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DESIGN OF MARINE DIESEL ENGINE CRANKSHAFTS: COMPARISON OF MEASURED AND CALCULATED STRESSES USING THE PROPOSED CIMAC RULES

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Design of Marine Diesel Engine Crankshafts: Comparison of Measured and Calculated Stresses Using the Proposed CIMAC Rules*

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1. SYNOPSIS

During 1972–1979 a Working Group of Conseil International Des Machines A Combustion (CIMAC) developed new rules for the design of marine diesel engine crankshafts, with the aim of proposing these for standardizing the different rules issued by the major classification societies. In 1979 a research group from Bureau Veritas (BV), Lloyd's Register of Shipping (LRS) and MAN started a project to measure and compare the measured and calculated stresses in crankshafts and output shafts using the different design rules. The authors describe how the work was planned and the points where the measurements were taken in a two stroke and a four stroke engine. The actual readings were then compared with calculated readings using the proposed CIMAC rules and although some of the readings coincided well, there were several discrepancies. However, all the discrepancies came within the safety margin and they can be reduced by relatively simple modifications to the rules.

2. CIMAC SUB-GROUP 'CRANKSHAFTS' PROPOSED RULES

Since the crankshaft is one of the most important components of a diesel engine, classification societies have detailed rules on the dimensioning of this component and manufacturers who market engines internationally require approval for their crankshafts from all the major classification societies. There are, however, considerable differences between the rules of individual classification societies (Fig. 1).

As a result of this situation, engine builders are very keen to have the large number of different rules replaced by unified requirements. The CIMAC Working Group 'Classification Societies', which is the engine builder's panel responsible for co-operation with IACS (International Association of Classification Societies), therefore made arrangements as early as 1972 for a sub-group 'Crankshaft dimension' to propose a method of calculation.[1]

The new method should be completely transparent and, if possible, cover all essential points and be in line with the latest state of the art, but be simple enough for day-to-day use. As engine builders have been handicapped in the past by those classification societies which demanded overdimensioned crankshafts, the stresses permitted in the new proposal must be higher rather than lower than those allowed at present.

In this paper the stresses measured will be compared with those calculated by the CIMAC proposal. The essential features of the calculation method are described below.

(a) Safety coefficient

A safety coefficient (S) is essential for the rating of a crankshaft and is defined as:

$$S = \sigma_{DW}/\sigma_v \qquad (1)$$

where σ_{DW} is the fatigue strength of the crankshaft and σ_v is the equivalent stress at the position exposed to the highest stress.

So as not to depart from past experience, the 1.15 minimum value for S has been maintained and was verified by recalculating crankshafts which have given satisfactory service.

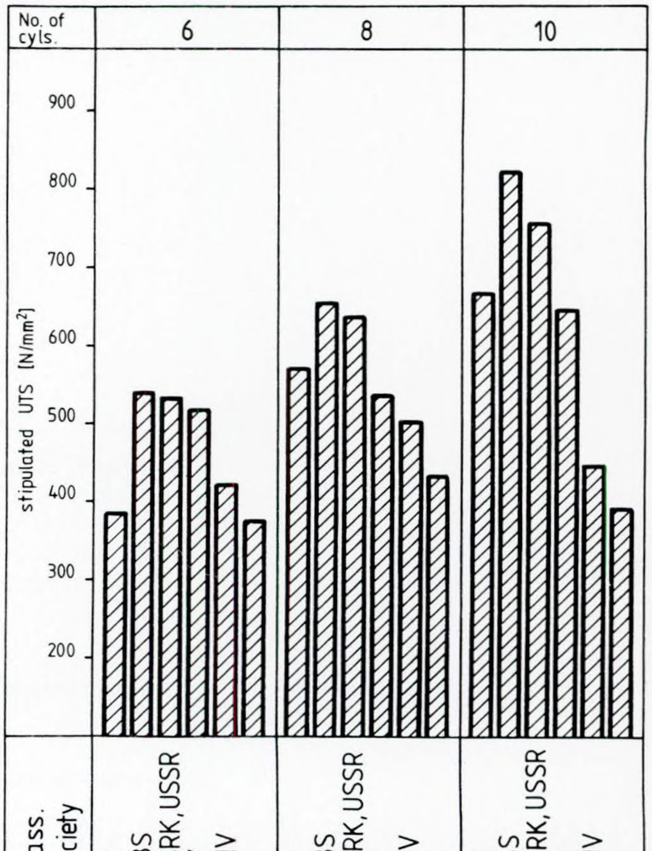

FIG. 1 Variations in a crankshaft UTS according to the rules of various classification societies for engines with three different numbers of cylinders

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† The activities of MAN Diesel Engine Division and of B & W Diesel, Copenhagen were combined into MAN-B & W Diesel in 1984.

(b) Fatigue strength

The fatigue strength of a crankshaft largely depends on the component. The literature available at the beginning of the work was of little assistance in calculating the fatigue strength values from the static strength values of the material. Almost 100 results of fatigue strength investigations made throughout the world on different sized crankshafts were collected and used to derive an empirical formula.[2]

(c) Definition of the equivalent stress

The equivalent stress used in Eqn (1) necessitates the selection of a suitable failure theory. It was, therefore, decided to adopt the von Mises criterion, which is widely used throughout the world and which, according to investigations of ductile materials,[3] errs on the safe side.

For the stress condition obtaining in the crankshaft, the equivalent stress is:

$$\sigma_v = \sqrt{(\sigma_B^2 + 3\tau^2)} \qquad (2)$$

where σ_B is the maximum alternating bending stress and τ is the maximum alternating torsional stress, the mean stresses being neglected.

When calculating the decisive safety coefficient, Eqn (2) is worked out for the crankpin and journal fillets of all cranks and the maximum value is inserted in Eqn (1).

(d) Stress concentration factors

When calculating the notch stress in fillets, the selected nominal stress is multiplied by the stress concentration factor. This method avoids complicated calculations of doubtful accuracy.

When the work of the CIMAC group commenced, a research project of the German Internal Combustion Engine Research Association (FVV) had just been completed. Within the scope of this project, stress concentration factors in bending and torsion were determined for a very large range of geometric dimensions, with use having been made of all important former investigations of other authors.[4] These stress concentration factors were used for the CIMAC proposal, which also made it necessary to take over the pertaining nominal stress definition.

(e) Bending stresses

According to (d) above, the nominal alternating bending stress used in calculating the maximum alternating bending stress is defined as the ratio of the bending moment in the web centre to the resistance moment in the web cross-section. The FVV investigations showed that, to obtain the nominal stresses in the journal fillet, an additional stress, proportional to the ratio of the transverse force to the cross-sectional area of the web, must be added. Separate stress concentration factors for this stress were worked out by FVV.

The CIMAC proposal uses either the single crank or the continuous beam method for obtaining the web bending moment and transverse force. In the single-crank method, the crank being examined is cut in the centre of the adjacent main bearings and is simply supported there. This statically determinate method is used by many classification societies and engine builders, but ignores the clamping effect of the adjacent cranks and the effect of the forces passing from the adjacent crank to the crank under review.

The continuous beam method is expected to be more accurate. Here, the crankshaft is represented by a substitute beam with a constant cross-section which is subjected to the loads from all forces acting on the entire crankshaft. For the cross-sectional area of the substitute beam, a circular diameter is determined on the basis of experience.

Although the continuous beam method requires more calculation work, it normally reveals stresses which are lower and coincide better with those measured. If the same safety factor is used in both calculation methods, the continuous beam method gives a lower hidden safety factor and therefore better utilization of the material.

(f) Torsional stresses

The nominal stress is defined as the ratio of the calculated vibratory torques to the polar moments of resistance of the respective crankpins or journals. The torsional moments are determined by the forced-damped vibration method[5,6] and, provided a substitute system is used consisting of single masses and stiffnesses of the crankshaft and the system components connected to it, satisfactory approximation to the measured stress can be achieved.

During the design phase of an engine, a torsional vibration system from which the highest equivalent stress according to Eqn (2) can be expected must be used for determination of the decisive torsional stress.

(g) Additional stresses

The bending stresses in (e) above refer to ideal bearing alignment and quasi-static crankshaft performance without any bending or axial vibration.

The ideal bearing alignment may be neither achieved nor maintained owing to production errors and hull distortion, resulting in higher bending stresses than calculated. To allow for this, and based on practical experience,[7] the CIMAC proposal recommends adding ± 30 N/mm^2 for two stroke and ± 20 N/mm^2 for four stroke engines. The reason for the larger value in the two stroke is that this type of hull deformation has only been found to affect these engines.

It is only in exceptional cases that the aforementioned bending and axial vibration-induced additional stresses occur. As a generally recognized calculation of such processes has not yet been established, it is the engine builder's responsibility to make allowance for these additional stresses.

3. PLANNING AND HANDLING THE PROJECT

(a) Sharing of tasks

After taking into account the facilities at the disposal of the three partners, the tasks were allocated as follows.

- MAN was to select suitable engines and shipyards prepared to co-operate; contact these shipyards; attend to the technical handling of the project on the testbed and onboard; and install the strain gauge systems on the crankshafts.
- Bureau Veritas was to measure journal displacement paths on the two stroke engine; the distortion of the engine frames and support structures; and the static stresses in the crankshaft of the four stroke engine, on a rigid table whilst the bending and support positions were varied.
- Lloyd's Register of Shipping was to measure the dynamic stresses on the testbed and onboard, using an eight channel telemetric system and a magnetic tape recorder.

Within the scope of the trial, all three partners would evaluate the stresses in the crankshaft in accordance with their respective in-house methods, MAN using the CIMAC proposed rules.

(b) Selection of engines

Owing to the well-known design differences in crankshafts, it was decided that the measurements should be taken on a two-stroke engine with a semi-built crankshaft and on a four-stroke engine with a monobloc crankshaft. The engines were mainly selected on availability, since a condition of the project was that static measurements on the crankshaft had to be taken prior to installation in the engine (only possible with the four stroke engine), before the shop test run and sea trials. The engines selected were:

Two-stroke engine: MAN. K7SZ 70/125 BL
Power output = 10 640 kW
Speed = 130 rev/min

Four-stroke engine: MAN 8L 40/45
Power output = 4400 kW
Speed = 600 rev/min

(c) Points of stress measurement

The main aim was to record the bending and torsional stresses in the crank plane and in the plane at 90 deg to the crank plane, caused by the gas and mass forces and any misalignment from the bearing track deformation. The measuring points selected are shown in Fig. 2. In addition, one torsion measuring point consisting of a full strain gauge bridge was provided either on the output shaft or on the line shafting.

(d) Reference quantities

The following were also measured:

- cylinder cyclic pressure for one of the cranks concerned and
- TDC of cylinder no. 1 (the cylinders and cranks being counted from the coupling end of the engine).

(e) Selection of cranks to be examined

- *Two-stroke engine:* End crank at the coupling end to record any influence of line shaft bending. End crank at the free end to record the anticipated highest bending stress.
- *Four-stroke engine:* End crank at the coupling end, to record any influence from the flywheel or coupling; and crank no. 4, as this is one of the two adjacent cranks with the same orientation in the middle of the engine with the highest bending stress.

(f) Measurement programme

Measurements were to be taken at no load, 50%, 75% and 100% load; and when turning the crankshaft under cold and warm conditions.

4. COMPARISON BETWEEN MEASURED AND CALCULATED STRESSES: GENERAL

As mentioned in Section 3, the results of the measurements were recorded by LRS on magnetic tapes. For further investigation, abstracts of these tapes were transferred to paper and circulated to the other two partners.

The aim of MAN's interpretation of the measurement results was to prove that there was a satisfactory coincidence between the stresses calculated by the CIMAC method and the actual measurements or, at least, that the use of the CIMAC method for dimensioning a crankshaft does not involve any risk. Since, in the CIMAC method, the operational reliability of a crankshaft is defined by the ratio of fatigue strength to equivalent stress, the latter point is proved if insertion of the measured values into Eqn (1) gives a safety coefficient value which is either the same or higher than the calculated value. It would be ideal if the calculated stresses, as a function of time, tallied with the measured stresses. To check this, the graphs provided by LRS were digitized by MAN and plotted together with the calculated results. These diagrams show satisfactory coincidence, but also some discrepancies. Since some of these discrepancies are typical of either two-stroke or four-stroke crankshafts, the results for the two engine types are dealt with separately in Section 5. However, some typical deviations are discussed here.

Deviations consisting of a parallel displacement between the plots of calculated and measured readings may be due to measuring equipment drift. Drift is due to temperature differences in the engine, which are often unavoidable with the units used for the strain gauge technique. Since only the alternating stress values are used to calculate equivalent stress in the CIMAC method, there is no need to attach a great deal of importance to parallel displacements.

The results from the four-stroke engine revealed discrepancies which were initially inexplicable and MAN took an opportunity to recheck the results from the co-operative research project on a four-stroke engine of the same type and with the same number of cylinders, which happened to be on the testbed in Augsburg. The work was confined to an assessment of the important measuring points. Since it was possible to clarify during this repeat measurement some of the discrepancies that occurred during the original measurements on the

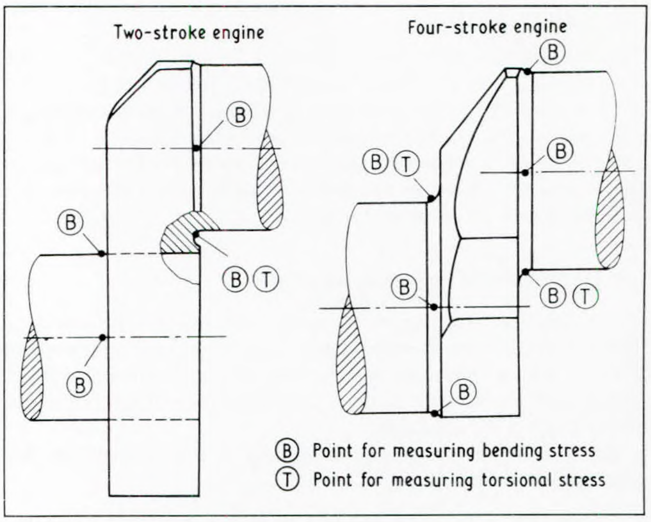

FIG. 2 Measuring points on cranks of both crankshafts

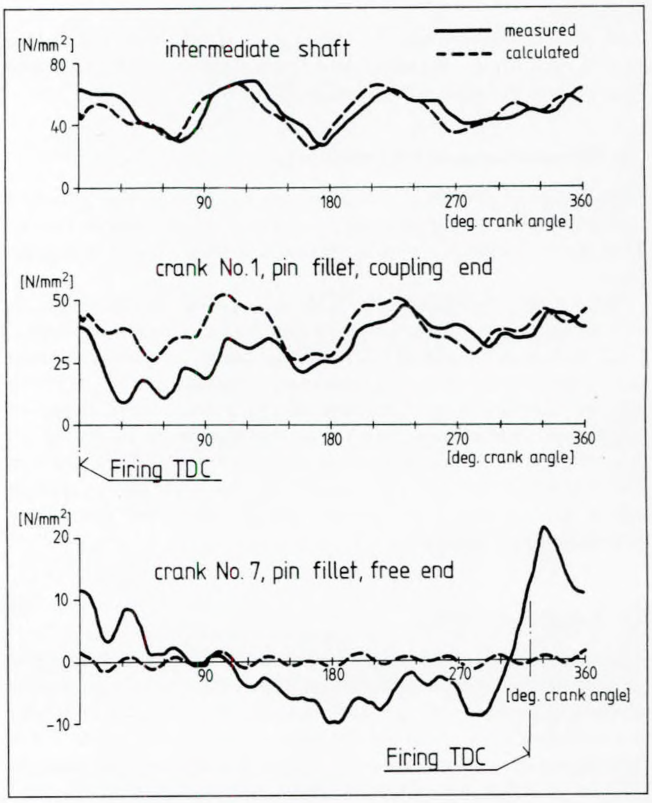

FIG. 3 Torsional stresses in the two-stroke engine

Table I: Comparison between measured and calculated results of the two-stroke engine (installed onboard)

	Torsional stress, τ (N/mm²)	Bending stress, σ_B (N/mm²)	Additional stress, σ_{add} (N/mm²)	Equivalent stress, σ_{equ} (N/mm²)	Deviation from measured value, $\Delta\sigma_{equ}$ (%)	Safety coefficient,[a] S
CRANK 1:						
1.1 Measured	19	47	—	57	—	3.51
1.2 Calculated, with additional stresses measured	14	42	—	48	−16	4.15
1.3 Calculated according to CIMAC, statically indeterminate	14	40	30	74	+29	2.73
1.4 Calculated according to CIMAC, statically determinate	14	84	30	116	+104	1.72
CRANK 7:						
7.1 Measured	16	69	—	74	—	2.70
7.2 Calculated, with additional stresses measured	2	75	—	75	+1	2.67
7.3 Calculated according to CIMAC, statically indeterminate	2	71	30	101	+36	1.99
7.4 Calculated according to CIMAC, statically determinate	2	84	30	114	+54	1.76

[a] S based on fatigue strength (σ_{fat}) = 200 N/mm².

four-stroke engine, only the results of the repeat measurements are used for the four-stroke engine in Section 5.

The measurements on the two-stroke engine during the co-operative research project did not reveal any fundamental differences between the measurements taken on the testbed and those taken onboard, apart from a different torsional vibratory state which was to be expected. To simplify matters, only the results of the measurements taken onboard are considered in Section 5.

As may be noted from Section 3, a large number of strain gauges were applied to every crank examined. To simplify the comparison, only the maximum bending and the maximum torsional stresses in the fillet are considered. However, the stresses in the fillet have a rather steep gradient and it is therefore possible that the maximum stress was not recorded, as the maximum stress could occur between two strain gauges. On account of this phenomenon, a 5–10% deviation from the measured results is feasible.

It should also be noted that only calculated stress concentration factors could be used in the stresses calculated, whereas experimental verification is the usual practice if high accuracy is required. For this reason, a further 5–10% deviation is possible in the case of the calculated stresses.

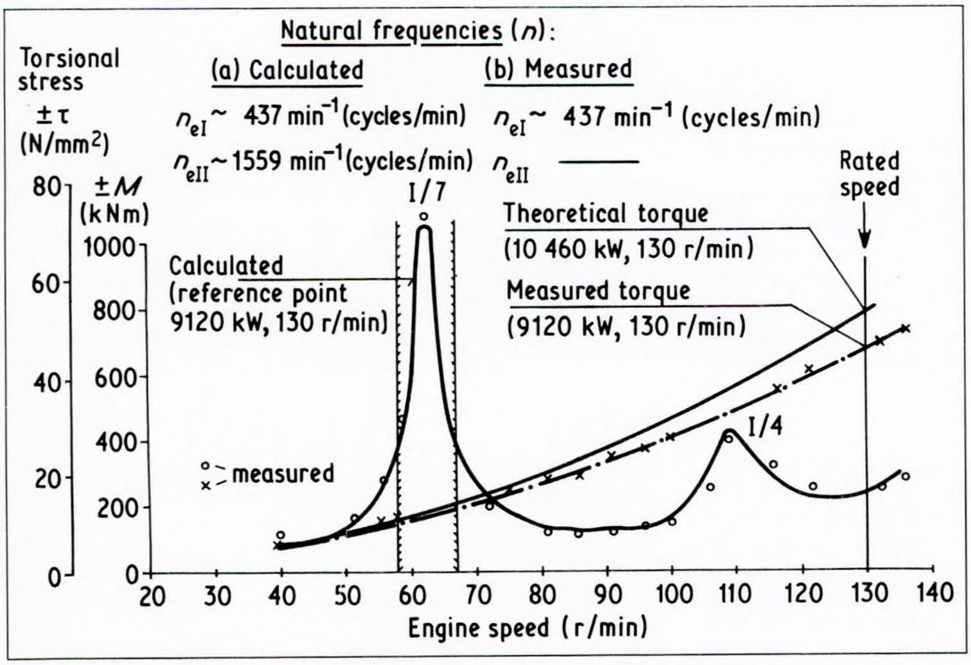

FIG. 4 Torsional stresses τ and torque M in the intermediate shaft of the two-stroke engine

5. COMPARISON BETWEEN CALCULATED AND MEASURED STRESSES

To start with, the calculated and measured stresses were compared on the basis of stresses plotted against time in the joint diagrams. For those fillets in which stresses were measured, the safety coefficient differences between the measured and calculated stresses by different methods were compiled in tabular form.

5.1. Two-stroke engine

It is not standard practice in the CIMAC and other calculation methods normally used to calculate stresses in the fillet at the journal next to the shrink fit, as the fillet hardly exists because the dimensions of the journal and the shrink fit diameters are only slightly different. The reason for the omission of this calculation is that, in comparison with the crankpin fillet, the stresses are low. The measurements taken have confirmed this and the journal fillet of the semi-built crankshaft does not have to be considered.

5.1.1 Torsional stresses

Figure 3 compares the calculated and measured torsional stresses in the intermediate shaft and in the two crankpin fillets at a power output of 10 302 kW at 136 rev/min. The crankshaft stress amplitudes from Fig. 3 are given in Table I.

Figure 4 compares the torsional stresses measured and the values calculated as a function of engine speed within the intermediate shaft.

Coincidence of the positions of the I/7 and I/4 resonances means that the substitute system theoretically selected for the torsional stress calculation reflects the plant performance very well. It is also apparent that the torsional stress in smooth shaft sections can be determined accurately by the forced-damped vibration method; the magnitude of the stresses calculated in the resonance itself largely depending on the damping values used for the calculation. In the example examined, the assumed overall damping agreed very well. Finally, on account of the low draught of the ship, the actual mean-torque-induced stress was lower than the one calculated and used in Fig. 4. The vibratory torque calculation used the mean torques measured.

In Fig. 3, the torsional stresses calculated in the intermediate shaft as a function of time coincide very well with the torsional stresses measured; but there are considerable differences between the calculated and measured torsional stresses in the two crankpin fillets. The easiest way of explaining these differences is to look at the graphs for crank no. 7, where the calculated torsional stress is virtually zero. This is understandable since there is only the mass of one web and one flange between the crankpin of crank no. 7 and the free end of the crankshaft. Since the torque must be zero at the free end, there can only be minimal stress in the free-end fillet of the last crank. The measured torsional stress shows a peak in the zone after the firing of cylinder no. 7, which gives rise to the assumption that there is some relationship with the firing force. This phenomenon is discussed in detail in Section 7.

5.1.2 Bending stresses

The calculated and measured stresses are shown in Fig. 5 for the two fillets measured. The deducible values have been included in Table I.

In order to assess the quality of the calculation using the continuous beam model, allowance must be made for the additional stresses produced in the fillets as a result of bearing misalignment, which play a role in the case of the two-stroke engine. These stresses were determined separately by means of a turning test in the warmed-up condition. The assumption that the stresses measured during operation contain these stresses which are superimposed on the stresses from combustion and mass forces is justified, as the calculation method can only calculate the latter. To permit an objective comparison, the calculated stresses were corrected by the additional stresses measured. The curves shown in Fig. 5 are therefore those that could be calculated provided the misalignment is known. The deducible amplitudes from this are entered in Table I for each crank.

Figure 5 compares the corrected calculated and measured bending stresses in cranks no. 1 and no. 7. It will be seen that in crank no. 1 the measured bending stress is about 10% higher than that calculated, whereas in crank no. 7 it is about 10% lower.

5.2 Four-stroke engine

None of the former rules distinguished between crankpins and journals in the stress calculation. However, the FVV results, which have been incorporated into the CIMAC proposal, make a distinction between the stress concentration factors for crankpin fillets and journal fillets. In the case of the four-stroke engine, the comparison between calculation and measurement has been extended to four fillets and the output shaft.

As discussed in Section 4, the results from M.A.N.'s repeat measurements taken on the testbed with the engine developing the full 4400 kW power output at 600 rev/min were used for the comparison between calculated and measured readings. Only the torsional measurements taken on the output shaft during the joint measurements were used, as they agreed with the calculated results and there was therefore no point in repeating the trials.

5.2.1 Torsional stresses

In Fig. 6, measured and calculated stresses are plotted against time. (The deducible amplitudes are given in Table II.) The diagrams of pin fillets in Fig. 6 also show the firing position of the cranks in question and of adjacent cylinders, and it can be seen, especially at crankpin fillet no. 4, that, at about 40 deg after firing, the difference between calculated and measured readings is relatively large. It may be assumed that the web torsion effect described in Section 7 and the tangential force on the cylinder under review and of adjacent cylinders play a part

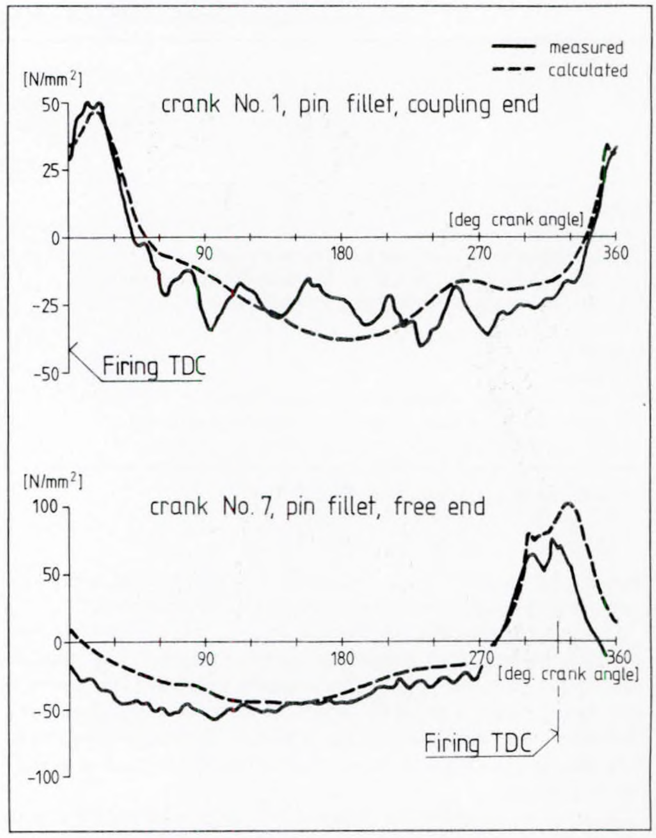

FIG. 5 Bending stresses in the two-stroke engine (measured onboard)

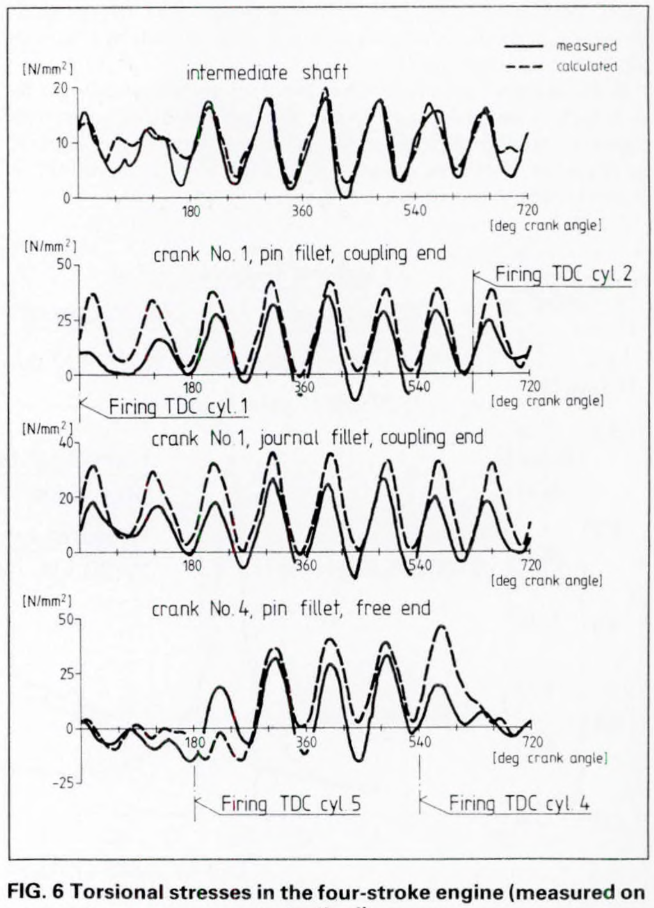

FIG. 6 Torsional stresses in the four-stroke engine (measured on testbed)

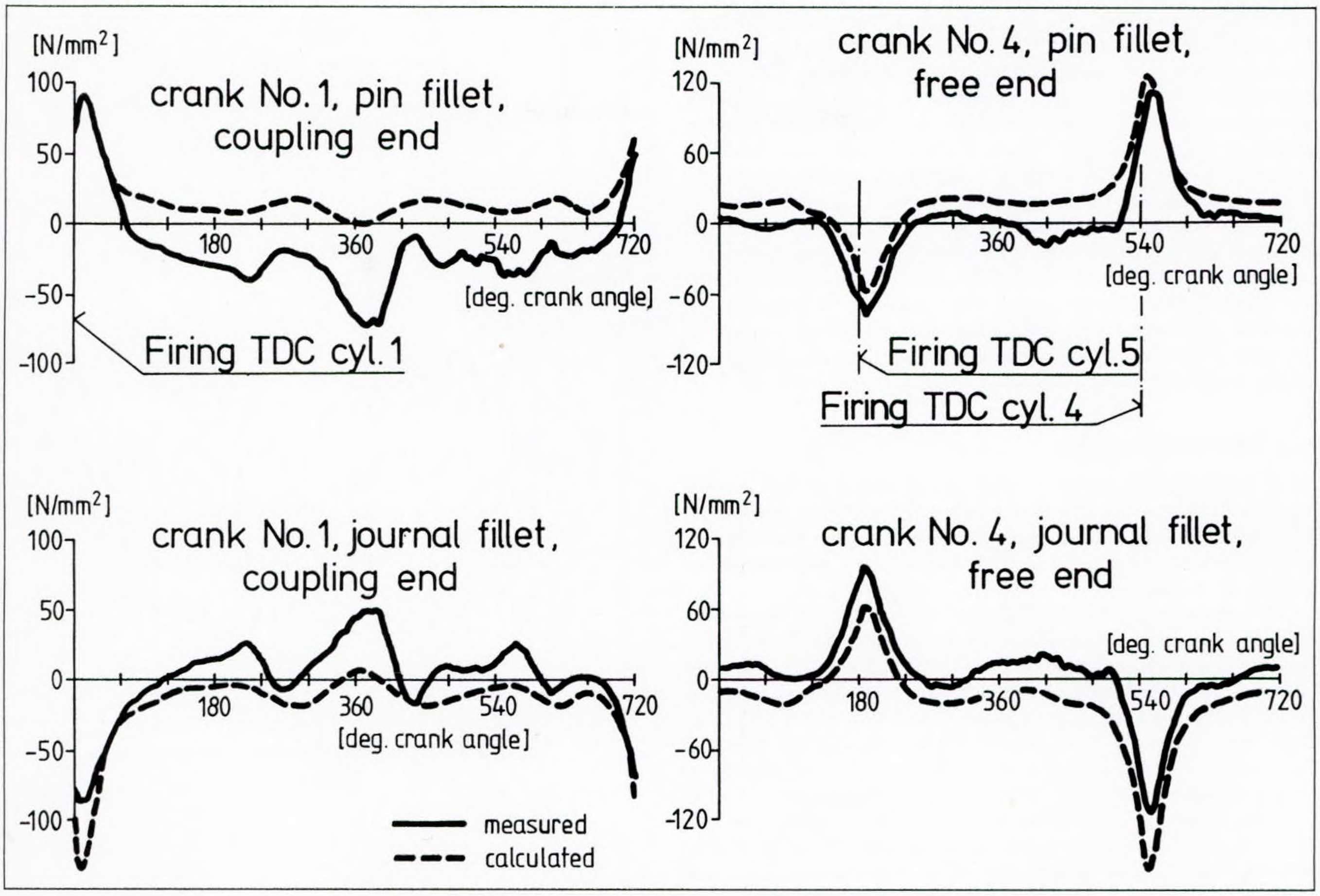

FIG. 7 Bending stresses in the four-stroke engine (measured on testbed)

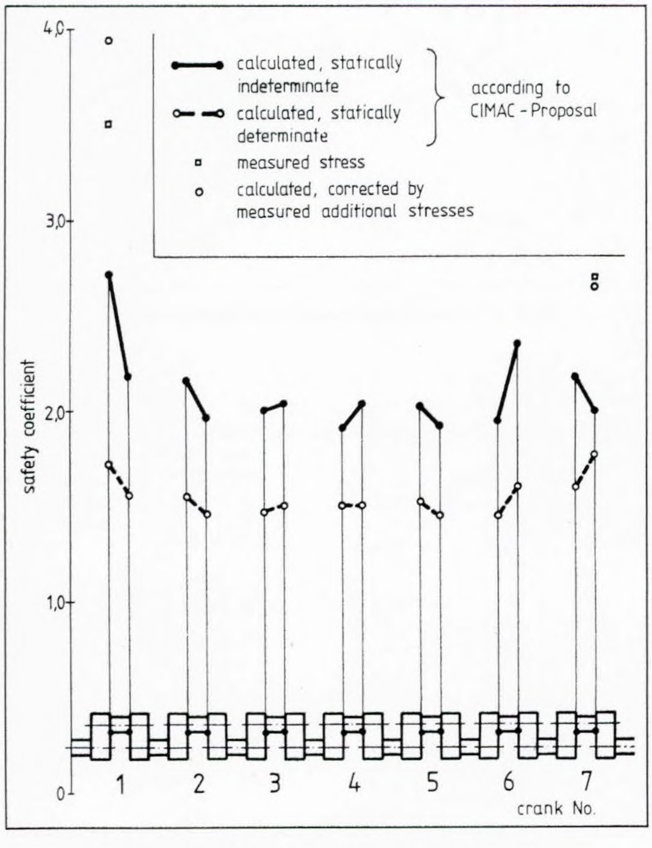

FIG. 8 Safety coefficients (two-stroke engine)

in this discrepancy. In the case of the output shaft, however, a good coincidence between calculated and measured torsional stresses is obtained.

5.2.2 Bending stresses

In Fig. 7, the measured and calculated stresses are plotted against time. The deducible amplitudes are also included in Table II. Whereas the curves and especially the amplitudes at the two fillets of crank no. 4 coincide quite well, a considerable deviation of the stresses in the crankpin and journal fillets of crank no. 1 can be seen at about 360 deg. A similar deviation had already been noticed in the measurements taken in the co-operative research project and was instrumental in the decision to repeat the stress measurements. The repeat measurements also included the journal displacement path of crank no. 1 and this record indicated the reason for the deviation. The observations made are outlined in Section 8.

6. ASSESSMENT OF THE DIFFERENCES BETWEEN CALCULATED AND MEASURED STRESSES WITH REFERENCE TO THE SAFETY COEFFICIENT

6.1 Two-stroke engine

Table I permits a critical assessment of the differences between the calculated and measured stresses and their resultant influence on the safety coefficient. It can be seen that the equivalent stress measured on crank no. 1 is a little higher than the calculated stress in the second line. However, if the safety coefficients are examined in accordance with the two methods in the CIMAC proposal, it can be seen that the safety coefficient of both cranks, even using the statically indeterminate variant in the CIMAC proposal, is markedly lower than the measured value; the reason being that the additional stresses contained in the CIMAC proposal are far higher than those actually determined during the measurement. The bending stress according to the continuous beam model, including the additional ±30 N/mm^2, is about 46% higher at the more highly loaded crank no. 7 than the stress measured and the equivalent stress used as a basis for approval is still about 35% higher than the one calculated from the measured readings.

In Fig. 8 the safety coefficients calculated by the two methods in the CIMAC proposal are plotted against engine length. An important finding from this graph is that the lowest safety coefficients that occur using the statically indeterminate method are likely to be either in the middle of the engine (on account of a high torsional stress) or at the free end (on account of the highest bending stress).

6.2 Four-stroke engine

Here again the differences between calculated and measured results must be assessed on the basis of the safety coefficient shown in Table II. The calculation method used is the CIMAC one described in Section 2, but without the improvements proposed in Section 8.

The main variation between the calculated and measured bending stresses is in crank no. 1 (discussed in Section 5.2.2), and results in the equivalent stress being about 35% too low and the calculated safety coefficient too high if no allowances for misalignment etc. are made when working out the equivalent stress (torsion was calculated using the forced-damped method and bending stress was calculated by the statically indeterminate method in order to obtain the equivalent stress).

Even if an allowance for misalignment is added to the bending stress, the equivalent stress in Line 1P3 of the table remains about 18% below the value determined from the measured results. However, in Line 1P4 the statically determinate calculation method furnishes an equivalent stress which is higher and a safety coefficient which is lower than the measured values.

In this context the compilation of the safety coefficients on the strength of the statically indeterminate and statically determinate calculated bending stresses in Fig. 9 is of interest. It can be seen that, according to the statically indeterminate bending stress calculation, the lowest safety coefficient is in the middle of the engine. This is where the statically indeterminate and determinate calculated bending stress, and thus the pertaining safety coefficients, are almost of the same size. This is due to the firing force effect of a cylinder and the mass effect coming from the adjacent cylinder, adding up in the dead centres at the 'inner' fillets of cranks nos. 4 and 5.

The resultant bending stress is almost as high as the one calculated according to the single-crank method with simply supported journals. This effect occurs in all crankshafts with an even number of cranks and is evidently well recorded by the calculation method, as may be noted from Lines 4P1 and 4P2 in Table II.

7. INVESTIGATIONS INTO DIFFERENT VALUES OF TORSIONAL STRESSES IN CRANKPIN FILLETS OF THE TWO-STROKE ENGINE

If the differences between the measured and calculated stresses in Fig. 3 are analysed, the largest differences always occur when large tangential forces act on the crank throw. This is particularly evident from the measurement taken on throw no. 7, where maximum stress occurs exactly at the moment when the maximum tangential forces act on the crankpin.

That the normal torsional vibration calculation cannot reveal effects originating from tangential forces becomes obvious from Fig. 10 when it is remembered to what extent the crankshaft calculation has been simplified.

The three-dimensional beam model of a crank shows that

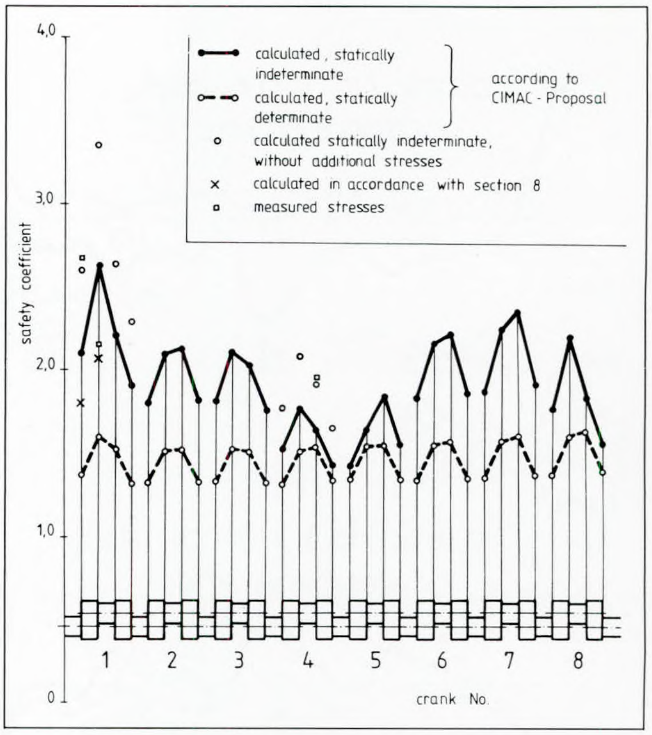

FIG. 9 Safety coefficients (four-stroke engine)

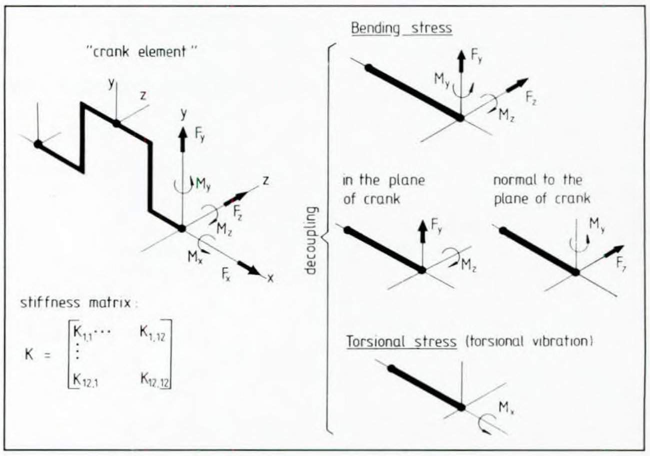

FIG. 10 Calculation models for crankshafts

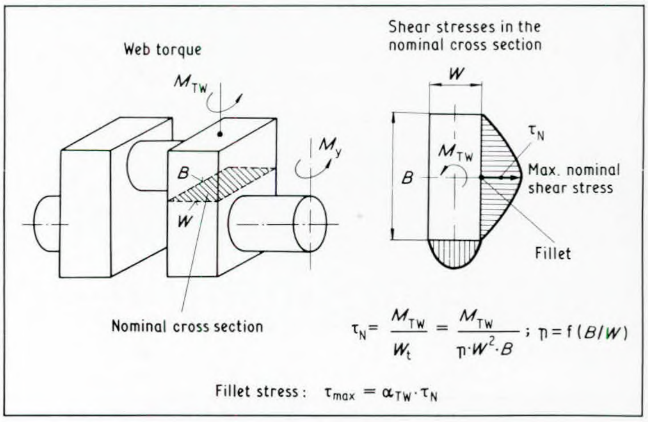

FIG. 11 Calculation of shear stress due to web torsion

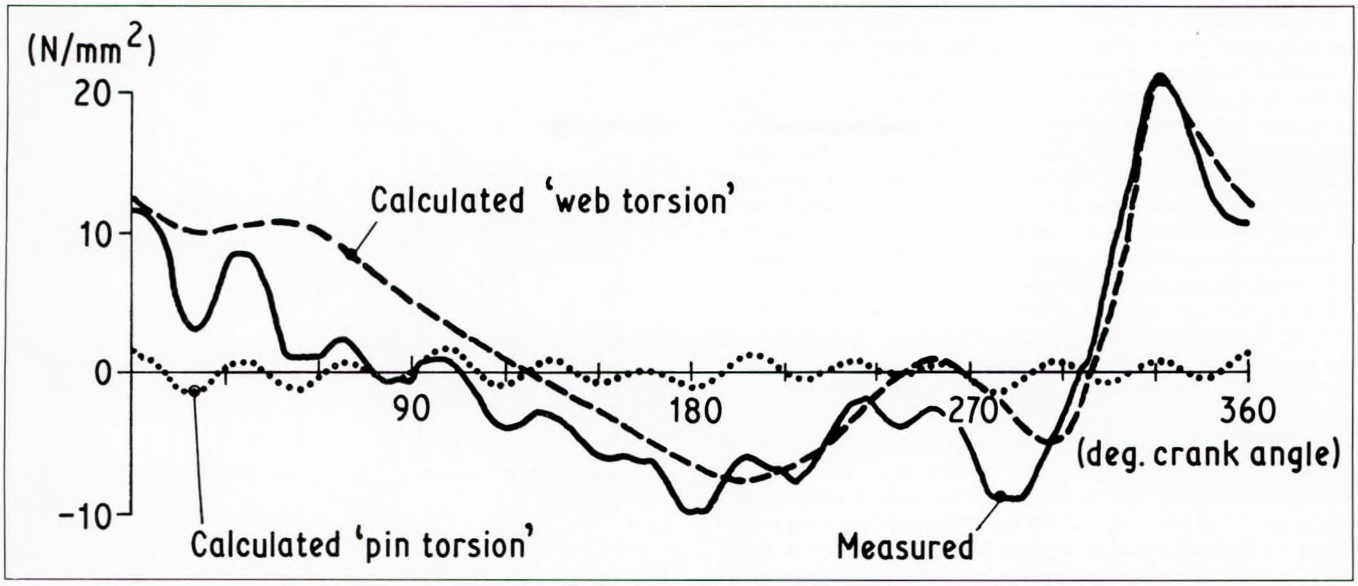

FIG. 12 Measured and calculated shear stresses at crank no. 7 in the two-stroke engine

three sectional forces and moments in the pins of a crankshaft have to be allowed for. The symbols used for the moments mark the plane which is defined by a couple of forces equivalent to the moment. Although such calculation models are sometimes used, such a procedure seems to be too complicated for an acceptance calculation.[8] For this reason, the normal procedure is to decouple in such a way that a one-dimensional continuous beam model is viewed; the bending and the torsional stresses being determined by separate calculations.

The torsional stress calculation only allows for the torques around the crankshaft axis and consequently force effects are not allowed for. A point of importance is that the normal bending stress calculation allows for all forces and moments in the crank plane and in a plane arranged at an angle of 90 deg to it; the calculation thus reveals the bending stress vector which actually exists in the centre of the web. However, to calculate the bending stress of each crank, only the projection of this moment in the crank plane is used.

In Fig. 11 moments occurring in the plane vertical to the crank plane are interpreted as bending if related to the pins but as torsional moments if related to the web axis.

If the same nominal cross-section is used as for the bending stress, a parabolic nominal stress distribution occurs at the edge of the cross-section when a torsional moment is applied; the maximum distribution being in the middle of the long side, i.e. where the fillet is. In the event of the torsional moment being known, this nominal stress can be calculated with the aid of the torsional resistance moment of the web cross-section. Adjustment by a stress concentration factor α_{TW} gives the fillet shear stress.

Table II: Comparison between measured and calculated results of the four-stroke engine (on testbed)

	Torsional stress, τ (N/mm²)	Bending stress, σ_B (N/mm²)	Additional stress, σ_{add} (N/mm²)	Equivalent stress, σ_{equ} (N/mm²)	Deviation from measured value, $\Delta\sigma_{equ}$ (%)	Safety coefficient,[a] S
CRANK 1—PIN						
1P1 Measured on testbed	24	85	—	95	—	2.15
1P2 Calculated statically indeterminate, without additional stresses	23	47	—	61	−36	3.34
1P3 Calculated according to CIMAC, statically indeterminate	23	47	20	77	−18	2.63
1P4 Calculated according to CIMAC, statically determinate	23	101	20	127	+34	1.60
CRANK 1—JOURNAL						
1J1 Measured on testbed	18	70	—	76	—	2.67
1J2 Calculated statically indeterminate, without additional stresses	19	71	—	78	+2	2.61
1J3 Calculated according to CIMAC, statically indeterminate	19	71	20	96	+27	2.11
1J4 Calculated according to CIMAC, statically determinate	19	125	20	149	+95	1.37
CRANK 4—PIN						
4P1 Measured on testbed	24	95	—	104	—	1.95
4P2 Calculated statically indeterminate, without additional stresses	31	92	—	106	+2	1.91
4P3 Calculated according to CIMAC, statically indeterminate	31	92	20	124	+19	1.64
4P4 Calculated according to CIMAC, statically determinate	31	101	20	132	+27	1.54
CRANK 4—JOURNAL						
4J1 Measured on testbed	[b]	107	—	—	—	—
4J2 Calculated statically indeterminate, without additional stresses	26	114	—	123	—	1.65
4J3 Calculated according to CIMAC, statically indeterminate	26	114	20	142	—	1.43
4J4 Calculated according to CIMAC, statically determinate	26	125	20	152	—	1.34

[a] S based on fatigue strength (σ_{fat}) = 203.3 N/mm².

[b] Not measured.

Figure 12 shows that the shear stress as a function of time can be well approximated by this calculation made with a continuous beam programme which is normally used for the statically indeterminate bending stress calculation. The only modification required was that the space-fixed component of the bending moment, calculated in the web centre, was not projected into the crank plane but into a plane vertically to it. Selection of the stress concentration factor in this case is difficult. Since no experimental value was to hand for this shaft, the maximum stress was matched to the value measured. For this purpose a stress concentration factor $\alpha_{TW} = 2.5$ was required but this stress concentration factor is not universally applicable. If the type of crankshaft is different, it will change considerably. In the case of a small four-stroke engine, for example, a markedly lower stress concentration factor was found experimentally and a correspondingly lower shearing stress was calculated.

Depending on the sign of the shear stress from the web torsion, the torsional stress resulting from the torsional vibration calculation can be increased or decreased. It will be noted from Fig. 3 that the influence of the web torsion at throw no. 1 leads to a reduction within a range of about 30 deg crank angle. Also, in the case of the four-stroke engine differences between measured and calculated readings can be seen, which are due to web torsion, e.g. in Fig. 6 at throw no. 4 and about 580 deg crank angle.

Conclusions on the CIMAC calculation method are dealt with in the final section. It may be taken for granted, however, that the shear stress level due to web torsion is markedly lower than the torsional-vibration-induced maximum torsional stress level that occurs in the engine, especially if resonance influences occur at the torsional vibration.

8. INVESTIGATIONS INTO DIFFERENT VALUES OF BENDING STRESSES IN THE FOUR-STROKE ENGINE

Figure 13 shows the loading and seating arrangements of a continuous beam model.[9,10] It is assumed that the resultant loads act on a point in the middle of each pin and web; the reciprocating masses and the gas forces act in the middle of the crankpin; the rotating mass of the crank and of the connecting rod is distributed between two web centres; existing counterweights are mounted in the same place and the beam is seated on linear springs, i.e. movements of journals within bearing clearances are neglected.

If the differences between calculated and measured readings in Fig. 7 are considered for crank no. 4, it becomes apparent that the difference consists mainly of a time-constant stress share, which is of a compressive nature in the crankpin fillet and of a tensile nature in the journal fillet. This means that the difference is caused by a constant bending moment. Such moments are caused exclusively by the effect of the rotating masses. Presumably, the distribution of the rotating masses in the calculation model is not correct in this case. Better coincidence could possibly be achieved if a larger share of the rotating mass was assumed to be in the crankpin centre.

At this crank in the middle of the engine, the coincidence of the dynamic stress values is excellent so that other simplifications of the calculation model are evidently unimportant in this case, which proves that the stress concentration factors calculated by the CIMAC proposal are correct. The comparison between calculation and measurement shown in Fig. 7 for throw no. 1 is less satisfactory. Apart from the aforementioned difference by a time-constant share, which is particularly visible in the crankpin fillet, it is noticeable that large differences occur mainly in the vicinity of the TDCs of the throws.

To explain this phenomenon it is necessary to explain first the conditions at the coupling end of this engine. Figure 14 shows that the crankshaft has been extended by an additional seated flange shaft. The power take-off for the camshaft is

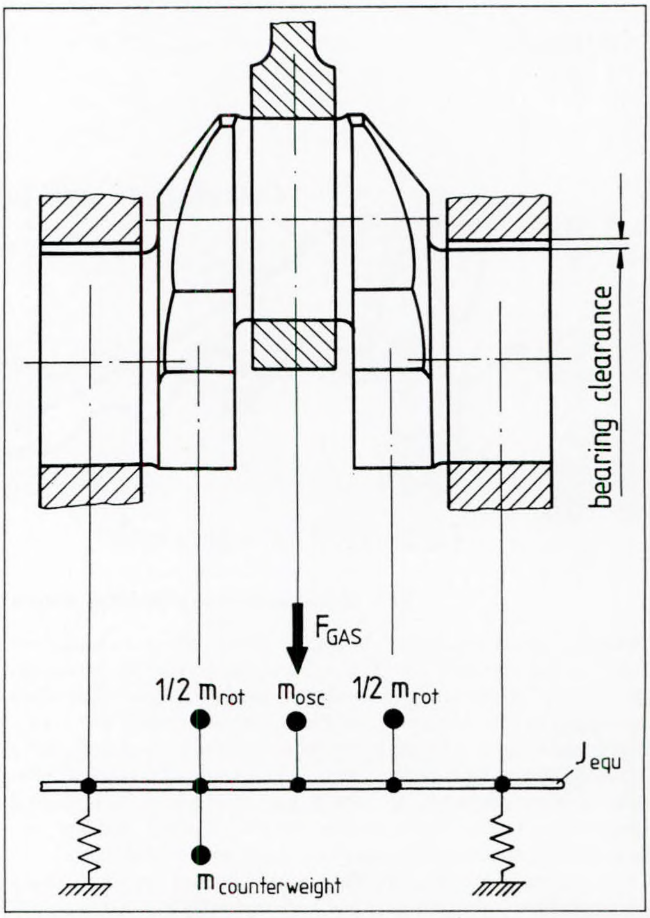

FIG. 13 Model for bending stress calculations (continuous beam)

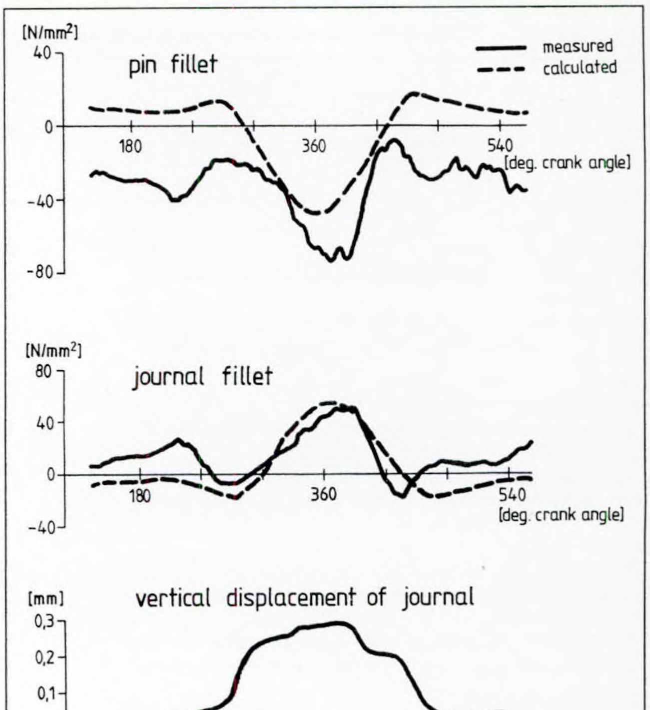

FIG. 15 Bending stress calculation for crank no. 1 of the four-stroke engine, allowing for the actual seating conditions

between the two outboard bearings. The bending stress calculation model allowed for the overhung mass of the flywheel and coupling with an additional outboard bearing O. As will be noted from the particulars stated below, introduction of this bearing, which produces a substantial clamping effect at the crankshaft end, is not justified.

It was found by measuring the movement of the journal next to crank bearing no. 1 that, owing to the mass force effect, the shaft is lifted within the range of about 360 deg crank angle and bears against the top shells of the crank bearings. The recalculation of this load case reveals that, at the same time, the shaft lifts off the bottom shell of the outboard bearing. It is apparent from the gas force load case, also shown in Fig. 14, that lift-off takes place at the outboard bearing as well, which leads to the conclusion that the reactive forces are negligible, i.e. that this bearing does not produce a clamping effect.

If a calculation around 360 deg crank angle is carried out on the basis of actual seating conditions (upward displacement of the pins at bearings nos. I and II by the bearing clearance and absence of the outboard bearing), the dynamic stresses as a function of crank angle are recorded quite well (Fig. 15) and there is quite a good correlation between calculated and measured dynamic bending stresses.

In crank no. 1, the measured stress in the crankpin fillet is larger than in the journal fillet. There is no explanation for this phenomenon as the ratio of crankpin fillet to journal fillet stress should be the same in every crank, if it is assumed that these stresses are produced by bending moments in the middle of the webs caused by transverse forces. A possible explanation is that the force acting radially on the crankpin at the coupling end deforms the crank axially (crank web deflection) and, due to the axial clamping effect at the adjacent locating bearing, the entire crankshaft mass moves axially towards the free end as in Fig. 14 (bottom), for instance due to the gas force.

The mass force developed during this pushing process gives rise to bending moments in the crank, which are superimposed on the transverse-force-induced moments so that the stress amplitude level reversal between crank pin and journal, described above, appears to be possible. Since this phenomenon was not investigated any further, the effect described must be regarded only as a theory.

It will be noted that the calculation method in the CIMAC proposal gives rise in this case to considerable differences between calculation and measurement. This is evidently due to the additional outboard bearing, which does not bear under certain load conditions. If the outboard bearing is left out of the calculation, the calculated stress value in the crankpin fillet is ± 88 N/mm^2, which is close to the ± 85 N/mm^2 stress measured. However, in the journal fillet, the calculated stress is ± 108 N/mm^2 and the measured one ± 71 N/mm^2; consequently the calculation is on the safe side.

In this context, owing to the integrated thrust bearing, the two-stroke engine also features a radial outboard bearing which had to be taken into consideration to ensure that the measured and calculated results tallied. In the two-stroke engine, no forces act upwards.

With regard to further refinements of the calculation methods for the bending stress in crankshafts, there does not seem to be any urgent need to introduce a higher-quality stiffness model[11,12] for the crankshaft itself on account of the aforementioned difference between measurement and calculation. More accurate recording of the boundary conditions obtaining at the continuous beam, making allowance for the bearing clearances, would be more important.

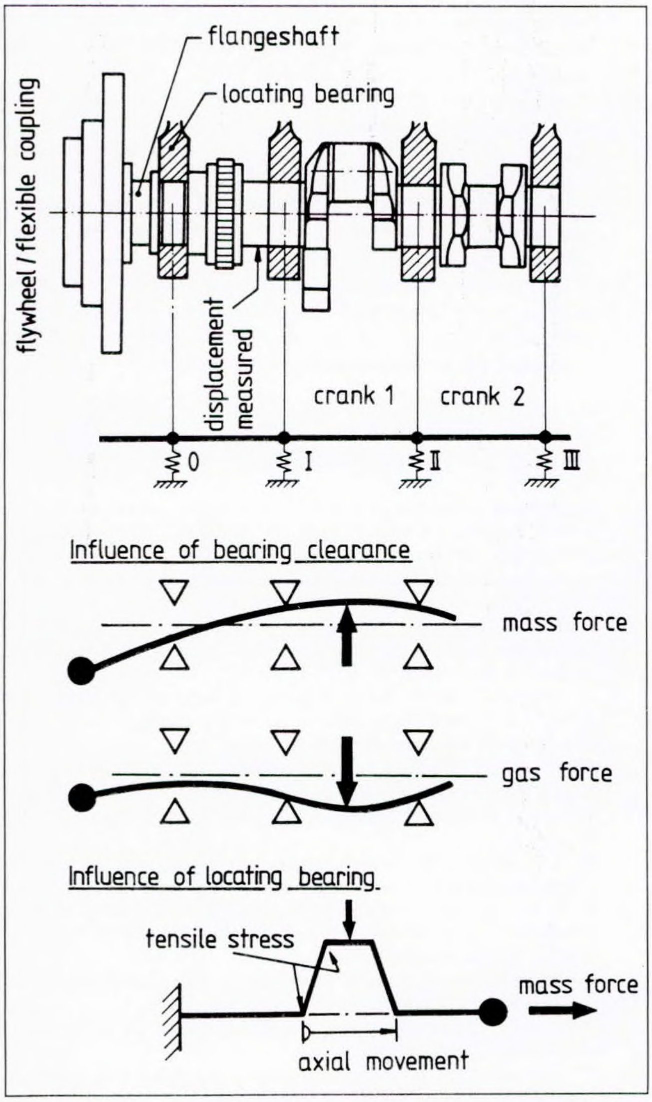

FIG. 14 Factors affecting bending stresses at coupling end of the four-stroke engine

9. SUMMARY OF RESULTS

As far as is known, this is the first time that identical and comprehensive stress measurements have been carried out on several cranks of different crankshaft designs. The more important results are as follows.

(i) Forced-damped vibration method using simple substitute systems[5]

This method is used world-wide for calculating torsional stresses and the results coincide well with the actual measurements taken on smooth shaft sections. The standard reduction method for slightly damped systems normally results in an accurate calculation of the resonance points, the amplitudes of the resonances being determined by the quality of the empirical damping values used.

(ii) Additional torsional stress

The opportunity to measure torsional stresses in a fillet through which virtually no torque passes revealed an additional torsional stress which has hitherto not been taken into consideration. This torsional stress is due to the bending moments originating in the crankpin or journal being transformed in the web into torsional moments acting around the web axis.

Whereas the determination of these moments is not difficult to quantify, determination of the related stresses is impeded as no stress concentration factors have been determined yet for

Dr Günter Donath obtained his doctorate in 1959 on vibrations. In 1963 he joined Rheinstahl-Wanheim GmbH, and in 1964 Schloemann AG: and worked on the design of rolling mill machinery and hydraulic forging presses. He joined the diesel engine division of Maschinenfabrik Augsburg-Nürnberg AG in 1969 as head of the mechanics and metrology department, in charge of sound and vibration mechanics, strength testing and calculation, and measuring techniques involved with diesel engine development. He is a member of the CIMAC committee investigating crank shaft design.

Mr Heinz Seidemann has worked at MAN Augsburg for 20 years on large-bore diesel engine development, specialising in engine component strength properties. He first worked on experimental stress analysis, using photoelastic model and strain gauge measuring techniques and, when high-capacity computers were introduced, he used them for numerical and, in particular, finite-element calculations. He is currently head of the strength calculation/testing development section within the engine development department.

this type of stress. The ± 14 N/mm^2 difference (Table I) between the calculated and measured readings for crank no. 7 of the two-stroke engine gives some idea of the magnitude of these stresses. This difference must be compared with the maximum ± 45 N/mm^2 torsional stress generated in this engine by vibratory torques.

(iii) Discrepancies between calculated and measured torsional stresses

Although the effect in (ii) above was first discovered in a vibratory-torque-free web, it can also be found in all stresses measured on the crankshafts of two-stroke and four-stroke engines. This effect is the main reason for the lack of coincidence between calculated and measured stresses. In a four-stroke engine, some idea of the additional torsional stress can be gained from the crankpin fillet of crank no. 4 (Fig. 6); this additional torsional stress amounts to about ± 13 N/mm^2 and is thus of a similar size as in the two-stroke engine.

When the maximum additional web-torsion-induced stress in a crankshaft has been estimated, the question of the effect on the equivalent stress is determined mainly by the phase angles. In the worst case the value of the torsional stress in the web can add itself algebraically to the conventional torsional vibration stress.

(iv) Allowance to be made for the additional torsional stress

To allow for this new torsional stress in a simple way, it is suggested that ± 15 N/mm^2 should be added to the torsional stresses calculated by the method proposed by CIMAC; the proposed method being identical to those used by almost all engine builders and classification societies. It is up to the engine builders to decide whether or not this stress should be verified more accurately by means of a test.

(v) Additional bending stresses

The measurements have shown that, in four-stroke engines, bearing misalignments do not cause additional bending stresses and this has been confirmed by other engine builders. This is due to compact construction, high stiffness of frames and quality of machining in the bearings.

The additional stresses do occur in large two-stroke engines but they are markedly lower than the ± 30 N/mm^2 proposed by CIMAC. It is recommended that the allowance for additional stress should be omitted in four-stroke engines and be amended to ± 15 N/mm^2 for two-stroke engines.

(vi) Bending stresses from gas and mass forces

The comparison of the statically indeterminately calculated bending stresses due to gas and mass forces has shown that satisfactory coincidence with the measured results, comparable with the accuracy of the torsional stress calculation, can be achieved as long as no special boundary conditions obtain at the coupling end of the crankshaft. The measurements taken in this range of the four-stroke engine revealed that the bearing clearance had a greater influence on the stresses. This is to be expected, particularly if there is a further outboard bearing next to the first crank bearing. It was proved that this bearing does not come to bear in the TDCs of the first crank so that the clamping stiffness of the first crank is considerably lower than the one allowed for in the calculation.

Also, it is not necessary to introduce the hydrodynamic theory for plain bearings to the calculation model, as reasonable boundary conditions for the calculation can be found by taking the reactive bearing forces and displacements into consideration. A simple but safe calculation model entails the omission of such a bearing. The safety coefficients calculated in this way and those derived from the values measured are given in Fig. 9.

10. CONCLUSIONS

(i) The statically indeterminate bending stress calculation method reflects the actual stress conditions in a crankshaft considerably better than the statically determinate method.

(ii) Comparison with the measured results shows that, when the statically indeterminate method is used, the decisive safety coefficient still contains a certain hidden safety factor.

The statically determinate method, however, involves variation of the hidden safety factor over a wider range and is therefore less reliable.

(iii) The statically indeterminate method should be adopted by the classification societies in their unified rules.

(iv) The CIMAC proposals would be fully satisfactory if they included the recommendations in this paper.

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ACKNOWLEDGEMENTS

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They also wish to thank Mr Archer of Lloyd's Register and Mr Volcy of Bureau Veritas and their colleagues for their excellent cooperation.

Discussion

B. LAW (Perkins Engines Ltd): I would first like to congratulate Dr Donath and Mr Seidemann on their interesting paper.

My experience of crankshaft-loading and safety-factor prediction relates to the high-speed diesel engines manufactured by Perkins Engines at Peterborough and Shrewsbury (formerly the diesel-engine division of Rolls-Royce Motors). Currently engines in the range from 40 to 820 hp have approval certificates from a number of classification societies.

The calculation procedures for crankshaft and main-bearing loading used at Perkins has been described in a CIMAC paper.[1] The crankshaft and engine stiffnesses are first computed using a finite-element computer program. Load against deflection equations based on these stiffnesses are then solved (in a separate computer program) throughout the engine cycle, with a main-bearing oil film representation based on the 'mobility' method. The procedure has been validated by comparison of predicted and measured crankweb strain in a high-speed diesel engine. The major conclusions in the comparison of predicted and measured strain were:

1. The statically determinate (or simple throw) method was inconsistent and generally grossly overestimated the crankweb loading.
2. The available forced damped techniques for torsional vibration were adequate in predicting the stress components induced by torsion.
3. Most of the discrepancy between predicted and measured strain at the crankweb adjacent to the flywheel was due to mass–acceleration and gyroscopic effects not being represented in the calculation procedure.

These findings are generally in agreement with those presented in the paper.

The procedure for crankshaft assessment used in Perkins follows the general principles reported in the paper. Accurate crankshaft-loading prediction (bending and torsion) is fundamental in the determination of the fatigue safety factor. On high-speed diesel engines modelling of the journal-bearing characteristics and the crankcase structure is necessary in order to compute the crank loading accurately. The importance of crankcase stiffness was highlighted in a recent experimental and theoretical study. It was found that a 30% reduction in the crankweb bending moment resulted from selective stiffening of the crankcase structure. The proposed CIMAC rules would lead to less accurate load prediction because the oil film and crankcase stiffnesses are grossly simplified. What provision is there in the proposed CIMAC rules for selection of appropriate spring constants to represent the crankcase supporting structure?

Having determined the crankshaft loading, the next stage in the assessment procedure is calculation of critical stresses. The stress concentration factor approach has proved to be effective for this calculation. The safety-factor calculation generally includes an allowance for mean stresses.

The allowance has a significant effect on high-speed diesel-engine crankshafts, because there are significant mean crankweb bending moments (usually tensile mean stress is induced in main-journal fillets) according to the indeterminate calculation method. Crankshaft failure statistics derived from engine test-bed running correlate very well with computed fatigue safety factors which include the mean stress factors, and correlate very poorly with safety factors which do not include the mean stress effect. For a typical proven engine design, the safety factors, based on loadings computed from the statically determinate (single throw) and statically indeterminate methods, are 1.2 and 1.7, respectively. A single safety-factor acceptance criterion should not be applied to results from both methods. Application of the criterion (1.15) suggested in the CIMAC rules indicates a margin of safety for extra loading when indeterminate calculations are used. In my experience of high-speed diesel-engine crankshafts, I judge this margin to be excessive and would not believe the crankshaft design to be viable if the safety factor was 1.15 based on the statically indeterminate method.

In conclusion, although I agree with many of the principles advocated in the paper, I must emphasise the necessity for inclusion in the rules of accurate cylinder-block stiffness models, mean stress in the safety-factor calculation and, in particular, acceptance levels for safety factors which are appropriate to the calculation method. The rules would not identify vulnerable high-speed diesel-engine crankshafts and their effectiveness in assessing other classes of diesel-engine crankshafts is questionable.

G. C. VOLCY (Bureau Veritas): The authors of this very valuable paper are to be complimented for the tremendous work performed on such a complex problem as the measurement and calculation of stresses in crankshafts. I am also impressed by their courage in undertaking such an arduous task. The crankshaft is the most mysterious and complicated piece of machinery I have encountered in my professional life, and I have been working on it over the last 25 years.[2–6]

Being fascinated by the numerous problems related to the crankshaft, I was lucky when, many years ago, I encountered A. Schiff, from MAN, with whom I have often spoken of the possibility of obtaining greater knowledge of crankshaft behaviour, particularly in view of the serious damage (including the breaking of crankshafts) which occurred all over the world at the beginning of the 1960s. The reasons for these difficulties were incompatibilities between the flexibility of the structure of newly built ships (always of greater tonnage) and

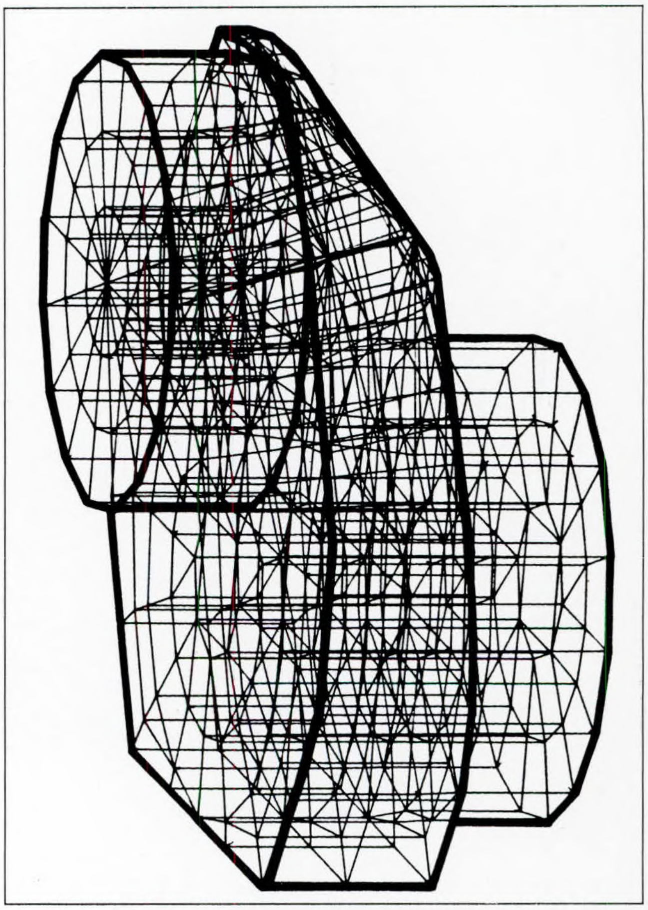

FIG. D1: 3D FEM model of a half-crank

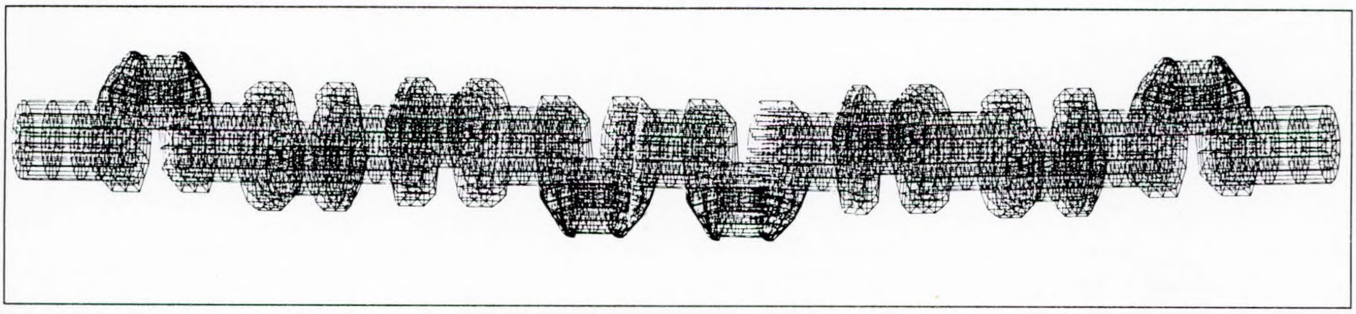

FIG. D2: 3D FEM model of a crankshaft

the increased stiffness of the line shaftings, including the crankshaft. This was also the reason why MAN and B & W decided to build the box-type structure of engine girders, now adopted world wide.[7]

However, the same could be said of the decision of CIMAC concerning the need to reanalyse the methods used by different classification societies to calculate the scantlings of the crankshaft which led to its stiffness. This handicapped the diesel-engine manufacturers. When the CIMAC Crankshaft sub-group, under the direction of Dr Donath, finished its work, it was necessary to compare the proposal for the future of shaft design with the reality, by finding the correlation between calculations and measurements. The aim was also to present convincing arguments to IACS for adoption as a rule of the proposal put forward by CIMAC.

In view of the above it was decided to create a joint venture between Companies which have, in the past, shown the most interest in the behaviour of crankshafts, and so Lloyd's Register of Shipping was invited to join MAN and Bureau Veritas in this research team.

The task of Bureau Veritas in this cooperative research project has been threefold:

1. To provide a theoretical check of the method used to calculate crankshaft scantling as proposed by the CIMAC working party and the supplementary method proposed by Lloyd's Register, which considers the crankshaft to be a spatial hyperstatic beam. The modern FEM methods were applied to the treatment of the crankshaft under static and dynamic conditions.
2. To use the theoretical and experimental results to provide supplementary data and information (for theoretical calculations and their correlation with experiments) on the effects of internal (trajectories of journals) and external (thermal effects, loading and sea conditions, influence of line shafting and propulsor) factors on the behaviour of crankshafts.
3. To establish, also by using FEM, a method which allows the spatial position of crankshaft journals in their bearings (ie the real alignment conditions of the crankshaft) to be determined from readings of crankshaft deflexions.

The above applied to both two- and four-stroke engine crankshafts.

This contribution will deal only with the essential results of the research undertaken by Bureau Veritas. Those interested in greater detail are referred to ICMES '84 held in Trieste in September 1984. The most important results are related to:

- Measurements performed on the crankshaft of a four-stroke engine and corresponding FEM calculations related to deformation and stress;
- Previous research related to the calculation of stresses with an equivalent hyperstatic beam representing the crankshaft.

Figure D1 shows a 3D FEM model of a half-crank, with the 3D FEM model of the corresponding crankshaft in Fig. D2.

By applying well known forces to the crankshaft, the equivalent von Mises/Hencky/Huber stresses can be calculated. An example of such calculations is presented in Fig. D3.

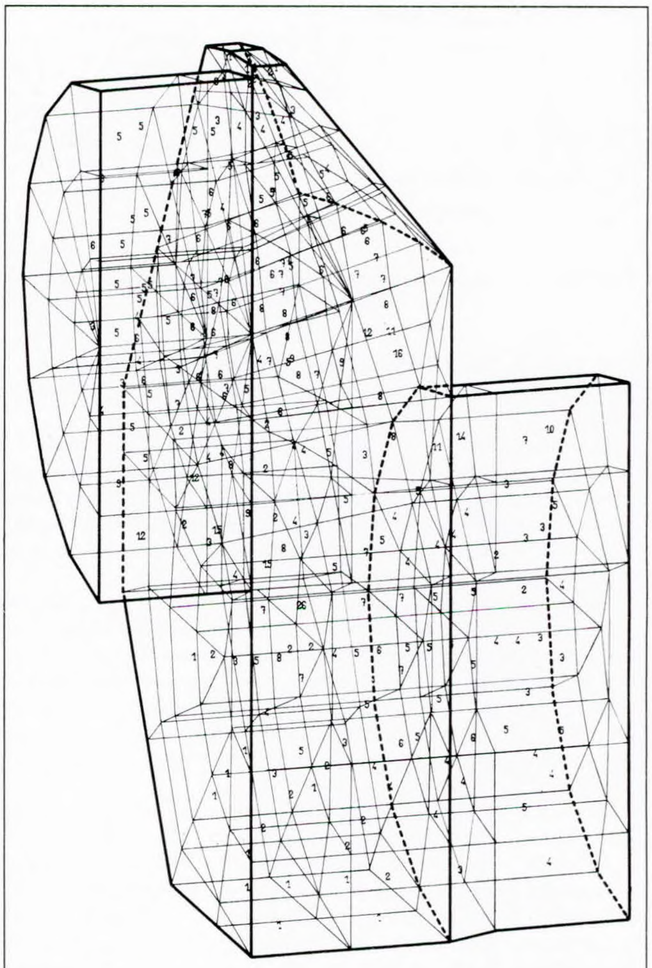

FIG. D3: Von Mises stresses in crank no. 4

Unfortunately, however, the results of these FEM calculations were disappointing. In fact, the long, laborious and expensive calculations by the most advanced theoretical means (FEM calculations on speedy computers) have proved that pragmatic solutions of technical problems are better than more sophisticated methods. The limitations of FEM are highlighted when practical and usable solutions are required.

The sub-structures method is particularly suitable for this kind of structure because of the geometrical repetition. Also the general stiffness and deformation of the crankshaft can be well represented. However, the exact supporting conditions are not easy to evaluate.

Another important conclusion which can be drawn from these calculations concerns the great difficulties of such an analysis. These difficulties appear at each level of the calculations. Another difficulty lies in the interpretation of the results.

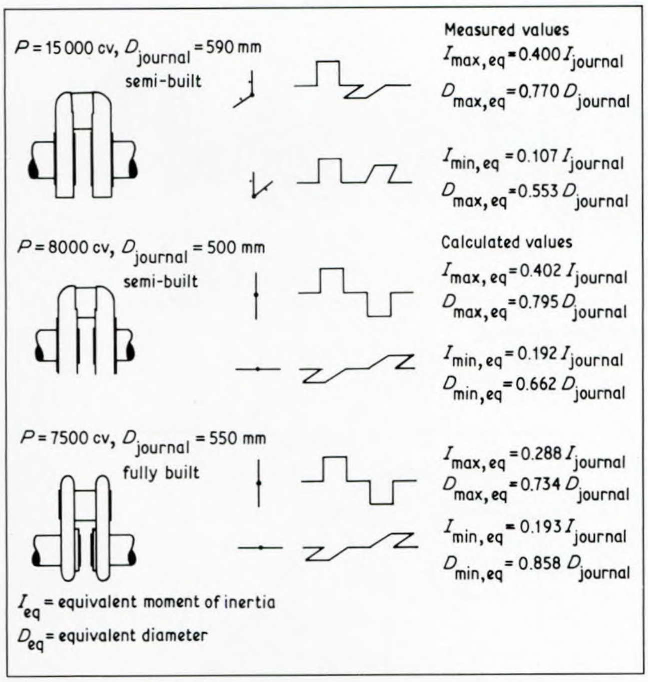

FIG. D4: Values of moments of inertia on bending and equivalent diameters of the webs

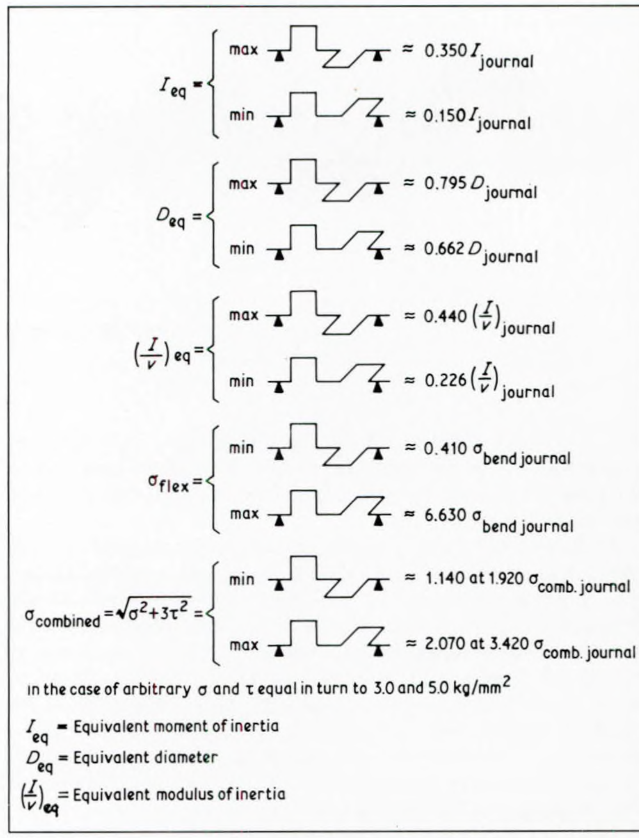

FIG. D5: Strains in the webs as a result of loss of contact between a pin and its bearing

The main difficulty, however, is in the modelling of such a complex spatial structure as a crankshaft and especially of the fillets.

It is important to note that the refinement of the mesh for stress calculation in the fillets is so far practically unsolved. This is because the fillets are spatial structures and refinement of the corresponding mesh presents serious problems for the software. We know of no such programme and no information could be found on the subject.

Hence there is a need for a less theoretical, simpler approach, which could be supplemented by input data obtained from experiments and concentration factors. Clearly the best calculation method uses either an equivalent hyperstatic beam or a spatial beam in the determination of the scantling of the crankshaft in view of the different loadings required to determine the bending moments and shearing forces.

The work of Bureau Veritas shows the need to find simpler and easier methods for determining the scantlings of crankshafts. In view of the above, the CIMAC Crankshaft sub-group must be congratulated for deciding to replace the crankshaft by a beam. I am glad that Dr Donath and Mr Seidemann came to the conclusion that the hyperstatic undetermined beam should be adopted by the classification societies as the unified calculation method.

I would like also to draw attention to some results of previous work on this subject which was published some years ago.[3-5,8] When studying seriously damaged crankshafts and their main bearings I found that a very important role is played by structure deformation,[2] as well as by the alignment of the crankshaft,[3-6,8,9] leading me to recommend a curved alignment of the crankshaft, predeformed by sagging during outfitting.

I have also had the chance to investigate the intrinsic properties of a crankshaft when looking for its equivalent moment of inertia in bending. The results of this investigation[3-5] are presented in Fig. D4, from which can be seen $I_{eq} \leq I$ of a beam having the diameter of the journal (or pin).

Having found realistic values of I_{eq}, the stresses occurring in the crankshaft when a journal is losing contact with its corresponding bearing could be deduced. Figure D5 shows the results, which are from 3.4 to 6.6 times more than the stresses of the beam of journal (or pin) diameter commonly used to fix the rules for determining the scantlings of the crankshaft.

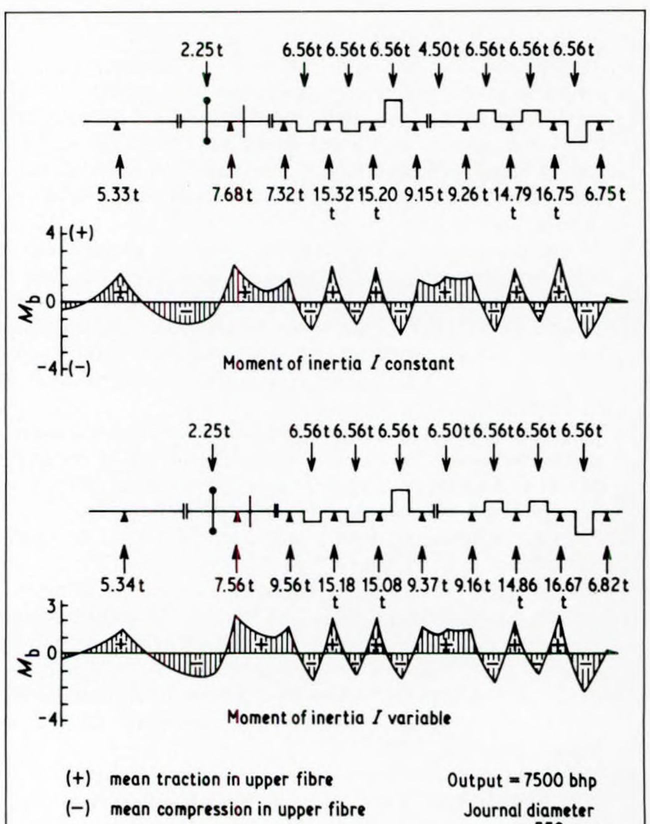

FIG. D6: Influence of the value of the moment of inertia of a crankshaft on the distribution of bearing reactions and bending moments

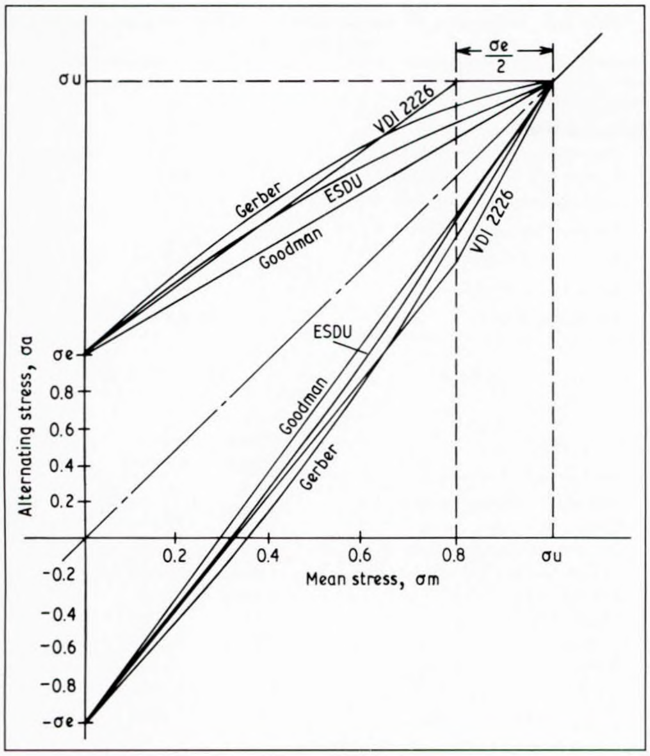

FIG. D7: R–M diagram, some comparisons

These values of the stresses explain the reason for the previous damage to bearings and stiff crankshafts when placed on a flexible engine girder (with transverse webs) mounted on foundations that are not stiff enough, the stiff crankshaft being subjected to the deformation of the ship's structure.[2,8,9]

Moreover, I_{eq} is not constant over the whole length of the crankshaft but varies as a function of the angle between adjacent cranks (see also Fig. D4). Fortunately the published[5] calculations and Fig. D6 show that this variability of I does not influence the distribution of bending moments and bearing reaction forces.

In view of the above, I would like to ask the authors how the equivalent moment of inertia I_{eq} of their hyperstatic equivalent beam crankshaft is determined. Also of interest is the question of which values of I_{eq} they have found previously in comparison with the moment of inertia I of a pin (or journal), eg for the dynamic calculations presented in their paper.

R. B. SIGGERS (Lloyd's Register of Shipping): Dr Donath and Mr Seidemann are to be congratulated on a very well presented paper, which will be of interest to engineers and stress analysts all over the world. The paper sets out honestly and clearly both the successes and the problems encountered.

The cooperative work between CIMAC and IACS should in the very near future result, after many years of work, in the publication of unified requirements for dimensioning crankshafts. This must seem an extremely attractive proposition to the marine engine manufacturers all over the world. The logic is irrefutable, as shown in Fig. 1 of the paper; the crankshaft is not aware of being classed by LR, BV or any other society so it is a good idea to have a unified approach. However, whilst the paper gives cause for some optimism, in my opinion we are still some way away from this ideal.

'In the beginning' we used mean stress, which was later superseded by mean and alternating stress (causing many divisions). Theories of Goodman and Soderberg were adopted almost as a fashion. A multitude of other theories are available: Gough's ellipse, ellipse arc, Gough–Pollard, VD1 2226, ESDU, modified Goodman and of course Gerber. Four of these are shown plotted on an R–M diagram in Fig. D7, from which an appreciable band is already forming.

Having selected a theory, arguments occur as to whether stress concentration factors should be incorporated on mean and alternating parts. No text book seems the same. Now we are presented with a neat solution: ignore mean stress.

The question of fatigue endurance limits and factors of safety can cause differences in the acceptance of crankshaft ratings. Heat treatments such as induction hardening and processes such as cold rolling of fillet radii can give enhanced fatigue properties, which are currently reflected in our Rules by the Z value, but these will also vary from society to society, as will the factors of safety.

Table DI shows the influence of mean stress on a case currently being dealt with. Please ignore the actual values as the engine's rating is unacceptable, but look at the changes in factors of safety. This engine is a Vee engine running at 1000 rev/min (where the inertia forces are comparatively low so mean stress is high). The right-hand column at the bottom shows the percent change in factors of safety when mean stress is introduced.

Table DII shows a smilar state of affairs and concerns an in-line engine running at 1800 rev/min. The inertia forces appear to have reduced the mean stresses but the differences in safety factor are still of significance.

I have taken some time to get to the point, which is as follows. Although the current work is of enormous value regarding unified values for alternating stresses and stress concentration factors, I would like to ask the authors if, in their opinion, the mean stresses really should be neglected.

A. H. SYED (Lloyd's Register of Shipping): First I would like to give some background to the Co-operative Research Programme outlined by the Authors.

Arising out of the need to have a calculation method which takes into account all the important factors influencing the stresses in crankshafts, a fairly sophisticated method was developed by Lloyd's Register of Shipping. The guiding criteria set for this development were:

(a) It should take account of all the important factors in a

Table DI: Influence of mean stress. Vee engine running at 1000 rev/min

Parameter	*Input data*	
Ultimate tensile strength	690.0000	690.0000
Ultimate shear strength	483.0000	483.0000
Endurance limit in bending	274.7000	274.7000
Endurance limit in torsion	158.6000	158.6000
Mean direct stress	0.0000	206.5000
Rev direct stress	306.8000	306.8000
Mean shear stress	0.0000	43.1000
Rev shear stress	97.4000	97.4000

Theory	*Factors of safety*		
	Without mean stresses	*With mean stresses*	*Change (%)*
von Mises–Goodman (Lloyds)	0.7846	0.6278	20
ESDU (M = 1.5)	0.7846	0.6877	12
Goodman–Gough ellipse quadrant	0.7846	0.5778	26
Goodman–Gough ellipse arc	0.7932	0.6505	18
Kerr–Wilson combination	0.7846	0.7620	3
ESDU–Gough quadrant	0.7846	0.6770	14
ESDU–Gough arc	0.7932	0.7157	10
Gerber–Gough quadrant	0.7846	0.7278	7
Gerber–Gough arc	0.7932	0.7508	5
DNV–Gough quadrant	0.7846	0.6751	14

$$\text{CIMAC factor of safety} = \frac{\text{endurance limit}}{\text{von Mises stress}} = \frac{274.7}{\sqrt{(306.8^2 + 3 \times 97.4^2)}} = 0.7846$$

transparent way so that the source can be traced and any modification or addition can be easily made as more information becomes available;

(b) It should be reasonably economical to use both in terms of human and computer resources.

For this purpose, the finite-element method was considered but rejected on the grounds of being too time consuming for a day-to-day calculation method. A numerical approach based on classical bending theory was therefore adopted which models the crankshaft as closely as possible to its actual shape rather than an equivalent straight beam. These are fully described in Refs 10, 11 and 12.

It was decided at the outset that a full-scale measurement would be attempted to verify the calculation methods. With this objective in view, Bureau Veritas and the authors' company were approached. It was found that in parallel with our development, MAN were actively involved in developing the CIMAC method, and out of this common interest the joint venture for measurements described in the paper was undertaken.

Lloyd's Register has a great deal of expertise in strain measurements, which is largely used for trouble-shooting investigations, but these measurements were done simply as an aid to design. I believe that this is the first time that such large-scale measurements have been successfully carried out on two different engines under actual working conditions on test-bed and at sea.

Turning to the paper, the authors' remarks regarding the shear stress in the crankpin being caused by torsion in crankweb, as showing in Fig. 11 of the paper, is interesting. However, the crankpin, even at the free end, is subjected to a torque due to the component of bearing reaction in a plane at right angles to the crankplane. The bearing reaction in turn is influenced by the movement of the journal in the bearing clearance under running conditions and the flexibility of the bearing support structure. In our method of calculation, initially the effect of bearing clearance was not taken into account but it has since been further developed to include this and predicts the journal excursions within the bearing through the engine cycle. Figure D8 shows the effect of bearing clearance on the bedding stress and Fig. D9 on the shear stress. Note that when bearing clearance is taken into account, the correlation of calculated and measured stresses is acceptable.

The authors have demonstrated the effect of journal movement in the bearing clearance, as shown in Figs 14 and 15 of the paper. As this effect may have a critical influence on the maximum stress range in the system for some designs of engine, I would like to ask the authors what procedure they adopt to predict this at the design stage as the CIMAC method has no provision for it.

A grey area at present is the flexibility of crankshaft seating and in this respect a very wide range of A values, with a recommendation for a mean, are proposed by CIMAC. The actual value to be used, no doubt, remains a matter of judgement. It would therefore be of interest to learn if the author's company has carried out any measurement or calculation of structural stiffness, static or dynamic, in way of bearing pockets and if so some details of this would be appreciated.

Finally, I would like to thank the authors for a very interesting paper and also for their full co-operation in the Co-operative Research Programme.

R. H. CROWTHER (Lloyd's Register of Shipping): I would like to congratulate the authors on a very interesting paper.

The problem of approval of crankshafts has been exercising the minds of the classification societies in recent years. Meetings of IACS have been held to consider and discuss the CIMAC proposals. At the last meeting a measure of agreement was obtained and the majority of the societies agreed to accept the document.

There are, however, aspects of the document which are viewed with some reservations, the most important one being

Table DII: Influence of mean stress. In-line engine running at 1800 rev/min

Parameter	Input data	
Ultimate tensile strength	690.0000	690.0000
Ultimate shear strength	483.0000	483.0000
Endurance limit in bending	282.5000	282.5000
Endurance limit in torsion	162.8000	162.8000
Mean direct stress	0.0000	67.9000
Rev direct stress	201.0000	201.0000
Mean shear stress	0.0000	17.9000
Rev shear stress	84.7000	84.7000

Theory	Factors of safety: Without mean stresses	With mean stresses	Change (%)
von Mises–Goodman (Lloyds)	1.1353	1.0111	12
ESDU (M = 1.5)	1.1353	1.0912	11
Goodman–Gough ellipse quadrant	1.1345	1.0456	9
Goodman–Gough ellipse arc	1.0968	1.0116	8
Kerr–Wilson combination	1.1345	1.1265	7
ESDU–Gough quadrant	1.1345	1.1087	2
ESDU–Gough arc	1.0968	1.0702	3
Gerber–Gough quadrant	1.1345	1.1268	1
Gerber–Gough arc	1.0968	1.0884	1
DNV–Gough quadrant	1.1345	1.0894	4

$$\text{CIMAC factor of safety} = \frac{\text{endurance limit}}{\text{von Mises stress}} = \frac{282.5}{\sqrt{(201^2 + 3 \times 84.7^2)}} = 1.135$$

the absence of any consideration of the mean stresses in the crankshaft, particularly the bending mean stress. The calculation for both pin and journal fillets are based on the bending moment of the centre line of the web. Additionally at the journal fillet allowance is made for the shear forces, which cause a direct stress in the web. In most cases the stress range in the journal fillet is higher than the stress range in the crankpin fillet, and thus the factor of safety at the journal fillet is the least. However, crankshafts rarely if ever fail through the journal fillet, and the explanation may be that the mean stresses are compressive in the journal fillet and tensile in the crankpin fillet. Could the authors comment on this aspect of mean stress?

Reservations have also been expressed that where the principal stress axis rotates the von Mises criterion of failure may not be the most appropriate.

Figure 1 of the paper is most instructive. The engine does not know which classification society stamp it bears and presumably will run happily at the lowest stipulated UTS shown, and it was to end such anomalies that agreement was obtained to the document.

I would take exception to the conclusions which ask the classification societies to adopt the statically indeterminate method. It was decided at the meeting that a simple calculation was required and the statically determinate method was adopted. In the case of trunk piston engines this gave a very reasonable estimate of the range of stress measured in this series of tests, and also in tests reported by Mr Guppo and Mr Gaudio of GMT in their paper 'Crankshaft bending stresses: experimental investigations and calculation methods' (Ref. 12 of the paper). The same does not apply to crosshead engines and the statically determinate method gives an overestimate of the stress. The classification societies, however, agreed to 80% of the determinate stresses.

In fatigue the range of stress is important and the maximum range is usually given by the stresses at TDC and BDC in the case of two-stroke engines, and TDC and BDC in the case of four-stroke engines. The simple calculation cannot explain what happens between these two points, but it will give a good estimate of the range.

I. J. BICKLEY (Mirrless Blackstone (Stockport) Ltd): The authors are to be congratulated on the clear way they have presented their paper on a far from straightforward topic.

My comments and queries are on behalf of the intended users of these rules who have an essentially practical approach and are familiar with the application of classification society rules, principally LRS (Part 5, Chapter 2) used to determine allowable power outputs and DNV (Part 4, Chapter 2) used to determine allowable nominal crank torsional stresses.

The rules should incorporate a simple, straightforward and clear method to enable use as an initial design tool. This would include graphs as well as formulae for the stress concentration calculations and a simplified method of radial-force calculation, particularly for Vee engines. The proposed rules are too cumbersome to enable such an assessement to be made.

The rules should also incorporate a more rigorous analysis method which would enable a more accurate assessment of the final design of crank to be carried out. We have used our current method of crankshaft stress calculations successfully for 15 years.[13] This method incorporates both a statically indeterminate approach and mean-stress effects. We entirely endorse, therefore, the authors' main conclusion that these rules should adopt the statically indeterminate approach, at least as an option. We do believe, however, that mean-stress effects must be included if the true state of stress within the crank, and hence its safety, is to be determined.

A factor of safety of 1.15 for such a vital component as a crankshaft is too low, especially in view of the authors' comments (Section 4) that two separate deviations of 5–10% in the measurements are feasible. If cranks have proved satisfactory in the past with factors of safety of 1.15 then this indicates that the method of calculation is inappropriate. It may be that the maximum torsional and bending stress concentrations do not coincide or that there is a phase difference between the bending and torsional loading. However, the reasons for the satisfactory operation of such cranks should be determined, rather than assuming that a factor of safety of 1.15 is, and will continue to be, acceptable. Generally speaking, when the operating conditions are well known the minimum factor of safety should be 1.3.

Did the authors also measure the free-end amplitudes as well as the torsional stresses? This would provide a further check on the accuracy of the calculations.

The low increase in fatigue strength factor (1.05) due to CGF forging of cranks does not accord with the higher benefits claimed by the crank manufacturers or allowed by the classification societies' rules as they now stand.

The current generation of crank designs incorporate a thin-webbed highly overlapped design, which is outside the parameters of these CIMAC rules. Have the authors an estimate of the extra amount of research necessary to enable the rules to cope with a crank of modern design or one which may be designed in the foreseeable future?

Although the rules deal with barred speeds and cylinder imbalance, they do not appear to consider the effect of a malfunction of a damper in damper-controlled torsional criticals where the bending stress is low and the torsional stress, allowed by the rules, is high. An agreed code of practice should be incorporated into the rules to discourage systems which, although they meet the rules when new, could give failures at a later date.

The observations in Section 9 that additional stresses should be allowed for in the torsional calculations over those calculated by the CIMAC method and that the additional bending stress proposed by CIMAC is halved for two-stroke engines and omitted altogether for four-stroke engines, surely highlights to the inadequacy of the present CIMAC proposals.

A set of rules which has been agreed by all the major classification societies and is a more accurate reflection of the true state of stress within a crankshaft is clearly in everyone's interest. However, the limitations of these proposals and the results and observations of the authors following their exercise with LRS and BV illustrate the size of the task to be completed before such a unified set of rules can be adopted.

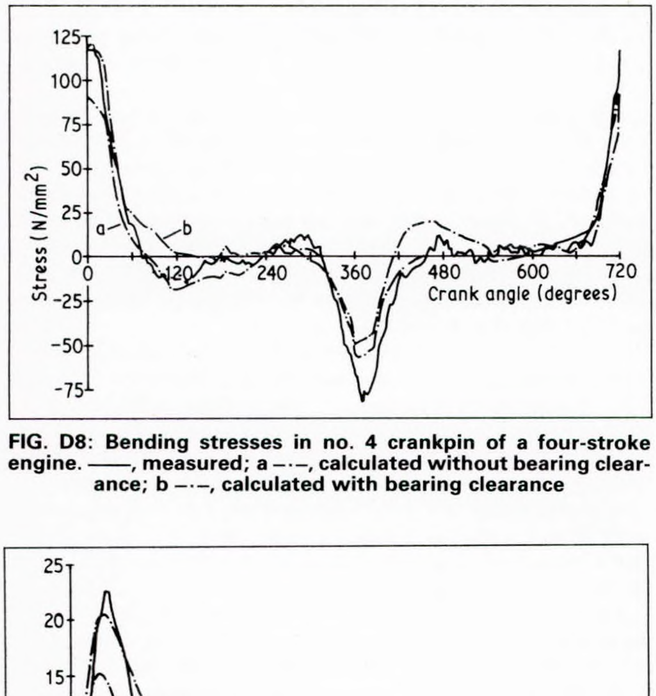

FIG. D8: Bending stresses in no. 4 crankpin of a four-stroke engine. ——, measured; a —·—, calculated without bearing clearance; b —·—, calculated with bearing clearance

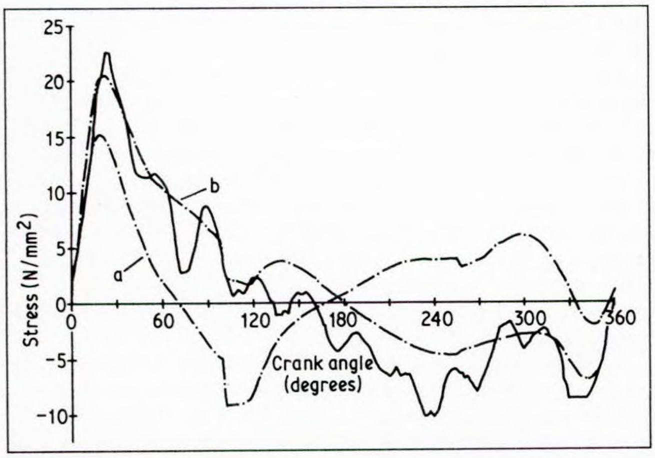

FIG. D9: Torsional stresses in crank no. 7 (free end) of a two-stroke engine. ——, measured; a —·—, calculated without bearing clearance; b —·—, calculated with bearing clearance

G. S. MOLE (Stork Werkspoor Ltd): This most interesting paper compares stresses calculated according to CIMAC rules with measured stresses in the crankshafts of two- and four-stroke in-line engines.

There is no reference in the paper to Vee configuration engines which, in my experience, have been subject to problems, specifically in relation to bearings but also on occasion to crankshafts. Can the authors give an indication of how the CIMAC rules are interpreted in relation to the double-firing load at each crank and whether any measurements on test-beds or in service have been made to verify the calculated values?

Can they also advise if the same factor of safety of 1.15 applies to Vee engines as for in-line engines?

C. GRAY (Ricardo Consulting Engineers): The development of unified rules for the design of marine diesel-engine crankshafts will clearly give many benefits, and it is encouraging to see that the programme of measurements has demonstrated good agreement between the calculated and measured stresses.

It seems surprising that only the crankpin and journal fillets have been considered as possible sources of fatigue failures. In a design office which deals with a wide range of engine sizes, from medium-speed designs down to high-speed engines of less than 1 litre capacity, the lubricating-oil hole break-out in the crankpin has to be considered as an additional, possibly critical, stress-raising feature. On engines having a simple through hole from journal to crankpin the break-out is not close to the neutral axis of bending, and this increases the total stress.

Although the paper is concerned only with large marine diesel engines it may be of interest to note that the crankshafts of automotive engines are often of nodular cast iron with the fillets rolled to increase their fatigue strength by 50% or more. Unfortunately, it has not so far proved practicable to apply a similar treatment to the oil hole, and it can therefore become the weakest point.

With regard to safety factors, the value of 1.15 proposed as the minimum allowable is below the usual range of values used by us for crankshafts and other major components. The main reason appears to be that the fatigue strength derived from the proposed CIMAC rules itself contains a safety factor. In principle, would it not be better to have a single overall safety factor instead of one safety factor for the calculated equivalent stress and another safety factor for the fatigue strength of the crankshaft?

One feature which we believe makes out own method for calculating crankshaft bending stresses applicable to a wide range of designs is the treatment of the web cross-section. Instead of basing the nominal stress on a section perpendicular to the web we base it on a section through the crankpin and journal fillets. This gives a closer approximation to the state of stress existing in crankshafts with large overlap and relatively thin webs. The fillet stresses are then determined more accurately because they are less affected by the accuracy of the stress concentration factors.

S. ARCHER: This paper will be welcomed by all marine engine builders and crankshaft designers, since it contributes powerfully to the argument for the unification of classification society crankshaft rules (Fig. 1 of the paper) and promotes the work of IACS in that context.

The paper is a rare example of cooperative research between the marine engine manufacturing industry and the classification societies. Hopefully it will encourage similar cooperative research in other marine engineering fields.

The authors use the CIMAC proposed method of calculation as a base against which to compare the various measured stresses, but unfortunately they have omitted to give relevant details of it for the benefit of those unfamiliar with it. Could they please provide details of the proposed method?

In the two-stroke engine it was regrettable that no stresses were measured in the crankpin fillets other than the end cranks. Presumably this was due to the extra difficulties of instrumentation such measurements would have incurred. However, if this had been possible, particularly at the most highly stressed crank, the value of the research would certainly have been augmented.

As regards the limiting fatigue strength of the crankshaft materials, it is noted that an empirical formula has been derived, based on experimental results of tests on a large number of crankshafts. Although details are presumably given in Ref. 2 of the paper, it would again be helpful if the formula could be presented. Could the authors also state the ultimate tensile strength of the (presumed forged) shaft materials for comparison with the assumed values of fatigue strength given in the paper?

On the stress concentration factors mentioned in Section 2, it would be useful if the authors could indicate what values have been used in the calculations for both direct and torsional stresses. Could they also state whether these are theoretical values or fatigue reduction factors, taking account of the notch sensitivity of the crankshaft materials?

The calculations for combining the direct and torsional stresses are based on the von Mises criterion, the authors observing that this errs on the safe side. In a different application I made use of this formula in my 1964 paper to this Institute,[14] and pointed out that it clearly shows the importance of torsional stresses compared with direct stresses.

The calculations appear to assume that the von Mises formula is applicable, even if the direct and torsional cyclic stresses are not in-phase. Would such a simplification significantly affect the calculated safety coefficients.

I am surprised that the authors, in their calculations of safety coefficients, have made no allowance for the effect of mean stress, both direct and torsional, in reducing the limiting fatigue strength of the crankshaft materials. This is surely well established in the history of fatigue research (eg Gerber and Goodman).

In my earlier paper[14] a six-cylinder, two-stroke engine with a forged semi-built shaft was investigated. The bending conditions were conventionally assumed as for a simple uniform cylindrical shaft, encastré at the mid-length of the main-bearing journals. This gives a bending moment approximating fairly closely to that obtained at the mid-thickness of web when using a simply supported assumption. Although the web-section modulus is only about 70% of that of the pin or journal, for simplicity all stresses were calculated on crankpin diameter, ie an underestimate of some 30%.

The shaft crankpin diameter was 550 mm and less than 1% over rule size with a minimum UTS of 32 ton/in^2 (494 N/mm^2 or 50.4 kg/mm^2). The crankpin fillet radius was 25 mm recessed 20 mm into webs.

One of the factors investigated was the effect of artifically increasing the flywheel moment of inertia, so as to bring the ninth order (crankshaft mode) into proximity with the service speed, giving a dynamically magnified stress equal to the maximum allowed in Lloyd's guidance notes of $\pm$1600 psi ($\pm$11 N/mm^2). The removal of this partly resonant critical vibration stress increased the calculated safety coefficient by some 20–25%.

The equivalent direct stresses due to bending and torsion (including the ninth critical) were calculated using various criteria of fatigue failure (Marin, Götaverken–Söderberg, and Götaverken–modified Goodman) and taking account of mean stress. At the most highly stressed position (found to be abaft no. 5 crank, no. 2 in the authors' convention) the calculated safety coefficients ranged between 1.52 for the Marin and Götaverken–Söderberg methods to 1.81 for the Götaverken–modified-Goodman method.

At the least stressed location abaft no. 1 crank, the calculated coefficients varied in a narrow range between 2.25 and 2.51. These values compare with the authors' CIMAC calculated statically determinate safety coefficients in Fig. 8 of the paper for the two-stroke engine. Since, presumably, in the case of the authors' seven-cylinder engine there are no resonant, or partially resonant, torsional vibration criticals, it is necessary, for a better comparison, to eliminate the effect of the ninth order, giving safety coefficients at no. 5 crank of 1.85 and 2.2 for the above respective methods.

The measured stresses for the end cranks in Fig. 8 of the paper indicate much lower values than when using either of the two proposed CIMAC calculation methods, but with less reduction compared with the statically indeterminate calculations. It would therefore seem that the latter, which, perhaps coincidentally, agree well with my results for the six-cylinder engine using the Götaverken–modified-Goodman basis, would represent a realistic preference in the proposed CIMAC calculation methods. It would produce a lower hidden safety factor and therefore more efficient utilization of the crankshaft material, as claimed by the authors.

To investigate the effect of axial vibration in the six-cylinder engine, the bending-stress range was increased by an assumed $\pm$4000 psi ($\pm$27.6 N/mm^2). (See CIMAC Report A.13, Copenhagen, wherein it is stated that such a high stress had been measured in no. 10 crank fillet near the service speed with the damper inoperative.) On the Götaverken–Söderberg calculation method the safety coefficient, at the most highly stressed section abaft no. 5 crank, was thereby reduced by only about 10%.

On the same basis, to estimate the effect of misalignment of the two-stroke crankshaft, a 50% increase in bending-stress range was assumed and gave a reduction of only 8% in safety coefficient.

The stress concentration factors for the six-cylinder crankshaft were taken as 3.0 and 1.6 for bending and torsion, respectively. However, according to Marin, quoting Lipson *et al.*,[15] the corresponding notch sensitivity factors for annealed steels are $q = 0.4$ and $q = 0.85$, respectively, giving fatigue reduction factors of 1.8 and 1.5 for bending and torsion, respectively. Using the Götaverken–Söderberg criterion, this would increase the safety coefficient at no. 5 crank by some 13%.

The authors are to be congratulated on the results of their research and in particular on their work in connection with the effect of shear stresses due to web twisting in adding to, or subtracting from, the torsional stresses. Note, however, that these are appreciably lower than the latter, especially when there is torsional resonance.

C. ARCHER (Lloyd's Register of Shipping): It is often the case that papers of a theoretical nature neither attract the attendance that they merit nor provoke a high-level technical discussion. Fortunately this occasion has proved that a topical subject can provide a basis for a successful technical meeting. Dr Donath and Mr Seidermann have provided a lucid paper, ably presented, and they are to be congratulated.

The proposed CIMAC rules for crankshafts have evolved over the last decade or so, largely due to the enthusiastic support of Dr Donath and his colleagues. Concurrently Lloyd's Register of Shipping had been conducting a programme of hardware measurements and calculation-method development which was reported widely, particularly in the ubiquitous 'Silver Book'.[10]

The joint research project decribed by Dr Donath brought together the different approaches to stress analysis which had been developed, with that of Bureau Veritas, and compared the calculations with direct measurement. I had the fortune to be actively involved throughout the project. The technical problems associated with the direct measurement of strain on an operating diesel-engine crankshaft were challenging. The expertise developed in the LRS Technical Investigation Department over many years resulted in the collation of the valuable measurements indicated in the paper. During the project a free flow of technical information between all the partners was maintained. The end result was a most worthwhile technical exchange which produced the conclusions, amongst others, given in the paper.

The discussion of this paper has been broad and lively, as indeed were many of those during the joint research project described. It has been particularly interesting to note the range of opinions expressed. The impression given is that industry is not necessarily agreed that the CIMAC proposal, which is soon to be adopted as the basis of a unified classification requirement by IACS, are entirely satisfactory. Some adverse comment has been made about the retrogression to simplified solutions whereas other remarks infer that the proposed methods may be overcomplex, when considered with the safety factors. I sincerely hope that this paper will promote further discussion in the near future so that some consensus can be obtained from British industry for the whole range of engine sizes.

On behalf of those present and the Institute of Marine Engineers I would like to thank Dr Donath for his presentation and to extend our collective gratitude to him and Mr Seidermann for their excellent paper.

Authors' reply

Comments on the contributions to discussion

We are very pleased that our paper on a relatively unusual subject has attracted so much attention and given rise to so many contributions. On many of the questions raised, we should like to give additional comments and information. However, this presents a problem: in the 10 contributions received, numerous problems are dealt with several times, although from slightly varying points of view. Replying to all the contributions in order would lead to repetition and our reply would become too lengthy. We have therefore summarized our replies by subject matter, with reference to the various contributors.

General

In some of the contributions (eg that of Dr Law) the CIMAC proposal has been criticised as being too imprecise because, for example, it only roughly allows for influences such as stiffness of the frame. In other contributions it has been stated that parts of the calculation are too complicated. These contradictory opinions confirm that the CIMAC sub-group has more or less reached the goal aimed at, as the CIMAC proposal must be a compromise calculation method with general validity for a large variety of engines while at the same time being as precise as possible with a justifiable calculation effort. The proposal was not drawn up for the primary purpose of providing a means of crankshaft pre-designing; engine builders can do this by simplifying the proposal.

We should like to emphasize that our paper was primarily intended to present the results of a joint investigation project which compared the stresses in the orginal crankshafts measured on large-bore engines with the calculated results. A detailed, fully comprehensive presentation of the CIMAC calculation method giving substantial reasons for the individual steps did not fit in the space limits set by the Institute. Reference should be made to Ref. 1 of the paper, which unfortunately only exists in the German language, to the Unified Requirements (UR)[16] which is in the possession of all classification societies, or to the text of the CIMAC proposal with the detailed explanations appended,[17] which was made available to all classification societies.

Substitute system

Mr Syed has remarked that the three-dimensional beam model in the 'Silver Book'[10] better represents the crankshaft than a constant cross-section substitute continuous beam. Theoretically, this is correct. We would, however, point out that for exemplified load cases we have shown the moment curve, with equal bearing stiffness, for a constant cross-section continuous beam as well as for a three-dimensional beam model. The moment curves obtained were not identical mathematically, but the differences were negligibly small. The calculation effort required for a constant cross-section continuous beam, however, is distinctly lower.

As to the origin and magnitude of the substitute inertia moment used by us and mentioned by Mr Volcy, we had intended to show in our paper the precision that can be achieved with the CIMAC proposal. Therefore, we have kept to the values given there: for the two-stroke engine, the substitute diameter was fixed at 85% and for the four-stroke engine at 80% of the crankpin diameter. A substitute diameter of about 86% of the crankpin diameter could be derived from Mr Volcy's bending tests made on the four-stroke engine crankshaft. The slightly higher value can probably be explained by the fact that on this crankshaft the journal had a larger diameter than the crankpin.

According to our experiene, this deviation of the substitute diameter has practically no influence on the precision of the result. Of more importance is the selection of the correct bearing stiffness. Adapting the calculated results to the measured results by means of bearing stiffness variation compensates for the error in the substitute diameter.

Dr Law mentions the influence of the frame stiffness on the calculation result. For a generally applicable calculation proposal for everyday use, the application of finite-element calculation methods for the bearing stiffness is, however, out of the question. Principally, the influence of the bearing stiffness is allowed for in the so-called A value of the CIMAC proposal, which represents the relationship between crankshaft stiffness and bearing stiffness. As has been established by Cuppo and Gaudio, in Ref. 12 of the paper, and as applied by us, an adaptation of calculated and measured stresses permits us to determine this A value for a typical engine or frame design and to transfer it to other engines. However, the A value cannot be determined principally by a static test or by calculation because it also makes allowance for the influence of the lubricating film and, to some extent, the bearing clearance.

Comparisons between measured and calculated results were made by our company to determine the magnitude of A for the engines it builds, with a relatively narrow scatter. For two-stroke engines we obtained $A = 3$ to 5, for medium-speed engines with the crankshaft embedded in the bedplate $A = 4$ to 6, and for the medium-speed engines with underslung crankshafts and for high-speed engines $A = 5$ to 8.

The use of the statically determinate calculation method in the UR is regarded by us as a first step. For the stress calculations of any individual throw, the statically indeterminate method is definitely more precise (see comment of Mr Crowther). For calculating the smallest safety coefficient of a whole crankshaft, eg of a medium-speed engine, the difference between the statically determinate method and the statically indeterminate method is relatively low, as illustrated by Fig. 9. For the throw with the smallest safety coefficient there is also satisfactory agreement between the calculated and measured results. According to Cuppo and Gaudio and as shown in Tables I and II of the paper, the statically determinate method of calculation is practically always on the safe side. Even though the statically determinate method provides for over-dimensioned crankshafts in certain cases, as mentioned by Dr Law, the UR in its present form is for engine builders distinctly superior to the great number of previously used and differing calculation formulae proposed by the classification societies, not least because it is easier to use.

Additional stresses caused by bending vibrations and gyration effects are known to us from the literature.[18] The CIMAC calculation proposal makes allowance for these only in a general form as the 'memory quantity σ_{add}'. When the proposal was discussed in the CIMAC group, none of the members had encountered such phenomena in practice, and they probably only occur on ultra-high-speed engines not falling under the UR. More distinct are the deviations between measured and calculated results on the end throws of engines with outboard bearings, as described in the paper, where it has been proposed, for the time being, that better agreement between calculated and measured results is obtained if the outboard bearing is disregarded in the calculation. The possibility of supplementing a calculation program for continuous beams to include directly the influence of bearing clearances may certainly be expected for the near future in view of the rate of computer development.

Although both of the engines dealt with in the paper were in-line engines, in reply to Mr Mole's comments the CIMAC calculation method applies to both in-line and Vee engines. The only essential difference is the modified calculation of the radial force, which is described in more detail in Ref. 16.

Stress concentration factors

The graphs and formulae considered necessary by Dr Bickley already exist: the graphs are contained in the explanatory notes relating to Ref. 17 and the formulae in Refs 16 and 17. Our experience is that when the calculation method is used relatively often, the programming of stress concentration factor formulae is more convenient than the use of graphs.

Table DIII: Stress concentration factors

Engine	*Pin*		*Journal*		
	bending	torsion	bending	shearing	torsion
Two stroke	4.07	1.71			
Four stroke	1.88	1.85	1.92	3.47	1.88

For the comparative calculations described in the paper the stress concentration factors were determined from the above mentioned sources. The actual factors used are given in Table DIII.

To answer the question as to how, for a crankshaft outside the scope of the CIMAC proposal, the stress concentration factors can be determined, it is a very laborious procedure to derive the stress concentration factor characteristics from a large number of test results, as described in Ref. 4 of the paper. An extension of the stress concentration factor characteristics in respect of the range of certain parameters would require the new determination of all characteristics and involve much effort.

If the calculation procedure described is only to be used for a specific crankshaft outside the scope, there is a relatively simple method of obtaining the necessary stress concentration factors by making a steel model with which to determine the stress concentration factor by the use of strain gauges. The possibility of re-working the steel model in individual parts allows several similar geometric variants to be treated with a single basic model. The details of the procedure are also described in Ref. 4 of the paper. The stress concentration factors mentioned in the paper are all statically measured values representing the relationship of the maximum fillet stress to the reference value. The influence of the stress gradient on the fatigue strength asked for by Mr Archer is contained in the fatigue strength formula.

As regards the reference cross-section for the nominal stresses, which on multiplication by the stress concentration factors yield the fillet stress, we agree with Mr Gray that this cross-section can also be placed obliquely between the fillets of crankpins and journals. In this case, stress concentration factors other than those used with the method contained in the CIMAC proposal must be used.

Considering the whole range of geometric variants of crankshafts covered by the CIMAC proposal, the exactness of the stress concentration factor to be used for a specific variant doubtless depends on the total number of test results used in determining the characteristics. As described in Ref. 19, the FVV test programme, the results of which were incorporated in the CIMAC proposal, comprised a total of 590 different measuring variants of its tests and 218 measuring variants of other authors for determining the characteristics. Not least as a result of the great characteristics density reached, it was possible to obtain a standard deviation of $<10\%$ for the bending stress concentration factors and $<8.5\%$ for the torsion stress concentration factors. To verify whether another definition of a reference cross-section with the pertinent stress concentration factors would yield better results for the stresses, the new definition would have to be applied to the above-mentioned 808 different test results, which would involve a great deal of effort. We believe that the high accuracy mentioned above will not be reached by stress concentration factors from another source.

We agree with Mr Gray's objection that under certain circumstances the outlet of the oil bore can represent a weak point on a crankshaft. It should, however, be recalled that the case of treated fillets, the most probable case where this might occur, is not covered by the UR. The philosophy of the CIMAC sub-group was that in the case of standard crankshafts the designer will always be in a position to shape the outlet of the oil bore so that the safety at this point is higher than in the fillet.

Until now an argument against the inclusion of a calculation specification has been the fact that it is difficult to find generally applicable stress concentration factors for this point that apply to all possible variants. For this reason the UR contains the remark that on request from the classification society the engine builder is, from case to case, to furnish proof for the safety at this point.

Additional stresses

It does not seem fair for Dr Bickley to argue that differences between the original form of the CIMAC proposal and proposals contained in our paper suggest general short-comings of the former. Note that the CIMAC proposal was drawn up in the years 1972–1979. Until now only empirical values could be used for the additional stresses due to misalignments which definitely exist but are disregarded in most of the present classification rules.

When the proposal was drawn up, it was evident to the CIMAC sub-group that the measuring results available for this aspect were scarce. In view of the discrepancies existing between practice-proven crankshafts and the design specified by some classification formulae (Fig. 1 of the paper), the engine builders could not afford to wait until any possible future developments had been included in the new calculation proposal. The findings on additional stresses mentioned in the paper only became known after submission of the proposal to IACS.

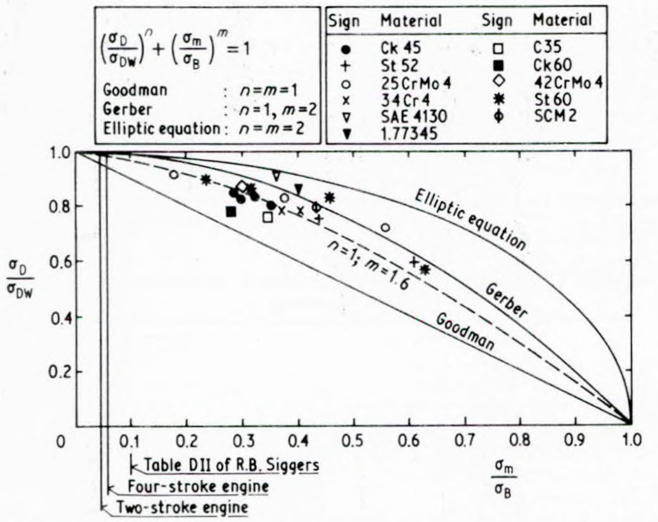

FIG. D10: Influence of the bending mean stress on the tolerable bending stress amplitude

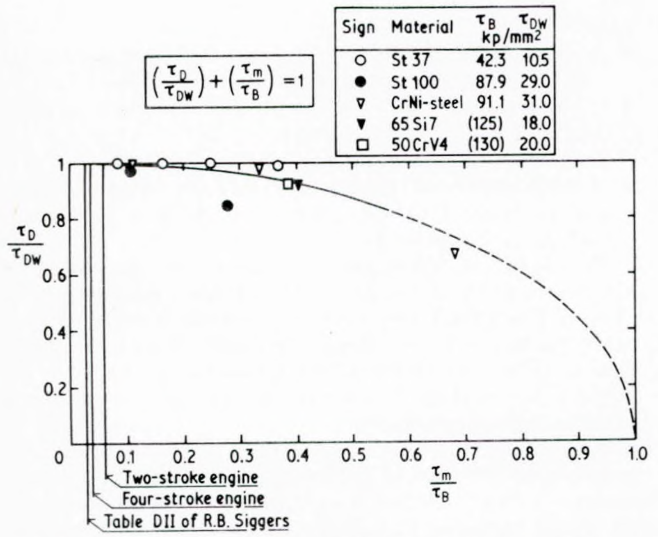

FIG. D11: Influence of the torsional mean stress on the tolerable torsional stress amplitude

Mean stress influence and failure theory

Several contributors complained that the CIMAC proposal does not make allowance for the influence of the mean stresses in determining the reference stress. This point was also a subject of importance in the discussions between the working groups of IACS and CIMAC, and therefore we should like to substantiate in detail why this decision was reached, which at first sight seems embarrassing.

It is generally known that the fatigue strength is reduced by tensile mean stresses; this applies both to uni-axial vibration stresses and torsional stresses. Since the effect of mean stresses depends on their magnitude and on the material, it is necessary for a quantitative statement to investigate how both parameters behave in the case of crankshafts.

In the dimensionless Haigh diagram, Fig. D10, a number of test points have been plotted, some of which were taken from the literature[20] and some from own investigations. The steels concerned fall within the tensile strength range of $\sigma_B = 500$ to 1000 N/mm^2 and are frequently also used for crankshafts. These axial fatigue tests therefore show the probable influence of the mean stress on the fatigue strength of crankshafts under alternating bending stresses. As can be seen, the Gerber parabola under the classical approach would be the best suited. Optimum agreement with the mean value of the test results is achieved if the function with the exponent $m = 1.6$ is chosen.

In Fig. D10 we have also entered the calculated mean stresses referred to the tensile strength that were obtained from our investigations on the two-stroke and four-stroke engines as well as that given in Table DII of Mr Siggers. As to Table DI, we should like to remark that we do not consider it acceptable that a decisive influence of the mean stress is proved by a fictitious case that is outside the scope of the CIMAC proposal; the safety coefficient is 32% below the admissible limit and the alternating bending stress alone exceeds the fatigue limit by 11%; accordingly the mean stresses are also unrealistically high.

By analogy to the treatment of the bending stresses, the torsional stress behaviour is represented in Fig. D11.[21] Owing to the test material, no influence of mean stresses can be seen in the range of torsional mean stresses of crankshafts; nevertheless, a formulation with the elliptic equation is used in the following. In this connection we should like to point out that the ultimate torsional strength τ_B is slightly higher than the ultimate tensile strength σ_B; it can be calculated in accordance with the approximation formula[22] given below. It would be interesting to know why in the discussion the ultimate shear strength is used in this connection and was assumed to be $0.7\sigma_B$, as shown in the examples of Mr Siggers.

If the opinions expressed later are ignored, the safety S (proposal made by VDI, described in Ref. 23 on p. 67) can be obtained with the following equations, using the above-mentioned mean stress dependences:

$$\left(\frac{\sigma_a}{\sigma_D}\right)^2 + \left(\frac{\tau_a}{\tau_D}\right)^2 = \frac{1}{S^2} \tag{3}$$

where

$$\sigma_D = \sigma_{DW}\left[1 - \left(\frac{\sigma_m}{\sigma_B}\right)^{1.6}\right] \tag{4}$$

$$\tau_D = \tau_{DW}\sqrt{1 - \left(\frac{\tau_m}{\tau_B}\right)^2} \tag{5}$$

$$\tau_B \approx 1.32\sigma_B - 0.00035\sigma_B^2 \text{N/mm}^2. \tag{6}$$

The fatigue limits are evaluated by Equations (4) and (5) with the mean stress influence and entered into the elliptic Equation (3), which in the special case of $\tau_{DW} = \sigma_{DW}\sqrt{3}$ contains the von Mises criterion. We suppose that Mr Siggers

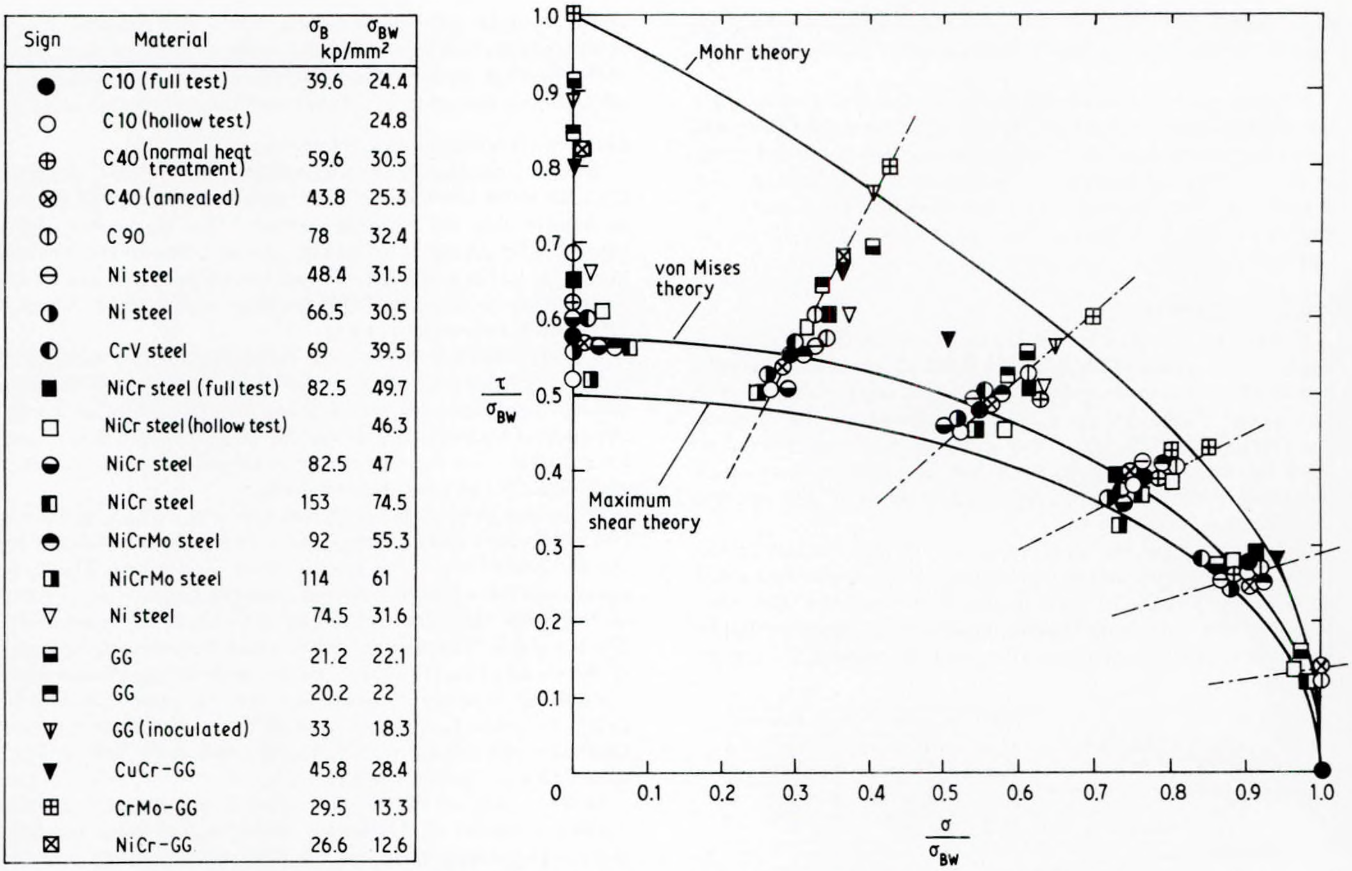

FIG. D12: Dependence of the torsional fatigue limit on the bending fatigue limit in ferrous materials

has applied a similar method to calculate the safety coefficients with other mean stress dependences being used.

Quantitative evaluation of the above relationships yields, for the crankshafts dealt with in our paper, a reduction in safety by the mean stress influence of less than 1% and of 1.6% for Table DII of Mr Siggers. We feel that compared with the scatter of other characterising quantities it is not worth making allowance for such a negligible influence.

In the absence of personal experience, we will not deal with the case of the high-speed diesel engines referred to by Dr Law, on which high tensile mean stresses are said to occur even in the journal fillet. We think, however, that the CIMAC proposal is not intended, at least not primarily, for such engines.

Dr Law and Mr Crowther state that crankshafts are known to break preferably in the crankpin fillets, and the suspicion is expressed that this is a result of tensile mean stresses. In the CIMAC working group nothing was known of such experience, and in our company there has been no case of a broken crankshaft due to fatigue failure for the last 15 years. Where crankshafts have broken it has always happened in connection with bearing damage. We can hardly imagine that other engine builders have statistical test material on real fatigue failure of crankshafts that are within the scope of the proposal.

Mr Crowther and Mr Archer refer to the question of the applicability of the von Mises criterion. As is shown in Fig. D12,[24] the applicability is sufficiently proved for steels if there is a synchronous and sinusoidal stress pattern of the same vibration frequency.

Recently, two new theories have become known in respect of stresses in crankshafts, with the magnitude and direction of the main stresses varying over one cycle: the shear stress intensity theory[25] and the theory of Simbuerger.[26]

No applications of the first theory to crankshafts have yet been published, but some interesting results, partly supported by tests, are contained in the research report, such as:

Table DIV: Safety factors calculated according to different theories

Engine type (two stroke)	*Safety factors* von Mises	Simbuerger
Bore 760 mm, stroke 1550 mm, power 2400 hp/cyl	2.5	2.7
Bore 760, stroke 1600 mm, power 2600 hp/cyl	2.8	3.0
Bore 580 mm, stroke 1700 mm, power 1920 hp/cyl	2.6	2.6

- A phase shift is of no influence on the fatigue strength in the case of sinusoidal stresses of the same frequency.
- If mean stresses occur, the phase shift does have an influence.
- Frequency differences affect the fatigue strength.
- A triangular stress pattern increases the fatigue strength and a trapezoidal stress pattern reduces it in comparison with sinusoidal patterns.

The theory of Simbuerger also allows the general stress conditions to be calculated. One of the members of the CIMAC Crankshaft sub-group, Mr Aeberli of Sulzer, tried to apply this theory to measured crankshaft stresses. As can be seen in Table DIV, the von Mises criterion contained in the CIMAC proposal tends towards the safe side in comparison with the Simbuerger theory.

We consider these new approaches to be promising, but inclusion in the CIMAC proposal was out of the question because extensive computer programs are required to evaluate the stress patterns. It remains to be seen whether future findings will call for a modification of the strength theory for crankshaft calculations.

Fatigue strength

It was pointed out by Mr Siggers that the CIMAC proposal does not contain uprating factors for treated crankshaft fillets. The question as to whether such uprating factors should be included was discussed in great detail, both within the CIMAC sub-group and in the discussions with IACS. The CIMAC sub-group was of the opinion that treating procedures such as induction hardening, rolling or shot-peening, depending on the methods used, differ so vastly that the existing test material does not suffice to establish the general applicability of such factors. The CIMAC proposal and the UR therefore contain the remark that such uprating factors have to be substantiated from case to case.

A special case of uprating factor is that for continuous-grain flow forging. The test material used for the CIMAC formula permitted an uprating factor of 1.05 to be derived for continuous-grain flow-forged crankshafts, which in comparison with the uprating factor commonly used by Lloyd's Register is regarded as too small by Dr Bickley.

After submission of the CIMAC proposal to IACS, an investigation programme on the fatigue strength of large crankshafts was conducted in Germany by FVV (Forschungsvereinigung Verbrennungskraftmaschinen e.V.). Unlike the CIMAC compilation, which was limited to the collection of available tests made at many different points and by different methods, this programme comprised investigations on 20 crankshafts by the most modern and strictly uniform methods, with the results being evaluated by one institution. Approximately equal numbers of the crankshafts investigated were open-die forged and continuous-grain flow forged. The investigations did not reveal any significant differences between these two forging processes. Note that the continued technical development apparently results in a permanently improving purity of steels; eg the grain flow is hardly visible in micrographs. It seems reasonable that improvements in strength as a result of the grain flow are therefore no longer of influence.

The fatigue strength formula worked out by the CIMAC sub-group and mentioned in the paper is contained in Refs 1 and 2 of the paper; as requested, it is repeated below:

$$\sigma_{DW} = K(0.42\sigma_B + 39.3) \times \left(0.264 + 1.073D^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R}}\right)$$

where σ_B is the tensile strength of the crankshaft material in N/mm^2, D is the crankpin diameter in mm, R is the fillet radius of the pin in mm, and K is the correction factor for the manufacturing process.

For comparison, the ultimate tensile strength values of the open-die-forged fatigues investigated in the joint project are:

K7SZ 70/125 B engine, 640 N/mm^2
8L 40/45 engine, 590 N/mm^2.

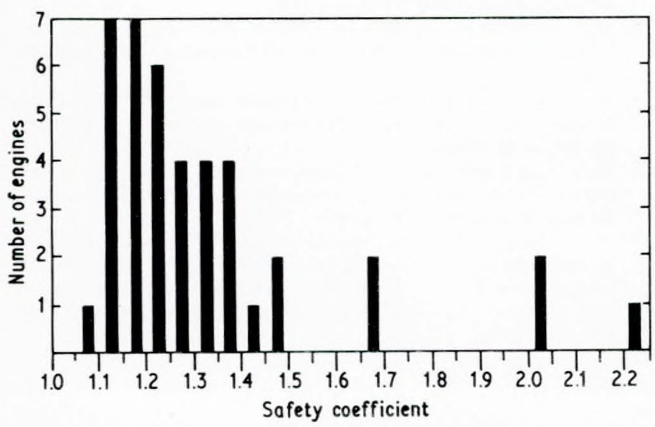

FIG. D13: Frequency of the safety coefficient in the case of four-stroke engines with statically determinate calculation of the bending stress

Safety coefficient

Dr Law and Dr Bickley complained that the admissible safety coefficient for the vital component of the crankshaft has been fixed at a very low value of 1.15.

It is true that such a low value is unusual for other components and distinctly higher values are recommended in the literature. It can, however, be proved that in connection with the entire calculation proposal the establishment of this value is reasonable and that the acceptance of crankshafts according to this proposal involves no risks.

In drawing up the CIMAC calculation proposal it was made a principle that the individual characterizing quantities yielding the safety coefficient should contain as few 'hidden' margins as possible. This is of great importance to the engine builder when using the calculation procedure as a dimensioning aid, and individual influences are to be included in respect of their effective importance. When taking into account the fact that all individual quantities are subject to statistical scatter, the most exact method would be to refer the calculation formula for every quantity to the average value of the scatter and to derive the necessary safety coefficient from this scatter by statistical methods.

Unfortunately, it shows in the final analysis that the mean value and scatter are only known relatively exactly in the case of the calculation formulae for the stress concentration factors. For all the other quantities this is not the case, and some necessarily still contain hidden margins that cannot be determined quantitatively, or at least only imprecisely. As was correctly assumed by Mr Gray, it can also be estimated on the grounds of more recent test results that the fatigue strength of about 80% of the crankshafts tested is higher than that calculated by applying the CIMAC formula.

As it seems that the safety cannot be expressed by an absolute value, a limit had to be fixed by recalculating for a number of engines for which sufficient safety had been proved in practical operation. This is demonstrated by the calculations made by Mr Scholz of Germanischer Lloyd to verify the CIMAC proposal.

Figure D13 shows the frequency distribution of the safety coefficient of 43 four-stroke engines with statically determinate calculation of the bending stress. The calculation results were grouped in classes with a width of 0.05.

Figure D13 shows that as many as eight crankshafts are within the safety range of 1.05–1.15 and would therefore not be acceptable according to the CIMAC proposal when applying the statically determinate method. When calculating the bending stress by the statically indeterminate method, these crankshafts would at least rise to class 1.15–1.20 and could thus be accepted. This effect was intended by the CIMAC sub-group because it involves the necessity of applying the more exact calculation method in cases of stringent material economy.

This is also the reason why in the CIMAC proposal the same admissible safety coefficient has been chosen for statically determinate and statically indeterminate bending stress calculations. Making reference to an example in which the statically determinate method yields about 40% higher values than the statically indeterminate one, Dr Law demanded that different coefficients should be fixed. The argument against this is that distinctly smaller differences occur for four-stroke engines with position-aligned throws. Table II of the paper shows that the difference on the throw with the least safety is only 7%. However, such cases have primarily been considered in supporting the admissible safety coefficient.

It has been pointed out that the UR yields higher safety than in Fig. D13 despite the statically determinate calculation method because in the UR the additional bending stress has been reduced from 20 N/mm^2 in the CIMAC proposal to 10 N/mm^2. We believe this change to be justified because the discussion of the CIMAC proposal revealed that no additional dynamic bending stresses resulting from alignment occur on modern four-stroke engines (trunk-piston engines) and so only part of the hidden safety is eliminated.

We think that the preceding comments show that the IACS and CIMAC working groups have dealt thoroughly with the question of the admissible safety coefficient, and that amendments would only be reasonable if new aspects in the assessment of individual characterizing quantities turn up in the future.

Test programme of the cooperative research project

It was assumed that the two-stroke engine throws selected for the investigation would not be those subject to the highest stress. The reason for the selection of the throws investigated was that at the coupling end the influence of the lineshaft was to be included while at the free end the highest bending stress must be expected because of the unilaterally missing restraint.

Figure 8 of the paper shows that the calculated safety value on some throws within the engine is as low as at the free end (at any rate it is not considerably lower). While the safety at the free end depends on the high bending stress, the torsional stress is predominant in the middle of the engine.

Introduction of the Unified Requirement

In several contributions it was pointed out that improvements are required before the UR can be implemented. It will doubtless have been noticed that some of these proposals are contradictory. For example, should we develop a simple and sufficiently exact calculation method for everyday use or draw up a sophisticated calculation method with an exactness making allowance for the latest state of the art.

This has been the subject of discussions within the CIMAC sub-group as well as those, extending over several years, between the IACS working party on engines and the CIMAC crankshaft sub-group.

Figure 1 of the paper illustrates the urgency of the problem: present circumstances do not allow the engine builders to tolerate the situation that for one and the same engines two different crankshafts have to be built, depending on the classification society in charge of acceptance. It is not difficult to imagine that one minor aspect of a calculation proposal like that presented here can be discussed for so long that new results turn up; thus possible good achievements are lost in the permanent search for even better ones.

We wish to thank all the members of the Crankshafts working group of the working party on engines who agreed with the opinion that only a pragmatic solution is possible, and we thank in particular Mr Crowther, who also supported this standpoint in his contribution to the discussion.

The UR necessarily constitutes a compromise between the two above-mentioned extreme attitudes. It has been agreed between the working party on engines and the CIMAC sub-group that the UR is but a first step in the right direction. The cooperative research project on which we have reported shows that the calculation method is on the safe side. Expectations were left unfulfilled on the part of both the working party on engines and the Crankshafts sub-group; they are to be realized in a future version. All contributions that may improve existing weaknesses of the draft are welcome. They must, however, make allowance for the practical limits of such a calculation proposal.

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