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# Stress Investigations on Large Diesel Engines

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An account is given of stress investigations carried out on models of certain components of the Sulzer RD marine engine series. The model tests were supplemented by measurements on engines in operation and strength tests on welded joints.

## INTRODUCTION

A draft design that is correct from the form aspect by no means implies optimum performance from a component with regard to strength. To achieve a good design it is sufficient for the designer to combine imagination with general experience, but a weight-saving configuration that is correct from the strength aspect calls for additional knowledge obtained from detailed investigations. If quantitative results are demanded, it is possible to get by with more or less straightforward calculations for machine components, but bigger, complex elements are usually outside the range of theoretical methods. In such cases resort must be had to measuring techniques, to produce experimental answers to the questions involved. In the field of strength investigations three main possibilities are exploited:

a) the brittle lacquer method;
b) photo-elastic investigation;
c) strain measurement by electrical resistance pick-ups.

It is assumed that the reader is acquainted with the basic characteristics of these measuring procedures[1, 2], and it should therefore suffice to recapitulate the fundamentals.

The first method entails spraying a brittle lacquer on to the component to be examined. When load is applied, the tensile stresses set up produce cracks in the lacquer oriented at right angles to the main stress axis. These crack patterns permit analysis of the principal stress axes and reveal the points at which maximum stressing occurs. Since this method is still very sensitive to temperature and humidity as yet, quantitative results that can be evaluated accurately are possible, at best, under laboratory conditions in practice.

Two-dimensional photo-elastic investigation is a model method. Deductions on the stress pattern are drawn from the way in which polarized light falling upon a loaded model of amorphous, isotropic material is refracted. Between the state of stress characterized by the principal stress difference ($\sigma_1 - \sigma_2$) and the double refraction resulting from the phase shift $\delta$ of the two component rays there exists the following relation:

$$\sigma_1 - \sigma_2 = \frac{S}{d} \cdot \delta \qquad (1)$$

where $\sigma_1 - \sigma_2$ = principal stress difference in kg/mm$^2$;
$S$ = stress-optic coefficient approximately 1·07 kg/mm and order;
$d$ = penetrated thickness of model in mm;
$\delta$ = phase shift (fringe order, isochromatic order).

In the three-dimensional photo-elasticity technique, the deformations, and hence the stresses, are frozen by a suitable method. If the three-dimensional model is then carefully sawn into plane sections, the frozen deformation and stress states are retained. The stresses in the plane of each section can be evaluated in turn by equation (1). But in this case the stress-optic coefficient is about 0·027 kg/mm and fringe order. Even though it is possible to determine the stresses and their gradients on the model exactly, and a test of this nature gives a good picture of the general distribution of $\sigma$, difficulties may arise with three-dimensional investigations—especially when dealing with problems of load application—because it is not possible to fulfil all the model laws exactly when transferring the results to the actual component.

It is also possible to measure deformations by assessing strains with electrical resistances (strain gauges). If an electrically-insulated conductor (wire coil or foil strain gauge) is bonded to a component, deformation changes in the latter result in variations in the length of the strain gauge. The accompanying changes in the cross-sections of the current-carrying conductor produce a measureable variation in the resistance, which will be proportionate to the deformation imposed if a suitable conductor material is employed. Thus:

$$\frac{\Delta L}{L} = \frac{1}{K} \cdot \frac{\Delta R}{R} \qquad (2)$$

where $\frac{\Delta L}{L}$ = elongation of component in the direction of the bonded strain gauge $\left(\frac{\Delta L}{L} = \epsilon\right)$ in per thousand;
$\frac{\Delta R}{R}$ = measured relative resistance change in the strain gauge, in per thousand;
$K$ = pick-up factor; about two with commercially available strain gauges.

Thus for materials obeying Hooke's law ($\sigma = E \cdot \epsilon$, $E$ being the modulus of elasticity), the following is valid for uni-axial stressing:

$$\sigma_1 = E \cdot \frac{1}{K} \cdot \frac{\Delta R}{R}$$

If it is desired to measure a plane stress, such as occurs on the surface of a component for example, under the most unfavourable conditions, i.e. when the main directions of stress are not known, the strains will have to be measured in three different directions at one measuring point. From these three measured values it is then possible to calculate the two principal stresses, $\sigma_1$ and $\sigma_2$, and their directions.

While the strain gauges now available will measure static and dynamic stresses in practically all temperature ranges occurring in general engineering, there are, nevertheless, three circumstances which may exercise a deleterious effect:

1) A strain gauge integrates the elongation over its measuring length, and thus gives only a mean value for $\sigma$. Where the stress distribution includes peaks with correspondingly significant variations in the stress gradients, it will not always be possible to select

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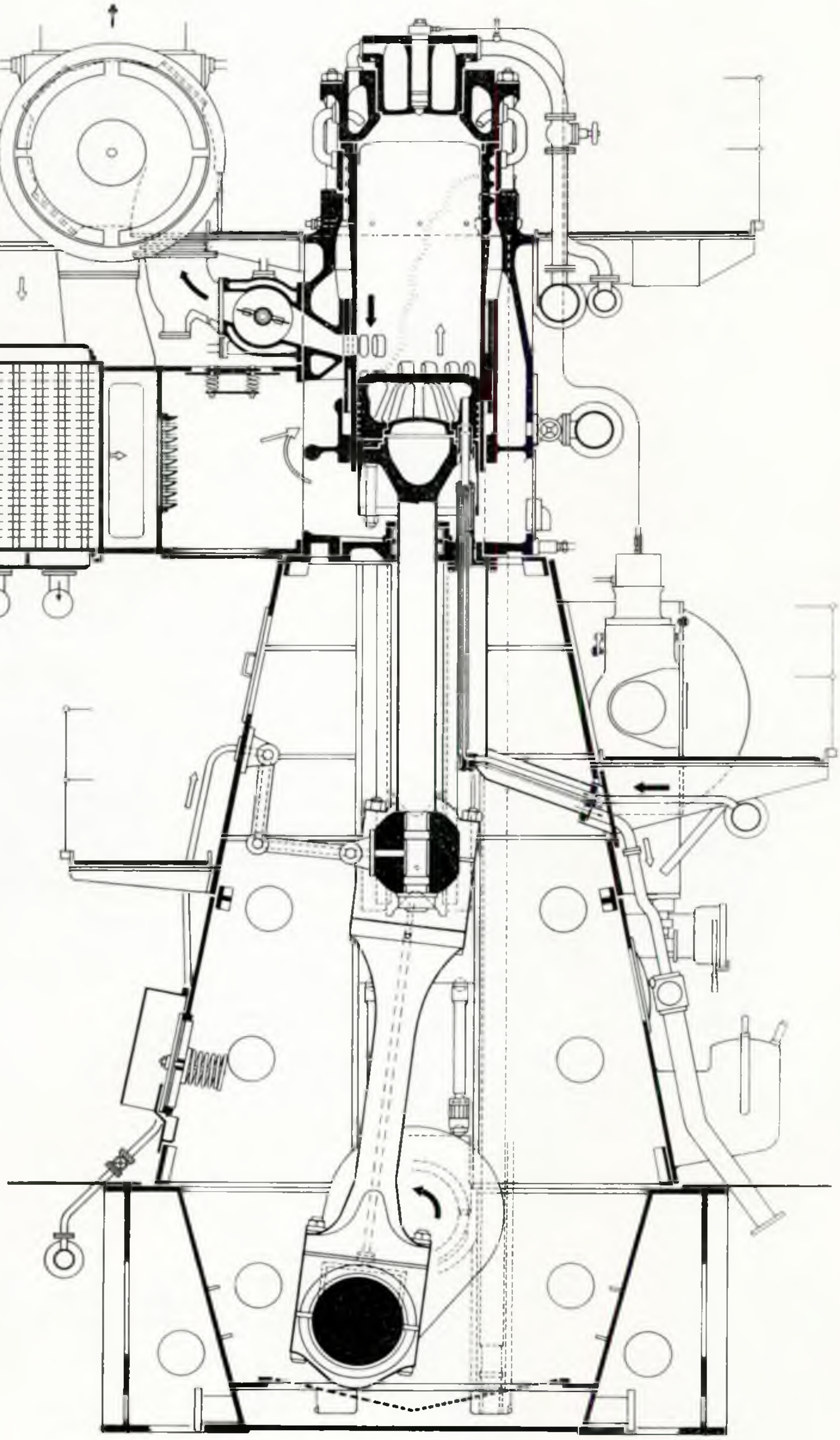


FIG. 1—*Cross-section of Sulzer marine Diesel engine of the RD series*

a sufficiently short measured length for the strain gauge to ensure adequately accurate detection of the stress peaks actually occurring.

2) Usually it is not possible to place the strain gauge exactly on the points of highest stress, for it is just these points that the measurements are intended to reveal. It is necessary to group several measuring points into measuring sections, so that the points under maximum stress can be ascertained by interpolation or extrapolation as the case may be.

3) Only the stress distribution can be measured on the surface of a component, the stress gradients within it remain unknown. But a knowledge of these may be very desirable in connexion with the fatigue strength.

Since the three-dimensional photo-elasticity technique is not subject to these shortcomings, photo-elasticity and strain gauges very often supplement each other, especially when all details are to be established. When investigating the structural strength of machine parts, similar criteria as set out below will be observed, dividing the investigation into three stages:

### 1—IDENTIFICATION AND DEFINITION OF A NEW PROBLEM

Before any start can be made with the investigation, the problem must be set. This statement may seem paradoxical, but it is necessary to establish first, from the available indications, where the actual problem lies, otherwise research may start off in the wrong direction, with attendant loss of time and material on a substantial scale. For this reason the most important factor in an investigation is close identification of the problem, so that it can then be defined as accurately as possible. It is thus indispensable that the possible, largely physical background is considered, and a working hypothesis for resolving the attendant difficulties deduced from it.

### 2—INVESTIGATING AND SOLVING THE PROBLEM

Having defined the problem, several avenues present themselves for attacking it. The path to solution is chosen according to the conditions in force:

i) Extrapolation of existing knowledge. There are many cases that can be mastered with sufficient reliability by extrapolating existing experience*. These are mostly problems in which a small number of known limiting conditions has to be fulfilled.

ii) If the limiting conditions have not been established with finality or if there are too many of them, experience alone will no longer suffice. Here the investigation will have to be widened to include certain detail studies of theoretical or experimental nature.

iii) If an entirely novel problem is involved, basic research —some of it comprehensive in scope—may be unavoidable.

* By "existing experience" the author means the knowledge of one or more persons, whether on the staff of a private firm or a scientific body, according to the standpoint from which the problem is being analysed.

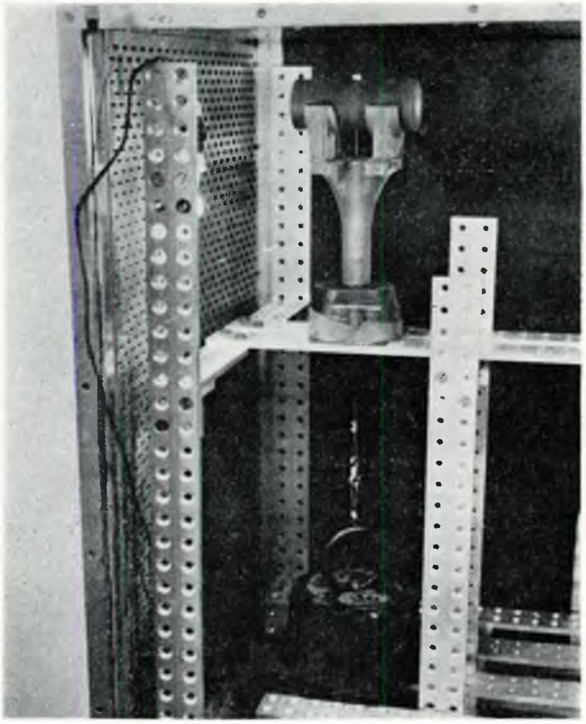


FIG. 2—*Photo-elasticity freezing test—Connecting-rod model rig in the furnace*

### 3—EFFECTIVENESS AND VERIFICATION OF THE ADOPTED SOLUTION

Once a solution has been reached by one route or the other —usually a compromise between the theoretically ideal, the practically feasible and the economically admissible solution— the selected improvement or new design will have to be examined to verify that it does, in fact, solve the problem as identified and defined according to Section 1. It is of primary importance to establish how reliably the new design envisaged meets the requirements placed upon it.


FIG. 3—*Isochromatic photographs of the shell section "s" of Fig. 4, for three different section thicknesses*

*Example* 1

All large Sulzer Diesel engines of the RD series are characterized by the same basic layout (see Fig. 1). For the smallest type in this range—the RD44 engine with 440-mm bore and 760-mm stroke—the demand for optimum weight economy in the design of the running gear was more imperative than for the bigger units. Thus the attempt was made in the design stage to produce the piston and connecting-rods as bodies of equal strength as far as possible. Three-dimensional photo-elastic model tests were carried out to establish how far this aim might be realized.

The connecting-rod has a shaft terminating in a square end at the top, which carries the two housings for the cross-head-pin bearings. A recess had to be provided in the top end of the connecting-rod, to enable it to miss the pipes of the water-cooling arrangements for the piston. But this recess entails stress concentrations, and clarity was desired regarding the magnitude of these.

Fig. 2 shows the test rig in the furnace, for the 1 : 4 scale model of the top end of the connecting-rod. A load of 20 kg was applied by a weight through a pin. The weight was suspended on a wire led down through a concentric "oil passage" in the shaft. From the pin the forces were transmitted to the connecting-rod through the two bearing housings.

When the results came to be evaluated, it soon emerged that the highest stresses did in fact occur in the recess, and the author will therefore confine himself to the stress patterns of the shell section "s" shown later in Fig. 4. Isochromatic photographs of this section are reproduced in Fig. 3; the thickness was gradually reduced in order to ensure optimum accuracy in the extrapolation of the maximum stresses at the edge. The result is shown in Fig. 4, all stresses being referred to the nominal stress $\sigma_N$ prevailing in the shaft. The highest values occur on the top face, in the rounded inner corners of the recess. At these points the stress peaks amount to 1·5 times the nominal stress $\sigma_N$ prevailing in the shaft. The highest values stress amounting to 1·35 times $\sigma_N$ where the recess runs out. It was possible to round the inner corners off with a bigger radius, thus minimizing the stress peaks still further. Since importance was attached to achieving exact replicas of the bearing housings in respect of stiffness, the force application conditions in the test corresponded largely to those of the actual component, for which the external forces were amenable to calculation with a high measure of reliability. Since, moreover, in this test series the stress gradients of importance for the fatigue strength were evaluated and comparable strain gauge measurements carried out on connecting-rods of larger engines were available, it was possible to define the admissible stressing of this component within pretty narrow limits.

On the piston rod the top end flares out into an annular flange, which is bolted to the piston. Here, in contrast to the connecting-rod, however, the conditions under which the forces are transmitted to the flange are known only approximately.

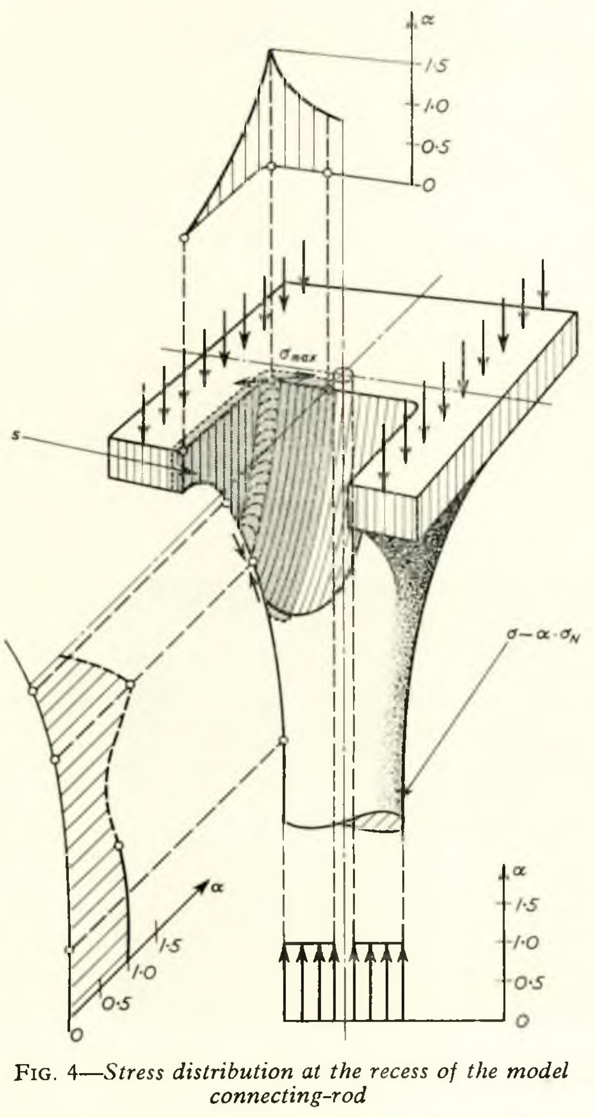



FIG. 4—*Stress distribution at the recess of the model connecting-rod*

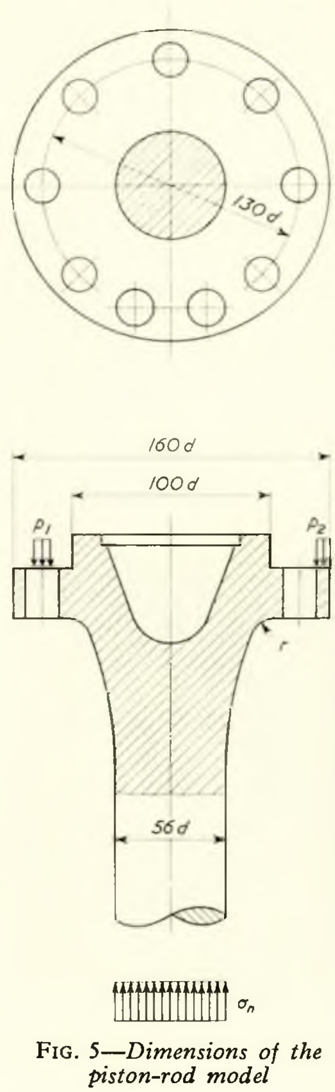



FIG. 5—*Dimensions of the piston-rod model*

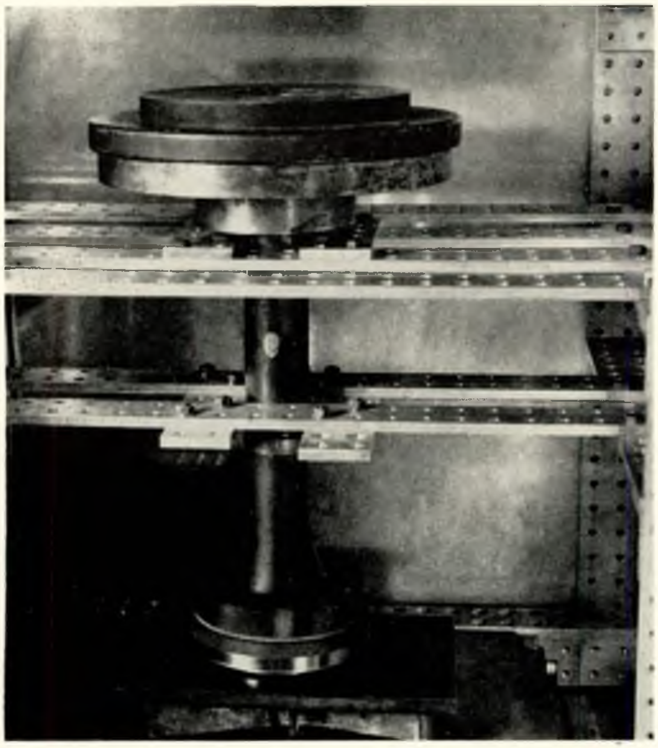


FIG. 6—*Photo-elastic freezing test—Piston-rod model rig in the furnace*

Using a model to a scale of 1 : 2·5, the photo-elastic test yielded piston-rod dimensions as shown in Fig. 5. To measure the stresses occurring under different flange thicknesses $h$ and transition radii $r$ between flange and shaft, several models were investigated. Also investigated was the influence of the point of force application, by transmitting the load through an annular area of 80 mm $< r <$ 72·5 mm (alternative loading I) and 69 mm $< r <$ 61·5 mm (alternative loading II). Although both assumptions are less favourable than actual conditions, they were chosen because strain-gauge measurements were available from the piston rods of RD76 engines, which were based on correspondingly similar loading assumptions.

FIG. 7—*Isochromatic photographic of the longitudinal section disposed at 90° to the plane of symmetry* ($d$=6·8 *mm*)

FIG. 8—*Isochromatic photographs of two longitudinal sections disposed radially between two flange holes* ($d$=4·1 *and* 4·9 *mm respectively*)

Fig. 6 shows the piston-rod model set up in the furnace ready for the test, with the flange at the bottom supported on a ring. The load was applied to the shaft through a point by means of a loosely guided metal cylinder and a ball, in order to obtain a stress distribution of the greatest possible rotational symmetry. The isochromatic photograph reproduced in Fig. 7 shows how well this was attained. It involves a longitudinal section disposed at 90° to the longitudinal symmetry plane. Fig. 8 shows isochromatic photographs of two longitudinal sections disposed radially between two flange holes. Here again the evaluated stresses are referred to the nominal stress occurring in the shaft. The influence of the load take-up is clear for example from the following figures, representing the values

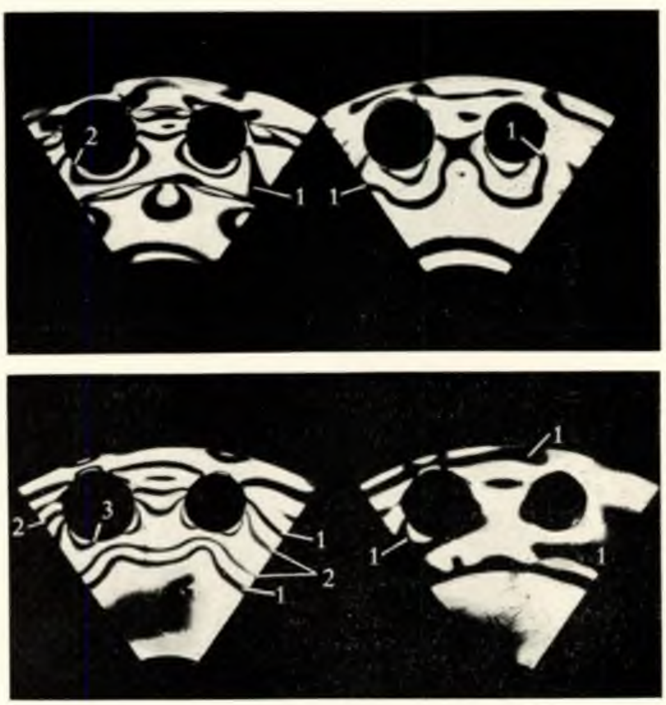


FIG. 9—*Isochromatic photographs of the horizontal sections H1 and H6* ($d$=2·9 *and* 3·4 *mm respectively*)

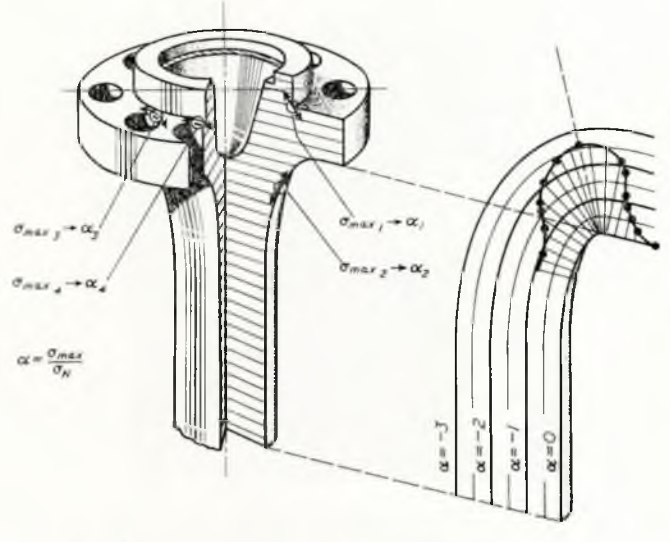



FIG. 10—*Piston-rod model—Position of evaluated stress peaks and behaviour of longitudinal stress in the transition from flange to shaft*

$\left(\alpha = \frac{\sigma_{max}}{\sigma_N}\right)$ obtained at the first dimensioning for the points designated 1 to 4 in Fig. 10:

| | $\alpha_1$ | $\alpha_2$ | $\alpha_3$ | $\alpha_4$ |
|---|---|---|---|---|
| Alternative loading I | 3·8 | −2·45 | 1·65 | 1·65 |
| Alternative loading II | 2·75 | −2·05 | 1·0 | 1·0 |

This will illustrate the difficulties involved in attempting to transfer experimentally ascertained values quantitatively to the actual component in the absence of precise knowledge on the conditions under which the piston bears upon the piston rod yet merely to know what influence the various load take-ups may have on the stress pattern is of value to the designer.

Furthermore, this test series revealed, for instance, that, by slightly increasing the flange thickness and improving the transition from flange to shaft, the form factor $\alpha_1$ can be reduced to 1·6. In addition the flange holes were subjected to close examination. Under alternative loading I the pattern shown in Fig. 11 emerged. Evaluation was performed on six horizontal sections $H_1$ to $H_6$, the positions of which can be seen from Fig. 11. The isochromatic patterns for sections $H_1$ and $H_6$ are reproduced in Fig. 11. Fig. 11 (b) shows the positions of the four radial sections $r_1$ to $r_4$, at which the circumferential stresses of the holes were evaluated. Sections X-X and Y-Y served to determine the stressing across the flange thickness $h$. The maximum stress occurs at the top of the flange along section X-X, where it amounts to about 1·7 times the nominal stress $\sigma_N$ in the shaft. Also recorded was the distribution along the circumference at the height of the sections $H_1$ and $H_6$. It may be imagined that the circumferential stresses acting on the top of the flange deform the holes into ellipses, with their minor axes oriented radially. The radial tensile stresses likewise occurring on the top counteract this deformation, but they are far too weak to compensate the influence of the circumferential stress even approximately. If, moreover, allowance is made at the outside edge for the influence of the load applied there, and for the reverse influence of the negative circumferential and radial stresses on the underside of the flange, then the patterns shown in Fig. 11 will be more understandable.

Since the results of these model tests can be transferred to the actual component only to a limited extent, the last work in this investigation rests with measurement on actual engines.

*Example* 2

Another investigation was devoted to a thorough analysis of the RD engine bedplate. This was prompted by unsuitably designed welded joints in an early series of RD bedplates, plus the desire to find out what performance boost is admissible with present-day bedplates from the strength aspect. While the

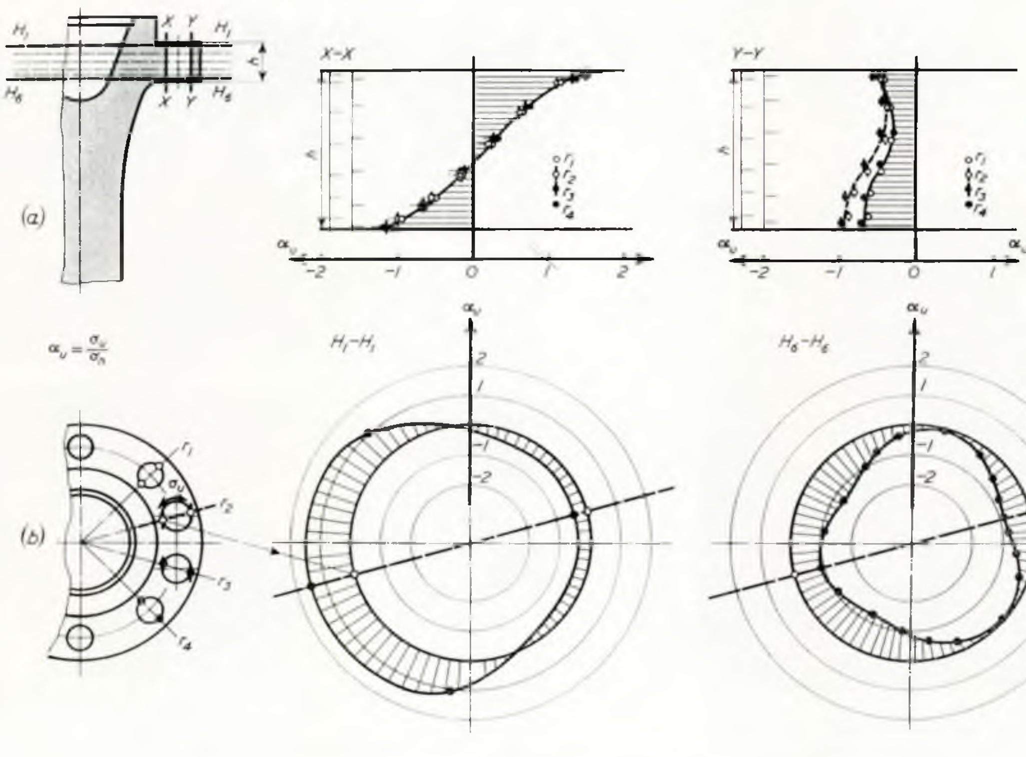



FIG. 11—*Distribution of the circumferential stresses in the flange holes*

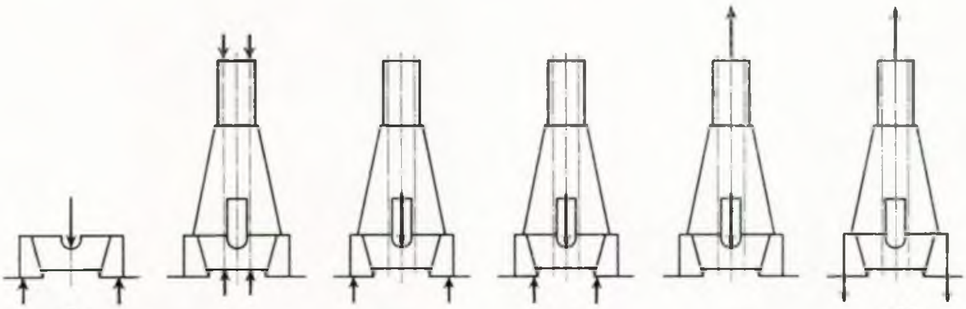



FIG. 12—*Bedplate model tests—Loading situations selected*

stiffness of a bedplate still lends itself to theoretical analysis, any attempt to ascertain quantitatively the distribution of the force flow by calculation is doomed to failure. If the dynamic stress fluctuations occurring at any given point on the bedplate are measured with the engine running, it does not follow that anything would be learnt about how the forces momentarily applied in the component are being distributed.

To make this question better understood it seemed right to first measure the stress patterns in the bedplate of a model, under different loading situations.

If the stiffnesses of the crankshaft and engine block are ignored, it is a simple matter to calculate the time curve for the main bearing forces, from the pressure curve in the cylinder, and the inertia effect of the running gear components for any speed on any engine. Schematically the firing force $P_z$ is transmitted to the cylinder cover at top dead centre (TDC) in a two-stroke engine. The same force also acts on the piston crown; but on its way down through the running gear components into the main bearings it is opposed by the inertia force $P_{M_{TDC}}$, so that finally only the ignition force relieved by the inertia force acts on the two main bearings: $P_{TDC} = P_z - P_{M_{TDC}}$.

At bottom dead centre (BDC) only the inertia force $P_{M_{BDC}}$ acts on the main bearings. This inertia force distributes itself sideways towards the outside in the bearing-saddle cross-member, and is led off downwards through the foundation bolts: $P_{BDC} = P_{M_{BDC}}$.

In order to define the flow of the forces through the bedplate, the basic loading situations shown in Fig. 12 were chosen for the vertically acting forces:

- O: loading at BDC, but ignoring the bracing effect from the columns and bolted connexions;
- I: tie-rod pre-stress;
- II: loading at BDC allowing for the influence of the supporting conditions;
  longitudinal girder supported outside: IIa;
  longitudinal girder supported inside: IIb;
- III: loading at TDC with regard to the forces diverted through the tie-rods;
- IV: it is assumed that the force component applied in the top of the cylinder, which is out of equilibrium with main bearing forces, has to be compensated by the foundation bolts.

*Model*

On account of measuring accuracy the largest model possible should be employed, but, on the other hand, there is a desire to keep it small enough so that the forces to be applied, the weight of the model and the loading apparatus do not become excessive.

Adequate measuring accuracy and small loadings call for an elastic material combining a low *E*-modulus with sufficient strength. For this reason the 1:5 scale model chosen for the RD76 type was welded together from light metal plate.

Since the model had to be made from commercially available plate material, not all the wall thicknesses emerged exactly true to scale. Initially there were difficulties in the welding of the first model M*, the welded joints produced being far from true to scale. The welds did not penetrate the plates and were considerably over-dimensioned.

Only after using copper backing profiles and welding with an air gap could satisfactory penetration be achieved on a second model M**. After subsequent machining of the last weld layers a more true-to-scale transition of the welded joints was finally obtained.

On the other parts, such as columns, cylinder block and tie-rods, attention was directed primarily towards correct reproduction of the stiffness conditions. Their cross-sections were not merely produced to the model scale, but in some cases reduced additionally, to allow for the different elasticity moduli of the material employed. The static compression diagram for tie-rods and engine parts measured on the model shows very close agreement with the diagram intended for the actual engine.

*Loading Rigs*

For loading situation O the loading frame shown in Fig. 13 was used. The bearing load was applied hydraulically. Screwing plates to the end of the bedplate model enabled allowance to be made for the influence of the adjacent bearing-saddle cross-members.

In the other cases, the loads were applied by bracing the parts against a larger frame (Fig. 14). The forces and the tie-rod preload of 4600 kg were measured by means of calibrated,

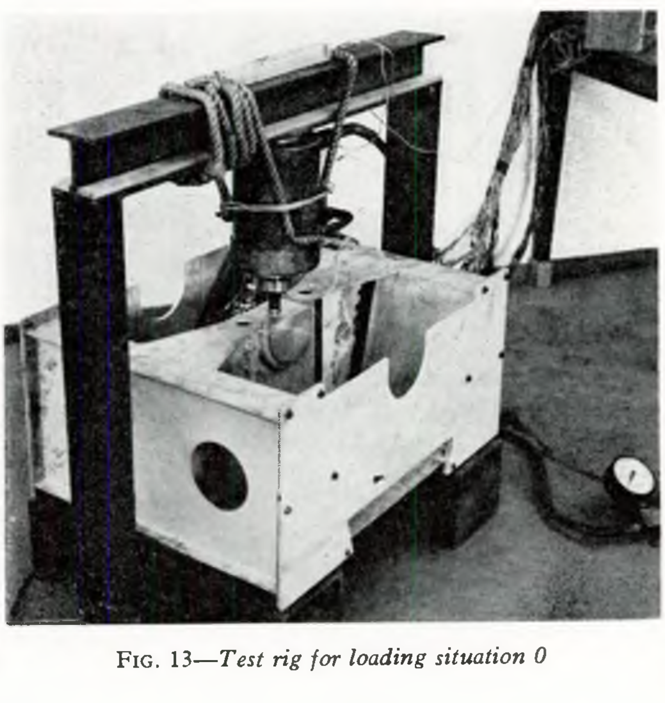


FIG. 13—*Test rig for loading situation 0*

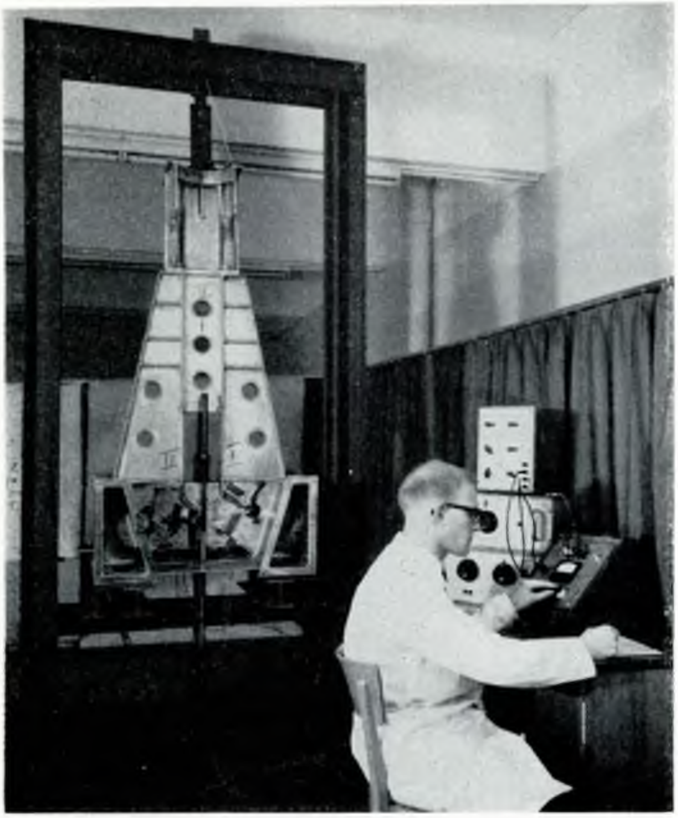


FIG. 14—*Test rig for the bedplate investigations*

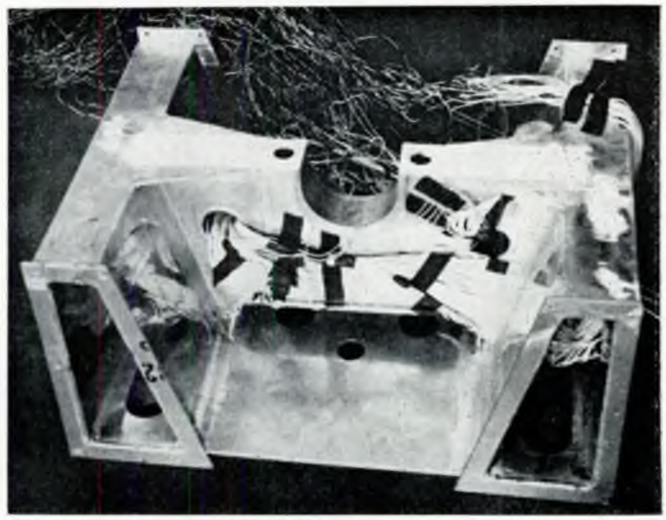
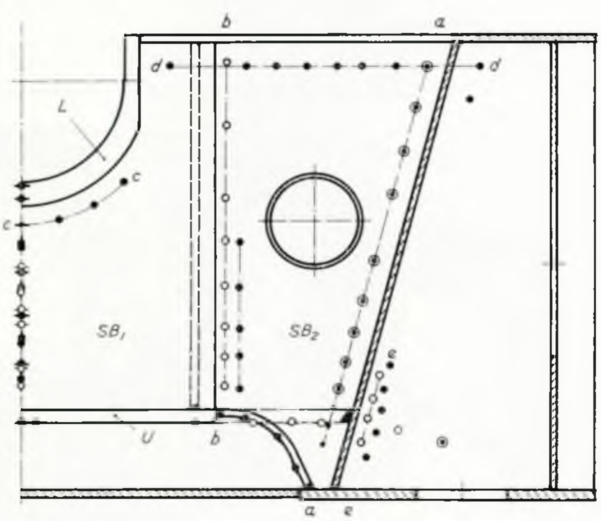


FIG. 16—*Bedplate model M***




○ Measuring points on model M*
● Measuring points on model M**
*U* Bottom flange of bearing-saddle cross-member
*L* Bearing saddle
*SB1* Inner web plate of cross-member
*SB2* Outer web plate of cross-member
a - a to e - e: sections

FIG. 15—*Positions of measuring points and designation of measuring sections*

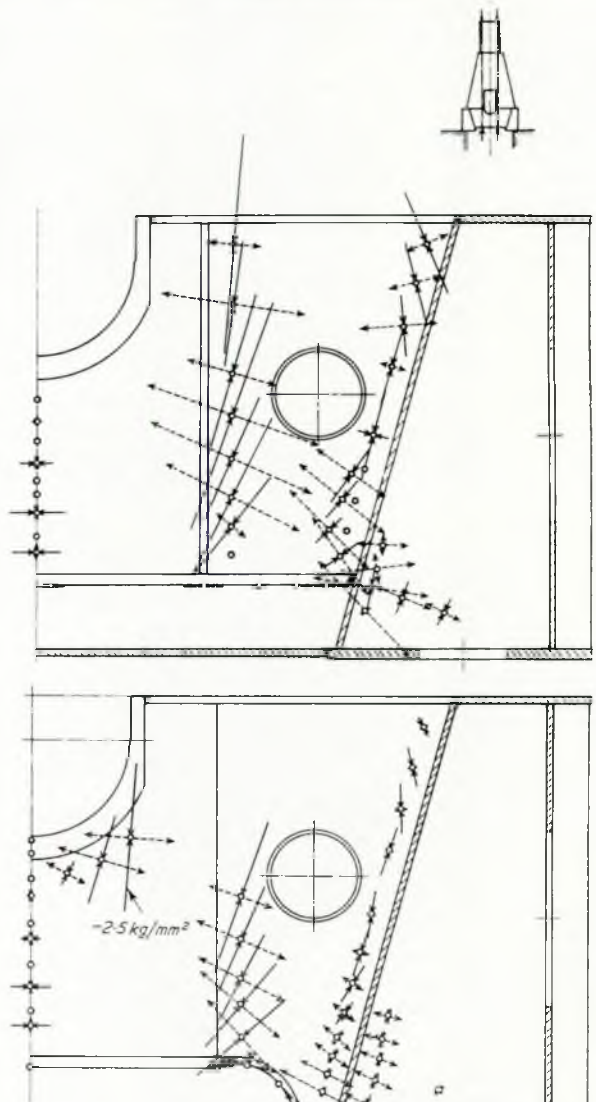



FIG. 17—*Distribution of stresses on models* M* *and M**—Loading situation I* (*tie-rod preload*)

bending and temperature-compensated strain gauge dynamometers, and monitored continuously during the test.

Despite initial misgivings, this loading system had acquitted itself well. The forces were transmitted to the main bearing by a ram, the diameter of which was fixed so that support conditions between crankshaft and bearing saddle, similar to those on the actual engine, were obtained under the maximum bearing load of 4000 kg chosen for the test. Moreover, different supports for the longitudinal girders were investigated, in order to clarify the conditions here also.

*Measuring Technique and Evaluation*

At all points where the principal stress directions were not known in advance, strain gauges in three directions (rosettes) were fixed. With the object of eliminating the effect of the stress gradients as far as possible, strain gauges with small measuring lengths were chosen. On the Budd rosettes employed, the three rosette measuring strips were accommodated on an area of 5·9 mm × 5·9 mm. In symmetrical sections and along the bottom flanges of the bearing-saddle cross-members, Hottinger strain gauges with 6-mm measuring length were used. To enhance the reliability of the investigation results, measurement was carried out at every deformation point, to the following procedure: zero measurement—measurement at 50, 75 and 100 per cent of maximum load—zero measurement. The measured results were then plotted against the loading for each point and linearized. From the strains ascertained in this manner the principal stresses $\sigma_1$ and $\sigma_2$ were calculated, and for the rosettes the principal stress directions also. Fig. 15 illustrates the position of the measuring points and the designation of the measuring sections. On the first model M*, 14 strain gauges and 26 rosettes were fixed. After measuring-out the loading situations sketched in Fig. 12, the facts obtained were checked on model M** with downswept bottom flange on the bearing-saddle cross-member (Fig. 16). Measurements were recorded from 20 strain gauges and 34 rosettes.

*Measured Results*

The stress patterns in both models for loading situations I (tie-rod preloading), IIa (BDC) and III (TDC) are indicated in Figs. 17 to 19.

*Situation I: Tie-rod Preloading*

In the section b-b, both models clearly show the pressure

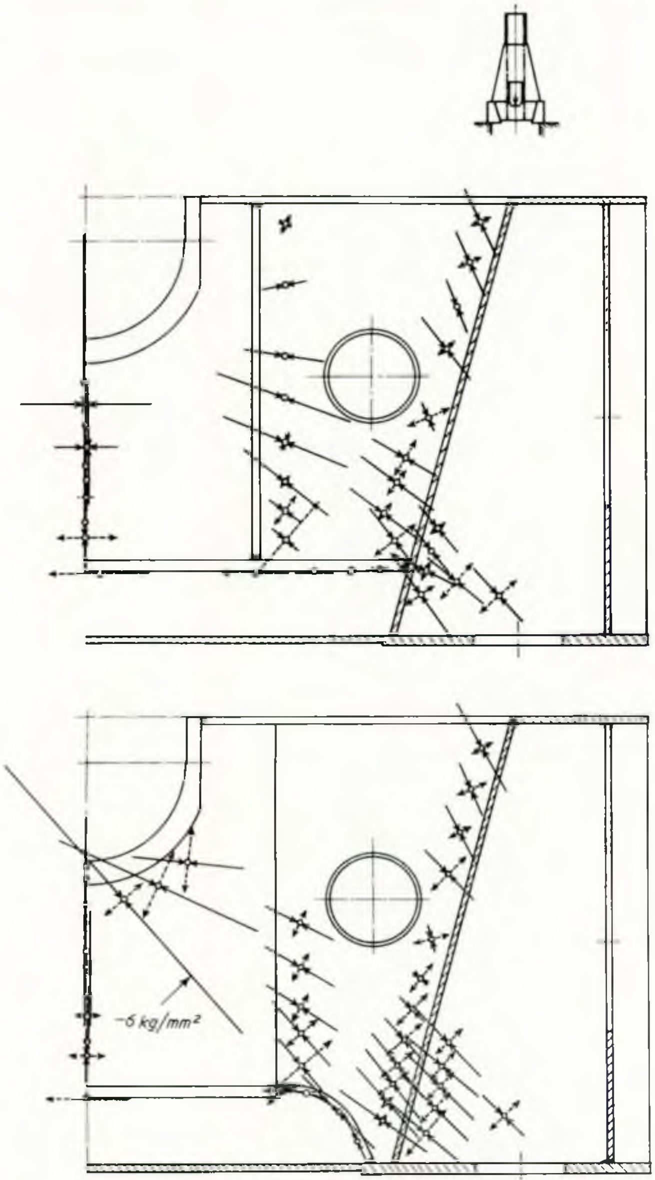



FIG. 18—*Distribution of stresses on models M* and M**—Loading situation IIa (BDC)*

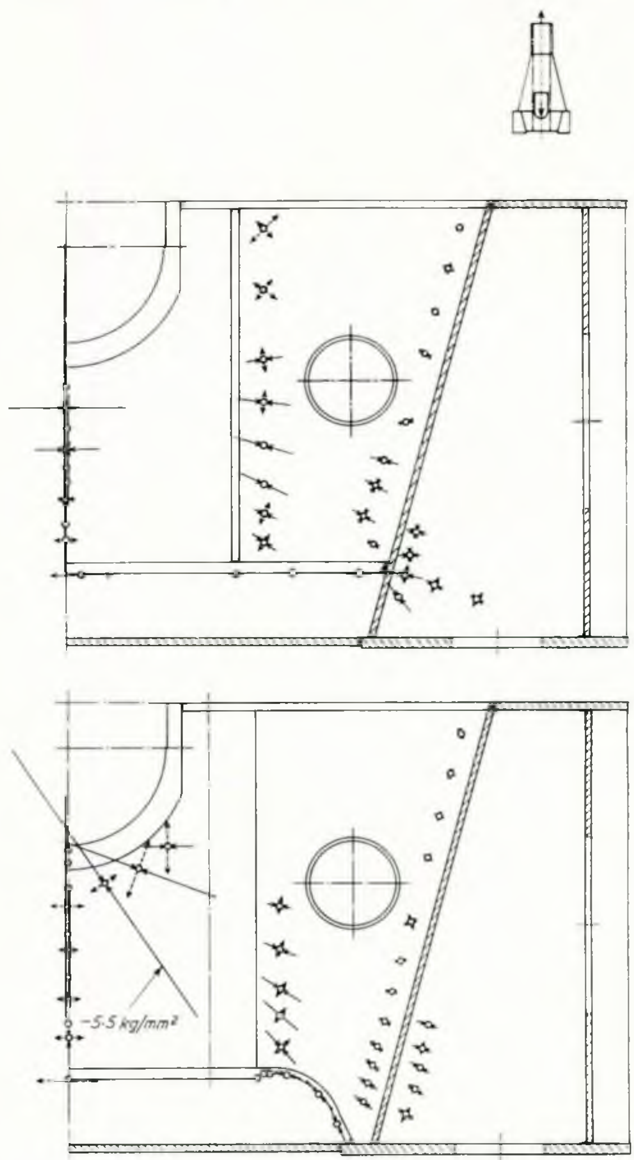



FIG. 19—*Distribution of stresses on models M* and M**—Loading situation III (TDC)*

cones underneath the preloaded tie-rods (see Fig. 15). Under this load the bottom flange moves upwards, setting up tensile stresses in the transistion from bottom flange to longitudinal girder. On model M* these stresses are transmitted primarily through the bottom flange, so that the tensile stresses in the measuring section e-e point to this transition. On model M** the tensile stress is transmitted through the web plates, resulting in a more regular pattern. In section a-a the patterns in the top part correspond only qualitatively. The disparities can be traced to differing support conditions between column and bedplate, which were compensated in later investigations.

*Situation IIa*: *BDC*

These "dynamic" stresses under loading at BDC are virtually identical on both models in the sections b-b and e-e, and are hardly affected by the different bottom flange shapes. The biggest compression stresses occur under the bearing saddle.

*Situation III*: *TDC*

Both model tests demonstrate clearly that the forces diverted through the tie-rods hardly influence the measuring section a-a. The forces applied have the effect of relieving the stressed parts, so that the tensile stresses have to be measured in the vertical direction.

Also revealing are the stress patterns along the bottom flange of the bearing-saddle cross-member (see Figs. 20 and 21). Comparison of load situations IIa and IIb on model M* shows that in the latter case the stress at the middle of the cross-member becomes less as the supporting width is reduced, though at the transition from bottom flange to longitudinal girder the stress deflexions are about 30 per cent higher, due to the sharper force diversion. Another important fact for the designer is the knowledge that the welds in the bottom flange on model M** are only weakly loaded.

To enable the accuracy and reliability of the resultant stress values to be judged, several checks were applied. Thus, for example, the normal stresses operative in section d-d were calculated from the main stresses ascertained, and these were regarded as an extraneous load acting on the bedplate. From this it was then possible to determine the transverse force curves for the bedplate set out in Fig. 22. The transverse forces operative in section b-b emerged as:

Loading situation I : transverse force = +1640 kg
Loading situation IIa : transverse force = −750 kg
Loading situation IIb : transverse force = −850 kg
Loading situation III : transverse force = −(380 ÷ 500) kg

On the other hand the pattern of the shear stresses can be determined in analogous fashion from the measurements in the section b-b, and by integration from the mean value a trans-

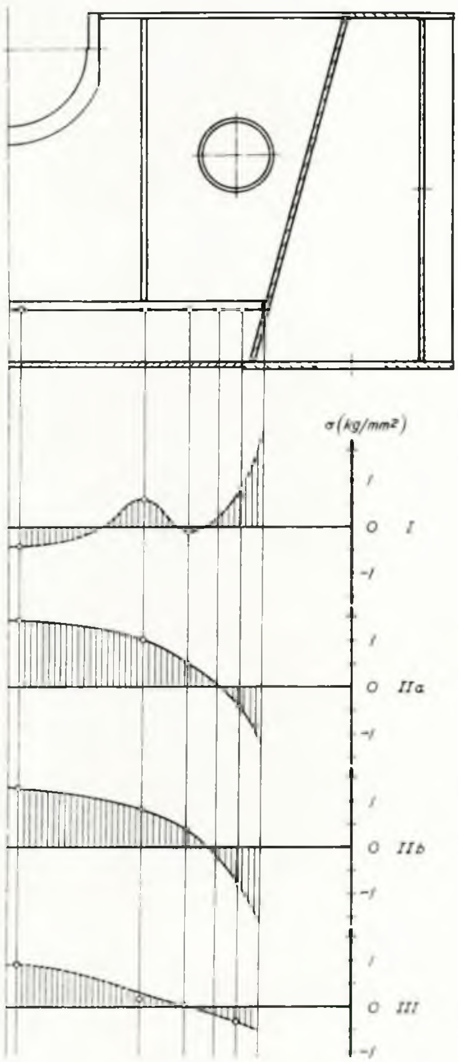



FIG. 20—*Stress patterns along the bottom flange of model M*—Loading situations I, IIa, IIb and III*

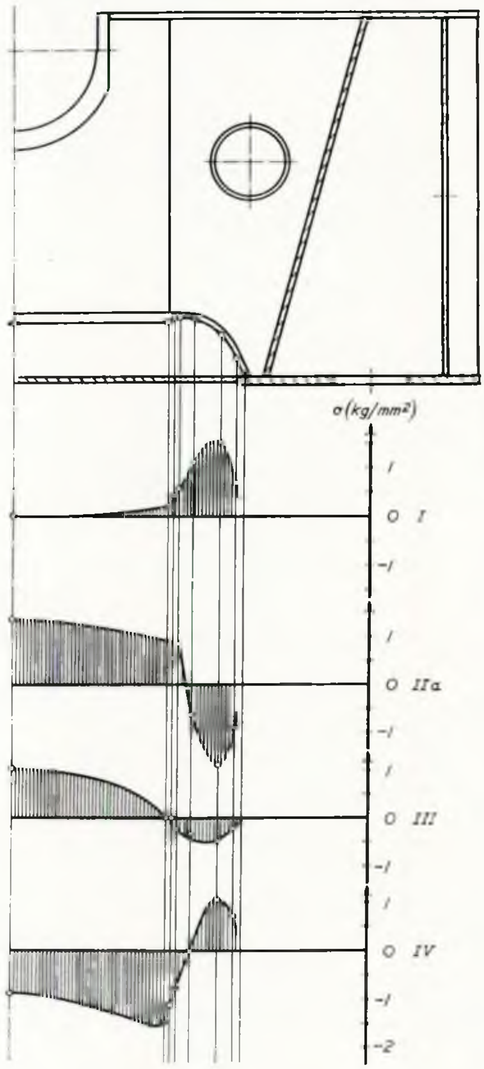



FIG. 21—*Stress patterns along the bottom flange of model M**—Loading situations I, IIa, III and IV*

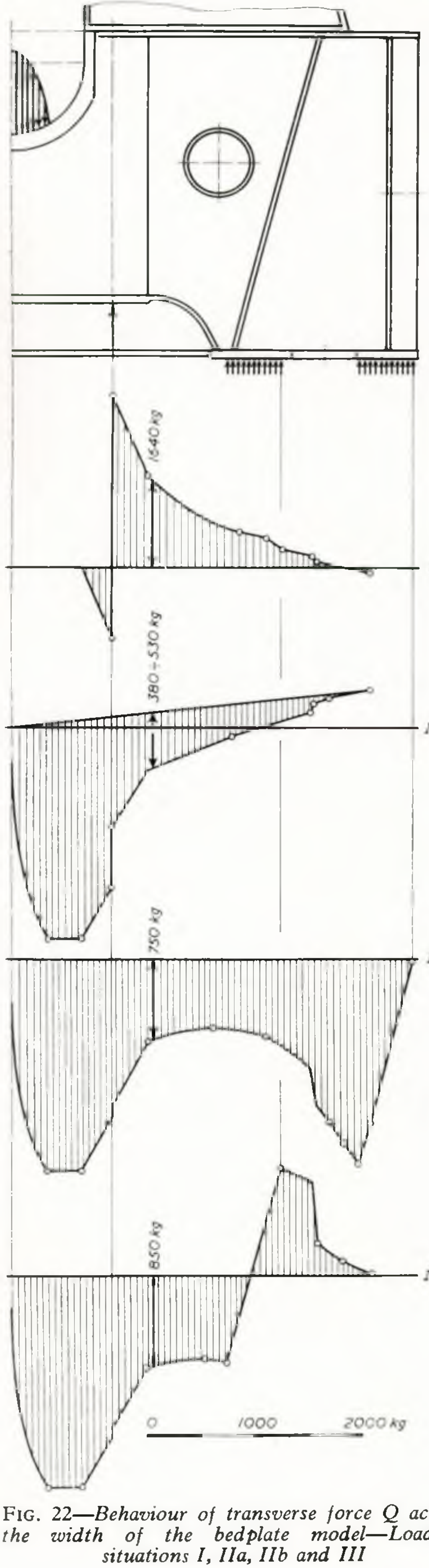



FIG. 22—*Behaviour of transverse force Q across the width of the bedplate model—Loading situations I, IIa, IIb and III*

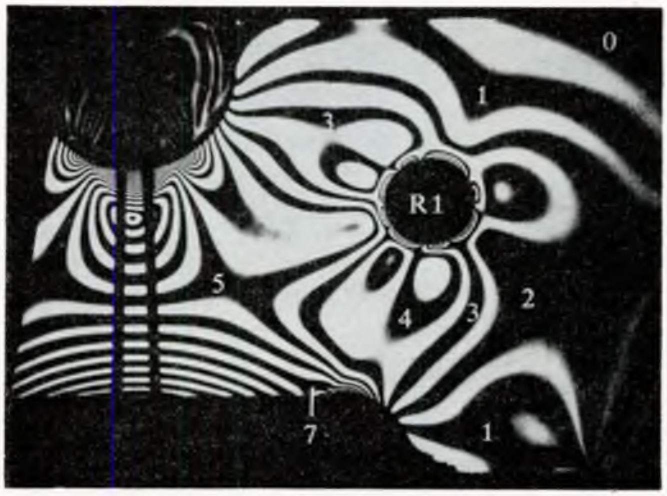



FIG. 23—*Two-dimensional photo-elastic model test for the bedplate—Isochromatic distribution of the stiffening tubes in the web plates of the longitudinal girders*

verse force is derived which ought to correspond with the foregoing figures:

Loading situation I : transverse force = +1780 kg
Loading situation IIa : transverse force = − 705 kg
Loading situation IIb : transverse force = − 825 kg

Situation III was not verified further, since, as will be seen from Fig. 22, the transverse force no longer returns to zero outside.

As these discrepancies were without significance for the continued investigations, their cause was not examined further. Nevertheless the author considers that the foregoing figures prove how valuably such measurements are able to assist the designer both qualitatively and quantitatively.

Once again, in this investigation also, supplementary photo-elastic tests were performed. The shape of the transition from bearing saddle to web plates was analysed by three-dimensional tests, while the influence of the stiffening tubes in the longitudinal girders was established on two-dimensional models (Figs. 23 and 24).

The conclusions drawn from such model measurements require verification on the actual engine. Thus, for example, on a 9RD76 engine with a bedplate corresponding to model M*, certain measurements from the model tests were verified. The positions are shown in Fig. 25 for the bearing-saddle cross-member, and in Fig. 26 for the web plate of the longi-

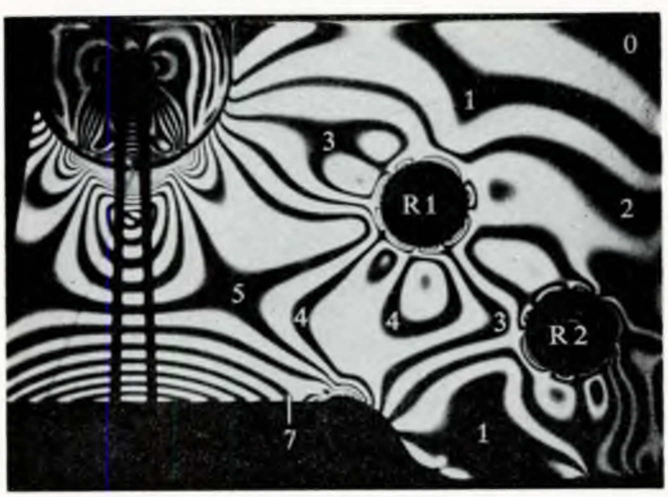



FIG. 24—*Two-dimensional photo-elastic model test for the bedplate—Isochromatic distribution allowing for the stiffening tubes in the web plates of the longitudinal girders*

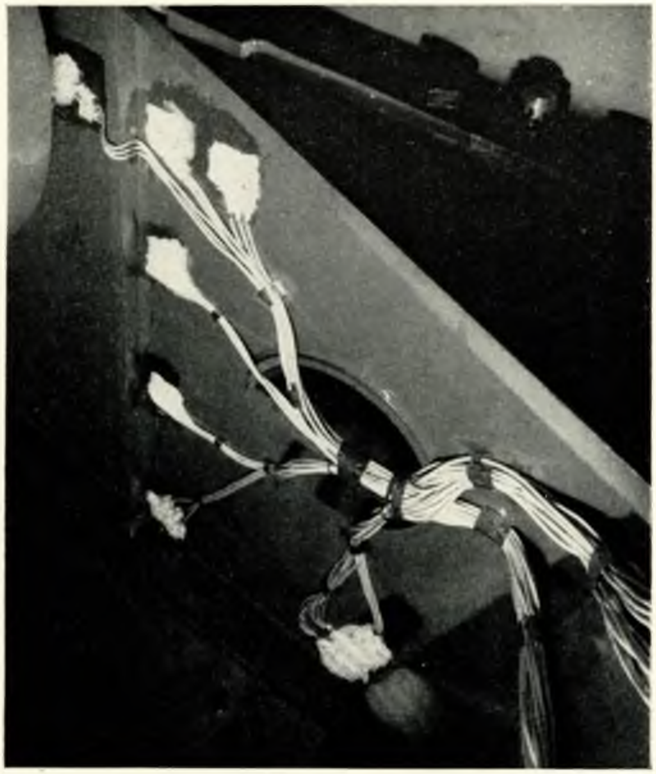


**Fig. 25**—*Strain gauge measurements on a 9RD76 engine: position measuring points on the bearing-saddle cross-member between cylinders numbers 6 and 7*

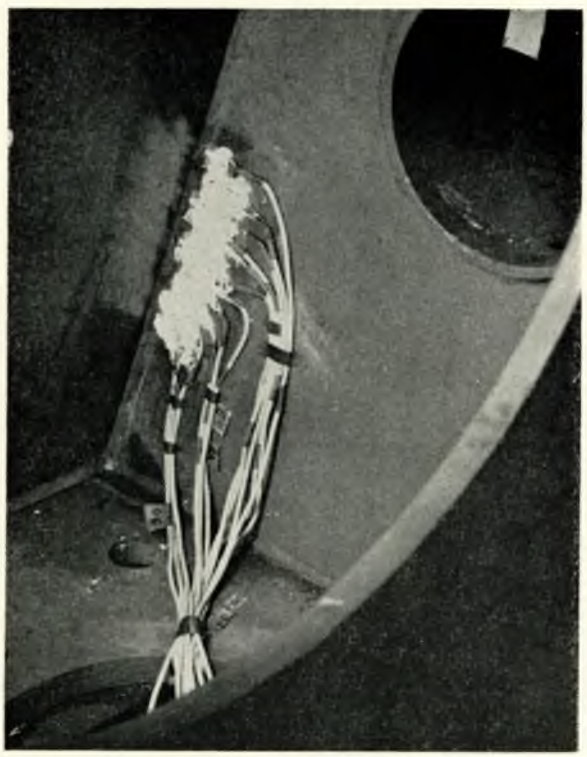


**Fig. 26**—*Strain gauge measurements on a 9RD76 engine: location of measuring points on a web plate in the longitudinal girder*

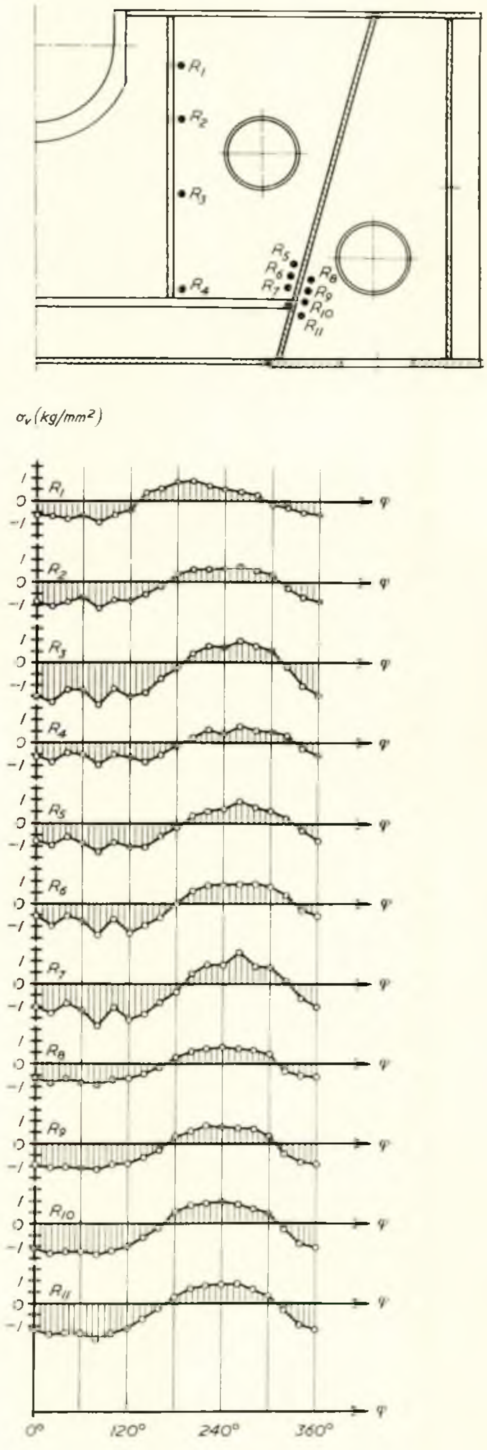



**Fig. 27**—*Strain gauge measurements on a 9RD76 engine — Working stress $\sigma_v = (\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2)^{0.5}$ through one complete revolution at 119 rev/min and 76 kg/cm² ignition pressure*

FIG. 28—*Welding sample for fatigue tests*

tudinal girder. The measurements were recorded on a cross-member at which the maximum inertia effects occurred owing to the minimum displacement between the two cranks affecting the cross-member. Measurement of the principal stresses and their directions was effected by means of rosettes. The working stresses calculated from them with the equation:

$$\sigma_v = (\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2)^{0.5}$$

are plotted in Fig 27 through one complete revolution.

In addition, strain gauge measurements were also carried out on the bedplate of an 8RD90 engine corresponding to model M**, likewise during operation. Thus, at the weld in the transition from the thick to the thin bottom flange plate, a stress deflexion of $\pm 0.7$ kg/mm² was measured, while at the second weld in the bottom flanges at the transition to the bottom plate only $\pm 0.4$ kg/mm² was recorded.

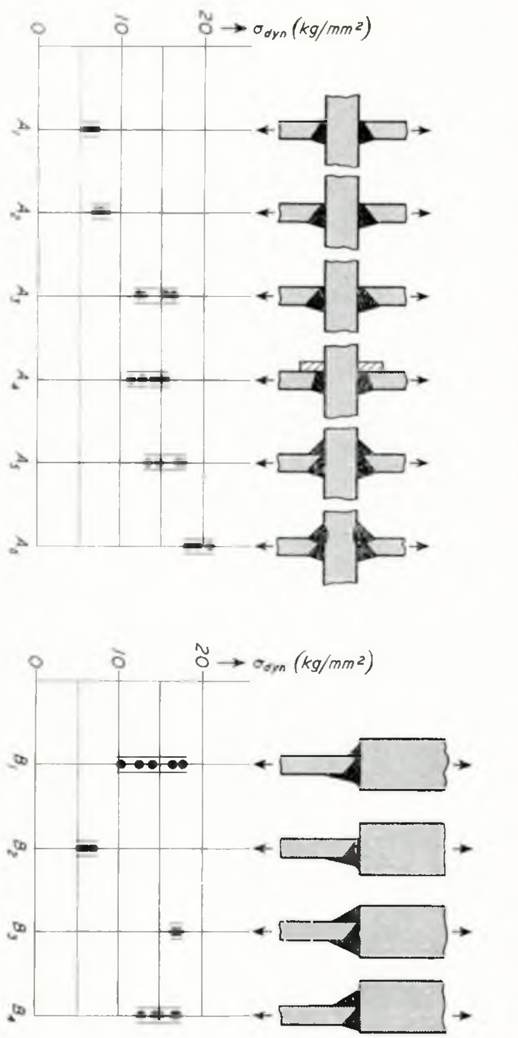



**FIG. 29—*Comparison of welded joints examined and tensile fatigue limits $\sigma_{dyn}$ ($\sigma_{dyn}$ = pulsating stress amplitude as nominal stress in the plate)***

Now that the values obtained from model tests have been checked and largely confirmed by full-scale tests, the stresses occurring in the bedplate may be known, but it is still necessary to establish what stresses are admissible. To this end, various welds were tested to ascertain their admissible dynamic stress deflexion $\sigma$ dyn ($\sigma$ dyn = fatigue limit, pulsating stress amplitude from zero to a positive maximum as nominal stress in the plate).

First, box-shaped structures 750 mm long were produced with a configuration similar to the bedplate, under conditions as encountered in the workshop. From these structures, of similar construction, samples were produced as shown in Fig. 28. The welds illustrated in Fig. 29 were investigated in connexion with the transition from the bearing-saddle cross-member to the longitudinal girder.

| | |
|---|---|
| Sample A1: | V-groove with 35° enclosed angle. |
| Sample A2: | V-groove with 40° enclosed angle. |
| Sample A3: | V-groove with 40° enclosed angle and 2-mm air gap between the plates to be welded. |
| Sample A4: | V-groove with 20° enclosed angle and 4-mm air gap, welded against plate strips. |
| Sample A5: | Sample A2 with root reinforcement. |
| Sample A6: | Sample A2 with the V-groove root widened by the Arcair process prior to reinforcement. |

The boiler plates, 12 mm thick, were welded with low hydrogen electrodes of 3·25-mm diameter.

In order to keep the size of the investigation within reasonable limits, the tensile fatigue strengths $\sigma$ dyn were determined by the staircase method described by Locati[3]; this enables the fatigue strength of a component to be ascertained from a single test piece by assuming that the damage sustained by it bears a linear relation to the number of load cycles (Miner's rule). This method has been used many times, revealing an accuracy which should fully satisfy the practical engineer.

To obtain an idea of the scattering of the measured values, three to five samples were examined for each different weld. From the results set out in Fig. 29 it is at once clear that the stresses occurring in the engine are well below the tensile fatigue limits ascertained.

Samples A1 and A2 possess similar fatigue strength. By contrast samples A3 and A4, welded with an air gap, as well as the reinforced sample A5, are characterized by a fatigue limit approximately double. The grooving of samples A6 before reinforcement raises the strength by a further 5 kg/mm². It should be noted that under bending stress the two sample welds A5 and A6 ought to display the same fatigue strength. Since all samples were stress-free annealed, the internal welding stresses may be largely ignored, more so as, with the high notch effect of the root, a static stress component would have only a negligible effect on the dynamic strength values. Analogous tests were made with sample B, to investigate the transition between two plates of unequal thickness, as for example on the bearing-saddle cross-members between the web plates SB1 and SB2 in Fig. 15.

If the problem is viewed purely from the aspect of fatigue strength, samples A1 to A4 have a similar notch radius on the root side. Only the non-penetration depth differs. When the foregoing test values are plotted against the penetration deficit $x$, measured on sectioned samples, a clear relation emerges between the penetration deficit and the fatigue strength (see Fig. 30).

The B samples also yield a corresponding curve. The constancy and the low scatter of the values plotted in this manner likewise confirm the accuracy and reliability of the Locati method.

Here again, a series of plane photo-elastic tests was carried out to determine the influence of deficient penetration and misalignment of the plates on the A samples. As shown in

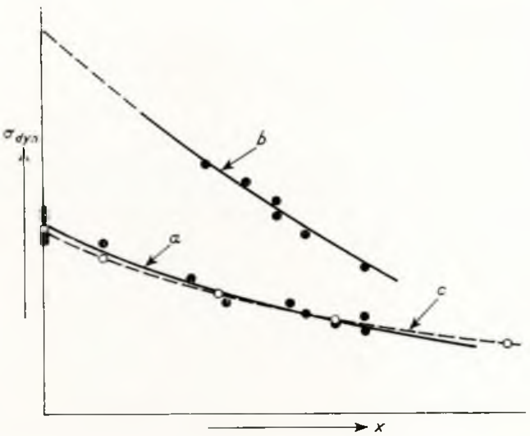



FIG. 30—*Tensile fatigue limit* $\sigma_{dyn}$ *of samples A1 to A4 (curve a) and samples B1 to B4 (curve b) against the penetration deficit x—Curve c was taken from a plane photo-elastic investigation into the influence of deficient penetration on the A samples*

Fig. 30, the photo-elasticity curve agrees closely with the fatigue tests. This in turn proves that practical usable results are yielded by this model method. It was also demonstrated that, provided the plates of a cross-beam joint are mutually displaced by not more than 25 per cent of their thickness, the influence of misalignment on the fatigue strength can be ignored in practice.

With these tests on welded joints the bedplate investigations were concluded. The procedure for strength investigations as sketched was followed, for the questions involved were answered by means of model tests and the correctness of the results obtained in this way was verified on the actual engine. To enable the operational reliability of the component to be judged as well, the stressings thus established were compared with the admissible stresses determined from fatigue tests.

REFERENCES


1) FROCHT, M. 1960. "Photoelasticity". John Wiley and Sons, New York.

2) HETENYI, M. 1954. "Handbook of Experimental Stress-Analysis". John Wiley and Sons, New York.

3) "Application de la méthode de L. Locati au contrôle de la limite de fatigue de pièces d'automobiles". *Revue de Métallurgie,* April 1960.

# Written Discussion

MR. T. R. GURNEY wrote that the paper was of considerable interest because it demonstrated the method of designing a complicated structure which was subjected to fatigue loading. Although it was preferable to carry out a strain-gauge analysis on a full-scale prototype of the structure whenever possible, the use of large-scale models was perfectly acceptable and was certainly better than carrying out no such analysis. More and more designers were coming to realize that this was the correct starting point and that it should be followed up by fatigue tests on critical regions of the structure.

The fatigue tests carried out in this particular investigation were, however, surprising. In the first place, the use of the Locati method in preference to normal fatigue testing seemed most unfortunate. The Locati method had been developed, originally, as a method of quality control and as such it was no doubt quite useful. However, it was not a satisfactory method for use in research* since very large errors (at least $\pm 50$ per cent) could result from selecting an incorrect basic $S$-$N$ curve or if the value of $\frac{n}{N}$ differed significantly from unity, which it could easily do. The investigation would have been more valuable if normal fatigue testing procedures had been used, particularly as in several cases sufficient specimens had been tested to establish the slope of the $S$-$N$ curve.

In any case, even if the results obtained were reasonably accurate, it must be remembered that they represented the fatigue strength at $2 \times 10^6$ cycles, which, for welded joints, was certainly not, as the author assumed, the fatigue limit. Indeed it had been shown that the fatigue limit was a very much lower stress and, in fact, there had been several failures which had occurred in service at considerably longer lives than $2 \times 10^6$ cycles. Use of these stresses for design purposes would therefore be most unwise.

It would be interesting to know the mode of failure associated with the various types of specimen which had been tested. He assumed that series A1-A3 failed from the weld root, series A4 from the junction of the weld with the backing bar and series A5 and A6 from the weld toe. It would also be interesting to know what the author defined as lack of penetration, particularly in the B type specimens. Fig. 29 showed all the specimens as being made with full penetration welds and he wondered if the author meant the difference in thickness between the two plates. Guessing the answer to the question was made impossible by the fact that no scales were given in Fig. 30.

Incidentally it seemed extraordinary, if the sketches of the B type specimens shown in Fig. 29 were to scale, that the use of such a joint should even have been contemplated in a structure subject to fatigue loading. Chamfering of the thicker plate, so that the weld was made between plates of equal thickness, would have been more satisfactory.

MR. A. R. HINSON (Associate Member) wrote that the investigations conducted by the author into the stress levels occurring in the connecting and piston rods indicated that the stresses and their gradients were sufficiently low to be in keeping with what was usually considered good design.

Unfortunately, while stress was in many cases an acceptable criterion of design, there was an objection to its sole use for connecting-rod forks.

The connecting-rod fork carried the crosshead bearing assembly which was notoriously the Achilles heel of the large slow-running Diesel engine. Satisfactory operation of the crosshead bearing depended largely on satisfactory alignment under load and alignment under load depended on the relative stiffnesses of the journal/bearing assemblies.

Even if stress levels were adequate, if the stiffnesses were different the design was suspect. It was not sufficient that the stiffnesses were equal, i.e. that, under load, any deflexion in the crosshead and journals should equal that in the bearings; the sense of the stiffnesses should also be the same. Deflexions would then have the same magnitude and direction, and alignment would be preserved.

An indication that the journal/bearing stiffnesses were not exactly matched could be obtained by examination of the wear pattern of the bearings and journals. If the majority of the wear occurred at the inner ends of the bearings, i.e. near the crosshead, the relative stiffnesses were such that the journal assembly sagged and the bearing assembly hogged.

It would be interesting if the author would give a typical wear pattern for the RD crosshead bearing and relate it to the stress levels and stiffness of the connecting-rod fork.

Examination of the records of Lloyd's Register of Shipping indicated that the operation of RD crosshead bearings compared favourably with other designs. It was probable that the inherent stiffness of the fork contributed significantly to this fact.

With reference to the stress analysis of the RD engine bedplate, prompted in part by unsuitably designed welded joints in an early series of bedplates, the value of the paper would have been enhanced if a diagram had been given showing the exact location of the cracks together with an explanation of the mechanism of failure, i.e. direction and rate of propagation and the methods by which the defect had been eliminated.

Fig. 12, I, II a and II b, and Figs. 17 and 18 did not prove that the tie-rods were taking their fair share of the load; but then, it was possibly not the author's intention that they should. Nevertheless, it was desirable to know quantitatively how much value the tie-rods were in reducing the bedplate stresses.

For half the load to be carried by the tie-rods, the stiffnesses of the tie-rods and transverse beam should be equal over the load range.

The geometry of the tie-rods was relatively simple and the load/deflexion curve could be calculated and would be linear within the elastic limit. For the bedplate, the same quantity could be determined experimentally using Fig. 12, 0, and measuring the deflexion at the points where the tie-rod nuts bore. The curves should, of course, be identical for equal load sharing.

Some evidence should be given to show that the stresses recorded approached the maxima which occurred in the structure. Where possible assurance should be given that the

*Gurney, T. R., and Chapman, G. "A Theoretical Analysis of the Locati Test Method and Some Exploratory Fatigue Tests". International Institute of Welding, XIII-271-61.

active portions of the strain gauges were located in the same relative positions as the cracks in the original bedplates. If the cracks originated in the welds and propagated into the plate, then the gauges should have been applied to the welds. However, even if these precautions were taken, a stress lower than actual would be recorded for the reasons the author himself gave in his masterly introduction.

This disadvantage could be mitigated to some extent by a re-distribution of the strain gauges. They were generally located parallel to welds, e.g. a-a; b-b; c-c; d-d; e-e; Fig. 15. Such unidirectional coverage had the disadvantage that the stress gradients, from the weld across the plate, were not recorded. Two-dimensional coverage was just as simple to effect. If both stress and stress-gradient were known, extrapolation became probably more accurate than when the stress only was recorded and gradients were estimated from photo-elastic models.

This was illustrated by Figs. 20 and 21 where the elimination of the stress raiser at the bottom flange of M* by the incorporation of the radius as in M** effected a striking improvement. The relatively high stress in the corner of the bottom flange of M* had not only been removed, but the sign of the stress gradient had been reversed.

Mr. M. Langballe (Member) wrote that detailed studies into the mechanical strength of engine structures were not often published by engine designers, although considerable research into such problems was no doubt carried out. This informative paper was therefore to be welcomed by those who had to consider the strength of Diesel engine components. All too often, engines were put into service, which, after some time, exhibited weaknesses and damage with considerable inconvenience to the owners. At that stage quite a lot of research might have to be undertaken, together with a trial and error process on the spot, and patching-up. This might be avoided to a large extent by an approach of attack and laboratory research, as proposed by the author and largely motivated by previous experience, design considerations and results of more or less elaborate precalculations at the design stage and, eventually, by dynamic strain measurements on a prototype engine. Model tests were, however, expensive and time consuming and, with this in view, theoretical methods should not be dismissed for the strength analysis of complicated fabricated Diesel engine structures at the design stage. Potentially powerful in this respect were finite element methods for solution on a high-speed digital computer.

In investigations of cracked engine structures, he had often been puzzled by the fairly low working stresses which might give rise to cracks in welded engine structures. Comparing the results of the dynamic stress measurements shown in Fig. 27 with the fatigue strengths of the welded specimens shown in Fig. 29, the weld along the section b-b (Fig. 15) should have a sufficient nominal safety. As far as he knew, however, difficulties with cracking around this weld had occurred, in which case the reason might have been a more severe lack of weld penetration than those investigated. Could the author comment on this?

Unlike most other makers of large engines, who had retained the cast-steel insert for the main bearing and tie-rod support in their welded bedplates, Sulzer seemed to be in favour of the all-welded design. This obviously put heavy demands on the proper design and workmanship of highly stressed welds of difficult accessibility. Some recommendations on what to accept in respect of lack of penetration would be most useful and he would, therefore, also ask the author to supply Fig. 30 with numbered scales and, if possible, a sketch showing the geometry of the weld roots of the specimens.

The assessment of stresses at sharp notches in terms of fatigue limit, which must have been made by the author in investigating the effect of lack of penetration, was always difficult. The stress gradient, as well as the peak stresses, had to be taken into consideration and a considerable idealization of the geometry of the weld root on photo-elastic models must have been made. This being of considerable general interest, he would like to know how the author had tackled this and arrived at results which matched the fatigue test results so well.

Regarding the model tests with bedplates and columns, it would be very useful if the author could give the dimensions and design particulars of these models, as well as the magnitude of the loads in each case. The dynamic stressing (Fig. 27) in the points of measurement was obviously due to inertia forces, which presumably had a considerable horizontal component. The cases of model loading were all vertical and the model stresses were therefore, in his opinion, hardly comparable with the measured dynamic working stresses as given in Fig. 27 unless an analysis of the separate components of the working stresses was carried out. A case of horizontal loading on the models should therefore be included, in order to predict the working stresses at the positions of dynamic measurements. Predictably, and also indicated by Fig. 18, the stresses in the web plate beneath the bearing-saddle were fairly high, although in compression. Had the working stresses been determined in this region, either by dynamic measurements or the model tests, and compared with the corresponding fatigue limits of the welds involved?

In some cases on other makes of engine, the horizontal components of the inertia forces had been a source of nuisance. If the transverse stiffness of the entire bedplate was not adequate, a part of the horizontal inertia force had to be transmitted from the bedplate to the engine seating, which had led to wear of the chocks and slackening of the holding-down bolts. It would be of interest to learn whether the author had had any similar experience and whether this had been considered in the design of the bedplate.

Three-dimensional photo-elastic methods were very effective in determining accurately the stress distributions in design details and the author's work on piston-rod models was impressive. The method was, however, expensive, as only one configuration could be tested with each model. In investigating crankshaft stresses, he had found that a combination with strain-gauge measurements on aluminium models had certain advantages. The latter were cheap and might readily be re-shaped to other variations of details.

# Author's Reply

In his reply to the discussion, the author wrote that concerning Mr. Gurney's comments with regard to the methods of carrying out fatigue tests, in particular the Locati method, their opinions differed considerably. The merits of this method had been appreciated by several investigators*† much more favourably than by Mr. Gurney and Mr. Chapman in their original paper. At Sulzers it was believed that the Locati method, if one was well aware of its weaknesses, did offer very good services, in a case just like that in the example. The procedure of fatigue testing could be different, according to the aim of the test series.

If one did not apply special laboratory methods for the fabrication of the test specimens, it was hardly possible to achieve test pieces with identical weld defects. It would therefore not have been possible to assess the slope of the S-N curve with, for instance, six test samples as suggested by Mr. Gurney, as each sample had a somewhat different lack of penetration. This would result in a different fatigue limit, giving a broad scatter at cycles less than $10^6$ and hence the impossibility of defining the slope accurately enough, even although all samples were taken from one and the same plate welded by the same welder under apparently identical welding conditions. To use "normal fatigue testing procedure" would have meant the fatigue testing of some thirty specimens for each joint configuration. Using the Locati method, it was quite agreed that the S-N curve must be correctly ascertained if one wanted reliable results, but this was duly considered in the investigations. As to how "unwise" it was to use the stress figures given for design purposes, the author would like to point out that these were cross-checked and fitted quite well with experience gained with fillet welds which failed under known service conditions and which were statistically analysed.

Generally speaking it was well known to Sulzers that, in practice, fatigue failures were detected or really did happen at what one might think to be a surprisingly high number of stressing cycles, and, if the company felt that at least $10^8$ cycles had to be chosen as a relevant figure, then this was due to the fact that one or more of the following points would influence the problem:

a) under variable service conditions, depending on the load factor of the engine considered, cumulative fatigue must be investigated;

b) when operating within the endurance range, i.e. when running with a safety factor equal to one, a very high scatter in time for failures must be expected;

c) on joints with high stress raisers and low nominal stresses there would be a considerable time delay between the moment of crack initiation and the time of detection of the developed crack;

d) the influence of corrosion might also have to be considered.

Concerning the mode of failure the author referred to

*de Leiris, H., and Cazaud, R. "Evaluation de la Limite de Fatigue des Structures soudées en Acier à l'Aide d'une seule Epreuvette ou Elément d'Essai". *International Institute of Welding,* paper XIII-277-62.

†"Advantages and Limits of the Progressive Action Method". *International Institute of Welding,* paper XIII-283-62.

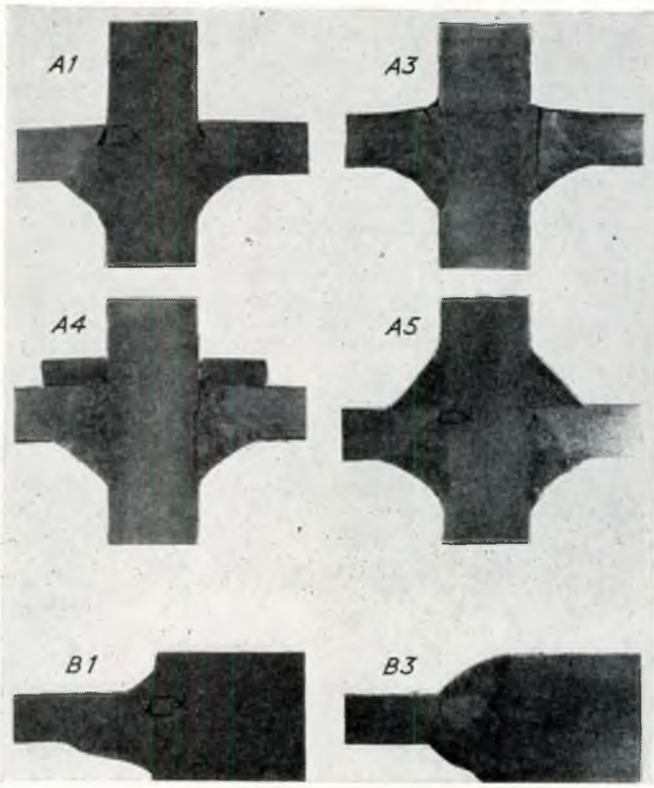



FIG. 31—*Mode of failure of different weld connexions*

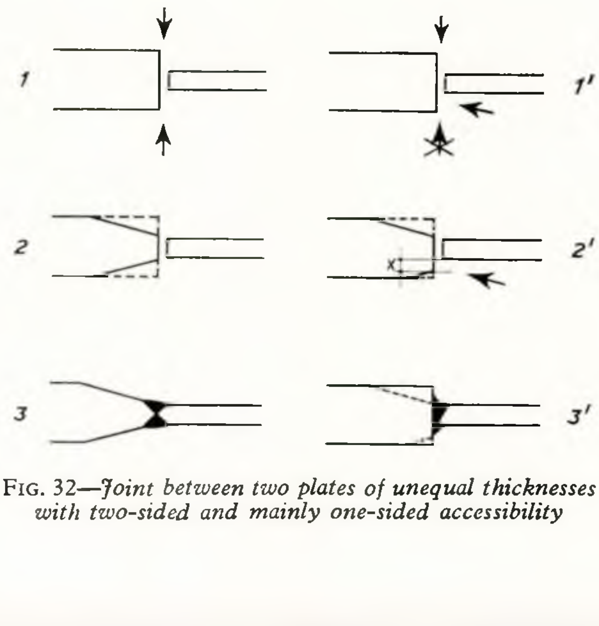



FIG. 32—*Joint between two plates of unequal thicknesses with two-sided and mainly one-sided accessibility*

Fig. 31 which he said was self-explanatory. In the same figure was shown what was to be understood as the lack of penetration X.

Fig. 32 showed that for a two-sided accessibility when welding, a chamfering of the thicker plate was certainly more satisfactory for the B-type specimens. However, in box-type structures, the accessibility was mainly one-sided (position 1).

For the back-weld it became necessary to provide for a flat, which gave the welder a guidance. This practically ruled out chamfering at the inner side. However, with this the chamfering at the outer side became a pure aesthetical argument as could be shown by fatigue tests and photo-elastic investigations.

Mr. Hinson had pointed out that, for the connecting and piston rod, to check the stiffnesses was as important as the stress investigations. This was also valid for a bedplate and, more or less, for every structure. The author had purposely restricted his paper to the possibilities of stress investigations and would like to refer to the paper by Mr. Kilchenmann‡ concerning the connecting-rod stiffness.

Concerning the bedplate stress investigations, Fig. 33 showed the welded joints for different types of Sulzer bedplates. The investigations were prompted by failures on Type II bedplates, restricted to an early series of RD76 engines. The cracks along the weld b-b of Fig. 15, to which Mr. Langballe referred, only occurred in the fillet weld of these Type II bedplates and showed a very low propagation rate. The lack of penetration of such a weld being equivalent to the plate thickness, it was quite obvious that the stress flow through that joint was not good and would result in a substantial reduction of fatigue strength compared with the A-type specimens. This fillet weld would only show about half the fatigue strength

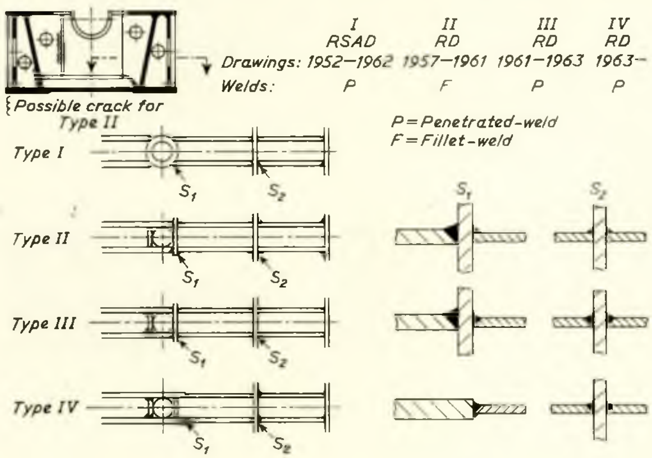



FIG. 33—*Welded joints in the different types of bedplate*

‡Kilchenmann, W. 1961. "The Development of Heavy Duty Marine Diesels During the Past Five Years". *Trans.I.Mar.E.*, Canadian Division Supplement, No. 5, p. 103; also *Trans.I.Mar.E.*, Vol. 73 (bound copies only), p. 476.

of specimen A1 and gave a similar level of stresses to be expected in service. For that reason a Type II bedplate might show cracks along the fillet welds, but must not necessarily do so as indicated by experience on bedplates of this type which had stood up to more than 20 000 running hours without failure.

This experience was the basis from which the designer would evaluate the admissible stressing for the weld joints shown in Fig. 29.

The weld joint between the forged main bearing saddle and the vertical web plates to which Mr. Langballe referred did not allow for any lack of penetration. A comparison of the working stress with the corresponding fatigue limit had not yet been properly done, as the author's company had not at its disposal a test rig permitting dynamic loads of a few hundred tons. This final confirmation was still pending.

Regarding the effect of the horizontal inertia force on the wear of the chocks or their counterfaces, the author did not know of any troubles with RD engines, as their longitudinal girders had been designed for stiffness. This problem was only known in connexion with some old SD68-type engines with cast-iron bedplates.

Coming back to the model testing, it was necessary to mention that the stress distributions had also been investigated for a load acting on the main bearing at various angles to the vertical. These investigations were complemented by photo-elastic tests.

As for the photo-elastic test results, partly shown in Fig. 30, it was mere luck that a geometry was found which was satisfactory and which was proved later on, by cross-checking, to allow for a qualitative comparison.

One could always discuss the right location of strain gauges. The author could not agree with Mr. Hinson that if a crack started in a weld the strain gauges should be applied to the weld, because the measured stress was meaningless. It gave neither an indication of the peak stress at the weld root nor the value of the nominal stresses (direct stress or bending stress). Comparison of measured stresses with fatigue data, could only be done by comparing nominal stresses, but these could only be measured on the plate in the neighbourhood of the weld.

The unidirectional coverage a-a, b-b, d-d (Fig. 15) was chosen to permit a check on the measured values as shown in Fig. 22. For the sections c-c and e-e, sections were placed perpendicular at the points where the maximum stress could be expected. This allowed for an extrapolation of the nominal stresses up to the welds. Only for section b-b was this not done, because of the low stress gradients. It was felt that, in fact, a two-dimensional coverage had been applied near the points of maximum stressing.

Mr. Hinson was also interested in the influence of the tie-rod on the bedplate stresses. When firing, the tie-rods took approximately 25 per cent of the load, the stress reduction being appropriate. It was also important to notice that, due to compressive pre-stress (static stress), the dynamic amplitude of the bedplate stresses became less dangerous in certain areas.

The author concluded by thanking all the contributors for their interesting and valuable contributions to the paper.

*At the Annual Conversazione held on Friday, 2nd December 1966 at Grosvenor House, Park Lane, London, W.1. President of the Institute, Sir Stewart MacTier, C.B.E., B.A., and Lady MacTier (on left), with the Chairman of the Social Events Committee, Mr. Stewart Hogg, O.B.E., and Mrs. E. Morrison*

*Chairman of Council, Mr. R. R. Strachan (left), with Miss E. Barrett, Mrs. J. McAfee, and Mr. J. McAfee (Vice-Chairman of Council) at the Conversazione held on Friday, 16th December 1966 at Grosvenor House*

# INSTITUTE ACTIVITIES

## Annual Conversaziones 1966

Two Conversaziones were held at Grosvenor House, Park Lane, London, W.1, on Friday, 2nd December, and Friday, 16th December 1966. The President, Sir Stewart MacTier, C.B.E., B.A., and Lady MacTier, together with Mr. Stewart Hogg, O.B.E. (Chairman of the Social Events Committee) and his daughter, Mrs. E. Morrison, received the members and guests on 2nd December. The Chairman of Council, Mr. R. R. Strachan, and Miss E. Barrett, together with Mr. J. McAfee (Vice-Chairman of Council) and Mrs. McAfee, received members and guests on 16th December. The numbers present on these two occasions totalled almost 2000.

Music for dancing was played by the Sydney Jerome Dance Orchestra and the Russ Henderson Steel Trio and among those who appeared in the cabaret, compered by Leslie Crowther, were The Charifiens, The Two Perrards, and Joan Turner. The floor show was "Champagne" on 2nd December and "Dazzle" on 16th December.

On the evening of 16th December carols were sung after dinner by the Wimbledon Girls Choir, and were greatly enjoyed by those present.

## Branch Meetings

### Devon and Cornwall

*Junior Meeting*

A Junior Meeting was held by the Branch on Tuesday, 1st November 1966 at Falmouth Technical College, when the paper "Marine Automation and Remote Control" by Ll. Young (Member) was presented by the author to an audience of approximately fifty students and members from the Falmouth area.

The interest of those present was aroused and the paper was followed by a very good discussion.

Mr. K. West (Member), Principal of the College, was in the Chair and the vote of thanks was proposed by Mr. J. Howie (Member).

### Kingston upon Hull and Humber Area

The Branch held a Joint Meeting with members of the Royal Institution of Naval Architects on Thursday, 17th November 1966, when the paper "Ships for Containerized Cargo" by A. Killen, M.R.I.N.A. was presented by the author.

Mr. Killen opened by saying that the great aim of containerization was to cut down the time a ship spent in port by making use of the ship more as a transporter than a floating warehouse. In some cases up to 50 per cent of cargo transportation costs were incurred in port.

Containerization was not new, in fact it had been in use for 60 years, but development had accelerated since the Second World War.

Mr. Killen then gave examples of container ships from all parts of the world illustrating these examples with drawings. Finally he discussed the various design considerations for a modern container ship.

A lively and wide-ranging discussion followed, touching on, amongst many others, the following topics: standard containers; maintenance of shipboard handling equipment; the

**North West England Branch**

*The Chairman of Council, Mr. R. R. Strachan (right), with from left to right: Commander K. I. Short, O.B.E., D.S.C., R.N. (Chairman of the Branch), Mrs. K. I. Short, and Miss E. Barrett, at the Annual Dinner and Dance of the Branch held at the Adelphi Hotel, Liverpool, on Friday, 4th November 1966* economic life of a container ship; the application of nuclear propulsion; space utilization and mooring and loading problems.

A vote of thanks was proposed by Dr. J. F. Leathard and seconded by Mr. G. D. Moore (Honorary Social Secretary).

### North Midlands

*Visit to British Railways, Derby*

On Wednesday, 30th November 1966, a party of forty members of the Branch were the guests of the British Railways Board at their Engineering Research Division, Derby.

The members were welcomed by Mr. S. Wise, Assistant Director Mechanical Engineering, who, in his introductory remarks, explained the work which the division carried out and pointed out many recent developments which were now being put to use in the railway systems of this country.

The party then made an extensive tour of the following departments: physics; computers; soil mechanics; non-destructive testing; strength of materials; dynamics and signalling. After this, Mr. Wise answered the many questions put to him by members arising from what they had seen.

Mr. H. V. Campbell (Chairman of the Branch) then thanked Mr. Wise and his staff for a most interesting visit to see "behind the scenes" at this most important branch of British Rail. The visit terminated at 10.00 p.m. and the members left by coach for Sheffield and Leeds.

*Joint Meeting*

A joint meeting with the Royal Institution of Naval Architects was held on Monday, 5th December 1966, in the Conference Room, Dock Board Offices, Pier Head, Liverpool, 3, at 6.00 p.m., when a paper entitled "Hydrodynamic Factors Influencing the Development of New Propulsion Devices for Ships" by A. Silverleaf, B.Sc., was presented by the author.

Sir Stewart MacTier, C.B.E., B.A., President of the Institute of Marine Engineers, was present at the meeting and opened the discussion which followed the lecture.

The audience, which included members of the Royal Institution of Naval Architects and students and staff of the City of Liverpool College of Technology numbered seventy-eight.

### Scottish

*Tenth Annual Dinner and Dance*

The Tenth Annual Dinner and Dance of the Scottish Branch was held at the Grosvenor Restaurant, Glasgow, on Saturday, 10th December, 1966, when Mr. T. W. Liddell (Chairman of the Branch) and Mrs. Liddell received the two hundred and eleven members and their guests.

Following the Loyal Toast, Mr. Liddell extended a warm welcome to all those present, and especially to the Ladies. He emphasized that this was a night for dancing and not for speeches, and he did not intend to make a speech.

The Chairman was pleased to welcome Mr. R. Beattie (Vice-President) and Mrs. Beattie, also Mr. H. Brady (Immediate Past Chairman of the Branch) and Mrs. Brady.

David Sibbald again, proved himself to be an excellent Master of Ceremonies and his Orchestra kept everyone on their feet throughout the evening.

The function was a great success.

*General Meeting*

A general meeting of the Branch was held on Wednesday 14th December 1966, at the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2., at 6.15 p.m., when a paper entitled "Further Progress in Auto-

**Scottish Branch Tenth Annual Dinner and Dance**

*At the Tenth Annual Dinner and Dance held at the Grosvenor Restaurant, Glasgow, on Saturday, 10th December 1966. Standing from left to right: Mr. L. D. Trenchard (Honorary Secretary of the Branch), Mr. R. Beattie (Vice-President), Mr. T. W. Liddell (Chairman of the Branch), Mr. W. McLaughlin (Vice-Chairman of the Branch) and Mr. H. Brady (Immediate Past Chairman). Seated, from left to right: Mrs. I. Walker, Mrs. H. Brady, Mrs. T. W. Liddell, Mrs. R. Beattie, and Mrs. W. McLaughlin*

### North West England

The Annual Dinner Dance of the North West England Branch was held on Friday, 4th November 1966, at the Adelphi Hotel, Liverpool, at 7.00 p.m., with Commander K. I. Short, O.B.E., D.S.C., R.N. (Chairman of the Branch) in the Chair.

The Branch was honoured by the presence of the Chairman of Council, Mr. R. R. Strachan who was accompanied by Miss E. Barrett. Other guests included Mr. A. E. Franklin (Assistant Secretary, Technical) and Mrs. Franklin, Captain J. A. Smith, D.S.C., V.R.D., A.D.C., and Mrs. Smith, the Principal of Birkenhead Technical College, Mr. C. V. Vinton-Fenton and Mrs. Vinton-Fenton, and Mr. A. R. Kinsman, Principal of Riversdale Technical College, and Mrs. Kinsman.

Three hundred and forty-three members and guests attended the very successful function.

mation" by R. Munton, B.Sc. (Vice-President) and J. McNaught (Member) was presented by Mr. McNaught.

Mr. T. W. Liddell (Chairman of the Branch) presided and welcomed the eighty-eight members and visitors present.

The paper was limited to considerations of automation in the machinery spaces of Diesel ships and was essentially concerned with the authors' policy and experience. Mr. McNaught expressed views upon the necessity for economic justification for each step made in centralized controls and machinery automation.

The installation in the *Clan Macgillivray* class was briefly described and reference made to some of the difficulties experienced. The speaker continued with the design considerations and the interpretation of these into practice in the *Southampton Castle* and *Good Hope Castle*. A comparison was made between the engine room complement of these ships and that of an older large Diesel passenger ship of similar power with no centralized controls or automation equipment. The installation in the *Clan Ramsay* "R" Class was described in some detail, the speaker pointing out that controls should not be installed for systems infrequently used, such as starting up and closing down, on entering and leaving port. Safety and reliability must be the first thought. Emphasis was placed on the need for rigid control of the entire system where there was a lot of machinery and small staff and this point must be given serious consideration during construction.

Looking into the future, Mr. McNaught expressed the opinion that there was still much to be learned from experience gained from day to day and the question of manned or unmanned engine rooms in the future must be seriously considered from all aspects.

With extensive use of slides Mr. McNaught gave a very good description of existing and proposed engine room layouts.

The Chairman opened the discussion by pointing out that present developments involved the deprivation of engineers' senses. He thought that consideration should be given to making the machinery spaces habitable and putting noisy machinery in soundproof boxes. He could not visualize engineers locking up the engine room and going happily to bed with £500 000 worth of machinery around.

The discussion was very interesting and controversial and it was with some regret that the meeting closed at 8.30 p.m. when Mr. J. J. W. Bryne expressed the thanks of the Branch to Mr. McNaught for the most interesting paper which he and Mr. Munton had prepared.

### West of England

A general meeting of the Branch was held on Tuesday, 13th December 1966, in Smith's Assembly Rooms, Westgate Buildings, Bath, at 7.00 p.m., when the 1966 Parsons Memorial Lecture "The Prospect for Steam Propulsion" by Captain N. J. H. D'Arcy, R.N. (Member of Council), was read by the author.

Mr. J. P. Vickery (Chairman of the Branch), presided at the meeting and welcomed the audience of thirty-five members and guests which included Mr. F. C. Tottle, M.B.E. (Local Vice-President).

Captain D'Arcy, who on previous occasions had given his lecture in full, gave a shortened version. This was followed by a lively and interesting discussion, and such was the interest shown that the Chairman had to reluctantly intervene and terminate the proceedings owing to shortage of time.

Mr. Vickery, in a vote of thanks to the speaker, expressed the appreciation of those present, and the meeting ended at 9.00 p.m.

## Election of Members

Elected on 12th December 1966

MEMBERS

*Elections*

Giorgio Beltrami, Dott. Ing.
Raymond A. C. De Raedemaeker
George Farkas
Samuel Elsby Harper
Herbert Alan Hunter
Syed Nasir Mehdi
Arthur Lionel Redford, M.B.E.
Albert William Rowe
Edward Hexter Upton

*Transferred to Member from Associate Member*

David William Akhurst
Sasadhar Datta
Gerald Taylor Gailey
Rudolph Johannes Kuhn
Royden John Whyte

*Transferred to Member from Graduate*

Adrian George Glentworth

ASSOCIATE MEMBERS

*Elections*

John Alfred Beech
Andrew Bulloch
George Ross Graham
James Young Hamilton Halliday
Kenneth Herbert Harding, Eng. Lieut., R.N.
Alan Edwin Makepeace, Eng. Lieut., R.N.
Derek Nelson
Leslie Arthur Newman, Eng. Lieut., R.N.
Leonard Frank Porter, Eng. Lieut., R.N.
Christopher Robertson
Alexander Charles Robertson Shand
Gordon Victor Shoesmith
Suresh Soota, Lieut., I.N.
James White Stevenson
James Ward

*Transferred to Associate Member from Associate*

John Vincent Fry
Maung Nyana
Harjit Singh Sidhu, Lieut., I.N.
Keith Edmund Tucker

*Transferred to Associate Member from Graduate*

James Crabb, B.Sc.
George Irving Papworth
John Sydney Michael Sutton
John Theodore Henry Willcox

*Transferred to Associate Member from Student*

David Harrold Blenkinsop, B.Sc.
George Ian Buchanan, B.Sc.
Antony William Burton
Philip John Martin

*Transferred to Associate Member from Probationer Student*

David Laurence Gifford

ASSOCIATES

*Elections*

William Arundel
William Cragg
John Day
Bernard Hamilton Gane
Dhirendra Kumar Hajela
Christos Joannides
Maurice Jobling
Walter Simpson Lewis
Derek Harold Luke
Donald Metcalfe
Brian Walter Nott
Aubrey Llewellyn Phipps-Jones
Alan Ray
Horace Remedios
Walter Gordon Richards
Trevor George Robinson
Dennis Charles Tarplee

Roy Whittingham
Saiyid Mohd. Kamaluddin Zafar

*Transferred to Associate from Graduate*
Adrian Harfield Hulse
John Terence Peters
Gwynfryn Thomas Roberts
Harry Tomlinson

GRADUATES
*Elections*
Robert James Adams
Nasir Ansari
Michael Stuart Biggs
Keith Ronald Corless
James Doyle
Frederick James Hay
Long Sey Hai
Gabriel Raftopoulos, B.Sc.
Vempala Venkata Rao
Raymond Edwin Sanders
Robin Hartlett Tandy, Lieut., R.N.
Abdul-Wahab Thabet
John Conroy Treeby

*Transferred to Graduate from Student*
Anthony Charles Davis
Solomon Waddoyi Ohingo
Jonathan Boyi Ojeh, Dip. Tech.(Eng.)

*Transferred to Graduate from Probationer Student*
Frank Massey
David Whitehead

STUDENTS
*Elections*
Ashok Kumar Agrawal
Mahmood Billah
Edward Alexander Britton
Pooran Prakash Chugani
Suresh Dagur
Harbhajan Singh Gandhi
Robert Begg Jack
Mahesh S. Jain
Jagdish Prasad Joshi
John Drever Leslie
Malcolm MacKinnon
Alan Paul Martin
Pradip Kumar Nayyar
Abdul Razap A. Owoyemi
Rabindra Nath Padhi
Thomas Donald Patterson
Melvyn Edward Poulter
Kesava Iyengar Raghuveer
Swastik Sandel
Michael William Sheret
Kuljit Singh
Rajendra Prasad Singh
Brian Richard Tremain
Radha Kant Verma
Pottayil Viswanath

PROBATIONER STUDENTS
*Elections*
Peter Edwards
Kenneth Leo Hagen
John Malcolm Hepworth
Martin Hillmansen
Neal Roy Hillsley
Christopher Noel Ingleson
Glyn Trevor Jones
Thomas Leith Taig
Malcolm Walker

# OBITUARY

CORNELIUS FRANCIS BUCKLEY (Member 6837) died on 17th May 1966.

Born in 1895, he served his apprenticeship with the Kerry Electric Supply Company. In 1915 he went to Malta with an erection team engaged on the construction of the island's first steam turbo-generators and in 1916 he was appointed assistant engineer with the Water and Electricity Department of the Government of Malta. He remained with the Department until his retirement in 1959, rising to the position of chief electrical engineer of the electricity branch.

Mr. Buckley, who was an Associate Member of the Institution of Electrical Engineers and a Fellow of the Royal Society of Arts, was elected a Member of the Institute in September 1939.

ADDISON STANLEY BURDIS (Member 3304) died on September 8th 1966. He was seventy-six years old.

He served his apprenticeship with Smith's Dock Co. Ltd., South Shields, after which he saw sea service for three and a half years, gaining his First Class Board of Trade Certificate just before the outbreak of the Great War, in which he served with the Royal Engineers. On leaving the Army he served for a time with the Valvoline Oil Company until he went to Africa to take up an appointment with the East African Trading Company. After this he again went to sea and in 1939 he obtained a First Class Motor Endorsement. He served as chief engineer in oil tankers until his ship was torpedoed in mid-Atlantic and it was for his action on this occasion that he received the M.B.E. Due to injuries which he had received he was unable to return to sea and he joined the staff of the Liverpool Technical College as a lecturer. Subsequently he obtained a transfer to the Erith Technical College, where he remained until his retirement in 1959.

Mr. Burdis was elected a Member of the Institute in July 1917. He was also an Associate Member of the Institution of Mechanical Engineers. He is survived by his wife.

DOUGLAS WALTER JAQUES (Associate Member 13227) died on 6th July 1966 at the age of fifty-eight.

From 1934 to 1939 he was apprenticed at H.M. Dockyard, Chatham. In 1934 he was awarded the Royal Drawing Society Gold Medal for Mechanical Design and in 1938 he was awarded a Whitworth Prize.

From 1939 to 1942 he was an acting second class draughtsman on the staff of the Commander in Chief, Nore, and from 1942 to 1945 on the staff of the Deputy Director of Dockyards, London.

He then served as naval architect and marine engineer with Rendel Palmer and Tritton, consulting engineers, of London, until 1951, gaining his B.Sc.(Engineering) in 1949. He was later with BP Trading Ltd., in various engineering appointments, until the time of his death.

He was elected an Associate Member of the Institute in March 1951. He leaves a widow.

JOHN JOHNSTON (Member 16407) died suddenly on 11th June 1966 at the age of fifty-five. Mr. Johnston was apprenticed to David Rollo and Sons, Liverpool from 1927 to 1932, when he joined the now BP Tanker Co. Ltd. as a junior engineer officer. He gained his First Class Motor Certificate in 1937 and in 1941 received his first appointment as chief engineer.

He became engineer commodore of the BP Tanker Company fleet on 1st July 1964, and was awarded the O.B.E. in January 1966. His last appointment was in m.v. *British Commerce.*

Mr. Johnston was elected a Member of the Institute in June 1955. He was also a Member of the Marine Engineers' Association.

His wife survives him.

ROBERT JOLLY (Associate Member 4528) died on 17th August 1966. He was eighty-four years old.

He was educated at Allan Glens School in Glasgow, and, after serving his apprenticeship with Bow and McGlaughlan, Paisley, joined the Glasgow office of the British India Steam Navigation Co. Ltd. in 1904. In 1920 he transferred to the company's head office in London as technical assistant to the superintendent engineer. Due to ill health, he retired in 1942, after thirty-seven years service with the company.

Mr. Jolly was elected an Associate Member of the Institute in April 1922.

ALEXANDER GRAHAM LISTON (Member 6832) died on 20th August 1966. He was seventy-four years old.

He started as an apprentice with the London and South Western Railway in 1908. In 1914 he joined the Royal Mail Line, serving with them for ten years, during which time he was torpedoed twice. He held a First Class Board of Trade Certificate and in 1924 joined the Southern Railway, serving in several cross-Channel steamers, including s.s. *Isle of Guernsey,* of which he was chief engineer. In 1931 he became assistant to the mechanical engineer at Southampton docks, becoming assistant superintendent marine engineer in 1938. In 1945 he became superintendent marine engineer and later became marine manager, retiring in 1958.

In 1945 he was awarded the M.B.E. for his services during the invasion of Normandy in 1944.

Mr. Liston was elected a Member of the Institute in September 1931. He is survived by his wife and daughter.

CHARLES JOHN MILES, M.B.E. (Member 13862) died on 16th October 1966. He was seventy-five years old.

He served his apprenticeship with the Great Western Railway in their marine factories at Neyland and Fishguard. From 1912 to 1914 he served with the White Star Line in *Majestic* and *Oceanic,* rising from sixth to fourth engineer. In 1914 he joined the Marine Department of the Great Western Railway Co. He obtained his First Class Steam Certificate in 1916 and was first appointed chief engineer in December 1929. From 1946 to 1948 he was resident engineer, new ship construction, Liverpool. His next appointment was as marine engineer in charge, Marine Department, British Railways, Fishguard Harbour, where he remained until his retirement in 1956.

Mr. Miles was elected a Member of the Institute in July 1952.