it is legitimate to ignore shear stresses due to bending. The net result is that beyond a certain point any increase in breadth/thickness ratio becomes ineffective. In the author's opinion, this ratio should be limited to a maximum value of 4, if at all possible.

Recessed fillets of generous radius are valuable, provided they do not too severely reduce web thickness locally and, for solid forged shafts, should preferably be used with a useful degree of positive overlap.

Shrunk Webs

With present proportions and, say 28/33 ton steel, there seems little to be gained in grip pressure by exceeding an interference fit of 1/600[28], owing to the local plastic yielding of the material around the eye holes which, however, may well serve to absorb any geometrical mating errors such as out-of-roundness, taper etc.

The fully built throw today is being progressively displaced by the semi-built, since the crankshaft pins and journals are increasing in diameter relative to crank throw and consequently the amount of material left in the bridge pieces between eyeholes is becoming rather critical so far as local yielding from shrinkage stresses is concerned. This, if excessive, has the effect of inducing non-uniformity of shrink grip pressure around the eye holes which is undesirable. In the author's opinion, depending upon the particular design, the ratio of bridge piece thickness to eye hole diameter should be not less than about 0·27.

A further point is the need to limit the minimum axial thickness of shrunk webs in order to avoid undue reduction in bending fatigue resistance tending to cause belling and slackening of web bores. A minimum value of 0·525 times the Rule diameter of shaft is considered appropriate.

The modern practice of omitting dowels has the advantage of more uniform grip and that in the event of sudden stoppage due, for example, to fouling the propeller, "hydraulicking" etc., slippage can sometimes occur without too much consequential damage. On the other hand, there is no denying that in the past dowelled crankshafts with slack shrinks have enabled ships to reach port.

Oil Holes

For large modern engines the trend today seems to favour the elimination of lubricating bores and holes in webs, pins and journals, with their attendant stress concentration effects. This is achieved by feeding the main bearings from the common rail and the bottom end bearings from the crossheads through holes bored in the connecting rods.[29] This is a welcome development which must inevitably yield dividends in the future so far as crankshaft life is concerned.

Where oil holes are drilled in crankshaft journals and pins, care should be taken that the machining of the holes is smooth and free from grooving and preferably the outer portion should be finished by reamering. There has been a number of failures recently where the torsional fatigue crack started in the journal oil hole which was rather rough. Experiment seems to suggest that the maximum stresses occur a little way down the hole rather than at the lip, provided this is well rounded.

Couplings

The knifing of fillets for bolt heads and nuts, if excessive and without adequate radiusing, has sometimes led to fatigue failures of crankshafts.[3] There is much to be said for increasing the outside diameter of the coupling, enabling the bolts to be located at a greater radius, thus both reducing their required size and at the same time minimizing or eliminating the need for arboring. Furthermore, the fillet radius for crankshaft couplings, having due regard to that adopted for crankwebs, need not be as large as is now often provided, and probably a value of about 5 per cent of the shaft diameter is fully adequate.

Bedplates

There is no doubt that cracked bedplates of both cast and especially of welded types, have in the past been the cause of crankshaft failures due to misalignment.[22]

The situation as regards fabricated bedplates around 1957/58 became so serious that in January, 1959, the Society introduced special requirements (new Section H.9) for welded structures for main steam reciprocating and oil engines. These call for plans of the proposed structures to be submitted for approval, together with details of welded joints, materials, electrodes and heat treatment, also an outline of welding procedure, fabrication method and sequence.

The Section includes requirements for materials (e.g. carbon content not generally to exceed 0·23 per cent), welded joints (e.g. main welds taking operating loads to be of continuous, full strength type with complete fusion of the joint).

In addition, with single plate cross-girders, the steel castings for main bearing housings are to be formed with web extensions which can be butt welded to the flange and vertical web plates of the girder. Also stiffeners on the transverse girders are to be attached to the flanges by full penetration welds.

There are also detailed requirements for construction, heat treatment and inspection, the latter including crack detection of transverse girder assemblies.

There seems little doubt that the introduction of these requirements has had a salutary effect on casualty rates in bedplates so far as cracking is concerned and by inference it is expected that this will also be reflected in some measure by a reduction in crankshaft failures due to misalignment. In this connexion Table X may be of interest in showing the reduction in reported cracking of bedplates since the introduction of the new requirements in 1959.

TABLE X.—NOS. OF CRACKED FABRICATED BEDPLATES REPORTED IN YEARS 1953 TO 1962 INCLUSIVE ON POST-WAR BUILT CLASSED SHIPS.

Year	1953	54	55	56	57	58	59	60	61	62
Total cracked bedplates	18	19	44	50	97	100	93	73	46	44

Workmanship

The avoidance of such undesirable features as sharp grooving or notching of shaft surfaces and the provision of a reasonably fine degree of surface finish, also the rounding of sharp edges on webs and bolt hole arborings, will, of course, all help to eliminate or reduce potential sources of fatigue cracking.

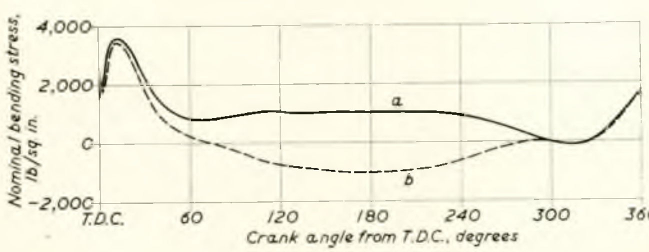
In the same way careful initial alignment of bearings and regular checking and maintenance in service, using both web deflexion readings and bridge gauge measurements (also, when practicable, taut wire or optical measurements of bearing levels) will undoubtedly assist in avoiding bending fatigue type failures.[4,30,31,32]

PART II—THE PROBLEM OF CALCULATING CRANKSHAFT SIZES AND FACTORS OF SAFETY

There must be extremely few crankshaft failures which are not due to some form of fatigue loading.[3, 4, 20, 33, 34, 35]

The actual loading of crankshafts is, of course, complex [21, 24, 36] and, in general, will be a combination of the following: —

i) Normal bending from combustion gas pressures, deadweight and inertia, with some contribution from misalignment of main bearings.
ii) Axial bending and direct stresses arising from:
  a) the "normal" bending moments in (i) above and/or
  b) external axial forces such as propeller thrust variations and/or
  c) inertia forces consequent upon, i.e. coupled with, twist deflexions between cranks due to torque variations or torsional vibrations.
iii) Variable torque forces exerted by the tangential components of the connecting rod loads due to combustion gas pressure, deadweight and inertia.[37]

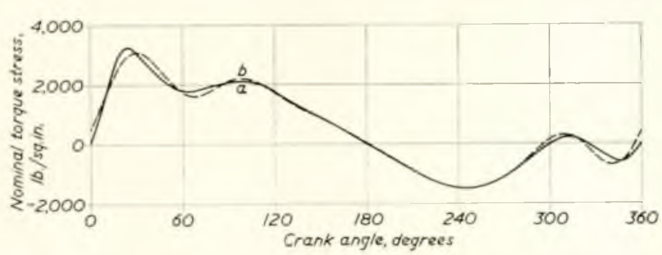


(a) Conventional variation (b) Maximum variation

FIG. 27—*Comparison of typical bending stress curves for a pressure-charged two-stroke cycle engine due to gas pressure, inertia and deadweight*

Fig. 27 shows typical normal bending stress diagrams corrected for deadweight and inertia of reciprocating masses (without stress concentration factor) for a modern, pressure-charged 2-S.C. engine.

Fig. 28 gives the corresponding diagram (similarly corrected) of nominal shear stress (without stress concentration factor) due to torque variation for a single cylinder of the same engine.



(a) Full variation (b) First six harmonics

FIG. 28—*Typical nominal shear stress due to torque variation for a single-cylinder of a pressure-charged two-stroke cycle engine*

These periodic curves may be represented by the synthesis of a series of simple harmonic components of the Fourier type, i.e. the summation of harmonic orders of frequencies which are integral multiples of crankshaft r.p.m. and having various amplitudes and phases referred to, say, No. 1 crank on T.D.C. Unfortunately for designers, each of these harmonic orders will excite some degree of torsional[4, 38] and axial vibration[4, 39, 51], the amount of dynamic magnification depending upon the proximity of the particular forcing frequency to a natural, or resonant, frequency of the complete shafting system and upon the amount of damping present. The amount of energy fed into the vibrating system as a whole will also depend upon the relative phase angles of the harmonic forcing torques and moments at each crank due to the angular intervals between cranks in conjunction with the order number of the particular harmonic. Clearly, from the fatigue point of view, it is important to limit the *total range* of stress, whatever the degree of dynamic magnification of the various orders from resonance effects. The permissible range of stress is also influenced to some degree by the mean values per cycle of the periodically varying dynamic stresses set up in the crankshaft. Another problem is how to combine the component axial and torsional stresses in order to relate them to known fatigue test data based on, in most cases, small smooth polished test pieces. Only a limited amount of data has been published on combined fatigue strength (e.g. references (40)(41)(42)) and in most cases the tests have been conducted with the bending and torsional stresses of the same frequency and in phase.

It is clear that if all the above factors are to be fully taken into account in assessing the strength of a crankshaft, this would represent a formidable task indeed, even with the aid of modern computer methods (see, for example, later).

The Society's rules for oil engine crankshaft scantlings were originally evolved[43] at a time when the significance of torsional vibration was little understood. They were based on the best known practice of the time and were semi-empirical in character taking into account the principal factors of design only, such as bore, stroke, span of bearings, number of cylinders, two or four-stroke cycle, single or double-acting, maximum firing pressure, mean indicated pressure. The formula included coefficients derived from a study of different firing orders and crank sequences and were, in general, so framed as to legislate for the most unfavourable of those likely to be adopted.

The torque variations at each journal were determined by tabular summation of indicated torque, phased in accordance with the firing order, and took no account of inertia of shaft masses, nor of shaft torsional stiffness. A nominal allowable stress of 7,300 lb./sq. in. was laid down in conjunction with a maximum combined bending moment (St. Venant) due to the simultaneous action of bending and torsion (based on indicated pressures). By and large these crankshaft rules have stood the test of time and were not so detailed as to be unduly restrictive to Diesel engine development.

As a result of increasing awareness of the nature and importance of torsional vibration, the Society in 1944 introduced requirements calling for the submission of torsional vibration calculations, the object being primarily to limit critical stresses in the vicinity of the service speed of the machinery, but also to ensure that any speeds within the operating range of the engines at which excessive vibration stresses were calculated to occur, should be restricted or "barred", i.e. not used for continuous operation.

Experience at that time[4] had indicated that a well-designed crankshaft should be capable of withstanding, under continuous operation at the service speed, additional torsional vibration stresses, over and above the normal operating stresses, up to about $\pm$ 2,000lb./sq. in. (nominal). At about the same period the Society issued Guidance Notes covering recommended limits of torsional vibration stress[6] and these have since been revised as experience indicated.

The crankshaft rules have also been revised from time to time, the most important being in 1952 when the diameter formula was amended to reflect implicitly continuous variations in maximum and mean indicated pressures, and otherwise adjusted and brought up to date to give substantially equal sizes as between the rules of Lloyd's Register and the British Corporation.

In this way, therefore, has grown up the present practice of first designing the crankshaft from the Rule formulae for leading dimensions and then calculating the torsional vibration characteristics and if necessary making any possible adjustments to the crankshaft scantlings, weight of running gear, moment of inertia of balance weights, flywheel etc.

This is a practical approach to the problem of crankshaft design, which has certainly to date worked well, in that, as shown in the statistical part of this paper, the incidence of crankshaft failures is extremely low in relation to the total numbers of shafts at risk.

However, there is no doubt that the present procedure suffers from several drawbacks.[44] One is that the same size of crankshaft is asked for, whether there is torsional vibration present at or near the service speed, or not. Another is that the location of the point of maximum torsional vibration stress (usually the nodal position) is not necessarily that section experiencing the greatest *range* of resultant torsional or combined bending and torsional stress. The author has become increasingly conscious of this over the past few years in the light of several crankshaft failures in which the shaft did not break at the torsional node but at sections where the range of total torsional stress variation was a maximum. The author pointed this out in his contribution to the discussion on the recent paper before this Institute by Atkinson and Jackson[25] (1960 loc cit).

It is for these and other reasons that the comparatively recent advent of digital computer methods of calculation[44, 45] has opened up tremendous possibilities for a more discriminating treatment of the problem of establishing minimum crankshaft scantlings for a given type of engine.

As a first step in this direction the Society's Engineering Research Staff has developed suitable programmes for use with the Society's IBM 1620 computer, which will rapidly calculate the total variations of torsional stress at each crank from a knowledge of the indicated torque curve for a single cylinder, together with details of firing order, weights, etc.

These programmes take into account the effective inertia at each crank and stiffness between cranks in addition to damping. As will be shown later, these values of torsional stress variation may be suitably combined with the appropriate values of bending and/or axial stress variation to give the maximum and mean values of stress due to combined bending and torque, from which, by comparison with basic fatigue test data, an assessment can be made of the margin of safety for any given design of crankshaft.

In the author's opinion computer methods similar to those outlined above will ultimately form the basis for the Society's approval of crankshaft sizes.

DETAILED CALCULATIONS FOR A TYPICAL MODERN TWO-STROKE CYCLE TURBOCHARGED ENGINE

With a view to assessing the theoretical factors of safety for an approximately Rule-size shaft, with and without a limiting torsional vibration stress at the service speed, and for the purpose of demonstrating the computer methods outlined above and described in detail in Appendix I, a six-cylinder, 2-S.C. turbocharged engine having the following particulars was selected:—

Bore	760 mm.
Stroke	1,550 mm.
Span of bearings	1,400 mm. (centres)
Span of bearings	1,010 mm. (inner edge to inner edge)
B.h.p.	9,600
R.p.m.	119
M.i.p.	10 kg/sq. cm. (142lb./sq. in.)
Max. pressure	70 kg/sq. cm. (1,000lb./sq. in.)
Crankshaft ...	Semi-built (see Fig. 29)
Material ...	Forged carbon steel of 32-36 tons/sq. in. ultimate tensile strength

Rule size of pins and journals (based on 28/32 ton steel) = 565 mm. diameter

Rule size of pins and journals (corrected for 32/36 tons steel) = 546 mm. diameter

Proposed size of pins and journals = 550 mm. diameter

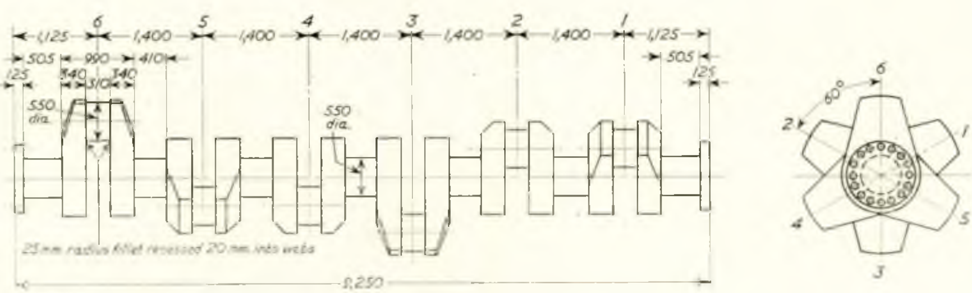


FIG. 29—*Semi-built crankshaft*

Fig. 30 shows a typical indicator card corresponding to the normal service conditions above, from which are derived the 72 gas pressure ordinates as input data to Programme 1. (Appendix 1.)

Crankshaft Torque

Programme 1 also indicates how the necessary corrections for inertia and deadweight of reciprocating masses are made and how the pressure ordinates are converted to shaft torque (see Fig. 28 giving nominal shear stress variation equivalent to shaft torque for one cylinder).

In the conventional method of calculation these torque or shear stress values are summed, either graphically or by tabulation, in accordance with the firing order to give the corrected indicated torques abaft each crank.

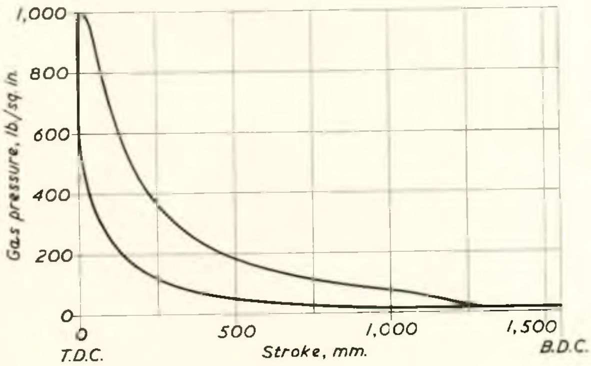


FIG. 30—*Typical indicator card*

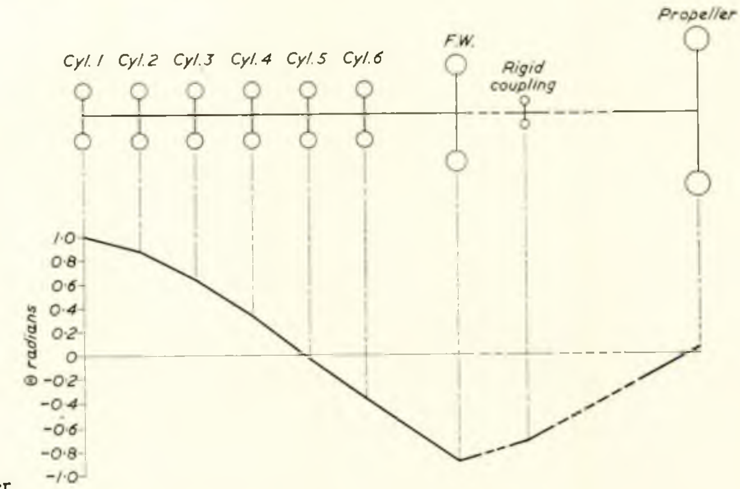


FIG. 31—*Equivalent mass-elastic system*

Reference to Appendix I will show how the computer method, starting from the same input data, makes equivalent calculations, but takes into account the torsional stiffness of the crankshaft and the effective moment of inertia of the shaft masses together with any damping and vibratory effects.

It will be seen that Programme 1 carries out a harmonic analysis to 18 orders and Programme 2 effects a damped forced vibration calculation of the various harmonic components, whilst Programme 3 re-combines them with due respect to phase, adding in increments of mean torque at each crank and finally giving the resultant torque stress variations abaft each mass.

Fig. 31 shows the equivalent mass-elastic system for the subject engine together with the 2-node swinging form for the crankshaft mode of vibration. Table XI gives the corresponding Holzer natural frequency tabulation. From this it will be seen that the 9th order critical would resonate at 122 r.p.m., or just above the service speed of 119 r.p.m.

For purposes of calculation, the inertia of the flywheel has been deliberately increased well beyond the engine builder's practice in order to achieve a sufficient flank effect to provide a torsional vibration stress just equal to the maximum value considered acceptable for continuous operation in accordance with the Society's Guidance Notes, or about ± 1,600lb./sq. in. in this particular case.

The resultant torque variations in terms of nominal shaft stress as given by Programme 3 are shown in Fig. 32. Points to note are the relatively large proportion of the total torque variation contributed by the partially resonant 9th order and the positions along the shaft of the maximum total torque variations. It will be seen that the highest value of the range occurs abaft cylinder No. 2, followed closely by those abaft cylinders Nos. 4 and 5 (see also Table XII). Later calculations will show that the lowest factor of safety applies to the shaft section abaft cylinder No. 5 having particular regard to the higher mean stress at that section, although the factor of safety at No. 4 is very little different.

With a view to comparing the conventional torque calculation with the computer method, it is obviously necessary to remove the dynamically magnified 9th order and it was found that a sufficiently close approximation to the original torque variation for one cylinder (Fig. 28, curve a) was obtained by using only the first six harmonics (Fig. 28, curve b). As will be seen, the maximum deviation in total range is less than 5 per cent. The ordinates of this approximate torque curve were put through the conventional tabular method and also through the computer programmes as described above. The resultant torque variations abaft each mass in terms of stress have been plotted in Fig. 33 and the agreement in total range between the two methods is remarkably good with the exception of cylinder No. 5, and especially No. 6 where the computed torque variation is only some 46 per cent of that derived by the conventional method.

TABLE XI—2-NODE HOLZER FREQUENCY TABLE

Frequency = 1,100 v.p.m. $p^2 = 0\cdot0133\times10^6$ rad.²/sec.²

Mass	$\frac{J}{g}$ lb.-in.-sec.²	$\frac{J}{g}p^2\times10^{-6}$ lb. in.	θ radians	$\sum\frac{J}{g}p^2\theta\times10^{-6}$ lb. in.	$C\times10^{-6}$ lb. in./radian
Cylinder No. 1	42,640	567·1	1·0	567·1	4,493
2	41,810	556·1	0·8738	1,053·0	4,493
3	41,810	556·1	0·6395	1,408·6	4,493
4	41,810	556·1	0·3260	1 589·9	4,493
5	41,810	556·1	−0·0278	1,574·4	4,493
6	42,640	567·1	−0·3782	1,360·0	2,694
Flywheel	121,520	1,616·2	−0·8830	−67·2	427
Coupling	4,950	65·8	−0·7258	−114·9	150
Propeller	230,890	3,070·8	0·0405	9·4	

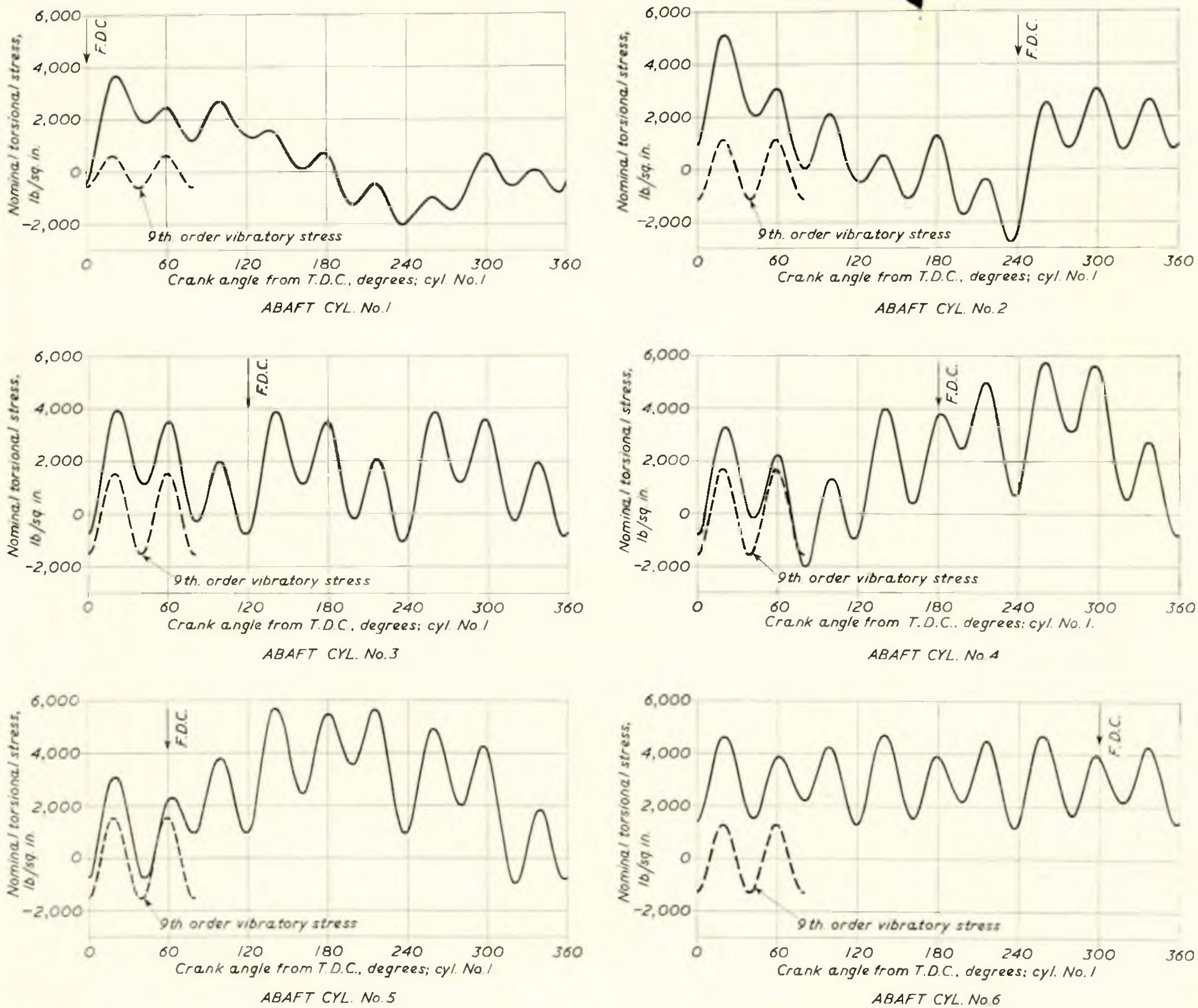


FIG. 32—*Total torsional stresses at service speed (operational plus vibratory)*

Crankshaft Bending Moment

The conventional calculation of bending stresses makes the following assumptions:—

a) The crank is imagined to behave like a straight beam of uniform circular cross-section and the position of the neutral axis is, for simplicity, taken constantly at right-angles to the line of stroke.

b) The span is taken to the *centres* of adjacent main bearings.

c) The constraint of the journals in the bearings is assumed to be "encastré."

d) The loads on the crankpin and journals are taken as concentrated at the centres of the bearings.

e) The level of the main bearings is taken as constant throughout the engine, i.e. no deflexion or unequal wear-down is allowed for.

f) Shearing stresses are neglected, although properly in a relatively short, deep beam such as the crank approximates to, these should strictly be allowed for.

In view of these assumptions, it is obvious that the bending moment so obtained cannot pretend to give more than a comparative estimate of the bending action as it actually occurs. Since, however, most of the above assumptions are on the safe side, the method has been adopted as a standard throughout all the calculations.

The values of bending moment are therefore assumed to be given by:

$$M = \frac{\pi}{4}\,\frac{D^2 p\, L_c}{8} = \frac{\pi}{32} D^2 p\, L_c$$

and the bending stress by

$$\sigma = \frac{32M}{\pi d^3} = \frac{D^2\, L_c\, p}{d^3}$$

where

D = cylinder bore
L_c = span of bearing centres
d = diameter of crankpin and journal
p = net cylinder pressure.

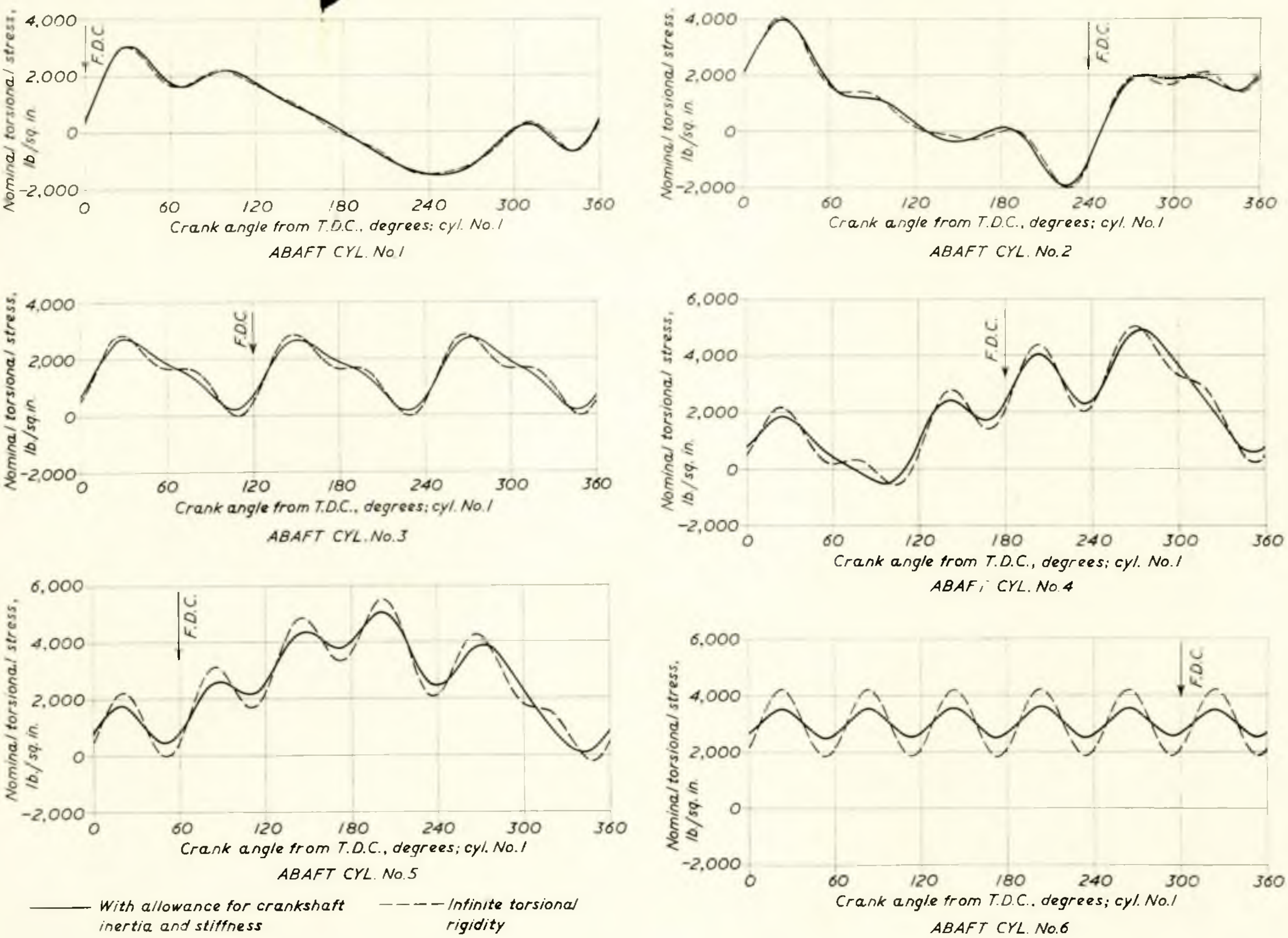


FIG. 33—*Torsional stresses at service speed due to first six harmonics of turning effort*

The foregoing forms the basis for the calculations of Figure 27, curve a.

Curve b represents the more exact calculation of bending stress taking into account the increase in bending load and variation in inclination of neutral axis consequent upon the angularity of the connecting rod. As will be seen, this gives the curve an appreciably different shape with a significantly lower mean stress due to the much larger negative area between about 75 and 280 degrees of crank angle. For the case considered the ratio of the total ranges of bending stress is $\frac{\text{curve a}}{\text{curve b}} = 0{\cdot}80$. It can be shown that the ratio of the ordinates of the two curves is given by:

$$r = \frac{\sigma_a}{\sigma_b} = \frac{\cos\phi}{\cos(\theta + \phi)}$$

where

θ = crank angle from T.D.C.

ϕ = angle of obliquity of connecting rod.

Table XII summarizes the various stresses, both steady and dynamic, as derived from the calculations described. It should be explained that the mean values given are *true* mean, whereas the half-range of dynamic stress is taken as one half of the difference between maximum and minimum.

Combined Torsion and Bending Fatigue

For combined torsional and bending stresses Marin [46] has proposed a fatigue failure relation based on octahedral shear stress, and assuming that the component bending and torsional fatigue stresses are of the same frequency and in phase.

Some justification for this latter simplifying assumption is contained in work reported by Nishihara and Kawamoto [42] who found that the greatest increase in fatigue strength obtained by de-phasing the component stresses was not more than 10 per cent for steels when varying the phase angle up to 90 deg.

The expression for determining the factor of safety according to Marin is given in Appendix II, together with a sample calculation carried out for the stress conditions abaft cylinder No. 5. In order to calculate the actual factors of safety along the crankshaft, it is necessary to choose suitable stress concentration factors to be applied to the calculated nominal values of bending and torsional stress. To this end, the factors 3·0 and 1·6 have been selected as being typical for bending and torsion respectively, and, it is assumed that the same factor applies to both the mean and dynamic parts of each stress, an assumption on the conservative side. These values correspond to the theoretical stress concentration factor, k_t, which therefore infers a notch sensitivity of unity, i.e.

$$q = \frac{k - 1}{k_t - 1} = 1$$

where k = fatigue reduction factor >1.

Some justification for this is contained in recent research

Table XII

		Stresses lb./sq. in.		
		1st—18th Harmonics Flexible	1st—6th Harmonics Flexible	1st—6th Harmonics Rigid
Bending Cylinders 1 to 6	Maximum	3,532	3,532	3,532
	Minimum	−101	−101	−101
	Mean	1,004	1,004	1,004
	Half-range	1,817	1,817	1,817
Torsion Cylinder No. 1	Maximum	3,631	3,074	3,085
	Minimum	−1,965	−1,493	−1,510
	Mean	506	506	506
	Half-range	2,798	2,289	2,298
Torsion Cylinder No. 2	Maximum	5,138	3,978	4,077
	Minimum	−2,771	−1,953	−2,014
	Mean	1,013	1,013	1,013
	Half-range	3,955	2,966	3,046
Torsion Cylinder No. 3	Maximum	3,912	2,744	2,865
	Minimum	−1,118	176	−3
	Mean	1,519	1,519	1,519
	Half-range	2,515	1,284	1,434
Torsion Cylinder No. 4	Maximum	5,753	4,930	5,049
	Minimum	−2,020	−518	−617
	Mean	2,025	2,025	2,025
	Half-range	3,887	2,724	2,833
Torsion Cylinder No. 5	Maximum	5,726	5,060	5,500
	Minimum	−988	79	−215
	Mean	2,532	2,532	2,532
	Half-range	3,857	2,491	2,858
Torsion Cylinder No. 6	Maximum	4,730	3,583	4,199
	Minimum	1,271	2,517	1,877
	Mean	3,038	3,038	3,038
	Half-range	1,730	533	1,161

work on both steel and aluminium, particularly work published by Frost [14] who found that for edge-notched steel plates under reversed direct stress, provided k_t was less than about 4, there was negligible difference between k_f and k_t. Above this value the limiting fatigue strength was apparently little affected by increasing sharpness of notch, i.e. by increasing k_t. Below this critical fatigue stress, which was about $\pm 3\frac{1}{2}$ tons/sq. in., there was a zone in which cracks formed but did not propagate, and below this zone again no cracks were formed. Similar results were obtained with notched round bars under rotating bending stresses.

In any case the assumptions made in the preceding paragraphs are, if anything, on the safe side and the effect on resultant factor of safety is small, as will be shown later.

In a recent C.I.M.A.C. paper [39] factors of safety have been estimated from measured strains on the aftermost crank of a large turbocharged ten-cylinder engine of Götaverken design and manufacture. The tests were carried out both on the test bed and at sea using electric resistance strain gauges of 3 mm. gauge length applied in the recessed fillets between pin, webs and journals. The results enabled both steady and alternating bending stresses, together with steady and alternating torsional stresses, to be derived. These were then combined, as indicated in Appendix II, to give the equivalent steady and equivalent dynamic combined stresses, using the octahedral shear formula in each case. In order to estimate the various factors of safety, these equivalent stresses were then used in the Söderberg [47] and Modified Goodman fatigue failure relations designed to take account of the effect of mean stress for uni-axial stress systems.

From Table XIII, it will be seen that for the subject crankshaft good agreement is obtained as between the Marin and Götaverken - Söderberg methods. The Götaverken - Modified Goodman results are naturally more favourable on account of the more optimistic fatigue characteristics assumed. From the work of Kawamoto and Nishioka (1954) [48] it would seem that the actual mean stress/fatigue limit relation should lie somewhere between the two hypotheses of Söderberg and Modified Goodman. As already indicated, the minimum factor of safety calculated by all three methods occurs abaft No. 5 crank and has a value of approximately 1·5 on the assumptions made.

Table XIII

Cylinder No.	Factor of Safety		
	Marin	Götaverken-Söderberg	Götaverken-Modified Goodman
1	2·31	2·25	2·51
2	1·78	1·75	1·95
3	2·11	2·11	2·49
4	1·60	1·59	1·86
5	1·52	1·52	1·81
6	2·00	1·93	2·21

Effect of Torsional Vibration.

The author is convinced that in the great majority of cases torsional effects will far outweigh bending so far as combined fatigue strength is concerned. Thus, for example, if in a given system there are equal direct and shear stresses of, say, one quarter the tensile yield stress and then in turn first the direct stress is halved and then the shear stress is likewise halved with the direct stress restored to its original value, the percentage reductions in equivalent combined stress on the octahedral shear criterion would be as follows:—

Table XIV.—Equivalent combined stress on octahedral shear criterion

Original	Condition 1	Condition 2	Percentage reduction	
$\sigma = \frac{\sigma_y}{4} = \tau$	$\sigma = \frac{\sigma_y}{8}$; $\tau = \frac{\sigma_y}{4}$	$\sigma = \frac{\sigma_y}{4}$; $\tau = \frac{\sigma_y}{8}$	Condition 1	Condition 2
$0{\cdot}236\sigma_y$	$0{\cdot}213\sigma_y$	$0{\cdot}156\sigma_y$	10	33·7

Other criteria would yield differences of a similar order[25].

Accordingly, the importance of minimizing additional stresses due to torsional vibration should be apparent.

To investigate this point, the factors of safety have been re-calculated omitting the semi-resonant 9th order, for which purpose, as already mentioned, only the first six harmonics of the original single cylinder torque curve (Fig. 28) were included to produce the modified torque curves, Fig. 33. As will be seen from Table XV, the increase in factor of safety is of the order of 20 to 25 per cent.

Table XV

Cylinder No.	Götaverken-Söderberg Factor of Safety		Percentage increase
	1st—18th Harmonics (includes 9th resonant)	1st—6th Harmonics (approximates driving torque)	
1	2·25	2·48	10
2	1·75	2·07	18·5
3	2·11	2·59	25
4	1·59	1·92	21
5	1·52	1·85	21·5
6	1·93	2·18	13

The calculations of bending stress have been made for the assumed encastré bending moment acting on the pin or journal only. In practice, of course, most bending failures start from the fillet on the underside of the crankpin and pass through the web rather than through the pin. One reason for this is that the modulus of a Rule-size web for a solid forged shaft in bending is only about 70 per cent of that of the crankpin, the bending moment being, of course, the same for each at the point of initiation of the fracture. However, as will be shown later, the effect on factor of safety of variations in bending stress is not very marked and therefore, for simplicity, the crankpin or journal stress has been used for both torsion and bending throughout the calculations.

Axial Vibration

With the steadily growing size and length of crankshaft in the modern high-powered turbocharged two-stroke engine, the dangers of axial resonance are becoming increasingly evident[25, 39, 49, 50, 51]. This form of vibration can be excited by one or more of the possible causes already mentioned in the introduction to Part II.

Very little published information is available on the important question of the magnitude of the additional stresses in crankshafts arising from axial resonance, whether independent[25] or of coupled torsio/axial type[4]. However, in reference [39] the authors measured axial vibration fillet stresses in No. 10 crank throw near the service speed at sea of about ± 4,000lb./sq. in. when the axial vibration damper was drained. This was stated to be engine excited from the 5th order harmonic of the crank bending forces, although, since a five-bladed propeller was fitted, this may well have influenced the result.

Accordingly, in order to assess the order of magnitude of the effect of resonant axial vibration stresses on the calculated safety factors, a further calculation has been made in which the normal bending stress range has been increased by ± 4,000lb./sq. in. The results are indicated in Table XVI:—

TABLE XVI

Cylinder No.	Factor of Safety		Percentage decrease
	Götaverken-Söderberg	With axial vibration ±4,000 lb./sq. in.	
1	2·25	1·83	18·5
2	1·75	1·53	12·5
3	2·11	1·72	18·5
4	1·59	1·41	11
5	1·52	1·35	11
6	1·93	1·55	19·5

Misalignment

To deal adequately with this vexed question would require a paper on its own and it is therefore proposed to limit discussion to an assessment of the effect of an arbitrary increase of 50 per cent in normal bending stress range. Some justification for this chosen limit may be found in the paper by Dorey [4] who for reasonable crankweb deflexion values (i.e. 0·02in. on a throw of 29·5in.) calculated an increase of crankpin bending stress of 27 per cent, and further, the Götaverken tests[39] gave substantially similar results when due allowance is made for the much smaller crankweb deflexions on an engine of roughly similar stroke.

The results are summarized in Table XVII:—

TABLE XVII

Cylinder No.	Factor of Safety		Percentage decrease
	Götaverken-Söderberg	With 50 per cent increase in bending range	
1	2·25	1·96	13
2	1·75	1·60	8·5
3	2·11	1·84	12·5
4	1·59	1·47	7·5
5	1·52	1·40	8
6	1·93	1·66	14

It will be seen that for the assumed conditions the reduction in safety factor at the most critical section, i.e. abaft No. 5, is only some 8 per cent.

Notch Sensitivity

In the foregoing calculations of factors of safety, it should be noted that values of 3·0 and 1·6 are selected as being typical stress concentration factors in bending and torsion. According to Marin, quoting Noll and Lipson[52], for theoretical stress concentration factors of 3·0 and 1·6, the corresponding notch sensitivity factors for annealed steels are $q = 0{\cdot}4$ and $q = 0{\cdot}85$ respectively see (reference (46) p. 217). The fatigue reduction factors, which are then used in the octahedral shear theory, are obtained as follows:—

$$\begin{aligned}\text{for bending } k_{fb} &= q(k_{tb} - 1) + 1\\ &= 0{\cdot}4(3{\cdot}0 - 1) + 1\\ &= 1{\cdot}8\\ \text{For torsion } k_{fs} &= q(k_{ts} - 1) + 1\\ &= 0{\cdot}85(1{\cdot}6 - 1) + 1\\ &= 1{\cdot}5\end{aligned}$$

Thus, to illustrate the effect of notch sensitivity, the values $k = 1{\cdot}8$ and $k = 1{\cdot}5$ are used in the calculation of factor of safety at the section abaft cylinder No. 5. The factor of safety so obtained is 1·72 using the Götaverken-Söderberg method, i.e. about 13 per cent increase on 1·52.

CONCLUSIONS

The calculated margin against fatigue failure of about 1·5 appears unexpectedly slender for a practically Rule-size crankshaft, especially as no attempt has been made to take into account the high hoop and appreciable radial static stresses set up by the shrinkage assembly of journals and webs, particularly in the fillets on the underside of the crankpin. Even without the additional allowable torsional vibration stress, which would increase the safety margin to about 1·85, the provision for meeting the occasional unpredictably severe operating condition might perhaps be thought none too generous.

As an approximate check on the figures obtained, it should be noted that the authors of reference (39) calculated a minimum factor of safety at No. 10 crank of 1·9, but taking into account appreciable axial vibration. However, this was on a shaft some 33 per cent above Rule section modulus, so that on a Rule-size shaft the safety margin would have been reduced by about 25 per cent, i.e. to about 1·4. Further, it is clear that in a ten-cylinder engine the maximum total torque variation and range of combined fatigue stress would not occur at No. 10 crank, but somewhere nearer the centre of the crankshaft.

There would thus seem reasonable confirmation that the figures obtained in the paper are at least of the right order of magnitude.

Of course, the test computation of a single Rule-size, six-throw crankshaft does not justify dogmatic conclusions but, nevertheless, the calculations do suggest that the margin of safety is sufficiently tight to demand from designers, metallurgists and operators every possible care in eliminating or minimizing unfavourable factors, including those discussed in the paper. Of these, in the author's opinion, torsional vibration and metallurgical quality merit special attention, whilst observation of the cardinal rules of design in the avoidance of severe stress concentration at highly stressed positions will assist in reducing still further the already low incidence of marine crankshaft failures in classed vessels.

* * * * *

Secondary Resonance and Sub-harmonics in Torsional Vibration[53]

These phenomena have been known theoretically for many years[54] [55], but have only been recognized in practice for marine crankshafts comparatively recently. Within the past few years several large main engines have indicated that measured stress values of certain orders have been much greater than those calculated by the normal linear vibration methods. A theory was developed by Draminsky[53] to explain this and a method of calculation based on non-linear vibrations was outlined in order that this phenomenon might be recognized in the design stage.

Fractures in the crankshafts of at least four ships have brought this phenomenon uncomfortably to light. The torsional fatigue fractures occurred roughly at the nodal positions after periods of service up to about ten years and, after repairs, measurements taken in some cases have indicated torsional vibration stresses at the service speed up to five times greater than originally anticipated. Fortunately, the number of recognized cases has so far been small.

The characteristics of this type of vibration are such that an *nth* order critical of small equilibrium amplitude occurs at or near the service speed, N_s, predicting a stress value which normally would be considered unimportant, while an *(n-2)th* order critical gives rise to a much greater equilibrium amplitude. The critical speed corresponding to this larger harmonic torque, being well above the service speed, is again normally disregarded. It appears that should the system have a considerable second order mass variation, the vibration amplitude of the smaller critical at the service speed can be magnified by the larger in secondary resonance.

The following are details of two recent typical cases.

Case 1

This was a ten-cylinder, 2-S.C. engine which fractured its crankshaft near the nodal point after some ten years' service whilst a sister ship sustained comparable fatigue cracks after a similar period.

The torsional vibration characteristics were as follows:—

Order	*Calculated stresses in crankshaft at resonance* (± *lb./sq. in.*)
II/8th at 1·025N_s	550
II/6th at 1·37 N_s	10,650
II/4th at 2·04 N_s	24,800

Based on these figures, the torsional vibration characteristics were originally accepted, it being considered that the 4th and 6th orders were sufficiently removed above the service speed.

The Society's Guidance Notes state:—"In general, the stresses apply to the effect of a single order only, but cases will arise where account must be taken of the simultaneous effect of the flanks of adjacent orders and these may require special consideration."

Conventional flank stress calculations carried out for the three harmonic orders in question at the service speed, N_s, gave the following results:—

Order	*Stress* (± *lb./sq. in.*)
II/8th	220
II/6th	955
II/4th	1,645
Arithmetical sum =	2,820

The Draminsky calculation (reference (53) pp. 64-65) indicates that the stress at the 8th order resonance point will be ± 2,700lb./sq. in.

It is doubtless only coincidental that the Draminsky 8th order calculated stress agrees substantially in value with the arithmetical sum of the flank stresses. The wave form for the two methods of calculation will, however, be very different, in that for the Draminsky case, the 8th order will predominate.

Damped-forced frequency tables (Programme 2) (see, for example, Table XXIII for Case 2 below) in conjunction with harmonic analysis of torsiograph records taken at the service speed, gave the following results:—

Order	*Stress* (± *lb./sq. in.*)
II/8th	2,840
II/6th	1,040
II/4th	1,580

It will be noted that the measured 4th and 6th flank stresses agree closely with the predicted values by conventional calculation, whereas the measured 8th order stress is some five times greater than the calculated amplitude at resonance.

Case 2

This was a 12-cylinder, 2-S.C. engine which sustained somewhat similar cracking after about 8½ years' service. The corresponding particulars of torsional critical speeds and stresses are as follows:—

TABLE XVIII

Order	Two-node crankshaft stress ±lb./sq. in.
	Calculated Resonant Stress
II/9th at 1·0 N_s	820
II/7th at 1·29 N_s	8,880
II/5th at 1·80 N_s	18,850
	Calculated Conventional Flank Stress at N_s
II/9th	820 (resonant)
II/7th	990
II/5th	1,480
Arithmetical sum	3,290
	Measured Stress at N_s
II/9th	2,480
II/7th	690
II/5th	1,250

Full details of the Draminsky calculation for this case are given in Appendix III, from which it will be seen that the measured 9th order stress of ± 2,480 lb./sq. in. is almost exactly three times greater than would normally have been predicted.

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Appendix I

ANALYSIS OF TORQUE ALONG CRANKSHAFT

Nomenclature

a_n = *nth* order harmonic sine coefficient of the tangential effort of a single cylinder.
b_n = *nth* order harmonic cosine coefficient of the tangential effort of a single cylinder.
j = subscript for mass in the frequency table.
l = subscript denoting position of ordinate of the gas pressure curve plotted on a crank angle basis.
n = harmonic order number.
p = phase velocity of torsional vibration frequency.
θ = angle of rotation of the crank, measured from top dead centre of No. 1 cylinder.
θ_j = torsional vibration amplitude of the *jth* mass in the frequency table.
ξ_j = angle of rotation of crank, measured from top dead centre of No. 1 cylinder at which the *jth* mass fires (if the mass is not a cylinder, $\xi_j = 0$).
ϕ_j = phase angle of vibratory torque abaft the *jth* mass relative to top dead centre of No. 1 cylinder.
ϕ_{nj} = phase angle of vibratory torque abaft the *jth* mass for the *nth* order harmonic.
ψ_n = phase angle of *nth* order harmonic component of the tangential effort of a single cylinder.
ω = phase velocity of crankshaft r.p.m.
ATO_j = operator for forcing vibratory torque for *jth* mass (if the *jth* mass is a cylinder, $ATO_j = 1$; if not, $ATO_j = 0$).
D = diameter of piston.
FH = order number of first harmonic to be investigated.
GP_l = gas pressure in cylinder at the *lth* ordinate when plotted on a crank angle basis.
HM_n = *nth* order harmonic component of the tangential effort of a single cylinder.
I_j = moment of inertia of *jth* mass in the dynamic system.
L = length of connecting rod.
LH = order number of last harmonic to be investigated.
M = total number of masses in the dynamic system.
N = crankshaft r.p.m.
NC = total number of stiffnesses in the dynamic system.
PD = diameter of crankpin.
P_l = net pressure (gas + inertia) at the *lth* ordinate.
R = crank radius.
RE_j = mass damping coefficient at the *jth* mass.
RI_j = shaft damping coefficient in the shafting abaft the *jth* mass.
S_j = stiffness of shafting over section between *jth* and $(j + 1)$*th* masses.
T_l = tangential effort of a single cylinder at the *lth* ordinate when plotted on a crank angle basis.
$TORM$ = mean torque of a single cylinder.
$(\Sigma T)_j$ = vibratory torque abaft the *jth* mass in the frequency table.
$(\Sigma T)_{nj}$ = vibratory torque abaft the *jth* mass for the *nth* order harmonic.
W = deadweight of piston with associated masses, and proportion of connecting rod.
Y_l = torque lever arm of net pressure for a single cylinder.

All torques are based on indicated gas pressures. Any consistent set of British or Metric units may be used.

The normal method used to determine the torques acting at any position along the crankshaft due to the piston forces is based upon the assumption of infinite torsional rigidity of the shaft and zero moment of inertia of shaft masses. This implies that all the forces transmitted to a crankpin in the direction perpendicular to the rotating plane containing the centre lines of crankpin and shaft are utilized in rotating the shaft and that there is no interchange or dissipation of energy in the twisting that occurs between the various sections and in the accelerating of the shaft masses. In the conventional method, the vibratory torques, mainly those under resonant conditions, are determined separately and are generally considered without reference to their phasing with the driving torques.

The computer method developed here takes account of the flexibility of the crankshaft and the inertia of its masses, together with the damping forces, and the vibratory effects, whether forced or resonant, are combined in their correct phases. The method has been separated into three distinct programmes for convenience, because the results of each one, in certain circumstances, may be required independently of the others. However, since the computer results obtained from each are used as the input data for the succeeding programme, the three may be combined into one complete programme, if desired, with a consequent saving in computer time.

Programme 1

This programme determines the harmonic components of the turning effort of a single cylinder due to gas pressure and inertia effects. Although the flow diagram is written for a two-stroke cycle, it can be modified to a four-stroke cycle by doubling the intervals of θ.

Two alternative entry points have been provided in the programme, depending upon the form in which the initial data is supplied. The more fundamental entry requires the gas pressures obtained from the indicator diagram to be plotted on a crank angle basis. The gas pressures at 72 equi-spaced angular intervals (5 deg. for 2-S.C. and 10 deg. for 4-S.C.) are obtained, commencing with the first ordinate after the top dead centre position of the cylinder and ending at 360 deg., or 720 deg., respectively. From these and the crankpin radius, connecting rod length, deadweight, piston diameter and crankshaft r.p.m., the resultant turning effort from the single cylinder due to the gas pressure, deadweight and inertia is calculated for each angular interval.

If the turning effort for each angular interval is available, these values can be inserted at entry point (ii) in the programme.

Whichever type of input data is employed, the computer then proceeds, by means of a normal 72-point harmonic analysis, to determine the mean torque and the magnitude and phase, relative to the top dead centre position of the cylinder, of each harmonic component up to and including the 18th.

The computer time for the more fundamental form of entry, from the initial loading of the programme to the final printing of the results on the Society's IBM 1620 computer, is approximately 25 minutes. If the results are required solely as input data for Programme 2, or are printed on an off-line printer, there will be a saving of approximately 12 per cent in computer time.

There is a parallel programme available in which a generalized harmonic analysis may be carried out, commencing at entry point (ii), for 4m ordinates (where m is any integer). These two programmes may also be used for the analysis of torsiograph or strain gauge records.

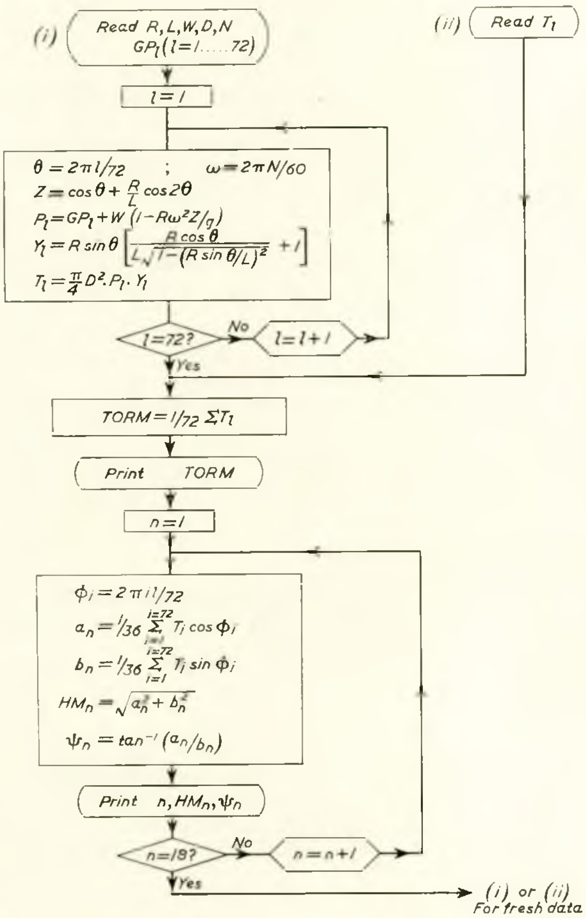


FIG. 34—*Harmonic analysis of single cylinder torque —(Programme 1)*

Programme 2

This programme calculates the forced-damped torsional frequency tables, for either resonant conditions or flank effects, due to the harmonic components of the turning effort for a specific crankshaft r.p.m. A similar programme may be used for investigating a natural frequency of the dynamic system instead of a single crankshaft r.p.m.

The programme can be required to investigate the effect of a range of any consecutive harmonic orders up to and including the 18th, by adjusting the values of FH and LH, or, alternatively, to consider a single order, by making FH = LH, as may be required for a resonant condition. The forcing harmonics are obtained from the results of Programme 1. Allowance is made in the tables for the effect of absolute damping at the masses or relative damping in the shafting.

The calculations are carried out on the well-known Holzer table[37] basis. However, it should be borne in mind that complex numbers are involved and the real and imaginary parts must be kept separate in the computer. These are denoted in the flow diagram (Fig. 35) by the subscripts *R* and *I*, e.g. θ_{1R} and θ_{1I}.

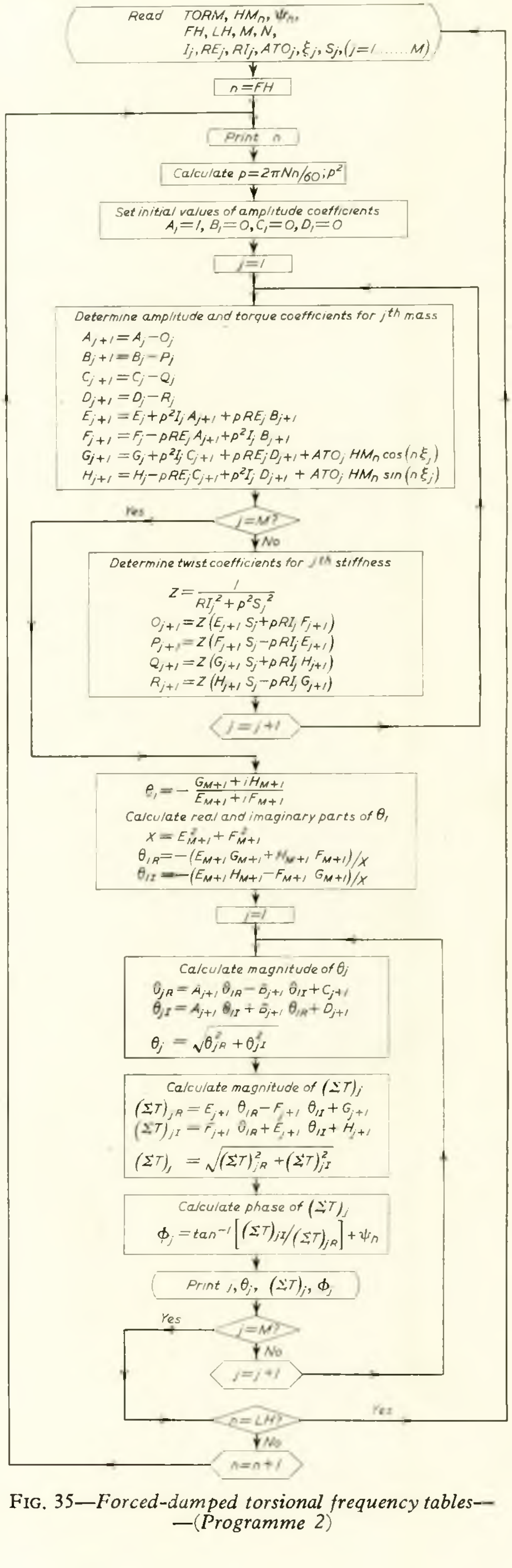


FIG. 35—*Forced-damped torsional frequency tables— —(Programme 2)*

The table is built up by a series of recurrence relations for amplitudes, torques and twists between the masses, all of which are expressed in terms of the amplitude of the first mass, i.e. of the form:

$$\theta_j = (A_j + i\,B_j)\theta_1 + (C_j + i\,D_j)$$
$$\Sigma T_j = (E_j + i\,F_j)\theta_1 + (G_j + i\,H_j)$$
$$\theta_{j+1} - \theta_j = (O_j + i\,P_j)\theta_1 + (Q_j + i\,R_j)$$

The first loop in the programme, i.e. from j = 1 to j = *M*, calculates the normal Holzer table with unit amplitude at the first mass. The final vibratory torque abaft the last mass is then equated to zero and thereby the value of θ_1 is determined. This amplitude will, itself, be of complex form. The second loop in the programme, over the same variables, then substitutes this value of θ_1 in the previously calculated expressions. The final values of the amplitudes at the various masses in the dynamic system and the torques abaft these masses are printed, together with the phase angle of these torques relative to the top dead centre position of cylinder No. 1.

The computer time from the initial loading of the programme to the final printing of the results, in an easily understood form, for a 9-mass, 18-order computation on the Society's computer is approximately 22 minutes. If the results are required solely as punched cards for the input data to Programme 3, or are printed on an off-line printer, there will be a saving of approximately 45 per cent in computer time. From the loading of the data to the final punched cards, one 9-mass frequency tabulation takes 30 seconds.

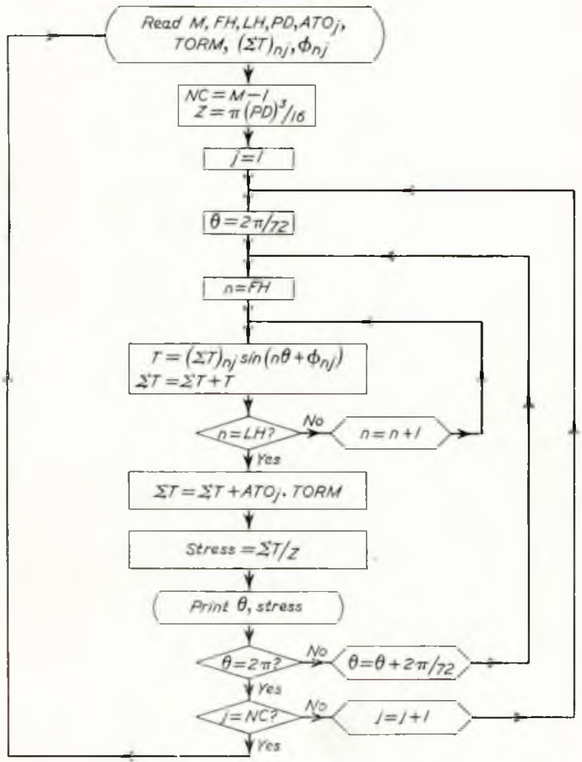


FIG. 36—*Synthesis of crankshaft torque—(Programme 3)*

Programme 3

This programme synthesizes the torques from the results of the previous two programmes and expresses them in terms of stress based upon the crankpin diameter.

If the calculation is dealing with a range of harmonics, the total (vibratory plus mean) torques will be computed. It takes each mass in turn, determines the absolute phase of each harmonic and hence its equivalent torque and then adds these torques over all the orders investigated. In moving aft from No. 1 cylinder it superimposes the mean torque for a single cylinder each time a cylinder is encountered, and finally evaluates the stress. The results are given for the same 72 equispaced angular intervals, measured from the top dead centre position of No. 1 cylinder, as used in Programme 1. These results are printed for each mass.

If the calculation concerns a single order only, such as might be the case when investigating a resonant condition, only the vibratory torques are included. The results may be printed in the same form as for a range of harmonics but the Society's programme is designed to scan the values and print out the maximum positive value of stress and the angle, relative to the top dead centre position of No. 1 cylinder, at which this occurs.

The computer time from initial loading of the programme to the final printing of the results for a 9-mass 18-order computation on the IBM 1620 computer is approximately 100 minutes. Again, if the results are printed on an off-line printer instead of on the console typewriter, the saving in actual computer time is approximately 25 per cent.

APPENDIX II

EQUATIONS USED IN THE CALCULATIONS OF FACTORS OF SAFETY TOGETHER WITH SAMPLE CALCULATIONS

Nomenclature:

n = factor of safety.
σ_m = nominal mean bending stress.
σ_r = nominal half-range of bending stress.
τ_m = nominal mean torsional stress.
τ_r = nominal half-range of torsional stress.
k_b = stress concentration factor in bending.
k_s = stress concentration factor in torsion.
σ_u = ultimate tensile strength.
σ_e = fatigue limit in bending.
σ_y = yield point in bending.

For the material of the crankshaft under consideration:

σ_u = 32 tons/sq. in. = 71,680lb./sq. in.
σ_e = 12 tons/sq. in. = 26,880lb./sq. in.
σ_y = 16 tons/sq. in. = 35,840lb./sq. in.

Typical stress concentration factors selected and used in calculations:

k_b = 3·0
k_s = 1·6

Method 1

For combined bending and torsional stresses, the general expression for the failure criterion as proposed by Marin is:

$$\frac{1}{n} = \frac{1}{\sigma_e}\left[\sqrt{(\sigma_m + \sigma_r)^2 + 3(\tau_m + \tau_r)^2} - \left(1 - \frac{\sigma_e}{\sigma_y}\right)\sqrt{\sigma_m^2 + 3\tau_m^2}\right]$$

With stress concentration factors, the expression becomes:

$$\frac{1}{n} = \frac{1}{\sigma_e}\left[\sqrt{(k_b\sigma_m + k_b\sigma_r)^2 + 3(k_s\tau_m + k_s\tau_r)^2} - \left(1 - \frac{\sigma_e}{\sigma_y}\right)\sqrt{(k_b\sigma_m)^2 + 3(k_s\tau_m)^2}\right]$$

The stress conditions abaft cylinder No. 5 (see Table XII) are:

σ_m = 1,004lb./sq. in.
σ_r = 1,817lb./sq. in.
τ_m = 2,532lb./sq. in.
τ_r = 3,857lb./sq. in.

Substituting these values and values for σ_e, σ_y, k_b and k_s in the above:

$$\frac{1}{n} = \frac{1}{26{,}880} \times$$

$$\left[\sqrt{(3\times 1{,}004+3\times 1{,}817)^2+3(1{\cdot}6\times 2{,}532+1{\cdot}6\times 3{,}857)^2} - \left(1 - \frac{26{,}880}{35{,}840}\right)\sqrt{(3 \times 1{,}004)^2 + 3(1{\cdot}6 \times 2{,}532)^2}\right]$$

$$= \frac{17{,}710}{26{,}880}$$

$$\therefore n = 1{\cdot}52$$

Method 2

Calculations of factors of safety according to the Götaverken-Söderberg method [39] is based on "equivalent" stress. The steady and alternating stresses are first combined and then factors of safety determined, using the Söderberg relation for fatigue failure:

$$f_m = \sqrt{(k_b\sigma_m)^2 + 3(k_s\tau_m)^2} = \text{Equivalent steady stress}$$

$$f_r = \sqrt{(k_b\sigma_r)^2 + 3(k_s\tau_r)^2} = \text{Equivalent alternating stress}$$

$$\frac{1}{n} = \frac{f_m}{\sigma_y} + \frac{f_r}{\sigma_e}$$

For the stress conditions abaft cylinder No. 5:

$$f_m = \sqrt{(3 \times 1{,}004)^2 + 3(1{\cdot}6 \times 2{,}532)^2} = 7{,}637$$

$$f_r = \sqrt{(3 \times 1{,}817)^2 + 3(1{\cdot}6 \times 3{,}857)^2} = 12{,}000$$

$$\frac{1}{n} = \frac{7{,}637}{35{,}840} + \frac{12{,}000}{26{,}880}$$

$$\therefore n = 1{\cdot}52$$

Method 3

The Götaverken-Modified Goodman method gives slightly higher calculated factors of safety by virtue of being based on a slightly more conservative fatigue failure relation.

It can be shown that the expressions for determining factors of safety are:

i) If $\dfrac{f_r}{f_m} < \dfrac{1 - \dfrac{\sigma_y}{\sigma_u}}{\dfrac{\sigma_y}{\sigma_e} - 1}$, $\qquad \dfrac{1}{n} = \dfrac{f_m}{\sigma_y} + \dfrac{f_r}{\sigma_y}$

ii) If $\dfrac{f_r}{f_m} > \dfrac{1 - \dfrac{\sigma_y}{\sigma_u}}{\dfrac{\sigma_y}{\sigma_e} - 1}$, $\qquad \dfrac{1}{n} = \dfrac{f_m}{\sigma_u} + \dfrac{f_r}{\sigma_e}$

iii) If $\dfrac{f_r}{f_m} = \dfrac{1 - \dfrac{\sigma_y}{\sigma_u}}{\dfrac{\sigma_y}{\sigma_e} - 1}$, $\qquad \dfrac{1}{n} = \dfrac{f_m}{\sigma_u}\left[\dfrac{\sigma_u - \sigma_e}{\sigma_y - \sigma_e}\right]$

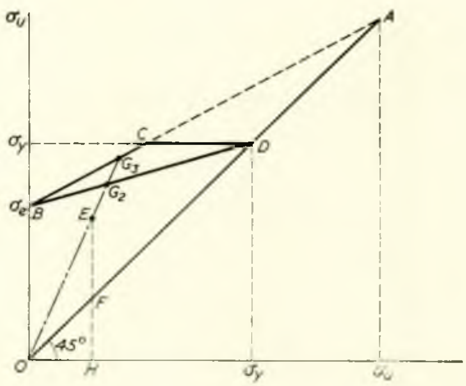
For the stress conditions abaft cylinder No. 5, f_m and f_r are the same as in Method 2,

$$\text{hence } \frac{f_r}{f_m} = \frac{12{,}000}{7{,}637} = 1{\cdot}57$$

$$\frac{1 - \dfrac{\sigma_y}{\sigma_u}}{\dfrac{\sigma_y}{\sigma_e} - 1} = \frac{1 - \dfrac{16}{32}}{\dfrac{16}{12} - 1} = 1{\cdot}5$$

$$\therefore \frac{1}{n} = \frac{7{,}637}{71{,}680} + \frac{12{,}000}{26{,}880}$$

$$n = 1{\cdot}81$$



FH = f_m
EF = f_r
$\dfrac{OG_2}{OE}$ = Factor of Safety, Götaverken-Söderberg (Method 2).
$\dfrac{OG_3}{OE}$ = Factor of Safety, Götaverken-Modified Goodman (Method 3).
BD = Söderberg line.
BCD = Modified Goodman line.

FIG. 37—*Graphical representation (Methods 2 and 3)*

Appendix III

SECONDARY RESONANCE AND SUB-HARMONICS IN TORSIONAL VIBRATION

It is not proposed to discuss the development nor the validity of the theory presented in reference (53). However, a brief outline of the calculation of "secondary resonance" in a 12-cylinder 2-S.C. engine which fractured its crankshaft is carried out and compared with the actual measured stresses.

TABLE XIX.—2-NODE HOLZER FREQUENCY TABLE.
Frequency = 1,085 v.p.m. $p^2 = 0{\cdot}01289 \times 10^6$ rad.^{2}/sec.^{2}

Mass	$\frac{J}{g}$ lb.-in.-sec^2	$\frac{J}{g} p^2 \times 10^{-6}$ lb. in.	θ radians	$\sum \frac{J}{g} p^2 \theta \times 10^{-6}$ lb. in.	$C \times 10^{-6}$ lb. in./radian
Cylinder No. 1	14,535·6	187·297	1·0	187·297	3,402·56
2	14,535·6	187·297	0·9450	364·291	3,402·56
3	14,535·6	187·297	0·8380	521·251	3,402·56
4	14,535·6	187·297	0·6848	649·507	3,402·56
5	14,535·6	187·297	0·4940	742·036	3,402·56
6	14,535·6	187·297	0·2760	793·725	1,962·55
7	14,535·6	187·297	−0·1282	769·716	3,402·56
8	14,535·6	187·297	−0·3542	672·605	3,402·56
9	14,535·6	187·297	−0·5519	569·234	3,402·56
10	14,535·6	187·297	−0·7191	434·547	3,402·56
11	14,535·6	187·297	−0·8468	275·946	3,402·56
12	14,535·6	187·297	−0·9278	102·172	2,065·84
Flywheel	13,280·4	171·187	−0·9773	−65·078	65·1
Propeller	182,280·	2,349·589	0·0223	−12·152	

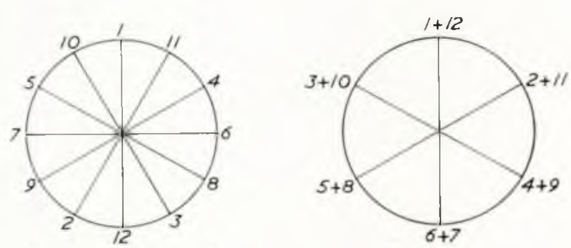
Calculation of stress according to Draminsky

Only a brief outline is given and reference should be made to the original paper. The notation used is as in that paper.

TABLE XX.—CALCULATION OF β FOR VIBRATION FORM A (2-NODE SWINGING FORM).

Mass	△	△^{2}	β_c	β_c△^{2}	$m_{rel.}$	$m_{rel.}$·△^{2}
Cylinder 1	1·0	1·0	0·3	0·300	1·0	1·0
2	0·945	0·89	0·3	0·267	1·0	0·89
3	0·838	0·70	0·3	0·210	1·0	0·70
4	0·685	0·469	0·3	0·141	1·0	0·469
5	0·494	0·244	0·3	0·073	1·0	0·244
6	0·276	0·076	0·3	0·023	1·0	0·076
7	−0·128	0·016	0·3	0·005	1·0	0·016
8	−0·354	0·125	0·3	0·038	1·0	0·125
9	−0·552	0·304	0·3	0·091	1·0	0·304
10	−0·719	0·516	0·3	0·155	1·0	0·516
11	−0·847	0·715	0·3	0·214	1·0	0·715
12	−0·928	0·860	0·3	0·258	1·0	0·860
Flywheel	−0·977	0·954	0	0	0·915	0·872
Total				1·775		6·787

△ = relative vibration amplitudes
β_c = coefficient of oscillating mass per cylinder
$m_{rel.}$ = relative inertia per cylinder



(a) Crank arrangement (b) 2nd order vectors

FIG. 38

Second Order Vector Sum of $\beta_c\Delta^2$

Vertical component = 0·784
Horizontal component = 0·205

$$\text{Resultant } (\beta_c\Delta^2) = \sqrt{0{\cdot}784^2 + 0{\cdot}205^2} = 0{\cdot}81$$

$$\therefore \beta = \frac{\text{Resultant } (\beta_c\Delta^2)}{\Sigma m_{\text{rel.}}\,\Delta^2} = \frac{0{\cdot}81}{6{\cdot}787} = 0{\cdot}12$$

Calculation of Stress

Stress is due to direct resonance of 9th order and secondary resonance of 7th order:

In vibration form A, $\beta_{aa} = 0{\cdot}12$

Equilibrium amplitude of 7th order, $a_{7s} = 6{\cdot}33 \times 10^{-4}$ radians

$$K = \tfrac{1}{8}\beta \frac{q(q-2)}{q-1} = \tfrac{1}{8} \times 0{\cdot}12 \times \frac{9 \times 7}{8}$$

$$= 0{\cdot}118$$

9th order "fictive" amplitude: $b_9' = K \times a_{7s}$
$= 0{\cdot}118 \times 6{\cdot}33 \times 10^{-4}$
$= 0{\cdot}747 \times 10^{-4}$

The direct resonance equilibrium amplitude of 9th order:
$a_{9s} = 0{\cdot}28 \times 10^{-4}$

Neglecting other vibration forms, total amplitude neglecting phase:

$$\theta_o = (0{\cdot}747 + 0{\cdot}28) \times 10^{-4}$$
$$= 1{\cdot}027 \times 10^{-4} \text{ radians}$$

Therefore the calculated stress by Draminsky with damping coefficient $\rho = 0{\cdot}02$, i.e. magnifier of 50,

$$\text{is: } \frac{T_n}{Z} \times \frac{\theta_o}{\rho} = \frac{793 \times 10^6}{1{,}285} \times \frac{1{\cdot}027 \times 10^{-4}}{0{\cdot}02}$$

$$= \pm 3{,}170 \text{lb./sq. in.}$$

where $T_n = 793 \times 10^6$ lb. in. (= Σ Torque)

$Z = 1{,}285\text{in.}^3$ (= $\frac{\pi d^3}{16}$, d = diameter of crankshaft)

Calculation of Vibratory Stresses by Harmonic Analysis of Torsiograph Records and in Conjunction with Forced-Damped Frequency Tables

Using Programme 1 for the harmonic analysis of records, this will give the mean value [YM] and for each order the sine component [$A(M)$], cosine component [$B(M)$], single amplitude harmonic magnitude [$H(M)$] and the phase of each harmonic [$PSI(M)$].

It is normally necessary to magnify the records optically in order that a suitable number of ordinates may be measured for the harmonic analysis.

Table XXI gives the results of a 48-point analysis of torsiograph records taken at the normal service speed of the above engine.

TABLE XXI

$YM = 5{\cdot}5020$

M	$A(M)$	$B(M)$	$H(M)$	$PSI(M)$
1	−1·4061	−1·4190	1·9977	3·9224
2	−0·5237	−0·2523	0·5814	4·2633
3	−0·0482	−0·1206	0·1299	3·5219
4	0·1459	−0·3143	0·3465	−0·4346
5	−0·7432	0·2684	0·7902	1·9173
6	−0·1318	−0 1238	0·1809	3·9583
7	0·4441	0·1271	0·4619	1·2920
8	−0·4269	−0·6387	0·7683	3·7308
9	−1·4312	1·1970	1·8658	2·2673
10	−0·8220	−0·5767	1·0042	4·1005
11	−0·1627	−0·0481	0·1697	4·4246
12	−0·1291	−0·1000	0·1633	4·0535

The measured amplitude of each order is given by:

$$A = H(M) \times 2 \times \frac{d}{S \times D^1} \times \frac{1}{M \times M_o}$$

where D^1 = driving pulley shaft diameter = 148 mm.
d = torsiograph pulley diameter = 148 mm.
S = standard diameter = 148 mm.
M = pen magnification = 3 : 1
M_o = optical magnification = 1·9

Forced-damped frequency tables are given at the end of this Appendix which have been calculated from Programme 2 for the 5th to 9th order harmonics.

Measured stress due to *nth* order.

$$= \frac{\text{Measured amplitude of } nth \text{ order } (A)}{\text{Amplitude at } F.E. \text{ of } nth \text{ order}} \times \frac{1}{Z} \times SUMT$$

where Z = shaft modulus = $1{,}285\text{in.}^3$

A summary of the results is given in the following table:

TABLE XXII

STRESSES (±lb./sq. in.) AT SERVICE SPEED

Cylinder No.	Harmonic analysis of records in conjunction with forced-damped tables		
	5th	7th	9th
1	320	170	760
2	620	330	1,380
3	880	470	1,880
4	1,070	580	2,180
5	1,200	660	2,390
6	1,250(1,480)	690(990)	2,580(820)
7	1,200	670	2,480
8	1,060	590	2,400
9	870	490	2,170
10	600	360	1,750
11	300	210	1,160
12	30	40	430

Note: Figures in brackets are conventional *calculated* flank stresses at service speed.

1) It can be seen that the sum of all flanks comes to ±3,290lb./sq. in. which again happens to be in close agreement with the calculated Draminsky stress.

2) It can be seen that the 9th order resonant stress is considerably magnified (about three times).

3) The arithmetical sum of the measured 7th and 9th orders gives a stress of ± 3,270lb./sq. in. which again happens to agree with the calculated Draminsky 9th order stress.

TABLE XXIII.—FORCED-DAMPED FREQUENCY TABLES
(CASE 2, APPENDIX III)

Harmonic Order No. 5

Mass	SUMT	Phi	Amplitude
1	484938·29	1·93732460	0·00222648
2	934772·19	2·13041840	0·00208399
3	1327358·00	1·93563070	0·00181390
4	1623051·40	2·12713980	0·00142380
5	1821408·70	1·93156250	0·00096028
6	1875841·60	1·73744100	0·00043048
7	1815864·30	1·93147650	−0·00052549
8	1612206·80	2·12827890	−0·00105503
9	1311094·40	1·93534210	−0·00151564
10	913832·41	2·13440670	−0·00190096
11	459046·20	1·93620400	−0·00216474
12	30096·87	5·09759510	−0·00229962
13	164111·14	5·09759520	−0·00228505
14	0·20	5·10885830	0·00023585

Harmonic Order No. 7

Mass	SUMT	Phi	Amplitude
1	277521·73	1·31366580	0·00138422
2	531712·45	1·19894040	0·00130266
3	764064·23	1·31601930	0·00114738
4	932010·77	1·20412100	0·00092283
5	1062719·20	1·32143300	0·00065149
6	1111451·30	1·43532890	0·00033981
7	1075569·80	1·32132250	−0·00022678
8	957146·30	1·20695810	−0·00054137
9	800933·97	1·31578940	−0·00081976
10	578690·11	1·20798660	−0·00105514
11	333580·79	1·31280800	−0·00122426
12	63023·52	1·30675040	−0·00132230
13	92482·76	4·44835040	−0·00135281
14	1·00	1·27800100	0·00006781

Harmonic Order No. 6

Mass	SUMT	Phi	Amplitude
1	196305·42	3·95865710	0·00002792
2	151·43	7·12816470	−0·00002976
3	196604·52	7·10029960	−0·00002971
4	288·27	7·12716800	0·00002806
5	196035·17	3·95862230	0·00002814
6	397·29	7·12462910	−0·00002946
7	196812·84	7·10031630	−0·00002926
8	454·03	7·11918530	0·00002857
9	195915·93	3·95863120	0·00002871
10	466·99	7·11162610	−0·00002886
11	196838·55	7·10028370	−0·00002873
12	434·96	7·10111140	0·00002911
13	2042·06	3·95951140	0·00002933
14	0·00	5·52910000	−0·00000203

Harmonic Order No. 8

Mass	SUMT	Phi	Amplitude
1	83456·03	3·73930070	0·00001479
2	82786·24	4·78058320	−0·00001128
3	3008·94	7·29406240	−0·00002811
4	87617·24	7·88265570	−0·00002724
5	88529·18	6·90574900	−0·00000968
6	4883·30	7·28897490	0·00001901
7	79883·64	3·72022380	0·00002149
8	80623·19	4·80123750	−0·00000817
9	4479·62	7·27452300	−0·00001975
10	87669·34	7·88079830	−0·00001851
11	87458·46	6·89923530	0·00000902
12	2389·71	7·22951780	0·00002778
13	1955·58	4·08791790	0·00002894
14	0·00	4·11130630	−0·00000109

Harmonic Order No. 9

Mass	SUMT	Phi	Amplitude
1	352306·28	2·07035330	0·00161246
2	646160·74	1·97128840	0·00150909
3	877587·36	2·04856930	0·00131936
4	1022161·80	2·03340220	0·00106160
5	1111137·20	2·02041890	0·00076131
6	1206301·20	1·97592960	0·00043487
7	1158538·30	2·02061050	−0·00017983
8	1114374·20	2·03266890	−0·00052007
9	1009547·90	2·04530190	−0·00084746
10	810563·39	1·98161290	−0·00114404
11	540317·95	2·05289820	−0·00138211
12	201489·44	2·01760180	−0·00154080
13	109827·13	5·15920170	−0·00163833
14	0·10	6·78230070	0·00004871

Discussion

Mr. P. Jackson, M.Sc. (Member of Council) said that he found the paper instructive and absorbing but also disturbing. All manufacturers and designers of large marine engines were concerned with torsional vibration problems and, with the resulting vibration stresses and total stresses in the crankshafts, to try to eliminate the failures that had occurred in the past. Lloyd's Register was a kind of parent, looking on, and guiding from analysis of their large quantity of statistical material and reports.

The first part of the paper gave an excellent analysis of the performance of crankshafts under Lloyd's survey for the ten years 1952-62. He wondered whether these analyses took account of results from engines built many years before. Everyone had learned lessons in the intervening period and some of those early shafts should not be taken as examples of modern engines.

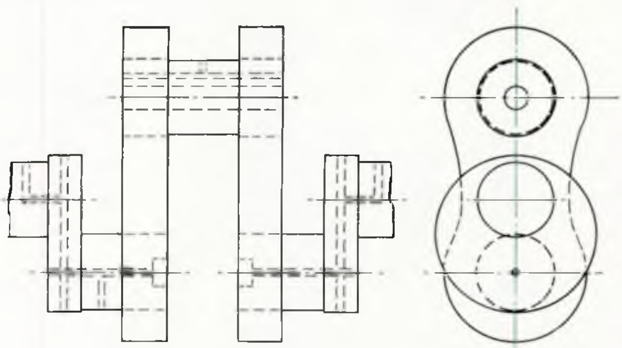
The analysis, as Mr. Archer had said, showed a high rate of breakage for triple cranks, but this had been explained. The series of Doxford 750 cranks did swell the number of breakages and the reasons were explained in a paper (reference 1 of the paper) by Atkinson and the speaker some three years ago, in that that particular crankshaft was unfortunately designed so that there was hardly any overlap of the crankpins and journals and, in addition, there were at least four forms of stress on certain of those shafts and where the stresses were cumulative there had been breakage. For example, in addition to the normal torsional and bending stresses, those engines operating between 105 and 109 r.p.m. were running on the flank of a critical speed. In addition there was some axial vibration at about 107 r.p.m. This was the first time that the company which Mr. Jackson represented had encountered axial vibration. The 750 engine was the biggest engine built by them up to that date. There was also some degree of misalignment.

If those shafts were taken out of the examples, the triple crank engine was equivalent to other engine types. He would have liked the analysis to go further and take more modern engines into account, e.g. there was no single case of breakage on the P-type engine, but as the author had pointed out, there was not sufficient long running experience of such engines.

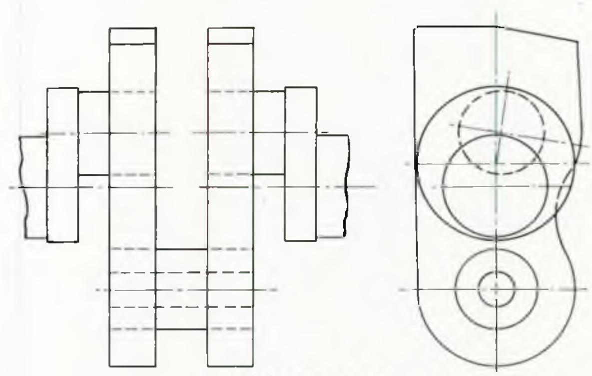
The question of slipped shrinks had also been covered by the author. Engines built prior to the P-type suffered through the possibility of water leaking into the cylinder during standby. This possibility had been eliminated on the P and J engines.

He wondered whether Mr. Archer and his colleagues would not consider modifying their rules to reduce the thickness of main webs at the shrinks, because he had never known a shrink slip due to normal operation. All the slipped shrinks he had known had been due to accidents such as water in the cylinder or somebody trying to start an engine with the turning gear engaged or the propeller fouling and in those circumstances something had to give and the shrink fits were the weak link. Precautions were now taken to guard against these various accidents. A device now prevented starting air flowing to the starting valves while the turning gear was engaged. Nevertheless, such accidents as shock stopping due to a propeller fouling something did occur, but apart from such cases he had known of no cases of shrinks slipping.

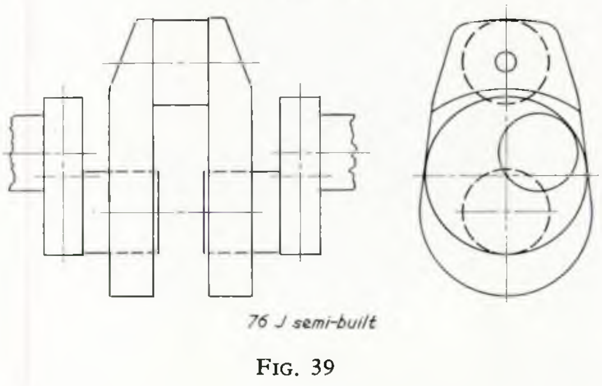
The thickness of main webs due to the shrink requirements was one of the features limiting the design of an engine, together with the requirement to provide adequate bearing area, and he wondered how many cases of failure of crankshafts were caused by inadequate bearings due to designers having to reduce total lengths to secure high natural frequencies of torsional vibration. If crankwebs were unduly thick then bearings had to be reduced. The opposed-piston engine was in a favourable position in this respect since the bearings did not have to carry the combustion loads.



67 LB fully built



67 PT fully built



76 J semi-built

Fig. 39

Mr. Archer had referred to the fact that he had no drawing of the J-type crankshaft to include in Fig. 4 and so Mr. Jackson showed this type relative to the shafts of previous Doxford engines in Fig. 39. The two shafts shown by the author were the 67LB shaft, of which there were well over 200 in service with only two cases of failure, one of them being a slipped shrink due to water in the cylinder, and that for the P-type engine which was designed with a big journal and a big overlap of crankpin and journal, which had removed the concentration of stress of the previous LB crankshafts, particularly of the 750 shaft. The new J-engine had a semi-built shaft as shown in Fig. 39, with the advantage of even greater overlap of the crankpins and increased rigidity due to the web being used as a journal. Being semi-built it was a much lighter shaft and yet more rigid, so that the torsional and axial vibration frequencies were much higher.

Some of the examples given in the paper were very interesting though he wondered whether quite the right explanation was given in every case. For example, in Case I of "Examples of Service Failures" the statement was made that balance weights were put on the crankshaft in order to reduce bearing loads. That would be correct, but the statement that aluminium pistons were fitted also to reduce the bearing loads could not be correct. His guess was that the fitting of balance weights reduced the critical speed to a dangerous level and the aluminium pistons were fitted to correct this. Some of these examples were very interesting and showed the danger of recessed fillets. Doxford's were no longer employing totally recessed fillets, though they did have a few examples of partially recessed fillets. The recesses were limited to one-third of the radius. This again was due to the rules requiring very thick webs.

Every slipped shrink he had known took place in the crankpin and not in the journal shrink of fully-built cranks; therefore there could be a reduction of the rule width for the webs of semi-built cranks. He hoped Mr. Archer would consider this point.

On the question of the loads on the centre main bearings on the six-cylinder four-cycle engines due to the centrifugal forces, all makers now realized that the centre bearing was heavily loaded and had either to be made bigger than the normal bearing or balance weights must be fitted to the crankshaft, but he wondered how far this point had been realized in regard to two-cycle engines. For instance, on the two-cycle nine-cylinder engine the cranks between Nos. 3 and 4 and between Nos. 6 and 7 were at only 40 deg. and, due to this, these bearings were subject to a centrifugal load of over 70 tons on a large single-piston engine which, however, was reduced to 17 tons on an equivalent opposed-piston engine.

He appreciated all the points which Mr. Archer made with regard to bedplates. Mr. Jackson's company had redesigned their bedplate for the P and J engines with longitudinal girders of increased strength to which the transverse sections were welded, but he had previously acknowledged that the difficulties with bedplates had been overcome largely due to Lloyd's specifying that bedplates must be welded from boiler quality steel plate, that no three welds must come together at any point and that transverse girders must be annealed. His company were now annealing complete bedplates.

He had said that the paper was disturbing; he referred to the latter part of course. He agreed that breakages were most likely to occur at the point of maximum stress even when that was not the same as the position of the node. The suggestion that the factor of safety of some crankshafts was as low as 1·35 was a bit frightening. One read of aeroplanes having factors of safety of 1·1 but their life was much less than that of a ship. Fortunately, the methods of calculation and the assumptions that the stresses were cumulative (i.e. the nominal stresses plus an additional torsional vibration stress plus an additional axial vibration stress plus a mis-alignment stress) occurred so very, very rarely, but it was always possible that they would once in a hundred thousand times. However, all these were now being calculated by computer programmes for his company's engine.

The latter section of the paper was also very interesting. He imagined that all those who had been concerned with torsional vibrations had at times measured stresses higher than calculated and had wondered why, and had been disturbed by the differences. He knew of three identical engines, not main propulsion engines but auxiliary types, where the stresses in one were 70 per cent higher than in the other two. Mr. Archer's paper gave some interesting explanations.

He had no doubt that this interesting paper would become a guide to marine engine designers for a long time.

Mr. E. J. Nestorides said that the stress concentration factors of Stahl and Leikin gave indications of the ranges of severity to be expected for crankweb fillet stresses. These could be used in Goodman diagrams, but there was not yet much knowledge of the actual fatigue strength factors (as distinct from the stress concentration factors) which should be associated with such stress raisers. Fatigue tests of crankshafts of various sizes would be needed as well, in order to determine the size effect. For plain shafts, for instance, it was known that a bending fatigue strength of 100 per cent for a 5-mm. specimen would be reduced to 57 per cent of its value for a 300-mm. specimen. For crankshafts the test data available were hardly sufficient for an attempt at a statistical assessment. It was thus understandable that low stress limits were required in practice. A question in this respect was that related to high-U.T.S. steel crankshafts. Should higher stress levels be allowed for these?

Static bending stress calculations of crankshafts were usually based on a single-throw analysis. Calculations by the continuous-beam method gave greater stresses for intermediate throws in some cases and lesser stresses in others. Strain-gauge measurements on crankwebs in running engines might help to clarify these conditions.

An effect of crankshaft vibration was to increase local oil pressures in bearings. At B.I.C.E.R.A. it had been found that the amount of solid contaminants embedded in bearing shells was increased by high pulsating pressures and this led to increased bearing and journal wear. Thus it seemed wise to limit crankshaft vibrations to moderate amplitudes for this reason, irrespective of the corresponding crankshaft vibration stresses.

The stresses due to crankshaft vibrations could also be accentuated by bearing flexibility and crankcase distortion. If the main bearing supports deflected axially, the slope of the journals increased the opening of the crankwebs and added to the web stresses.

The evalution of complex-number Holzer frequency tables was now being extended to gear-branched systems. The results agreed with those obtained with an analogue computer. For exploratory work of this kind he felt that the analogue computer (in fact, an electronic integrating machine, also known as a differential analyser) was more versatile than the digital computer.

In particular, for a gear-branched system investigated with the analogue computer, it was possible to determine the vibratory torques, taking account of gear flexibility, and thus to obtain resonance curves similar to those determined from torsiograph records. In this connexion he was indebted to Mr. Zdanowich, who first pointed out to him the importance of gear housing flexibility. Static load/deflexion tests of gears (one gear clamped to the crankcase and another loaded by means of a torque arm) recently carried out on an engine, showed that the additional gear deflexion due to housing flexibility could be 50 to 100 times greater than the flexibility of two spur-gear teeth, calculated by the usual formulae. This affected the frequency and amplitude calculations appreciably.

Regarding Dr. Draminsky's interesting evaluation method for sub-harmonic vibration, he wondered if it had been used to indicate not only large effects but also negligible effects, for engines in which sub-harmonic effects were found to be negligible. The validity of calculations rested on experiments and further examples would therefore be useful, particularly since for Case 1, quoted by Mr. Archer, the usual damped-forced frequency tables already gave something like the desired stress level.

MR. R. W. ZDANOWICH, M.A., said that the paper was one to be treated seriously: it was based on the author's long and varied experience as the head of a section of an organization for which everyone had respect, and thus summarized the effects of a large variety of working conditions on an equally large variety of types of engines.

His first question referred to the first case of the examples of service failure described on page 82 of the paper. It was presumed that the inertia of the generator was large compared with that of the engine. Although no details were given, it was further assumed that there was some kind of quill or coupling between the engine and generator. One could reasonably argue that in such an installation the node was close to generator and the engine system at the anti-node. Under these conditions the large displacements of the engine, combined with negligible damping coming from the generator, may well have resulted in vibratory conditions more severe than those estimated with the aid of usual assumptions and, in the absence of an efficient damper, had led to engine failures. Incidentally, for " generator " one could well read any large output mass. Extending the argument one could reasonably deduce that so many auxiliary plant failures, like magnetos, superchargers, camshafts, water pump drives, etc., which seemed to plague the industry from time to time, even though, on paper, very lightly stressed, could well be due to the same phenomenon i.e. to the existence of the so-called auxiliary modes in which the part in question vibrated in opposition to the rest of the engine, with the node very close to the latter, resulting in negligible damping coming from the engine and the magnifier being mainly that due to the hysteresis effects in the shaft driving such auxiliaries. It was thought that the value of such a magnifier might well approach even 200 thus leading to severe stresses. The auxiliary modes were difficult to evaluate on a desk machine but an electronic computer, if properly programmed for, could give them with ease. The author's well-considered views on the subject would be appreciated.

The second example on page 83 was a warning of how unwise it was to be penny wise and to try to salvage a nitrided crankshaft by regrinding it.

Reading the descriptions of other failures one was forced to a further conclusion that it was again unwise to be penny wise and dispense with the assistance of Lloyd's when building or repairing ship's machinery.

Turning to the problem of axial vibrations, Mr. Zdanowich said he would be the last person to suggest that axial vibrations did not exist. Quite a number of papers had been written on the subject and he had read a good many of them, but he wished to be convinced beyond any shadow of doubt that what had been recorded as severe axial vibrations were not in fact periodic shortening and lengthening of crankshafts due to pure and simple torsional oscillation. A paper* was read in 1939 by Dr. S. F. Dorey before the North East Coast Institution of Engineers and Shipbuilders, and in the discussion which followed most of those who took part were of the opinion that axial vibration was only of importance when of the same frequency as the torsional. This could be interpreted in two ways—either that the frequencies of the two types of vibration coincided (an unlikely occurrence) or that, what was measured was, in fact, a forced axial oscillatory motion, caused by torsional vibration. What he wished to see was some examples of failures which were due to pure and simple axial vibrations and not to the proximity of torsional vibration. The foregoing was a very abridged version of his contribution† to the paper by Mr. R. Poole to the Institution of Mechanical Engineers. Since there appeared to be a revival of interest in the subject of axial vibrations, Mr. Zdanowich suggested that the contribution could well be referred to by those interested.

His next point concerned the so-called secondary resonance and sub-harmonics in torsional vibration. It was a well known fact that in multi-cylinder engines the forcing impulses contributed at any given instant by various cylinders were by no means equal. Quite substantial variations from the mean were usual and if an allowance were made for these inequalities, the severity of some minor orders could easily become ten times greater than in the case of equal contributions.

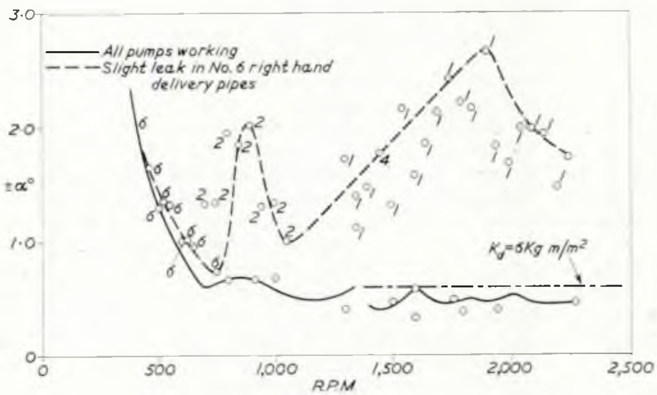


FIG. 40—*Junkers Jumo V—recorded vibration and amplitudes*

Fig. 40 illustrated this point. It was prepared from vibration records on their old friend the Junkers Jumo V engine. The dotted line showed the effects of a slight leak in the right-hand fuel delivery pipes; the left-hand pump and pipes were working normally. There was nearly a five-fold increase in the severity of the first output order at under 2,000 r.p.m. and about a four-fold increase in the corresponding second order at about 800 r.p.m. When the leak was rectified, the amplitudes were reduced to normal, shown in plain lines. The sixth order node-at-gears at under 500 r.p.m., being a major one, was only slightly affected. The important thing to note is that the phenomenon observed was not in any way affected by the dynamic relationship described by Dr. Draminsky.

Another interesting example was the appearance of the second minor order in a 12-cylinder, six-throw crankshaft, two-stroke engine. The calculated value of the free-end displacement was slightly over ±1 deg. In practice the measured displacement varied between under ±1 deg. and over ±7 deg., depending on the shape of the induction manifold fitted. To repeat, variations of the induction manifold led to free-end amplitudes varying between approximately 1 deg. and over 7 deg., again a phenomenon not in any way dependent on the Draminsky relationship.

Coming to more recent examples, a well faired elbow in the induction pipe of an 18-cylinder engine could produce a three-fold difference in the severity of the minor orders, compared with a straight-through induction pipe. Such examples could be multipled *ad infinitum,* but the speaker was not going to suggest that Dr. Draminsky had evolved his theory just to fit the facts; his standing and reputation as a serious scientist was second to none. It would, however, be interesting to know whether the inequalities of impulses had been considered by Dr. Draminsky in the preparation of his paper and whether his conclusions would have been the same or whether his paper would have been written at all if that phenomenon had been taken into account.

Professor D. C. Johnson of Cambridge said, when approached by the speaker, that the phenomenon undoubtedly existed. Some experiments were done in Cambridge a few years ago demonstrating the effect and previously it had been observed on a single cylinder air compressor by a worker in Newcastle. In a multi-cylinder engine the effects arose equally well from unequal firing impulses in the various cylinders and

* Dorey, S. F. 1939. " Strength of Marine Engine Shafting." Trans.N.E.C.I.E.S., Vol. LV, p. 203.

† Poole, R. 1943. "Axial Vibration of Diesel Engine Crankshafts". Contribution to discussion by R. W. Zdanowich. Proc.I.Mech.E., Vol. 148, p. 200.

for this reason there was rarely much need to worry about the Draminsky effects.

The single cylinder case was exceptional because there was then no other source of sub-harmonic and one might get into trouble if sub-harmonic resonance were not considered at the design stage. This was probably only on importance in the last compressor.

The final part dealt with damping or its reciprocal, the dynamic magnifier. It was an elegant method whereby damping was included in the modified Lewis or Holzer tables. Providing one could be reasonably certain of damping and providing one dealt with simple ungeared systems, as described by Mr. Archer, the results were obviously reliable.

When dealing with complex-geared systems consisting of anything up to 80 masses, half of which might be gears, such a method had nothing to commend it. There were excellent reasons to believe (as would be shown shortly) that most damping came from the gears, but not to the same extent from each pair, so it would be quite wrong to put the damping at each crank throw only. In such cases, a statistical approach to determine the overall damping for each type of engine would seem to be the only rational one. An attempt had been, in fact, made by the speaker and data collected, relating to close on a hundred readings on many types of engines, both geared and ungeared.

Figure 41 showed that about 60 different methods had been tried in an attempt to correlate some of the engine parameters with the value of the dynamic magnifier. Of all these only the two marked with crosses seemed to show any promise.

Figure 42 showed method No. 2 and was proof of how important the presence of gears was in reducing the severity of torsional oscillations. Other things being equal, gearing more than doubled the damping and so, for a directly-driven large slow speed marine engine, the magnifier could well approach

A = Swept area of cylinder wall
γ = Equilibrium stress = f_0
V_E = Engine swept volume
V_C = Cylinder swept volume
f = Frequency
n = Order number
N = R.P.M. of n

No.	'A'	'B'	No.	'A'	'B'
1	$\frac{M}{V_E}$	n	30	$M\gamma/V_E\Sigma\theta$	f
2		f ×	31	$1/M$	$1/n^2$
3	$\frac{M}{V_E f}$	n	32	$1/M$	$1/n$
4		f	33	M/V_E	$\gamma = f_0$
5	$\frac{M}{V_E n}$	n	34	M	$\gamma = f_0$
6		f	35	$(M/V_E)^{\frac{1}{3}}$	γ
7	$\frac{M}{V_E f^2}$	n	36	$Log(100M/V_E)$	$Log\ \gamma$
8		f	37	V_E/M	f
9	$\frac{M}{V_E n f^2}$	n	38	M	L.H.P
10		f	39	Mf	L.H.P.
11	$\frac{M}{V_E n f}$	n	40	M/f	L.H.P.
12		f	41	M	L.M.E.P.
13	$\frac{M}{V_C N}$	n	42	Mf	L.M.E.P.
14		f	43	M/f	L.M.E.P.
15	$\frac{M}{V_C N f}$	n	44	M/f	1/L.M.E.P.
16		f	45	$Log(Mf/V_E)$	$Log\ V_E N$
17	$\frac{M}{D^2 n f}$	n	46	$Log(Mf/V_E)$	$Log\ V_E$
18		f	47	$Log(Mf/V_E)$	$Log\ f$
19	$\frac{M}{V_E N}$	n	48	$Log(Mf/V_E n)$	$Log\ V_E$ ×
20		f	49	M/f	1/L.M.E.P.
21	$\frac{M}{V_E N f}$	n	50	$Log(M/n)$	$Log\ V_E$
22		f	51	$Log(M/n)$	$Log\ f_1/f$
23	M	V_E	52	M	$10^5 f/V_E^2 N f_1$
24	M	$V_E f$	53	M	$10^5/N V_E^2$
25	M	$V_E f^2$	54	M	Vibration stress
26	M	$V_E N$	55	$M/\sqrt{V_E}$	f
27	$M V_C/V_E$	f	56	M/f	V_E
28	M/A	f	57	MN/V_E	V_E
29	M/A	f	58	$Mf/V_E N$	$1/V_E N$
60	M	$1/f$	59	$Mf/V_E N$	$f/V_E N^2 n$
61	$Mf/V_E N$	$1/V_E N^2$	62	$Mf/V_E N$	$f/(V_E N^3 n)$

FIG. 41—*Dynamic magnifiers: correlation of experimental data relationships plotted "A" being plotted against "B"*

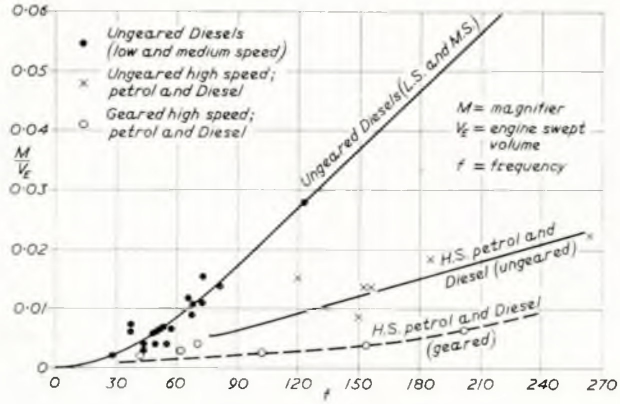


FIG. 42—*Dynamic Magnifiers*

a hundred, for a small, high speed, geared engine, it could be under ten. There was, unfortunately, a large scatter of points, and he only showed the graph as a matter of interest because the problem could well be approached from other aspects by other people.

Various investigators had tackled this problem of gear flexibility in a number of ways. Some said it was a transverse deflexion of the shaft on which the gears were mounted which was the main source of flexibility, others argued that it was only the bending of the gear teeth. In his firm's experience, the chief cause of gear flexibility and the resulting de-tuning was the distortion of the crankcase, as measured by the angle which a line joining the centres of two gears under load made with the same line when the load was removed. Angular displacement of this line was made up of actual distortion of the casing and movement due to the displacement of oil from the bearings. The speaker had reason to believe that it was this displacement of the oil and shearing of it between the teeth which were the two major sources of damping. Admittedly, his experience was based mainly on high speed, lightly built engines.

It is highly gratifying to feel that such an authority on vibration problems like Mr. Nestorides was likewise inclined to believe that the casing distortion was the one major source of detuning.

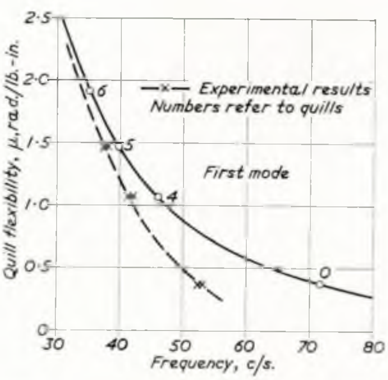


FIG. 43—*Calculated frequency versus quill flexibility*

Fig. 43 showed the effects of quill shaft flexibility on frequency. For a very flexible quill the de-tuning was a matter of a few per cent. For a rigid quill such de-tuning approached 40 per cent, power output, speed and all dimensions, except for the diameter of the quill, remaining the same. The quill was so designed that it did not transmit bending and the gear was independently mounted in two ball bearings, one on either side of the gear flange.

The firm with whom he had the privilege to be associated had drawn up an elaborate programme of research towards the

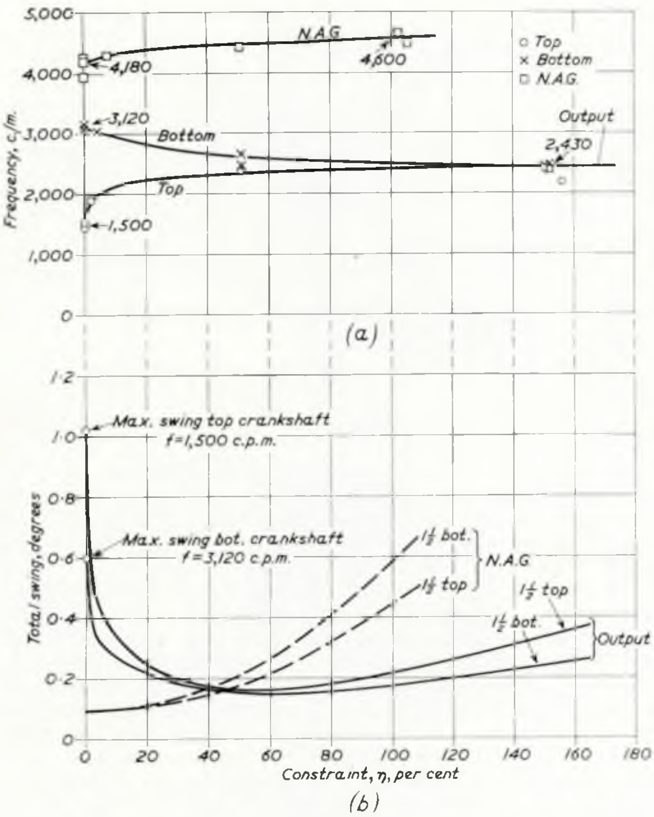


FIG. 44(a)—*Recorded frequency versus constraint*
(b)—*Recorded amplitudes versus constraint for 1½ order*

end of the war, unfortunately, except for some preliminary tests, the work was not proceeded with.

Fig. 44 showed how involved the problem was. It was an engine of the Sabre type with two crankshafts, normally geared to one propeller shaft. In this version each crankshaft was geared to its own airscrew shaft and carried its own propeller —the two shafts being coaxial—and there was a synchronizing gear between the two crankshafts.

The top graph showed the relationship between so-called percentage constraint and the frequency, and the bottom graph showed again the same percentage constraint versus displacement due to the 1½ order. By constraint was meant the ratio of the total torque passing through the synchronizing gear to total engine torque, it could thus be more than 100 per cent. Looking at the top graph (a) it could be seen that with zero constraint the output modes for the crankshafts were completely independent of each other, the two frequencies being 1,500 and 3,120 cycles/min. and the corresponding swings 1·2 deg. and 0·6 deg. respectively. The node-at-gears (N.A.G.) was negligible, under 0·1 deg. swing, with a frequency of 4,180 cycles/min.

Application of constraint very rapidly reduced the swings of output mode until, with 50 per cent constraint, the two displacements became similar, but it was not until 150 per cent constraint was reached that the two crankshafts merged into one system possessing a common frequency of 2,430 cycles/min.

Regarding the node-at-gears mode, increase of constraint rapidly increased the severity and, if the strength of the component parts of the engine made it possible to increase the constraint in this mode to beyond the 110 per cent reached, displacements higher than those shown could well be attained.

The interesting thing about the node-at-gears mode was that the increase of constraint stiffened up the system, thereby raising the frequency from 4,180 at zero constraint to 4,600 at 110 per cent constraint.

These tests led to two important conclusions.

The first suggested that the backlash between the teeth was of no consequence. There were no torque reversals throughout the tests, i.e. no separation of the teeth, and the results shown in Fig. 44 could not have been in any way affected by the magnitude of the backlash.

The second explained the reason why the dynamic magnifier, other things being equal, was so drastically reduced by virtue of the presence of the gears. Taking the case of an engine with two crankshafts geared to a common output shaft, it was clear that the constraint could never exceed 50 per cent. Reference to the graphs would show at once that, under these conditions, neither the node-at-gears mode nor the output mode could reach the full severity. This was, of course, additional to the advantage of extra damping, already referred to.

It was hoped that this short summary would clearly show that the question of gear effects was a highly complex one, fully deserving some methodical and really careful research. Hitherto, the few investigators who endeavoured to tackle the problem concentrated on one or two less important aspects of it. This, unfortunately, did not help to elucidate this involved subject, the more so as their findings were frequently obscured by quite unnecessarily involved mathematical reasoning. A practising engineer who was invariably pressed for time and frequently did not know which way to turn, simply could not make the effort to study in detail such otherwise excellent papers, with the result that the findings were either lost to the industry or insufficient notice taken of them, or even wrong conclusions derived from them.

He wished to thank D. Napier and Son Ltd. but particularly the firm's chief engineer, Mr. R. H. Chamberlin, for their permission to read this contribution.

MR. S. OLSSON expressed his appreciation of the paper and said that as a crankshaft designer he was well aware of the difficult problems involved in the calculation of the stresses and the designing of the features of big crankshafts. It was, therefore, very valuable and encouraging to see the satisfactory statistics on crankshaft failure in spite of the very low safety factors calculated by the author for a rule-sized crankshaft.

He was very glad and proud to hear that the author had found that the experimental investigations and theoretical work, carried out by Mr. Olsson's company and published at the C.I.M.A.C. in Copenhagen in 1962, worthwhile describing in his paper and had to a certain extent served as a basis for his calculations of the safety factors.

There were, however, some facts to which Mr. Olsson wished to draw attention. When calculating the safety factors for the subject crankshaft, the author used what he called the Götaverken-Söderberg method. As was pointed out in the paper, this method was based on the failure criterion described by a straight line between the fatigue limit at zero mean stress and zero dynamic stress at yield point. However, a great number of tests had shown that the mean stress had a rather small influence on the fatigue limit. Therefore his company were of the opinion that the second method described in their paper, referred to by the author as the Götaverken-Modified Goodman method, was more realistic. If this method was applied, the safety factors, as pointed out, were more favourable. However, the author had in his calculations used a bending fatigue limit of 26,880lb./sq. in. for the crankshaft material. This figure was a little too optimistic when due consideration was taken of the "size effect" and surface roughness. According to tests, published by several investigators, such as Horger, Faulhaber, Eaton, Lehr, Moore, etc., the bending fatigue limit on big shafts was about 30 per cent lower than on small specimens. In recent Japanese tests on shafts with shrink fits, a size effect of the same magnitude had been found. In the C.I.M.A.C. paper they had used the fatigue limit arrived at by push-pull tests on small specimens which, for the material in question, was about 20,000lb./sq. in. This was considered to include the lowering effect due to the surface condition. When applying this fatigue limit to the calculation of the safety factors these would again decrease to values about equal to or just less than those found by the author. For example, on the journal abaft cylinder No. 5 the safety factor would be 1·41 instead of 1·52.

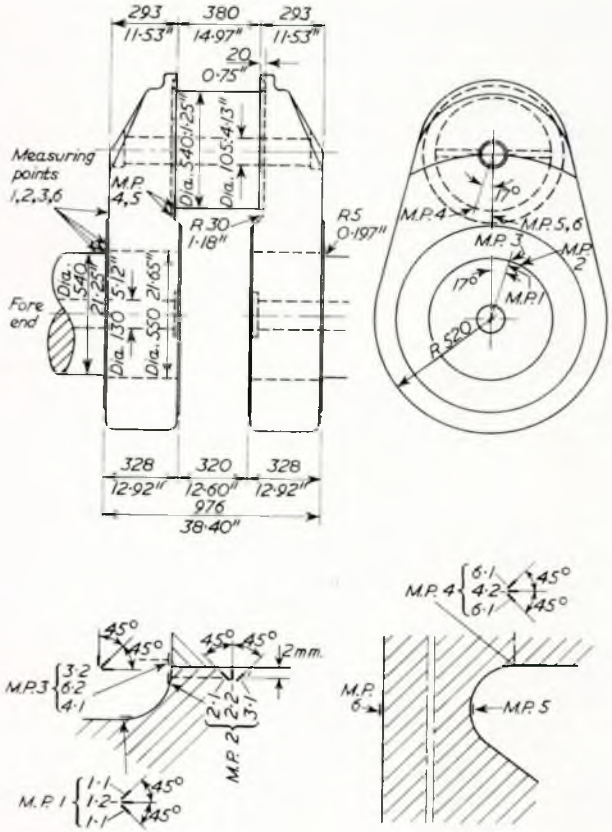


FIG. 45—*Engine DM 760/1500 VGS 7U: crank 7, crank dimensions and arrangement of gauges*

As a complement to, and confirmation of, the stresses calculated in Fig. 32 and Table XII he wished to show the results of some strain-gauge measurements carried out, by his company, on the crankshaft of a seven-cylinder engine, after the C.I.M.A.C. paper had been written. The crankshaft had a diameter of only one per cent over the rule diameter. The measurements were performed on the fore web of crank No. 7 and in the adjacent fillets of both pins during the shop trials. The scantlings of the crank and the measuring points could be seen in Fig. 45. Due to the short shaft between the engine and the rather heavy brake a torsional vibration critical of the seventh order occurred above and close to the service speed. The pure torsional vibration stress in measuring point No. 1, in the fillet on the journal between crank Nos. 6 and 7, at the service speed was ±1,560lb./sq. in. as shown in Fig. 46. The nominal stress in the journal was calculated to ±970lb./sq. in. based upon the forced vibration frequency tabulations and the amplitude at the fore end of the shaft measured by torsiograph. The safety factor was 1·46 when using the Götaverken-Modified Goodman method and taking into account that the shaft material had an ultimate tensile strength of about 63,000lb./sq. in. and an estimated fatigue strength of about ±17,000lb./sq. in.

At the same trials, the stresses measured in the crank pin fillet were much smaller (*vide* measuring point 4). The safety factor for this point was 2·37. To what extent this depended on the bigger fillet radius or the restraining action of the bearings on the transverse movement due to twisting of the crankpin was very difficult to say.

Concerning the effect of axial vibrations on the safety of a crankshaft he agreed with the author that generally they were of less importance than torsional vibrations.

However, if the resonance occurred within the full speed range the permitted nominal torsional vibration stress was ±1,560lb./sq. in. Assuming a stress concentration factor of 1·6 and converting it into bending stress by multiplying by 1·73, the result was ±4,300lb./sq. in. The axial vibration stress measured in the fillet on the crankpin No. 10 and published in the C.I.M.A.C. paper, was without damping effect in the damper, ±5,350lb./sq. in. corresponding to an amplitude at the forward end of the crankshaft of ±0·055in.

As could be seen from these figures, axial vibration ought to be taken into consideration when the amplitude was of the magnitude of ±0·05in.

The author's conclusion that the computer methods had opened up tremendous possibilities of making better technical calculations was fully agreed with. Mr. Olsson's company had, for four years, made all their torsional vibration calculations on a digital computer. Now they used it also for the calculations of axial vibration, ship vibration, whirl, etc. They had also made

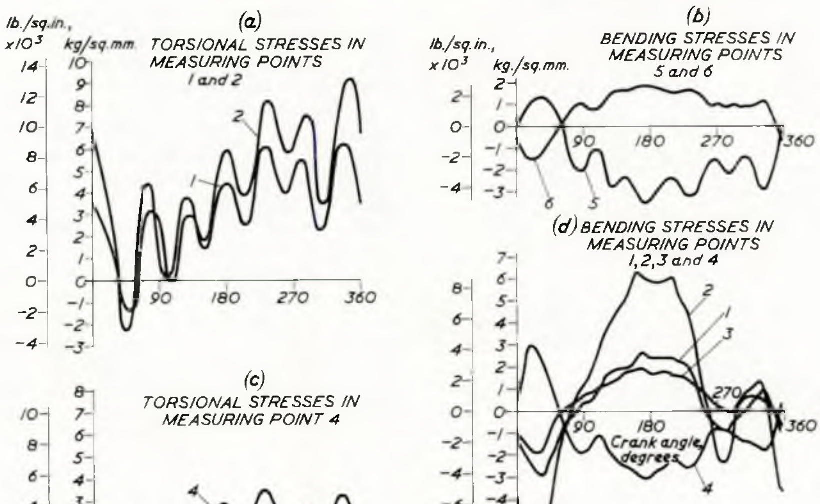


FIG. 46—*Stress measurements during shop trials on crank 7, engine—DM 760/1500 VGS 7U: 8,750 b.h.p., 112 r.p.m., m.i.p. = 8·8 kg./sq. cm.; P maximum = 55 kg./sq. cm.*

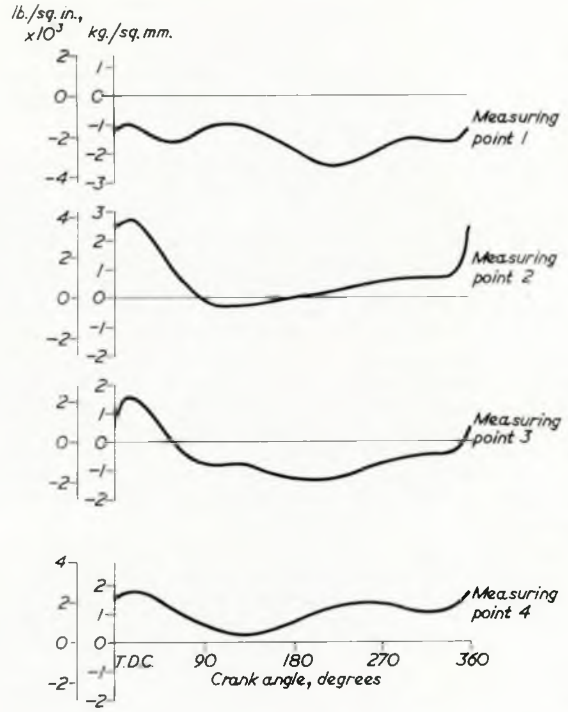


FIG. 47—*Calculate stresses for engine 850/1700 VGALO at 18,300 b.h.p. 155 r.p.m. m.i.p. = 8·5 and P maximum = 55 kg./sq. cm.*

a programme for calculation of bending stresses in a crankshaft taking into consideration all forces acting on three adjacent crankthrows and the thrust shaft. The constraints of the journals were derived by using the slope deflexion method. The computer calculated the tangential and radial forces on the crankpins, the

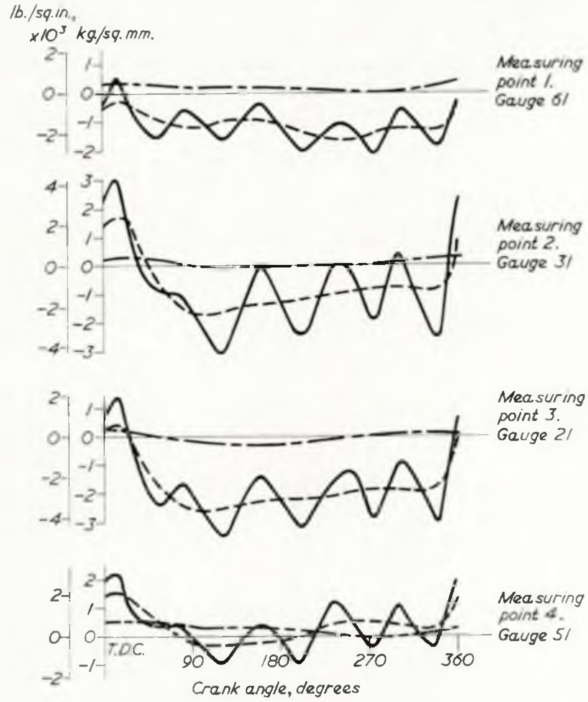


FIG. 48—*Stress measurements during sea trials 18,000 b.h.p., 155 r.p.m., m.i.p. = 85 P. maximum = 55 kg./sq. cm. with axial damper*

bearing forces on the journals, the torque in the pins and the bending moments in the pins and webs from the input data, cylinder pressure, piston weights, etc. The curves in Fig. 47 showed the calculated bending stresses on the crankshaft in the ten-cylinder engine referred to in their C.I.M.A.C. paper. The measured stress curves shown in Fig. 48 and those calculated were rather congruent.

As a complement to the statistics of crankshaft failure he could mention that there were about 620 Götaverken main engines in service and with a total of about 4,900 years of service. Up till today only ten crankshafts had been cracked or broken, which meant an incidence figure of 0·20 per hundred years of service.

MR. H. B. SIGGERS (Member) said that a quick glance at the photographs of various failures reproduced in the paper had a some-what blood-chilling effect and it was reassuring to see from Tables III, IV, V and VI that the incidence of failures of crankshafts per 100 years' service was in fact remarkably low.

In Case III on page 84 the author, with commendable restraint, referred to the design as inadequate, but he could have been excused had he used stronger language.

He believed it was correct to say, in fairness to the manufacturers, that this was the only shaft of its design they made. The design was not all that it might have been.

The design shown in Fig. 15 (Case V) on page 85 was unusual, to say the least, and he was in entire agreement with the author's view that it was unacceptable for marine service.

With regard to the statement that the fracture was due to the net and fishing gear fouling the propeller, it was quite remarkable how much machinery damage appeared to be caused by "the propeller striking a submerged object". Although to the best of his belief no one had ever identified one of these mysterious submerged objects, the reports that came in left one in no doubt that they were to be found all over the world, waiting to be struck by the unwary propeller.

On page 90 under the heading "Materials" the author rightly deprecated excessive "stepping" of the inside surfaces of webs and consequently one was a little surprised to find on the next page, under "Web Scantlings", that he advocated the use of recessed fillets of generous radius provided they did not too severely reduce web thickness locally. Mr. Siggers suggested that recessed fillets should be avoided altogether, no matter how generous the radii.

It seemed to him that the continual searching after shorter and lighter engines sometimes led to the overlooking of fundamental engineering principles and that, in spite of the keen competition that existed in the industry, it might well pay designers to concentrate more on producing a strong, rugged engine of the utmost possible reliability and which could safely withstand a fair amount of mishandling in operation.

He would have thought this was even more important today in view of the rapidly increasing trend toward more and more automatic and remote control of machinery, presumably accompanied by reduction in engine room staff. Would the author agree with this?

In the section on "Bedplates" attention was directed to the fact that cracked bedplates, especially those of welded type, had been the cause of crankshaft failures. Another point might be mentioned in this context, namely, that great care must be exercised when repairs were made and/or reinforcement fitted to welded bedplates. There was a case a little while ago where reinforcing plates were fitted each side of some of the crossgirders, and rather less than a year later the shaft fractured completely through No. 5 web by bending fatigue.

It subsequently emerged that, after the reinforcement of this bedplate had been fitted, it was found that No. 5 main bearing was subject to excessive and rapid wear down and was twice re-metalled before the shaft broke.

There was little doubt that the bearing pocket had been pulled out of line when the welding of the strengthening plates was done and this was the cause of the rapid wear down of the bearing and final failure of the shaft.

There was, of course, always the exception that proved the rule, and in a similar engine to that just mentioned it was found after about seven years' operation that several of the bearing pockets were to all intents and purposes completely detached from the surrounding structure but the shaft had suffered no harm. In point of fact, the bedplate was reinforced and the shaft was still running well three or four years later.

He suggested with some diffidence that the true value of Figs. 5 and 5a might have been enhanced if the cause of failure could have been indicated where known. Mr. Jackson had already mentioned the presence of water in the cylinders, causing slipped shrinks. There were other causes of failure which could not be attributed to crankshaft design and possibly it was a little unfair to lump them all together like that.

MR. J. H. MILTON (Member of Council) said that most marine engineers associated Mr. Archer with gearing. Crankshaft failures in oil-engined vessels and gearing failures in turbine vessels were, up to about five years ago, the most usual cause of total machinery disablement. It seemed very fitting, therefore, that Mr. Archer should, after his well-known papers on gearing, now turn to crankshafts.

He had found the paper extremely interesting, apart from some of the highly involved mathematics, as on page 32. There were some points he wished to mention and upon which he would welcome Mr. Archer's opinion.

Firstly, he felt that probably most engineers preferred the forged semi-built to the cast semi-built crankshafts, as, part from better physical properties, they felt that, even if there were any cavities or fissures in the original ingots, the amount of work done during forging would in all probability forge them up, whereas in castings the possibility of hidden inclusions or gas pockets was always present. However, he understood that the forging cost more than the equivalent casting and no doubt this factor influenced the number in use of each type. At least one engine builder installed either forged or cast units and it would be extremely interesting to know the relative crankshaft prices.

Secondly, Fig. 5 showed more failures with after-end installations than with amidship installations, in the ratio of 53:45, and of the 53 a large proportion had failed in the after part of the crankshaft. Was this lower percentage of failures in amidship installations due to the accommodating nature of the line shafting or was it that with engines aft, vibratory stresses were more difficult to cater for, and bending stresses due to tailshaft wear down were more severe? In this connexion perhaps Mr. Archer would care to comment on the effect, if any, on after crank bending stresses when a $\frac{1}{4}$-in. tailshaft wear down was present in an after-end installation, bearing in mind that sometimes the intershafting was oversize to give satisfactory torsional vibration characteristics.

Thirdly, in Fig. 15, on page 85, a crankshaft was shown with a concealed fillet which, with the adjacent enlarged pin diameter, must, he imagined, *produce* rather than *reduce* stress concentrations between pin and web. He well remembered, as a zealous surveyor, his first encounter with this form of construction—the oil oozing from the hidden fillet and his 0·005-in. feeler gauge going straight into the pin instead of the shrink. Mr. Archer pointed out that such shafts were of unusual design, but if Mr. Milton remembered rightly they came from a very well known engine builder.

Fourthly, in Case No. II, page 83, a crank was shown which originally was nitrided all over, presumably to operate in conjunction with a special bearing metal. This shaft appeared to have failed in bending fatigue and the fact that a regrinding operation to take off only 0·005-in. broke through the case hardening was stated to have produced a serious weakness. Was this correct? One would have thought that if the pins and journals were scored by contaminated oil the bearings must also have been affected, to the probable detriment of the alignment, and production of bending stresses.

The majority of crankshaft failures he had examined had been of the bending failure type in the fillet between pin or journal and web. The bending stresses causing such fractures could arise from several causes, the most usual being malalignment of bearings. In this respect, had Mr. Archer any comments to make on the possible bedplate flexure of long ten and twelve-cylinder engines under heavy weather conditions?

MR. J. R. MICHAELIS (Associate Member) thanked the author for a most interesting paper dealing with what was, after all, the heart of the reciprocating engine. The cost resulting from a broken crankshaft was large, both in terms of the actual repair bill for the engine and also the loss of earnings of the ship while it was out of service. Everyone associated with shipping would therefore welcome this paper as an aid to reducing the number of such failures in the future.

His experience had been confined to the building of crankshafts for engines of up to about 10,000 b.h.p. and his remarks would be restricted to production aspects. But he first wished to ask the author to enlarge on the statement he made in Part I, in the section dealing with materials on page 90. The author mentioned that the practice of stepping webs was to be deprecated, but, in Figs. 10 and 24, the end of the fillet between the pin and web appeared to stand proud of the web, with quite a sharp step. He appreciated that the stepping referred to by the author was of a more pronounced nature, but were these small steps also a source of danger?

In an effort to reduce manufacturing costs, many more of the engine builders were now considering investing in crankpin turning machines, to reduce the setting times required for the final machining of crankshafts. When using one of these machines it was advantageous to have the pin to web fillet also undercutting the pin by a small amount to give the tool a "run out". Was such an undercut likely to seriously weaken the shaft? What would be its maximum permissible depth? The pin, of course, should be blended into the fillet.

Another operation which offered scope for reducing costs was the shrinking of the journal pins into the webs. A very effective way of making joints, where there was a heavy interference fit, was the oil injection method. For this method the mating surfaces were tapered and the bore was slightly smaller than the pin, so that the latter would only pass between half and two-thirds of the way into it. Oil under very high pressure was then pumped in between the surfaces to expand the outer ring and allow the pin to slide in. When the pressure was released the interference was such that the parts became rigid. The beauty of this method, which was used for building small crankshafts and putting half-couplings on shafts, etc., was that the parts could be dismantled by merely applying the high pressure oil again and withdrawing the pin. Did the author see any reason why this method should not be used for building the largest size of crankshafts now being produced?

Finally, did the author have any information to show whether vertical or horizontal shrinking was the best method for making an accurate crankshaft and what order of misalignment could be tolerated between the pins themselves and the journals?

MR. G. YELLOWLEY (Member) said that Mr. Archer in Part II of this most informative paper, had referred to conventional methods of calculating the stresses in the crankshaft and it might be of interest to give an example of one method for comparison with the results from the computer method.

The examples chosen were for crankshafts of a design incorporating a cast crank unit comprising webs and crankpin as shown in Fig. 49.

A major reason for adopting the cast unit was that a rational shape could be obtained to give a more even stress distribution between the web and pin and also the minimum weight of material was utilized consistent with strength requirements.

The diagrams which followed were for a two-stroke turbocharged engine having the undernoted particulars and these might be compared with the example given on page 93 of the paper:

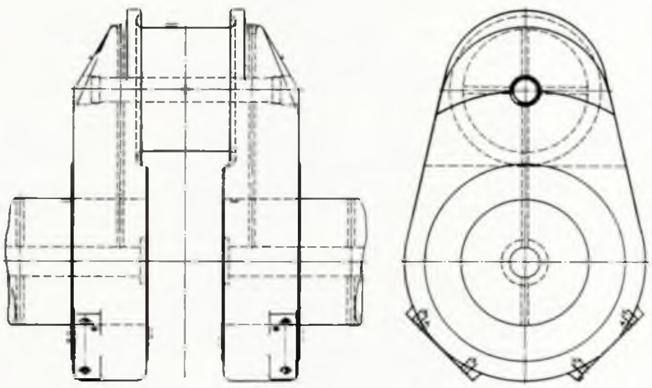


FIG. 49—*Crankshaft element*

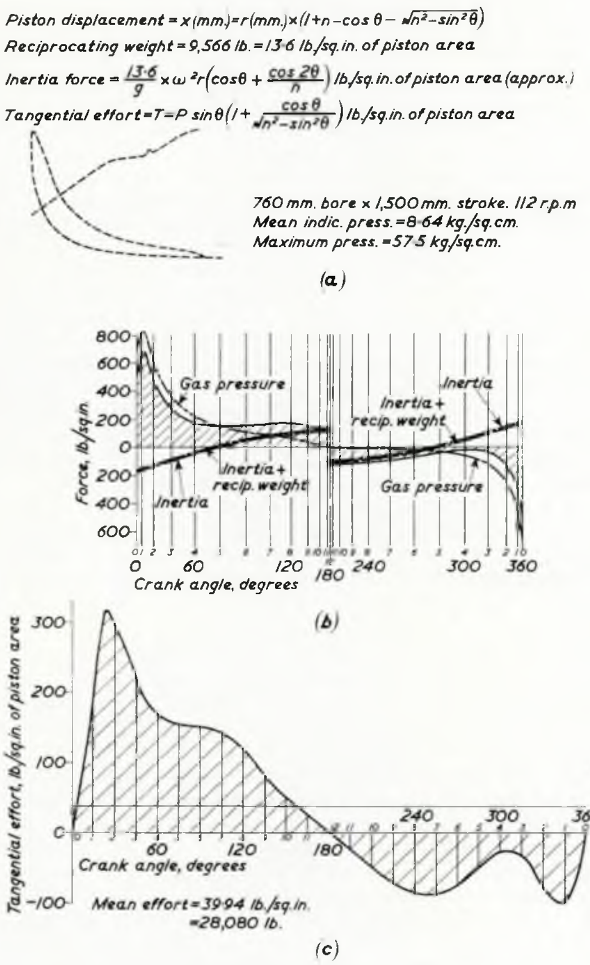


FIG. 50(a)—
(b)—*Resultant force diagram*
(c)—*Tangential effort diagram—1 cylinder*

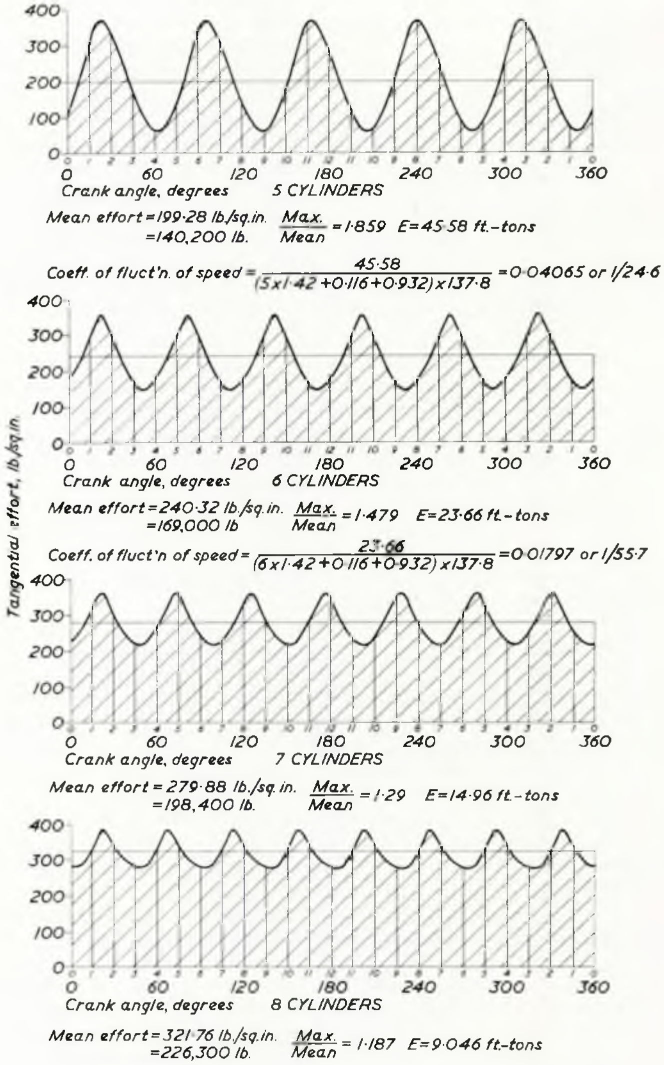


FIG. 51—*Tangential effort diagrams—5, 6, 7, and 8 cylinders*

Bore of cylinder	760 mm.
Stroke of piston	1,500 mm.
Span of bearings (centres)	1,400 mm.
B.h.p.	7,500
R.p.m.	112
M.i.p.	125lb./sq. in.
Maximum combustion pressure	797lb./sq. in.
Rule size of pins and journals	527 mm. (5 cylinders) 531 mm. (6 cylinders)
Design size of pins and journals	540 mm. (5 cylinders) 540 mm. (6 cylinders)

The cylinder indicator card for these engines and the resultant force per unit area of the piston derived from the indicator card were shown in Fig. 50(a) (b). Allowance had been made for the inertia and weight of the reciprocating masses.

A limited number of ordinates only had been taken for the desk calculation compared with 72 for the computer. (40 and 36 for 5 and 6 cylinder engine respectively).

Torsional Stresses

The tangential effort at the crankpin for one cylinder was

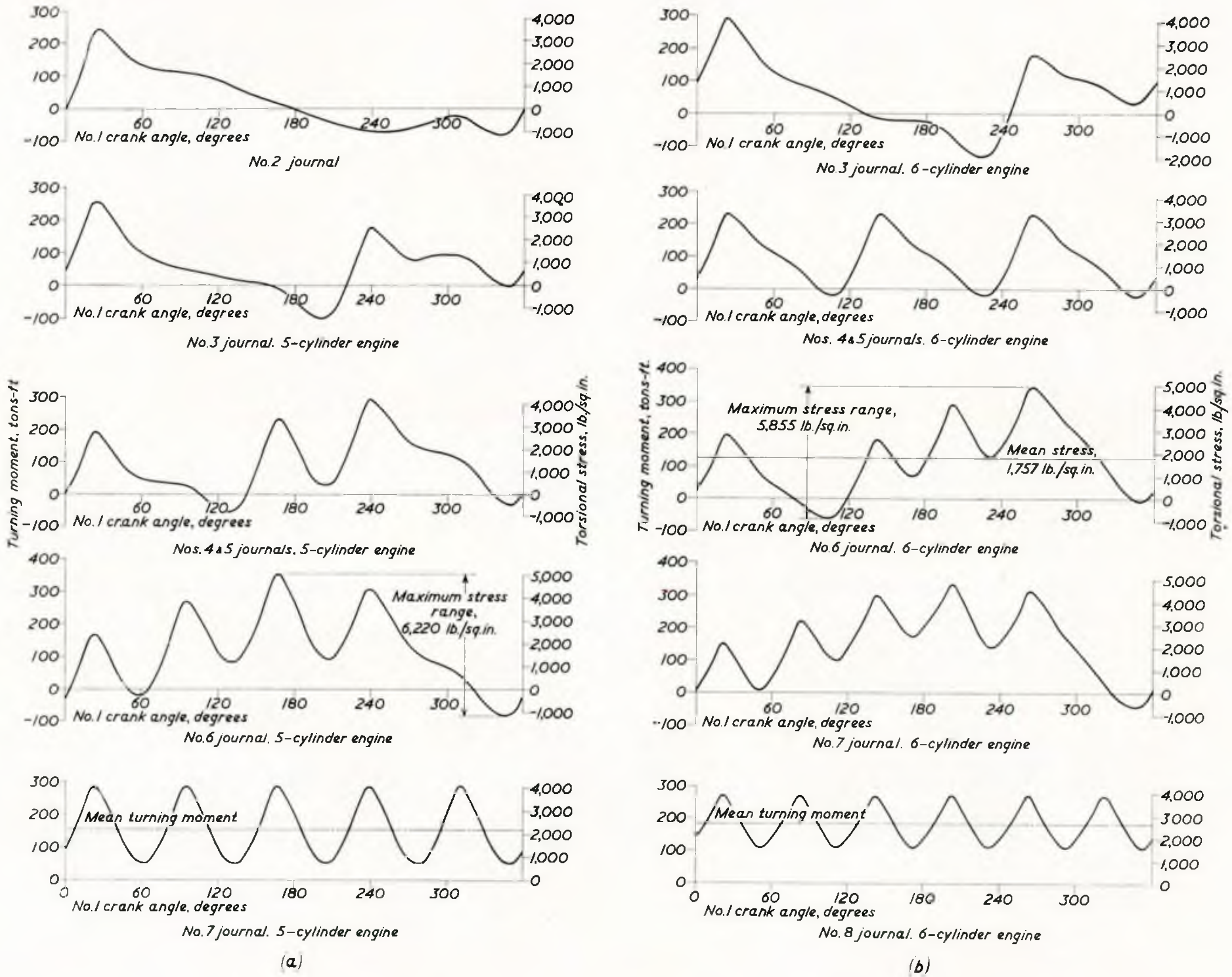
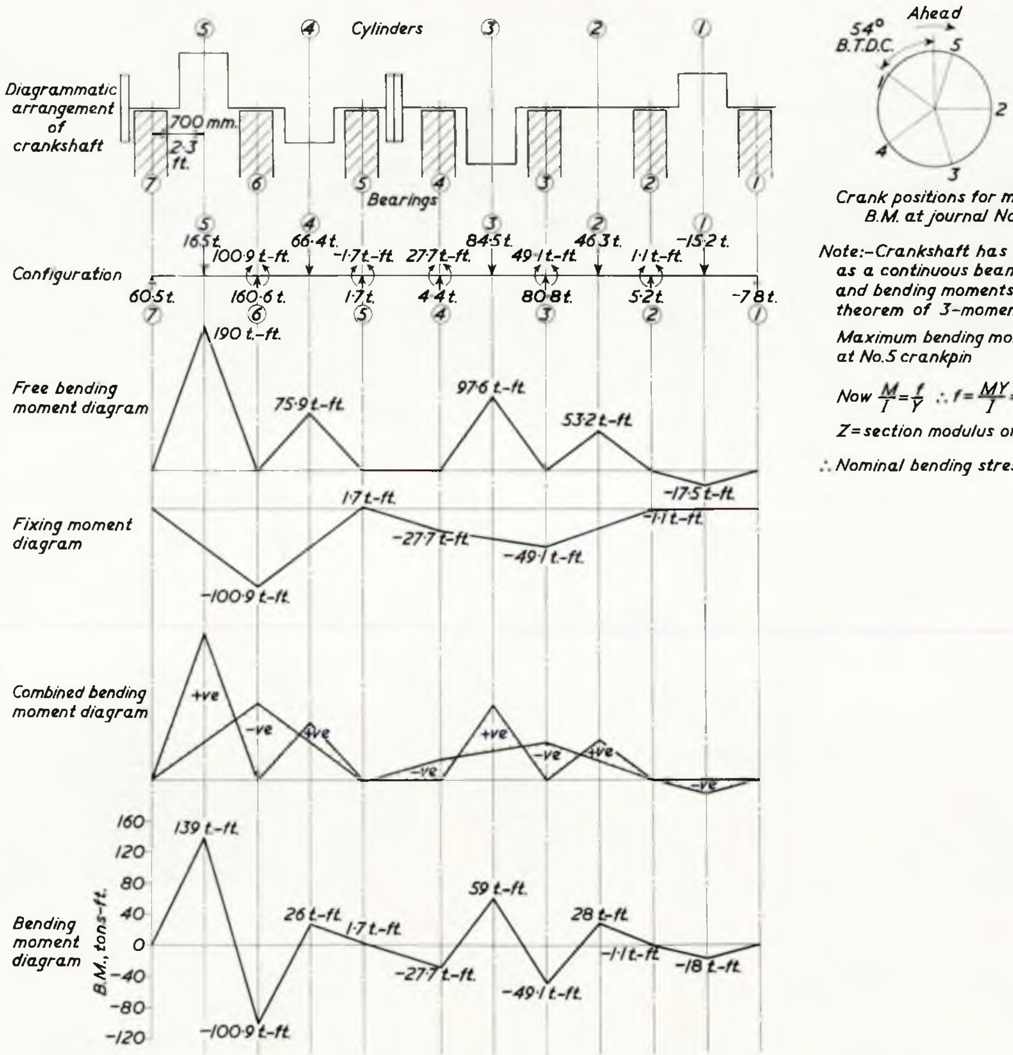


FIG. 52(a) and (b)—*Journal turning moment diagrams 760/1500 engine at full load—112 r.p.m.*



Crank positions for maximum B.M. at journal No.6

Note:—Crankshaft has been considered as a continuous beam of uniform section and bending moments evaluated using the theorem of 3-moments

Maximum bending moment = 139 t.-ft. at No.5 crankpin

Now $\frac{M}{I}=\frac{f}{Y}$ $\therefore f=\frac{MY}{I}=\frac{M}{Z}$

Z = section modulus of crankpin = 942·2 cu.in.

$\therefore$ Nominal bending stress $f=\frac{139 \times 2{,}240 \times 12}{942 \cdot 2}$

$= 3{,}966$ lb./sq.in.

FIG. 53—*Crankshaft bending moment diagrams for 760/1500 VGS—5U engine*

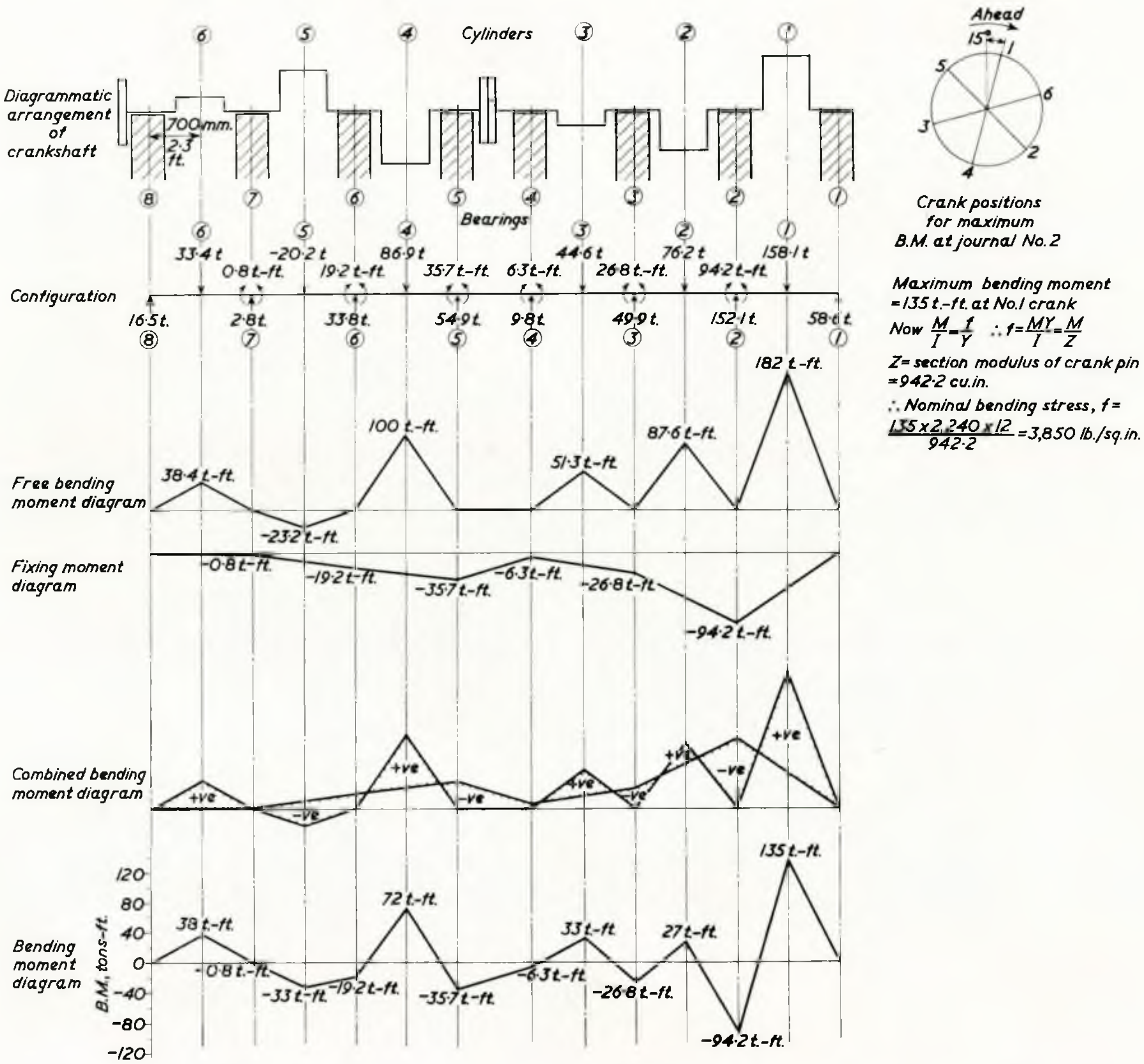


FIG. 54—*Crankshaft bending moment diagrams for 760/1500 VGS—6U engine*

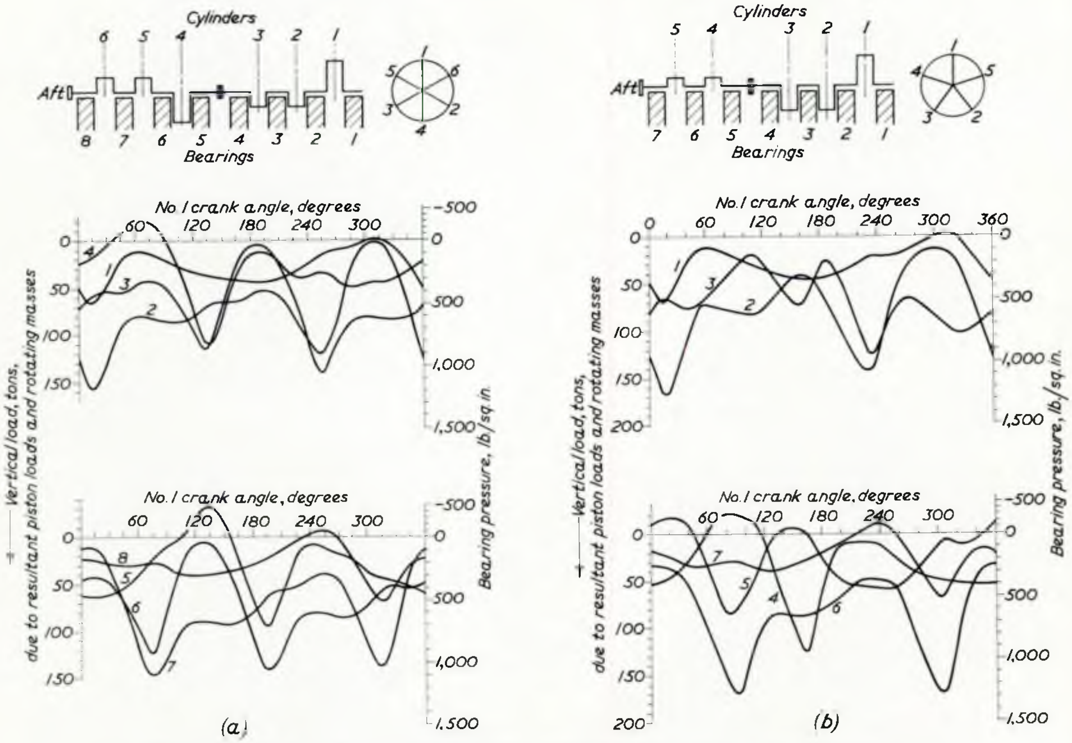


FIG. 55(a)—*Main bearing loads—6-cylinder 760/1500 engine at full load 112 r.p.m.*
(b)—*Main bearing loads—5-cylinder 760/1500 engine at full load 112 r.p.m.*

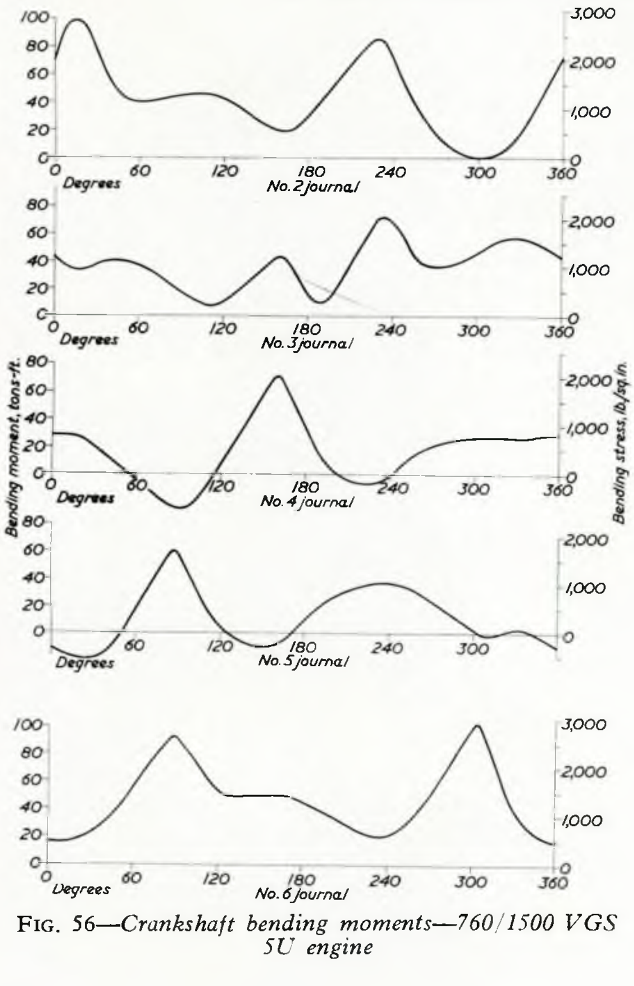


FIG. 56—*Crankshaft bending moments—760/1500 VGS 5U engine*

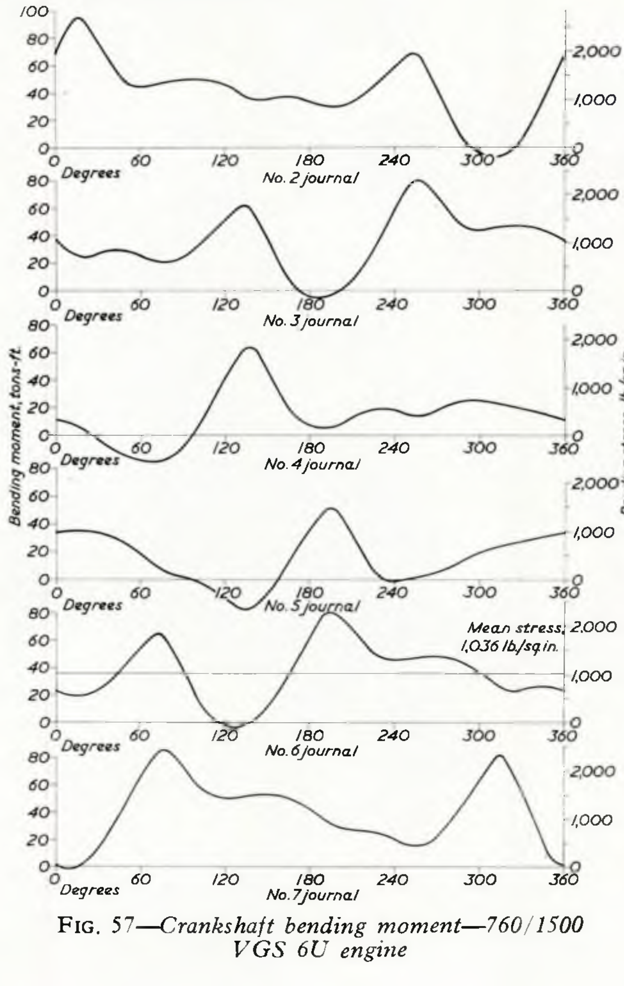


FIG. 57—*Crankshaft bending moment—760/1500 VGS 6U engine*

shown in Fig. 50(c) and at the output end of the crankshaft for engines of 5 to 8 cylinders in Fig. 51. The ratios maximum torque/mean torque were also given.

The nominal torsional stresses at each journal of the 5 and 6 cylinder engine crankshaft were given in Fig. 52. It would be noted that in the case of both engines the maximum torsional stress range due to gas pressure occurred at No. 6 journal. This engine design incorporated a chain drive casing between cylinders 3 and 4 and this was the reason for the additional main journal.

As the author had stated in his paper the torsional vibration stress due to any order of vibration might be superimposed on these curves. It was of interest to compare these curves with those of Fig. 32 of the paper.

Bending Stresses

The bending moment diagrams due to the vertical loads were derived using the assumption given by the author on page 95 except that the crankshaft had been treated as a continuous beam. Figs. 53 and 54 indicated typical bending moment diagrams at the crank angles corresponding to maximum loading.

From the data of Figs. 53 and 54 the loads acting at each crankshaft main bearing throughout the cycle were determined and these were shown plotted in Fig. 55. The vertical loads on the crankpin including the centrifugal component were similarly indicated as seen later in Fig. 59.

Figs. 56 and 57 showed the nominal bending stress at the journals abaft each cylinder for both designs. It would be noted that in general the bending stresses were only about half of those due to torque.

Direct shear stresses had been neglected as referred to by the author. The reason for this was that in the case of a circular beam subject to direct shear the shear stresses were zero at the extreme upper and lower fibres where the bending stresses were a maximum. Further, the maximum shear stresses were smaller than those due to bending and it was therefore considered the endurance of the crankshaft was unaffected.

Equivalent Stresses

The derived nominal bending and torsional stresses had been combined using a simple strength criterion to determine the equivalent tensile stresses in the crankshaft material at each journal and the results were given in Fig. 58.

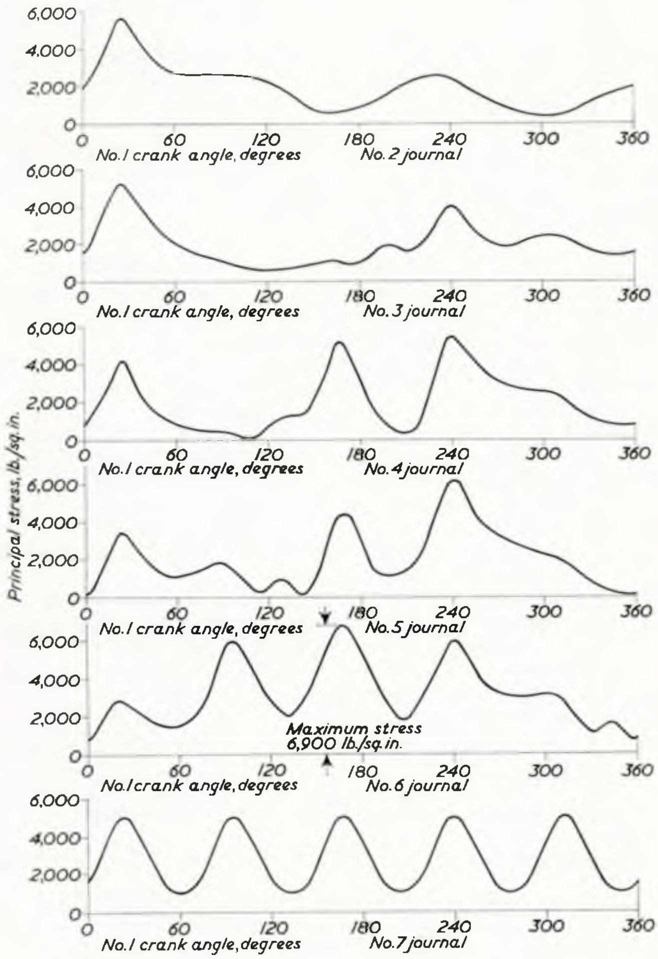
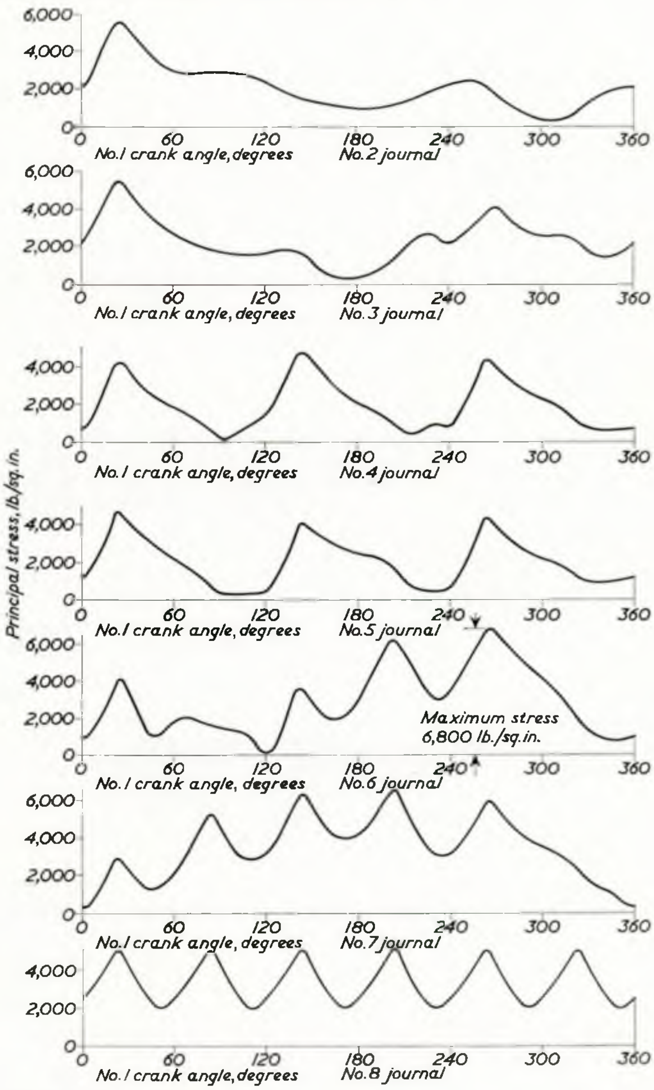


FIG. 58—*Principal tensile stresses in crankshaft journals*



760/1500 VGS 5U engine

Principal stress = combined bending and torsional stresses

From $f_p = \frac{3}{8}f_b + \frac{5}{8}(f_b^2 + 4f_s^2)^{\frac{1}{2}}$

Where f_p = principal tensile stress fb = bending stress

f_s = torsional stress

All stresses are nominal stresses based on journal diameter.

These values were nominal stresses without any reference to stress concentration factors or to additional stresses due to vibration. The maximum values of 6,900lb./sq. in. might be compared with the value of 7,300lb./sq. in. on which Lloyd's Rules were based and mentioned by the author on page 92 of the paper. He believed it was correct to say that this Rule value dated from a paper given by the then Chief Engineer Surveyor, Mr. J. Milton, in 1911.

In his paper Mr. Archer gave examples using three different criteria for combined alternating and steady torsional and bending stresses. Interpolating values derived from this desk calculation into Professor Marin's criterion (method (1) of Appendix II) and using stress concentration factors applicable to the fillet design $K_b = 3{\cdot}2$ and $K_s = 1{\cdot}4$ the factor of safety

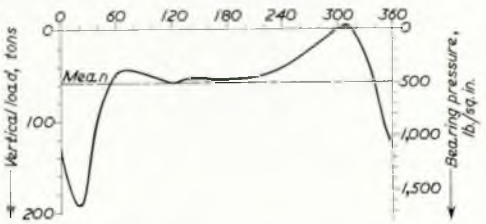


FIG. 59—*Crankpin bearing loads 760/1500 engine at full load*

without reference to torsional vibration was about 2·2. This factor might be compared with the author's minimum value in Table XV for the non-resonant condition of 1·85.

He was not clear on Mr. Archer's note regarding the shafts of the larger engines and, although there was a rider added to the effect—that it was only fair to note that comparatively speaking this type of shaft had so far recorded one of the best performances from this point of view—he could not see why Mr. Archer formed this opinion from the data in the paper. It was a pessimistic note for the large crankshafts which was a bit disturbing.

MR. S. G. CHRISTENSEN (Member of Council) said that after listening to all that had been said he felt almost punch drunk, but before reaching this state he had picked up two matters in the paper about which he wished to ask some questions. The first was relative to Case failure No. II, where Mr. Archer said that definitely there was no evidence of bearing misalignment. Mr. Milton had touched on this point already, but he himself felt, from examining an engine which had been run without lubricating oil for some short period, that there was every possible chance that there was serious misalignment with this engine. Taking the wear pattern on the crankshaft journals where an engine had had an abrasive material in the oil, or where the engine had run without oil, it would be seen that the journals had what might be referred to as circular flats on them. That flat portion of the journal occurred at a certain angle. Further, the flat portion was not the same on the adjacent journals. From the fact that 0·005in. had to be ground off the crankshaft it might be assumed that 0·004 was worn off one part of one journal and 0·004 off a part of an adjacent journal in a different angular position there could be a case of very serious misalignment. Looking further at the case here, was a twin-screw propulsion installation driven, he presumed, from two engines to each screw. Was there ever a very long period where one engine only was run. According to Lloyd's Rules, in designing a crankshaft, one of the important factors was the mean effective pressure. With two engines geared to one propeller shaft it could be shown that with only one engine in operation the mean effective pressure could rise to a value greater than that for which the crankshaft was designed. Did this set of circumstances occur in the case mentioned, and could it have been a contributory factor in the cause of failure?

Some guidance in the Rules of the Society would enlighten operatives of multiple engine installations on this point.

With regard to some of the special factors affecting crankshaft life, from what had been seen and heard it was obvious that misalignment must have been one of the reasons for a lot of the crankshaft failures that had been seen. Mr. Archer stated at the end of this section the need for regular checking of alignment in service using both web deflexion readings and bridge gauge measurements.

Why take both when one shows no more than the other. The bridge gauge and the deflexion clock gauge when used alone or together did not overcome the possibility of a journal bridging a bearing, and the dangers that come about if a bridged bearing remains undetected. He said, in his own experience, it was better to remove bearing keep, crown shell, adjustment shims and then jack the journal down on to the bearing. If the deflexion clock gauge is fitted between the crankwebs during the jacking down process the gauge will show a deflexion if a bridged bearing is present. This was the only positive check for bridged bearings and it is surprising that the author had not mentioned this most important point in the paper.

On page 86 the author said that the Society's records contained adequate examples of failures in important forgings and castings, and that a particularly serious view was taken where things were done without the surveyor's knowledge. The author said that this disturbed the mutual confidence which should exist between surveyor and builder or manufacturer.

He perhaps represented the shipowning fraternity who paid for all these failures! No doubt a lot of failures were put on the underwriters but eventually the shipowner had to pay for the failures in the form of premium. The more important aspect here was the loss of confidence between the shipowner and the Classification Society, it was the shipowner who ultimately kept the Classification Society.

DR. A. W. DAVIS (Member) said that the wealth of trouble described so ably by the author could be rather discouraging if an attempt to achieve proper perspective was inhibited by reflections on the adequacy of scantlings.

The stresses arising in a complicated machine part such as a crankshaft could usefully be divided thus:

A) Those that are understood and assessed.
B) Those not properly understood and indifferently assessed as KA where K is the factor of ignorance.
C) Those due to unsatisfactory features which can be eliminated given proper thought.
D) Those due to vibrational or other characteristics not previously recognized, innocently embraced by B, but in fact capable of analysis.

The above stresses were those arising in parts having an acceptable degree of finish and were assumed to occur under full load conditions with fair handling. They were magnified M times by undetected bad workmanship or bad material and N times by severe service operating conditions.

It could then be said that:

Calculated stress $= A + B = A(1 + K)$

Actual stress $= (A + B + C)MN$

When a failure occurs, the post mortem will usually reveal the significance of C, M and N, and the actual stress can be rewritten as $A(1 + K) + \int (C, M, N) + X$ where X is the balance quantity to give the failure stress.

If X is assessed at more than zero, it can be said in more mundane language that the failure is not adequately understood. It will then be thought (as D axiomatically is not known to exist) that K has been under-assessed, in other words, that the factor of safety is too small. After a few such incidents rule scantlings will be increased, and engines will become heavier and more costly to build—indeed we have a hint of the author's macabre thoughts in this direction.

The value of the author's paper would be very much increased if he would in fact indicate the number and proportion of failures which are not understood, that is in which X is positive.

Another point to be emphasized was that the more research work that could be put into extricating new factors from D, the pool of latent knowledge, the fewer evils would be attributed to K and the *less* would be the likelihood of scantlings being increased. Draminsky's work fell in this category and, while the author was to be congratulated on bringing this theory to bear in practice on the study of failures (and in so-doing going to prove that it was not only a theory), it was hoped that an even more urgent approach to the subject, affecting approval at the design stage, would help to forestall thoughts in the direction of increasing scantlings generally.

Finally, it was only when better practice minimized the effects of C and M and when the growing knowledge of items under D had the effect in time of reducing to insignificance the number of failures for unknown reasons, that the time would be ripe to consider whether in fact present scantlings could be *reduced,* and this provided an even greater impetus to the need for more active research.

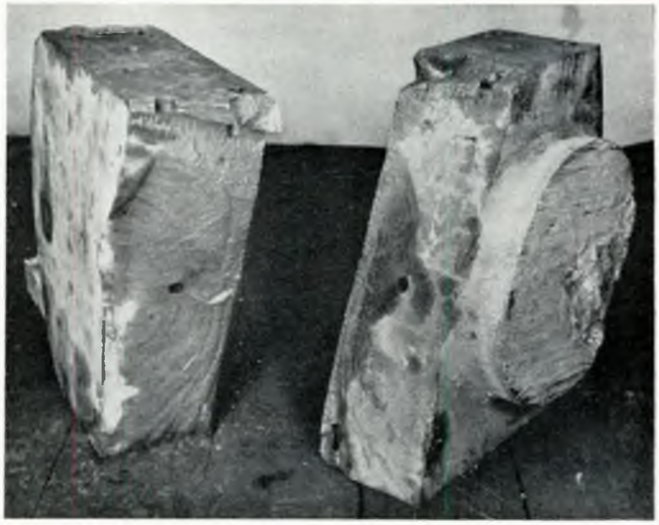


Fig. 60

Mr. N. G. Leide said that some of the earlier speakers had stressed the possibilities of calculating and measuring the stresses in crankshafts, and Mr. Archer presented a detailed calculation of a special crankshaft. With modern data machines with strain gauges and modern devices it was now possible to get a much more detailed knowledge than before of stresses in crankshafts as well as in other constructions. Based on this knowledge, people found stress patterns in most cases in an ideal material and they were designing with the use of smaller dimensions, as re-calculations showed that the safety factor would still be sufficient. It was unfortunately impossible to know what the safety factor would cover and all those who had been working with engineering material knew that these were by no means isotropic ideal materials. There were a lot of internal faults, surface faults and other defects in these materials. Mr. Archer mentioned some faults that could be found in forged crankshafts. Definitely similar types of defects were present in cast steel parts, as cast throws for semi-built crankshafts. He wondered whether the author thought that the modern ways of calculating and measuring stresses would be followed also by more modern ways of checking the materials' quality, using both dye check testing and a more detailed magnetic particle testing than was applied today to cast steel or forged crankshafts. The importance of close control could be seen in the paper, in Fig. 14, showing the sulphur print from origin of fracture (Case IV). He had found that sulphur prints which were used for gear rims, for instance, would be most valuable also for checking crankshafts in order to find if there were segregations in the crankshaft. It could be used as an acceptance test in order to make sure that the best possible material would be obtained. The use of ultrasonic testing was very valuable and it was necessary to learn how to interpret the results from these tests. Nevertheless, a few larger or smaller faults would always be found, and it would be interesting to know if Mr. Archer had any indications to show how much could be tolerated, how many mm. of cracks could be accepted in different parts of the crankshaft. One could never think of getting a perfect crankshaft. It was well known that some fractures would never propagate; they would stay as they were. Which were the conditions for that? One of the manufacturers of crankshafts of cast steel had just carried out some very simple bending tests and found a fatigue strength in bending which was less than those figures mentioned earlier in the discussion. In these tests they had taken a throw, put strain gauges in the fillet and measured the stresses during pulsations, and found a fatigue limit of something of the order of 12 to 15 kg./sq. mm. for that kind of material, which was supposed to be a good material.

There were two things he had come across which might be of some interest with regard to the cause of crankshaft failure. In one case it was a solid forged crankshaft which broke right through the web (see Fig. 60). A close examination showed that there were signs of an imperfection in the material. It had first been thought to be a weld which was done in the fillet but later, on closer investigation under the microscope, it was found that there was a high carbon content in the material at the starting

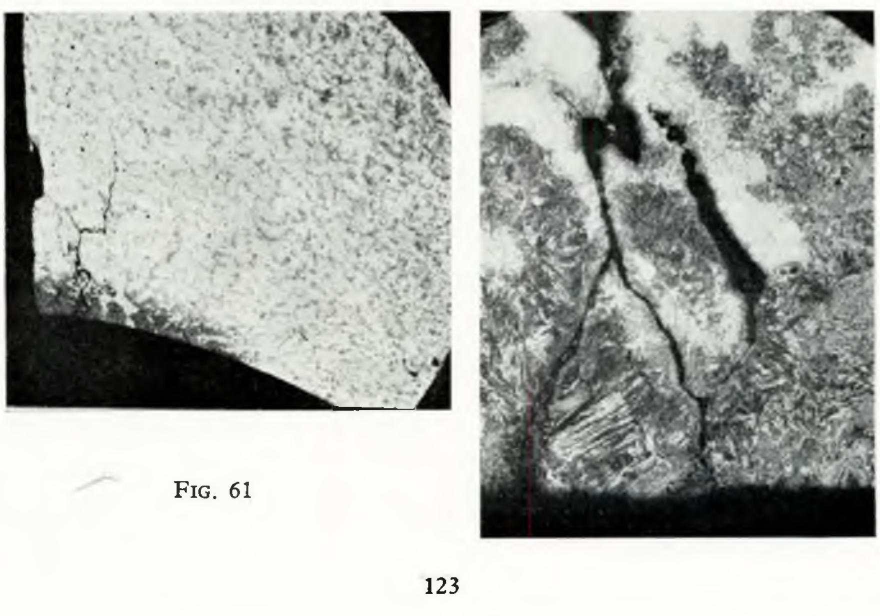


Fig. 61

point (see Fig. 61). It was possible that it was due to the bearing running hot and that the oil carbonized the material, martensite appeared in a very hard spot and from that the fatigue failure eventually started, resulting in a broken crankshaft.

The other matter concerned corrosion in some engines where there were traces of water in the lubricating oil. An explanation which he had heard, but which he did not know much about, had been put forward by a colleague. He wondered if the author could comment on that explanation. When electrostatic potential was formed in the lubricating oil due to splashing, water or humidity in the crankcase could give free hydrogen by electric action and that might enter some of the small surface fractures of the cast steel or a forged part. Hydrogen entering could result in hydrogen embrittlement, and this gave rise to internal stresses which eventually could result in fatigue failure.

Correspondence

DR. P. DRAMINSKY wrote that it was of great interest to him to see Mr. Archer's applications of his calculation method for secondary resonance. He wished to say that the calculation of the "fictive" force (or impulse) was pure and exact mathematics, and in this respect there could be no doubt about the validity of the theory. Furthermore, the calculation was very easy to carry out, even at an early design stage.

It was a little more troublesome to find the exact phase angle between the "fictive" impulse and the direct impulse of the same order, but in practice it would always be appropriate, and on the safe side, to calculate with the arithmetic sum of the two impulses.

In Case 1, the ten-cylinder engine, the "fictive" and direct impulses of the 8th order were (as he knew it) almost exactly in phase together, and therefore the measured resonance stress was very close to the calculated. In Case 2, the 12-cylinder engine, the measured resonance stress of the 9th order was some 20 per cent below the calculated, and this might be due to a certain phase difference between the fictive and the direct impulse of this order. Still the actual stress was sufficient to cause fracture after some ten years of service.

He thought, therefore, that classification societies would be fully justified in declaring, that no case of secondary resonance at all could be tolerated at normal service speed. When he said this it is not from a personal desire that the Draminsky formula should be known by all designers of crankshafts, but he hoped that it would be of assistance to them and give them a feeling of safety.

Besides, we must be prepared in the future for more complicated plants, e.g. direct-coupled, geared plants with high speed engines. In his opinion, such plants could be designed with absolute safety, even without the use of slip couplings, when the designer knew all possibilities of secondary resonance and sub-harmonics. There could be no other risk of unexpected and "mysterious" vibrations.

MR. C. J. HIND (Member) wrote that it was with considerable diffidence that he offered comment upon this paper, bearing in mind the wide experience of the author and the exceptional facilities available to him through the Society with which he was connected.

Over the last 40 years Mr. Hind's company had produced engines ranging in horsepower from 300 to 3,000 and the incidence of crankshaft failure had been extremely small. The sum total they had on record was ten, of which seven had given clear evidence that they had failed through torsional fatigue. Of these seven only one had failed as a result of the damper becoming inoperative. During the last 20 years many hundreds of engines had been supplied with vibration dampers designed to control continually the effects of a major critical speed and yet only one had resulted in crankshaft failure. This one was due to faulty manufacture which occurred in the early days of the production of this particular type of damper and, with this background, he would suggest that the statement made on page 83 should be treated with some reserve. The other six crankshafts which failed due to torsional fatigue, were as a result of auxiliary flywheels, supplied for attachment to the forward end of the crankshaft not being fitted, by mistake, and cracks appeared some six years later. All the ten shafts referred to were of the solid forged type and since transferring their entire manufacture to the CGF process the company had had no failures.

Although crankshaft failures in large powered, slow speed engines, as recorded in Mr. Archer's paper, had not been excessive, nevertheless the percentage of failure had been considerably higher than in the case of medium speed engines. It was suggested that the adoption of the CGF process for the manufacture of crankshafts eliminated many reasons for failure, such as unreliability of steel castings and the weaknesses induced with shrink fits. Medium speed engines of up to 10,000 h.p. could now be made available using crankshafts which eliminated these weaknesses and it was fair to suppose that a much lower incidence of crankshaft failure could result.

Mr. Hind wished also to comment on Lloyd's Rules for crankshaft scantlings. When these rules were first introduced, and in all subsequent modifications, no definition was placed on the manner in which maximum cylinder pressures should be measured and yet maximum cylinder pressures were a vital factor in establishing crankshaft scantlings according to the rules. Improved instrumentation, developed by engine manufacturers, had now shown that maximum pressures were occurring in the cylinder much in excess of the originally measured figures and yet crankshafts, subjected to these conditions more severe than anticipated, had behaved entirely satisfactorily in service. As a result of the improved instrumentation, engine manufacturers now submitted revised figures of maximum cylinder pressures to the Society with the result that they penalized themselves, relative to the required scantlings. This factor must be taken into account, particularly when crankshafts of the CGF type were used, and he would like to have Mr. Archer's comments on this factor.

He also considered that urgent action was required to provide rules for crankshafts using surface hardening treatment as a standard part of the manufacturing technique, as this process would undoubtedly be adopted for larger crankshafts in the very near future.

He also made a plea that Lloyds should extend their rules to cover guidance and dimensions for Vee-type engines of which there was an ever increasing number on the market.

He had read, with interest, Mr. Archer's comments on secondary resonance and was tempted to ask the question as to whether his company had been fortunate in not running into trouble due to this complex phenomenon or whether such phenomenon was more of theoretical interest than practical importance. His company had for many years relied upon simple methods in the assessment of torsional vibration and had never considered secondary resonance, yet they had been remarkably free from crankshaft troubles. The examples quoted on page 99 implied that the problem was real and yet there must be many cases where a low amplitude *nth* order critical speed and the flank of an *(n-2)th* order critical speed appeared to co-exist quite safely. He would appreciate Mr. Archer's comments on this point.

It was recorded that a number of crankshaft failures had occurred due to excessive fatigue stress range and he was tempted to suggest that the adoption of a variable pitch propeller, which would permit the operation of the engine over a minimum speed range, would have the effect of keeping the fatigue stress range to a minimum and hence reduce the possibility of crankshaft failure due to this cause. He would be interested to know whether, in Mr. Archer's experience, such a solution had been contemplated.

Finally, he congratulated Mr. Archer upon the presentation of a most excellent paper and thanked him for the service he had done to the industry in presenting this in such a readable form.

DR. ING. F. SCHMIDT, in a written contribution, commented that the author had compiled a remarkable quantity of facts and experiences which required a careful study of the content page by page.

As mentioned on page 91, for the fully-built shaft the bridge between the eye holes of the web should not be less than $0 \cdot 27 \times$ shrink diameter.

In his opinion, for a semi-built shaft, the distance between the eyehole and the crankpin or its fillet should also have a minimum limit. Otherwise the stress concentration in the fillet combined with the shrink stresses would be dangerous.

Could the author give some idea or quote some experience of this limit, preferably as a ratio to crank diameter. In other words, what was the smallest stroke permissible for a semi-built crank of a certain shaft diameter.

MR. G. P. SMEDLEY, B.Eng., B.Met., wrote that Mr. Archer had provided a most useful paper on the reliability of crankshafts. He had also illustrated some methods of calculating the steady and cyclic stresses in a crankshaft and of estimating the factor of safety.

In a study of fatigue strength for combined stresses, Marin* pointed out that evaluation depended on the accuracy of the failure relation between mean and variable stresses. The proposed relationships could be expressed by the general formula:

$$\left[\frac{f_r}{\sigma_e}\right] \quad \left[\frac{kf_m}{\sigma_u}\right]^m = 1$$

The nomenclature of the stress terms was the same as on pages 103 and 104 of Mr. Archer's paper; *n, m* and *k* were constants having the following values:

Relationship	*n*	*m*	*k*
Söderberg	1	1	1/Yield ratio
Goodman	1	1	1
Gerber	1	2	1
Ellipse	2	2	1

For the general case, the Söderberg relationship on which Methods 1 and 2 (Appendix II) were based was the most conservative. Experimental data generally lay between the Goodman and the Gerber relationships. The indications were that for ductile steel *n* could be about $(2 - f_m/\sigma_u)$, *k* and *m* being unity. The work of Gough, Pollard and Clenshaw† showed that where f_m/σ_u was less than $0 \cdot 2$, the influence on fatigue strength was reasonably small. The factor of safety for the case considered in the paper should therefore lie between $1 \cdot 81$ and $2 \cdot 24$. Even the minimum of this range might appear to be small but account had been taken of stress concentration factors expected at changes of section. The resulting values were more realistic than those given by the common practice in which the stress for failure was divided by the nominal stress on uniform section.

Some increases in vibration stress must be expected during service. These were essentially of bending type and arose from tolerable misalignment, wear down and flexing of the bedplate. The lower the initial cyclic stresses, the greater the latitude for any increases during service. Mr. Archer had shown clearly the importance of limiting torsional vibration stresses, in this respect.

With reference to the calculation of the working stresses it must not be forgotten that irrespective of the technique, the answers were only as accurate as assumptions and approximations permitted. In looking to the future the aim must be to eliminate the greatest of the uncertainties. Some of these were referred to in the discussion on the paper. Several important factors were as follows:

1) The damping characteristics which limit the vibratory stresses in a crankshaft are not known with a sufficient degree of accuracy. It has been established that the internal damping capacities of a steel are very small. Friction at bearings and rubbing surfaces therefore controls the stress levels. The dearth of information is confined to this feature.
2) Cyclic bending stresses associated with acceptable misalignment and wear down, also require more exacting methods of assessment.
3) Accurate evaluation of stress concentration factors for notches such as fillet radii is essential. In this connexion web dimensions and pin and journal dimensions must be considered. Reasonable factors are only available for simple shapes.
4) As Mr. Archer has pointed out, the full stress concentration factor of a notch should always be used in design calculations, i.e. the steel should be assumed to be fully notch sensitive in fatigue. Coyle and Watson‡ have shown the importance of this in a recent investigation of bending fatigue failures at notches in certain steam turbine rotors.

Size effect must also be eliminated from data on the unnotched fatigue strengths of different steels. There was an urgent need for design data sheets giving reliable fatigue strengths of different steels and applicable to large section sizes.

On page 90, the author referred to the work of Frost§ at the National Engineering Laboratory. There appeared to be a slight misunderstanding of the results of his work. He and his colleagues undertook an investigation to determine:

1) the level of cyclic stress which is required to propagate a fatigue crack of a particular length or size;
2) the rate of growth of fatigue cracks.

They found that the level of cyclic stress to cause a crack to grow in a laboratory atmosphere could be expressed by the formula:

$$\sigma = \sqrt[3]{\frac{\text{Constant}}{l}}$$

where σ is the level of the cyclic stress
l is the length of the crack

This expression was not applicable to mechanical notches such as grooves, fillet radii and oil holes. Moreover the conditions were modified by a corrosive atmosphere.

IR. A. HOOTSEN, in a written contribution, congratulated the author on his most informative and valuable paper and wished to know whether he could give more detailed information about the Society's requirements for magnetic crack detection of cast steel crankwebs (Chap. P 515), as mentioned by him on page 87 of the paper at the end of Case VII. The relevant chapter gave only prescriptions about surface preparation prior to magnetic testing but no specifications as to which defects, and where situated in web or throw, would be allowed or might be removed, when detected. In this connexion, he pointed out that the areas around the fillets between crankpin and crankweb, specially at the inside facing the shaft centre, were the most vulnerable parts of the shaft and thus called for more severe requirements for acceptance than for example the outside of

* Marin, J. 1956. "Interpretation of Fatigue Strength for Combined Stresses", I.Mech.E.-A.S.M.E. Proc. International Conference on Fatigue of Metals, Session 2, p. 184.

† Gough, H. J., Pollard, H. V. and Clenshaw, W. J. "Some Experiments on the Resistance of Metals to Fatigue under Combined Stresses" Ministry of Supply, Aeronautical Research Council Reports and Memoranda, H.M.S.O. 1951.

‡ Coyle, M. B. and Watson, S. J., "Fatigue Strength of Turbine Shafts with Shrunk-on-Discs", I.Mech.E., preprint P6/64, October 1963.

§ Frost, N. E., Holden, J. and Phillips, C. E., "Experimental Studies into the Behaviour of Fatigue Cracks" Conference of the Hungarian Academy, Budapest, October 1961.

the crankwebs. As the author would agree that no casting could be absolutely free from even small surface defects or inclusions; a specification of the Society's requirements for acceptance would therefore be most helpful.

MR. W. F. DOWIE (Associate Member) wrote that he was surprised to hear it stated during the discussion of the paper that the effect of mean stress was not great, as in a paper presented to the Institute by B. Taylor* there was a figure which showed that for mild steel, the loss in limiting fatigue stress range was about half the increase in mean stress, i.e. an increase of mean stress from 0 to 20 tons/sq. in. caused a decrease in limiting stress range from ± 16 to ± 11 (10 tons/sq. in. loss in stress range). Thus the stress range (A) followed the relation

$$A = A_0\left(1 - \frac{\frac{1}{2}S}{u.t.s.}\right)$$ within the limits of mean stress (S)

from 0 to half the u.t.s.

It would be seen that this was a modified Goodman relation.

As 80 per cent of the examples of service failures given in the paper were of fatigue at the fillet radius of crankpin or journal and as nitriding had been mentioned, it would be interesting to hear the author's opinion on fillet rolling.

Fatigue tests in reversed bending on small crankshafts (3-in. diameter journals and 2½-in. diameter crankpins) had been carried out by Motor Industry Research Association†‡ and the results gave a 60 per cent improvement due to fillet rolling for both forged steel and cast iron specimens. M.I.R.A.§‖ had also carried out an extensive programme of fatigue tests in rotating bending on stepped bar specimens, of steel, cast steel and cast iron, which had been rolled in the fillets and improvements ranging from 25 to 240 per cent were obtained.

It seemed reasonable to suggest that the residual compressive stress due to fillet rolling, reduced the mean stress at this critical region thus increasing the permissible stress range before crack initiation.

Devices could easily be designed to roll the crankpin and journal fillets with the crankshaft in position on the ship.

DR. F. ØRBECK wrote that the author should be congratulated on this very constructive paper. Although all parts of it had been read with great interest, this contribution would be confined to Part II—The Problem of Calculating Crankshaft Sizes and Factors of Safety.

The development leading to the present method of design using a rule formula and limits for additional stresses due to torsional vibrations was of great interest and the value of this design approach over a long period was appreciated. As pointed out in reference 44 of the paper, however, present means might now turn further refinements into practical propositions. The author and Lloyd's Register of Shipping should be congratulated on taking a lead in this respect and the calculations presented would be of great value.

Similar computer programmes to those described in the paper had now been in operation by his company for some time. These calculations gave results which were valuable for assessing the relative strength in torsion of different installations and as experience increased limits could be established for the total nominal stresses in torsion. Would the author consider that such limits could be of value as a stage further than the limits purely for stresses due to torsional vibrations or was it better to go the whole way and work out resultant stresses due to bending and torsion, accepting the greater complications with this second alternative?

On page 97 under "Effect of Torsional Vibration" the author wrote as follows: "The author is convinced that in the great majority of cases, torsional effects will far outweigh bending so far as combined fatigue strength is concerned". The writer was inclined to disagree and considered that bending stresses were more important than was suggested in the paper. His company had concentrated much of their effort towards the study of bending stresses. Stress concentration factors in bending had been measured with strain gauges on a number of shafts and calculations had been developed to obtain the nominal bending stresses due to working of the engine with a straight shaft and due to misalignment.

The stress concentration factors were measured on the shafts in their shops after being finished machined. The aft side web of cylinder No. 1 was subjected to bending by letting this cylinder section overhang. Strain gauges were cemented in the side pin to side web and side web to journal fillets and readings were taken for the side crank on top and on bottom. The bending moment on the web could be calculated but the method gave a considerable change in bending moment over the thickness of the web, i.e. in the axial direction. To which bending moment should the nominal stresses be referred? As the same problem arose for the shaft under normal working conditions, he would be pleased to hear the author's opinion on this point. The following stress concentration factors referred to nominal stresses based on the bending moments at the centre of the webs, acting on the full web section, and might be of interest: for the original 75LB6 engine 4·2, for the 725SB6 engine 3·2, for the 67PT engines 2·05.

The difference between the stress concentration for the first two engines is mainly attributed to the first engine having deeply recessed fillets and the second external fillets.

The difference between the 725SB6 and the 67PT engines was caused mainly by the greater overlap between the side pin and the journal for the 67PT engines.

The importance of external fillets and overlap was clearly demonstrated.

His company's calculation of the bending stresses in the crankshafts was based on the following assumptions which were lettered as the assumptions given on page 95 of the paper:

a) The same assumption as in the paper.

b) and c) The shaft is considered as a continuous beam ranging from the forward end to the propeller and simply supported at each bearing.

d) As in the paper. This assumption will introduce no inaccuracy to the bending moments at the crankwebs.

e) A separate calculation has been developed to give the stresses due to misalignment calculated from web deflexion readings.

f) At the present only bending stresses are considered but the direct and shear stresses can be incorporated where necessary.

The calculation was carried out for a number of crankshaft positions equally spaced over one revolution. The forces on the crankpins due to gas pressure in the cylinder and deadweight and inertia of reciprocating parts were considered and the bending moment variations along the shaft in the vertical and horizontal planes due to these effects were found. A separate calculation then gave the bending moments due to the deadweight of the rotating parts. Finally, the effect of misalignment was incorporated and the bending and twisting moments on the crankwebs were evaluated. The calculations were mainly carried out on digital computers and the programmes were available for Ferranti "Pegasus" and Elliott 803.

The bending moment distribution due to misalignment was calculated from the web deflexions in the following way. When misalignment alone was considered the bending moment distribution was linear between any two bearings. The sum of the bending moments at the main webs of a cylinder section was related to the deflexion of those webs by a deflexion

* Taylor, B. 1952. "The Strength of Large Bolts Subjected to Cyclic Loading." Trans.I.Mar.E., Vol. 64, p. 233.

† Love, R. J. "The Influence of Shot Peening and Cold Rolling on the Bending Fatigue Strength of Cast Crankshafts." M.I.R.A. Report No. 1952/3.

‡Love, R. J. and Waistall, D. N. "The Improvement in the Bending Fatigue Strength of Production Crankshafts by Cold Rolling." M.I.R.A. Report No. 1954/2.

§ Wright, D. H. and Love, R. J. "Improving the Fatigue Strength of Steel Components by Fillet Rolling." M.I.R.A. Report No. 1959/2.

‖ Wright, D. H., Love, R. J. and Nixon J. "Improvement of Fatigue Strength by Fillet Rolling—Five Cast Irons and a Cast Steel." M.I.R.A. Report No. 1960/6.

coefficient which was determined by the shaft dimensions alone. This deflexion coefficient had been obtained by calculation and measurements for a number of shafts. The bending moments at the supports were obtained from the bending moments at the centre webs by geometry. The bending moment at the first main bearing was equal to zero and it therefore followed from the foregoing that the bending moment at the second main bearing could be obtained from the web deflexion of cylinder No. 1. It was assumed that this bending moment was independent of the angular position of the crankshaft and the bending moment at main bearing No. 3 could therefore be obtained by making use of the web deflexion on cylinder No. 2. Continuing this process the complete bending moment distribution along the crankshaft was found and hence the nominal stresses could be obtained for any position.

Axial vibrations were treated in much the same way as torsional vibrations. The engine was represented by a vibrational system as shown in the author's contribution to reference 25 and a computer programme had been developed for the forced damped vibrations. Values for thrust block stiffness, damping constants and propeller excitation had been obtained through repeated measurements and calculations and nominal stresses of the webs in bending were obtained from measurements and by calculation.

This work was still in a state of development and a complete set of calculations and measurements had not yet been carried out for any installation. For the 75LB6 and 67PT6 engines there was, however, sufficient information available to present a picture of the relative importance of the various factors involved. Some of this information is given in Table XXIV. For the 75LB6 engine the worst case had been assumed in which the installation ran on a 4th order I-node axial vibration resonance peak and a 7th order III-node torsional vibration resonance peak.

The stress concentration factor referred to the deeply recessed fillet design of side webs. For the reconditioned crankshafts external fillets were used and these gave a considerably lower stress concentration factor in bending. No measurements were available for the stress concentration factor in torsion and the values quoted by the author were used in Table XXIV. The equivalent alternating direct stress had been calculated with allowance for a misalignment stress which was not unusually high. In conjunction with torsional and axial vibrations serious misalignments could therefore, cause failures of the original crankshafts of the 75LB6 engines.

The results for the 67PT6 engines were obtained in the same way as for the 75LB6 engines. They referred to a typical installation but the alignment stresses should be considered as limits rather than actual values. The stresses given in Table XXIV suggested that bending played by far the most important part in the load on the side webs of a Doxford engine and he would welcome the author's comments on this point as well as on the improvement due to external or only partially recessed fillets. A further stress reduction in the 75LB6 shafts could be obtained by using a combined torsional and axial vibration damper.

In Table XXIV misalignment stress had been added to the other stresses considered by the author. Had this been done in the author's example the factor of safety arrived at would have been still lower.

The factors of safety given in the paper and in Table XXIV were the result of much more exact knowledge than had been available in the past and therefore must not be compared with previously used high factors containing a very big margin of ignorance.

MR. R. MACIOTTA, was of the opinion that Mr. Archer's paper, especially in its second part, had perhaps given one of the most complete pictures of the various causes which could give rise to fatigue stresses in a crankshaft.

Some of such causes were fairly evident or anyhow known and long since investigated: load from connecting rod, torque fluctuation, torsional vibration and misalignment of main bearings.

Only during recent years, by way of the technical literature, had the designers' attention converged on axial vibrations as a possible cause of high stresses, although the phenomenon was already long known; even more recently had the influence of the non-linearity in the inertial system formed by crankshaft and crank gear been pointed out, i.e. the possibility of quite

TABLE XXIV—TOTAL CRANKSHAFT STRESSES IN SIDE WEB TO SIDE PIN FILLETS

Engine Type		75LB6		67PT6	
Stress lb./sq. in.		maximum	minimum	maximum	minimum
Bending	Nominal stress due to gas pressure, deadweight and inertia of reciprocating parts	+1,710	−1,990	+2,904	− 611
	Nominal stress due to axial vibration	+ 900	− 900	+ 205	− 205
	Nominal stress due to misalignment (expected limit in service)	+2,000	−2,000	+2,000	−2,000
	Total nominal stress	+4,610	−4,890	+5,109	−2,816
	Total nominal stress (approximate)	−140 ±4,750		+1,146 ±3,963	
	Stress concentration factor in fillet from Doxford measurements	4·2		2·05	
	Total fillet stress	−590 ±19,900		+2,350 ±8,120	
Torsion	Total nominal stress including vibration stress	+1,290 ±3,200		+1,680 ±2,070	
	Stress concentration as from the author's paper	1·6		1·6	
	Total fillet stress	+2,060 ±5,120		+2,680 ±3,320	
Equivalent fatigue stress by Marin's method		±22,100		±13,500	
Fatigue limit. Steel 28-32 tons/sq. in. u.t.s. Proportioned from author's fatigue limit Appendix II		±23,600		±23,600	
Final factor of safety after taking into account all consideration and stress concentration factors		1·07		1·75	

high stresses occurring, due to the so called "secondary resonances" of the "sub-harmonics" (see reference 53 in the paper).

Finally, a study* issued during recent months, proved mathematically the interdependency of axial and torsional vibrations; it seemed that, under particular conditions, this could give rise to vibrations having a magnitude higher than that expected with the normal methods.

Thus the problem of crankshaft fatigue stresses appeared more and more complex, as newer and newer aspects were evinced.

He wished to make a few remarks concerning the newest aspects of the problem.

Mr. Archer stated that there was very little information on the magnitude of additional stresses due to axial vibrations: in this connexion he recalled that, at the recent C.I.M.A.C. 1962 Congress, Mr. Guglielmotti and himself presented a paper dealing with the results of axial vibration measurements for 14 different propulsion sets consisting of two-stroke Diesel engines with 6 to 12 cylinders, arranged at both stern and amidships; the crankshaft stress values were deduced from the amplitude recordings, taking into account the elastic line of the vibrations.

Such recordings had indicated maximum nominal stresses in the crankpin of some 2 kg./sq. mm. which, based on an experimentally deduced concentration factor equal to 4, corresponded to stresses in the crankweb fillet of some 8 kg./sq. mm.

It was easily noticeable that the stresses of such magnitude, jointly with the normal stresses, would practically nullify the safety margin of the crankshaft.

Therefore, axial vibrations required, in their opinion, very careful attention.

Taking into consideration the vital importance of the crankshaft, it was his company's practice to limit the stresses due to axial vibrations within a quarter of the aforesaid values, applying, when necessary, a damper.

The author's conclusions, which pointed out the moderate value of the safety factor of crankshafts, even if they were proportioned according to Lloyd's Register specifications, appeared to them to be good confirmation of their standpoint.

As far as torsional "secondary resonances" were concerned, to date, in no case had they found discrepancies between calculated and measured values, which could be explained by the above theory; however, it was to be noticed that, both in the example given in the paper (Appendix III), and in the example given in the original Draminsky paper (reference 53), the crankshafts were provided with flywheels having a somewhat low moment of inertia (lower than that of a single crank).

It was easily noticed that a larger flywheel reduced the coefficient β thus reducing the excitation of the "secondary resonance"; in fact a flywheel having higher inertia made the system approach a linear system.

The use of large-dimensioned flywheels was therefore advantageous also from the above point of view; probably due to the employment of rather large flywheels, in none of the cases examined to date had they experienced the disturbing effect of "secondary resonances".

In conclusion, it was considered that such continuous effort to obtain better knowledge of crankshaft stresses, would contribute to the improvement, more and more, of the reliaability of crankshafts, in spite of undoubtedly greater difficulties deriving from the continuous increases of performance.

It would be most interesting if the author could give some information about the relative frequency of crankshaft failures in the individual years.

It was not known whether the amount of available data was sufficient to prove a trend; anyway their particular experience in the matter permitted fairly good prevision.

Mr. A. Kleiner wrote that the excellent paper by Mr. Archer was a further very valuable and helpful contribution to the designers of marine Diesel engines. He deserved not only congratulations for it but also thanks. The paper was particularly valuable because it contained a lot of practical experience which would never come to the knowledge of each individual designer. The statistics of damaged crankshafts resulted indeed in a very fortunate statement. If, from all the 4,464 crankshafts with a total of 27,068 service years, one eliminated the triple crankshafts, which could not be considered as a very happy design, 3,598 crankshafts of conservative design with a total of 20,594 service years would remain. The probability of such a crankshaft failing was lower than 2½ per 1,000 service years The only exceptions were the solid forged crankshafts of four-stroke engines exceeding 2,000 h.p. in which the rate of failures was 15·4 cases in 1,000 service years.

Considering that the primary reason for most of the failures was bad maintenance of the main bearings or corroded shrink fits or even faulty material which was covered by bad welding, one could draw the conclusion that nowadays crankshafts were properly designed. The very cautious rules from Lloyds Register of Shipping for the dimensioning of crankshafts together with the very well known "Guidance Notes on Torsional Stresses and Critical Speeds . . . " were certainly contributing considerably to the excellent safety factor. This of course also applied to other classification societies having very similar rules. Absolute safety which would also prevent failures caused by bad servicing or faulty material (which, today, were less and less frequent) was hardly possible. Anyhow, such security could never be obtained by making the current rules more severe.

The reference to a more exact calculating method for stresses in the crank throws compared with the method presently applied was also very helpful. However, such an exact calculation required a computer, but made it possible to work out a safety factor which should specify the quality of the design. However, conclusions based on such safety factors should only be drawn from a good number of years of practical experience. It should never be forgotten that all methods of stress calculation were based on some simplifying assumptions of the actual problem and that one was never quite sure which "theory of strength" corresponded best to such a complicated machinery part as a crankshaft. If such a new method of calculation gave a safety factor of 1·4 or 1·5 for a crankshaft design which proved to be very satisfactory in the past, there was no reason just because of this calculated safety value to ask for a more comfortable safety factor. It would therefore be very interesting to learn the calculated safety factors of all typical large modern crankshaft designs mentioned on pages 76 and 77 of the paper.

Mr. A. R. Hinson (Associate Member) wrote that his observations of crankshafts, fractured through corrosion fatigue, had led him to believe that severe crankshaft corrosion was not necessarily the long, drawn-out process with which one sometimes associated corrosion. It could occur rapidly, in days rather than months, and with very little warning other than pitting of centrifuge discs (or three-wings) and the steel laminations of filters of the autoclean type.

Routine maintenance of the cooling and sealing arrangements, lubricating oil changes or water washing while centrifuging removed corrosive elements and the deterioration ceased. It was then very difficult to determine the cause of corrosion and the period during which it occurred.

Areas on the crankshaft where stresses concentrated, e.g. fillet radii, became anodic and the unevenness of the corroded surface increased the stress concentration. These areas should be buffed to a mirror finish; a hand power-tool driving a flexible emery disc gave good results.

It seemed that corrosion of bearing journals usually took place when the engine was stopped. The oil in the bearing grooves stagnated and deteriorated although the rest of the oil in the system might remain in good condition. Crevice type corrosion occurred in the clearance between the edge of the oil grooves and the journal; the oil was held there by a kind of capillary action. This resulted in the corrosion patches

* Van Dort, D., and Visser, N. J. "Crankshaft coupled free torsional-axial vibrations of ship's propulsion system"—Studiecentrum T.N.O. voor scheepshouw en navigatie—Report No. 39M —September 1963.

FIG. 62

running roughly fore and aft along the journal and having one side bounded by a more or less straight line (see Fig. 62). Journals which had been attacked in this way caused rapid bearing wear down with consequent crankshaft malalignment. The crankshaft in Fig. 62 fractured.

Mr. Hinson believed that severe corrosion of this kind *could* probably be alleviated by slightly inclining the oil grooves in the bearings instead of machining them parallel to the journal axis. This would permit the warm oil to drain more completely from the bearing on shut-down.

There was much to commend the practice of turning a shut-down engine once daily with the lubricating oil pump on. When this was not possible a ten or fifteen degree rotation of the crankshaft should be sufficient to disturb the stagnant oil and prevent crevice type attack.

Mr. Hinson thanked the author for a very interesting paper.

MR. J. F. BUTLER, M.A. (Member) thanked the author for an extremely instructive paper, both for its analysis of crankshaft failures and for its explanation of the basis of Lloyds Register of Shipping Rules on crankshafts, it would form a most valuable reference for many years to come.

A point raised on page 91 of the paper was most thought-provoking. The author suggested that in a fully-built shaft the width of the bridge piece between the two eyeholes could safely be as little as 0·27 of the eyehole diameter. The records for fully-built shafts, for both breakage and slipped shrinks, confirmed this figure; yet this bridge piece had to carry a total tensile load due to the shrink fits of twice the amount carried by other parts of the webs around pin and journal.

With a shaft of rule diameter (d) and web thickness $0{\cdot}625d$, the rule radial eye thickness was $d\sqrt{\frac{0{\cdot}12}{0{\cdot}625}} = 0{\cdot}438d$. With shrinkage allowance of one part in 550 the radial shrink fit pressure calculated for a thick ring was 19,500lb./sq. in. and the maximum hoop stress on the same basis 35,100lb./sq. in. With a friction coefficient of 0·15, the torque to produce slip corresponded to a shear stress in the shaft of 14,600lb./sq. in.

Assuming that in the bridge piece, $0{\cdot}27d$ wide, of an actual shaft, the hoop stress was constant across the section, the stress to maintain the same grip as in a full ring would be $19{,}500 \times 2/0{\cdot}27 = 145{,}000$. Clearly, therefore, yield must take place at this point and, taking the yield strength of 28-ton steel as 36,000lb./sq. in., the transverse grip pressure would be reduced to approximately $19{,}500 \times \frac{36{,}000}{145{,}000} = 4{,}850$lb./sq. in. and the overall shrink grip reduced in the ratio $\frac{19{,}500 + 4{,}850}{2 \times 19{,}500} = 62$ per cent.

Many slipped shrinks were the result of exceptional occurrences such as water in cylinders and, in such cases, slipping was bound to occur unless the grip was sufficient to break the shaft or some other part. The records suggested that the occurrence of slipped shrinks might not increase with moderate reduction of grip. Possibly no effect would be felt until the grip torque nearly reached the normal running torque with allowance for torsional vibration.

Was there not, therefore, a strong case for reducing the eye radial thickness in webs with ample bridge piece width and in semi-built webs? If the overall breadth transverse to the crankthrow must be kept large to maintain web bending strength, could not the outside circle of the web be made eccentric to the pin? In this way the minimum radial thickness in line with the crankthrow could be reduced to a figure giving the same total grip as in a fully-built web with minimum bridge piece.

The writer would be most grateful for the author's comments on this matter since the suggested reduction in web scantlings could result in considerable weight reduction and corresponding improvement in torsional vibration characteristics of built shafts.

MR. B. K. BATTEN, M.Sc. (Associate Member) wrote that Mr. Archer in highlighting some outstanding service failures had demonstrated that the calculated safety margin of the present-day crankshaft would tolerate little that was substandard on the part of designer or manufacturer.

Considering some of the special factors affecting crankshaft life, Mr. Archer had rightly drawn attention to the machining and finish of oil holes drilled transversely through journals and pins. Experiments on transverse holes in 3-in. diameter shafts under torsional vibration* had clearly demonstrated the beneficial effect of a lip radius, the interesting point being that in each test the initial cracks were formed at the junction of the lip radius and the parallel portion of the hole. Calculation showed that, for a constant diameter of hole, the stress for crack initiation at point J (Fig. 63) remained sensibly

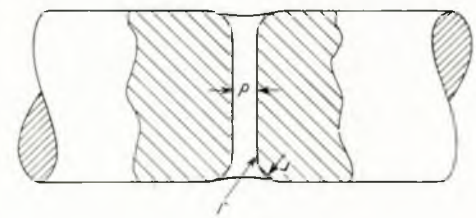


FIG. 63

constant, irrespective of lip radius. Thus a larger radius r (at least up to $r = d$) allowed a greater shaft surface stress before cracking occurred, point J now being nearer to the neutral axis. On this basis the smoother the transition at point J, and the further down the hole this smoothness could be carried, the greater the chance of avoiding fatigue failure from this source. Incidentally, Case IX provided an interesting conclusion regarding the comparative vulnerability of fillets and oil holes. Having regard to the quoted proportional dimensions of oil hole diameter and fillet radius in terms of crank journal diameter, and referring to the work quoted above—also that by Dorey and Smedley† it would seem that for oil holes there was only a small size effect upon stress concentration factors

* Smedley, G. P. and Batten, B. K. 1961. "Fatigue Strength of Marine Shafting", N.E.C.Inst. Eng. and Shipb.

† Dorey, S. F. and Smedley, G. P. 1956. "The Influence of Fillet Radius on the Fatigue Strengths of Large Steel Shafts", I.Mech.E. Int. Conf. Fatigue of Metals.

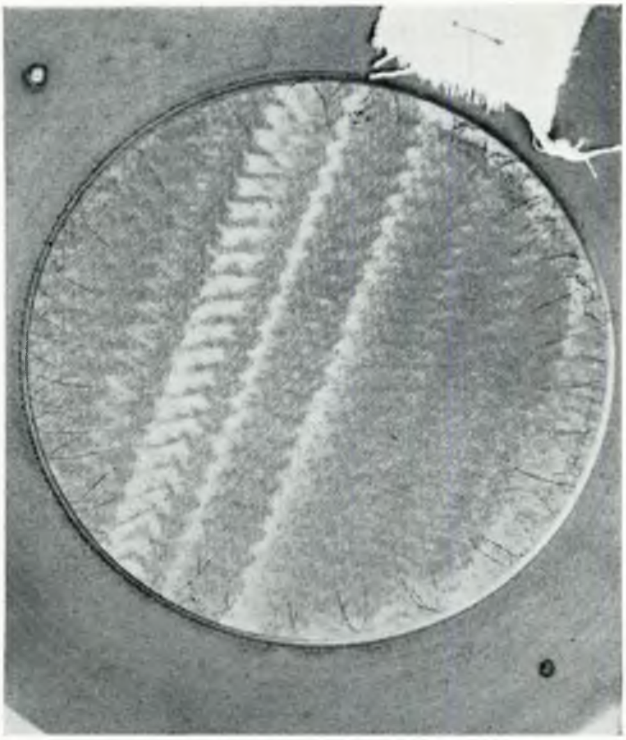


FIG. 64

with increasing diameter of shaft. This size effect would appear to be opposite to that for fillets, in that the larger the shaft diameter the less was the strength-reducing effect of the comparative oil hole.

The argument for nitriding smaller diameter shafts would, in spite of the evidence of Case II, appear to be a good one, for figures quoted by Lessells* showed an increase of as much as 50 per cent in fatigue strength with nitriding. However, since the depth of the nitrided layer was more or less constant at 0·025in., and as with this type of process there was a fairly sharp change of residual stress pattern at this depth, it would seem that any notch in the material, unless it was itself nitrided, was a potential danger. This would imply that nitriding should not be used on shafts with oil holes, and in any case should be carried well up onto the radius of the fillet. The author's comments on this would be appreciated.

The conclusion in Part II that, for combined bending and torsional fatigue, a notch sensitivity of unity might be inferred, was interesting. Frost, in his recent work on non-propagating cracks, showed the relationship between the critical propagation stress and crack length was given by $\sigma^3 l = C$ where σ is the semi-range of alternating stress based on gross area, l is the crack length, and C (for mild and alloy steels) is 5·5 tons-inch units. If $\sigma^3 l$ is greater than C a crack will grow, and if $\sigma^3 l$ is less than C a crack will remain dormant. It was widely suggested that the mechanism of crack propagation was distinct from that of crack initiation. Using the figure of $\pm 3\frac{1}{2}$ tons/sq. in. for critical fatigue stress we have the condition that if the initial crack length exceeded about $\frac{1}{8}$in. then the crack would continue to propagate. This had to some extent been demonstrated in the work on mild steel shafts with fillets†. Fig. 64 showed a section from an unbroken end of one of these 3-in. diameter shafts with a sharp fillet. Here a distinct ring of non-propagating cracks could be seen while the shaft had broken through a single propagation crack at its other end.

The major significance of the existence of distinct critical conditions both for crack initiation and propagation lay in the realization that the real damage to a crankshaft might be done in an extremely short time, such as through the failure of a damper (Case I). Provided the initial crack was of sufficient length then the propagation could proceed under normal running under much lower stresses.

* Lessels, J. M. 1954. "Strength and Resistance of Metals", Wiley.
† See footnote on p. 129.

MR. N. J. VISSER wrote that, starting from the first attempt to "turn something round", with the help of a primitive crank-mechanism, the author had progressed to a study of the most modern possibilities of mastering the difficult problems, which brought about the complex phenomenon of the highly loaded crankshaft of today.

His excellent review of the difficulties with marine crankshafts, put into service since 1947, was disturbing. It was a warning to be careful and to do the utmost to avoid trouble. It was an appeal to engineers to increase their knowledge, making use of the modern possibilities of investigation and calculation.

A very interesting point mentioned in the paper was the improvement to the calculation of the dynamical behaviour of crankshafts with the help of the electronic computer.

The Engineering Research Department at Werkspoor N.V. had had experience with electronic computing for about five years now; during the past year it had acquired, for its own use, an EL X1 electronic computer from Electrologica N.V., provided with an Algol (algorithmic language) compiler. A number of successful calculations had been made, including those for torsional and axial vibration problems.

In the experience of his company the possibilities afforded by electronic computing were many and far reaching. Nevertheless, besides the problems connected with the analysis and programming of the calculations, there was one rule which had to be observed, and which all calculations had in common, viz. "the result of any calculation depends not least on the exactness of the input data".

In the author's opinion a more exact calculation of the total torsional stresses at service speed (operational and vibratory) was now possible, with the help of electronic computers, and his results seemed to be hopeful.

Nevertheless the question remained of how to derive the right input data, viz., the right pressure ordinates at the exact crank angles. The common indicator diagram was not exact at all and was valuable only for a more qualitative estimation of the Diesel process. The question was—how could we be sure that the pressure ordinates read off from any indicator diagram finally gave the right harmonic forces and phase angles, especially the higher ones?

From all kinds of indicator diagrams, directly registered or photographed from an oscilloscope screen, it was only possible to derive the gas pressure ordinates by reading them off. As this had to be done by individual persons, it was necessary therefore not only to consider the errors, caused by the imperfection of the measuring instruments, but also the human factor.

In order to make some contribution to the solution of this problem, his company had analysed a number of Farnboro indicator diagrams (see Tables XXV, XXVI, XXVII and XXVIII).

A) 2-S.C.S.A. trunk-piston engine; type TEH 452, turbocharged two-cylinder test engine, N_{max} = 990 b.h.p. at n = 250 r.p.m.

Table XXV: Comparison between different cylinders at various m.e.p. values.

Comparison of the tangential harmonic components of six gas pressure diagrams, viz. for two cylinders and m.e.p. of 8·02, 7·56 and 5·48 kg./sq. cm.

Table XXVI: Influence of diagram strewing and reading-off errors.

a) Comparison of the tangential harmonic components of two gas pressure diagrams of cylinder No. 2 with an m.e.p. of 5·5 kg./sq. cm. for maximum and minimum ordinates due to strewing in the diagram.

b) Comparison of the tangential harmonic components of four gas pressure diagrams of cylinders No. 1 and No. 2 with an m.e.p. of 8·02 kg./sq. cm. for reading-off by two different persons.

Table XXVII: Influence of T.D.C.-indication errors.

a) Comparison of the tangential components of a gas pressure diagram of cylinder No. 1 with an m.e.p. of 8·02 kg./sq. cm. for a T.D.C. indication error of +1 deg. and −1 deg.

b) The same for the diagrams of cylinder No. 2.

B) 2-S.C.S.A., six-cylinder main engine, type W.S. 546, turbo-charged, N_{norm} = 3,650 b.h.p. at n = 160 r.p.m.

Table XXVIII: Influence of the pre-injection angle.

Comparison of the tangential harmonic components of the gas pressure diagrams of cylinder No. 6 at an m.e.p. of 6·50 kg./sq. cm. and at different pre-injection angles.

Farnboro gas pressure diagrams on a crank angle base were used, as this type of diagram had a large scale and was more precise than the common indicator diagrams on a piston base.

The conclusions from the comparisons were:

i) The influence of different cylinders on the tangential harmonic components is only significant above the 12th harmonic; the influence of different cylinders on the phase angle degrees appears negligible, but multiplied by the harmonic number it is clear that the phase angle differences for the higher harmonics can be about 90 deg.

ii) The influence of relatively small diagram errors caused by strewing of the diagram in the range near the maximum combustion pressure on the harmonic components is very important, however the influence on the phase angle is negligible. The influence of reading-off errors cannot be neglected.

iii) The influence of T.D.C.-indication errors on the harmonic components and on the phase angles is moderate.

iv) The influence of pre-injection angle adjustments on the harmonic components and phase angles appears to be not very important.

Mr. Visser also commented upon two other problems mentioned by the author in his paper.

1) Mr. Visser's company had always been very careful with the use of dampers in the service speed range (see page 83 of the paper), knowing that damper installations might be affected by time in service.

As the author had mentioned, in the case of damper installations, with bearings and gearings, the wear and tear of the bearings and teeth might change the characteristics of the damper. The degeneration of the damping medium could be fatal to the life of the crankshaft.

With the introduction of the viscous damper, filled with silicon-oil as a damping element, a new period of hope began. His company had never had any trouble with the viscous dampers in either low or moderate speed engines. Unfortunately after a good beginning, with high speed railroad Diesel engines, after some time in service, breakdowns of crankshafts occurred.

An investigation brought to light the fact that failures of the dampers, caused by a too high heat loading and thereby degeneration of the silicon-oil, were the source of the trouble.

Therefore their conclusion was that if extra torsional vibratory stresses, exceeding the limit for continuous running in the service speed range, could not be avoided, the use of a viscous damper was admissable when the heat loading of the damper did not exceed a certain limit.

Experience made it seem unlikely that heat-loading below 400 B.t.u./sq. ft./hr. would cause difficulties. Care should be taken that the damper did not have to work in a region of high temperature, or under the influence of heated oil.

2) On page 96 the author gave suitable concentration factors for bending and torsion stress of 3·0 and 1·6 respectively.

From stress measurements on crankshafts, Mr. Visser's company had found concentration factors of 4·0 and 1·9.

The factor of 3·0 for bending stress concentration especially, seemed to them to be somewhat too small if the results of their investigation and those of others were compared with this value.

The fact that the concentration factor for bending was

TABLE XXV.—COMPARISON BETWEEN DIFFERENT CYLINDERS AT VARIOUS M.E.P.-VALUES.

2—S.C.S.A. type TEH 452; N=495 b.h.p./cylinder; n=250 r.p.m.

n of comp.	M.e.p.=8·02 kg./sq. cm. T_{n_1}	ΔTn 1) per cent.	ψ_{n_1} deg.	Δψn 2) deg.	M.e.p.=7·56 kg./sq. cm. T_{n_1}	ΔTn per cent.	ψ_{n_1} deg.	Δψn deg.	M.e.p.=5·48 kg./sq. cm. T_{n_1}	ΔTn per cent.	ψ_{n_1} deg.	Δψn deg.
1	7·360	−1·9	68	−0·5	6·953	0	68	−2	5·345	+0·5	68	0
2	6·938	−0·7	46	+1	6·572	+1·8	46	+0·5	5·223	+1·3	46	−0·5
3	4·141	+2·2	35	0	3·933	+5·1	35	−0·3	3·243	+1·9	34	−0·3
4	2·528	+3·0	27	0	2·461	+2·9	27	0	2·041	+2·9	26	+0·3
5	1·758	+4·6	23	0	1·729	+6·1	23	0·6	1·449	+1·4	22	0
6	1·105	+2·0	22	−0·5	1·049	+7·5	22	+0·1	0·910	−2·0	21	−0·7
7	0·564	+2·1	20	+0·3	0·545	+3·1	20	+1·1	0·497	+5·0	20	+0·1
8	0·325	+13	18	−1·8	0·298	+18	17	+1·9	0·303	+11	16	+0·4
9	0·235	−1·5	18	−1·8	0·306	+3·6	17	+0·7	0·265	+1·9	17	−0·8
10	0·215	+7·4	20	+1	0·250	−8·4	19	+1·7	0·198	−11	19	−2
11	0·084	−2·4	23	−0·5	0·103	−5·8	21	−1·1	0·092	−39	20	−3·4
12	0·080	−7·5	17	−2·2	0·092	+28	15	−0·9	0 088	−2·3	15	−0·7
13	0·133	−6·0	18	+0·6	0·134	−10	16	+1·8	0·099	+18	15	+0·8
14	0·058	+31	19	−0·8	0·110	−4·5	19	+1·0	0·052	+48	15	+1·6
15	0·010	+33	7	+7·3	0·061	−10	18	−2·5	0·030	+40	13	+1·9
16	0·049	−4·0	15	+1·8	0·040	+48	17	−1·9	0·040	−5	13	+1·3
17	0·044	−20	17	+1·4	0·033	+79	17	+0·3	0·051	−7·8	14	+2·6
18	0·022	−14	0	+5·3	0·036	+56	13	+4·1	0·037	+16	14	+1·4

1) T_{n_1}=tangential harmonic components due to gas pressures of cylinder No. 1 in kg./sq. cm.
T_{n_2}=tangential harmonic components due to gas pressures of cylinder No. 2 in kg./sq. cm.

$$\Delta T_n = \frac{T_{n_1} - T_{n_2}}{T_{n_1}} \times 100 \text{ per cent.}$$

2) ψ_{n_1}=phase-angle of harmonic components due to gas pressures of cylinder No. 1 in kg./sq. cm.
ψ_{n_2}=phase-angle of harmonic components due to gas pressures of cylinder No. 2 in kg./sq.cm.
$\Delta\psi_n = \psi_{n_1} - \psi_{n_2}$ in crank angle degrees.

Table XXVI.—Influence of diagram strewing and reading-off errors.

2—S.C.S.A type TEH 452; N=495 b.h.p./cylinder; n=250 r.p.m.

n of comp.	M.e.p.=5·5 kg./sq. cm.; cylinder No. 2				M.e.p.=8·02 kg./sq. cm.; cylinder No. 1				M.e.p.=8·02 kg./sq. cm.; cylinder No. 2			
	T_n 1)		ψ_n 2)		T_n 3)		ψ_n 4)		T_n 3)		ψ_n 4)	
	T_u	ΔT_b per cent.	ψ_u deg.	$\Delta\psi_b$ deg.	T_1	ΔT_p per cent.	ψ_1 deg.	$\Delta\psi_p$ deg.	T_1	ΔT_p per cent.	ψ_1 deg.	$\Delta\psi_p$ deg.
1	6·353	−7	69	+1	7·345	+0·3	68	0	7·162	+0·1	67	0
2	6·198	−0·6	45·5	0	6·963	−0·8	46	+0·5	6·874	+0·8	47·5	0
3	4·083	−1·1	33·3	+0·3	4·154	0	35	+0·3	4·267	+0·5	35·3	+0·3
4	2·638	−2·2	25·5	+0·3	2·522	+2·1	27	0	2·594	0	35·6	−0·5
5	1·781	−3·3	21·8	+0·4	1·773	+1·7	23	0	1·828	+2·6	26·8	+0·4
6	1·107	−5·2	20·3	+0·2	1·113	+0·1	22	0	1·165	+2·5	22·6	+0·8
7	0·647	−8·7	18·9	+0·3	0·567	−2·6	21	+0·1	0·598	−13·0	21·2	−0·4
8	0·422	−13	16·4	+0·1	0·341	−5·3	18	+0·1	0·358	+15	17·0	−1·3
9	0·315	−16	16·1	+0·1	0·327	−8·3	18	−0·1	0·326	+15	16·6	+0·7
10	0·195	−21	17·6	+0·7	0·208	−3·8	20·1	+0·5	0·225	+7·1	20·4	+1·1
11	0·123	−30	16·5	+0·6	0·095	−17	21·5	+0·1	0·097	−6·1	22·2	+4·7
12	0·125	−26	15·0	−0·1	0·089	+7·8	16·4	−1·1	0·057	+75	14·8	−0·9
13	0·103	−20	17·3	+0·8	0·105	+18	16·8	+0·8	0·092	+91	17·7	−0·9
14	0·070	−20	17·6	+1·0	0·062	+18	18·6	+1·1	0·093	+20	19·4	+1·0
15	0·061	−25	15·4	+0·3	0·032	+16	17·1	−1·5	0·040	−57	17·9	−17
16	0·064	−16	14·5	0	0·051	+11	15·8	−2·1	0·047	+68	14·8	−2·9
17	0·061	−9·8	14·2	0	0·040	+7·5	16·4	−5·3	0·056	+25	14·7	−0·4
18	0·052	−5·7	13·8	0	0·018	+105	19·9	−2·4	0·019	+147	13·9	+5·8
	← strewing effect →						← 2 persons →					

1) T_u = tangential harmonic components due to upper boundary of gas pressures in kg./sq. cm.
T_1 = the same for the lower boundary.

$$\Delta T_b = \frac{T_u - T_1}{T_u} \times 100 \text{ per cent.}$$

2) ψ_u = phase-angle of tangential harmonic components due to upper boundary.
ψ_1 = the same for the lower boundary.
$\Delta\psi_b = \psi_u - \psi_1$ in crank angle degrees.

3) T_1 = tangential harmonic components of gas pressures read off by person 1.
T_2 = the same read off by person 2.

$$\Delta T_p = \frac{T_1 - T_2}{T_1} \times 100 \text{ per cent.}$$

4) ψ_1 = phase-angle of tangential harmonic components of gas pressures read off by person 1.
ψ_2 = the same read off by person 2.
$\Delta\psi_p = \psi_1 - \psi_2$.

TABLE XXVII—INFLUENCE OF T.D.C.—INDICATION ERRORS.

2—S.C.S.A. type TEH 452; N=495 b.h.p./cylinder; n=250 r.p.m. M.e.p.=8·02 kg/sq. cm.	Cylinder No. 1				Cylinder No. 2			
n of comp.	ΔT_n^+ per cent.	$\Delta \psi_n^+$ deg.	ΔT_n^- per cent.	$\Delta \psi_n^-$ deg.	ΔT_n^+ per cent.	$\Delta \psi_n^+$ deg.	ΔT_n^- per cent.	$\Delta \psi_n^-$ deg.
1	−2·0	+2	+2·1	−2	−2·1	+3	+2·2	−2·0
2	−0·9	1	+0·9	−1	−0·7	+1	+0·8	−1·0
3	−0·8	0·7	+1·0	−0·7	−0·8	+0·7	+0·9	−1·0
4	−1·4	0·8	+1·4	−0·2	−1·2	+0·5	+1·3	−0·5
5	−1·4	0·4	+1·4	−0·4	−1·3	+0·4	+1·3	−0·2
6	−1·3	0·3	+1·3	−0·3	−1·3	+0·3	+1·3	−0·3
7	−2·1	0·3	+1·9	−0·3	−1·6	+0·4	+1·8	−0·3
8	−3·5	0·3	+3·5	−0·1	−3·6	+0·3	+3·9	−0·1
9	−2·4	0·1	+2·4	−0·2	−2·8	+0·1	+2·8	−0·1
10	−1·4	0·3	+1·4	−0·2	−1·3	+0·2	+1·3	−0·3
11	0	0·5	0	−0·4	+1·0	+0·5	−1·0	−0·5
12	−3·4	0·3	+4·4	−0·2	−8·8	+0·3	+8·8	−0·2
13	−1·9	0·2	+1·9	−0·1	0	+0·2	+1·1	−0·2
14	−1·6	0·2	0	−0·2	+2·1	+0·1	−2·1	−0·1
15	0	0·2	3·1	−0·2	+2·5	+0·2	−5·0	−0·3
16	−1·9	0·1	0	−0·1	+2·1	+0·1	0	−0·1
17	0	0·1	0	−0·1	+1·7	0	−1·7	0
18	+5·5	0	0	0	+5·3	0	0	−0·1

T_n° = tangential component due to gas pressures for T.D.C.—error = 0 deg.
T_n^+ = tangential component due to gas pressures for T.D.C.—error = +1 deg.
ψ_n^+ = phase-angle of component due to gas pressure for T.D.C.—error 1 deg.

$$\Delta T_n^+ = \frac{T_n^+ - T_n^\circ}{T_n^\circ} \times 100 \text{ per cent.}; \ \Delta \psi_n^+ = \psi_n^+ - \psi_n^\circ.$$

T_n^- = tangential component due to gas pressure for T.D.C.—error = −1 deg.
ψ_n^- = phase-angle of component due to gas pressure for T.D.C.—error = −1 deg.

$$\Delta T_n^- = \frac{T_n^- - T_n^\circ}{T_n^\circ} \times 100 \text{ per cent.}; \ \Delta \psi_n^- = \psi_n^- - \psi_n^\circ$$

TABLE XXVIII.—INFLUENCE OF THE PRE-INJECTION ANGLE.

2—S.C.S.A. type W.S. 546, N=3,600 b.h.p., n=160 r.p.m. M.e.p.=6·5 kg/sq. cm.	15 deg.		18 deg.	
n of comp.	T_n	ψ_n	ΔT_n per cent	$\Delta \psi_{no}$ deg.
1	7·378	69	−4·6	−2
2	7·355	46	−2·2	−0·5
3	4·662	35	−2·4	−0·3
4	2·907	27	−3·9	−0·3
5	1·976	23	+5·4	0·
6	1·188	23	+4·4	−0·5
7	0·596	22	+0·3	−0·9
8	0·318	20	+35	−1·4
9	0·301	21	+11	−0·6
10	0·243	24	+4·1	−1·3
11	0·209	26	−33	−5·9
12	0·100	23	+27	−5·3
13	0·069	21	+140	−1·5
14	0·097	22	+40	−2·7
15	0·089	22	−15	−3·0
16	0·063	21	+13	−1·6
17	0·042	21	+9	−1·2
18	0·045	21	−18	−0·3

T_n = harmonic components due to gas pressures in kg/sq.cm.
ψ_n = phase-angle of components due to gas pressures in crank angle degrees.

$$\Delta T_n = \frac{T_n\,(18 \text{ deg.}) - T_n\,(15 \text{ deg.})}{T_n\,(15 \text{ deg.})} \times 100 \text{ per cent.}$$

$$\Delta \psi_n = \psi_n\,(18 \text{ deg.}) - \psi_n\,(15 \text{ deg.})$$

always higher than that for torsion, was a reminder to be careful with bending effects, e.g. those caused by axial vibrations, and consequently it was necessary to endeavour to avoid excessive bending moments. The use of an electronic computer was of great assistance for that purpose.

MR. TH. WILSE wrote that after having studied Mr. Archer's valuable work he thought the paper would be very useful and instructive for those like himself who were surveyors or superintendents.

Torsional vibrations had been dealt with very thoroughly. Together with the statistics of defects it was of interest to note that the average years of service before failures were *6·08 years* (Table VII). The time indicated that the crankshafts could have carried out some 10^8 revolutions and possibly more than 10^{10} stress variations due to torsional vibration.

The statistics unfortunately did not differentiate between failures due to bending and those due to torsion.

One might, however, expect that if the stresses had been constant through the whole service period, the failure would have occurred at an earlier date.

Was there any reason to believe that torsional failures could occur from gradual altering of the torsional system? Bearing in mind the increased resistance to the hull because of roughness, is was evident that the velocity of water at the propeller decreased during the ship's lifetime. Furthermore the propeller blades were often liable to considerable erosion, especially at the blade tips. The propeller, including entrained water might thus obtain a decreased moment of inertia. Had torsional vibrations, measured for instance after four or eight years of service, been compared with the corresponding measurements taken when the ship was delivered from the builders?

The formula: IF^2 = constant (approximate) indicated that a change in the moment of inertia I of 10 per cent however, only altered the frequency F by about 5 per cent. Even a relatively small change in a critical frequency might be of importance if near enough to the service conditions.

Author's Reply

In reply to the discussion the author said that Mr. Jackson had enquired whether the analyses of crankshaft performance given in the paper took account of results from engines built many years before. As indicated in the paper, all shafts considered were post-war built (between 1947 and 1962 inclusive (16 years)) and the records of defects covered the ten-year period, 1953 to 1962. Inevitably, there had so far been relatively little total experience with the larger and more recent designs and, in consequence, insufficient time in which possible points of weakness might show themselves. It was to be hoped they would indeed be few and far between.

The author agreed with Mr. Jackson that slip of oil engine crankshaft shrinks rarely occurred under normal operating conditions and were usually the result of severe shock or vibration. Nevertheless, cases were on record where, owing to misalignment, bellmouthing of the shrink bores had taken place after a number of years in service, allowing oil seepage and final failure of the shrink to occur.

Fig. 5 showed that for shafts, other than of triple-crank type, the ratio of number of slips of journal shrinks to crankpin shrinks was 5 to 1 for fully-built shafts, which was contrary to Mr. Jackson's stated experience. It was clear that this would in general be much less serious than in the triple-crank, combination type where slip of one shrink, either side pin or centre pin, could cause severe misalignment of main bearing journals. In fact, in the former type of engine, journal shrink slip acted as a kind of relief device in emergency, thus helping to avoid more serious consequential damage, a further argument also against the use of dowels, now rarely fitted.

Mr. Jackson's appeal for easement of the Society's Rules in respect of axial thickness of webs at the shrink had already been met, in that, in the latest revision, up to 16 per cent reduction was permitted if an equivalent strength of web and shrink were provided.

In the author's opinion, where it was necessary to stiffen up the crankshaft torsionally on account of critical speeds, it was better to increase the diameters of pins and journals rather than cut down on web thickness and this had the added advantage of increasing overlap of side pins and journals. As Mr. Jackson pointed out, the journals on a Doxford shaft carried no combustion load, consequently he should be in a better position to reduce axial length of main bearings than other designers. The author did not agree that semi-built shafts should be allowed a smaller axial thickness of web than fully built shafts, the rates of incidence of slip per *shrink* at risk being approximately equal (Table VII).

He was grateful to Mr. Jackson for his explanation as to the reason for fitting the aluminium pistons in *Case I*. This was, in fact, the correct one.

In Mr. Jackson's stated agreement "that breakages were most likely to occur at the point of maximum stress even when that was not the same as the position of the node", he concluded that, in fact, he meant "point of maximum stress *range*".

The author would remind Mr. Jackson that the factors of safety calculated in the paper assumed that the torsional vibration additional stress co-existed with only *one* other adverse influence at a time, which did not seem an unreasonably remote possibility.

Replying to Mr. Nestorides, the author said that he was familiar with the work of both Stahl and Leikin in respect of crankweb fillet stresses and stress concentration factors, but of course their work was confined to quite small specimens and thus extrapolation to marine-sized crankshafts would be hazardous, except possibly for reversed direct stresses where size effect was likely to be less important. The work of Frost (reference 14 in the paper) on mild steel suggested that it would be prudent to assume that up to theoretical stress concentration factors, k_t, of about 4, the fatigue reduction factor k_f was of equal value, i.e. the notch sensitivity factor q was unity. From the work of Kuhn and Hardrath[56] and Heywood[57] it would seem even more advisable to work to k_t for higher tensile steels unless the size of notch was particularly small. From the results of these and other researchers it was clear that allowable stress levels could not be increased *pro rata* with tensile strength.

Mr. Nestorides's evidence, from work at B.I.C.E.R.A., of increased bearing wear under conditions of severe torsional vibration lent useful support to the need for limiting torsional vibration amplitudes in the vicinity of engine service speeds, quite apart from the shaft fatigue strength aspect. This consideration was, of course, emphasized in Lloyd's Register of Shipping's Guidance Notes for T.V. Stress.

The use of analogue computers for gear-branched systems had been elegantly described by Yates[58] in 1955, and Mr. Nestorides might perhaps care to refer to that work.

Mr. Nestorides's remarks on the unexpectedly large influence of gear housing flexibility on torsional natural frequency and amplitude were of much interest but the author doubted if this factor was of over-riding importance except in very stiff installations and for node-at-gears modes.

Mr. Nestorides's speculations on the Draminsky effect were in fact correct, in that from calculations already made by the Society it appeared that fortunately the effect was of importance in only a small minority of cases. His remarks on *Case* 1 in the paper were, however, inappropriate, since although fortuitously, the arithmetical sum of the predicted flank and resonant stresses calculated by conventional forced frequency tables happened to agree with the total measured stress, the 8th order measured stress on resonance was, as stated in the paper, some five times greater than predicted, in fact, the wave form, to the eye, was almost pure 8th order.

Mr. Zdanowich was correct in presuming that in *Case I* of the failure examples the inertia of the generator was indeed large compared with that equivalent to the engine (about 12 times) but no quill or coupling was fitted.

Mr. Zdanowich's remarks on what he termed "auxiliary" modes were of interest but, in the author's view, the comparable magnifiers in marine service would rarely be as high as even 100. He doubted if many marine auxiliary drives had to rely solely on the weakly restraining effect of gearing or hysteresis damping alone.

As regards Mr. Zdanowich's doubts on the Draminsky effect he would refer him firstly to Dr. Draminksy's contribution. Secondly, he would assure him that the Society was fully alive to the possibility of the effect he had postulated and attempts had been made in a number of previous cases to ascribe discrepancies in amplitude to it. In the author's view, however, this was definitely not the cause of the higher than normal stresses in the cases cited.

Thirdly, the author knew of at least five examples of crankshaft cracking or damage in large 2-S.C. marine engines attributable, from calculation and/or measurement, to the Draminsky effect and four of them were of identical size, type and firing order.

The single-cylinder work stated to have been referred to by Professor Johnson as bearing on the existence of the Draminsky effect was most probably that carried out in Newcastle by Professor Goldsbrough, published in the Proceedings of the Royal Society in 1927 (reference 54 in the paper) which the author would commend to Mr. Zdanowich. It was believed that further work along similar lines was done by Dr. Wylie Gregory at Cambridge a few years ago.

In Mr. Zdanowich's lengthy dissertation on dynamic magnifiers he had suggested that for large slow speed direct-coupled installations values of D.M. might approach 100, whereas in comparable geared installations it could be under ten. Assuming that Mr. Zdanowich's definitions of dynamic magnifier and equilibrium torque were those, for example, as used by Ker Wilson, viz:

$$\text{D.M.} = \frac{\omega_c \Sigma\left(\frac{I}{g}\,\theta^2\right)}{\Sigma(k\theta^2)}$$

$$\text{and } T_{no} = \frac{T_n \Sigma\theta}{\omega_c^2 \Sigma\left(\frac{J}{g}\,\theta^2\right)}$$

it was indeed rare, in the author's experience, for D.M. values to exceed 50, even for minor orders. For 1-node, or "output", modes involving gears the magnifier would depend vitally upon the amount of external damping present. For example, propeller damping would almost invariably swamp any small damping from gears.

The remainder of Mr. Zdanowich's contribution hardly called for reply since the subject matter lay mostly outside the scope of the paper and in any case, dealt with certain highly specialized aspects of vibration engineering applied to particular types of high speed geared engine, not very representative of marine practice.

The author was grateful to Mr. Olsson for his encouraging and valuable contribution. The bending fatigue limit of 12 tons/sq. in. had been arrived at by assuming a 50 per cent ratio of reversed bending fatigue strength to ultimate tensile strength for a forging of 32 tons/sq. in. tensile, based on small polished specimens. Since, however, the material in a large crankthrow subjected to bending and torsional stress gradients would experience at any given instant negligible variation of stress over a sectional area equal to that of a small polished specimen, it was considered more appropriate to use the reversed direct stress rather than the reversed bending stress fatigue strength. If the ratio of the two were taken as 0·75 (a rather pessimistic value in relation to most published data) the value of intrinsic unnotched fatigue strength was obtained as:

$$\sigma_e = \pm \tfrac{1}{2} \times 32 \times \tfrac{3}{4} = \pm 12 \text{ tons/sq. in.}$$

The author believed this value would be about right for throws made by processes such as folding or continuous grain flow, but that a lower value of no more than about $\pm$ 10 tons/sq. in. could be relied upon from throws made by the block forging process. (References 3 and 4 in the paper).

In any case the use of a notch sensitivity factor of unity should more than outweigh any slight optimism as to scale effect.

Furthermore, as the author had stated in his "Conclusions", no attempt had been made to allow for the adverse effect of the relatively high static shrinkage stresses in the region of the crankweb/pin junction.

The author would emphasize that the object of the sample computation in the paper was not so much to try to arrive at any precise figure of safety factor, but rather to assess its order of magnitude, and especially, the relative importance of the various adverse influences postulated.

For reasons already stated, the author thought Mr. Olsson's figure of $\pm$17,000lb./sq. in. for the 63,000lb./sq. in. u.t.s. material in the seven-throw crankshaft unduly pessimistic and would therefore have assessed a higher safety factor of about 2 for measuring point 1 on the Götaverken-Modified Goodman method.

The much lower stresses measured in the crankpin recessed fillet were interesting. Assuming normal main bearing clearances, it seemed more likely that the chief cause was, in fact, the restraining action of the bearings rather than the larger fillet radius. The cumulative effect of torsional "wind-up" under mean torque would increase towards the after end of the crankshaft leaving reduced effective clearance to absorb extra twist due to torsional stress variations.

The author was glad to note Mr. Olsson's agreement that, in general, axial vibrations were of less importance than torsional vibrations. His limiting amplitude of $\pm$0·05in., whilst useful as a rough guide to permissible free-end axial amplitude, was hardly of general applicability. Clearly, this must depend on the design of the crankshaft, such as number of throws, firing order, angles between adjacent cranks, etc., etc. For example, a triple-crank type shaft for opposed piston engines could accept very much higher amplitudes without large stresses and, similarly, for long stroke engines compared with short stroke engines.

The measured bending stress curves in Fig. 48 for the ten-cylinder Götaverken engine were particularly interesting. The shape of the mean dashed curves (eliminating vibration amplitudes), especially Gauge 31, was almost exactly duplicated by Fig. 27 (b) of the paper in which connecting rod angularity effects were allowed for. However, taking into account the 25 per cent greater maximum pressure in the author's example and assuming the same bending stress concentration factor of 3·0 for the Götaverken shaft, it would seem that either the conventional methods of calculation exaggerated the stress range by a factor of about 2, or alternatively, the true stress concentration was appreciably less than 3. From static strain gauge results, the latter seemed unlikely, hence it could legitimately be concluded that the conventional stress calculation for combustion bending was well on the safe side. In either case, however, as shown in Table XVII, the effect of quite large variations in bending fatigue range was relatively unimportant compared with the torsional vibration stress ranges.

The author was glad to note the rather better than average overall incidence of cracked and broken shafts claimed for Götaverken engines, viz. 0·2 per 100 shaft years. The Society's own records for ships classed with Lloyd's taken over the same periods as given in the paper, gave an incidence rate for semi-built cast throw shafts of 0·27 for Götaverken-built engines.

Mr. Siggers' strictures on the use of recessed fillets were considered rather too extreme, in that, as stated in the paper, provided adequate radius was given and, in the case of solid forged shafts, sufficient overlap, there was experimental evidence to show that the undoubted advantages accruing were not too dearly bought. For semi-built shafts it was almost universal practice, as reference to Fig. 4 would confirm.

The author would agree with Mr. Siggers in respect of the need for greater emphasis on ruggedness and reliability with the increasing adoption of automatic and remote control.

Mr. Siggers' diffident suggestion with reference to Figs. 5 and 5a had been considered when planning the paper but had regretfully been rejected owing to space limitations in an already lengthy manuscript, and also owing to the difficulty of establishing prime causes from the somewhat terse survey reports. In any case the surveyors themselves were only too often unable to obtain reliable evidence as to the actual course of events leading to a failure. However, an attempt had now been made to comply with this suggestion which had also been voiced by Dr. Davis and other contributors. The results, for what they were worth, were given in Tables XXIX and XXX corresponding to Figs. 5 and 5a, respectively. It would be noted that, unfortunately, in almost 50 per cent of the cases the cause of failure was not established.

Mr. Milton's remarks were much appreciated. In reply

Table XXIX

Probable Causes of the Crankshaft Failures (Fig. 5)

1)	Material Defects:		
	a) Casting	5	
	b) Forging	2	
	c) Inclusions, etc.	3	
	d) Indiscriminate welding of crankshaft without surveyor being informed	5	15
2)	Mal-alignment:		
	a) Mal-alignment	—	
	b) Bending fatigue	2	2
3)	Design:		
	a) T.V.	4	
	b) Oil hole	1	
	a) + b) T.V. + Oil hole	2	
	c) General design	1	8
4)	Others:		
	a) Water in cylinders	—	
	b) Shock loading (buoy chain round propeller, etc.)	2	
	c) Break up of hardened surface due to overheating	1	
	d) Lubrication failure	1	4
5)	Unknown*:		
	a) Moved shrinks	8	
	b) Cracks and breaks	21	29
	Total		58

* In the majority of these cases the cause was not investigated.

Table XXX

Probable Causes of the Crankshaft Failures (Fig. 5a)

1)	Material Defects:		
	a) Casting	—	
	b) Forging	1	
	c) Inclusions, etc.	1	2
2)	Mal-alignment:		
	a) Mal-alignment (includes 3 cases accelerated by corrosion)	10	10
3)	Design:		
	a) T.V.	2	
	b) Oil hole	—	
	a) + b) T.V. + Oil hole	—	
	c) General design	8	10
4)	Others:		
	a) Water in cylinders	7	
	b) Shock loading (excess fuel, etc.)	2	9
5)	Unknown*:		
	a) Moved shrinks	4	
	b) Cracks and breaks	13	17
	Total		48

* In the majority of these cases the cause was not investigated.

to his first point the author shared his implied preference for a forged crankthrow, preferably, however, when made by a process such as folding. The casting was indeed cheaper than such a forging by about 30 per cent so far as U.K. prices were concerned. On his second point, statistics showed that for post-war built direct-drive oil engine installations the distribution between amidships and aft end was about 50/50. Of the 53 aft end cases of failure in Fig. 5 only about half had failed in the after half-section of the shaft. However, the author would agree that Mr. Milton's suggested causes probably accounted for some part of the greater incidence of trouble in aft end installations. Mr. Milton's suggestions concerning the cause of breakdown in *Case 1* seemed reasonable, but the author thought the failure was probably a result of the combined effect of the two factors considered. On the general question of bending fatigue failures whilst it obviously was most important to limit bed plate deflexions due to ship loading and/or hull straining action in heavy weather, in the author's opinion, such deflexions were less dangerous than those resulting from differential wear-down of adjacent main bearings since the former were much less localized.

In reply to Mr. Michaelis the author would point out that only in Fig. 24 was there a slight step associated with a locating face on the side of the web. He could not recall any cases of failure having originated at such points, also he did not consider some slight undercutting of the pin at the fillet to be of consequence provided it were well blended into the pin surface. The Society had approved a few quite small diameter crankshafts assembled by the oil injection method. The difficulty with very large shafts, however, was to achieve the very fine tolerances required and, in general, greater security would be expected by hot shrinkage, which, in his view, was better carried out vertically rather than horizontally.

The author was indeed grateful to Mr. Yellowley for his massive (if he might be excused the term) contribution to the discussion, which had undoubtedly added greatly to whatever value the paper might have.

It was interesting to note that the six-cylinder example chosen by him was very similar to that given in the paper, but of somewhat lower m.i.p., maximum pressure and r.p.m.; otherwise the same ahead firing order, 1-5-3-4-2-6, but with slightly smaller diameter semi-built cast throws instead of semi-built forged.

Considering the torque curves of Fig. 52 derived by the conventional summation method, Mr. Yellowley invited comparison with those of Fig. 32 in the paper. This was interesting but an even more striking comparison was with the dashed curves of Fig. 33 which had been derived by a similar method but using a closely approximated (first six harmonics only) basic torque curve. The agreement in general shape was indeed very close. Numerically also, the maximum stress ranges tallied extremely well, viz:

	Fig. 33	*Fig. 52(b)*
Abaft No. 4	5,666lb./sq. in.	5,855lb./sq. in.
Abaft No. 5	5,716lb./sq. in.	5,300lb./sq. in.
Abaft No. 6	2,322lb./sq. in.	2,300lb./sq. in.

As pointed out in the paper, if due account were taken of crankshaft inertia and torsional elasticity, the stress ranges towards the after end of the engine tended to be less than calculated by the conventional method and for the engines in question would probably not greatly exceed 1,000lb./sq. in. abaft No. 6 crank.

Mr. Yellowley had pointed out that the bending stress ranges were only about half of those due to torque, but in the author's example the ratio was nearer two-thirds owing to the 25 per cent higher maximum combustion pressure. It was remarkable that the range of crankshaft bending stress, which was a maximum of 2,800lb./sq. in. at No. 2 journal (Fig. 58), agreed almost exactly with the value calculated for the author's example by the simple encastré beam method, namely 3,634lb./sq. in., when due allowance was made for the 20 per cent lower maximum combustion pressure in the former engine. Whatever the calculated stresses might be, it was encouraging to note from Mr. Olsson's strain gauge measurements that actual bending stress ranges were likely to be appreciably lower, possibly no more than about half of those calculated.

The combined strength criterion correctly quoted by Mr. Yellowley (Fig 59) as having been introduced by Mr. Milton (later Dr. Milton, Chief Engineer Surveyor) was, of course, based on the St. Venant hypothesis of maximum principal strain. It was therefore not strictly correct to refer to it as a principal stress. It was in fact the equivalent simple uni-axial direct stress which would induce the same strain as the maximum set up by the principal stresses replacing the combined bending and twisting stresses.

It was interesting to note that when Mr. Yellowley's numerical data were applied to the Marin equations in Appendix II, *Method* 1, the factor of safety, without reference to torsional vibration, amounted to 2·2. Doubtless this value, 20 per cent greater than the author's 1·85, could be ascribed partly to the lower value of k_s and partly to the somewhat greater excess over Rule size. This could be regarded as reliable confirmation of the author's own results.

The author would assure Mr. Yellowley there was no intention to be pessimistic, only to suggest caution in view of the limited shaft service years so far accumulated by the really large engines. Problems of forging and casting inevitably became more acute with increasing size, hence it would be reasonable that the incidence of defects should also tend to rise with size, quite apart from the size effect in fatigue. Tables III and V definitely suggested that above 2,000 b.h.p. there was a trend of increasing incidence with horsepower.

Mr. Christensen's doubts about the cause of failure in *Case II* had been shared by Mr. Milton, but the author could only state what was actually reported. However, it did seem not unlikely that some degree of misalignment existed either before or after the re-grind. It was unlikely that the misalignment was initiated during the regrinding process since presumably all journals would be ground true at the same setting and with the shaft in its original centres. It was not known whether there had ever been a very long period during which only one engine per screw was run. Under such conditions, of course, the propeller revolutions *for the same m.i.p.* would drop to $\sqrt{0 \cdot 5} = 0 \cdot 707$ times normal service power r.p.m. Only if means were provided (or deliberately taken) to increase the lift of the fuel pumps beyond maximum continuous rating, e.g. by removing the overload stops, could increased torque and ship speed be obtained. There was no evidence to indicate that such action had been taken in this case. So far as the Society's Rules were concerned, although not specifically mentioned, it was common practice to allow overload-testing for a short period at up to 10 per cent above maximum continuous rating, for proving purposes only. It was not envisaged that this margin would ever be used in service.

Mr. Christensen complained that the author had not gone into the practical details of crankshaft gauging in the paper. It was, however, assumed that most members would be familiar with the necessary precautions to avoid the kind of pitfalls so well described by Mr. Christensen.

On his final point, whilst the author admitted he might well have mentioned owners among the interested parties concerned with clandestine welded repairs, he felt Mr. Christensen's remarks were a little unjust. After all, the classification surveyor acted in good faith to safeguard owners' interests. It might surely be fairer to say that such cases should lead to loss of confidence of owners in those *builders* who resorted to such practices.

Dr. Davis seemed to have read more into the paper than perhaps it merited. Certainly the author had at no time envisaged any increase in crankshaft scantlings. The general run of statistical evidence to date certainly would not warrant any such increase. He would refer Dr. Davis to his reply to Mr. Olsson regarding the underlying inspiration for the paper. The author hoped that improved methods of calculation might lead to a more *discriminating* appraisal of different design features which could not hope to be fully covered by the semi-empirical approach so far generally followed in the industry.

The author was intrigued by Dr. Davis's algebraical analysis of the design problem in generalized terms. It was obvious that provided the factors governed by the summation sign could be minimized, this would allow higher values of the stresses under A and a corresponding reduction of K.

Unfortunately it was not always just a question of a failure not being understood. In many cases it was rather that all the relevant facts and circumstances surrounding the failure were not ascertainable.

In response to Dr. Davis's request, shared by Mr. Siggers, for more information concerning the number and proportion of failures, the causes of which were not identifiable (either because they were not understood or because information was lacking), he would refer him to Tables XXIX and XXX and his reply to Mr. Siggers on this point.

Mr. Leide's contribution, coming from an experienced applied metallurgist, commanded close attention. The author was only too well aware that the materials used for large crankshafts today were far from being ideal and isotropic, and, of course, castings, although less directional in properties, shared these deficiencies with forgings. He would support Mr. Leide in calling for the use of the most modern inspection methods, even beyond those demanded by the classification societies, in the interests of ultimate safety. The whole question of possible standards of acceptance for defects in forged and cast crankthrows was currently under review by Lloyd's Register.

The author was familiar with the full scale fatigue tests mentioned by Mr. Leide. He understood that in these tests, which were in fluctuating tension, the minimum stress was kept constant at about 5 kg./sq. mm. and the maximum cycle stress was then varied until the fatigue limit, taken at 2×10^6 cycles, was reached. This upper limit was understood to be about 30 kg./sq. mm. On the Modified Goodman basis, the equivalent *reversed* fatigue limit would be given by:

$$\pm\sigma_e = \frac{f_{max} - f_{min}}{2\left(1 - \dfrac{f_{max} + f_{min}}{2 \text{ u.t.s.}}\right)}$$

The value of u.t.s. for the material in question was stated as 52·7 kg./sq. mm., or 33·5 tons/sq. in., giving a reversed fatigue limit of $\sigma_e = \pm 18 \cdot 8$ kg./sq. mm. or $\pm$ 12 tons/sq. in. This, as suggested to Mr. Olsson, was about right for the small-scale reversed *direct* stress fatigue limit to be expected from material of this tensile strength.

Mr. Leide's finding of a high carbon content at the origin of failure might perhaps find a parallel in some recent serious failures in turbine thrust bearings of pad and collar type in which, under certain conditions, débris of a "wire wool" consistency was found embedded in pockets in the white metal of the pads, the material of the wire having been "machined" off the collar by severe grooving action. The wire itself was found to be extremely hard (of the order of 900 V.P.N.) and largely martensitic in structure.

The author did not believe there was anything in the theory of nascent hydrogen generated by electrolytic action due to water in the oil. The classic experiments of Amari and Ando (1952)(59) on the "bubbling" crankshafts in Japan would tend to negative this suggestion.

The author thought that the marine Diesel industry was greatly indebted to Dr. Draminsky, not only for having recognized the importance of the phenomenon of secondary resonance but also for so ingeniously providing the necessary procedure for its quantitative mathematical analysis. He was also grateful for Dr. Draminsky's amplifying remarks, including his comments on the relative phase of "fictive forces", which lent added weight to the author's presentation of this important factor, undoubtedly capable of adversely influencing the life of marine crankshafts. Lloyd's Register were now carefully examining every case submitted for approval to ensure that no Draminsky effect of consequence existed near the service speed.

Mr. Hind's interesting account of his firm's crankshaft experience over 40 years was an important contribution for which the author was grateful. He took note that no less than 70 per cent of all the firm's casualties had been due to torsional vibration, which again tended to support the author's contention concerning the predominating influence of torsional effects on crankshaft life. The good performance of the C.G.F. shafts was noteworthy. Reference to Table V showed an incidence rate for 4-S.C. solid forged shafts as low as 0·15 per 100 shaft years for engines between 1,001 and 2,000 b.h.p. Doubtless this range would embrace many of the products of Mr. Hind's company.

On the subject of continuous control by dampers of criticals near the service speed, Mr. Hind would be interested to learn that in the latest revision of the Society's Guidance Notes (which were soon to be published in Chapter R of the Rules) this was now accepted, where unavoidable, provided the damper was adequately cooled and of a type not subject to mechanical wear and tear. This amendment had been introduced in deference to the trouble-free performance of modern dampers, for example, of the viscous friction type.

On his remarks concerning maximum cylinder pressures in relation to crankshaft scantlings, the author appreciated the point but doubted if it was of very great significance so far as modern multi-cylinder engines were concerned, as the following tabulation indicated:

	Maximum Combustion Pressure			
	55 kg./sq. cm.		*70 kg./sq. cm.*	
	6-cylinder	*10-cylinder*	*6-cylinder*	*10-cylinder*
Rule diameter (mm.)	530	561·4	543 (+2·45 per cent)	575 (+2·42 per cent)

The above figures were for 2-S.C.S.A. engines as given in the paper but with a moderate mean indicated pressure of 120lb./sq in.

In practice, for the higher maximum pressure the m.i.p. would be about proportionally increased which would reduce the relative importance of increased maximum pressure below the percentage given.

For comparable 4-S.C.S.A. engines the above percentage increases in Rule crankshaft diameter might be about doubled, but, of course, the mean indicated pressures for 4-S.C.S.A. engines would be much higher for the same power, which would in turn also tend to reduce these percentage increases.

Mr. Hind's remaining points concerning the need for rules covering surface-hardening of crankshafts and shaft dimensions for Vee-type engines were well taken and were already being considered.

The author was glad to note that Mr. Hind's company had apparently not so far encountered evidence of secondary resonance. One possible reason could be that the phase of the "fictive force" relative to that of the *nth* orders concerned was such as not to add appreciably to the *nth* order resonant stress, or might even reduce it. Another possibility could be that the 2nd-order vector sum (Fig. 38) was not large, both of which possibilities would of course depend upon firing order and to some extent upon swinging form. Again, the larger the ratio of reciprocating/total equivalent rotating masses per cylinder line, i.e. the larger the coefficient β_c (Table XX), the greater would be the relative importance of "fictive force". It was known that this was greater for large slow-running long-stroke, crosshead type marine engines with negligible flywheels than, for example, with relatively small, trunk piston, short-stroke, medium or high speed engines, especially if these were four-stroke with comparatively heavy flywheels.

The author was glad to have Dr. Ing. Schmidt's contribution, since he had raised an important question on which little experimental data were available, namely, how far could one safely reduce the radial distance between shrinkage bore and underside of crankpin in a semi-built crank? This was a difficult one to answer; however, in the author's opinion, it would be unwise to reduce this distance below 15 per cent of the web bore diameter, i.e. stroke not less than 2·3 times web bore diameter, and even then the axial thickness of the web should be fully up to Rule size, namely 0·625 d_R. Where a smaller axial thickness was provided, it would be prudent to increase the limiting (stroke/web bore) ratio in inverse proportion to axial thickness fitted. In any case, where small ratios were adopted, advantage should be taken of the increased axial width on the above basis to keep shrinkage allowances to a minimum.

The author was indebted to Mr. Smedley for some particularly useful comment on fatigue strength. It was perfectly true that the two failure relations between mean and variable stress used in the paper, i.e. the Söderberg and Modified Goodman, were the most conservative of the four different relationships he had listed, and that, other things being equal, the actual safety factor for the example in the paper might well be higher than the seemingly rather pessimistic value of 1·52 (allowing for torsional vibration) indicated and might even approach a value of 2 as Mr. Smedley had suggested. However, as pointed out in the paper, no account had been taken of the high hoop and appreciable radial static stresses set up by the shrinkage process in a semi-built shaft, especially in the fillets on the underside of the crankpin. Consequently, the author's Söderberg calculations might not perhaps be as conservative as would at first sight appear. In any case, the object of the exercise was as much to assess the relative importance of the various unfavourable influences to which crankshafts could be exposed as to attempt very precise calculations of minimum margins of safety, the range of variation of which, as indicated in the paper, must remain somewhat conjectural pending more reliable large scale fatigue research.

The author was in general agreement with Mr. Smedley's list of subjects for further research bearing upon crankshaft design and service reliability. Unfortunately, owing to various bugbears, including that of problematical size effect, fatigue testing would ideally require to be done at full scale if further progress in our knowledge of fatigue strength of large crankshafts was to be made, and, of course, that cost money. It was essentially a matter of priorities.

Concerning item (1) of Mr. Smedley's list, he had stated that "friction at bearings and rubbing surfaces controlled the (vibration) stress levels". Although doubtless some energy was absorbed in that way, both Draminsky[60] and Nestorides (B.I.C.E.R.A.) had found good evidence that a major part of the damping energy was contributed by the hydrodynamic "pumping" action of the journals in their bearing clearances, the locus of shaft centre movement being a closed multi-lobed circular path.

Comparison of the original work of Frost[14] with the reference cited by Mr. Smedley (Frost, Holden and Phillips, 1961) had convinced the author that his own interpretation of Frost's relation, $\sigma^3\, l$ = constant, as stated in the paper, was, in fact, correct. Frost stated* that although beyond the critical value of k_t, the alternating stress required to propagate a crack was independent of the notch radius, it was dependent on the crack length. Further, the $\sigma^3 l$ relation was only applicable when l was taken as the combined length of the notch and crack. Now the length of a non-propagating crack at the root of a sharp notch was usually small compared with the depth of the notch so that the minimum stress required to propagate a crack could be approximated by substituting the depth of the notch only for l in the above relation. Thus for mild steel, with geometrically similar notches, the minimum fatigue strength (reversed direct stress) for a notch depth of 0·2in. was about $\pm 3\frac{1}{2}$ tons/sq. in., whereas for a notch depth of 0·05in. it increased to $\pm 5\frac{3}{4}$ tons/sq. in. and for a 0·005in. notch up to about ± 10 tons/sq. in. Thus, the Frost equation could indeed be used to justify the inference that for geometrically similar notches (and presumably also fillets, or oil holes) in a given material, provided they were sharp enough, i.e. beyond $k_{crit.}$, the smaller the absolute "notch" size, the greater the minimum fatigue strength required to cause crack propagation. Mr. Smedley had also stated "the conditions were modified by a corrosive atmosphere", but of course the tests in question were not conducted in such an atmosphere, so it was concluded that he really meant "would be modified" as, for example, in an engine crankcase.

Ir. Hootsen had raised an important practical question concerning the Society's requirements for magnetic crack detection of cast steel crankwebs. As the author had stated in his reply to Mr. Leide, the Society had so far no specific standards of acceptability for surface defects in cast steel crankwebs but these were currently under consideration. In the meantime, of course, each case had to be judged by the surveyor concerned on its merits as to nature, size, depth and location. However, as Ir. Hootsen had pointed out, and as

* See also Forrest, P. G. 1962. "Fatigue of Metals". Pergamon Press, pp. 146-149 and Figs. 75 and 76.

most surveyors were well aware, the areas around the fillets between crankpin and crankweb, especially on the underside of the pin, were particularly critical. Naturally, the problem was complex since the acceptability of a given defect depended upon many factors, e.g. design and shape of crankthrow, dimensions of fillets, oil holes, etc., position of crankthrow in the engine (if known at inspection stage?), etc.

The author was grateful to Mr. Dowie for his reference to Taylor's fatigue results for mild steel bolt material showing the effect of mean tensile stress on limiting fatigue range. As he had pointed out, these showed a by no means negligible influence of mean stress, namely a loss in fatigue *range* of about half the increase in mean stress. Of course these results were based on smooth $\frac{1}{2}$-in. diameter test pieces and, furthermore, it was likely that for notched specimens the limiting fatigue range would be somewhat less sensitive to mean stress. Nevertheless, these tests did tend to show that the Modified Goodman relation was probably about right for 28/32 tons/sq. in. crankshaft steels. However, for relatively low values of mean stress, as usually was the case for marine crankshafts, there was little to choose between the Goodman and Söderberg relations, the divergence increasing with mean stress.

There was no doubt at all that the application of residual compressive stress at the surface in way of stress raisers in crankshafts, such as, for example, by nitriding or fillet rolling, could very substantially increase the fatigue strength of crankshafts and the author was glad to have the further references bearing on the latter subject. In his opinion, both these processes were specially useful for relatively small shafts, say up to about 6in. in diameter, but were less favourable in the larger sizes. This was because the depth of the hardened zones in both processes was relatively shallow and owing to the reduced stress gradient in the larger shafts, they would therefore be less effective against bending and torsional fatigue stressing. Some useful information on fillet rolling in larger-sized shafts had been given by Simonetti[36].

Dr. Ørbeck's kind remarks were much appreciated by the author, coming as they did from a pioneer in the application of computer methods to marine shafting vibration problems.

The first question he had posed was a difficult one and the author had not as yet fully crystallized his own views as to whether it might not perhaps be sufficiently accurate to determine the maximum total torsional stress range (including torsional vibration) for the particular crank at which the maximum net combustion bending stress (calculated by a simplified method) was nearest in phase with the maximum of the torque variation. In this respect he would refer Dr. Ørbeck to his replies to Mr. Olsson and to Mr. Yellowley, in particular to the close agreement reached by such a simplified bending stress calculation with the vastly more laborious and complex calculations carried out by Mr. Yellowley which seemed to have much in common with those used by Dr· Ørbeck. In any case, for multi-cylinder engines, at least in the absence of serious misalignment, the contribution of the bending stress to the equivalent combined stress, whatever criterion of failure might be taken, would normally be relatively minor for the conventional crankshaft. Admittedly, this might not be quite so true for engines of his company's type, but nevertheless it would not be expected that even for such engines having upwards of six cylinders, bending stress would be a major factor in the life of the crankshaft. This was especially true of the P and J designs with their much more rigid crankshafts, provided, of course, alignment was maintained. From the foregoing it would be apparent that the author took up a somewhat different position to Dr. Ørbeck on this particular aspect of crankshaft design. It could well be that the latter had been swayed by the high bending stresses measured on some of the 75 LB6 side webs, which were exceptionally weak in flexure.

Dr. Ørbeck's description of his company's extensive programme of research into axial vibration and bending stresses in the three engine types he had indicated was of very great interest. Especially striking were the wide variations in stress concentration factors for the side web fillets in the three designs.

On the particular point raised as to which bending moment should be taken across the axial thickness of the side web, one would normally assume that the bending moment at the mid-thickness of the web would be appropriate, but this might have to be modified if the variation was very marked.

As Dr. Ørbeck had inferred, the failures of the 75 LB6 crankshafts had been due primarily to the combined effect of the 4th order, I-node axial vibration resonance peak and the 7th order, III-node torsional vibration resonance peak rather than to the intrinsically poor design of side web alone. Had these vibration effects both been absent, the author was convinced that the 75 LB6 crankshaft would have been quite satisfactory in service. It was extremely unlikely that *all* the failed 75 LB6 crankshafts had been seriously misaligned as the prime cause of failure. Table XXIV was interesting, showing comparative calculations of side pin fillet stresses for the 75 LB6 and 67 PT6 engines on similar lines to those in the paper. However, although the *comparison* seemed fair enough, he noted that all the various bending stress ranges had been taken *in phase,* which was not necessarily justified. On that point he would refer him to Mr. Jackson's contribution (and the author's reply), since that suggested a very different viewpoint Furthermore, the misalignment contribution for such a relatively flexible shaft as that of the 75 LB6, namely a total range of actual fillet stress of $4{\cdot}2 \times 4{,}000 = 16{,}800$lb./sq. in., seemed somewhat exaggerated. For those reasons the author remained not fully convinced of Dr. Ørbeck's arguments on that point.

Regarding the use of dampers for the 75 LB6 engines, as Dr. Ørbeck would be aware, two dry cargo ships had been operating fully successfully for about six years with somewhat modified designs of shaft, but fitted with combined axial and torsional dampers of the silicones viscous fluid type.

The author was in full agreement with Dr. Ørbeck's final comment on calculated safety factors using more advanced methods of calculation, which he himself had also tried to express in his replies to Mr. Olsson and Dr. Davis.

The author thanked Mr. Maciotta for drawing his attention to the paper on axial vibration of crankshafts presented at the 1962 C.I.M.A.C. Congress. The author's comment in the paper had really been aimed at measured stresses due to axial vibration, e.g. using strain gauges in the fillets as had been described in reference[39], rather than by *calculation* from measured *amplitudes.* Nevertheless, the C.I.M.A.C. paper was a valuable contribution to our knowledge of these additional stresses.

The author was of the opinion that axial vibration stresses of the magnitude quoted were very exceptional and from Table XVI Mr· Maciotta would note that for fillet stresses of about ± 3 kg./sq. mm. the effect on overall safety factor was quite small. Nevertheless, the prudence of Mr. Maciotta's company in limiting such stresses to ± 2 kg./sq. mm. was laudable since thereby a greater margin was available for other unpredictable service stresses.

For large 2-S.C.S.A. engines, the size of flywheel required to reduce any "Draminsky" effect appreciably would be expected to be impracticably large, except, of course, by its detuning effect. He would refer Mr. Maciotta to his reply to Mr. Hind on this point. The majority of such engines today had negligible flywheels.

On the chronological distribution of the crankshaft failures listed in the paper he would suggest that Figs. 5 and 5a provided all necessary information. No obvious trends seemed to be revealed.

The author was grateful for Mr. Kleiner's kind remarks and as he rightly pointed out, the probability of failure in "straight" crankshafts today was less than $2\frac{1}{2}$ per 1,000 shaft years in service. His favourable comments on the influence of Lloyd's Rules and T.V. Guidance Notes on this good performance were much appreciated. As he knew, these had recently been revised and the latter would shortly be published in Chapter R. Mr. Kleiner's general remarks on factor of safety in relation to calculation methods were fully agreed, as had been indicated in the author's replies to Dr. Davis and Mr. Olsson. It would indeed be interesting to know the calculated safety factors of all the large modern crankshaft designs shown in Fig. 4, but it would, of course, be impracticable for the author to undertake this task in the time available. However, designers were themselves free to do so for their own crankshafts, using, for example, the methods and computer programmes outlined in the paper. The author would indeed be glad to receive any such results.

The author thanked Mr. Hinson for his interesting and practical remarks on corrosion fatigue based as they were on many years of observation and experience in the investigation of service troubles in Lloyd's classed ships. His suggestions for the avoidance of such alarming and unfortunately all too common, corrosion defects, so well illustrated in Fig. 62, were constructive and useful.

Mr. Butler's encouraging remarks on the paper were greatly appreciated. On the question of shrinkage in fully built shafts in relation to crankshaft scantlings he felt that Mr. Butler had over-simplified the problem. For example, the bridge piece material did not, in fact, have to withstand twice the mean shrinkage tension carried by the outer parts of the web since considerable support would be given by the web material between pin and journal remote from the minimum bridge piece ligament. In this connexion he would refer him to the work described in references[4] and [28]. The eye radial thickness in fully built throws was based on the yield strength of the material and on a margin of grip torque against slip justified by long experience. Between the eye holes, where the radial ligament was reduced, it would be expected that additional strength would be provided, e.g. by increase in mid-throw web width. No such possibility of reinforcement would be available for the material around the outer arc of the web shrink if reduced below $0{\cdot}438d$. Accordingly, Mr. Butler's proposal for making the outside circle of the web eccentric to the pin, with this in view, would not be considered acceptable.

Mr. Batten, like his colleague Mr. Smedley, was well known for his researches on fatigue strength of marine shafting and his contribution was therefore much appreciated. The author fully agreed with him on the importance of adequate lip radius for oil holes, and especially on the need for more care in the quality of surface finish given to the oil hole. He was glad to have Mr. Batten's support on the value of nitriding, on which he would refer him to his reply to Mr. Dowie. He could not see why it should be difficult to nitride the interior of the oil holes, but he agreed that unless all significant notches and stress raisers were so hardened, it was pointless to apply the process at all.

Mr. Batten's comments on the work of Frost cited in the paper were interesting and his inferences from it were generally agreed. The example (Fig. 64) he had contributed of non-propagating cracks in a sharply filleted torsional fatigue test piece was most intriguing and certainly lent further experimental support to Frost's conclusions on the laws governing the propagation of fatigue cracks in steel. As Mr. Batten had pointed out, if initial cracking from overstress due to whatever cause, such as, for example, racing, vibration, impact, etc. were sufficient to exceed the critical crack length, then propagation could indeed proceed under the much lower normal running stresses (see also, Archer 1949)*.

The author was much indebted to Mr. Visser, so well-known for his many valuable contributions throughout the years to the theory and practice of mechanical vibrations, for his characteristically constructive contribution. In particular, the author was in full agreement with his warning on the need for accuracy in the input data for computer calculations. The accuracy of the answers depended fundamentally upon that of the basic data. The results of Mr. Visser's careful analysis of the effect on the accuracy of the harmonic orders of the indicated torque diagram of various sources of error were most instructive and revealing.

Although the results were strictly only applicable to Farnboro type pressure diagrams, nevertheless most of Mr. Visser's four general conclusions were also applicable to other types of indicator diagram and his contribution would, in any case, be of much value for reference purposes in the future.

The author was glad to learn Mr. Visser's general experience of the use of dampers, in particular his company's preference for avoiding their use to control criticals near the service speed, which of course agreed with the Society's long-established recommendation. However, this had now been somewhat relaxed as indicated in the author's reply to Mr. Hind. The heat-loading limit of 400 B.t.u./sq. ft./hr. given by Mr. Visser was very useful design data.

It was interesting to note that Mr. Visser's company had measured stress concentrations in their particular designs of crankshaft of 4·0 and 1·9 in bending and torsion respectively, presumably for solid forged shafts. Naturally, these values would be sensitive to design variations, such as fillet radius, recessed or external, overlap of pin and journal, breadth to thickness ratio of web section, etc. However, it could well be that the value of 3·0 in bending used in the paper for a semi-built throw (for illustrative purposes) would be somewhat optimistic for certain designs, in which consequently special care might be needed to avoid extra bending stresses such as could arise from axial vibration, etc.

The author was grateful to Mr. Wilse for his contribution.

He would point out in connexion with Table VII that the overall average period of service given as 6·08 years did not apply to the average *endurance* of the shafts before failure, but to the average *total service* for all shafts during the ten-year period in question.

He agreed it would have been valuable to be able to differentiate between bending and torsional type failures but, unfortunately, such information was often lacking in the surveyor's report and in any case might often be difficult to establish with certainty.

The author did not believe that the kind of slow and slight changes commonly occurring in the torsional systems of propelling machinery would be likely to exert any significant effect on critical frequencies. Mr. Wilse's arithmetic was correct in his single-mass estimate of the effect on frequency of changes in propeller moment of inertia but in an actual case the approximation would have to be made on the 2-mass formula for natural frequency (c.p.m.) viz:

$$F = \frac{60}{2\pi}\sqrt{\frac{(J_e + J_p)C}{J_e J_p}}$$

where J_e = equivalent engine moment of inertia.
J_p = propeller moment of inertia.
C = equivalent torsional shaft stiffness between equivalent engine mass and propeller.

It would be found that on this basis the effect of slight changes in propeller inertia would be very much smaller and usually quite negligible.

Finally, the author wished to add his personal thanks to all those who had contributed to the discussion. The obviously keen interest shown in the subject of crankshaft design and service reliability had been impressively reflected in the weight and diversity of the many contributions, which was sufficient reward for the effort expended in the preparation of the paper.

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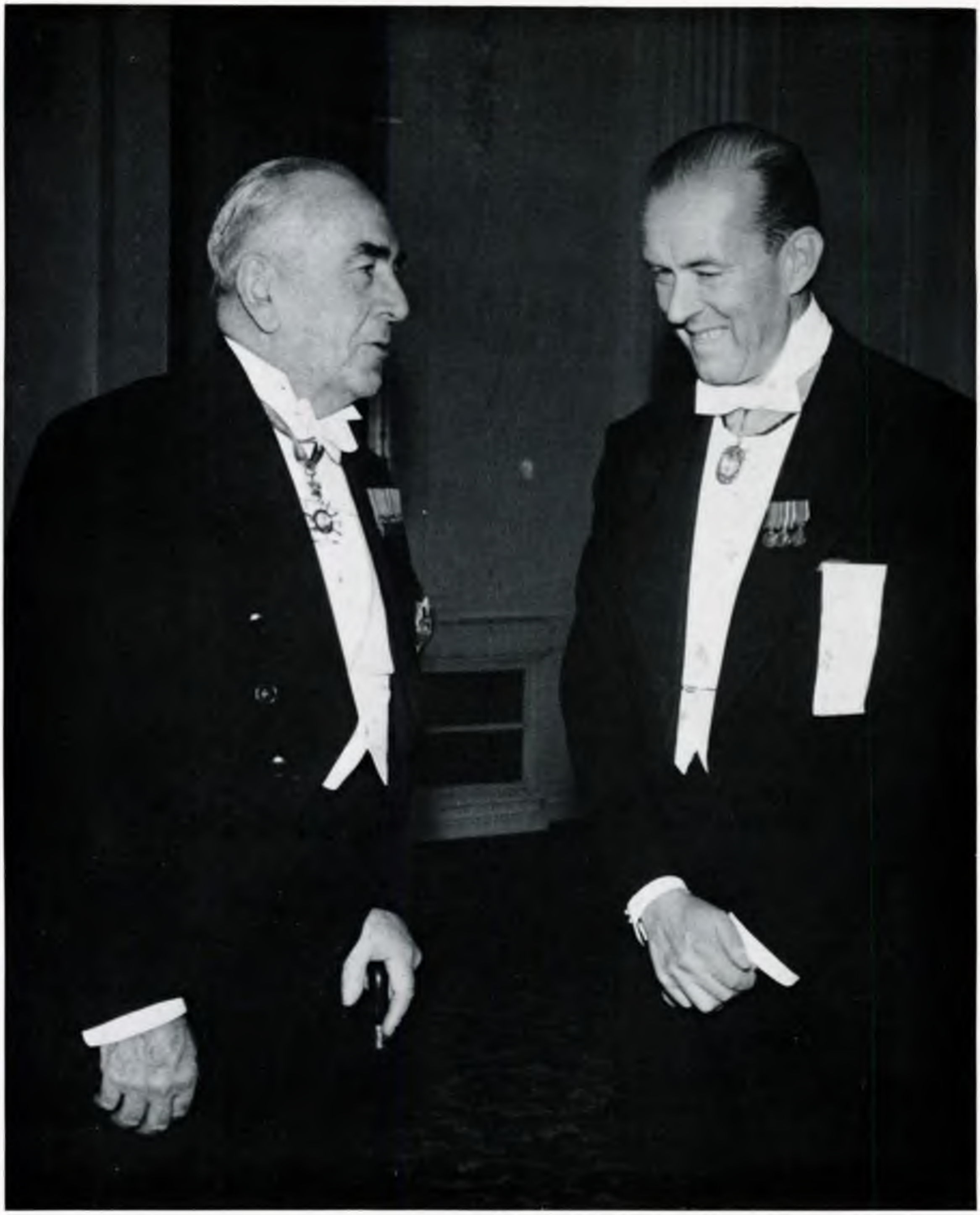
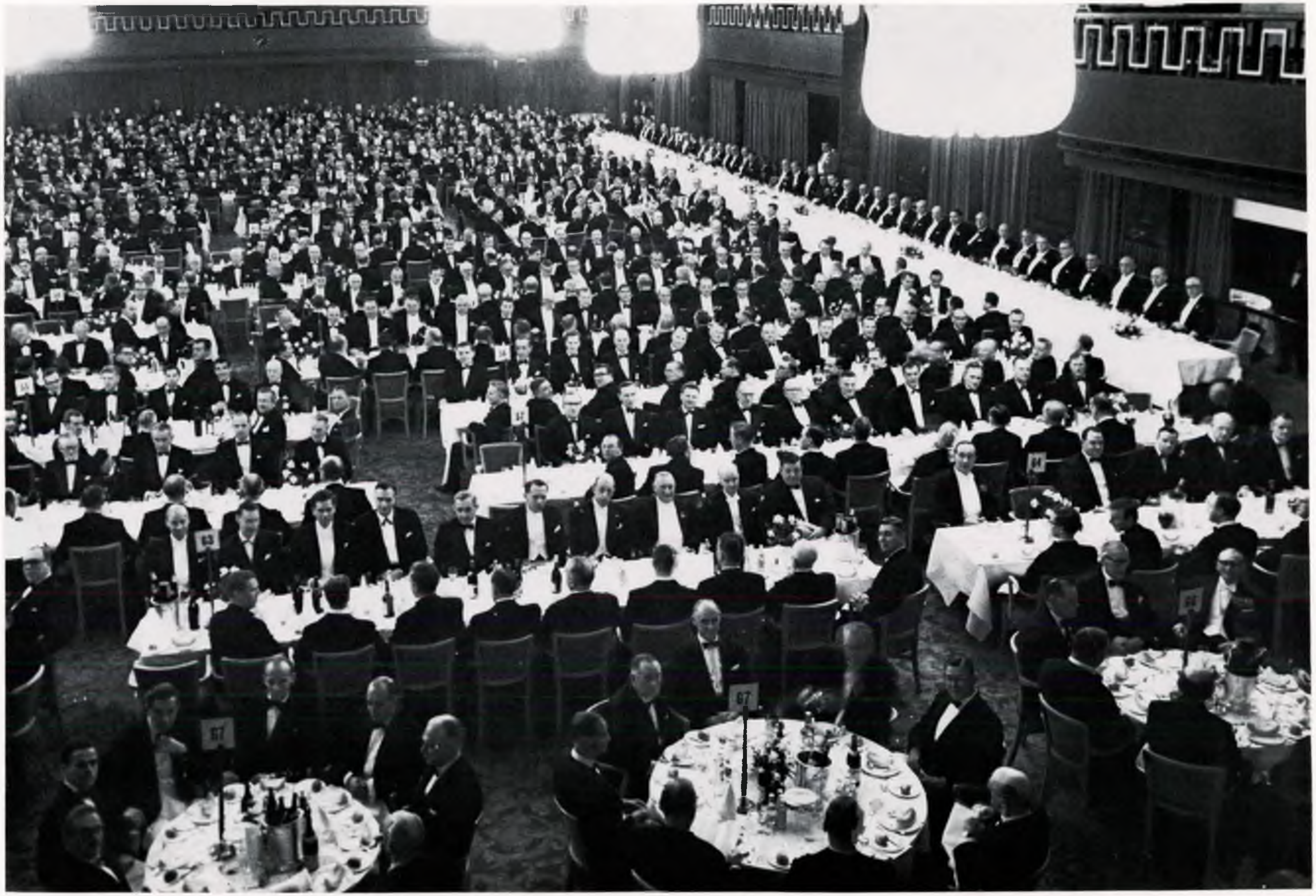


The President of the Institute, Sir Nicholas Cayzer, Bt. (right), with His Excellency The Right Honourable Sir Eric Harrison, K.C.M.G., K.C.V.O., The High Commissioner for Australia, at the Annual Dinner, held at Grosvenor House, Park Lane, London, W.1, on Friday, 20th March 1964



Annual Dinner 1964

Annual Dinner

The Sixty-first Annual Dinner of the Institute was held at Grosvenor House, Park Lane, London, W.1, on Friday, 20th March 1964 and was attended by 1,494 members and guests.

The President, Sir Nicholas Cayzer, Bt., was in the Chair.

The official guests included: His Excellency The Right Honourable Sir Eric Harrison, K.C.M.G., K.C.V.O., The High Commissioner for Australia; His Excellency Herr Doktor Hasso von Etzdorf, The German Ambassador; His Excellency Mr. Timothy Bazarrabusa, The High Commissioner for Uganda; His Excellency Dr. G. P. Malalesekera, The High Commissioner for Ceylon; His Excellency Dr. Carel de Wet, The South African Ambassador; Monsieur Johannes Tjaardstra, First Secretary (Commercial), representing His Excellency The Netherlands Ambassador; A. E. C. Drake, Esq., C.B.E., President, The Chamber of Shipping of the United Kingdom; Sir Victor Shepheard, K.C.B., Honorary Treasurer, The Royal Institution of Naval Architects; Sir Gordon Sutherland, Sc.D., LL.D., F.R.S., Director, The National Physical Laboratory; Sir Harold Roxbee Cox, D.Sc., Ph.D., Chairman, The Council for National Academic Awards; Sir Leslie W. Phillips, C.B.E., Chairman, The Baltic Exchange; C. C. Pounder, Esq., Past President; W. C. Agnew, Esq., C.V.O., Clerk to The Privy Council; I. E. King, Esq., C.B., C.B.E., R.C.N.C., Chairman of Council, The Royal Institution of Naval Architects; Commander F. M. Paskins, O.B.E., R.D., R.N.R., Chairman of Council; R. C. Chilver, Esq., C.B., Deputy Secretary, The Ministry of Transport; Professor A. S. T. Thomson, D.Sc., Ph.D., A.R.C.S.T., President, The Institution of Engineers and Shipbuilders in Scotland; The Reverend Maurice Dean, B.A., R.N.V.R., The Rector, St. Olave's, Hart Street, London, E.C.3; A. J. Marr, Esq., President, Shipbuilding Conference; R. B. Shepheard, Esq., C.B.E., B.Sc., Director, The Shipbuilding Conference; Dr. D. C. Martin, C.B.E., Executive Secretary, The Royal Society; A. Logan, Esq., O.B.E., President-elect; Dr. T. W. F. Brown, C.B.E., S.M., Director of Marine Engineering Research, British Ship Research Association; R. W. Sturge, Esq., Chairman, The Corporation of Lloyd's; Captain L. W. L. Argles, C.B.E., D.S.C., R.N., Captain Superintendent, H.M.S. *Worcester;* Commander H. E. Morison, D.S.C., R.D., R.N.R., Master, The Honourable Company of Master Mariners; H. N. G. Allen, Esq., M.A., Vice-President, The Institution of Mechanical Engineers; R. W. Bullmore, Esq., M.B.E., Assistant Secretary, The Ministry of Transport; A. W. Wood, Esq., Assistant Secretary, Ministry of Transport; Captain W. E. Heronemus, U.S.N., United States Assistant Naval Attaché; Commodore J. M. Ramsay, R.A.N.; Commodore M. K. Heble, I.N.; Captain S. F. Mercer, R.N.Z.N.; Commodore M. M. Hussain, P.N.; Stewart Hogg, Esq., O.B.E., Chairman, The Social Events Committee; C. H. Bradbury, Esq., President, The Diesel Engineers and Users Association; J. C. Duckworth, Esq., M.A., President, The Institute of Fuel; W. S. Douglas, Esq., President, The Institute of Refrigeration; G. H. R. Towers, Esq., J.P., President, The North East Coast Institution of Engineers and Shipbuilders; R. W. Reynolds-Davies, Esq., O.B.E., B.Sc., Secretary, The Institute of Fuel; L. A. Tiltman, Esq., Secretary, The Royal Institution of Naval Architects; S. E. Tomkins, Esq., O.B.E., Secretary, The Salvage Association; R. Munton, Esq., B.Sc., Denny Gold Medallist 1963; J. McNaught, Esq., Denny Gold Medallist 1963; J. N. Mackenzie, Esq., Denny Gold Medallist 1963; Dr. A. J. Johnson, B.Sc., A.C.G.I., Institute Silver Medallist 1963; W. McClimont, Esq., B.Sc., Institute Silver Medallist 1963; V. Wilkins, Esq., F.R.I.B.A.; R. Ward, Esq., F.R.I.B.A.; together with the Chairmen and Honorary Secretaries of the Sections.

The Loyal Toasts having been duly honoured, His Excellency Herr Doktor Hasso von Etzdorf (the German Ambassador) proposed the toast of "The Royal and Merchant Navies of the British Commonwealth".

He said: You have very kindly invited me tonight to be your guest at this Annual Dinner of the Institute of Marine Engineers. I have followed this invitation with great pleasure but, I am afraid, also with some anxiety. I considered it, of course, a great privilege to address this distinguished company and to propose such an important toast. On the other hand, I had to ask myself: Am I really the right person for this honourable task? After all, my professional experience with naval and marine affairs is the least justification for me to address you here tonight. But I can assure you that I have always had great admiration for the shipping world and for the role Britain has played in it. As a matter of fact, this began early in my life.

When I was a boy my parents lived in Berlin in a street by the name of Drakestrasse—a great name, of course, to you—and it was there that I got my first idea of Britain's historic role as a seafaring nation. Sir Francis Drake, who had given his name to the street (I believe it was the only street in Germany bearing this illustrious name) was not only the man who brought us the potato, though this in itself would have been sufficient to earn him eternal fame in Germany, for, as everybody knows, we are very fond of potatoes! The town fathers in Berlin chose his name as a symbol of one of the great sea powers of Europe, and honoured it in that way. So my imagination was much influenced by this great character, who, we are told, would not go out to sea to fight the Armada until he had finished his game of bowls.

I hope you will forgive me this short excursion into the past of your country and into my own. But even today, after I have learned a great deal more about Great Britain and her place in the world, there is hardly any other figure that I can more easily identify with the subject of my address tonight, the Royal and Merchant Navies of the British Commonwealth. (*Applause.*) Men like Sir Francis Drake were indeed the founders of the Empire and Commonwealth, the communications of which were and are provided by the merchant fleets and protected by the Royal Navy. This Commonwealth of yours is not only a manifestation of British vitality and strength towards the world; it is also an important element in the power and order of the whole West. It could be said of the British Commonwealth, as of certain other historic institutions, that if it had not existed it would have been necessary to invent it! (*Applause.*)

Today we talk about Europe. Who would deny that Europe without Great Britain is not the Europe we need? (*Hear, hear and applause.*) But we know, too, that the United

Kingdom can only become a member of this Greater Europe while at the same time retaining its bonds with the countries of the Commonwealth. (*Hear, hear.*) We recognize these two factors, and their reconciliation is one of the great tasks of European policy.

Common interest has brought the Merchant Navies of the Commonwealth and the German Merchant Marine closer together. On the one hand, we see our cargo ships and passenger liners engaged in keen competition throughout the world. Yet competition is essential to maintain and improve technical and commercial efficiency. On the other hand, we have important avenues of co-operation, I think, in the field of technical development and in our shipping policies. We in Germany benefit from the vast experience which the British Merchant Navy, as leaders in world shipping, has acquired throughout the centuries.

In this context I would like to point in particular to the international conference system, which developed by the United Kingdom, has been adopted by all nations traditionally engaged in merchant shipping. Only in this way has it been possible to have an internationally accepted common shipping policy among ourselves and with the newly emergent nations. It has also been responsible for our agreed policy of free and fair competition against the evils of protectionism and flag discrimination.

The maritime success of your country would not have been possible without the pioneering spirit and the great achievement of British shipbuilders and marine engineers. I am therefore particularly happy to find myself tonight among their most prominent representatives.

The Commonwealth, with its wide shipping links, would not have grown and flourished without the constant protection of the Royal Navies. It seems to me that the Royal Navies of the Commonwealth of today have a new but not less significant global task to perform. May I mention only NATO, SEATO and CENTO from among the defence pacts protecting the free world in which naval power plays a decisive part. Thanks to the Royal Navies, we in the free world do enjoy a feeling of security. (*Applause.*)

The German Navy has now joined the Royal Navy within our great defence alliance of NATO. This is a factor of unique historic significance. We all remember only too well that before 1914 Anglo-German relations were over-shadowed by the tragic rivalry of our navies. Instead of rivalry we now have co-operation. I would like to consider this co-operation as being one of the strong elements in our mutual relations, which, as your Prime Minister recently said, have not since the turn of the century been better than they are today. (*Applause.*)

I think we could hardly find better evidence of this collaboration than in the six frigates which we obtained from your Navy to use in our fleet, and the fast patrol boats which our Navy bought last year from a United Kingdom shipyard. Another venture in which our naval personnel is to be jointly involved is the experimental destroyer, which could well be the forerunner of a multi-lateral naval force within our NATO alliance. The excellent spirit of these naval relations has been manifest time and again when naval visits have taken place both here and in Germany during recent years. The sentiments of comradeship shown to our officers and men by the Royal Navy are wholeheartedly appreciated in my country.

In all our common efforts for greater interdependence within the framework of the alliance you gentlemen from the Institute of Marine Engineers have played a significant part, the importance of which remains just as great today. I understand that you commemorate this year, the 75th Anniversary of the Foundation of your Institute, and to congratulate you on the achievements of the Institute throughout that period must be the desire of whoever has shipping at heart. (*Applause.*) To be given the opportunity to do this here tonight, as the German Ambassador in London, is a great honour to me. With my best wishes for the Institute I should like to couple the toast to the great might and tradition which has formed the background for the development of the Institute of Marine Engineers, "The Royal and Merchant Navies of the British Commonwealth." (*Applause.*)

A. E. C. Drake, Esq., C.B.E. (President of the Chamber of Shipping of the United Kingdom) replied, also proposing the toast of "The Institute of Marine Engineers".

He said: I am very happy indeed to be with you tonight and to be entrusted with the dual role of responding to the toast of the "Royal and Merchant Navies of the British Commonwealth" so ably given by His Excellency the German Ambassador, and also to propose the toast of "The Institute of Marine Engineers". I know you would like to join with me in congratulating His Excellency on the magnificent way in which he proposed that toast. (*Applause.*) It is not often that a shipowner has the chance and the honour to reply on behalf of the Royal and the Merchant Navies together. Tonight, to have had a person of the calibre of Herr Doktor Hasso von Etzdorf to propose this toast is a tribute, not only to the Institute but also to shipping generally. (*Hear, hear.*) I feel as if I ought at least to be wearing some kind of ceremonial sword or some other outward token of authority in deference to our gallant friends of the Royal Navy, but your invitation was quite explicit about the dress to be worn, and I did not after all have to call upon the services of that well known Covent Garden store .(*Laughter.*)

This, I believe, is one of the few occasions where a toast so happily links the two great sea services, but, of course, it is a very logical thing, because they are very closely linked by bonds of common heritage and of mutual respect. This close association in the defence and the service of our country is part of the fabric of Britain's history. A few centuries ago, indeed, it was difficult to tell where the Royal Navy ended and where the Merchant Navy began! The ships of the old East India Company and others of that period were run on strictly naval lines and they carried, as indeed did most other ships of that period, almost as many supernumeraries as seamen, whose job it was to fight off the murderous pirates who infested parts of the ocean. Nowadays the kind of people who make things awkward and difficult for the shipowner, and thus for the seamen, do not brandish cutlasses or run down a ship before boarding her with blood curdling yells. Usually they appear before us like ethereal spirits dressed as politicians, and sometimes even as lawyers. They are always well-heeled, well-educated, and anything but well-meaning. (*Laughter.*)

Your Excellency, I should like to claim a close relationship with that famous man who brought your potato and after whom your Berlin street was named. Alas; I cannot be sure about this, and perhaps as he officially at any rate had no children (*Laughter*) the less I say about it the better! (*Laughter*). Whether as shipowners, naval men, marine engineers or politicians, we are all in some degree affected by the challenge of this atomic age whose benefits can enrich life and whose misuse can overshadow or destroy it. Recently a newspaper in British Guiana, offering its readers guidance and help, as newspapers are rather apt to do, gave this advice: "In the event of nuclear attack, run like hell." (*Laughter.*)

Whatever benefits or dangers are to emanate from the nuclear age, the respective roles of the Royal and Merchant Navies must remain impregnably vital to Britain's well-being, and it is surely unthinkable that any defence or strategic decisions of the future should fail to take proper account of their irreplaceable functions. In these functions the two Navies have continued to specialize, to equip themselves still better for their respective roles. The Royal Navy, the fighting unit, has done this with its nuclear submarines, its commando carriers, and so on. The merchant service has been ordering more specialized types of ships like tankers, bulk carriers and refrigerated cargo liners, to meet trade requirements. As my shipping interests are on the tanker side of the business I was very interested to learn that the tanker element is soon going to be even more strongly represented in the inner councils of the Institute.

When my good friend Sir Nicholas Cayzer ends, what I am certain will have been a most successful term of office as your President, he is to be succeeded by Alec Logan. (*Hear, hear and applause.*) As you all know, Alec Logan, until his recent retirement, was extremely prominent in the tanker world. I'm sure that in the history of that strange breed, the tanker animal, it will then be the first time that they have produced a President of the Chamber and the Institute at the same time. (*Applause.*)

I would like to talk now about those things which I regard as important to the British shipping industry today, faced as it is by so many fascinating changes, internally and in the climate in which it operates.

World shipping has been suffering from nearly seven years of unremitting gloom, but that British shipping is weathering the storm in good heart and strong spirit is testimony, I suggest, to the fact that as shipowners we at least know our business. (*Hear, hear.*) But is it enough these days simply to know our business? Is there nothing more that we can do and therefore must do to meet the challenge that the future holds forth? More and more people in British shipping are asking themselves these questions. The reason is that it is becoming increasingly clear that the protection being afforded now to so many of our competitors could be a feature of world shipping for quite some time to come, so that we must make an extraordinary effort if we are to hold our own, let alone go further ahead. We shall have to strive to the very utmost for peak efficiency in every aspect of our work, and we shall have to ensure, too, that the tools of our trade, our ships and their machinery and equipment, reflect all the latest ideas insofar as we can afford them. More than that, the effort must be made, certainly by all levels of management, but ideally by everyone in shipping, whatever their job, to look beyond their immediate horizon. It is in the growing realization of this that today the emphasis in British shipping is on the need to probe, question and explore. In Great Britain we pride ourselves on our engineering skill, and not without good reason. On the marine side nuclear propulsion remains an exciting prospect. Our physicists have not yet produced a reactor which would give us an alternative means of power which is commercially viable; but neither, so far as I am aware, have the physicists of any other country. Of the power plants that have to "pay for themselves" at sea we have produced a good share, and we can claim credit for many of the technical advances that have been made in marine engineering. Even so, can we be satisfied —are you all personally satisfied—that everything possible is being done to take full advantage of the most up to date technology? Technology implies a constant flow of new thought and ideas and what is new today may well be dated in the light of tomorrow's knowledge. So the urge to explore must be ceaseless.

Tonight's excellent dinner reminds me of a remark by that great American inventor, Thomas Edison. He said: "The stomach is the only part of man which can be fully satisfied." (*Applause.*) He went on: "The yearning of man's brain for new knowledge and experience and for more pleasant and comfortable surroundings never can be completely met. It is an appetite which cannot be appeased". I believe it to be of paramount importance that everyone connected with British shipping is imbued with the spirit of this technological age. I wholeheartedly applaud, therefore, the recent action of your Council in setting aside a substantial sum each year to promote higher technical education. This suggests that it may well be necessary to consider the whole manning and staffing arrangements for the operation of merchant ships arising from higher horsepowers, automation and the remote control of ship's machinery. Now I do hope I have conveyed to you the message that the shipowning side of the British maritime industries is resolved to keep fully abreast of the times. (*Hear, hear.*)

I have the greatest pleasure in proposing the toast to "The Institute of Marine Engineers", which so rightly enjoys an international influence and respect, and I couple it with the name of Sir Nicholas Cayzer, your distinguished President, who added much lustre to his splendid reputation as a virile, go-ahead leader of the British shipping industry during a notable year as President of the Chamber of Shipping. (*Applause.*)

The PRESIDENT, Sir Nicholas Cayzer, Bt., replied.

He said: If a Rip Van Winkle had fallen asleep in about the year 1900 and had awoken in this year of grace, I think he would have been totally bewildered at what had happened while he slumbered; for in the past sixty years change has been more radical and more violent than ever before, I think, in this world's history. The nature of this change has been very largely scientific and technical. For nearly a century *Pax Britannica* held sway in the world, but two catastrophic wars have changed all this, and the cost of modern weapons of war has altered the position of Great Britain in relation to the other world powers. I am inclined to think that this was probably inevitable and in fact returns us to an historical role that we played, not without success, in the past, that is, of forming alliances. I believe that the "go it alone" mentality is doomed to failure, but what really is worrying me is our industrial side. We are not, I think, keeping up. Our national productivity is backward compared to many other countries, and to a large extent since the war ended we have drifted along. Our declining share of world trade is most ominous. As a nation we hate facing facts until they are thrust upon us, but we are capable of great things when we have grasped the nettle. I think, however, it is very necessary that we should understand what our problem and purpose is in the world today.

Napoleon called us a nation of shopkeepers, and that is what I think we should be and in fact must be if we are to survive. Our population is large, our natural resources are small. We must sell to the world if we are going to live, let alone hold our place in the world. If we fail industrially we shall fail in every other way. A great deal of our industrial and financial machinery, however, is archaic. Both private industry and trade unions could do much more in raising productivity if they could forget the past and re-assess their roles. It is our failure to make the essential changes in this field that more than anything else has allowed Japan and other shipbuilding countries to get ahead of us in the matter of cost. An efficient end product or service at the right price is vital to our survival, but too much time is wasted on manœuvre and argument, to the great advantage of our competitors. (*Applause.*)

Both unions and employers in Great Britain are staunch conservatives. They do not like change and are very suspicious of it. I know that there is an awful lot of the past in these attitudes, of grievances, real or imagined, of cherished beliefs, but this little island will sink in the wake of its past dissensions if we are not careful. I quite understand and sympathize with some of the trade unions' aims, but they will only be achieved through industrial efficiency.

The man in the street—and, for that matter, the woman too—is heartily sick of indecision and lack of a clear lead, and too many good brains have given up in despair and departed. Let us face it: there are even those in our midst who would like us to fail.

The great problem, therefore, for this country is to find a way by which all work together to a common end, and that end must be to keep Great Britain in the van of progress. This is an urgent and not just an academic problem.

Here at home I believe that both shipowner and shipbuilder have changed in their thinking in the last few years and have grasped the fact that in both their industries they must co-operate with each other in regard to broad policy, although there is still plenty of room for individuality, but individuality must be subservient to the success of the whole. Research and training, as Mr. Drake has said, is of prime importance, and co-operation between shipowner and shipbuilder in research is developing promisingly. Higher education, more technical and scientific training, are now realized to be a "must". The younger generation that is growing up is fine material but it must be properly led and rightly inspired. The Robbins Report, with its recommendations, is only just in time. The follow-through must be concentrated on in its various fields, and certainly in the engineering field higher qualifications will be demanded.

The Institute of Marine Engineers has a vital part to play. It now has a membership of 17,000 spread throughout the world, and this Institute can only flourish if we are business-like and farseeing in our actions and thought and are not afraid of change.

Mr. Drake also mentioned the sum that has been laid aside by the Institute to finance scholarships, to develop knowledge and technology of engineering, and in this way help to lay the foundation stone of higher education for the marine engineer of the future.

This is the beginning. Increasing horsepower, automation, remote control of ships, will, I think, make it necessary to reconsider the staff requirements of future ships, and we must try to project our thoughts at least ten years ahead. Indeed, the whole question of training and education of marine engineers is a matter of the first importance. (*Hear, hear.*) This is a problem which has engaged the attention of both the Government and the shipping industry, and is one about which the Institute is thinking hard. I think that we should all co-operate together in deciding the right course for the future.

I did mention in my Presidential Address the desirability of potential engineering officers being trained not only in their profession but as leaders. A start has been made in *Conway* in training engineering officers and deck officers together, and I warmly welcome this departure. The secret of the public school, so much decried today, is that, having worked and suffered together, a comradeship is carried into life that is never entirely forgotten. We need to cultivate a sense of responsibility and a feeling of pride in our young merchant navy officers, whether they serve on deck or in the engine room. (*Applause.*)

A new development is the formation of the Engineering Institutions Joint Council, and we are happy to be one of the thirteen founder members. Decisions have been taken by the Council of the Institute before discussion with members or even the local sections could be arranged, and it is hoped that members will have confidence in the actions that have been taken. As your President for this year I feel sure that this is a move in the right direction. It means wider consultation, which must be of benefit to the Institute. A Royal Charter is being sought and it is hoped that this will be granted by the end of the present year.

The Chairman of the Council of the Institute, Commander Paskins, and our Secretary have just completed a very successful world tour, when personal contact was made with many members throughout Canada, the United States, New Zealand and Australia, and I myself was privileged to meet some of your members in South Africa recently. It is most heartening to see the enthusiasm and interest there is in all overseas sections.

I should like to take this opportunity of congratulating Admiral Sir Frank Mason, the immediate past-Chairman of the Council, on becoming President-Elect of the Institution of Mechanical Engineers. (*Applause.*) We wish him and Lady Mason a very successful and happy year of office.

I have been especially pleased to note during my year of office the important part that officers of the Royal Navy play in the affairs of our Institute. They bring a wealth of experience to our councils, and co-operation with the Royal Navy is of the very greatest value.

I should like also to congratulate Mr. Logan, who, if it is your will, will be my successor. (*Hear, hear and applause.*)

On your behalf may I warmly thank the Chairman of the Council, Commander Paskins, and his Council for the time they have given to the affairs of the Institute during the past year. I have been conscious of their desire to see that, in a changing and challenging world, all that can be done to keep the Institute in the van of progress shall be done. They have been most able and energetically supported and backed up by Mr. Stuart Robinson and the staff of the Institute, and we are greatly indebted to them. (*Applause.*)

Finally, may I say how very privileged I have felt in being your President during the past year. I wish you the success that will surely come if you are ever mindful of our great sea heritage and the need to keep abreast, if not ahead, of the times. As I have said, Great Britain's position is very different from what it was at the turn of the century, yet I am convinced that she has a great contribution still to make to the world.

Our ability to achieve this aim will depend on our realism, our ability to think clearly, and, above all, our determination to work together in industry. (*Applause.*)

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 12th November 1963

An Ordinary Meeting was held by the Institute on Tuesday, 12th November 1963, when a paper entitled "Some Factors Influencing the Life of Marine Crankshafts" by S. Archer, M.Sc., M.I.Mech.E., M.R.I.N.A., M.N.E.C.I. (Member), was presented by the author and discussed.

Mr. W. Young, C.B.E. (Vice-Chairman of Council) was in the Chair and one hundred and forty members and visitors attended the meeting.

Eleven speakers took part in the discussion which followed.

The Chairman proposed a vote of thanks to the author which was greeted by prolonged and enthusiastic acclamation.

The meeting ended at 8.25 p.m.

Section Meetings

Auckland

A meeting of the recently formed Auckland Section was held on Monday, 25th February 1964, at the Ellen Melville Hall, High Street, Auckland, at 7.10 p.m.

Mr. H. W. Whittaker (Chairman of the Section) was in the Chair and twenty-five members were present. The main business of the meeting was to discuss the programme for the following year and to report the financial arrangements approved by Head Office.

The meeting was followed by a lecture on "Recent Developments in Hovercraft Design" by H. L. Mirams. Mr. Mirams illustrated his paper with working models, films and diagrams. The lecture was most informative and was very well received. Among the thirty-five guests present the Section were honoured by the presence of Mr. V. G. Boivin, M.I.Mech.E., M.R.I.N.A., F.N.Z.I.E., an Honorary Corresponding Member in New Zealand of the Royal Institution of Naval Architects, formerly Chief Surveyor of Ships for the New Zealand Marine Department.

The meeting terminated at 11.00 p.m.

Bombay

Annual General Meeting

The Annual General Meeting of the Section was held on Wednesday, 26th February 1964, at the Nautical and Engineering College, at 5.00 p.m. Chairman of the Section, Mr. B. S. Sood (Local Vice-President), was in the Chair and twenty-four members were present.

Messrs. R. C. Rohan and B. Ananda volunteered to act as scrutineers for the election of committee members and were appointed by a unanimous vote.

Mr. D. Dyer (Honorary Secretary of the Section) moved, and it was agreed, that the Annual Report of the Committee, which had been previously circulated, should be taken as read.

Mr. R. S. Rawal felt that discrimination should be exercised as to what constituted a paper. In the past many manuscripts, which were little more than lectures, had slipped through with a doubtful effect on the prestige of the Institute. Mr. A. T. Joseph said that sheer "book work" should not be called a paper. An author should have actual experience in his subject. Mr. R. McIntosh suggested that more symposiums be held. The Chairman said that the views expressed would be considered by the Committee and passed on to the Sub-committee, which would be constituted to draft rules for the acceptance of papers. The report was unanimously adopted.

The Honorary Treasurer, Mr. C. S. Sundaram, presented the financial statement for 1963 and explained the significance of the various items. The meeting unanimously adopted the statement.

The scrutineers submitted the result of the election for the Section Committee and the following were duly declared elected: Messrs. S. A. Samson, M. K. Jagtianie and J. M. Trindade. The chairman pointed out that subsequent to the issuing of the ballot paper, Mr. Trindade had been transferred away from Bombay. Should he wish to withdraw, the nominee with the next highest number of votes, Cdr. V. S. P. Mudaliar, I.N., would be considered as elected. As no nominations had been received for the offices of Honorary Secretary and Honorary Treasurer, Messrs. D. Dyer and C. S. Sundaram respectively, would be considered as elected for a further term.

The Committee for 1964 is as follows:

Local Vice-President, Bombay: B. S. Sood
Chairman: S. A. Samson
Committee: B. Ananda
E. R. Dastoor
M. K. Jagtianie
Cdr. V. S. P. Mudaliar, I.N.
A. N. Mukherjee
K. Parthasarathy
S. Ratra
R. S. Rawal
Honorary Secretary: D. Dyer
Honorary Treasurer: C. S. Sundaram

In his address, the Chairman of the Section, Mr. B. S. Sood, said that in spite of the brevity of the report, it was clear that the Section had been active during 1963. He wished, therefore, to record his appreciation to his colleagues, particularly those who were retiring, for their co-operation and assistance. He was conscious of the honour accorded him in his new appointment as Local Vice-President for Bombay.

He had attended meetings of the Institution of Engineers (India), Bombay and, although they had a much larger membership, he could not help noting that the meetings of the Bombay Section of the Institute of Marine Engineers were much better attended.

He and several other members had attended the Shipping and Shipbuilding Conference in Calcutta. Both the technical sessions and social functions were well supported and the marine companies represented there must have benefited immensely, particularly due to the presence of experts from all over the world. The arrangements were magnificent, but it had to be conceded that Calcutta had an edge on Bombay due to the absence of a "dry policy".

Insufficient publicity was being given to marine engineering in India, in fact some people in Northern India did not even know what a marine engineer was. Perhaps now, with marine engineers finding their way into many other industries, further information on their capabilities would inspire a greater demand for their services.

Mr. R. S. Rawal, commenting on the Chairman's address, said that he too felt that marine engineers should be more conscious of publicity. Whereas several boards and committees had been formed by the Government for various marine engineering purposes—even for dealing with the training of marine engineers—the Institute had no representation on them. He suggested that the Government should be approached on this matter.

Mr. R. C. Mohan agreed with Mr. Rawal and said that even in the case of the manufacturing of Diesel engines, no properly constituted engineering society was represented.

Mr. T. M. Sanghavi, B.E. (Secretary, Indian Division) said that the Indian Division had approached the Government, but the reply was not encouraging.

Mr. R. McIntosh wondered whether a purely technical body should involve itself in these matters.

Mr. Rawal said that, unfortunately, on these committees, proper representation was not accorded to the views of marine engineers and, as a result, some alarmingly false notions were propagated.

It was decided that these matters affected the other Sections of the Institute too, and since a meeting of the Indian Division Committee was to be held shortly, they would be considered.

Mr. Mohan said that a very good topical subject for a symposium would be the reasons for marine engineers leaving the sea. Mr. E. J. D'Sa suggested that the title be "Talking Ships from Ashore". Mr. Ananda believed that the project would just be so much more talk. Mr. P. D'Abreo said that if marine engineers preferred to come ashore, nothing should be allowed to prevent them. Mr. Ratra believed that, in a few cases, they should, in fact, be encouraged. Mr. Mohan was convinced that in any case a lot of interesting views would see the light of day.

The Honorary Secretary was asked to make the necessary arrangements for holding a symposium on this topic.

Mr. A. T. Joseph said that he was sure that all present would agree that the Committee for 1963 had performed excellent work and that he would like to propose a vote of thanks. Mr. McIntosh seconded the motion and all present felt that the appreciation was well deserved.

Mr. Mohan proposed a vote of thanks to the Chair and said that while the attendance at this Annual General Meeting was better than hitherto, it would have been better if more members were present. He suggested that in future the senior members should take the initiative and persuade their subordinates to attend. Mr. Rawal seconded the motion, which was carried with acclamation.

The meeting closed at 7.15 p.m.

North East Coast

A general meeting of the Section was held on Thursday, 12th March 1964, at the University of Newcastle upon Tyne, Stephenson Building, Newcastle upon Tyne, at 6.15 p.m.

Mr. G. Yellowley (Chairman of the Section) presided at the meeting which was attended by eighty-two members and guests.

The Chairman introduced the speaker, Mr. D. Bradley, B.Sc., M.Sc., who presented his paper "Fuel Preparation and Lubricating Oil Treatment in the Automated or Unattended Engine Room".

In his presentation of the paper, which took just over an hour, Mr. Bradley covered the definitions, the reasons for automation, the economics and the degree to which they have been carried out.

The paper which was practical rather than theoretical, was particularly well presented and was closely followed by those present. Amongst those who contributed to the discussion which followed were: Professor G. H. Chambers, D.S.C. (Member of Council), Mr. J. F. Butler, M.A. (Member of Committee), Mr. E. C. Cowper (Local Vice-President, Newcastle upon Tyne), and Mr. W. H. Menzies.

The meeting closed after a vote of thanks to the author, by the Chairman.

North Midlands

A meeting of the Section was held on Wednesday, 26th February 1964, at the College of Art, Green Lane, Derby, at 7.15 p.m. Mr. J. W. Batey (Chairman of the Section) was in the Chair and some eighty-five members and visitors attended the meeting.

A paper entitled "The Design and Construction of the Trawsfynydd Nuclear Power Station" was ably presented by Mr. J. N. Bishop, B.Sc., M.I.Mech.E., M.I.E.E., who commenced his talk by showing the location, referring to its remoteness and the difficulties that had to be overcome in order that heavy items of equipment could be taken to the site.

A description of the various components of the reactor, together with detailed slides showing the heat exchangers, CO2 circuit, the complicated system of biological shields and the method adopted to renew the reactor fuel, followed.

The generators were briefly mentioned, together with the associated condensers, cooling water for which is drawn from Lake Trawsfynydd and which, by a series of baffles, ensures that the discharged water circuits the lake (some 5½ miles) before reaching the circulating pump inlets.

Mr. Bishop showed specimens of the carbon blocks used in the reactor and a container used for holding the uranium charge, which was of great interest to his audience.

A film was shown which illustrated the work in progress on the site during 1961-62 and brought the magnitude of the work into perspective and showed some of the heavier pieces of plant, weighing up to 425 tons, being lifted and placed in position.

A discussion lasting almost an hour followed, during which Mr. Bishop's apt and prompt answers showed his profound knowledge of his subject and at 9.00 p.m. the Chairman reluctantly had to bring the question time to a close.

A vote of thanks to Mr. Bishop, proposed by Mr. A. M. Jarvis, B.Sc., was heartily endorsed by those present.

Singapore

Annual Report

Active membership at the end of 1963 totalled 68 corporate members (i.e. Members and Associate Members) and 44 Students, making a grand total of 112. It will be noted that while the number of corporate members has remained static as compared with the end of 1962, there has been a substantial increase in the student membership.

During the year various invitations were received from the Joint Group to participate in visits arranged by that Group, including an invitation to attend the Joint Group Annual Dinner, a visit to the Johore Bahru Power Station, and to a lecture entitled "British Nuclear Power Programme" in the Shell Theatrette. Another visit of note was to the new fast cargo liner m.v. *Flintshire,* which was much enjoyed. Finally an invitation was extended to members to attend a demonstration given at the S.H.B. in connexion with the use of L.P.G. gas with oxygen for welding, etc.

A prize valued at $50 was awarded to a student selected by the board of the Singapore Polytechnic. The prize selected was engineering manuals.

J. O'B. Canavan (*Chairman*)
J. Snadden (*Honorary Secretary*)

Annual General Meeting

The Annual General Meeting of the Section was held recently. Of the sixty-eight nomination papers sent out, only three were returned and as these nominations did not exceed the number of vacancies on the Committee, a ballot was not held.

The Committee for 1964 is therefore constituted as follows:

Local Vice-President: S. A. Anderson, O.B.E.
Chairman: L. K. Wong
Vice-Chairman: J. A. Boater

Committee: J. McA. Brown
J. O'B. Canavan (co-opted)
E. Daniels
N. Gartland
B. B. Girling
Honorary Secretary: J. Snadden
Honorary Treasurer: J. McC. Mair

South East England

Senior Meeting

A senior meeting of the Section was held on Tuesday, 17th March 1964, at the Clarendon Royal Hotel, Gravesend, at 7.30 p.m.

Mr. G. F. Forsdike (Chairman of the Section), was in the Chair and welcomed Mr. J. H. Milton (Member of Council) who presented his paper entitled "Marine Water Tube Boiler Surveys" to some fifty members and visitors present.

The paper, illustrated by slides, was received with keen interest and affirmed that Mr. Milton was to be considered an authority on this subject.

In the discussion which followed many pertinent questions were asked and a forthright answer was given in each case.

The meeting closed at 9.20 p.m.

Junior Meeting

A junior meeting of the Section was held on Wednesday, 18th March 1964, at the Medway College of Technology, at 7.00 p.m. when a paper entitled "The Launching of Ships" was presented by the author, Mr. R. S. Hogg.

The meeting was arranged with the co-operation of Dr. H. R. Orr, B.Sc., Principal of the College, and Mr. F. B. Tucker, Senior Lecturer.

Mr. J. A. Andrew (Member of Committee) was in the Chair and introduced the speaker to the fifty-five members and visitors present.

Mr. Hogg displayed a complete command of his subject and gave a step by step literary and pictorial description of the techniques necessary to ensure a safe and successful launching.

The meeting was subsequently opened to discussion and the author was able to answer, and elaborate upon, the questions asked, and thus satisfy the technical appetite of those present.

A vote of thanks to Mr. Hogg was proposed by the Chairman, and was acclaimed by the meeting.

The meeting closed at 8.50 p.m.

Victoria

Annual Report

The activities of the Victoria Section held during 1963 were as follows:

8th February	Annual General Meeting at the Royal Society Building, Melbourne.
10th May	A lecture entitled "Development of Mirrlees Engines" was delivered by Mr. H. B. M. Vose of Hawker Siddeley Australia Pty. Ltd.
25th July	A visit by members of the Section to the Commonwealth Engine Works, Port Melbourne, to inspect the Sulzer engine.
9th August	A lecture entitled "For Mechanical Advantage" was delivered by Mr. D. Bartaby of W. H. Allen Sons and Co. Ltd.
22nd August	The Apprentices' Night was held at the Radio School Theatre of the Royal Melbourne Institute of Technology.
18th October	The Seventh Annual Dinner was held in the University Union Private Dining Room; the guest speaker was the Honourable H. R. Petty, M.L.A., Minister of Public Works.
15th November	A buffet dinner was held at Berkeley Hall, 11 Princes Street, St. Kilda, to welcome the visitors from London, Commander F. M. Paskins, O.B.E., R.D., R.N.R. (Chairman of Council) and Mr. J. Stuart Robinson, M.A. (Secretary of the Institute). After the dinner the meeting was addressed by the visitors from London.
16th November	A picnic to Maroondah Dam was arranged for Commander and Mrs. Paskins, and Mr. and Mrs. J. Stuart Robinson, followed by a visit to Healesville to the Colin McKenzie Sanctuary.

Annual General Meeting

The Annual General Meeting of the Section was held on Friday, 14th February 1964, at the Royal Society Building, Melbourne.

The Committee for 1964 is constituted as follows:

Local Vice-President and Chairman: A. J. Edwards
Committee: P. Bossen
V. F. Harris
J. E. North
G. Seales
J. B. Thomson
Lt. Cdr. D. W. K. Vagg, R.A.N.
Honorary Secretary: K. Paxton
Honorary Treasurer: Lt. Cdr. J. H. Coles, R.A.N.V.R.

West of England

A general meeting of the Section was held on Monday, 9th March 1964, at the Small Engineering Lecture Theatre, Queen's Buildings, University of Bristol, when a paper entitled "Hovercraft" was read by the author, Mr. G. C. Keen.

Captain A. C. W. Wilson, R.N. (Chairman of the Section) presided at the meeting, which was attended by thirty-three members and guests.

Mr. Keen gave his lecture with the aid of slides and a thirty-five minute film. He began by giving a history of the hovercraft from its inception, and outlined the fields in which it might play its greatest part, stating that the greatest application lay in the marine sphere, particularly over areas of land and sea where conventional craft could not operate.

Three fundamental types of air cushions were explained; the power requirements for these air cushions and for the propulsion of the craft, were also discussed.

As in all new developments, problems arose which had to be overcome, and one of these was the ingress of dust, debris, sand, sea water, etc., to the fans. The second problem was that of difficulty in travelling over rough seas and high waves. This had been overcome to a large extent by the fitting of a flexible rubber skirt around the periphery of the craft, so reducing the impact loads on the structure and smoothing out the ride.

This fitting of a flexible rubber skirt had been the greatest development, in hovercraft, yet.

The lecture ended with the film of a hovercraft operating a scheduled fare-paying passenger service—the first of its kind in this country.

The many questions put to Mr. Keen were admirably answered, and a vote of thanks on behalf of the members was proposed by the Chairman.

The meeting closed at 9.30 p.m.

Election of Members

*Elected on 10th February 1964**

MEMBERS

Philip Sayer Armstrong
George Samuel Boffey
Nigel Caffyn
Richard Ellis
Giovanni Giuliana, Dott. Ing., Lt. Col. Engineer, Italian Navy
Theodore Iatropoulos
Vladimir J. Lebedev, Capt., U.S.S.R.N.
Donald MacInnes
Ronald Ewen Mackinnon

* It is regretted that the February Election of Members was published incorrectly in the March issue of the TRANSACTIONS and therefore appears again here.

David Anderson Nicol Martin
Earl Matthews
Donald Stephen Townend, B.Sc. (Eng.), London
Prof. Dr. Ir. W. P. A. Van Lammeren
Ronald Watson

ASSOCIATE MEMBERS
Leonard Murray Short Bell
Charles Kenneth Bragg
Raymond Neil Callaby
John William Carlisle
Peter Davies-Carr
Michael Ferryman
James Henry Gerrard
Joseph Gillan, B.Sc. (Belfast)
Andrew Alexander Hope
William Lamb
James Lyness
William James McAnally
Allan Wilson McAra, Lt. Cdr., R.N.
Stanley John Marchant
James Brian Mason
Donald Alexander Matheson
Daniel Fairbairn Matthews
Peter William Benjamin Stoodley
Robert Laslett Thomas, B.Sc. (Glasgow)

ASSOCIATES
Reginald Sydney Abbott
Charles Belli
James Henry Emmerson
Md. Abdus Samad
Peter Ash Vie

GRADUATES
Geoffrey Francis Dart
Derek Clement Davies
David Emerson
George Douglas Hornsby
Gulab Rijhwani
Hira Lal Sharma
William Alan Stewart

STUDENTS
Howard Barnes
Alan Nicholas Campion
Bruce Michael Faulder
Edward Bruce Gibson
Brian John Hall
Michael Anthony Jackson
Brian Thomas Kevin
Michael Constantine Lainas
David George Malton
Sarosh Homi Marker
Michael Hugh Mateer
Brendon Michael Pickthall
Francis Edward Pleavin
Peter Priestnall
Martin John Reeves
John Stephen Riley
Peter Derek Stanworth
Brian Norman Stonely
Roger John Timms
Norman Travis
Robert Hubert Vart

PROBATIONER STUDENTS
Roland Aylwin Bainbridge
Thomas Joseph Bampton
Peter Ross Noel Barrar
Michael Anthony Brown
Anthony Victor Chivers
John Robert Clark
Terence William Davies
Edwin Stuart Garrett
Kenneth Graham
Nicholas John Harris
Anthony George Hines
Jeremy Victor Hockin
G. Hopkins
Ian Melvin Horrocks
Meirion Hughes
Anthony Jackson
Paul Jessey
Colin Charles Knill
Ross Richard Larsen
Alan Curtis Littlewood
Irvine Oswald Long
John Maling
Christopher John Matlock
Peter Thomas Mitchell
Christopher John Morton
Douglas Mackenzie Nettleton
George Edward Nicholson
Ian Clarence Pickering
Nicholas Michael Pope
David Warwick Preston
Simon Mellersh Rendall
Anthony Reid Russell
Stephen John Sherring
Anthony Charles Smith
Dennis Richard Michael Smith
James William Smith
John Probyn Sproat
Barry Patrick Sullivan
Christopher Paul Tamblin
Brian William Wright Taylor
John Tomlinson
Daniel Turner
Terry Grant Wise

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER
Robert William Anderson
Derek Charles Patrick Crowe
William Francis Dowie
Frank Hipson

TRANSFERRED FROM ASSOCIATE TO MEMBER
Douglas Royston Matthews

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER
John Gabriel Creen
Robert Howe
Desmond William John Phillimore
Davinder Nath Sabharwal, Lieut. (E), I.N.

TRANSFERRED FROM STUDENT TO GRADUATE
Michael Douglas Spear
Alan Frank Wilde

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT
Sunday U. Akpan
John Brian Gray
Christopher Guy Scott Wilson

Elected on 9th March 1964

MEMBERS
Eldon Elliott Appleby
James Cassels
Arnold J. F. de Soria
John Brian Luard Gilmore, Cdr., D.S.C., R.N.
William Hallsworth
Douglas Hay
Richard Bonner Humphreys
James Ivor James, Lt. Cdr., R.N.
Samuel Kinghorn
Richard Nathaniel Montgomerie Lea, Lt. Cdr., R.N.Z.N.

Murray Albert Macmillan
John Reid Martin
William Henry Mercer
Alexander Morrison
Benjamin Robert Randles
Ronald Stott
Oswald Toshe
George Mitchell Whitelaw
Ernest Wiberg

ASSOCIATE MEMBERS

Kenneth Malcolm Batchelor
Ramsay Kinghorn Birrell
Lyndall Cook
Eric William Forshaw
William Anthony Hepworth
Edward Joseph Howlin
Kuldip Singh Hundal
Arangath Chacko Joseph
William Kinghorn
Adekunle Shamusi-Deen Lawal, B.Sc. (Lond.), Sub-Lieut., Nigerian N.
David Edward Marshall
Omparkash Tejbhan Mehta
Robert Bain Middleton
Colin John Mountford
Rashidi Ayinde Raimi, B.Eng. (McGill), Lieut. (E), Nigerian N.
Patrick David Revans
George Ritchie
Leonard George Shaw
Thomas Taylor
John Graham Ismay Tyson, Lieut., R.N.
Viranshu Vimal
Henry Maddison Walker
Clive Anthony Wiles
Colin David Wilkie

ASSOCIATES

John Henry James Berry
Harry Bland
Donald William Bridges
Geoffrey Falconar
Prem Saran Gupta, Sub-Lieut. (SD) (ME), I.N.
Peter William Hebdon
George Derek Leek
Alexander Michael McGhie
Appala Narasiah
Thomas Richard Pearce
Jackie Rathsam
William Tortike

GRADUATES

Peter Allan
Peter Barrie Hamer
Harry Mathias
Subhas Chandra Roy
Keith Martin Townley
James Runcie Troup

STUDENTS

Eleazer Chimezie Akwiwu
Safa Y. Ashkuri
Michael Keneth Blow
Brian Peter Daly
George Birrell Fraser
Alan Roger Head
Roger John Howe
Gordon Jeffrey
Douglas McInroy
Ralph Philip Robson
Michael Douglas Still
David Harold Swallow
Alastair Wells
John Christopher Wright

PROBATIONER STUDENTS

Ian Gordon Campbell
John Fox
Andrew James Gibb
Michael John Part
Michael John Kingsmill Pope
James Joscelin Sisson

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER

Robert Anthony Babington
James Elliott
Robert Gemmell
Derek George Reeves Hall
Antony Hubbard
John Edward Lowther
Robert Colum McCartney
John Gray Richards, Lt. Cdr., R.N.

TRANSFERRED FROM ASSOCIATE TO MEMBER

Piero Balestrino
Edward Henry Barth
Timothy James Gerard Cronin
Kenneth Pike

TRANSFERRED FROM GRADUATE TO MEMBER

Cecil Scott

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER

David John Eyres, B.Sc.
William Osborne Gray
Colin Wyatt Hall
John George Howard
David Michael Jones
Oliver John Mark
Robin Gore Rees
Ian Ramsay Ritchie
Malcolm Hunter Tait
John Campbell Thompson
Wilfred James Thompson
John Anthony Vost
Gordon William Whitehead

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER

Donald Young Campbell
James William Christie
John Thompson Newton, B.Sc. (Durham)
David Robert Speirs

TRANSFERRED FROM STUDENT TO GRADUATE

Ian Jeffrey Day
David Alfred Nazzaro
Laurence Roy David Saunders

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE

Beverley William Archer
Clive Gray
Guy Nettleship
Alan Robinson
Gerald Malcolm Colvin Taylor

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT

William Geoffrey Adams
Charles Barry Connoly
David Wickham Cory
David Russell Crowther
William Farrell
Kenneth Gray Lee
John Anthony Share
Allan Godfrey Willis

OBITUARY

JOSEPH HEDLEY BROWN (Member 7277) died on 26th October 1963, after a brief illness. He had been a Member of the Institute since 13th March 1933.

Born on 3rd January 1889, Mr. Brown began his engineering career as an apprentice with Clelands Slipway and Forge Masters, Willington. During his many years at sea he served in vessels of the Stag Line and the United Africa Company (later the Palm line). He was chief engineer of s.s. *Conabrian* when she was torpedoed from the air off Aberdeen in 1940. For his efforts in beaching her and in the subsequent salvaging of the ship, he was mentioned in the *London Gazette* and commended for brave conduct. He continued to serve the Palm Line as chief engineer of various steamships until his retirement in 1957.

Mr. Brown held a First Class Board of Trade Steam Certificate and also a First Class Certificate of the British Oxy-Acetylene Welding Association.

BERNARD GAVIN (Associate Member 13832) was born on 27th January 1905. From 1920-1925 he was a Royal Naval Artificer Apprentice receiving technical training at Devonport and aboard H.M.S. *Fisgard,* Portsmouth. On completion of his training, he commenced his service with the Royal Navy, joining H.M. Repair Ship *Sandhurst*. During the following fifteen years he served in H.M. Submarines, for three years, and aboard various vessels of the Fleet, including H.M.S. *Acasta* and *Rodney*. He also served in the Shipfitting Department, H.M. Dockyard, Portsmouth.

On leaving the Navy, he became a technical instructor and, continuing his own studies, obtained an Ordinary National Certificate in 1946 and a Higher National Certificate in 1948. He continued in his career in the teaching profession and, at the time of his death on 20th December 1963, was a lecturer in the Engineering Department at Highbury Technical College, Cosham, Portsmouth.

Mr. Gavin was elected an Associate of the Institute on 16th June 1952 and transferred to Associate Member on 10th February 1958. He was also an Associate Member of the Institutions of Mechanical Engineers and Production Engineers.

FRANCIS HESLOP (Member 4794) was born on 9th January 1889, in South Shields, and served his apprenticeship with Mountstuart Dry Docks Ltd., Cardiff, from 1904-1910. He spent a short period as a draughtsman in London before going to sea. He gained his First Class Certificate in 1915 and, while he was chief engineer in H.M.T. *Argyll* in 1917, he was severely injured and spent a year in hospital in Tunis and London.

He joined his father's marine consulting practice at Cardiff in 1919 and continued on the death of his father. In 1927 he was appointed Consultant to the Powell Duffryn Coal Co. Ltd. and sailed in the s.s. *Somme* to South American ports for extensive tests and trials, using as bunkers, the company's Penallta Washed Mixture. He later made three further successful trial trips on other vessels.

He was Marine Superintendent to Pardoe Thomas and Co. Ltd., and superintended the building of their *Knight* fleet in 1929 at Lithgow's yard in Glasgow and Sir John Priestman's in Sunderland.

Mr. Heslop acted for many years as Marine Engineer Consultant to the Powell Duffryn Co. Ltd. and subsequently, on nationalization, acted in the same capacity to the National Coal Board. He also practised as a private marine surveyor and consultant. Shortly before his retirement in 1958 from active business, his son joined him as a partner and is now carrying on the practice.

Mr. Heslop was elected a Member of the Institute on 27th March 1923 and was also an Associate Member of the Royal Institution of Naval Architects. He died at his home on 29th November 1963 and leaves a widow and two sons.

CHARLES JACKSON (Member 10557) died suddenly on 17th July 1963, aged 60 years.

Mr. Jackson served his apprenticeship on the Tyneside, one year with Eltringhams Ltd. and four years with Sir W. G. Armstrong-Whitworth and Co. Ltd. He went to sea in 1926 as fourth engineer and for the next twenty-three years served almost continuously as a seagoing engineer, with various companies. He held a First Class Ministry of Transport Certificate and was employed in the grade of chief engineer from 1942 onwards.

He left the sea in April 1949 and for the last fourteen years of his life worked as a chargehand fitter for British Resin Products Ltd. (Distillers Co.) and actually died at work.

Mr. Jackson had been a Member of this Institute since 4th December 1945. He leaves a widow.

GEORGE BURGESS LOCKLEY (Member 11671) died suddenly on 29th February 1964, at Altrincham, Cheshire. He had been a member of the Institute since 1947, and was also a member of the Royal Institution of Naval Architects, and the Society of Consulting Marine Engineers and Ship Surveyors. He leaves a widow and daughter. Born on 23rd December 1902, at Altrincham, he was educated at Wallasey Higher Elementary School, and served his apprenticeship with Cammell Laird Ltd. He went to sea with Elders and Fyffes, gaining his First Class Steam Certificate before returning to Cammell Laird as Assistant Ship Repair Manager.

In 1939, Mr. Lockley took up the position of Assistant Marine Superintendent with the Bolton Steam Shipping Company, and was appointed Chief Marine Superintendent in 1943, which position he retained until his death.

REGINALD STANLEY MILLS (Member 16698) died suddenly on 21st December 1963, age 34 years. Born at Oakleigh, Melbourne, Australia, he attended the Springfield North State School and the Caulfield Technical College, afterwards serving a five-year apprenticeship in engineering with Robison Bros. and Co., Shipwrights, of South Melbourne, which he completed satisfactorily.

He joined Shell Tankers Ltd., as fifth engineer, in 1949 and remained with that company until 1962. During this period he served as fourth to chief engineer, in a number of motor vessels and steamships of Shell's tanker fleet, and gained his First Class Motor and Steam Certificates, in 1954 and 1961 respectively.

In 1962 he joined the Union Steamship Company of New Zealand as second engineer and served in the coastal ships

Raroon and *Resdon* until the time of his death.

Mr. Mills was elected an Associate Member of the Institute on 2nd November 1955 and transferred to full membership on 11th May 1962.

EDWARD HARRY PATTERSON (Associate Member 6297) died on 4th March 1964, at the age of 64. He was educated at Leith Academy and Leith Technical College, and afterwards served an apprenticeship with Brown Bros and Co. Ltd., Edinburgh. In 1922, he became a draughtsman with the same firm. In 1925, he returned to Leith Technical College as a lecturer in Machine Drawing. He also served as a draughtsman with G. and J. Weir Ltd. In 1940 he was responsible for design and estimates for heaters and evaporators, etc., with this Glasgow firm.

Ten years later he became Chief Draughtsman with Messrs. R. Y. Pickering and Company Ltd., Wisham. However, in 1952 he returned to G. and J. Weir, where he was in charge of the land evaporating plant section of the drawing office. He became Chief Engineer (Heat Exchange) in 1958, specializing in the design of multi-stage evaporating plants. In connexion with these he made a visit to Kuwait which brought about the erection of a large plant to obtain fresh water from sea water.

At the end of 1959, he had a very serious illness and, although he made a good recovery, he felt it necessary to retire in 1962. He leaves a widow and two sons, one of whom is an architect in Glasgow, and the other a physicist in Dorset.

He had been elected an Associate Member in 1929.

JOHN RICHMOND (Member 9232) was born in Glasgow on 4th August 1892 and received his education there at Allan Glen's School. He served his apprenticeship with G. and J. Weir Ltd. and, in 1913, commenced his seagoing career by joining, as a junior engineer, s.s. *Salamis,* a vessel owned by Glen and Co. of Glasgow. He spent the next two years in the Far East.

On returning to the United Kingdom in 1916 he joined the Mine-sweeping Patrol at Dover as an Engineer Sub-Lieutenant, R.N.R. After being demobilized in 1920, he took up an appointment as an engineer surveyor with the British Engine and Boiler Company. He remained with this concern until 1927 when he returned to sea, serving in a number of general traders until 1936. In that year he joined the staff of the Navigators' and Engineer Officers' Union as Assistant District Secretary in Glasgow until 1939, when he once more returned to sea. When hostilities broke out he was serving in tankers and remained with these vessels throughout the war, having been torpedoed and suffered bombing attacks on a number of occasions.

From 1945 onwards he served as chief engineer in tankers and large bulk carriers. In 1957 he was in attendance during the construction and fitting out of the ore carrier m.v. *Dalhanna* eventually sailing in her as chief engineer for some two years. In March 1963 he reluctantly retired. However, after a short period ashore, he contracted pneumonia with complications and in a comparatively short time had to enter hospital where, after six months, he died on 16th December 1963.

Mr. Richmond was elected a Member of the Institute on 24th March 1941.

EDWIN SHACKLETON (Associate 9370) died suddenly on 10th March, 1964, aged 60. He had been connected with the Institute since 1942 when he was elected as an Associate. He was apprenticed with Henry Widdop and Son, Keighley and, in 1925, joined the design staff of Messrs. Thornycroft, Reading. He left in 1928 to join English Electric at Rugby, in a similar capacity, but came back to reading in 1930 to accept a position as design draughtsman with the signals section of the Great Western Railway.

In 1933, he joined R. A. Lister at Dursley where he, among other marine projects, was largely responsible for the marine installation of Blackstone engines in wartime motor fighting vehicles. In 1954, he became chief draughtsman of Lister Blackstone Marine and at the time of his demise had served the Lister organization for 31 years.

He also served as a lecturer in engineering design at the Dursley Technical College. He was a gold medallist, Keighley Engineers' Association. He leaves a widow.

JAMES LEGGATT SMITH (Member 9986) was born on 4th January 1891 in Kilwinning, Ayrshire. He was educated at Kilwinning High School and Irvine Royal Academy, and served an engineering apprenticeship in Glasgow.

In 1912 he went to sea as fifth engineer with the Hall Line, in the service of which company he obtained his First Class Certificate of Competency. He was appointed chief engineer in the *City of Brisbane* in 1925. In 1947 he became assistant superintendent engineer to the Ellerman Bucknall Line and superintendent engineer in 1948. Mr. Smith had been elected a Member of this Institute on 18th July 1944.

He retired from professional life in December 1957 and enjoyed good health until the Autumn of 1963 when he was compelled to enter hospital. Despite the most careful attention, he developed pneumonia and died on 4th December last.

GEORGE HENRY STRONG (Member 10890), died in February 1964, after a long illness. He was 53 years of age. Mr. Strong was elected a Member of the Institute in 1946, and, at the time of his death, was acting as a ship surveyor and consulting engineer. He served his apprenticeship with Earle's Shipbuilding and Engineering Company, followed by a period at sea with the Blue Star Line and W. H. Cockerline and Company. He then entered the family firm of John H. Strong, founded by his grandfather. When this firm linked up with Broderick, Wright Ltd., under the style of Broderick, Wright and Strong Ltd., he became a director. In that capacity he was well-known in North East Coast shipping circles. He was also a member of the Society of Consulting Marine Engineers and Ship Surveyors. He practised as a consultant for 25 years at Hull, East Yorkshire.

He leaves a widow, and a son, and daughter.

JAMES LEASK SUTHERLAND (Member 8907), managing director of the S.T.S. Engineering Co. Ltd., died on 17th December 1963.

Born in Leith on 7th September 1909, he was educated at Trinity Academy, Leith, and, from 1924 to 1929, served an apprenticeship with Menzies and Co. Ltd. After completing his indentures, he went to sea as an engineer with the Royal Mail Lines, remaining in the company's service for ten years and obtaining a First Class Motor Certificate with Steam Endorsement. He came ashore in 1939 to set up his own small engineering company, the S.T.S. Engineering Co. Ltd., at Kingston upon Thames and was actively engaged in business up to the time of his death.

Mr. Sutherland was elected an Associate of the Institute on 1st May 1939 and transferred to full membership on 13th December of the same year.

T. SEABROOK WALLIS (Honorary Life Member 1779) died in August 1963 leaving a widow. He had been a member of the Institute since 1905, when he was elected an Associate. He became an Associate Member in 1907 and a Member in 1914.

Mr. Wallis served his apprenticeship with Jas. Simpson and Co. Ltd. and the R.M.S.P. Co. and was afterwards employed for a time by the latter company in their engine works.

He held an Extra First Class Board of Trade Certificate, with four years sea service and had also been an engineer at a glass works in Surte, Sweden. In 1949 he retired after 23 years as Station Superintendent for the Bristol Corporation Electricity Department and, later, the South Western Division of the British Electricity Authority.

GORDON POLLOCK WATT (Member 4327) died on 7th December 1963, at the age of eighty-one.

Mr. Watt served his apprenticeship with Cowans, Sheldon and Co. Ltd. of Carlisle. He was, at one time, an engineer with the Port and Lights Administration, Sudan Government, at Port Sudan and later became Resident Engineer with the Port of London Authority, Bulk Grain Division.

In 1921, he went into business on his own account as a director of the firm of Harley, Conyngham and Watt Ltd., with offices in Moorgate, London. His design work covered a variety of engineering applications, including conveyors, elevators, cranes and trucks. However, he did specialize in the guarding of machinery in mills and factories, and designed installations for several well known companies. He also represented several overseas firms, principally Canadian and United States concerns.

After the last war his work was severely handicapped by a series of surgical operations which involved the loss of both legs. Despite this and other infirmities, he carried on working until a few weeks before the time of his death.

Mr. Watt was elected a Member of the Institute on 22nd July 1921.

Tom George White (Member 5515), a well known figure in shipping circles in Cardiff and also in the North East, died in Darlington, Co. Durham, on 20th November 1963.

Born in Middlesbrough, on 20th October 1889, Mr. White served his apprenticeship in the Tyneside shipyards of the Shields Engineering Co. Ltd. and Smith's Docks Ltd. During the First World War, he served in France, at first with the Royal Engineers and later with the Royal Flying Corps. He was demobilized in 1919 and joined Elliott Jeffreys, of Cardiff, as manager, later becoming assistant superintendent to Stephenson, Clarke and Co. Ltd., also of Cardiff. In 1926 he became superintendent engineer to Sir R. Ropner and Co. Ltd., succeeding his father, and in 1939 was promoted to chief superintendent to Ropner's at West Hartlepool. He moved to the company's head office in Darlington in 1946 and retired in 1954.

Mr. White was elected a Member of this Institute on 8th February 1926 and was also a Member of the Institute of Consulting Engineers.

James Wood (Member 17616) died on 24th December 1963, aged 71. He was educated at Robert Gordon's Technical College, Aberdeen, and served an apprenticeship with Aberdeen Trawlowners and Traders Engineering Co., from 1908-1913. In the latter year, he became a junior engineer with the Red Star Line. From 1914-1919, he served as junior and senior engineer with the Union Castle Steamship Co. In 1918, he gained his First Class Certificate of Competency. From 1919-1921, he was engineer to the Rhodesian Trading Co. of Southern Rhodesia, after which he became engineer superintendent to the Thesan Steamship Co. Ltd. In 1927 he was appointed ship and engineer surveyor to Lloyd's Register of Shipping, for the port of Cape Town. From 1936 on, he was surveyor to Bureau Veritas for the Transvaal and Orange Free State, and also freelance surveyor to various mining companies and insurance concerns.

He had been a Member of the Institute since 1956 and was also an Associated Member of the South African Institute of Mechanical Engineers.