Investigations into the Stressing of Crankshafts for Large Diesel Engines

M. LANGBALLE, Siv. ing. (Member)*

Based on measurements of service stresses in crankshafts, methods for the theoretical calculation of stressing of crankshafts and their dynamic behaviour are evaluated, and explanations of some of the observed phenomena are proposed. By means of model experiments, the experimental data necessary for theoretical analysis are found. The agreements between theory and experiments are particularly good for torsional stresses, and the bending-stress calculations are also acceptable.

For simplified calculations, some empirical data are given, derived from the measurements. By means of the model tests, attention is drawn to some specific problems of semi-built crankshafts: transmission of torque in shrink-fits and residual stresses in crankpin fillets due to shrinkage. Methods for evaluation of structural defects in crankshafts are discussed. Approximate analysis of service stresses in both fractured and trouble-free operational crankshafts is carried out.

To include secondary resonance effects, the theory of forced torsional vibrations for multi-mass systems is extended to take time-variable masses into account.

INTRODUCTION

The design and dimensions of crankshafts for Diesel engines are generally laid out to comply with the requirements of the classification societies. Their rule formulae may seem rather simple and are, in principle, methods of extrapolation of the vast amount of past experience by means of dimensional analysis. Nevertheless, with the present-day design practice, the rate of failure of large crankshafts is very low compared to the risk they are exposed to, as has been demonstrated before this Institute by Dr. Archer[2]. Despite the reliability and apparent conservative dimensions given by the rules, crankshafts fail now and then, and a closer scrutiny of crankshaft stressing may be justified in the light of current development towards increased loads and outputs, and the larger values at stake. The several aspects and methods of theoretical crankshaft calculation have been exhaustively treated in the literature, and the present position and availability of high-speed computer techniques now permit a fairly free use of elaborate mathematical formulations and solutions. However, to apply these sensibly and with confidence, there seems to be a certain need for experimental evidence and verification of the methods.

By means of theoretical calculations which reproduce service stresses with reasonable accuracy, potential possibilities exist for optimizing the crankshaft designs based upon safety and economical requirements, as well as means of formulating requirements to the quality of steels. To this end, a transposition of current empirical practice and experience into terms of realistic stress levels is desirable.

The object of the investigations to be described in this paper is to evaluate existing theoretical methods, by comparison with measured service stresses, and eventually establish a basis for further experiments or theoretical development which may seem desirable. An endeavour is also made to extract from the measurements empirical data which seem to have at least some general character.

*Senior Engineer, Research Department, Det norske Veritas, Norway.

I.—SOME PROBLEMS OF THEORETICAL SHAFTING CALCULATION

The crankshaft of a Diesel engine, with its complicated geometry and appended masses, constitutes a dynamic, elastic system capable of several modes of deformation and coupled vibrations. A rigorous solution to determine its stressing would require the establishment of a unified, coupled system of differential equations with numerous experimentally determined terms.

Such a formulation is, no doubt, possible and capable of a computer solution, but perhaps of doubtful value, due to the many approximations and simplifications which in any case have to be incorporated. The approach chosen here is the traditional one, dealing separately with the different cases of stressing: torsion, bending due to radial forces, axial deformation and misalignment. Coupling terms are only considered when necessary to explain important observed phenomena.

In order to evaluate the different separate methods of calculations, some simple measurements were carried out, by means of strain gauges, on a four-stroke, eight-cylinder, high-speed engine intended for electric power generation. The most important particulars of the engine are given in Fig. 1(a). The strain gauges were arranged on cranks Nos. 4 and 8, to record, separately, stresses due to bending and torsion. Initially, the gauges were calibrated in place by statical forces on the shaft to determine the stress concentration factors in bending and torsion, referred to nominal stresses in the crankpin. Simultaneously, with measurements of dynamic stresses during operation, cylinder pressures were recorded. The results and a theoretical analysis of these will be described in the following paragraphs.

Torque and Torsional Vibrations

The impulses which form the basis for vibration calculations are usually derived from the gas-pressure measurements. For computer applications, it is somewhat awkward to have to feed the computer memory with relevant impulse data for every set of calculations and, to overcome this, a theoretical calculation of the *P-V* diagram is carried out, using characteristic data for the engines. Constant polytropic exponents for compression and expansion are used and certain empirically derived mathematical expressions are introduced to describe the pressure variations, during the combustion period and scavenging process. Tangential pressures, based on measured and theoretical time pressure diagrams and their harmonic components, are compared in Fig. 1(b), for the engine to be investigated. A similar procedure is available for two-stroke engines.

Forced vibration calculations are now becoming accepted as a standard procedure, to obtain flank and resonance stresses or the total fluctuating torque in shafting systems. The method used here is to establish the differential equations of motion for the torsional system and to solve them by matrix inversion on a digital computer. The forced-vibration calculations may readily be extended to take variable engine masses into account, developing, for a damped, multi-mass system, the fundamental equation for the variable single-mass system, which has been established and solved by Draminsky[3], who has drawn attention to the important dynamic phenomena which may occur in such systems. Some details of the mathematics involved are given in Appendix I.

A comparison between resonance and flank stresses calculated on the assumptions of constant and variable masses, is also given in Appendix I, applied on an engine where considerable secondary resonance amplification, due to the variable masses, was found by torsiograph measurements.

The usual concept of transmission of torque through a crank is based on the assumption that a fraction of the torque is transmitted through the crankpin as torsional stress, the remaining torque bypassing it by means of a lever action and shear force in the crankpin. To investigate this question, a dynamic calibration of the torsional gauge in crankpin No. 8 of the experimental engine was carried out in the following way: the load of the engine was increased by steps, and the increments in mean torsional stress, found by integration of the strain records, were compared to the corresponding torque increments read on the brake × 7/8. A dynamic stress concentration of 1·85 was found by this experiment compared to the static value of 1·91, indicating that the major part of the torque appeared as torsional stress in the crankpin.

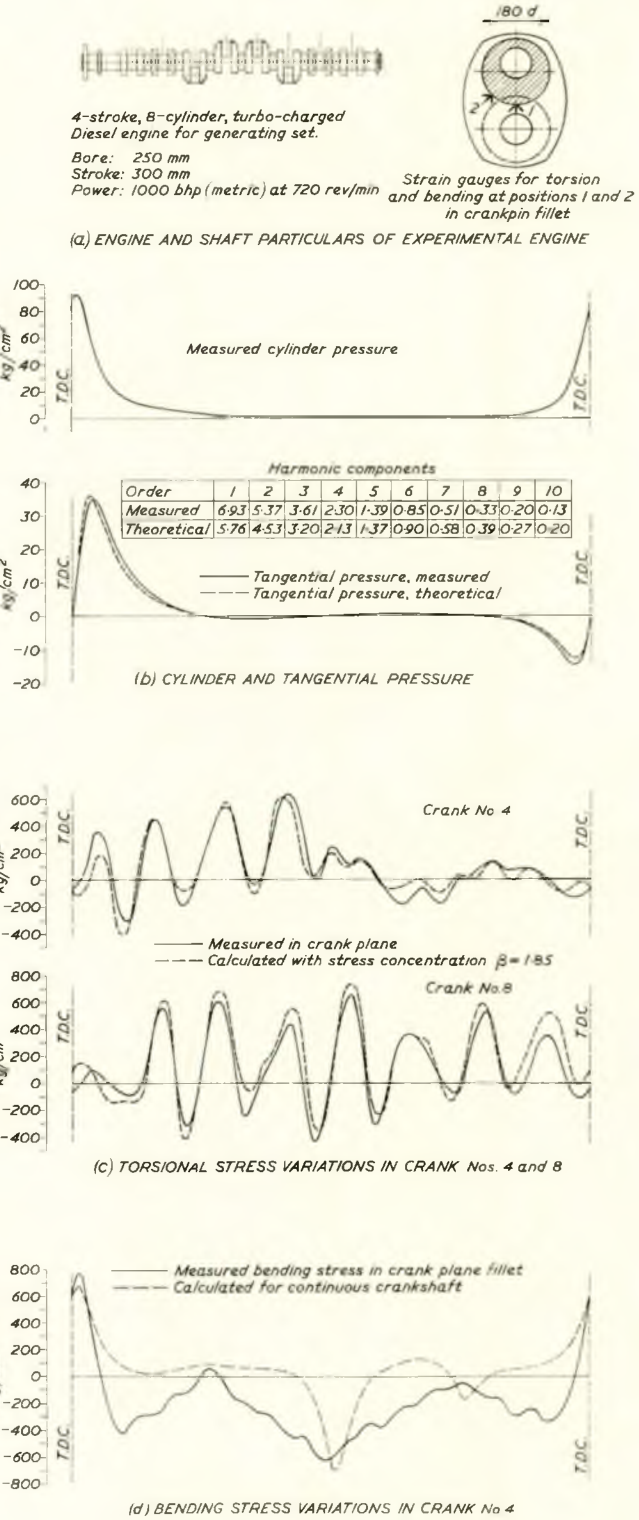
Comparisons of measured and calculated torsional stress, using theoretical impulses, static stress concentrations and coefficients of cylinder damping according to Maciotta[5], are shown in Fig. 1(c).

It is the general experience of the author that calculated torsional stresses and those found by strain gauge measurements on the shafting of operating engines, usually agree very well, within the uncertainties of damping and impulses. For determination of fluctuating torsional stress throughout a crankshaft, a theoretical calculation may thus be relied upon, relevant data for impulses and damping being available or determined by simple routine measurements.

Bending Stresses

For calculations of crankshaft bending, a straight beam analogy is adequate for an approximate analysis. By means of the slope-deflexion method, the bending calculations may, however, be carried out in a more rigorous way, introducing the elastic parameters of a single crank throw. These elastic parameters are defined as the angular deformations at both journal centres of a single, simply-supported crank throw, which is loaded by a unit moment in two planes at one end, a unit torque and unit radial and tangential forces.

The complete set should preferably be determined by model experiments, but, in the calculations carried out here, a method[9] is used, due to Timoshenko who has given the most important elastic parameters by means of beam theory. The distribution of bending moments being determined by solving the set of linear equations, which can be established by requiring moment equilibrium and continuity at the journal supports, the fillet stresses are found anywhere in the fillet by means of the stress concentration factors referred to the nominal bending stress in the webs. These stress concentration factors have to be found by loading a single crank in two perpendicular planes.



Order	1	2	3	4	5	6	7	8	9	10
Measured	6·93	5·37	3·61	2·30	1·39	0·85	0·51	0·33	0·20	0·13
Theoretical	5·76	4·53	3·20	2·13	1·37	0·90	0·58	0·39	0·27	0·20

FIG. 1—*Comparison of measured and calculated impulses and stress variations. The different measurements were made at approximately the same loading conditions of the experimental engine (maximum load)*

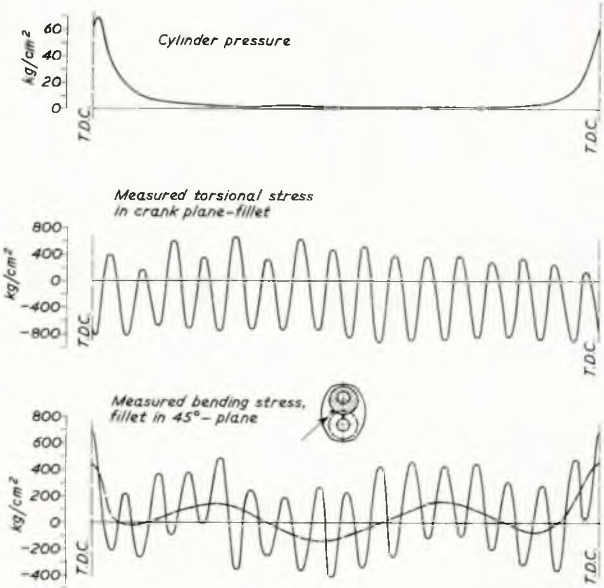


FIG. 2—*Stressing of crank No. 4 at major torsional critical rev/min*

This procedure is programmed for solution on a digital computer, to give stresses at every 10° of crank angle, loading each crank with its gas and inertia forces.

The cranks may also be loaded with the torques found by the forced torsional vibration calculation described previously. A restriction may also be introduced at the journals, to give restraining moments at the supports proportional to the angular deflexions at the centre of the journals.

The bending stress variation measured on the test engine is compared to the calculated fillet stresses, based on nominal values at the middle of the web thickness and the stress concentrations at the point of measurements. The theoretical variation of stress does not agree very well with measured stress variation, although the stress ranges are a good fit (see Fig. 1[d]). It is necessary here to have in mind some of the limitations and approximations of the bending stress calculations, which may account for the discrepancy:

a) the forces and reactions are assumed to attack at a point;
b) the centres of the journals are assumed to be restricted from lateral displacement;
c) dynamic amplification of torsion-coupled axial deformation is neglected;
d) dynamic magnification of bending stresses due to impulse loads is neglected;

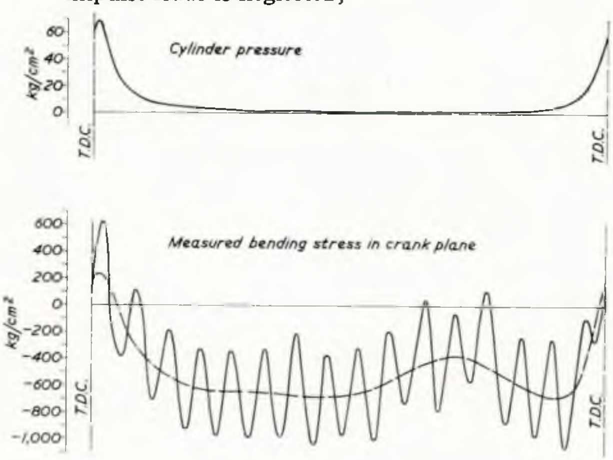


FIG. 3—*Stressing of crank No. 8 at major axial critical rev/min*

e) some elastic parameters of the crank throw are omitted (axial deformations due to the unit loads).

A particular feature to be noted, is the strong coupling between torsion and bending stress in the 45° position in the fillet of crank No. 4 at the major torsional resonance, which was found by the dynamic measurements (see Fig. 2). This may be thought to be due to a restriction of angular deformation of the journals imposed on them by the main bearings, but a theoretical analysis taking this into account was unable to describe this coupling. A similar effect is found to be present in the stressing of the large crankshaft to be dealt with later, and as will be shown there, it seems to be caused by a local elastic effect due to the asymmetric location of the point of measurement.

Axial vibration stresses are usually associated with the large slow-running engines. As a point of interest, it is mentioned that for a high-speed engine of the type mentioned here, a major critical axial vibration resonance may well appear within the operating range of rev/min, contributing substantially to the bending stress level as demonstrated in Fig. 3. Minor adjustments to the weight of flywheel or counterweights are usually sufficient to remove them from the service speed range.

With the deficiencies of even elaborate methods of bending stress calculations and the various assumptions which have to be introduced, such calculations may seem to be of doubtful value. Meanwhile, for the four-stroke, high-speed engine, the conventional simple formula seems to be adequate for the practical determination of bending stress in the crank plane due to gas pressure, torsion coupling and dynamic effects excluded:

$$\sigma = \pm k \cdot \frac{p \cdot l \cdot D^2}{d^3} \cdot \alpha \qquad (1)$$

where

k = constant (from measurements: $k = 0{\cdot}55$)
D = diameter of cylinder bore (cm)
p = maximum gas pressure (kg/cm²)
l = distance between centres of journal bearings (cm)
d = crankpin diameter (cm)
α = stress concentration referred to crankpin section.

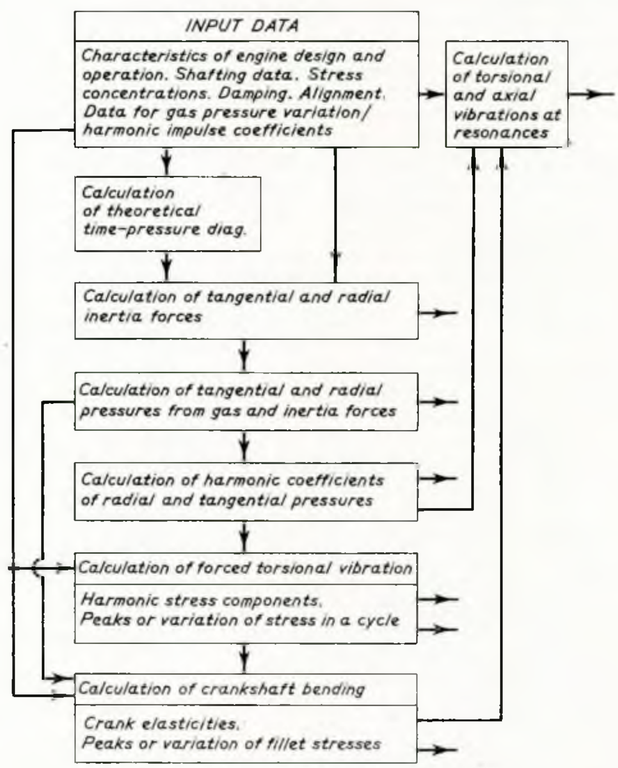


FIG. 4—*Flow diagram for computer calculations of crankshaft stresses*

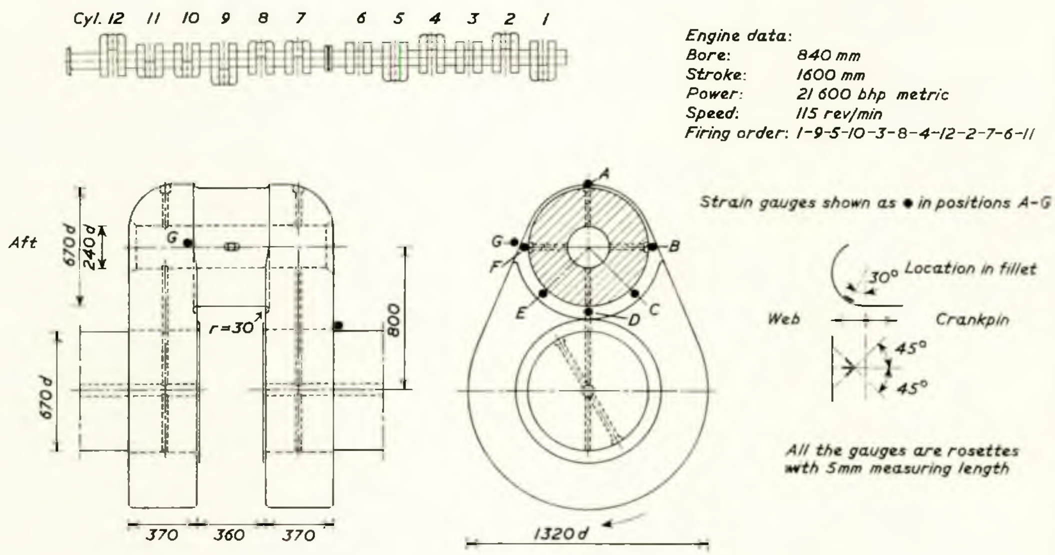


FIG. 5—*Particulars of experimental engine and shafting*

A further evaluation of bending-stress calculations will be given in the following section, dealing with large slow-running engines.

Computer Calculations

The theoretical methods outlined are programmed in the FORTRAN Language for a UNIVAC 1107 digital computer for routine applications. The separate chains of computer programmes may be linked up to perform calculations according to the requirements of specific cases, with the flow of information indicated in Fig. 4. When specific, relevant data for an engine, regarding impulses, damping, stress concentrations and various stiffness parameters are unavailable, they may be computed on the basis of engine specifications and dimensions using theoretical or empirical methods.

II—EXPERIMENTAL INVESTIGATIONS ON LARGE ENGINES

Comparatively few thorough investigations on service stresses in the crankshafts of large engines have appeared in the literature. Those available principally deal with bending stresses in the aftermost crank[1] and some need was felt for more knowledge of the stressing of those cranks exposed to the largest torque variations which usually appear somewhere midway between the fore and aft end of the crankshaft. The measurements to be described here were carried out on a type of large-bore, 12-cylinder engine, with the particulars shown in Fig. 5. This engine is quite representative of its kind regarding torsional stress level and crankshaft design, compliance with the requirements of classification societies being to within good margins.

Measurements were initially carried out during operation at sea, but an opportunity was later offered to duplicate and extend some of the measurements on a similar engine during workshop trials. The principal object was to determine the distribution and origin of service stresses around the crankpin fillet. Attention was principally confined to crank No. 5, but some measurements were also made on cranks Nos. 1 and 12 in the fore and aft end respectively. The dynamic measurements have been supplemented by static stress measurements on scale models, to be described in Section III.

Preliminary photo-elastic investigations on crank throw models indicated that the torsional-stress concentration is maximum and fairly flat 30° into the fillet, the bending-stress concentration, however, being somewhat less than maximum in this location. The positions of the gauges are shown in Fig. 5. All the strain gauges were of the three-filament rosette type, the filaments being recorded separately and simultaneously on a pen recorder via strain gauge bridges of the carrier-frequency type.

As it was considered of some importance to determine a reference line on the recordings corresponding to zero dynamic stress, the strain gauges were carefully temperature-compensated by means of dummy gauges close to the active rosettes with identical wiring. Upon each run of measurements, the engine

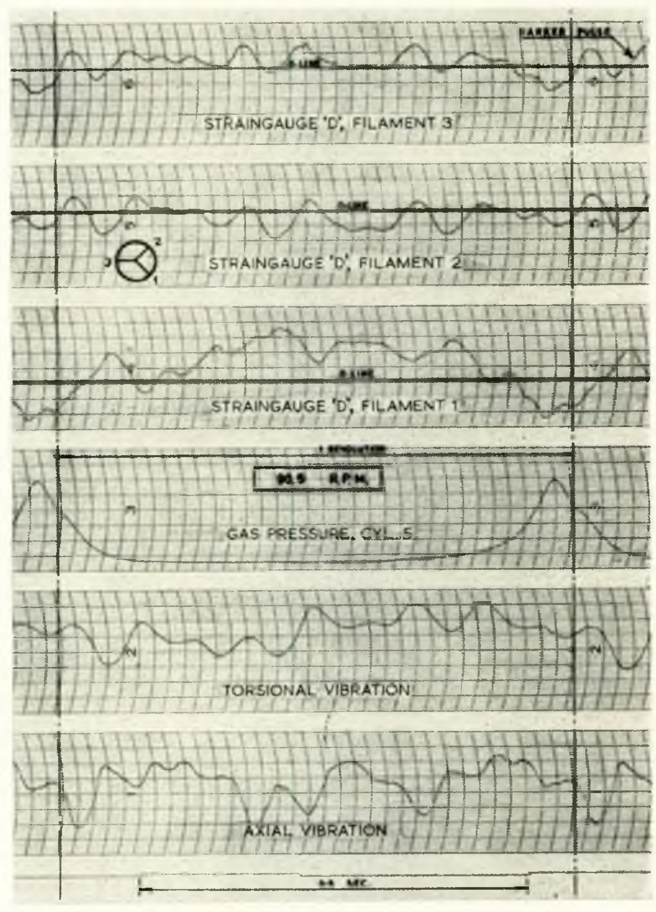


FIG. 6—*Dynamic recording*

was stopped, and the crankshaft turned to locate an average zero position of the recorders' writing styli.

By means of electrical transducers, deflexions of torsional and axial vibrations at the free end of the crankshaft, as well as cylinder pressure, were recorded simultaneously with the strain gauge rosettes, producing sets of recordings as shown in Fig. 6. Recordings of this kind were taken at several operating conditions from all the strain gauge rosettes, enabling a correlation between external forces, dynamic behaviour of the shafting and the stressing of the fillet.

Systematic measurements and investigations on engines in normal production and service are difficult and may be frustrating, as the schedule of commercial operation should not be hampered unduly. The practical approach, under such circumstances, is to record and collect as much information as possible within the time assigned, reject suspicious material and select reliable and useful information upon careful scrutiny. This naturally increases the amount of analytical work considerably. The most important results of the measurements will be mentioned in the following sub-sections.

Torsional Vibration

Some consideration was given to providing a gauge supplying a signal proportional to the torque in the crankpin and which could be calibrated in terms of nominal torsional stress. The gauge "A", on the top of the crankpin, was selected for the purpose, it has a stress concentration of approximately 0·9-1·0 in torsion and 0·8 in bending according to model tests. An approximate calibration of the transducer was obtained by the following methods:

1) The mean recorded torsional stress at several loads was determined by integration, the increments being compared to the corresponding torque increments determined by the water brake or torsionmeter; the relative distribution of mean torque within the crankshaft was determined by means of indicator cards;

2) at the torsional resonances, the harmonic components of torsional stress determined from torsiograph records, were compared to the corresponding components of the measured torsional stress by the gauge "A".

Results from the two methods did not differ significantly, and stress concentration factors, $\beta_A = 1{\cdot}08$ and $\beta_A = 1{\cdot}14$, were found, from shipboard and shop measurements respectively. The stressing at the other gauges may now be referred to the nominal torsional stress. Another interesting conclusion to be drawn from this experiment is that the torque, introduced into the crank from the preceding torques, is largely transmitted as torsional stress through the crankpin, in agreement with the finding from the corresponding experiment on the high-speed four-stroke engine described previously.

The torsional characteristics of the propulsion installation are shown in Fig. 7. The distribution of torsional stress along the shaft is established with the aid of theoretical calculation, and is a convenient means of evaluating the term "additional stress due to torsional vibrations" which is subjected to limitations by the classification societies. To be correct, it should be defined as the difference between the peak stresses calculated by vectorial summation of flank stresses and the peak stresses found by vectorial summation of the plain torque contributions from the individual cylinders. The amplitude of the total variable torsional stress is, however, a better basis of evaluation of torsional stressing.

Axial Vibration

The axial vibration characteristics of the ship's installation, as determined from bending-stress measurements in the crank plane, are shown in Fig. 8.

Harmonic analysis of the records reveals a 5th order first-mode resonance (propeller excited), as well as a 9th order second-mode resonance. The latter is accompanied by an 18th order third-mode vibration. Throughout the range of revolu-

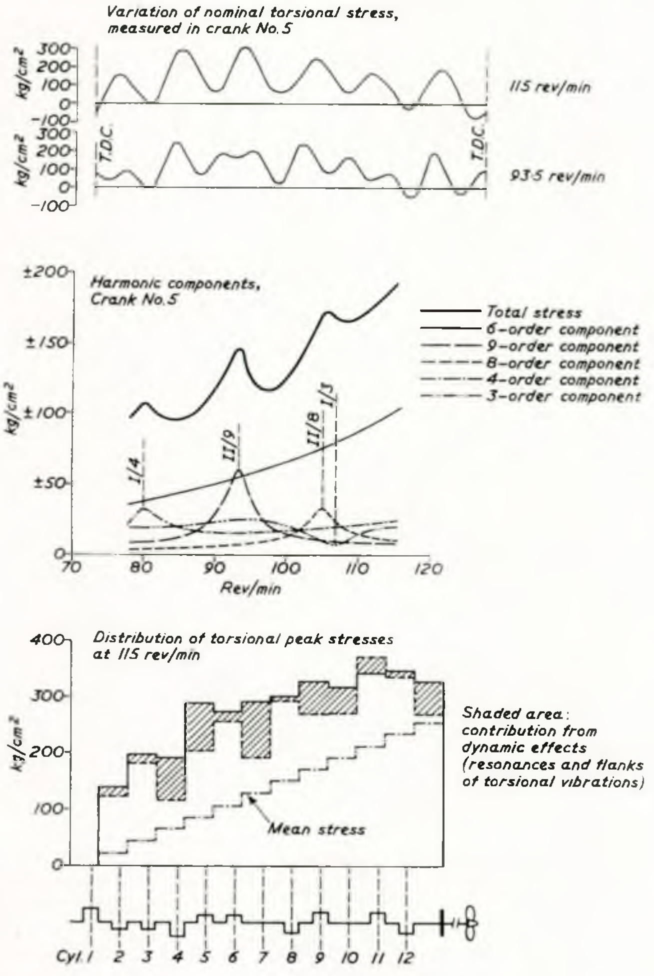


FIG. 7—*Torsional vibrations characteristics (ship installation)*

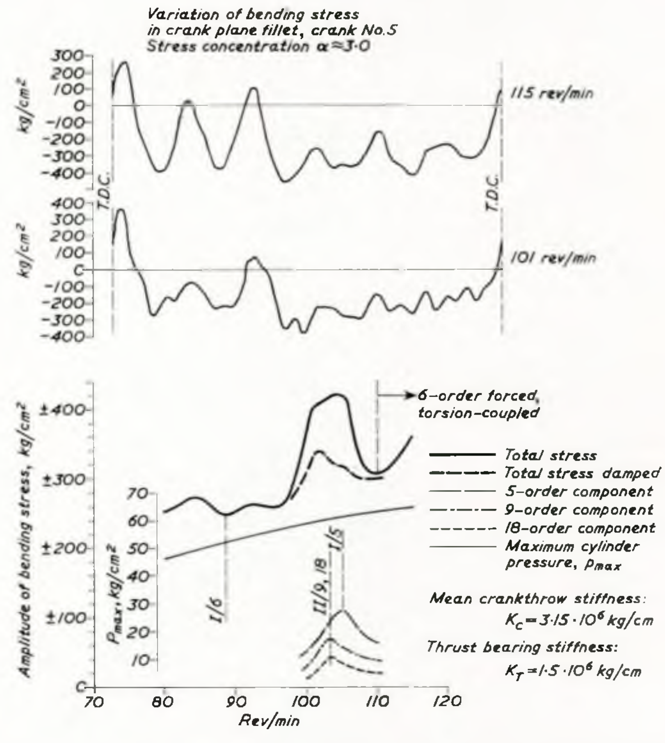


FIG. 8—*Axial vibrations characteristics (ship installation)*

tions, a 6th order is present. In the upper range it is particularly strong, probably owing to torsio-coupling. The introduction of an axial vibration damper reduces the 1/5 resonance to an insignificant level, but does not affect, very much, the forced torsion-induced 6th order vibrations.

The stress amplitude of the second-mode vibration is unimportant, but a study of it is nevertheless of some interest. Theoretical calculation of axial frequency of the freely-floating crankshaft for the four-stroke high-speed engine gave very close agreement with measured axial resonance frequency for a similar mode of vibration, using the calibrated axial stiffness of the crankshaft and lumped masses of the shafting system. The masses of connecting rods do not seem to participate in the vibration. In the test-bed trials of the large engine, the same assumptions are probably valid and, due to the absence of thrust and the slender thrust bearing pedestal on the water brake, the shafting can be considered as freely floating. Measurement of the axial resonance frequency of the second-mode on the test bed, thus offers a simple means of determining the average axial stiffness of the cranks, using the known masses in a trial and error calculation of natural frequency to find the fitting stiffness. The thrust-bearing stiffness of the shipboard installation is determined in a similar way. As a point of interest, the thrust bearing stiffness is lower at the second-mode vibrations than at those of the first mode. This may be explained by its non-linear character, the oil film stiffness being involved at small amplitudes.

By harmonic analysis of bending stress and vibration amplitude at resonance, the following approximate relation between measured axial deflexion, at the fore end of the shaft, and nominal bending stress may be established for the last crank:

$$\sigma = \pm \frac{f.\Delta a}{R.a_1} .0{\cdot}38.10^6 \text{ kg/cm}^2 \qquad (2)$$

where

f = measured amplitude at fore end (mm)
R = crank radius (mm)
Δa = relative axial deformation of the last crank
a_1 = relative deflexion at fore end.

Δa and a_1 are taken from the Holzer Table for axial vibrations.

Due to the effect of shrinkage of semi-built crankshafts giving the fillet around the crank plane large residual compressive stresses, the importance of axial vibration stresses are somewhat disputable. Away from the crank plane, the axial-vibration stresses decrease rapidly. However, other detrimental effects of axial vibrations should also be taken into consideration.

Fillet Stresses

The strain records from the rosette filaments were analysed to give the principal stresses and their directions at 70-100 angular positions of the shaft during a revolution. The principal stresses change continuously in direction and magnitude, and are somewhat awkward to evaluate. They were therefore processed in a digital computer to find the stress components in fixed directions, as demonstrated in Fig. 9. The direct stress

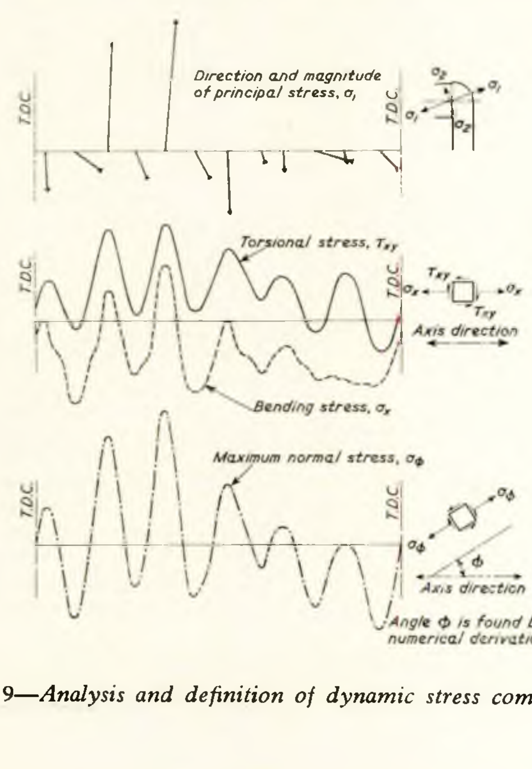


FIG. 9—*Analysis and definition of dynamic stress components*

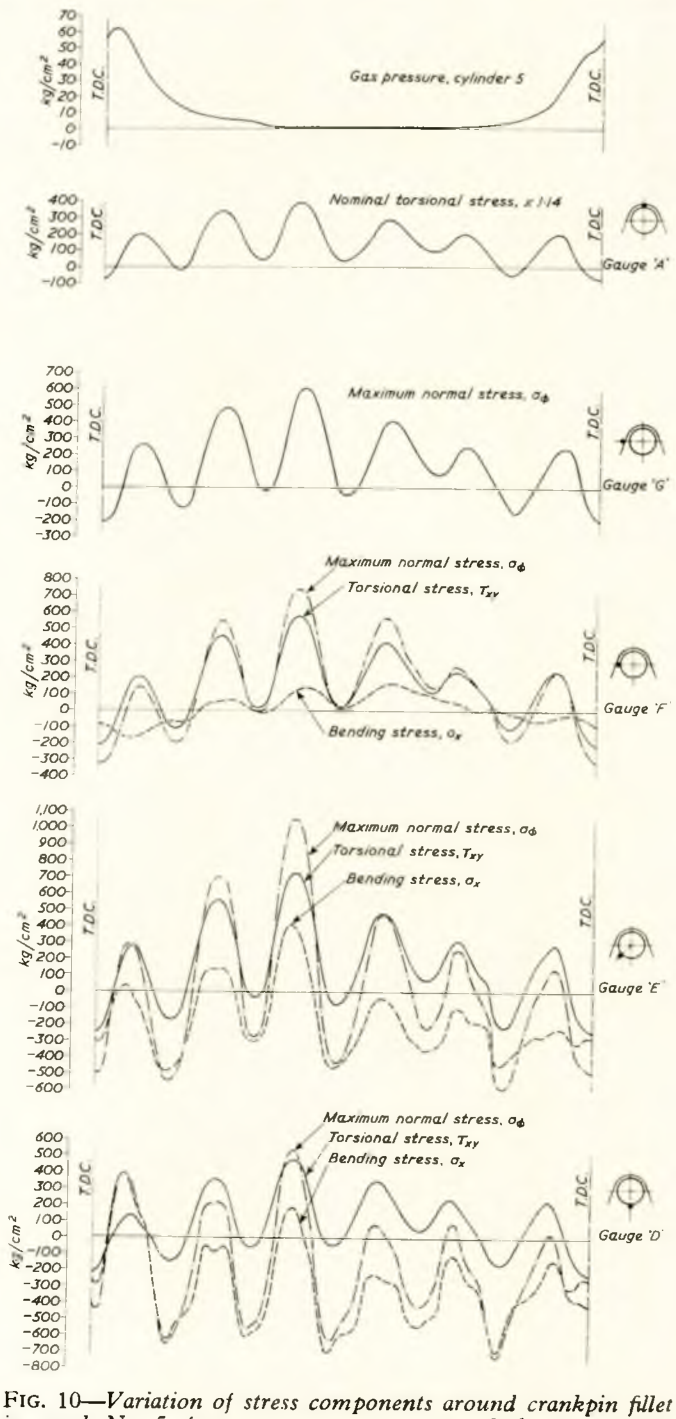


FIG. 10—*Variation of stress components around crankpin fillet in crank No. 5. from measurements on test bed—Load 21600 bhp, 115 rev/min*

in axial direction, σ_x, and shear stress in tangential direction, τ_{xy}, are defined as bending and torsional stress, respectively.

In addition to these basic components, a stress component, $\sigma\phi$, is computed, which is the normal stress in the fixed direction in which the variation of this normal stress is maximum. Its direction and magnitude are found by numerical derivation in the computer. This stress contains components of both torsion and bending according to the relative magnitude of these, and its variation gives a good idea of which is more important to the total stress level. The peak value is approximately equal to the maximum principal stress.

Throughout the analysis, extensive use of harmonic analysis was employed to isolate and correlate stress components and recorded dynamic phenomena.

Some typical samples of the stress variations measured by the gauges on crank No. 5 at full load and rev/min on the test bed are shown in Fig. 10. Comparing the various stress components with the pure torsional stress and the gas pressure, it will be noticed that their variation is closely related to torsion. The largest variation of the maximum normal stress, $\sigma\phi$, occurs in the fillet in positions 45 deg. off the crank plane. One reason for this circumferential distribution, is the torsional fillet stress which upon static model measurements is found to have a maximum around the 45 deg.-location. Similar distributions of torsional stress have also been found in quite different crank configurations of the solid type. The distribution of stresses around the fillet is shown in Fig. 11, in which they are referred to nominal torsional stress. It will be noted that the torsional stress concentrations agree almost perfectly with those predicted from model experiments.

A possible explanation of the evident coupling between torsion and bending stress at the 45 deg.-location, which was also significant in the stressing at the corresponding location of the four-stroke high-speed engine mentioned previously, may be restraints imposed by the journal bearings and adjacent cranks on the lateral angular deflexion of the journals. In effect, such restraints are equivalent to the introduction of additional bending moments perpendicular to the crank plane with a variation over the crank angle similar to the torque in the crank, and add to the stressing off the crank plane.

Contributing substantially to this effect is also the component of normal stress in the axial direction, which appears in the fillet off the crank plane even in the case of pure torsion load. This is demonstrated through the model experiments described in Section III. These observations render the definition of stress components in terms of pure torsional and bending stresses comparable to corresponding modes of crank deformations, somewhat irrelevant.

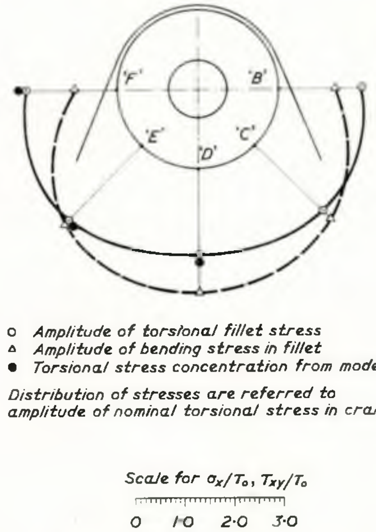


FIG. 11—*Distribution of dynamic stress amplitudes around fillet of crank No. 5 at 115 rev/min*

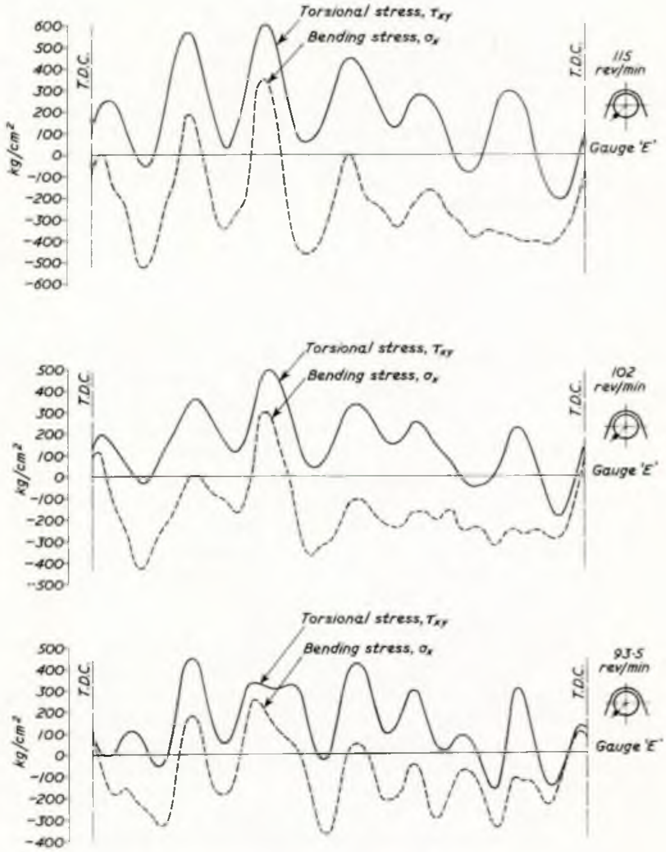


FIG. 12—*Simultaneous torsional and bending stresses in crank No. 5. Measurements in service conditions at sea*

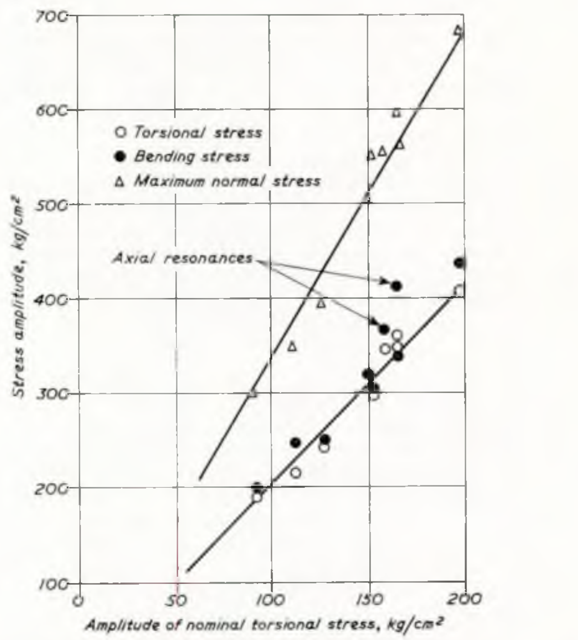


FIG. 13—*Relation between nominal torsional stress and fillet stresses at 45° to crank plane (gauge "E"), crank No. 5, at various loads and rev/min*

In Fig. 12 are shown the simultaneous torsional and "bending" stresses at the 45 deg.-position at several values of rev/min, taken from the measurements at service conditions at sea. Several records of this type are used to establish the diagram in Fig. 13, which seem to indicate a fairly linear relationship between the ranges of variation of the various stress components: nominal torsional stress, τ_0, torsional fillet stress, τ_{xy}, bending stress, σ_x, and the maximum normal stress, $\sigma\phi$. They may be expressed as follows:

$$
\begin{aligned}
\tau_{xy} &= 2\cdot12\cdot\tau_0 = \beta\cdot\tau_0 \\
\sigma_x &= 2\cdot12\cdot\tau_0 \\
\sigma\phi &= 3\cdot73\cdot\tau_0
\end{aligned}
\qquad (3)
$$

where β = stress concentration factor in torsion.

Axial vibration resonances do not seem to contribute significantly to the stressing at the $\pm$ 45 deg.-positions, although the deviation of some points from the straight line in Fig. 13 may be attributed to this cause. In the crank plane the bending stress is largely due to torsion-induced, forced axial vibrations.

Examining the direct stress variation at the edge of the crankweb (gauge "G" Fig. 10), it will be noticed that it is quite similar to the variation of nominal torsional stress, and that its stress level corresponds to a stress concentration in torsion of $\sigma/\tau_0 = 1\cdot9$, which is in good agreement with the value found by static tests, $\sigma/\tau_0 = 1\cdot97$.

According to the experience gained from the investigations on two widely different engines, the stress variations in the 45 deg.-position in the fillet may be of the same magnitude or even larger than those in the crank-plane fillet. This is particularly important to the strength of semi-built crankshafts, in which the superimposed system of residual stresses set up by the shrinkage forces tends to shift the critically-stressed zone away from the plane of symmetry of the crank (investigations into these residual stresses are described in Section III). The torsion and torsion-coupled bending which are the prevailing modes of stressing in this area, are therefore decisive to the strength of the crank.

In cranks Nos. 1 and 12 in the fore and aft end of the shaft, the bending and torsional stresses as measured on the test bed, are considerably lower than in crank No. 5. Their variation is in agreement with that to be anticipated from the action of radial and tangential forces, without any additional dynamic effects of importance. This situation may, however, be different in cases of first-mode axial resonance, particularly when coinciding with torsionals of the same mode and order. In such cases, the aftermost crank may become interesting from a strength and fatigue point of view. Here it refers to 1 and 12.

Theoretical Analyses of Stresses

Some theoretical analyses have been made to verify, by theory, the variations of fillet stresses found by the measurements, using the methods previously outlined and the necessary experimental data. The agreement between measured and calculated torsional stress is very good, using experimentally determined values for gas impulses and damping. The variations of torsional stress are determined by vectorial summation of forced-vibration components of harmonic orders 1-12 and including the constant torque components (see Fig. 14).

For the analyses of bending stresses, the Timoshenko method was used, with stiffness parameters for the crank throws computed by means of the beam theory and corrections for crankweb stiffness determined through model experiments. The variation of bending stress in crank No. 1 (the foremost) is reproduced very well (see Fig. 14). The calculation of fillet stress is based on the instantaneous values of bending moment in the middle of the crankweb thickness and the maximum stress concentration factors in bending, determined through model experiments. The complete crankshaft is incorporated in the calculation. The good agreement obtained upon this direct-bending stress calculation, is no doubt due to the absence of torque in this crank.

For crank No. 5, the agreements leave much to be desired when using the straight bending calculation. The introduction

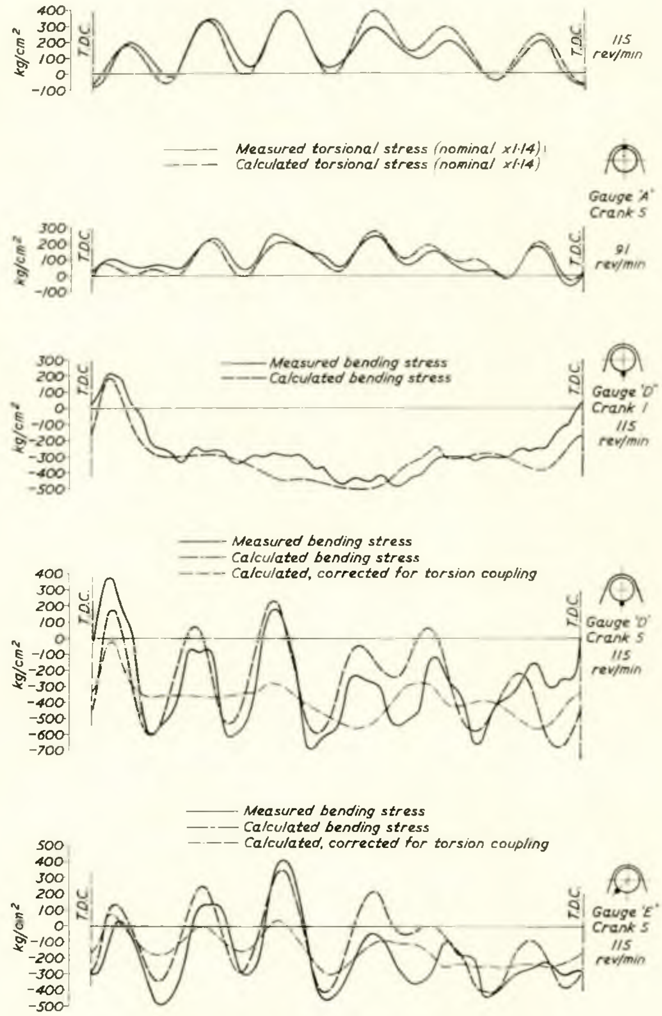


FIG. 14—*Comparison of calculated stresses and stresses measured on test-bed trials*

of bearing constraints, proportional to the angular deflexion of the journals, and loading the crank with the measured variable torque, was not sufficient to account for the torsion bending coupling of the stressing at the location 45 deg. off the crank plane, even by journals, nearly "encastré". By means of model experiments to be described in the following section, a purely local coupling between torsional stress τ_{xy}, and bending stress, σ_x, was found in this position with a stress concentration:

$$\sigma_x = 1\cdot12\cdot\tau_0 \qquad (4)$$

where τ_0 is the nominal torsional stress.

To obtain an approximate agreement with the measured variation, the bending stress can formally be written:

$$\sigma_x(t) = \sigma_0(t)\cdot\alpha + \tau_0(t)\cdot\gamma \qquad (5)$$

where:

$\sigma_0(t)$ = nominal stress calculated by the slope deflexion method, as a function of crank angle

$\tau_0(t)$ = nominal torsional stress, computed upon vectorial summation of harmonic components of resonance and flank stresses

α = stress concentration in bending

γ = factor of bending-torsion coupling.

In the plane of symmetry, at gauge D, the coupling factor has to be approximately $\gamma \simeq 2\cdot0$, and at gauge E in the 45 deg. plane $\gamma \simeq 1\cdot3$. In the latter location the value $\gamma = 1\cdot12$ was found by model experiments, and the stresses here can therefore be calculated in a fairly rational way, adequate for stress analysis of semi-built cranks.

The complete mechanism of this torsion coupling cannot be accounted for by the present theoretical methods. Until an adequate theory is available, experimental data for the coupling factor γ have to be relied upon. In cases of coinciding natural frequencies of torsional and axial vibrations, the torsion-induced axial vibrations may, however, be particularly strong, as shown by van Dort and Visser[10].

III—MODEL EXPERIMENTS WITH SEMI-BUILT CRANKSHAFTS

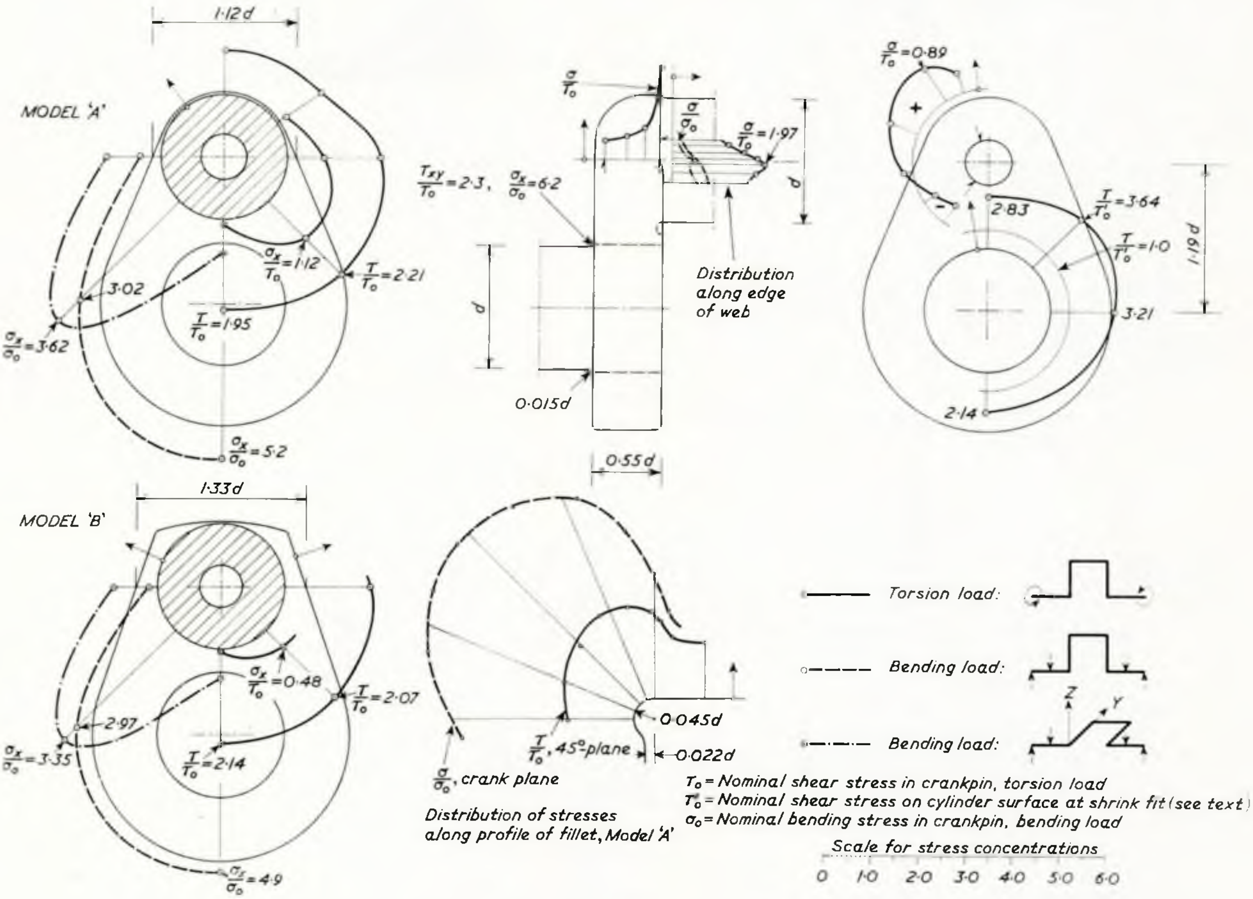
A large amount of systematic research work has been carried out by several investigators on stress concentrations in solid crank throws for marine applications. On the contrary, little is known about the particular problems of built crankshafts, that is, the shrinkage stresses and transmission of torque through the shrink-fit. To obtain some more knowledge in this field, as well as to enable a certain evaluation of the dynamic measurements of fillet stresses, described in the previous section, some experiments were carried out by means of strain gauge analyses on steel models with 100 mm shaft diameters, as well as three-dimensional photo-elastic analyses with 80 mm diameter shafts made of epoxy plastic. Great care was exercised to reproduce, in as great detail as possible, the configuration of the full-scale cranks of the dynamic investigations, with shrunk-in journals. Two varieties of cranks were tested, differing in the width of the webs.

The method used for theoretical calculation of fillet stresses requires the knowledge of stress concentration at a point due to torsion, bending in the crank plane and perpendicular to the crank plane with a constant bending moment. The loading devices were designed to avoid spurious forces and bending moments and produce pure bending or torsion.

In order to enable the determination of the peak fillet stresses, strain gauge rosettes of the foil type with very short filament length (0·7 mm effective grid length) were located along the fillet profile at several angular positions around the periphery of the crankpin.

For more detailed mapping of stress distribution, the three-dimensional photo-elastic stress analysis was employed, partly for the purpose of developing and evaluating practical methods of stress analysis on scale models of heavy machinery and structural parts.

For reliable results, the models, which are machined from heat treated epoxy-castings, have to be free from initial stresses which may introduce errors. With loads applied, the models are heated to 140° C (284° F) and slowly cooled down to room temperature in order to "freeze" the stresses, which are determined by cutting the model into slices for examination in polarized light. To avoid fracture and excessive distortion during the stress freezing cycle, great care must be exercised in design of loading devices and determination of load magnitude. In



Correction factors to fillet stresses for other proportions of models "A" and "B".

$$\text{Bending stress in crank plane: } K_\alpha = \frac{0{\cdot}11}{\left(\frac{r}{d}\right)^{0\cdot45} \cdot \left(\frac{t}{d}\right)^{1\cdot15} \cdot \left(1-\frac{do}{d}\right)^{0\cdot28}}$$

$$\text{Torsion stress in 45° plane: } K_\beta = \frac{0{\cdot}5}{\left(\frac{r}{d}\right)^{0\cdot2} \cdot \left(\frac{t}{d}\right)^{0\cdot0} \cdot \left(\frac{do}{d}\right)^{0\cdot07}}$$

(r = fillet radius t = web thickness d = crankpin bore diameter)

FIG. 15—*Distribution of stresses in semi-built crank throws with dimensions as shown*

order to analyse the stresses in sliced models, the standard polarizing microscope has been found ideal for accurate quantitative measurements. Once established, the photo-elastic method seems to be superior to strain gauges in analysing crankshaft stresses, expense, time consumption and obtainable information compared. The correspondence to the strain gauge method is very good.

Stresses in Bending and Torsion

Some results of the model tests are given in Fig. 15. A particular feature to be noted is the distribution of torsional stress around the fillet, and the associated normal stress in the axial direction as measured by the strain gauge filament in this direction. Any external forces, apart from those producing the pure torque, are precluded in the design of the loading rig, and this state of stress is therefore due to the asymmetry of the crank throw configuration. The importance of this phenomenon in explaining the dynamic behaviour and stressing of crank throws was mentioned earlier. As already shown in Fig. 11, the agreement with stress concentrations in torsion, found by means of the dynamic measurements of stresses, is very good.

The stressing of the fillets is naturally of the major importance to the design of crankshafts. However, in other locations around the crankweb the stress concentrations in torsion may attain the values of nominal stress and above, and should consequently be of some concern in evaluating the effects and severity of structural deficiencies in the castings and forgings.

The results of some measurements around the shrink-fits are also given in Fig. 15. As shown, the stress concentration in the transition fillet of the journal is considerable, but these stresses are believed to be of minor importance because this fillet must be assumed to be in a state of high compressive stress from the shrinkage forces, which increases the fatigue strength.

Shrink-fits

The simplified conception of torque transmission through a shrink-fit implies that twisting occurs when the torque T exceeds the moment of the frictional grip, as expressed by:

$$T > p.d^2.l.\pi.\mu.0{\cdot}5 \qquad (6)$$

where p = surface pressure according to thick cylinder theory
d = diameter of bore
l = length of bore
μ = coefficient of friction.

It supposes that the unit frictional force, $\mu.p$, is evenly distributed on the inter-surface in the process of slipping. Prior to slipping, however, the shrink-fit surface will be exposed to a unit shear load which is determined by the elastic properties of an equivalent monolithic web-pin assembly. This has been dealt with by Dr. Ørbeck[11], in connexion with his research into damping effect of shrink-fits.

To investigate this on the models, the shear stresses on the face of the web under torsional load of the steel crank were measured by means of strain gauges with very short filament length, located close to the inter-surface between pin and bore at the loaded end (the middles of the filaments were about one mm from the inter-surface). The distribution of this stress is shown in Fig. 15, and the average shear stress is:

$$\tau \simeq 2{\cdot}9\tau_0 \qquad (7)$$

where τ_0 is the nominal shear stress on a cylindrical surface with diameter d, and length l, concentric to the inter-surface:

$$\tau_0 = \frac{T{\cdot}2}{\pi{\cdot}d^2.l} \qquad (8)$$

Fretting at the ends may be incipient when the peak of the alternating torque exceeds $1/2{\cdot}9$ times the nominal slipping torque of the shrink-fit, as given in equation (6). The factor of safety should therefore be at least 2·9, basing the design of

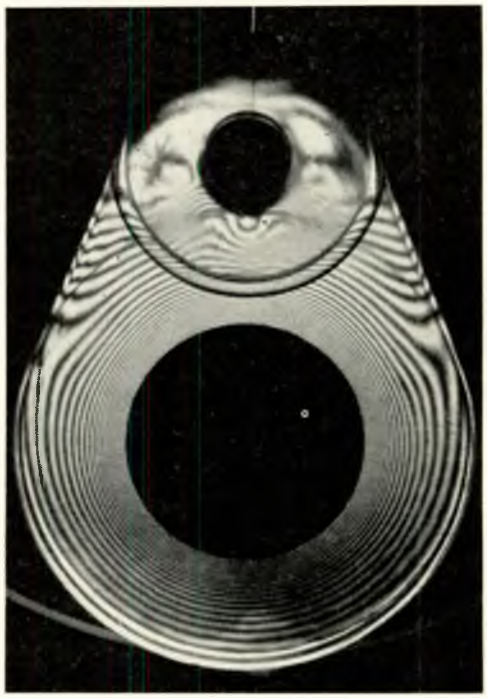


FIG. 16—*Isochromes in photo-elastic model of a shrunk crank throw*

the shrink-fit upon the peak values of the vibratory torque and to avoid fretting.

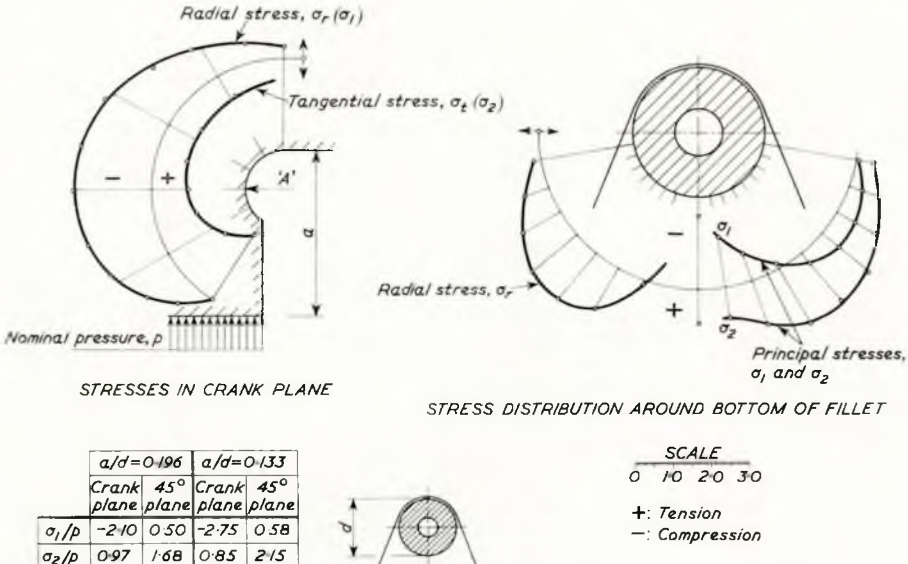
Equation (6) was used to analyse a shrink-fit which failed due to the passing of a major torsional critical c. approximately calculable magnitude, and it was found to fit in with a coefficient of friction in the order of $\mu \simeq 0{\cdot}18$—$0{\cdot}22$. Cases of twisted shrink-fits and cracks in the transition fillets of the journals have occurred where accidental overload is ruled out, the factor of safety, however, probably being less than the value 2·9 stated previously.

Shrinkage Stresses

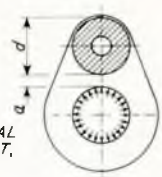
The distribution of stresses in the web and crankpin fillet due to the shrinkage forces in semi-built cranks has been investigated by means of the photo-elastic stress-freezing method. An oversize journal is cooled to −20°C (−4°F) and inserted into the web which is preheated to 130°C (266°F), the assembly then being submitted to a stress-freezing cycle.

In polarized light, a fringe pattern, as shown in Fig. 16, is produced and the detailed stress distribution is found by cutting the model into slices to be examined under a polarizing microscope. The corresponding stress distribution is shown in Fig. 17, where the stresses are referred to the mean, nominal surface pressure deduced from the thick cylinder theory. Of some interest are the components of radial and tangential stress, referred to the crankpin axis. In the crank plane and ± 20° away from it, the radial component is compressive. In the fillet, further from the crank plane, the radial component as well as the principal stresses change the sign into tension, as shown. The effect of the residual stresses in the crank plane is, in principle, to improve the fatigue strength in this area due to the large compressive stress superimposed on the variable stresses.

Around the 45 deg. region, however, its effect is reversed detrimentally, which will be evident when examining the distribution of principal stresses. Quantitatively, the effect of the residual stresses may be evaluated by criterion of fatigue fracture taking mean octahedral stress into account. With the usual surface pressures of shrink-fits, 1000-1400kg/cm², the stress components may attain values up to the yield limit in the fillet.



	a/d=0·196		a/d=0·133	
	Crank plane	45° plane	Crank plane	45° plane
σ_1/p	-2·10	0·50	-2·75	0·58
σ_2/p	0·97	1·68	0·85	2·15



VARIATION OF DISTANCE BETWEEN SHRINK FIT AND CRANKPIN (PRINCIPAL STRESSES AT THE BOTTOM OF FILLET, POSITION 'A', ARE GIVEN)

FIG. 17—*Residual stresses in the crankpin fillet due to the shrinkage of the journal into the web-bore*

Stiffness of Crank Throws

By means of crankweb deflexion measurements on the steel models, the following formula for the relation between nominal bending stress in crankpin and web deflexion was established:

$$f = \sigma \cdot 1 \cdot 91 \cdot 10^{-6} \cdot R\left[l/d\left(1 + \frac{a}{R}\right) + \left(\frac{R + 2a}{d}\right) \cdot 0 \cdot 56 \frac{d^4}{t^3 b}\right] \quad (9)$$

where f = crankweb deflexion (mm)
$\pm\sigma$ = nominal bending stress in crankpin (kg/cm^2)
R = crank radius (mm)
d = diameter of crankpin (mm)
l = length of crankpin (mm)
t = thickness of web (mm)
b = width of web, in way of crank plane fillet (mm)
a = position of gauge, measured from centre of journal (mm).

The formula is believed to be valid for the usual types of semi-built crankshafts. The deflexion f, is taken to be the maximum difference read on a dial gauge during one revolution of the crankshaft.

IV.—ANALYSIS OF CRANKSHAFT STRENGTH

Apparently being able to predict the critical stresses in large crankshafts with fairly good accuracy, one might perhaps raise the question of the practical significance aside from some satisfaction of academic interests. The establishment of factors of safety is a doubtful exercise due to the variable quality of, and hidden defects in, the steels which go into large crankshafts. Anyhow, a detailed knowledge of the stressing may offer some assistance to those looking for the defects by means of destructive and non-destructive inspection methods, and might enable a rational evaluation of them. It is a fact that many cracks nucleate in areas containing serious structural defects, not necessarily at unquestionably severe stress raisers. The stressing of these crankshafts being largely a matter of torsion, the variable torsional stress may constitute a suitable basis of comparison.

Twenty-five shafts have been analysed for amplitude values of torsional stress, using theoretical impulses and damping factors according to Maciotta[5] when specific data are unavailable. Crankshafts with trouble-free records, chosen at random, as well as cases of fracture are included in the analyses, the results of which are given as histograms in Fig. 18. The stresses given are the largest within the speed range believed to

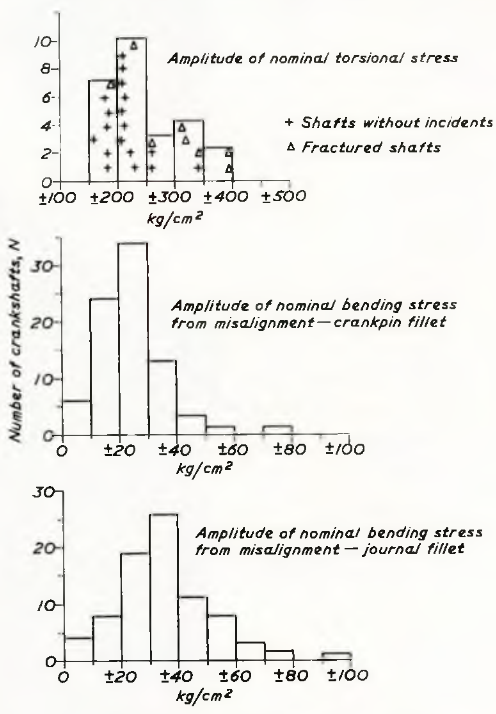


FIG. 18—*Analysis of stresses in crankshafts for engines in service*

be the normal service revolutions. The stressing of the fractured shafts is apparently on the high side, but fractures may well occur at quite moderate levels of torsional stress. This may happen when major structural defects or initial cracks are involved. Cases of repeated cracking are however, usually associated with the higher stress levels and with crack initiation at severe stress raisers. For instance, a stress level of approximately $\pm$ 400 kg/cm^2 seems to be a realistic fatigue limit for large, mild steel shafts with lubrication holes.

Furthermore, the additional stresses due to misalignment are analysed for 89 engines in service, based upon reported web deflexions and the relation between deflexion and nominal bending stress at the middle of crankpins, derived in the preceding section (the distributions of bending stress up to the chain drive have been found by supposing the foremost journal to be freely supported, the bending moments at the middle of crankpins being computed from their web deflexion readings and assuming a linear distribution of bending moment between journal supports). None of the higher-stressed shafts has suffered any casualties, and it may seem as if fairly severe misalignment is not necessarily a catastrophe.

Some failures at lower stress levels have been initiated at major structural deficiencies, such as solidification cracks and hot tears, cracks opened by accidental overload, and large inclusions and voids. In such cases, cracks may start at locations where stress concentrations usually are considered to be low and a certain relaxation in the requirements to structural soundness have been practised.

To assess the effect of structural defects, it is necessary to have some knowledge on the stressing of crankwebs, where structural defects most frequently appear. In torsion, the webs may in some locations have stress concentration factors in the order of 1·0-2·0 as shown previously, which may be enough to propagate initial cracks in otherwise moderately-stressed cranks. In Fig. 19, a typical example of a crack is shown which initiated at the edge of the crankweb which has been repaired by extensive unauthorized welding of extremely bad quality.

The science of microfractography is now increasingly being put to use to shed some more light on the mechanism of fracture initiation and propagation, and the role of stress levels and structural defects in this process. Some practical results have been obtained through this method, which is now a routine procedure in the examination of fractured machine members in the Metallurgical Department of Det norske Veritas.

The stress level and its history are decisive for certain features in the topography of the crack surface, which is revealed through the strong magnification obtainable in an electron microscope (usually 7000-10 000 times). To illustrate the method, a case of fracture of a large, cast steel crank throw will be dealt with.

The fracture initiated at a point on the crankweb exposed to a practically uni-axial service stress with a peak value of approximately 540 kg/cm^2 (calculated, with some support from measurements).

Initially, the crank had been exposed to an accidental overload of short duration with stresses exceeding the fatigue limit. Under the subsequent normal service, a fatigue crack developed. The surface around the crack origin, which is shown in Fig. 20(a), appeared to have three zones of distinctly different appearance. Zone 1 which is the nucleus of the crack, contains segregations of inclusions in connexion with hot tear cracks. The microsurface of Zone 2 shown in Fig. 20(b) is characterized by the dimples which are formed by local plastic ruptures under a high nominal stress. Zone 3 has the normal appearance of slowly propagating cracks. The striations which are found in this zone (see Fig. 20 [c]) are due to the stepped propagation of the crack front under a moderate alternating stress.

The conclusion to be derived from this investigation is

FIG. 19—*Fracture initiated at the edge of the crankweb of a small, cast steel crank throw (semi-built)*

that the crack initiated due to overstressing of short duration, in the zone containing structural deficiencies. This incident opened an initial sub-surface crack (Zone 2) with a characteristic extension of approximately 22 mm, which subsequently propagated as on ordinary fatigue crack under the normal service stresses.

Very similar developments of fatigue cracks from initial cracks have been found in other cases, where the findings of microfractography correspond well with the likely stress history. An attempt has been made to correlate the size of the initial cracks and the corresponding estimated propagative peak

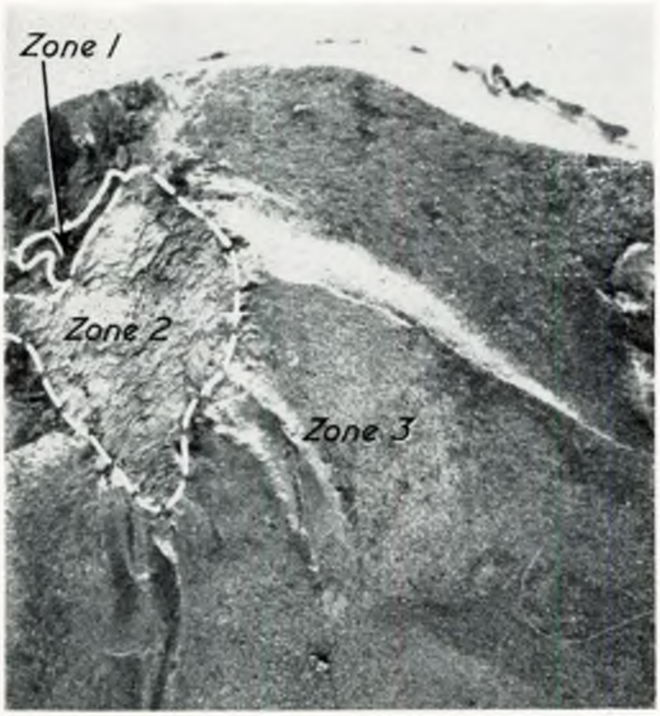


FIG. 20(a)—*Area of crack initiation on a large, broken crank throw*

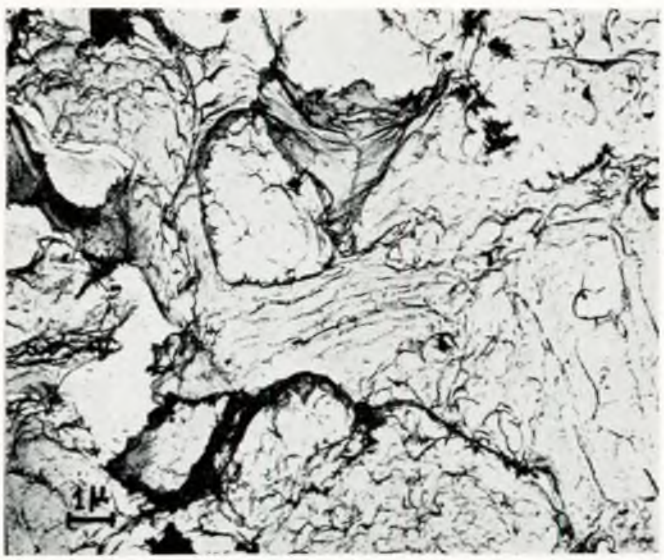


FIG. 20(b)—*Microfractographic photograph of crack-surface in zone 2, with dimples formed by high stresses and fast crack propagation*

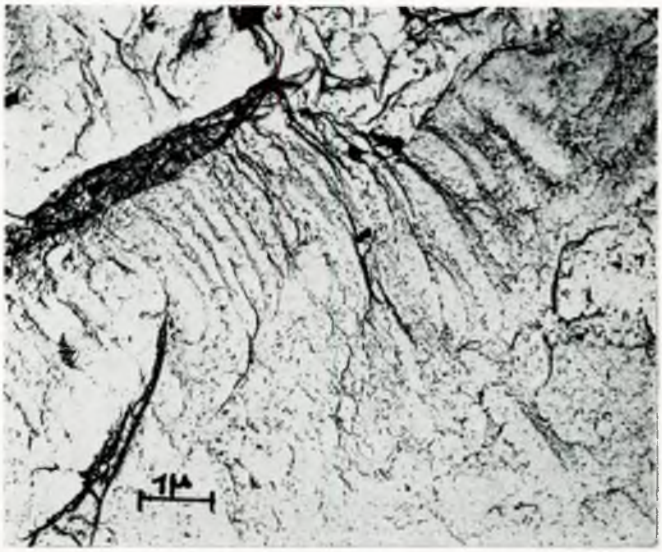


FIG. 20(c)—*Microfractographic photograph from zone 3, showing striations left by the front of a normally propagating fatigue crack at a moderate stress level*

stresses with the theory of crack propagation established by Frost[4] (see Fig. 21).

The stress level of the first case mentioned exceeds the propagative stress predicted according to Frost, which may be due to the mainly sub-surface location of the initial crack. This may be explained by the theory of elasticity according to which the

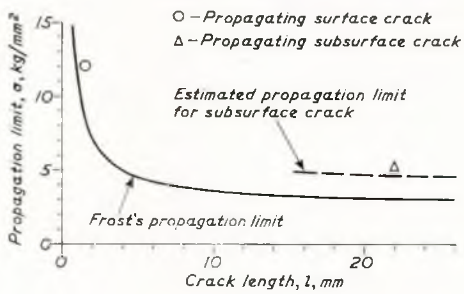


FIG. 21—*Cases of initial cracks in crankshafts which have propagated to failure. The estimated service stresses are compared with Frost's experimentally determined stress propagation limit*

ratio of peak stresses of the pre-cracked specimens, used in Frost's experiments, to that of an entirely enclosed crack of corresponding dimensions, is in the order of 0·65.

Although more practical verification is desirable, Frost's theory seems potentially useful for a quantitative evaluation of defects and an estimation of probability of crack propagation.

A further examination of the fatigue strength of steels for large crankshafts is given in Appendix II.

CONCLUDING REMARKS

The experimental investigations and their theoretical analyses described in this paper, seem to indicate that large crankshafts in general are quite amenable to a theoretical stress analysis with an accuracy satisfactory for practical applications. Particularly, the calculation of torsional stress yields results corresponding very well with measurements, even with theoretically derived impulses. In cases of resonances, however, data for damping should be available. Theoretical calculation of bending stresses is in need of some refinements to include coupling effects between torsion and axial deformation. For practical purposes, it is believed to be sufficient to base the stress analysis of large shafts on torsional stresses and using empirical relations between bending and torsional stress.

To utilize a knowledge of stressing, data are necessary for the fatigue strength, eventually the crack propagation stress of the steel in relation to the types, size and frequency of structural defects to be expected from the particular methods of production. Thus, the principal problems of producing crankshafts economically, and with statistical controllable lower limits of fatigue strength, remain with the metallurgists. Some research in these fields is in progress, in co-operation with manufacturers of crankshafts.

It is hoped that the kinds of investigations described in this paper may contribute to the design of economical crankshafts with optimum reliability, taking the problems of materials, manufacture and inspection into account. As to the stress analysis of crankshafts, it may be said that some more experimental data from other types of large engines are desirable, to outweigh the shortcomings of the purely theoretical calculations of bending stresses. Knowing what to look for, this approach may pay off better than theorizing the problem into obscurity.

ACKNOWLEDGMENTS

The experimental investigations have been made possible by keen co-operation of the engine manufacturers and shipowners in giving access to their engines, and the crankshaft manufacturer for supplying crankshaft models of remarkable quality. The valuable suggestions, encouragements and sharing of experience with Civ.ing A. Bj. Nielsen are particularly appreciated. The theoretical and experimental analyses are the result of teamwork by members of the research staff at the Research Department of Det norske Veritas. Substantial contributions to the project have been given by Siv.ing. O. Larsen, who has programmed and carried out the computer calculations, by Dipl.-ing. M. Gibstein, with his careful analyses of dynamic recordings and photo-elastic models, and Dipl.-ing. Nowinsky, who assisted in the preparation of the drawings.

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APPENDIX I

FORCED TORSIONAL VIBRATIONS IN VARIABLE MASS SYSTEMS

Taking the time-variable mass of the running gear of the individual cranks into account, Draminsky[3] has brought attention to the important phenomenon of secondary resonance which in some cases may amplify resonance stresses many times beyond those predicted by the constant mass theory of torsional vibrations. These secondary resonances may be of importance at critical revolutions which have large, adjacent resonance peaks of orders ± 2 with respect to the one examined. A need was felt to account for this effect in computer calculations of forced vibrations, and Draminsky's equation of motion for a variable mass system has been extended to a damped multi-mass system with an arbitrary number of variable masses.

For any mass, *s*, in the torsional system, the equation of motion can be written:

$$I_s(1 - \beta \cdot \cos 2\omega_o t) \cdot \ddot{\theta}_s + 2\beta \cdot I_s \cdot \omega_o \cdot \sin 2\omega_o t \cdot \dot{\theta}_s \qquad (1a)$$

$$+ D_s \cdot \dot{\theta}_s + k_{s,\,s+1}(\theta_s - \theta_{s+1}) + k_{s-1,\,s}(\theta_s - \theta_{s-1}) = \sum_{1-r}^{\infty} T_{s,r+m}\, e^{j\omega_o(r+m)t}$$

or:

$$I_s\left(1 - \tfrac{1}{2}\beta\left[e^{2j\omega_o t} + e^{-2j\omega_o t}\right]\right)\cdot\ddot{\theta}_s$$
$$+ j\omega_o\beta I_s\left(e^{-2j\omega_o t} - e^{2j\omega_o t}\right)\cdot\dot{\theta}_s + D_s \cdot \dot{\theta}_s \qquad (1b)$$
$$+ k_{s,\,s+1}(\theta_s - \theta_{s+1}) + k_{s-1,\,s}(\theta_s - \theta_{s-1}) = \sum_{1-r}^{\infty} T_{s,r+m}\, e^{j\omega_o(r+m)t}.$$

In this equation:

I_s = mean moment of inertia
β = coefficient of mass variation of a crank
D_s = damping coefficient
$k_{s,s+1}$ = stiffness of shafting between mass s and s + 1
$k_{s-1,s}$ = stiffness of shafting between mass s and s − 1
ω_o = angular velocity of rotation
r = order of primary impulse
t = time variable
j = $\sqrt{-1}$
$T_{s,r+m}$ = complex impulse vector of order r+m
θ_s = complex time variable vibration-deflexion at mass s (suffixes s+1, s−1 designate those of the adjacent masses).

The steady state solution of equation (1) is an infinite series of the form:

$$\theta_s = \sum_{n=-\infty}^{+\infty} F_{s,2n} \cdot e^{j(r+2n)\omega_o t}$$

where $F_{s,2n}$ is the complex vector of vibration. To render a numerical treatment manageable, a truncated expression is assumed to yield a solution of sufficient accuracy:

$$\theta_s = F_{s,-2} \cdot e^{j(r-2)\omega_o t} + F_{s,o} \cdot e^{jr\omega_o t} + F_{s,2} \cdot e^{j(r+2)\omega_o t} \qquad (2)$$

This solution and its derivatives are inserted in equation (1b), and equating coefficients of exponential terms of equal exponential order ($r-2$, r, $r+2$) a set of simultaneous, complex equations is obtained, for an order of primary excitation r:

$$A_s(r-2)^2 \cdot F_{s,-2} + B_s \cdot r^2 F_{s,o} + C_s \cdot r \cdot F_{s,o} + jE_s(r-2) \cdot F_{s,-2} + k_{s,s+1}(F_{s,-2} - F_{s+1,-2}) + k_{s,s-1}(F_{s,-2} - F_{s-1,-2}) = T_{s,r-2} \qquad (3a)$$

$$A_s r^2 F_{s,o} + B_s(r-2)^2 F_{s,-2} + B_s(r+2)^2 F_{s,2} + C_s(r+2)F_{s,2} + C_s(r-2) \cdot F_{s,-2} + jE_s r F_{s,o} + k_{s,s+1}(F_{s,o} - F_{s+1,o}) + k_{s,s-1}(F_{s,o} - F_{s-1,o}) = T_{s,r} \qquad (3b)$$

$$A_s(r+2)^2 \cdot F_{s,2} + B_s r^2 F_{s,o} - C_s r F_{s,o} + jE_s(r+2)F_{s,2} + k_{s,s+1}(F_{s,2} - F_{s+1,2}) + k_{s,s-1}(F_{s,2} - F_{s-1,2}) = T_{s,r+2} \qquad (3c)$$

where:

$$A_s = -I_s\omega_o^2$$
$$B_s = \tfrac{1}{2} I_s\beta\,\omega_o^2$$
$$C_s = -I_s\beta\omega_o^2$$
$$E_s = D_s \cdot \omega_o$$

Expanding the foregoing equations to incorporate all the masses in a system, gives a set of real linear equations (six for each variable mass and two for each constant mass), with a corresponding number of unknowns which can be rapidly solved by rational matrix methods on a high-speed computer. The solutions, in terms of complex vibration vectors, are dealt with in a way similar to that of a constant mass forced calculation, to give torsional stresses. The method was used to calculate the resonance and flank stresses of a seven-cylinder two-stroke engine with the shafting system shown in Fig. 22. For comparison, a constant mass flank stress calculation was carried out, using the same method and equations, but putting the coefficient of mass variation, β equal to zero. This is equivalent to an ordinary forced vibration calculation of a system with constant masses. As a simplified approach, the flank stresses in the variable mass system may be approximated by introducing a correction factor derived from Draminsky's method for the equivalent one-mass system. The calculated stresses are compared with stresses measured with the torsiograph (the 7th order stress is not quite accurate, as the record contains components of other orders).

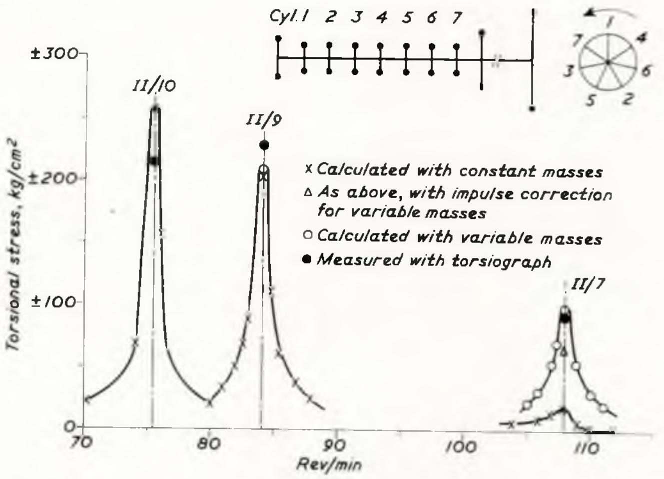


FIG. 22—*Calculation of forced, torsional resonances and flank vibrations with constant and variable masses*

APPENDIX II

FATIGUE STRENGTH OF CRANKSHAFTS

The fatigue strength of crankshaft steels is largely a question of which mechanism of fatigue fracture is at work and the presence of inherent, structural defects in the material. For small, forged shafts the stress necessary to nucleate a microcrack and henceforth propagate it, is probably a reasonable definition. For large sections and volumes of steel with increased liability to all types of inherent flaws, the fatigue strength may tend to the limit given by the stress necessary to propagate initial cracks and defects. In this case, the fatigue strength may vary according to the size and nature of the fracture-initiating defects. Fig. 23 may serve to illustrate the situation. In Fig. 23 are given the results of fatigue tests in rotary bending with 65 mm diameter cast steel specimens, taken from large-section castings. The average fatigue limit seems to be $\pm$ 15-16 kg/mm², but the scatter is considerable due to the presence of flaws in the specimens. There is apparently a need for statistical treatment of fatigue strength and its relation to the flaws, in order to warrant an assessment of the fatigue properties of large volumes of different types of steels. Aside from the methods of manufacture, the methods and principles of detection and rejection of structural defects in stressed areas should obviously be brought into these considerations.

Those values to be expected for the fatigue limit of steels with structural defects, may be indicated by some published results of alternating direct stress fatigue tests with large, cylindrical specimens having artificial defects: specimens with artificial surface porosity (drilled grid of 0·3 mm diameter holes):

$\sigma_A = \pm$ 14·6 kg/mm² (reference [8]);

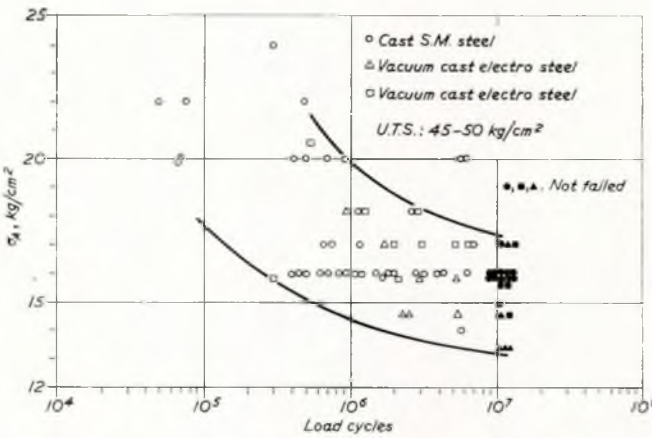


FIG. 23—*Fatigue test in rotary bending with 65mm diameter cylindrical specimens from three different steel works*

specimens with weld deposits containing inclusions:
$\sigma_A = \pm 10{\cdot}5 - 13{\cdot}7$ kg/mm^2 (references [6] [8]);

specimens with artificial microcracks on the surface:
$\sigma_A = \pm 11{\cdot}0$ kg/mm^2 (reference [7]).

Reference (12) which gives some results from fatigue tests with large specimens having various types of defects, also bears on this.

The ratio between fatigue strength in alternating torsion and bending, τ_A/σ_A, seems to be $0{\cdot}71 - 0{\cdot}77$ for inhomogeneous or porous steel, and probably $0{\cdot}85 - 1{\cdot}00$ for surface cracked specimens. The apparent fact that the fatigue strength in torsion and shear is less affected by flaws than the bending fatigue limit, may be one reason for the fairly low incidence of failure of large crankshafts, which are primarily stressed by torsion.

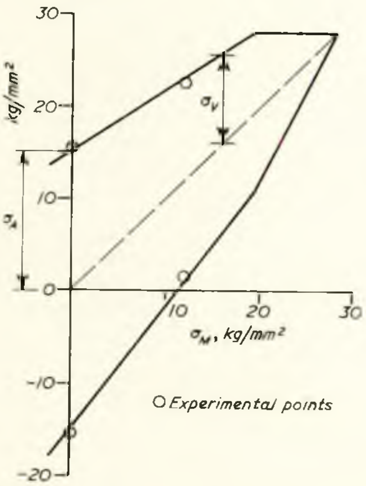


FIG. 24—*Goodman diagram for cast mild steel of average quality*

At variable levels of mean stress, the Goodman diagram shown in Fig. 24 is a good fit for the fatigue limits of cast steel of average quality. For the combined state of stress, the simple elliptical criterion of failure seems to describe failures in the fillets fairly well:

$$\left(\frac{\sigma}{\sigma_V}\right)^2 + \left(\frac{\tau}{\tau_A}\right)^2 = 1. \qquad (1)$$

or, introducing the alternating fatigue limit according to the Goodman diagram:

$$\left[\frac{\sigma}{\sigma_A\left(1 - \frac{\sigma_M}{\sigma_B}\right)}\right]^2 + \left[\frac{\tau}{\tau_A}\right]^2 = 1. \qquad (2)$$

In the foregoing equations:

$\pm\sigma$ = bending stress in fillet;
$\pm\tau$ = torsional stress in fillet;
σ_M = mean, direct stress;
$\pm\sigma_A$ = fatigue limit by alternating direct stress;
$\pm\sigma_V$ = fatigue limit by mean stress σ_M;
$\pm\tau_A$ = fatigue limit by alternating torsion;
σ_B = ultimate tensile strength.

For the case of fracture of a semi-built crank throw, in which the crack started about 45° away from the crank plane in the crankpin fillet, the maximum stress in the normal service range of revolutions has been analysed on the basis of measured impulses and damping, and assuming the principles of stressing outlined previously to be valid. The stresses are, including concentrations, approximately:

Torsional stress: $\tau = \pm 650$ kg/cm^2;
Bending stress: $\sigma = \pm 700$ kg/cm^2.

At the location of fracture initiation, the mean, residual tensile stress due to shrinkage of the journal may be estimated from the results of the photo-elastic measurements described in Section III:

$$\sigma_M = 700 \text{ kg/cm}^2.$$

Using equation (2), $\sigma_B = 50$ kg/mm^2 and assuming $\tau_A = 0{\cdot}75 \cdot \sigma_A$, the corresponding alternating stress fatigue limit of the steel would be:

$$\sigma_A = \pm 1190 \text{ kg/cm}^2.$$

It is emphasized that these calculations are highly approximate and, particularly, the treatment of the mean stress due to shrinkage is disputable. In the foregoing, the octahedral mean stress is used:

$$\sigma_M = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3} = 700 \text{ kg/mm}^2 \qquad (3)$$

(σ_1, σ_2 and σ_3 are principal stresses).
However, if the mean stress is computed according to von Mise's criterion, it amounts to:

$$\sigma_M = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2} = 1250 \text{ kg/cm}^2. \qquad (4)$$

This yields an alternating stress fatigue limit of the steel:

$$\sigma_A = 1280 \text{ kg/cm}^2.$$

These values of σ_A may fit well with the quality of the steel met with in this, as it contained some defects at the crack start.

Discussion

Dr. S. Archer, M.Sc. (Member of Council), opening the discussion, said that the Institute was fortunate to have the opportunity of discussing this most important paper, which he had no doubt would come to be regarded as a classic, on this fundamental subject, for Diesel engineers. It followed a number of papers in recent years dealing with crankshaft design and fatigue strength, in particular perhaps those of Anderson, Olsson and their colleagues in 1962, Pasetti in 1963 and—if he might be forgiven for the reference—his own paper in 1963, giving some of the results of the work of Lloyd's Register of Shipping in the same field (see references (1), (2) and (8) of the paper).

Mr. Langballe and his colleagues had taken the subject a great deal further, inasmuch as they had actually measured the dynamic stresses of large crankshafts in service, both at sea and in the shops. They had also supported this work by other tests, both on smaller crankshafts with strain gauges, as shown in the paper, by photo-elastic methods, and by measurements of other kinds. Consequently the paper contained a vast amount of information and, unfortunately, within the compass of a few pages, it would be impossible for the author to set out in detail all the various steps leading to his various results. This involved some difficulty for people wishing to discuss the paper in detail. He was sure that when people had the opportunity of studying the paper in greater detail they would know a great deal more about the stress conditions and the factors of safety applicable to the modern crankshaft.

On page 498, he was interested to see that, in order to overcome the problem of computerizing gas-pressure values, polytropic expansion and compression curves had been assumed, together with suitable mathematical expressions for the combustion and scavenging processes; would the author be prepared to indicate what these were? They might be of value to others.

The author had not mentioned, in his presentation, the Draminsky effect referred to in Appendix I. Dr. Archer said he was very interested in the work done by the author on this aspect. It was a relatively new subject to most vibration engineers, particularly so far as large crankshafts were concerned. Draminsky had developed his theory for a single-mass system only and used an approximate method to obtain results for multi-mass systems, whereas the author had carried the mathematics a good deal further, for he had applied the variable mass theory directly to a multi-mass system right from the start.

Referring to the example given in Appendix I, Dr. Archer noted that the two-node, 7th order critical given in Fig. 22 (open circles) was the stress calculated with variable masses, presumably by the theory given, and that measured by torsiograph was shown by the black dot. However, the author stated that this had not been corrected for the presence of 5th order components. It seemed a pity that this had not been analysed out, since one would like to have seen the extent of the increase in dynamic magnification of the two-node 7th order alone. Had the author done this and, if so, what did it amount to?

In the paper on the work done by Lloyd's Register of Shipping, two cases out of four, they knew had been reported. The first was a two-node, 8th order, in which the measured stress of torsiograph was five times that calculated with constant masses; the second case was a two-node, 9th order, in which it was three times that calculated. Looking at the author's two-node, 7th, this would appear to be somewhere about three to four times, which was of roughly the same order. This could be a very important effect, especially if this apparently minor critical should occur at or near service speed, as so frequently seemed to happen.

He was interested in the method of establishing beyond reasonable doubt that the crank pin transmitted the full torque. Some years ago he had made a rough calculation on this. He had compared the lateral bending deflexion of the webs due to transmitted torque and the magnified torsional deflexion of the pin with the main bearing clearance. He had calculated that the clearance would have to be very tight indeed, if the pin was not to take the full torque, and he was glad to have confirmation of this.

The author, in Fig. 1 (c)—Torsional stress variations in crank Nos. 4 and 8—used the theoretical impulses and coefficients of cylinder damping according to Maciotta. This, he took to be the paper (author's reference 5) that he and his colleague Merlino, had read to C.I.M.A.C. in 1965. He (Dr. Archer) and his colleagues had analysed the results in that paper and had compared them with their own measurements, calculating the corresponding damping factors. They had found that, in general, the Maciotta engine damping factors seemed to be much too low and had therefore given an unduly high calculated crankshaft stress compared with measured volumes. He would be glad to know whether Mr. Langballe had checked any of the Maciotta results against his own measurements.

In Fig. 1(c) he took it that the stress concentration factor should be 1·85 and not 0·85.

Dr. Archer said that in Fig. 1(d), he assumed that the three-lobed full-line graph was that applicable to the three-crank system for the comparison of bending-stress variation in crank No. 4, although the legend did not seem to indicate that in the figure.

Dr. Archer thought it was a little difficult to follow the author's description of the forces coming on a single crank, mentioned on page 498 under the heading "Bending Stresses". The author had quoted Timoshenko's work on this. For those unfamiliar with it, it might have been useful if a small sketch had been shown in the paper indicating the unit forces and the torques to which the author referred.

On page 500, dealing with measurement techniques and showing the position of strain gauges, particularly Fig. 5, he was not clear from the sketch exactly where the strain gauge "A" was attached. Was it on top of the pin on the fillet? Dr. Archer also asked whether they were all in the same transverse plane.

Dr. Archer said that he took it that Mr. Langballe's recorder was only capable of six channels and that therefore he had had to repeat his tests for each separate rosette to ensure that he had exactly the same conditions during each of these measurements, of which he must have taken a very great number, or possibly he was able to vary the engine speed gradually for each rosette and pick off corresponding speeds. Perhaps the author would explain how he had done this.

It was interesting to read on page 502, dealing with the experiments on axial vibration modes, that the author had found that the masses of the connecting rods did not seem to participate in the vibration. This was something which had been suspected for some time, but there had hitherto been no real confirmation. This was certainly a step forward in the theory of axial vibration. It was curious to note that thrust bearing stiffness seemed lower for the second mode than for the first mode. Probably the author's explanation was correct.

In Fig. 15, the author referred to distribution of stresses in semi-built crank throws. This was a most important piece of work. It seemed to throw a great deal of light on why it was that these shafts failed where they did in service. It was interesting also to note the photo-elastic investigations which seemed to confirm that the material of the web immediately under the pin was in a state of radial compression due to shrinkage. It might help to explain why it was that some designs had had a very small radial distance between the pin and the bore of the web and had still seemed to operate satisfactorily. Indeed, he knew of some cases in which it had actually been negative, with the pin surface below the bore of the web. This was something that had initially been viewed with considerable alarm. Probably the explanation was on the lines the author suggested.

The author then dealt with shrinkage stresses in the 45° region where reversal of radial stress occurred. This radial tension was clearly at a dangerous position where so many fractures started.

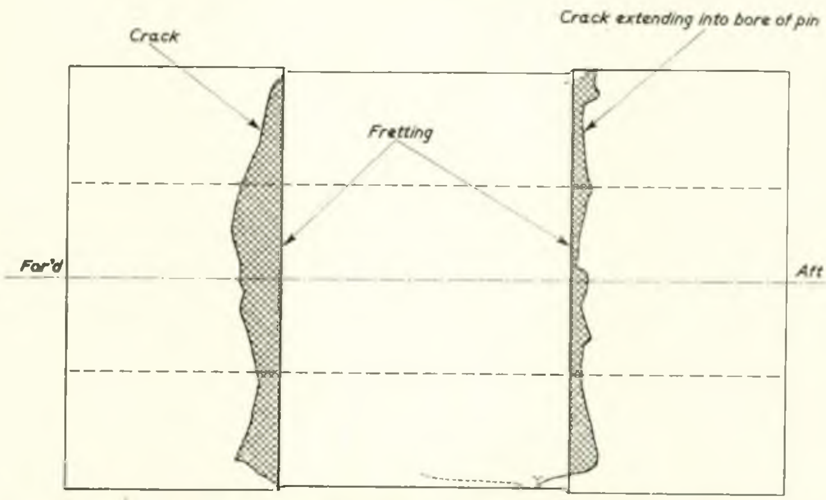
On page 507, author said that cases of repeated cracking were usually associated with higher stress levels and with crack initiation at severe stress raisers. He added that, for example, a stress level of approximately $\pm$ 400 kg/cm^2 seemed to be a realistic fatigue limit for large mild steel shafts with lubrication holes. This was presumably nominal torsional stress and, if so, compared very well with the earlier work of Lehr and Ruef* on some forged 245 mm diameter carbon steel shafts (St.C.35,61) hollow-bored pins and journals, where they had got a limiting value of $\pm$ 420 kg/cm^2 in pure torsional fatigue. Dr. Archer himself fully supported the assumption that torsional stresses were by far the most important influence on the fatigue life of crankshafts, as he had also indicated in his own paper.

He had been surprised and somewhat relieved to note that the effect of misalignment did not appear to be as important as people hitherto had assumed. However, more work was needed on this, as probably there were many cases of failure where misalignment, much more severe than the author had mentioned, had taken place.

The author had used Gough's elliptical formula for his equivalent stress criterion. Dr. Archer supposed that this was as good a theory as any and seemed to have been supported by earlier experimental work.

He had been struck by the microfractographs in Fig. 20 (a, b and c), which represented a relatively new development of the use of electron microscopes for the examination of material failures, and the author's description of the likely causes of the

* Lehr, E. and Ruef, F. 1943 "Fatigue Strength of Crankshafts of Large Diesels". M.T.Z., Vol. 5, Nos. 11/12, p. 349.



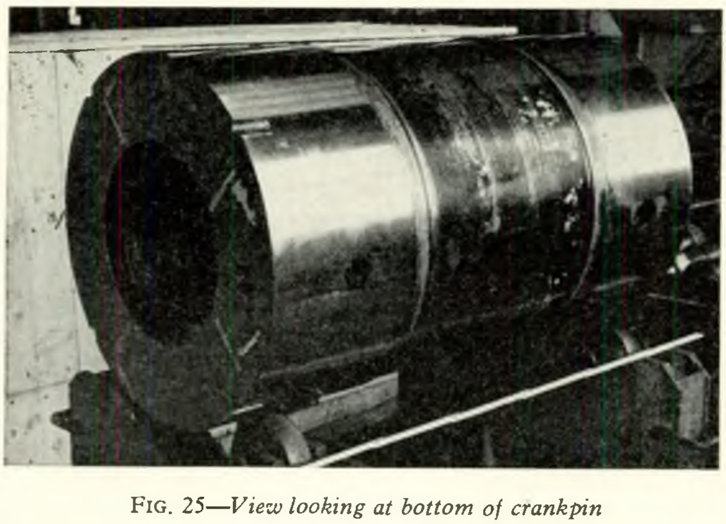


FIG. 25—*View looking at bottom of crankpin*

particular failure illustrated was informative. This, Dr. Archer believed, was the shaft which was supposed to have run away during the test-bed trials. Whether it had run through torsional criticals, he did not know. But at least he understood that it had reached quite high speeds and had been sufficient to slip the shrinks on some of the throws. The author had said that the crank surface in zone 2, shown in Fig. 20(b) was characterized by dimples formed by local plastic ruptures under high nominal stress. Dr. Archer was not quite clear from the photograph what the author would call "dimples". Perhaps he could state what he meant in his reply.

It was interesting to see the comparison with Frost's work on the stress required to propagate cracks of a given initial length. The suggested reason for the values obtained for subsurface cracks, which were higher than those for surface cracks, as shown in Fig. 21, seemed fairly reasonable, although Frost's work had been done mainly on plate material.

Had the author much experience of fretting fatigue cracks in shrinks? Recently, Dr. Archer and his colleagues had come across a case of a nine-cylinder two-stroke Diesel, which after 14 years, had developed two cracks in the crankpin, but originating inside the shrink. The cracks which occurred on the under side of the pin, had been found to follow almost exactly the limits of the fretting corrosion which had taken place and had propagated into the pin-bearing surface on one side. This was a quite exceptional example showing the dangerous effects of fretting.

Dr. F. Ørbeck said that the author had presented a most interesting paper on stresses in crankshafts and behind it one could visualize a high degree of organization and skill in both the theoretical and the experimental work. It was undoubtedly the right approach to study the problem both theoretically and experimentally, as the comparison of calculated and measured results gave a good indication of present achievements and where further improvements could be made.

There had been a considerable change in the study of crankshaft stresses since the Second World War, aided by the use of computers and improved measurement techniques and earlier theoretical works had gained in practical importance. It was interesting, for instance, that the work by Timoshenko, which the author had chosen as a basis for his bending calculations, had been carried out long before the war, although the author quoted a reference dated 1953. These calculations were, however, so complicated and tedious that only the introduction of computers could yield any practical results. Some results from his own (Dr. Ørbeck's) Ph.D. thesis (reference 11 of the paper) might throw more light on the accuracy of Timoshenko's method and the validity of some of the approximations made on page 499 of the paper.

An accurate analysis of the elastic behaviour of the crankshaft of a six-cylinder high-speed engine was required. Timoshenko's method was used and almost exactly the same approximations listed on page 499 of the paper were also made for the purpose of his own calculations. Having found the forces on the main bearings, the displacements of the journals were calculated from available journal-bearing theory. The centrifugal forces on the crankshaft and the forces due to the 5th and 7th orders of torsional vibrations were considered.

The displacement of the journal in main bearing No. 4 was later measured by means of capacitive transducers built into the bearing. The agreement between calculation and measurements was surprisingly good with the measured displacements being on an average ten per cent lower than the calculated. These tests had been carried out on a high-speed engine which would have a stiffer crankshaft compared with the bearing clearance than a slow-speed engine and it was therefore to be expected that the approximations were more accurate for the slow-speed engines. This applied mainly to assumptions (a) and (b) on page 499 of the paper.

The author had mentioned these approximations in order to explain the considerable discrepancies shown in Fig. 1 (d). A suggestion on this point might be of value. It would be seen that the measured bending stress in the crank-plane fillet contained a considerable component of apparently 2nd order. The engine, however, was a four-stroke engine with two revolutions in each cycle and this component of the bending-stress variation was therefore of the 1st order with reference to the crankshaft position. It was consequently likely that a large part of the variation was due to misalignment. Similar effects could also be seen in the bending stresses presented in Figs. 2 and 3.

When Doxford had difficulties with broken crankshafts, strain-gauge measurements were made in the side web fillets of two 75LB6 engines by Lloyd's Register of Shipping. Stress variations of $\pm$ 420 kg/cm^2 and $\pm$ 670 kg/cm^2 in bending were measured. Large parts of these were 1st order and probably mainly due to misalignment. Since then, Dr. Ørbeck's company had been very misalignment conscious and in 1964 a new instruction for permissible web deflexions was issued.

Because of the long distance between the main bearings on the company's engine, it was essential to make an allowance for the effect of deadweight on the web deflexions. This effect was calculated by considering the shaft as a continuous beam of constant cross-section. The results of these calculations for one particular case were compared with measurements to get some indication of the accuracy and this comparison was shown in Table I.

Table I

Crank No.	1	2	3	4	5	6
Measured deflexion $\times 10^{-3}$ in	16·8	6·9	9·5	9·1	6·1	11·5
Calculated deflexion $\times 10^{-3}$ in	15·9	6	12·4	12	6·1	11·5

The comparison between calculated and measured web deflexions indicated that the straight shaft assumption gave adequate accuracy in bending for this purpose. The author's calculation based on Timoshenko's method should be more accurate and the figures in Table I therefore also gave some support to the author's calculations.

Stress concentrations factors of 3·1 and 4·2 were measured in the side web fillets for webs with external fillets and webs with deeply recessed fillets, respectively, and the limits for the web deflexions were then set to give fillet stresses of less than $\pm$ 340 kg/cm^2. Several cases with stresses exceeding this limit had been reported for old engines and re-alignment was then required. It was noticed on page 507 that the author calculated the bending moment distribution due to misalignment according to the method adopted by Dr. Ørbeck's company. Although there were considerable differences between the side webs of opposed-piston engines and the webs of single-piston engines, it appeared to them that the author had made an under-estimation of the dangers of misalignment. In this connexion, it was interesting that from Fig. 1 (d) in the paper the fillet stress due to misalignment appeared to reach about $\pm$ 250 kg/cm^2. Were any alignments or web deflexion readings recorded?

Although the importance of misalignment was perhaps greater than suggested by the author, it was agreed that the torsional stress generally was the most important single factor in the complex stress system, as indicated in Fig. 18. In this connexion, it was of interest that Lehr and Ruef and Wupperman, Pfender and Amedick had established fatigue limits of 420 kg/cm² and 500 kg/cm² respectively for pure torsion.

The author would probably agree that since he had found fractured crankshafts with torsional stresses down to ± 240 kg/cm² the sum of the remaining contributive factors such as axial vibration, misalignment and material defects, was about as important as the torsional stresses.

The problem of finding the effect of a number of harmonic orders, all with some dynamic amplification, was of interest in connexion with a number of Doxford installations and in 1963 a calculation to deal with this problem had been proposed. Computer calculations were carried out for seven installations in conjunction with measurements, and the highest total alternating stress in torsion was found to be ± 244 kg/cm² for a six-cylinder engine after cylinder No. 4. The stress caused some concern, but the engine seldom ran at the corresponding speed and no failure had occurred.

This calculation, which involved synthesis of all the significant harmonic orders, had been enlarged upon by Dr. Archer (Reference (2) of the paper) and had now been further improved by the author by incorporating the effect of variable inertias according to Draminsky. In this connexion, some clarification in regard to Fig. 22 of the paper would be appreciated. What was meant by "as above with impulse correction for variable masses"?

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MR. A. A. J. COUCHMAN, B.Sc. (Associate Member) said that the paper was one in which the apparent problems of torsio-axial vibration coupling again appeared to cloud the issues involved.

Referring to Figs. 7 and 8, the 6th order torsional and, particularly, the 6th order axial, which the author pointed out existed throughout the range of shaft revolutions, caused him some concern, in that this type of phenomenon had been observed before without satisfactory explanation.

Torsional vibrations could induce axial vibrations, either at the same frequency, the transmission taking place via the propeller, or at twice the torsional frequency due to wind-up. Past experience suggested that, for transfer axial vibrations to be noticeable, the system must at least be near or on a torsional resonance. If this was going to occur with the non-resonating 6th order condition throughout the range of revolutions, then one would expect it also to happen with respect to the other torsional criticals, particularly the second mode, 9th order, but this did not seem to occur.

To his mind, torsio-axial coupling did not appear to account for the prolonged presence of 6th order axial vibration, together with the increasing axial stress levels at the higher revolutions. He began to wonder whether the crankshaft was vibrating as two separate lengths, each associated with six cylinders, except when it was called upon to respond to the overall system resonant frequencies. One had to bear in mind that there was probably the restriction of the gearwheel at the centre of the crankshaft and that the crankshaft was both long and flexible, its stiffness being about 655 tons, which was approximately a sixth of that of the thrust block.

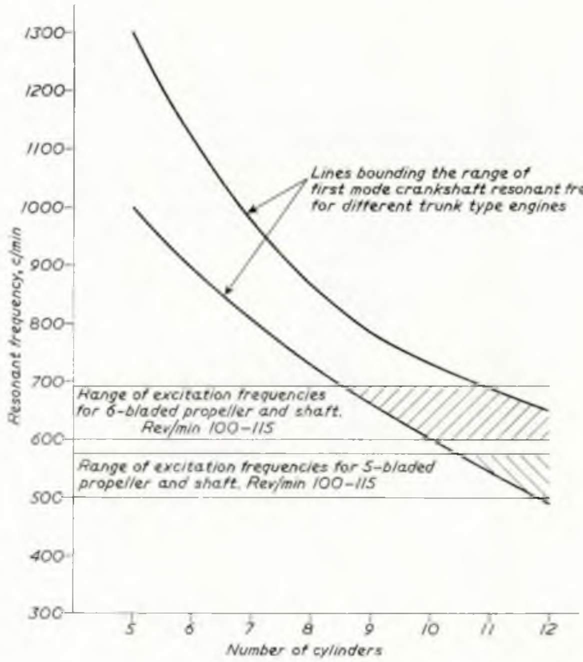


FIG. 26—*Relationship between first mode crankshaft resonant frequency and excitation from five and six-bladed propellers*

If one accepted that the crankshaft could vibrate as two separate lengths, probably with the ends moving in anti-phase, then the large 6th order torsional and axial components measured began to fit into place. Extending this line of thought a little further, he wondered whether one might expect an axial resonant frequency associated with half the length of the crankshaft, which would be placed in the region of 125 rev/min excited by 6th order forces and this could account for the strong axial vibration measured at 115 rev/min. The system second modes of torsional and axial, if excited by 6th order forces, would be at approximately 140 and 165 rev/min respectively, well above the maximum operating shaft speed.

The question of propeller excitation of the first mode of axial vibration in long crankshafts could, in some circumstances, be a serious matter, particularly if the propeller engine phasing was unsuitable, or if the propeller thrust variation should be high.

This concept could easily be shown by means of a graph (see Fig. 26). Plotting the first mode resonant frequency against the number of cylinders, two curves could be constructed which would bound the range of frequencies encountered with different types of trunk engines. Plotting on to this the excitation frequencies for five and six-bladed propellers at shaft speeds from 100 to 115 rev/min, it would be immediately obvious that some 12-cylinder crankshafts would be excited into resonance when coupled with a five-bladed propeller, while 9, 10 and 12-cylinder crankshafts could suffer when six-bladed propellers were fitted.

He was most interested to read that the author had also experienced difficulty in rationalizing the measured first and second modes axial frequencies, using the same system constants in each case. However, Mr. Couchman would not go to the extent of saying that thrust-block non-linear characteristics were involved, but rather that a difference in thrust stiffness appeared to be the easiest way of making measurements fit the calculations. In his own case, a ten per cent reduction in the thrust-block stiffness was necessary to equate calculated and measured second mode frequencies. It would be of value if the author would provide figures relating to his own case.

Regarding the oil-film stiffness and its effect on the thrust-block stiffness, calculations suggested that provided the shaft remained in contact with the pads, the ratio of stiffness for oil film and thrust block in the author's case would be of the order of 30 to 1, so that there would need to be a considerable change in the oil film characteristics to effect even a ten per cent charge in thrust stiffness. If non-linear characteristics were involved, it would seem more realistic to think that this was associated with the thrust-block structure.

The author had made reference to major axial criticals within the range of revolutions in high-speed Diesel alternator sets. Although it seemed an unlikely possibility, he wondered if the author would consider that axial vibration could excite the electro-mechanical coupling frequency between alternating-current generators running in parallel. As the author would be aware, this phenomenon resulted in considerable power surging from one machine to another and *vice versa*, and speeding and slowing of the sets. If the author had had experience of this condition, Mr. Couchman would like him to give details regarding frequencies and causes. He would also ask to what extent the author would expect the stresses in the crankshaft to be modified by a power transfer of ± 100 per cent.

Finally, he wished to stray a little off the subject of crankshafts and raise the question of crosshead bearings. In the highly-rated engines of today there was the tendency for the pins to bend, with the result that a limited or reduced area of the white metal bearing was subjected to loads considerably above the normal instantaneous firing load. The result was that the white metal was flattened and extruded out of the bearing; a very unsatisfactory circumstance. He wondered whether the author would care to comment on the acceptable limit of bending or bending stress which should be applied to the crosshead fillets to obviate this problem. Further it would be interesting if the author had information relating to stresses and temperatures in the white metal.

MR. S. OLSSON said that it was very interesting to see the records showing the coupling effect between torsional stress and bending stress in the crankpin fillet, especially the bigger effect in the plane 45° to the crank plane. The company he represented had measured the bending stress in the corresponding fillet in the crank plane on the aft crank on a seven-cylinder engine of 760 mm cylinder bore and 1500 mm stroke. The measurements were made on the test bed and a rather dominant critical torsional vibration of the 7th order occurred near the speed at which the measurements were made. Measurements were also made at the 10th order torsional vibration critical speed (see Fig. 27).

It was found in both cases that the bending stress in the bottom of fillet MP 5, i.e. on the vertical part of it, was somewhat larger than half the torsional stress in the starting point of fillet MP 4. In the latter point, which also was placed 17° out of the crank plane, the bending stress was about one quarter of the torsional stress. The 10th order torsional stress was about ±130 kg/cm². The bending stress due to torsion was consequently larger in the bottom of the fillet than at its starting point. The coupling effect was not so apparent as according to the records described in the paper.

Strain gauge rosettes were also applied at the small fillet of the journal between cylinders 6 and 7, MP 1 and 2. The bending stresses were rather high, but showed no influence from the torsional stresses.

The torsional stress in MP 1 at the beginning of the fillet was 1·7 times that in the crank-pin fillet. At MP 2—that was to say, at a point in the journal fillet close to the shrink-fit—the torsional stress was about 2·6 times that in the crankpin fillet. Mr. Langballe had put forward the opinion that the compression stress due to the shrink-fit improved the fatigue strength. Pre-stressing the surface for instance, by cold-rolling, was known to have a good influence on the fatigue strength. However, it was stated in the literature that a shrink-fit reduced the fatigue strength to one-half or one-third if the shaft had the same diameter outside and inside the fit. Experiments had shown that cold-rolling of the shaft before mounting improved the fatigue strength again. Therefore, he was somewhat sceptical regarding the improving effect of the shrink-fit, especially on torsional fatigue strength.

Mr. Langballe had mentioned on page 502, regarding axial vibration, that the importance of axial vibration stresses was somewhat disputable, due to the effect of the shrink-fit compressive stresses around the crank plane. Mr. Olsson was not convinced about this. However, this would only occur when the two neighbouring cranks had an intermediate angle of 0 and 180°. At angles between these extreme positions the bending stress probably would have its maximum in a plane other than the crank plane. General axial vibrations caused no problems, but he believed that they ought not to be neglected.

Recent tests in Japan, on shrink-fitted members, showed that a small increase in the diameter of the shaft inside the

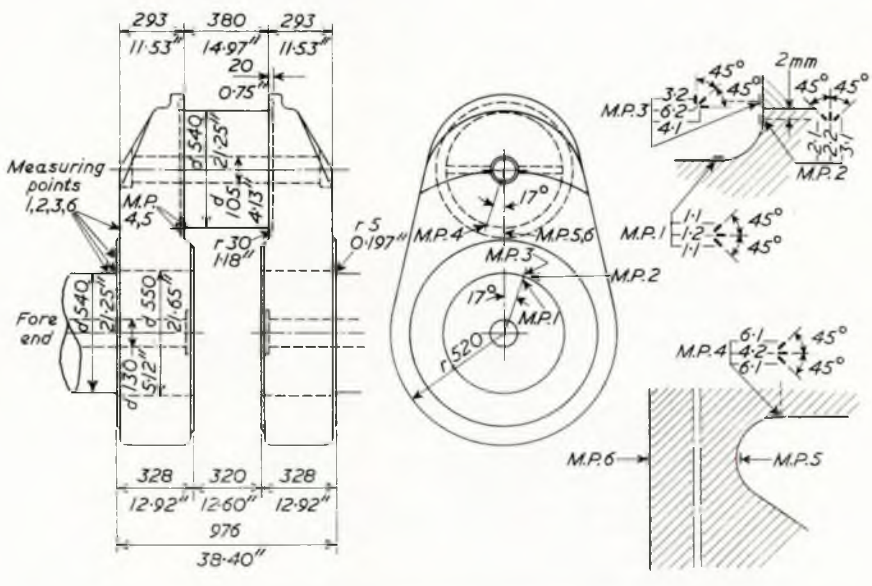


FIG. 27—*Engine DM 760/1500 VGS 7U, crank 7. Crank dimensions and arrangement of gauges*

shrink-fit also improved the fatigue strength in bending. It could also be shown mathematically that the transmission ability of both torque and bending moment through a shrink-fit was more dependent on the diameter and less on the length of the bore than shown in the formula on page 506, when due account was taken of the elasticity of the members and no slipping was allowed. Therefore he thought it would be practical to make the web bore large and the web thickness small and thus get shorter engines.

MR. G. P. SMEDLEY said that Mr. Langballe had drawn attention to the need for rational design of crankshafts. However, despite substantial progress, further experimental data were required to evaluate a number of parameters which were essential to accurate calculation of crankshaft stresses and the factor of safety. The most important were the stress distributions at the fillet radii between web and pin or journal, the influence of geometry on these stress distributions, the correction factors for web stiffness, the locations and magnitudes of the reactions at the main bearings, also the variations with misalignment of the bearings and the criterion of failure of heavy steel sections under combined mean and cyclic stresses with stress concentration.

At Lloyd's Register of Shipping, evaluation of web stiffness factors was attempted with plastic models of crankshafts. It was found that creep of the plastics under stress had a significant influence on the results. It was considered essential that tests should be made on either metal models or actual steel crankshafts. As it was easier and cheaper to use actual crankshafts, tests were now in hand on shafts of different forms supplied by some engine builders. In general, it had been found that stress concentration factors at the fillet radii were in reasonable agreement with data published by Stahl*. With reference to web stiffness factors, these were studied for bending of a crankshaft in the two principal planes, and for twisting of the crank. The q factor proposed by Yamada† for a bending moment, which opened or closed the webs, was almost constant at 0·7 for geometrics of normal crankshafts. The factor for transverse bending was likely to approach unity, while the factor for twisting was of the order of 0·4, although this factor was of little significance.

The main fatigue tests which had been made on crankshafts involved fluctuating torsion or bending. Doubt remained regarding the influence of mean stress on the endurance limit of a crankshaft. Evidence from small-scale tests on notch steel test pieces indicated that mean stress had a small influence on the fatigue strength. Design practice in many fields was to neglect the mean stress, when of low order, or to apply stress concentration factors to the cyclic components only. The author appeared to favour the elastic condition in which stress concentration factors were applied to both mean and cyclic nominal stress components. While this approach was theoretically correct and safe, it prejudiced the permissible load-carrying capacity of a crankshaft. He agreed fully with the use of the Goodman relationship, but considered that fatigue tests, on sizable notched-steel test pieces subjected to both mean and cyclic stresses, were necessary to resolve the problem. He would be pleased to have the author's views on this.

The electron microscope was an additional aid in the investigation of fractures which occurred during service. Unfortunately, in many cases the origins and early stages of crack propagation were damaged by fretting or hammering, or by impact at rupture of the section. Where the fracture surfaces were in good condition, it was possible, by other means, to distinguish zones of fracture initiation, rapid crack propagation and slow fatigue-crack propagation. Such zones were obvious in Fig. 20(a). The main hope for this new aid in microfractography was to obtain better guidance on the levels of stress associated with different stages of fracture. At present, interpretation in this way was very difficult. The author might have over-simplified the position by reference to the basic relationship proposed by Frost. Mean stress could have a marked influence on rate of propagation of a fatigue crack and on the critical level of cyclic stress. Environment also had a significant influence. Moreover, the mean stress could arise from residual stress, shrinkage and engine loadings. The mean stress could be highly localized in some cases. In his opinion, there was need for a great deal more fundamental work before reasonably approximate quantitative estimates could be made from studies of service fractures by the electron microscope.

With reference to built and semi-built crankshafts, a further criterion of strength was associated with the shrink-fits. Under cyclic stress, the mating surfaces of the pin or journal and web were prone to fret. So far, very few failures had arisen from fretting fatigue. This was most probably due to the heavy section sizes of these parts, which gave rise to low nominal cyclic stresses. However, the limitations on the levels of cyclic stress at shrink and interference fits precluded any significant reductions in the scantlings of large crankshafts, irrespective of changes in geometry to reduce say fillet stress concentration. He agreed with a previous speaker that cold-rolling of the seat of the journal was advantageous. This should raise the fretting fatigue strength appreciably as the heavy cold work provided stressing which slowed down the rate of propagation of fretting cracks, which inevitably formed as a surface fretted.

*Stahl, G. 1958. "Dynamische Spannungsmessungen auf Kurbelwellen." M.T.Z., Vol. 19, p. 267.

†Yamada, S. "Investigation on the Strength of Crankshafts (4th Report)." Jap.Soc.N.A., 107 91960, p. 351.

Correspondence

MR. G. A. BOURCEAU (Member) wrote that first of all, he was glad to note that this paper dealt with one of the most important problems concerning Diesel engines intended for the propulsion of large ships. Bureau Veritas had been making important investigations into the problem for many years. Also he was interested to note that Mr. Langballe applied, or hoped to apply, to the studies of this problem, a method very similar to that adopted by Bureau Veritas: the concluding remarks of the paper were that experimental data were desirable and led to better results than theorizing the problem into obscurity. In fact, he thought that it was better to carry out experimental researches closely approaching the true working conditions.

From his society's experience, he knew the difficulties which must be solved during the carrying out of the measurements as well as during the analysis of the phenomena recorded.

He said that he had just presented a paper‡ at the last Session of A.T.M.A. (Association Technique Maritime et Aéronautique), in co-operation with Mr. Wojcik, who was in charge of such investigations in the research departments of Bureau Veritas. The subject of their paper was large marine Diesel engine crankshafts.

He said that he would be very interested in discussing this subject with the author.

He then drew attention to a few points in Mr. Langballe's paper.

On page 498, Mr. Langballe stated that, for determining

‡Bourceau, G. A., and Wojcik, Z. 1966. "Au Sujet des Arbres Manivelles de Moteurs Marin". Paper read to l'Association Technique Maritime et Aéronautique.

the bending stresses, a straight beam analogy was adequate for an approximate analysis. Mr. Bourceau agreed with the principle, but thought that the main difficulty might be to specify what value would have to be taken for the moment of inertia of this straight beam. In various cases, other authors had considered that the crank-throw moment of inertia was equal to that of a circular section of the journal. From the latest investigations by Bureau Veritas, it was not possible to draw absolute conclusions concerning this point of view. However, in special working conditions for the crankshaft, interesting results were obtained; those results came from the measurements carried out on large Diesel engine crankshafts under working conditions; it was found that when the engine was operated, it was possible that one or several bearings lost contact with the crankshaft journal. Fig.10 of the A.T.M.A. paper showed the vertical vibrations of a journal in such a case. The investigations showed that, in this case, the moment of inertia of this part of the crankshaft was only 10-40 per cent of the moment of inertia of the journal section of the crank throw.

These researches concerned three types of crankshafts, as shown in Fig. 11 of the A.T.M.A. paper; these results concerned three Diesel engines of 7500, 8000 and 15 000 hp; the first two were of the semi-built and fully-built types respectively and the last one (15 000 hp) was of the semi-built type.

Considering the effect of the firing of two cylinders on three bearings, that in the middle being not in contact with the corresponding journal, it was found that the resulting stresses were about three to six times higher than those calculated for a beam having the same journal section; these results were detailed in Fig. 12 of the A.T.M.A. paper.

Mr. Bourceau thought it important to specify that such cases might occur very often during the running of Diesel engines of large ships; this was especially the case in heavy sea conditions.

He said that he would very much appreciate any comments from Mr. Langballe on the opinions expressed in this contribution about stresses induced in crankshafts, with regard to marine Diesel engines under working conditions.

DR. ING. F. SCHMIDT wrote that on page 501 the author described the method of supplying a signal proportional to the torque in the crankpin by using the gauge "A" on the top of the crankpin. In his opinion this gauge should be placed in the middle plane of the crank, but Fig. 5 did not clarify this question.

Concerning axial vibrations, the method of determining the stiffness of the crank throw and of the thrust bearing was very interesting. Further investigations in this direction would eliminate some uncertainties when calculating the natural frequencies, if a similar method to that shown by Dr. Ing. Alban Rupp, in his dissertation at the Technical High School, Karlsruhe, in 1964, were applied.

Some practical experience on this question was given in a paper* read before this Institute in January 1964.

Dr. Schmidt drew the author's attention to a very interesting paper† read at the Sixth Shipbuilding Conference, Bombay, in February 1966. These very comprehensive investigations shed light on the influence which the main dimensions had on the strength of a crank.

MR. A. KLEINER commented, in a written contribution, that this paper would certainly be appreciated by all Diesel experts, in particular by all Diesel designers. It represented, for quite a number of designers, confirmation of results which had been obtained through their own investigation, and was appreciated as such.

However, not everyone had the most modern and precise measuring apparatus at his disposal, as had the author, for investigating all the different factors which influenced the stresses in the crankshaft.

*Sörensen, E., and Schmidt, F. 1964. "Recent Development of the M.A.N. Marine Engine". *Trans.I.Mar.E.*, Vol. 76, p. 197.

†Hoshino, J., and Arai, J. December 1966. "Strength Analysis of Diesel Engine Crankshafts". *Trans.I.Mar.E.*, Indian Supplement, No. 19, p. 39.

One of the most important results of this investigation was the fact that no unexpected results, which would involve modifications to the present design of the crankshafts, were found. That design was based on development and experience over a long period, and practically no major improvement was possible. This conclusion had been reached already, as the result of Dr. Archer's paper.§ It was, however, very valuable to have confirmation of this finding, based on a quite different point of view.

The proposal to introduce "the variable torsional stress as a suitable basis of comparison" was very valuable for judging the general stresses in the crankshaft. This was a value which nowadays should be calculated very carefully for each Diesel engine. The stress level of approximately $\pm$ 400 kg/cm^2, as a realistic fatigue limit for large mild steel crankshafts, compared very nicely with experience over many years in the design of large stationary engines, driving generators. These engines had lower torsional natural frequencies than ships' main propulsion engines because of the large moment of inertia of the generator rotor. This was the reason why the stress level of the crankshafts of those engines was much higher than the values which most of the classification societies (including Det norske Veritas) would accept for marine engines, the more so as the constant service speed of land engines was, in general, higher than the full-load speed of the same engine when installed in a ship. Since the designer of large land engines—from the point of view of dimensioning the crankshaft—was, in general, independent of the very conservative rules of the classification societies, he had the opportunity of making a crankshaft of an efficient design, benefiting by the quality of the material, and still maintaining a high enough safety margin.

As designers of Diesel engines, the company which Mr. Kleiner represented hoped that the very conservative rules of the classification societies, concerning the additional torsional stresses in the full-load speed range of main propulsion machinery, would, in future, be relaxed, in view of the fact that the stress level, the safety of which was proved by long experience, had been confirmed by the very serious investigations of a well known classification society.

MR. H. H. SCHOLTEN, in a written contribution, commented that with respect to formula (1) on page 499 of the paper it was concluded that the rather low value of the constant k (0·55) was due to the relieving effect of the inertia forces per cylinder line. Theoretically this value should be 2 in the case of a freely-supported beam and 1 assuming a secured beam, the actual value being somewhere in between. Perhaps the author could give an idea of the value of this constant, taking into consideration the resultant pressure on the piston, thus decreasing the maximum combustion pressure with the inertia pressure at top dead centre.

With respect to secondary resonance (Appendix I of the paper) it might be of interest to mention a quite simple method which was developed in 1958 in close co-operation with Professor Dr. Ir. J. J. Koch from Delft Technical University:

Taking:
$$T = \Sigma\, I \cdot \alpha^2,$$
$$A = \Sigma\, I_r \cdot \alpha^2 \cdot \cos 2\,\theta,$$
$$B = \Sigma\, I_r \cdot \alpha^2 \cdot \sin 2\,\theta;$$

where: I = mass moment of inertia of each mass,
α = relative amplitude in Holzer—table,
I_r = mean mass moment of inertia of the reciprocating parts of each cylinder,
θ = crank angle relative to no. 1 cylinder;

the additional energy-input, by a harmonic of order $p \pm 2$ to a harmonic of order p, was equal to:

$$E_p = \frac{p}{8} \cdot \frac{\sqrt{A^2 + B^2}}{T} \cdot E_{p \pm 2}.$$

§Archer, S. 1964. "Some Factors Influencing the Life of Marine Crankshafts". *Trans.I.Mar.E.*, Vol. 76, p. 73.

As the foregoing formula was reached by means of a number of simplifications—though proving in itself to be sufficient to explain the discrepancies met with—an accurate investigation into this matter was carried out later, in which the influences of the order $p \pm 3$ were incorporated:

$$
\begin{aligned}
\text{Taking:}\quad T &= \Sigma\, I \cdot \alpha^2 \text{ as before,}\\
A_1 &= \Sigma\, I_r \cdot \alpha^2 \cdot \cos\theta,\\
A_2 &= \Sigma\, I_r \cdot \alpha^2 \cdot \cos 2\theta,\\
A_3 &= \Sigma\, I_r \cdot \alpha^2 \cdot \cos 3\theta,\\
B_1 &= \Sigma\, I_r \cdot \alpha^2 \cdot \sin\theta,\\
B_2 &= \Sigma\, I_r \cdot \alpha^2 \cdot \sin 2\theta,\\
B_3 &= \Sigma\, I_r \cdot \alpha^2 \cdot \sin 3\theta,\\
\lambda &= \text{crank/connecting rod ratio.}
\end{aligned}
$$

With these values the additional energy-input of the various adjacent orders would be:

$$E_p = \frac{p^2 \pm p + 0{\cdot}5}{2 \cdot (2p \pm 1)} \cdot \lambda \cdot \frac{\sqrt{A^2_1 + B^2_1}}{T} \cdot E_{p \pm 1}$$

(order-difference $\pm$ 1).

$$E_p = \frac{p^2 \pm 2p + 2}{4 \cdot (2p \pm 2)} \cdot \frac{\sqrt{A^2_2 + B^2_2}}{T} \cdot E_{p \pm 2}$$

(order-difference $\pm$ 3).

$$E_p = \frac{p^2 \pm 3p + 4{\cdot}5}{6 \cdot (2p \pm 3)} \cdot \lambda \cdot \frac{\sqrt{A^2_3 + B^2_3}}{T} \cdot E_{p \pm 3}$$

(order-difference $\pm$ 3).

By means of the foregoing formulae it could easily be found that, particularly, the secondary resonance with an order difference of $\pm$ 2 might be substantial due to the absence of the factor λ.

MR. R. MACIOTTA wrote that this research was to be considered as pioneer work and one must welcome any report on data and experience in this field, even if it was not always possible to present the details of the experience in a completely theoretical arrangement.

This research carried out by the classification societies' technicians was worth particular interest as these societies laid down rules for the proportioning of the crankshafts, which formed a valuable reference to the engine builders.

One of the most interesting results of the experimental research reported by Mr. Langballe concerned the occurrence of bending stresses (σ), besides the torsional stresses (τ), in the crankpin fillet as a consequence of a torque applied to the crankshaft itself. However, the mechanism through which these stresses took place was not completely clear; resulting from the recordings mentioned in the paper, the coupling between σ and τ could be referred to two different mechanisms:

a) an external coupling, consequent upon the fact that the crankshaft torsion entailed some transverse reactions in the journal bearings, which in their turn gave rise to bending moments in the crank;

b) an internal coupling, by virtue of the fact that the crankpin fillet was subject to stresses ranging gradually from pure torsional stresses, occurring in the crankpin, to pure bending stresses, taking place in the webs.

Mr. Maciotta believed this scheme was in accordance with the author's thoughts; anyway, it was only introduced with the purpose of facilitating the following discussion.

Concerning the combination between torsional and bending stresses, he wished to ask the following questions:

1) From Fig. 11, the dynamic bending stresses in the fillet (σ_x) had a value about 2·4 and 2·1 times the nominal torsional stress (τ_o), in the crank plane (position D) and 45° off the crank plane (position E) respectively. These results apparently did not agree with the values of the coefficient γ to be introduced in formula (5) to obtain the compliance of the calculated stresses with the recorded stresses. Could the author give any explanation of the matter?

2) The dynamic stress recordings made on an engine in operation and reported in Fig. 10 showed a high degree of coupling between torsional stresses (τ_{xy}) and bending stresses (σ_x) in the fillet both at position D and E (the results approximately complied with the coefficients of Fig. 11). The static stress recordings made on a model, however, proved that the internal coupling was high at position E (45°), but practically negligible at position D on the crank axis. The foregoing meant that the bending stress recorded at D was to be considered imputable to what had been referred to as an external coupling: but the external coupling implied the presence of an axial vibration having the same frequency and phase of the fluctuation of the torsional moment. Had the author had the opportunity of checking the said axial movement?

3) In Figs. 1 and 2 the author reported on two examples of bending stresses recorded in the crank No. 4 of a four-stroke engine crankshaft; in the first example, Fig. 1(d), in the crank plane, out of resonance, in the second example, Fig. 2, in the 45° plane and in resonance with a torsional critical. In the first example, no coupling was noticed between the bending stress fluctuation and the torsional stress pattern, Fig. 1(c); on the contrary, in the second example the torsional and bending stresses had a very similar pattern. Could the author make any comment on the different behaviour?

4) In Fig. 15, for the crank at the top left, the author gave the diagram of the ratio between the stresses in the fillet and the nominal torsional and bending stresses. Mr. Maciotta believed that the expression $\frac{\sigma_x}{\sigma_o} = 1{\cdot}12$ contained an error and should read: $\frac{\sigma_x}{\tau_o} = 1{\cdot}12$. The above-mentioned curve was plotted for half a crank only; did this mean that the two halves were to be considered symmetrical? This would not comply with the results of experience by Mr. Maciotta's firm: in the recordings with skin stress they had ascertained that breaking of the skin occurred in one of the web sides first.

Mr. Maciotta also remarked on formula (9) which established a relation between the web deflexion and the bending stress in the crankpin, and was a basis for the evaluation of the allowable crankshaft misalignment limits.

His company had also worked out a similar formula taking, as a basis, the relation between the nominal deflexion stress in the crankpin and the variation of the distance between the facing sides of the webs (under the action of two equally-opposed moments applied to the journals and acting in the crank plane) which was obtained by assimilating both the crankpin and the webs to De Saint Venant's solids: upon comparison with this base formula Mr. Maciotta felt that formula (9) should be written as follows:

$$f = \sigma R \cdot 10^{-6}\left[1{\cdot}91\,\frac{l}{2d}\left(1 + \frac{a}{R}\right) + \frac{R + 2a}{d}\,0{\cdot}56\,\frac{d^4}{bt^3}\right].$$

This formula represented, however, a mere theoretical relation which did not give accurate results, as both the crankpin and the webs were solids, considerably different from the De Saint Venant solids.

A better approximation could, however, be obtained by introducing as the deflexion length of the crank, an equivalent reduced length expressed by:

$R - Kd$ in which:
R is the crank radius
d is the crankpin diameter
K is an experimental coefficient.

Therefore the formula could be written as follows:

$$f = \sigma \cdot 10^{-6}\left[1{\cdot}91\,\frac{l}{d}\left(\frac{R + a}{2}\right) + \frac{(R - kd)(R - kd + 2a)}{d} \right.$$
$$\left. 0{\cdot}56\,\frac{d^4}{bt^3}\right].$$

Upon comparison of the calculated values with the measurements made on various crankshafts, the average value of the coefficient K was 0·55. The crankpin length (l) was however, considered equal to the useful length plus the thickness of one web.

A further question concerned the stress concentration in the fillet of the journal for which, in Fig. 15, the value $\frac{\sigma_x}{\sigma_0}$ = 6·2 was indicated. It was evident that this value depended on the radius of the fillet between the journal and the shrink-fit portion, the diameter of which was normally some millimetres larger than the journal radius.

No limits were specified by the classification societies concerning this value; however, it was the practice of Mr. Maciotta's firm to give this radius a value equal to that of the fillet between web and crankpin. Could the author specify which, based on his experience, was the minimum value of this radius referred to the journal diameter, to obtain a safety coefficient not lower than that obtained with the dimensioning specified by classification societies for the web crankpin fillet?

Finally, Mr. Maciotta remarked upon the author's statement on page 499, concerning the possibility of removing the axial vibration critical from the service speed range, by adjustments of the weight of the flywheel and counterweights (in Mr. Maciotta's opinion, this could not always be easily accomplished). He believed that the author had made reference here to generating sets' engines operating at constant speed.

In this connexion, it was perhaps worth mentioning the case of a four-stroke vee-engine with 16 cylinders, 300-mm bore, 450-mm stroke, operating at 500 rev/min, in which his firm had succeeded in eliminating noticeable axial vibrations by merely increasing the clearance of the crankshaft thrust collar.

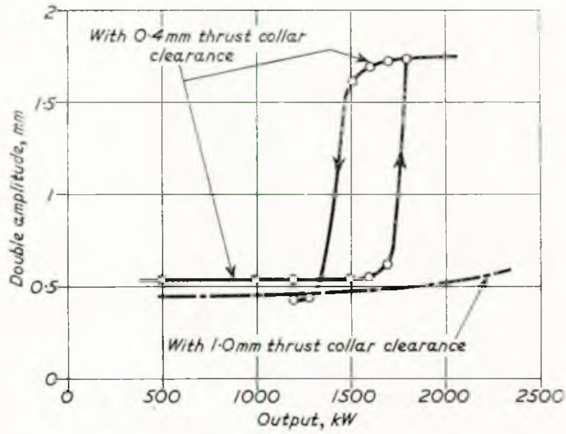


FIG. 28—*Axial vibrations of crankshaft in a four-stroke engine for power generation as a function of the output and the thrust collar clearance*

This was a singular case also from another point of view, in fact, as shown in Fig. 28 the vibrations depended essentially on the output; for outputs lower than 1700kW the vibrations were within practically negligible limits, but as soon as this output was attained, the vibrations appeared suddenly with such amplitudes as to give rise to sensible stressing of the engine bedplate. Upon reduction of the load, the vibrations disappeared as quickly as they had arisen, around 1500kW, as shown in the diagram.

Investigations made by Mr. Maciotta's firm, proved that in this plant, despite the absence of axial thrust, the bedplate had a share, as an elastic component, in the crankshaft vibrations transmitted through the thrust collar.

As mentioned earlier, modification of the thrust collar, consisting of increasing the axial clearance from 0·4 to 1 mm, proved sufficient to break up the elastic coupling between crankshaft and bedplate, thus avoiding priming of the vibrations.

Author's Reply

The author wished to express his thanks to those taking part in the discussion for their many important and valuable contributions. The subject was admittedly a rather specialized one, and he was sure that the discussion might elucidate some obscure points and thus enhance the value of the paper.

Dr. Archer's comments were highly appreciated. Dealing first with his comments regarding the measurements, the author said that to justify a simultaneous comparison of stresses at the different points of measurements at the same speed of revolution, the stresses being obtained in different runs of measurements, the following procedure was followed to ensure equal conditions:

One channel on the recorder was reserved for a reference strain gauge which recorded torsional stress during all the series of measurements, and the speed of the engine was adjusted carefully, with the aid of a very accurate electronic tachometer. The records exactly at the specified speed and eventually having congruent shapes of the reference trace were chosen for analysis.

The strain gauge "A" was located in the crank plane, 30° into the fillet, as shown in the detailed sketch in Fig. 5.

Regarding variable masses in torsional vibrations, the author had subsequently submitted the recordings of torsional vibrations in the case of secondary resonance referred to in Fig. 22 to a harmonic analysis.

The result yielded a much better agreement between the measured 7th order resonance stress and that calculated by the method outlined in Appendix I. This was now shown in Fig. 22. Also shown there were the stresses obtained by a conventional forced vibration calculation and the 7th order resonance stress obtained by introducing a correction factor to the constant-mass calculation, derived from the equivalent one-mass system according to Draminsky's method.

Secondary resonance was also found to be present in the eight-cylinder four-stroke engine investigated (Fig. 1 in the paper), at the 6th order critical and excited by the adjacent 4th and 8th order major criticals. The 6th order resonance stress, calculated by the method given in Appendix I, was found to be ± 153 kg/cm^2 as against ± 53 kg/cm^2 given by the constant-mass calculations (stresses between flywheel and the last crank).

The damping factors according to Maciotta were in good agreement with the values for the engines investigated here, which, however might be accidental. As to the polytropic exponents, the following values were used for the four-stroke engine: compression $N_k = 1{\cdot}38$, expansion $N_e = 1{\cdot}30$.

The author stated that Dr. Archer's question on the micro-fractographic photographs, was referred to in the reply to Mr. Smedley.

Mr. Scholten's presentation of the method of calculation of secondary resonance devised by himself and Professor Koch, was a very welcome contribution. It would be interesting to have assessed by this method the contribution to excitation from adjacent $\pm$ 4th order impulses. For such an analysis, the method outlined by the author would become too involved.

In order to explain the apparent discrepancies in the torsion-bending coupling factor, γ, mentioned by Mr. Maciotta, it was convenient to summarize the possible mechanisms of coupling between torsional and bending stress in the fillet:

1) transverse bending of the shaft introduced by torsional loading;
2) axial deformations of the crank throws, induced by torsional loading, eventually amplified by flank and resonance vibrations;
3) a local elastic effect within the fillet itself, or more specifically, the principal stresses in the fillet by torsional load having such an orientation as to give an axial normal stress component which appeared as if it were a bending stress.

The first mechanism was in principle taken care of by a complete bending-stress calculation, and was expressed by the first term of equation (5). The effects of mechanisms 2) and 3) were then contained in the factor γ. The local or internal coupling factor being 1·12 at gauge "E" in the 45°-plane and 0·0 in the crank plane at gauge "D", the differences ($\gamma - 1{\cdot}12$) and ($\gamma - 0{\cdot}0$), was due to axial deformations. The relation $\sigma_x = 2{\cdot}12 . \tau_0$ in equation (3), was due to the combined effect of all the three mechanisms of torsion-bending coupling in the 45°-location at gauge "E". In this connexion it should be mentioned that the 6th order stress variation was associated with a corresponding 6th order forced-axial vibration, measured at the free end of the crankshaft.

Regarding the distribution of the stress concentration σ_x/τ_0, shown in Fig. 15, this was equal on both sides of the crank plane, but of opposite sign.

The different behaviour of the measured bending stress, in response to the simultaneous torsional stresses, shown in Fig. 1(d) and Fig. 2, was due to the local coupling being absent in the plane of symmetry, but having a substantial effect in the 45°-plane.

The observations on torsion-bending coupling reported by Mr. Olsson might seem to indicate that the local type of coupling might be involved in his case, although to a lesser extent due to the geometry of his type of crank and the measuring points mentioned being closer to the crank plane.

The mechanisms of axial vibration proposed by Mr. Couchman were probably impossible and should not be necessary in order to explain the existence of the dominating presence of the 6th order axial vibrations at the higher revolutions. There was also some coupling at the other torsional criticals but to a lesser extent. It should be noted that the natural frequencies of the corresponding modes of torsional and axial vibrations were fairly close, being 840 c/min and 940 c/min respectively, in the shipboard installation (in test bed conditions 835 c/min and 1070 c/min). Evidently, the 6th order torsional-axial coupling was amplified by the more or less coinciding flanks of the associated major critical of these modes at vibrations. The need of predicting such conditions was evident, but an adequate mathematical theory seemed to be lacking.

The author agreed with Mr. Couchman that there might also be other explanations than a non-linear thrust-bearing stiffness, of the lack of agreement between the first and second mode axial vibrations. A further analysis of this had been carried out, introducing larger additional masses at the propeller and the thrust bearing. Adding 100 per cent virtual mass for entrained water to the propeller and assuming the oscillat-

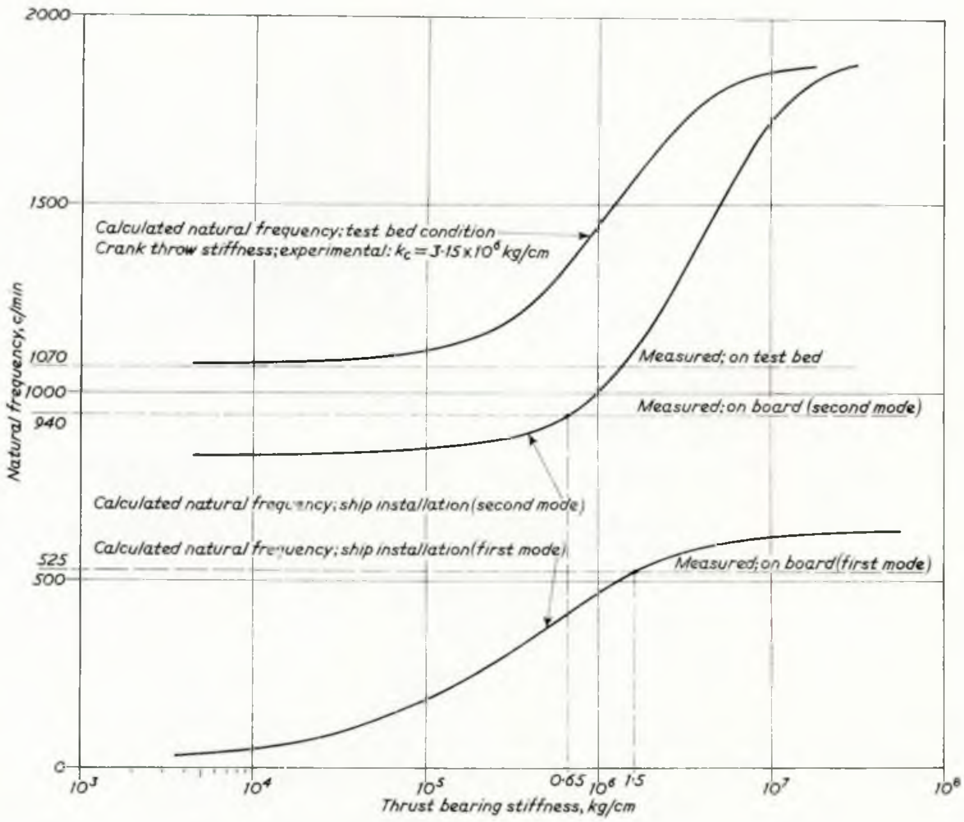


FIG. 29—*Natural frequencies of axial vibrations as functions of thrust-bearing stiffness*

ing weight of the thrust block assembly to be 10 tons (metric), still left a considerable difference in the thrust block stiffness, as shown in Fig. 29, where the natural frequencies were calculated as functions of the thrust-bearing stiffness. These virtual masses were probably too high, but hydrodynamic investigations into the entrained water masses of axially-oscillating propellers might throw more light on this problem.

Mr. Maciotta's observations on axial vibrations in a generating set were very interesting. In his own case, the author had come to the conclusion that the crankshaft behaved as though floating freely, in which case the effect of reduced thrust collar clearance would be the reverse of that obtained by Mr. Maciotta. In any case, it seemed probable that manipulations with the clearance might cure such problems in generating sets.

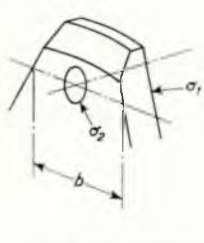
Mr. Smedley emphasized the necessity of gathering more knowledge on the influence of various proportions of the crank throw, on the stress concentrations. Some experiments had been done on this, by varying some of the proportions of the models shown in Fig. 15. The results had been condensed into formulae for correction factors which had been included in Fig. 15. The stress concentrations on crank web edges might also be of some interest, and their dependence upon the diameter of the crankpin bore and the width of the web were summarized in the table in Fig. 30. The best type of crank throw seemed to be the fairly wide one, with a crankpin bore diameter of maximum 0·5 times crankpin diameter.

Mr. Olsson proposed reducing the web thickness of semi-built shafts, and increasing the diameter of the shrink-fit to maintain the strength of the shrinkage grip. The author agreed with this principle, but the effect of the residual tensile stresses in the crankpin fillet, due to shrinkage, to some extent limited this possibility.

Mr. Maciotta also raised the question of strength at the fillet of the shrink-fit. In the large engines investigated by the author, the stresses measured in the journal fillet of crank No. 5 were not particularly critical, the ranges of stress being as shown in Fig. 31. This might of course be very different in other engines.

A more likely risk than cracking in the journal pin fillet was that of fatigue failure due to fretting, a very interesting case of which had been given by Dr. Archer. An alternative development of such frettings might be the slipping of the shrink-fit to some extent, accounting for such incidents where "brute forces" had not been at work. The safest design was no doubt as large a radius as possible at the shrink-fit. As stated by Mr. Smedley, the limitations imposed by the shrink-fit did

Width of crank web, $\frac{b}{d}$	1·12 (Model 'A')		1·33 (Model 'B')	
Diameter of crankpin bore, $\frac{d_0}{d}$	0·36	0·50	0·36	0·50
Peak stress at edge of web, $\frac{\sigma_1}{\tau_0}$	1·97	2·00	1·25	1·28
Peak stress at edge of bore, $\frac{\sigma_2}{\tau_0}$	0·89	1·51	0·80	0·98



d = Diameter of crankpin
τ_0 = Nominal torsional stress in crankpin

FIG. 30—*Stresses at the edges of crank webs in torsional loading of crank throws*

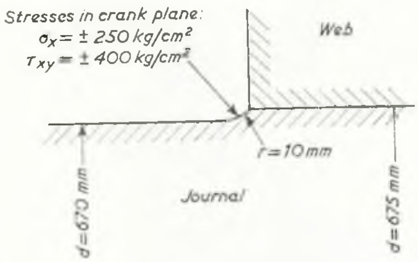


FIG. 31—*Dynamic stressing in fore journal fillet of crank No. 5 (measured in ship installation at full load, 115 rev/min)*

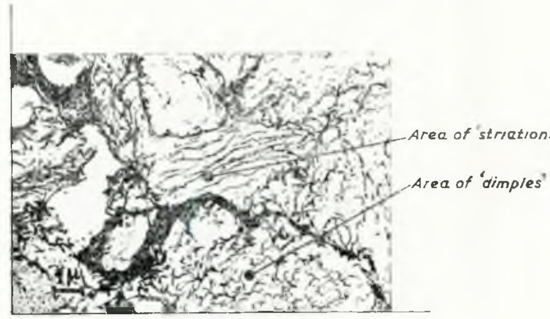


FIG. 32—*Some characteristic features of the microfractograph shown in Fig. 20(b)*

not leave much room for improvements in design of semi-built crank throws, but they should be explored more thoroughly by dynamic tests.

In a further reply to Mr. Maciotta, the author considered the crankweb deflexion formula (9) to be correct. It was, however, noted that the deflexion, f, was taken to be the maximum difference read on the dial gauge during one revolution of the crankshaft, as added in the text. The formula was derived on the basis of bending by a constant moment of the actual pin length and the effective elastic length of the web found to be 0·95.*R* by the model tests. (The factor 0·95 is included in the number 0·56). For the type of crank shown in Fig. 1, the elastic length was found to be 0·85.*R*.

It had been very interesting to see the good agreement obtained by Dr. Ørbeck between measured crank-web deflexions and those calculated using a straight beam analogy.

The crank-web deflexion and misalignment of the generating set had been very slight, and could not possibly account for the first order bending stress recorded in Fig. 1(d). By a subsequent static measurement, the stress variation in the fillet due to misalignment had been found to be $\pm 65\ kg/cm^2$ (the journals had been jacked down to ensure contact with the bearings during these tests). The author thought it would be useful to carry out a number of such simple static strain-gauge measurements in the fillets of large crankshafts, in conjunction with ordinary deflexion measurements, in order to draw more definite conclusions as to the importance of crank-web deflexions to the total safety of the shaft.

At low levels of nominal torsional stress, the author certainly agreed that other circumstances had to initiate failure such as structural defects, axial vibrations, misalignments and, it might be added, excessive residual stresses due to shrinkage.

Regarding Dr. Ørbeck's question on the case of secondary resonance, this had been referred to in the reply to Dr. Archer.

Mr. Bourceau's interesting measurements of the lateral displacement of the journals during operation seemed to indicate that it was too rough an approximation, in the theoretical calculation of bending stresses, to assume that the points of support of the journals were restricted from lateral movement. The assumption that one bearing should not carry any combustion load at all seemed, however, to be an unlikely and too unfavourable a position. It would have been useful to have calculated the additional stresses resulting from the engine bedplate deflexions measured in different loading conditions.

The approximate formula for the bending stress in the fillet, (1), was based on the nominal bending moment in way of mid-thickness of the webs. The factor 0·55 incorporated correction for the degree of fixation as well as the variable stiffness of the equivalent shaft. The factors proposed by Mr. Scholten were apparently based upon the bending stress at the point of application of the load on the crankpin, and not in the fillet.

The author could see no reason why the actual values of the mean stress should be left out of the calculations of safety factors, as proposed by Mr. Smedley. He himself was of the opinion that the absolute level of fillet stress should be applied in conjunction with the fatigue strength at corresponding levels of mean stress, obtained from large cylindrical specimens. The latter seemed to agree well with the Goodman relation. From the author's experience, this seemed to be a good approximation for the fillets of large crankshafts, which had comparatively low stress gradients. The mean stress was particularly important if crack-like defects were encountered, and in such cases the absolute peak stress determined the propagation of the crack (the stresses in Fig. 21 referred to the absolute dynamic peak stresses). Mr. Smedley's view that more experimental data were needed to verify the hypothesis of failure under combined stress and at various levels of mean stress was fully endorsed.

Some additional information on fatigue strength of cast steel had been given in Appendix II.

Microfractographic analysis might uncover many features of the fracture which were not readily revealed by conventional means, particularly if parts of the fracture surface were hammered. A fast propagating or ductile fatigue crack, such as that of zone 2 in Fig. 20 (a), might not be readily distinguishable from other types of cracks, for instance those formed under the influence of hydrogen. By microfractography, the dimples mentioned in connexion with Fig. 20 (b)—indicated more clearly in Fig. 32—generally appeared by ductile cracks, but lines left by the crack fronts, or striations, revealed that the fracture had been propagated by a cyclic stress in zone 2. In this particular case (which was dealt with in detail in the first additional reference*), a conventional fracture analysis might have led to different conclusions as to the origin and development of the crack. However, much experience and a background of systematic research was necessary to develop the method into a useful tool, in conjunction with stress analysis, for putting the numerous circumstances usually involved in fractures in the right order.

Mr. Couchman raised the question of power oscillations in parallel-running alternator sets and the consequences of these to the crankshaft stressing. For a detailed treatise on these oscillations the author would refer him to the second additional reference†. Briefly, the sources and remedies seemed to lie in the following circumstances:

a) too low an electrical damping, or instabilities in the alternator-exciter system;
b) too low a speed drop of the engines;
c) friction and backlash in the Diesel engine speed governor system;
d) too low a natural frequency of the Diesel engine speed governor, and too much damping of the governor.

The oscillations might be of a self-sustained type, excited by random, sub-harmonic torque components from the engine.

*Wintermark, H., and Harsem, Ø. 1966. "Lessons to be Learned from Crankshaft Failures". Det norske Veritas, Publication No. 55 (In Press).

†Brodin, G. 1962. "Power Oscillations in Parallel Connected Synchronous Alternators Driven by Diesel Engine". Det norske Veritas, Publication No. 31.

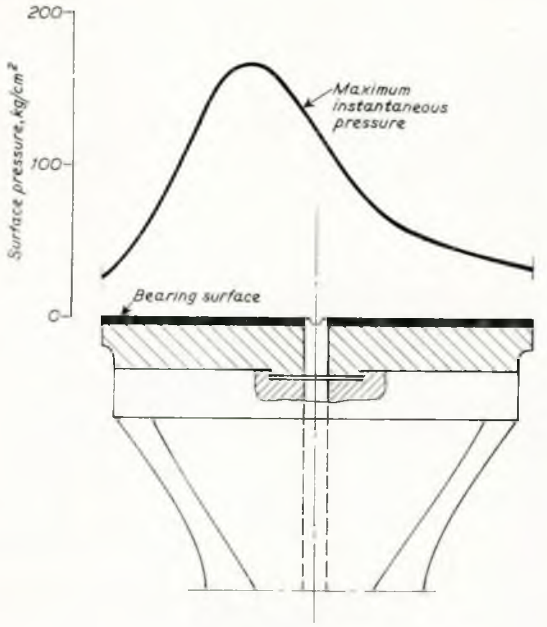


FIG. 33—*Instantaneous distribution of surface pressure in a crosshead bearing due to the combustion load (measured with pressure cells). The distribution shown is due to poor fitting, misalignment and bending of pin*

Because the natural frequency was usually very low compared to those of torsional vibrations (in one specific case 85 c/min against 3900 c/min for the torsional) the oscillations were as likely to act on the crankshaft stressing as any other comparatively slow load variation.

The problem of crosshead bearings was rather involved and controversial, but a few remarks from the author's own experience might be offered in reply to Mr. Couchman's question. On a bearing of the type shown in Fig. 33, the local surface pressures were measured during the running-in of a new bearing. At low speed, a distribution as shown was obtained and these pressures were associated with a slight temporary heating of the bearing. This distribution was partly due to the elastic deformation of the crosshead pin and not too good a fitting and alignment. As was well known such initial heatings of a newly fitted bearing were common and subsequently the bearing could operate reliably. However, if the initial heating of the white metal was excessive, the tin oxides which then formed might lead to increased wear and destruction of the essential super finish of the pin. Difficulties might then arise at a later stage.

The author was grateful for Dr. Schmidt's additional references. His question regarding the position at gauge "A" was clarified in the reply to Dr. Archer.

Regarding Mr. Kleiner's remarks on the classification societies' rules, their scantlings and permissible torsional stresses for large crankshafts at the normal speed might give a total nominal stress up to about ± 300 kg/cm^2. Accepting the fatigue strength of the crank throws in torsion to be ± 400 kg/cm^2, the factor of safety against pure torsional failures would be about 1·33 (with optimum design and absence of any structural defects, the fatigue limit was probably higher).

The application of more modern methods of calculation of the total torsional stressing in conjunction with appropriate permissible levels, should be of advantage to the designers. Rule formulations based on safety factor considerations and taking crank geometry into consideration, are quite feasible and will no doubt soon replace the present rules.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 10th May 1966

An Ordinary Meeting was held by the Institute on Tuesday, 10th May 1966, at 5.30 p.m., when a paper entitled "Investigations into the Stressing of Crankshafts for Large Diesel Engines" by M. Langballe, Sivilingeniør, was presented by the author and discussed.

Mr. W. Young, C.B.E. (Member) was in the Chair and fifty members and guests were present.

In the discussion which followed five speakers took part.

A vote of thanks to the author was proposed by the Chairman and received enthusiastic response.

The meeting ended at 7.55 p.m.

Branch Meetings

North West England

A lecture meeting was held by the Branch on Monday, 7th November 1966 at 6.00 p.m. in the Conference Room of the Mersey Docks and Harbour Board Building, Liverpool, when the paper "The Design of Air Registers for Oil-fired Boilers" by Lieutenant Commander J. P. D. Hakluytt, B.Eng., R.N. and B. C. North, B.Sc., was presented by the authors.

The presentation was illustrated by a film and the discussion which followed was opened by Professor J. Livesey.

Fifty-one members and visitors were present.

Scottish

A general meeting of the Branch was held on Wednesday, 9th November 1966, at the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2, at 6.15 p.m., when a paper entitled "The New Polar Four-stroke Engine" by Lars Th. Collin, M.Sc. was presented by the author.

Mr. W. McLaughlin (Vice-Chairman of the Branch) presided at the meeting and welcomed the 118 members and visitors.

Professor Collin opened his address by giving a brief history of the development of the Polar engine from the signing of a licence agreement in 1898, between Rudolph Diesel and a Swedish financial group.

It was interesting to note that the name Polar, which was included in the name of the United Kingdom licencees, British Polar Engine Ltd., Glasgow, was used for all engines manufactured after the Antarctic explorer Amundsen, reached the ice barrier at the South Pole in 1911 with his ship *Fram* powered by a 180 bhp Polar engine. In the same year the Atlantic was crossed for the first time by the motor ship *Toiler* equipped with two 180 bhp Polar engines.

Professor Collin described the development of the two-stroke engine with particular reference to the concentration, in 1950, on the development of the N and T-Type two-stroke engine, both utilizing the special Polar scavenging system—a development which reduced the number of types from seven to two.

The author went on to describe a number of considerations behind the introduction of the four-stroke working cycle, and the complications which could arise in the restricted air and gas passages on the smaller two-stroke engines, involving comparatively small turbochargers. Further development of the small two-stroke engine would suffer from the comparatively high resistance and low possible efficiencies and a greater susceptibility to deposits and disturbances in the smaller cross-sections due to the building up of deposit formations in the different passages.

It was emphasized that the main demand on a modern medium and high-speed engine must be the ability to work for prolonged periods between overhauls, a target more difficult to obtain with a two-stroke than with a four-stroke engine.

With extensive use of slides Professor Collin described the engine and performance results and particular reference was made to the adaptability of the engine for multi-engine systems and the introduction of a special synchronizer and synchrophaser governor equipment.

The speaker closed with slides of existing and proposed engine-room layouts showing the extensive available head room, an essential feature in roll-on/roll-off ships together with complete accessibility.

An interesting discussion reaching high technical level, followed and Professor Collin and Mr. Nordquist proved eminently capable in dealing with all points raised.

In proposing a vote of thanks Mr. R. Pike pointed out that the speaker had not referred to the number of F type engines built and he thought it only fair to state that over 100 engines had been delivered since 1962, the majority during the last two years. Mr. Pike thanked Professor Collin for the excellent paper and stated that the Scottish Branch were most impressed by the way in which he had handled the discussion.

The meeting closed at 8.10 p.m.

South Wales

Senior Meeting

The first lecture of the session, "Developments in the Operational Efficiency of a Cargo Fleet" was presented at 6.00 p.m. on Monday, 3rd October 1966, by Mr. N. V. McAslan of J. and J. Denholm Ltd., Glasgow. The lecture, which was held at the South Wales Institute of Engineers, was extremely successful and attracted a full attendance. It was followed by a discussion which went on well beyond the usual time and which finally closed when the lecture theatre had to be vacated.

Senior Meeting

A senior meeting of the Branch was held on Monday, 7th November 1966, at the South Wales Institute of Engineers, Cardiff, at 6.00 p.m. when a paper entitled "Further Progress in Automation" by R. Munton, B.Sc. (Vice-President) and J. McNaught (Member), was presented by Mr. McNaught.

Mr. N. C. James, B.Sc. (Vice-Chairman of the Branch) was in the Chair and after apologizing for the absence of Mr. T. W. Major, due to business commitments, welcomed Mr. McNaught and invited him to present his paper.

Mr. McNaught outlined the history and justification for the installation of automation equipment in the *Clan*

Macgillivray and the *Southampton Castle*. He went on to describe the problems and successes experienced with his ideas on the future of automation within the shipping industry.

An enthusiastic discussion followed which was reluctantly terminated by the Chairman.

In proposing a vote of thanks to Mr. McNaught, Mr. H. S. W. Jones (Member of Committee) directed some of his remarks to the Student Members present, indicating that they would require courage and initiative to fully participate in the radical changes which the industry was now undergoing. The vote of thanks was warmly received by those present.

A vote of thanks to Mr. James for taking the Chair at such short notice was proposed by Mr. A. H. John (Member of Committee).

The meeting closed at 8.15 p.m.

Annual Dinner

The Annual Dinner of the South Wales Branch was held on Friday, 21st October 1966, at the Park Hotel, Park Place, Cardiff.

The evening was overshadowed by the dark cloud of the Aberfan disaster which was uppermost in everyone's thoughts.

Mr. T. W. Major (Chairman of the Branch) presided at the Dinner at which 155 members and visitors were present. The principal guests included the President of the Institute, Sir Stewart MacTier, C.B.E., B.A. (Companion), and Mr. J. Stuart Robinson, M.A. (Director and Secretary), also local representatives from the many facets of the shipping industry.

After the Loyal Toast, the toast to "The Shipping Industry" was proposed by Mr. E. O. T. Blanford, General Manager and Director of Fairfields (Glasgow) Ltd., who gave mention to the radical changes taking place in the shipbuilding world and its plans for the future.

In responding to the toast Mr. T. B. Hutchison (Member) spoke of the problems arising from managing a tanker fleet and the solutions that were being tested in an effort to overcome them.

Sir Stewart MacTier, in proposing the toast to "The South Wales Branch", spoke of the need for a higher standard of engineer at sea, and thought that university graduates should be encouraged to join the shipping industry.

In reply the Chairman paid tribute to the Honorary Secretary, Mr. W. G. Fox, for his capable organization of the Dinner. Mr. Major then went on to point out the amount of capital investment for which chief engineers were now responsible and likened the chief engineer to the managing director of a medium-sized firm.

It was with much pleasure that the Chairman then proposed the toast to "Our Visitors".

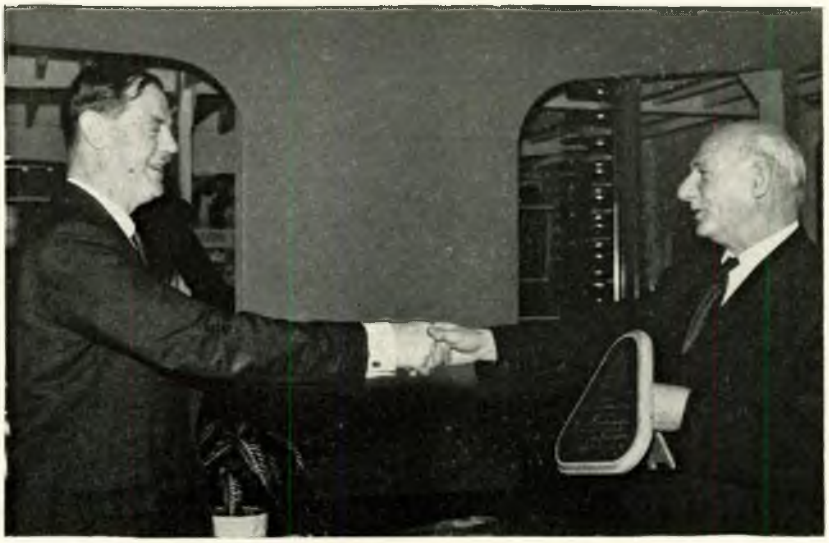
Mr. Neville Williams of Mountstuart Drydocks Ltd., replied on behalf of the visitors, his ready wit and humour considerably enlivening the proceedings.

Kingston upon Hull and Humber Area, North West England and North Midlands

Following the very successful meeting on 18th May 1966 when members of the Kingston upon Hull and Humber Area Branch, North West England Branch and North Midlands Branch visited Yorkshire Imperial Metals Ltd., a presentation of a plaque to the company was made on Monday, 17th October 1966.

Mr. W. R. D. Macdonald, Managing Director of Yorkshire Imperial Metals, received the plaque from Mr. G. Skelton, M.B.E. (Chairman of the Kingston upon Hull and Humber Area Branch Committee). Mr. D. A. Taylor (Honorary Secretary of the Branch) together with Mr. H. V. Campbell and Mr. G. Wilkinson (Chairman and Honorary Secretary respectively of the North Midlands Branch) was also present. It was not possible for the North West England Branch Committee to be represented on this occasion.

Presentation of Plaque



Mr. G. Skelton, M.B.E. (Chairman of the Kingston upon Hull and Humber Area Branch) seen here presenting Mr. W. R. D. Macdonald (left) Managing Director of Yorkshire Imperial Metals, with the plaque presented to the company by members of the Kingston upon Hull and Humber Area, North West England, and North Midlands Branches of the Institute, following their visit to Yorkshire Imperial Metals last May

Election of Members

Elected on 16th November, 1966

MEMBERS

Elections

Duncan Macnaughton Campbell
George Carter
William Cochran
George Theodore Alexander Darley, Eng. Lieut. Cdr., R.N.
William John Davies, B.Sc.
John Noel Patrick Douglas
Peter Hansen
Ronald Murray Hardie

William James Holden
Aubrey Constantine Karagianis, Lt. Cdr., C.D., R.C.N.
Norman Metcalfe
Richard Lawrence Mohan, Capt., U.S.N. (ret.)
Francis Joseph O'Connor
William Ovens
David Henry Pepper, Lt. Cdr., R.N.
John Sidney Rushton
Alexander Pattullo Simpson
Manus Tolland
John Walker
Norman James Walker
Goschtchevsky Yerzy, M.Sc.

Transferred to Member from Associate Member
Mohindar Pal Singh Bhinder, Lt. Cdr., I.N.
Edward Ronald Evans
John Angus Lazarus
Norman Silcock
Alan Raymond Stamp
Charles Norman Staines

Transferred to Member from Associate
David George Welton, Lt. Cdr., D.S.M., R.N.R.

Transferred to Member from Graduate
Michael John Neeves, Lt. Cdr., R.N.

ASSOCIATE MEMBERS
Elections
Dennis Arrow
Thomas Claude Ball
James Sydney Brownson
Michael Dunleavy Bryce
William Cochrane
Roy Curtis
Thomas Anthony Emmett
Arthur Gordon Green
David Roderick Inglis
Harry Jenkins, Lieut. (E) (LD), R.C.N.
Brian Jones Lewis
James Moore McAuley
Michael Davenport Medlicott
John Bertram Mills
Prodyot Kumar Mitra, Lieut., I.N.
Eric William Poingdestre
Roy Edward Rowe
Charles Reginald James White, Eng. Lieut., R.N.

Transferred to Associate Member from Graduate
Roger Josiah Brunton
David Clifford
Ronald George Head
Paul Hepburn
John Arthur Hodges
Ronald Arthur Johnstone
James Derrick Judge
Charles Rex Sadgrove
Alexander Duncan Tosh

Transferred to Associate Member from Probationer Student
Robin Darley Ebsworth

ASSOCIATES
Elections
James Henry Brennan
Frederick Neal Brookes
V. V. Chetty
Michael Anthony Easter
Vernon Horace Irvin
Courtney John Lee
Hugh Alan MacDonald
Robert F. Manning
Kenneth Morton, Eng. Sub. Lieut., R.N.
John Michael Nolan
Timothy Michael O'Leary
Melvin George Painter
Sayed Ali Khallad Raza
Gordon John Stewart
Ronald McDonald Summers
Stuart Bernard Thorn, Eng. Sub. Lieut., R.N.
Jack Albert Townsend
Keith Lawrence Walker
George Edward Wilson
Valiyaparampil Kurian Zachariah

GRADUATES
Elections
Peter Bruce, Lieut., R.N.
Md. Ainul Haque
Anwar Naimat Lodhi
Ian Arthur Maurice McRae, Lieut., R.N.
Kevin Mercer
Waseem Ahmad Qurashi
Michael Joseph Koolhoven Riches
John Alfred Slade
Michael Smith
Colin Ivor Richard St. John-Browne
William John Twose
Richard Thomas Jon Wall

Transferred to Graduate from Student
Michael John Brown
Albert William Errington
Robert Ian Harper
Norman Leslie Pallin
Afolabi Sorungbe

Transferred to Graduate from Probationer Student
Michael Baddeley
Roderick Barry Cotton
Gordon Charles Flaxman
Ronald James Groombridge
Keith Daniel Hole
Geoffrey Cowper Taylor

STUDENTS
Elections
Jashimuddin Ahmad
Iftikher Ahmed
David Butler
Theodore W. J. Da Silva
Md. Ekramul Haque
Mohammed Akhter Hossain
A. K. M. N. Huda
F. B. M. K. Iqbal
Md. Monirul Islam
Ashok Kumar Jain
Liaqat Ali Khan
John Mitchell Macfarlane
Amir Mamoor
Robbin S. Phillips, B.Sc.
Md. Abul Quasem
Md. Azizur Rahman
Nazrul Islam Siddiqui
Swaminathan Sivapathasunderam
Jeremy Whitaker
Alexander Whyte

Transferred to Student from Probationer Student
Michael John Endersby
Edward Finn
John Stanley Owen
David John Solomon

PROBATIONER STUDENTS
Maurice Edward Button
Alistair Colvin Gillone
David Pullen

OBITUARY

Sir E. Julian Foley, C.B.

Sir E. Julian Foley, C.B. who died on Friday, 18th November 1966 was President of this Institute in 1938-1939.

Sir Julian was born in Liverpool on 19th October 1881. He received his education at Liverpool College and at Liverpool and London Universities.

He joined the Civil Service in 1907 in the Transport Division of the Admiralty. He became Assistant Director of Transports in 1915 and two years later, in 1917, he was appointed director of Sea Transport in the Admiralty and Ministry of Shipping and served in this capacity for the remainder of the Great War.

In 1920 he became Principal Assistant Under Secretary to the Mines Department in which he served until his appointment in 1927 as Director of Sea Transport under the administration of the Board of Trade. In 1929 he was appointed to the position which he held during his term of office as President of the Institute, that of Under Secretary in charge of the Mercantile Marine Department at the Board of Trade.

From 1939 to 1942 Sir Julian was Liaison Officer, Ministry of Supply (Raw Materials). He was with the East African Governors' Conference in 1943 and in 1945 served as the representative of the Ministry of Supply (Raw Materials) in India. He was Leather Controller at the Board of Trade from 1947 until 1950.

As well as art and architecture, his interests included golf, fencing and walking and he was the author of a *Manual of Psychology*.

Sir Julian who was a holder of the Jubilee and Coronation Medals, was made a Companion of the Bath in 1919 and received his knighthood in the Jubilee Honours of 1936. His foreign honours included those of Chevalier, Ordre de la Couronne (Belgium); St. Anne of Russia (second class); Officer of the Legion of Honour; Order of the Crown of Italy and the Liberty Medal (Denmark).

Charles Chapman (Member 9021) died on 23rd July 1966.

He was born on 25th August 1891 and served his apprenticeship with Gourlay Bros., Dundee, Vittoria Foundry and Crown Harvey, Glasgow.

The holder of a First Class Board of Trade Certificate, he saw sea service with several companies from 1913 and joined Ellerman Lines, with whom he served until his retirement in 1957, as a third engineer in 1921, rising to become chief engineer in 1938.

Mr. Chapman was elected a Member of the Institute in December 1939. He is survived by his wife.

Arthur Vernon Fisher (Associate 19251) died on 22nd August 1966.

From 1915 to 1918 he was a deck officer cadet with the British India Steam Navigation Co. Ltd., with whom he served until 1928, gaining the rank of first mate by 1927, when he became assistant superintendent of the company's coal dock.

In 1928 he joined Lardner North and Co., marine superintendents, surveyors and assessors, Calcutta, as an assistant. He became a partner in the firm in 1930 and senior partner in 1935.

Mr. Vernon Fisher was also Honorary Secretary of the

Owners and Trainers Association of the Royal Calcutta Turf Club for more than thirty years.

He was elected an Associate of the Institute in September 1957. He is survived by his wife.

Wilfrid Goddard (Member 6152) died on 18th April 1966. He was born on 7th October 1900.

Mr. Goddard served his apprenticeship with J. Samuel White and Co. Ltd., Cowes, Isle of Wight, from 1916 to 1921, after which he joined the British India Steam Navigation Co. Ltd., with whom he served until 1936. In 1938 he joined the Board of Trade as an engineer surveyor and remained with them until his retirement in October 1965.

Mr. Goddard held a First Class Board of Trade Certificate and an Extra First Class Certificate. He was elected a Member of the Institute in March 1929.

Frederic Gray (Member 11535) died on 5th August 1966. He was sixty-seven years old.

He served his apprenticeship with R. and W. Hawthorn, Leslie and Co. Ltd., Newcastle upon Tyne from 1914 until 1919, when he joined the British India Steam Navigation Co. Ltd. as a fifth engineer, rising to third engineer by 1926, the year in which he gained his First Class Steam Certificate. From 1928 to 1931 he was engineer of a sisal plantation in Kenya, of which he became manager in 1932. In 1933 he joined Kenya and Uganda Railways and Harbours as Mombasa port services engineer.

On leaving East Africa in 1953 he joined the Niarchos Group and served in s.s. *Saxonmead* as chief engineer. After serving in tankers for several years he served in cargo, fast cargo and passenger vessels on the U.K.-South America run until his retirement in 1964.

Mr. Gray was elected a Member of the Institute in November 1947. His wife survives him.

Kenneth E. Greig (Honorary Life Member 2691) died on 27th October 1965. He was eighty-three years old.

He served his apprenticeship with the Caledonian Railway Co., Perth, and with the North East Marine Engineering Co. Ltd., Wallsend.

In 1905, after two and a half years sea service during which he gained his First Class Board of Trade Certificate, he rejoined the North East Marine Engineering Co. as assistant manager.

In 1909 he went to Hong Kong as engineering manager of the Taikoo Dockyard and Engineering Co. Ltd., becoming general manager in 1927. He retired from this position in 1937.

In 1939 he joined the Department of Merchant Shipbuilding and Repairs and two years later returned to Scotland as managing director of Scotts Shipbuilding and Engineering Co. Ltd., Greenock, where he remained until his retirement in 1945.

Mr. Greig was elected a Member of the Institute in December 1912. He was elected an Honorary Life Member in January 1963. He leaves two daughters.

Oswald Philip Horrobin (Associate Member 26095) died on 23rd May 1966 when the m.v. *Kaitawa* sank off the North Cape of New Zealand. He was forty-four years old.

He served his apprenticeship with Winstone Ltd., Auckland, New Zealand. From 1945 to 1947 he was a fuel injection serviceman in charge of C.A.V. depots in Christchurch, Dunedin and Invercargill. In 1948 he became a fitter/shift engineer with the British Phosphate Commission, Nauru Islands. He then joined the Canterbury Shipping Co. as second engineer in the m.v. *Gale* and in 1949 joined Andrew Weirs Ltd. as senior third engineer in m.v. *Clydebank*. Later he served for four years as mechanical overseer at Nandi airport, Fiji Islands. In 1956 he joined the Union Steamship Co. as a fourth engineer, rising to chief engineer by 1962.

He held a First Class Steam Certificate, a Second Class Motor Certificate and a First Class New Zealand Coastal Motor Certificate.

Mr. Horrobin was elected an Associate Member of the Institute in May 1963. He is survived by his wife.

William Keggin (Member 17947) died on 18th May 1966 at the age of sixty-three. He served his apprenticeship with H. and C. Grayson Ltd., Liverpool, from 1918 to 1923, gaining his National Certificate in Mechanical Engineering in 1922.

From 1923 to 1925 he worked for Cammell Laird and Co. Ltd. and William Beardmore and Co. Ltd. For the next three years he served as junior engineer with Alfred Holt and Co. He then entered service with T. and J. Harrison Ltd., rising from fourth to second engineer and gaining his First Class Certificate in 1930. In 1943 he was appointed foreman engineer of a shore gang with the company. By 1948 he had become assistant superintendent engineer and remained in this post until 1960 when he was appointed Liverpool dock repair superintendent.

He suffered a cerebral haemorrhage in May 1965 and died twelve months later.

Mr. Keggin was elected a Member of the Institute in October 1956. He is survived by his wife.

Andrew Cecil Kennedy (Member 6227) died on 12th September 1966. He was seventy-seven years old.

He served his apprenticeship from 1906 to 1909 with the Urban Electric Supply Co. Ltd. of Berwick-on-Tweed. After this he served as electrician in s.s. *Mantua*.

During the First World War he served as a pilot with the rank of Second Lieutenant in the Royal Flying Corps and later the Royal Air Force.

Between the wars he served with Kennedy and Son, Galashiels, C. H. Bailey, Graham and Co., Cardiff, and as resident electrical engineer with the Milford Haven Urban District Council.

He was in the Territorial Army Reserve and was recalled for service with the Royal Engineers in 1940. He remained in the Army until 1954, receiving the Territorial Efficiency Decoration in 1945 and becoming a Major by 1952.

On leaving the Army he was with the science department, Wellington College until his retirement in 1958.

He was elected an Associate of the Institute in July 1929 and became a Member in October of the same year.

Douglas Henry King (Member 4596) died on 24th December 1965. He was sixty-two years old.

Educated at Merchant Taylors School, he served his apprenticeship with R. H. Green and Silley Weir Ltd., Blackwall.

From 1925 to 1935 he served in vessels of Furness Withy Ltd., rising from junior engineer to senior second engineer and obtaining a First Class Board of Trade Certificate. For the next three years he was in charge of the erection of ventilating and drying equipment for Heat and Air Systems Ltd., of London. In 1941 he became works director and company secretary of the Lion Stamping Co. Ltd.

Mr. King was elected a Graduate of the Institute in August 1922; in 1927 he was elected an Associate Member and became a full Member in July 1944. He was also a Member of the London Association of Engineers.

He is survived by his wife.

Donald Livingstone (Member 19180) died on 14th September 1965. He was fifty-five years old.

He was apprenticed to Richardson Westgarth and Co. Ltd. and started his career at sea with the British Tanker Co. Ltd. in 1932, rising from sixth engineer to third engineer. From 1937 he served as a draughtsman with Harland and Wolff Ltd., Belfast, until 1938 when he joined the New Zealand Shipping Co. Ltd., leaving as chief engineer in 1943.

Subsequently he served with Trinder Anderson, J. and T. Harrison and Co. Ltd., the Shell Tanker Co., the Standard-Vacuum Transportation Co. Ltd. and the Esso Petroleum Co. In 1951 he obtained a First Class Steam and Motor Certificate.

Mr. Livingstone was elected a Member of the Institute in September 1957.

He leaves a widow.