LLOYD'S REGISTER'S TECHNICAL INVESTIGATION DEPARTMENT
SOME INTERESTING INVESTIGATIONS AND EXPERIENCE GAINED FROM 50 YEARS OF OPERATION

BY

J.S. CARLTON
and
I. BANTHAM
(Lloyd's Register Marine Division)

This is an edited version of the paper first published in the Transactions of Lloyd's Register's Technical Association, *Session 1997/8, October 1997.*

ABSTRACT

The article marks the fiftieth anniversary of Lloyd's Register's Technical Investigation Department and considers some of the experience gained during troubleshooting and problem solving activities since the Department was formed. Following a brief resumé of the origins of the Department and its work, attention is turned to the various full scale experimental and computational techniques that have been developed. The most common forms of failure encountered today are identified and discussed, as are the more usual types of shipboard vibration excitation. Several examples of interesting or novel investigations in the marine, land-based and offshore industries which embrace specific engineering lessons are outlined. The article finally moves to a discussion on the Department's work in the condition monitoring of marine machinery and land and offshore structures before concluding with some comments based on the more general experience gained during the course of the Department's activities.

Introduction

'Those who do not remember the past are condemned to repeat it'.

So wrote the Spanish born philosopher George SANTAYANA in the early years of this century.[1] Although this caution was expressed in the context of general philosophical and political argument, it is nevertheless true for engineering design and operation. This is because engineering is not a precise science; rather it is a combination of scientific method and engineering experience combined with manufacturing and operational practice and capability. This article, drawing as it does on the experience gained during the course of engineering investigation, seeks to contribute to the body of knowledge on the causes of engineering failure and unexpected performance together with the methods by which this can be investigated.

The Technical Investigation Department (TID) of Lloyd's Register (LR), since its formation fifty years ago, has been invited to investigate some of the more intractable engineering problems posed by the marine industry. It has done this by a combination of trial measurement, theoretical analysis and engineering judgement based on the Department's accumulated experience. This experience has embraced a full range of failure and mal-performance based engineering problems from the marine, land-based and offshore industries. To address these problems TID comprises a multi-disciplinary group of engineers and scientists which includes:

- Mechanical, marine, civil and structural engineers
- Naval architects
- Physicists
- Electrical and electronic engineers
- Mathematicians
- Computer scientists
- Statisticians.

The talents of this relatively unique group of highly qualified engineers and scientists are combined to provide a high quality, rapid response, technical consultancy and advisory service to the marine industry, the other industries that LR serves and to LR itself. These activities are supported by instrumentation and material investigation laboratories. The Department also conducts medium and long term marine engineering research and development to equip LR, in keeping with its divisional strategies, with the necessary capabilities to meet future requirements based on a continuing review of technological development. Clearly, by the nature of the Department's primary work, many of the research initiatives derive from the failure scenarios presented to TID.

Many engineering investigation activities involve an iteration which commences with a hypothesis about the failure mechanism or the underlying cause of an unexpected performance. From the initial working hypothesis, either a suitable instrumentation fit and trials programme is prepared or, alternatively, a theoretical model is derived to test the validity of the hypothesis. In the former case, following the full scale trials stage of the investigation, an analytical or numerical model may be formulated in order, first to obtain correlation with the quantitative trials data set and then, secondly to extend this domain so as to formulate proposals for appropriate remedial action. In the latter case a similar procedure applies but without the benefit of measured data, the correlation being undertaken by comparing the predictions with the observed qualitative behaviour and other standard test cases. In these cases significant demands are placed on the selected analytical or numerical procedures which, in turn, require that the chosen methods are well correlated and that their predictions can be generally supported by less detailed methods or heuristic insights into the appropriate behaviour. For both situations the accumulated experience of past failures and the factors involved in their manifestation are an essential ingredient. Consequently, to satisfactorily execute an investigation a unique blend of measurement and theoretical analysis combined with a sound historical knowledge of failure situations are essential for each case.

Origins and work of the Department

Lloyd's Register's Technical Investigation Department was formed in 1947 under the then Chief Engineer Surveyor, Dr S.F. Dorey. His primary purpose in forming the Department was:

> 'To give LR a capability to explore marine failures and to research technical problems with a view to improving the Rules.'

This vision has remained the guiding principle for the Department to the present day. However, when dwelling on the work of TID it should not be concluded that LR commenced its interest in failure mechanisms and Rule development at the time of TID's formation. Clearly, prior to this time LR had undertaken significant amounts of research and development as witnessed by both the stage of development of the 'Rules' at the time TID was inaugurated and the various references in LR's Annals to Principal Surveyors for Research.

At the time of its formation much of the Department's early work centred on torsional vibration problems, hull vibration and material fatigue. In the case of hull vibration, an extensive series of studies were undertaken in 1947 on a wartime ship in the River Fal in Cornwall using a purpose built exciter made in the Department. In contrast, the work on metal fatigue took place in an old earth floored laboratory, once used by STEPHENSON, in the Stavely Iron Works. Recognizing the importance of specimen size on the results of fatigue tests, in order to maximize specimen size the equipment used for this work was designed by members of the Department to function on an electrically induced resonance principle. Although designed by them and their colleagues, this equipment was not entirely liked by the members of the Department assigned to undertake the experiments as during the lengthy trials the resonance properties of the machines varied in sympathy with the electricity grid loads and needed constant fine adjustment by the Surveyors around the clock.

The earliest consultancy report that survives in the Departmental archives is dated the 18 September 1948 and relates to a torsional vibration investigation on the main machinery of the M.V. *Turoy*, conducted during trials on the River Humber. Since that time a complete set of TID's reports is maintained in the Department's library. Apart from the longer term research work in ship and machinery vibration which formed an underlying theme of the Department's work, consultancy investigations rapidly settled down into two relatively distinct types. These were problems with ship's machinery or structure which persisted while the ship was operating at sea and which, therefore, had to be investigated at sea and, secondly, major failures which occurred at sea and incapacitated the ship, consequently causing serious delay. This is a distinction which continues to the present time.

From the beginning TID needed to provide a rapid response service to any part of the world and this was made possible by the growing international airline network. However, even by 1951 the novelty of TID surveyors providing this kind of service still warranted mention in the Annual Report:

> "Surveyors of the engineering research department have flown many thousands of miles in the past year to investigate urgent problems on behalf of shipowners and ship and engine builders."

Today the Department's ability and reputation for responding rapidly to calls for assistance are rightly taken for granted by clients and LR alike. Since the Department's formation it has undertaken 4,775 investigations up to the end of 1996 and the growth of these with time is seen in (FIG. 1). An increasing demand for the services on offer can be seen with areas of significant activity in the late 1950's and early 1970's. The fall off in the last year has been due to the separation of LR's environmental activities into a specialized department which had previously been grown to a mature state within TID through LR's Marine Exhaust Emissions Programme[2] and work on Ro/Ro vehicle deck emissions and habitability. Indeed, throughout the life of TID there have been several examples of particular activities forming and being developed to a suitable stage where specific services could be offered to industry.

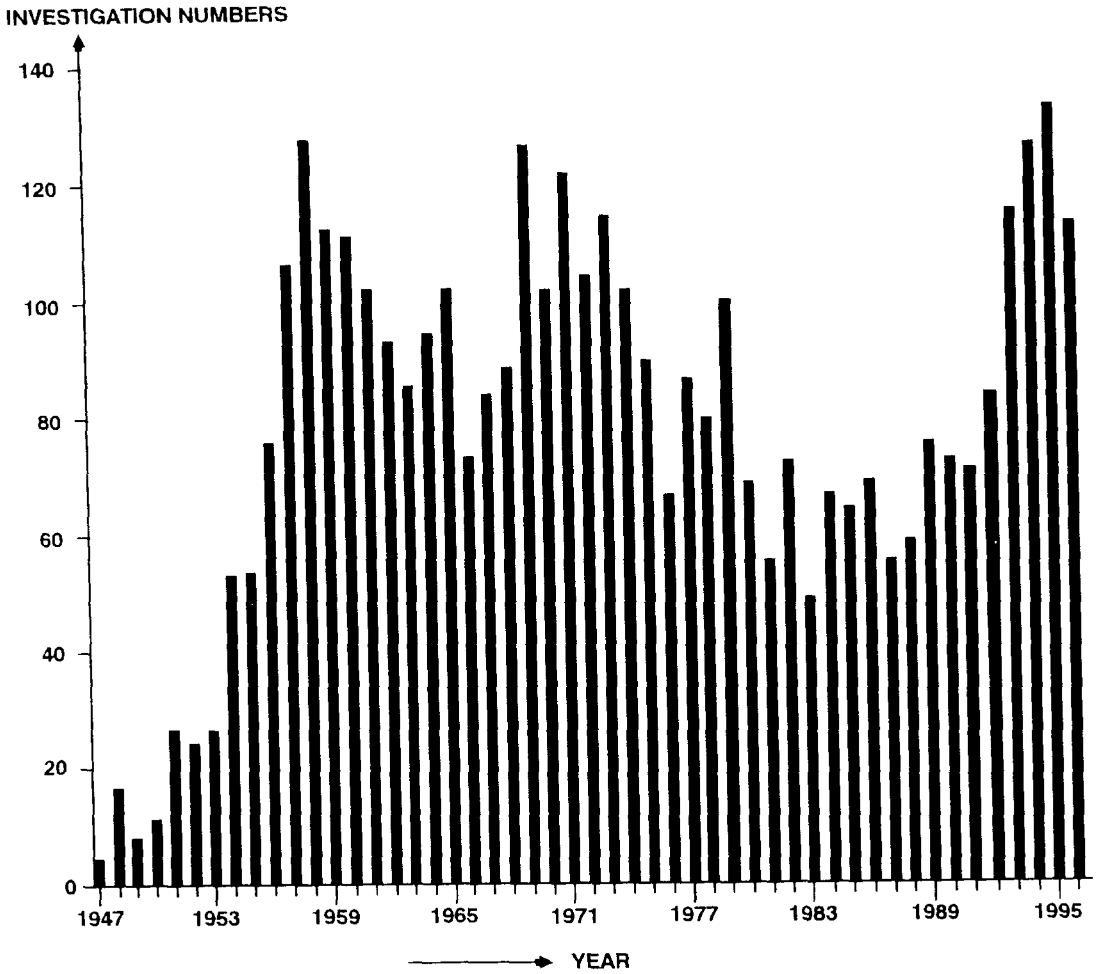


FIG. 1—DISTRIBUTION OF INVESTIGATIONS BY YEAR

Analysis of the Department's database, Table 1, shows the distribution of investigation types that have been undertaken in each of the marine, land based and offshore sectors. The subdivision of the marine categories in the Table is more extensive because this area forms the major part of the Department's work. As such, the miscellaneous category in the marine grouping is significantly smaller than the other two where, for example, a wide range of investigation work is undertaken that does not conveniently fit into the standard categories.

TABLE 1—*Primary Classification of Investigations 1947—1996*

Marine	%	**Land Based Industries**	%	**Offshore**	%
Hull/Mach. Vibration	25.8	Petro-Chem/ Indust	21.7	Platforms	58.3
Shafting	24.1	Power Stations	20.3	Miscellaneous	33.3
Diesel Engines	14.1	Miscellaneous	18.5	Pipelines	8.4
Gearing	9.1	Buildings	17.0		
Power Absorption	5.8	Nuclear	14.5		
Noise	2.7	Docks and Harbours	4.7		
Turbines	2.5	Bridges	3.3		
Electrical	2.4				
Propellers	2.1				
Auxiliary Machinery	2.0				
Condition Monitoring	1.9				
Boilers	1.6				
Couplings/Clutches	1.5				
Ship Structural Failure	1.2				
Rudders	1.1				
Environment	0.9				
Miscellaneous	0.8				
Pipelines	0.4				

When considering the data presented in Table 1, clearly these assignments have not been equally distributed over the years. Indeed, in TID's experience, it is a general characteristic that investigations tend to be grouped into subject areas for discrete periods of time. (FIGS. 2, 3 and 4) show typical examples of the variable trends with time for hull and machinery vibration, shafting, and diesel engine investigations; these being the three largest categories shown in Table 1.

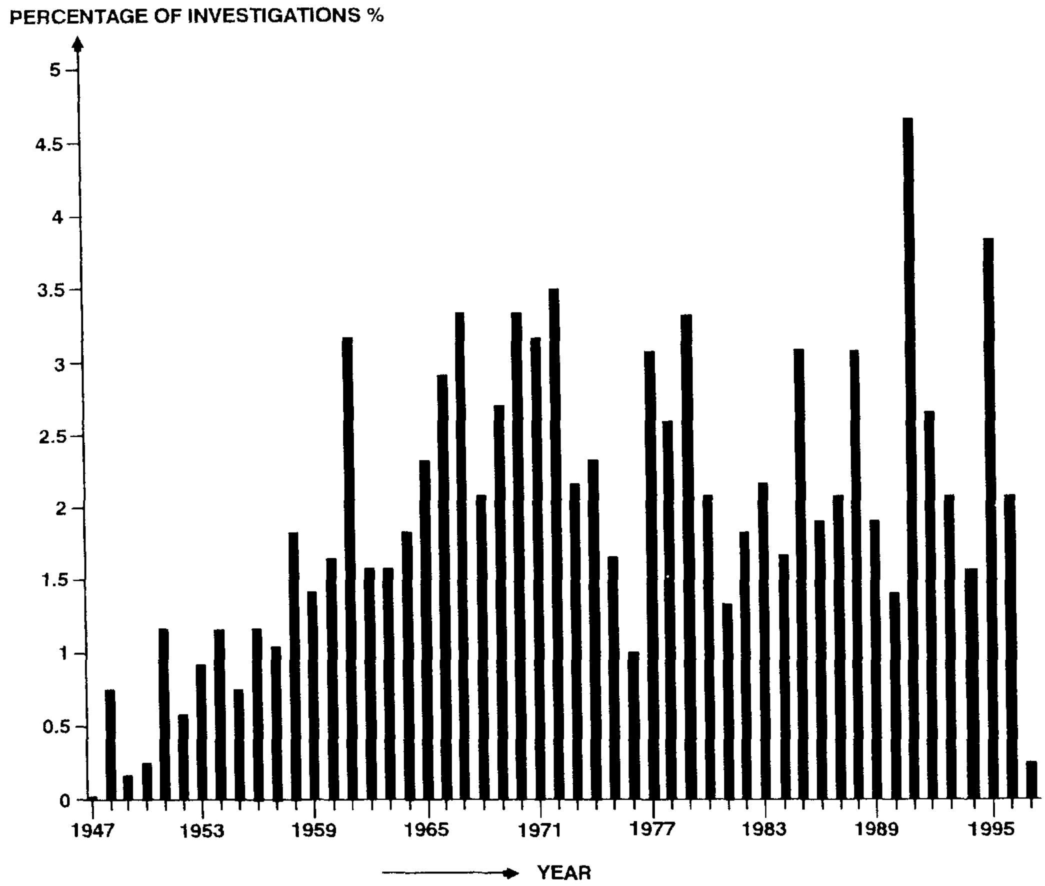


FIG. 2—DISTRIBUTION OF HULL AND MACHINERY VIBRATION INVESTIGATIONS

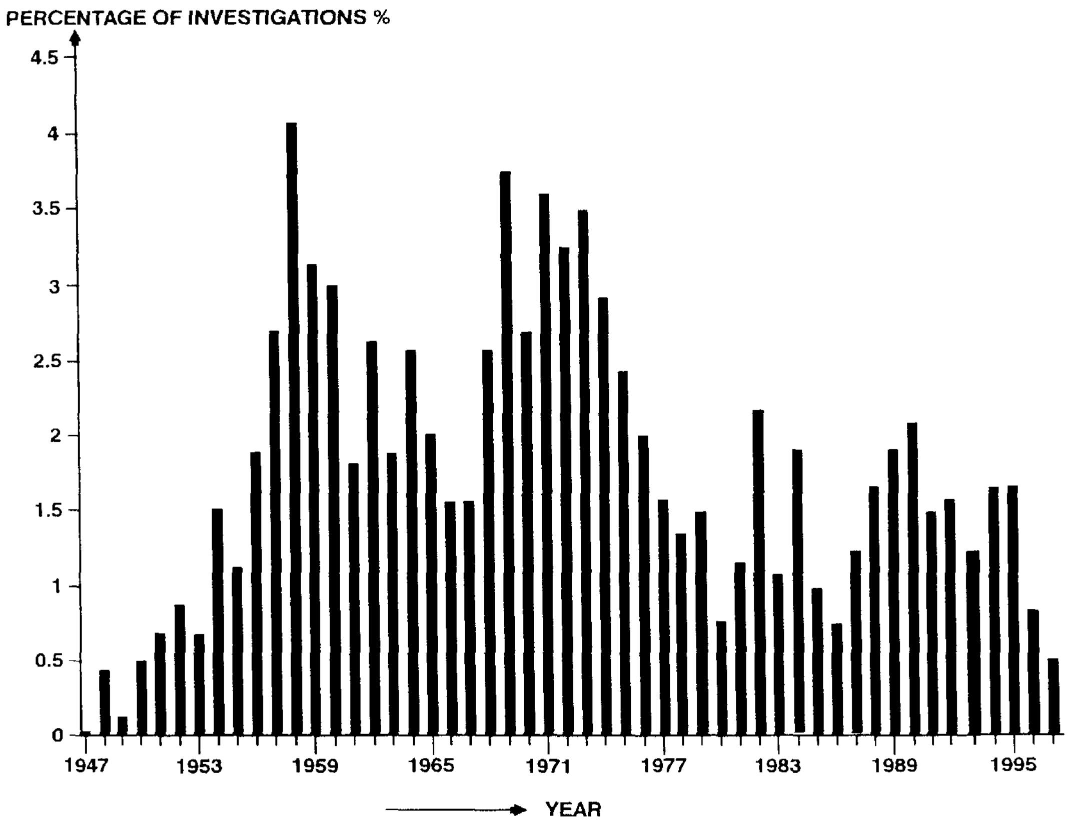


FIG. 3—DISTRIBUTION OF SHAFTING INVESTIGATIONS

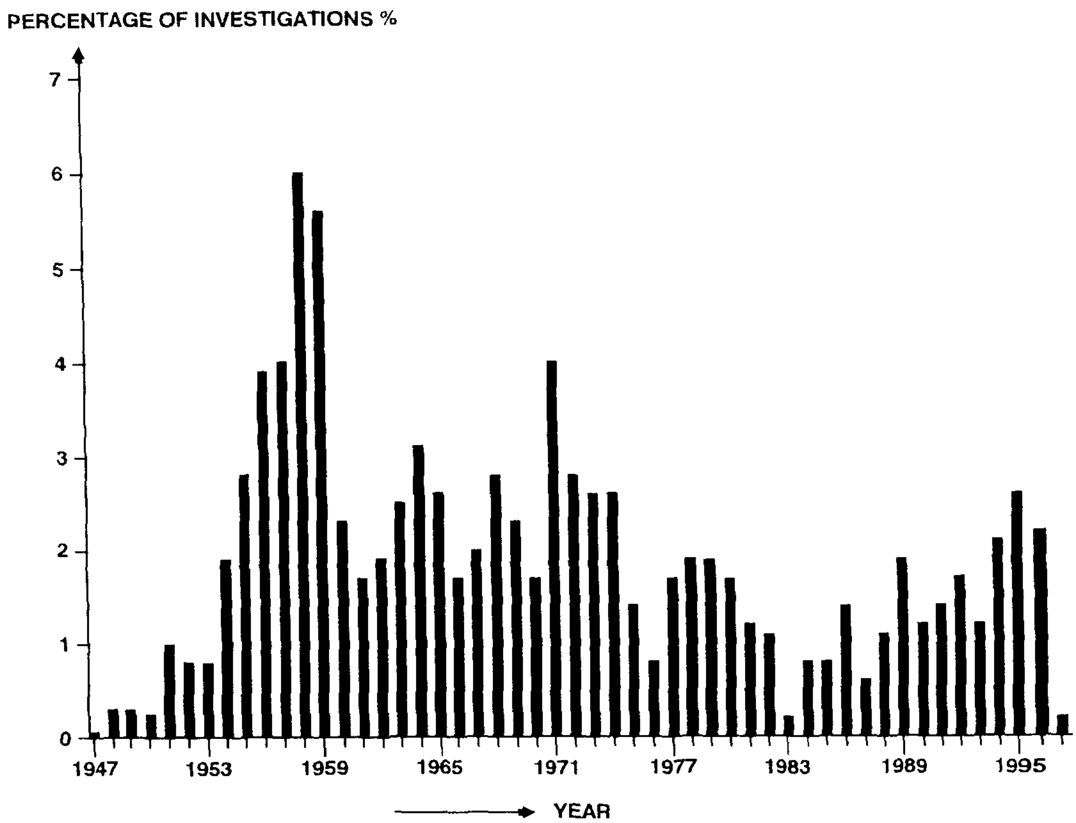
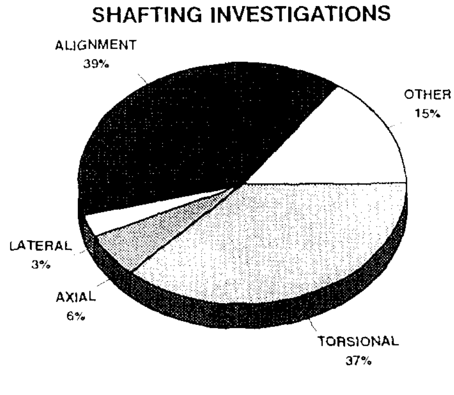
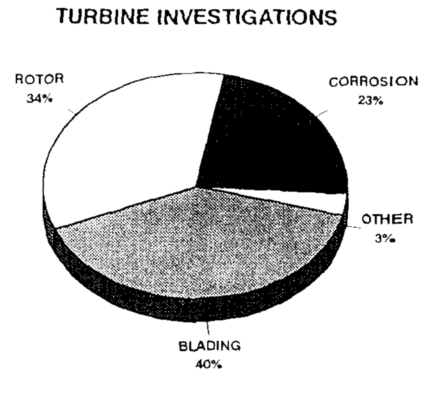


FIG. 4—DISTRIBUTION OF DIESEL INVESTIGATIONS

Within the various general categories there are some cases where subdivisions of investigation activity are of interest. These are shown in (FIG. 5) for the general categories of:

- Shafting
- Turbines
- Diesel engines
- Boilers
- Propulsors
- Electrical investigations.





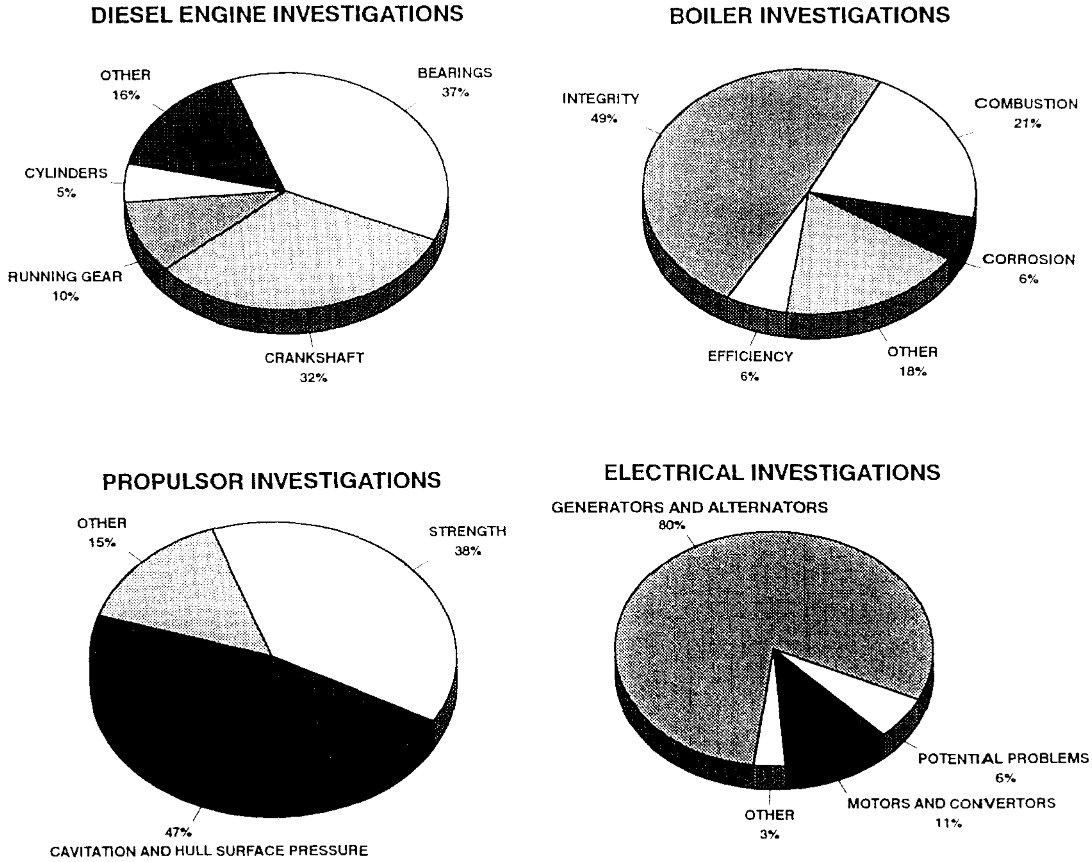


FIG. 5—SUBGROUPING OF SOME INVESTIGATION CATEGORIES

The ability to take engineering measurements in a wide range of environmental and in-service conditions, when coupled to an analysis and interpretation of the results, has enabled the Department to undertake a wide range of technical verification tests. LR's reputation for independence, integrity and impartiality provides an important cornerstone for these services.

Apart from the routine work associated with ships' acceptance trials; power measurements, noise and vibration surveys and bollard pull certification for tugs, there have been many unusual applications for TID's expertise, especially in measuring strain, displacement and acceleration.

The manufacturers of aviation flight simulators have adapted their product to perform fun rides in modern theme parks. These motion machines are subjected to stringent design safety assessments and subsequent operating licence regulation by the relevant National authorities. Cyclic loading of the fabricated lattice structures is a difficult design problem and fatigue cracks have occurred in some installations. Full scale stress measurements have become an integral part of the design verification procedures. Strain gauge positions are selected after studying the results of finite element calculations, and typically about 50 three-element rosettes are fitted. The strain measurements together with the hydraulic actuator loads and displacements are recorded by data-loggers, not only for the normal ride sequence, but also under simulated failure conditions. TID analyse the data to give the principal stress fluctuations which are then used in conjunction with welded joint fatigue life design limits to estimate the endurance of the structure.

Similar strain gauge measuring techniques are used in pressure vessel hydraulic tests and also for occasional pneumatic tests. For this work it is normal practice to fit the strain gauges on each side of the vessel's shell to differentiate between membrane and bending stresses. Ultrasonic methods are used to align the external and internal gauges on the shell or re-inforcing plates. Techniques have also been developed to waterproof the internal gauges and their electrical connections for pressures up to 200 bar.

One particularly interesting use of strain monitoring is during the mechanical stress relief of large containment structures. This procedure is usually applied when thermal stress relief is impractical due to the size of the pressure vessel. The principle of mechanical stress relief is to subject the vessel to hydraulic pressure cycles such that it attains a stress-stable condition with fully elastic behaviour up to the 'standard' test pressure. Structural response in the vicinity of stress concentrations is measured by strain gauges and acceptance is based on a demonstration of elastic characteristics at the gauge positions during two consecutive cycles up to the 'standard' pressure. ASME VIII Division 1 specifies the appropriate requirements for the test procedures.

Acceleration measurements were central to the testing of a novel capsule which was intended as an emergency escape from offshore platforms. The capsule was designed to operate as a free vertical fall unit with adjustable ballast so that on impact with the sea it would have a slightly negative buoyancy. A small propulsion motor could then propel it as a submerged craft beneath any fire surrounding platform, before the ballast was discharged. It was hoped that drops exceeding 25 metres might be feasible if suitable suspension arrangements for the occupants could be devised. When the capsule was released from a barge mounted crane the best seating arrangement, according to the dummy occupants on board, was supported by a rigid suspension. The various alternative arrangements incorporating springs, hydraulic and gas dampers gave higher accelerations, which in some cases were sufficient to break the safety harnesses during the recoil. It is suspected that the project test programme may still be waiting for its first human volunteers!

Not all investigations fall into the traditional mould of marine and industrial problems. Some, while being serious in their own right, are memorable for the circumstances in which they were carried out. One such example from the 1960's concerned an upper floor dance hall above some shops with large plate glass windows. Dances were held on Saturday nights together with a non-stop lunch time session. Problems arose when shoppers were on occasions startled to see the plate glass windows bulging rhythmically and tins bouncing on the shelves in harmony. After several evenings spent measuring vibration on and around the dance floor, with notebooks bearing such unusual entries as *Quickstep—2.5 cycles per second*, TID surveyors established that the dancers, particularly in their interpretation of the latest hit *March of the Mods*, were exciting the resonant frequency of the main girders supporting the floor. The problem was solved by placing pillars at strategic points under the girders. This is one example where building vibration has provided an interesting investigation: other examples have been associated with the:

- Piling action for locks on a river and its effect on ancient buildings
- Condition monitoring of post tensioned concrete bridges
- Effect of passing heavy London traffic on historic buildings.

Development of measurement techniques

As highlighted earlier, the diagnosis of machinery failures or mal-performance requires many skills in order to develop a satisfactory solution. The use of appropriate measurement capabilities is one aspect of the problem and TID uses both standard 'off-the-shelf' instrumentation and also, where necessary, develops equipment or measurement techniques to meet the needs of particular or a class of investigations. In all cases of standard or developed instrumentation, to be of value the instruments must have traceability to national standards and TID rigorously adheres to this policy.

In marine field investigations ease of instrument application is a critical parameter; both with respect to the time required to install equipment and to the level of instrumentation skills needed. To illustrate this point, strain gauges have been fitted by Departmental staff in the last twelve to eighteen months in temperatures as low as $-25°C$, in snow storms and rain or in the humidity and temperatures of the tropics. These various installations have, when complete, been required to work in nominally dry or wet conditions, interfacing with a variety of fluids, to be fitted to shafting systems which could not be stopped and be exposed to operating temperatures of several hundreds of degrees. As for time constraints, these can be exacting; recently, for example, in order to coincide with a ship's sailing schedule a system of ten shaft alignment strain gauges was fitted to the shaft, wired up, calibrated and the alignment trial undertaken in 4 hours.

The underlying philosophy relating to engineering investigations is that, if possible, measurements, or for that matter computations, made by one generic form of procedure are to be supported by those from a different generic form. Additionally, as an investigation proceeds and particularly if it is of an exploratory nature, it may be necessary to improvise or develop instrumentation in the field which can be a difficult undertaking in some of the remoter locations of the world.

The development of new instrumentation or methods can be either short or long term, depending upon whether the result is required for a particular assignment or for a generalised investigation procedure. By way of example of the short term development, the expansion of the UK Nuclear programme in the late 1950's and early 1960's led to an urgent requirement for the TID Laboratory to develop new encapsulation methods for strain gauges; the annual usage being of the order of 10,000. The majority of the technical development work is normally carried out in our laboratory, (FIG. 6), and in some cases may extend over a number of years. Examples of longer term developments have been strain gauge shaft alignment, gear tooth root strain techniques, propeller blade strain measurement, data transmission and remote trial control methods.

Shaft alignment

The self weight of the shaft, the overhung propeller and its associated operational loads, gear wheel meshing and the flexibility and loading of the ship's structure combine to make the alignment of the shaft supports, both in height and attitude, a non-trivial process. In addition to the operational loads, deflections from the nominal may also arise from:

- Structural deformation as the hull is welded,
- Thermal distortion of the structure due to the sun,
- Variations in attitude between dry docking and afloat, and
- Residual stresses which may relax once a service load has been experienced.[4]

As far back as 1951, methods by which the alignment of marine shafting could be undertaken without breaking the couplings were being developed by members of the Department. This early work resulted in an alignment indicator which was a three point support instrument comprising an Invar chassis and a strain gauged Tufnol cantilever with a hardened steel insert. The instru-

FIG. 6—TID LABORATORY—EQUIPMENT PREPARATION

ment was clamped to the shaft and measured the change of strain as the shaft was rotated using the turning gear. This measurement procedure was superseded in the early 1960's by the direct use of strain gauges bonded to the shaft. The practical interpretation of these measurements, the corresponding theoretical calculations and the results of other alignment techniques have been continuously developed to the present day.

The basis of the strain gauge method is that the bending moment at any station in the shaft can be considered as comprising two components: the moment arising from self weight with the bearings positioned in a straight line and the moment arising from the deflections from the straight line condition. To measure the actual bending moment in a shaft, (FIG. 7), strain gauge half bridges are fitted, usually at two stations per span. By using pre-encapsulated, pre-wired gauges, with fast curing resin or cyanoacrylate adhesives, reliable installations can be applied very quickly. After instrumentation the shaft is rotated through 360° using the turning gear, with strains measured at 90° intervals to obtain bending strain data for the vertical and transverse alignments. The differences in strain across the diameter are used to evaluate the shaft bending moments. The straight line bending moments are calculated from a theoretical representation of the shafting system together with the influence coefficients for the load-deflection response at each bearing. By subtracting the straight line bending moments from the measured bending moments and using the computed influence coefficient data a desired alignment to meet the operational conditions can be derived. Furthermore, from a proper treatment of the system boundary conditions, loads on inaccessible bearings in the gearbox or at the stern tube bearings can be determined. The measurement procedure is applied in either the hot or cold state and the ballast and loaded conditions. In more complex situations, the procedure can be employed dynamically by using radio telemetry systems for signal transmission.

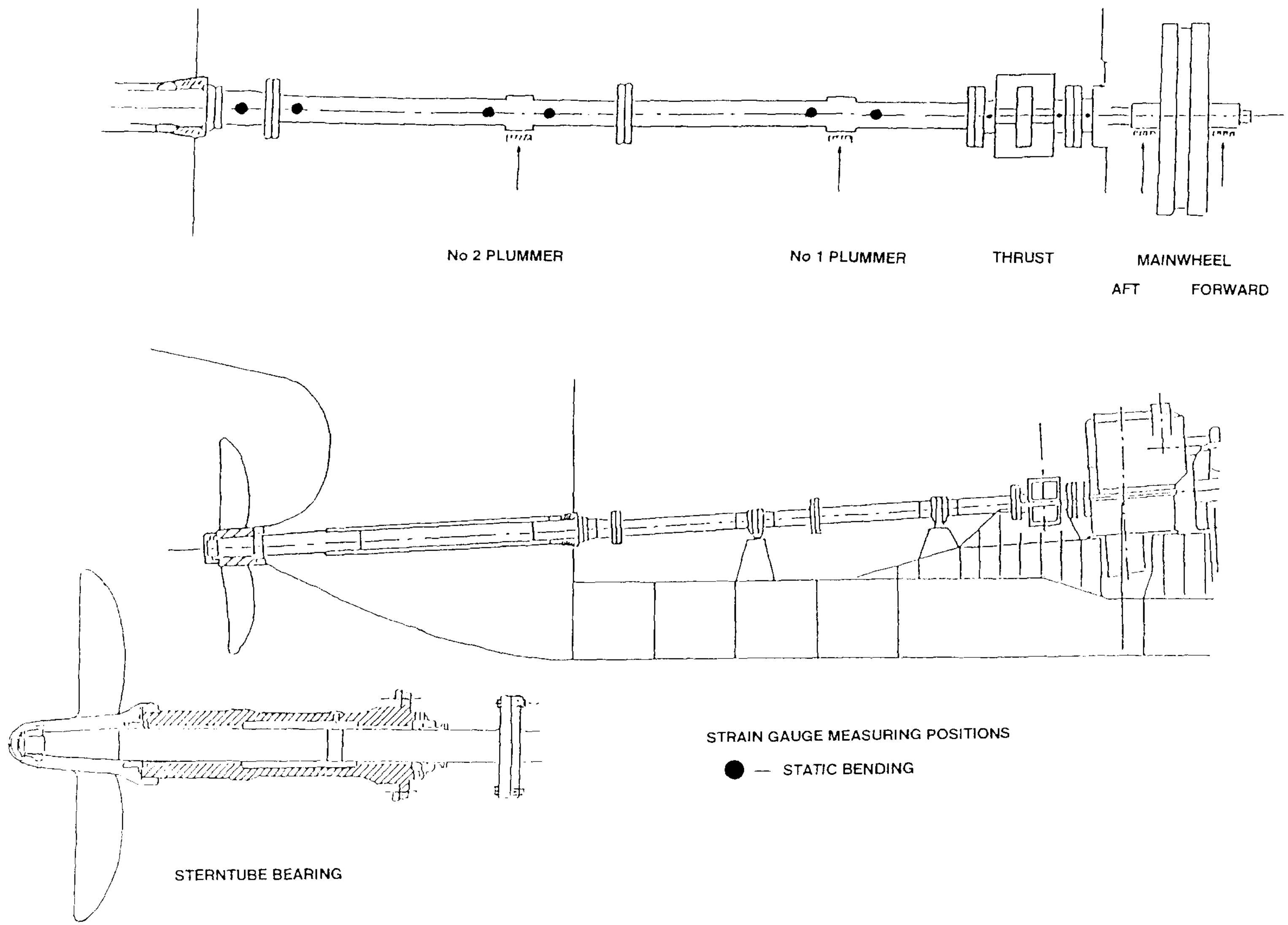


FIG. 7—TYPICAL ALIGNMENT STRAIN GAUGE CONFIGURATION

This method of alignment is particularly useful in investigation situations. However, it does have some disadvantages, as indeed do the other more commonly employed methods of jacking and measuring gaps and sags. In line with the Departmental philosophy strain gauge alignment results would, where possible, always be confirmed with a jacking trial as the two methods tend to complement each other. Some relative merits of the three methods are compared in Table 2.

TABLE 2—*Some relative merits of alignment methods*

Method	Advantages	Disadvantages
Strain Gauge Method	• Shafting is fully coupled • Journals are in their normal static attitude in relation to the bearing surfaces • Gives results for inaccessible bearings • Gives vertical and transverse alignment results • Quantifies the loads imposed by the propulsion shafting on direct drive diesel crankshafts • Easily applicable to investigate the effects of draught and machinery temperature	• Time and skill needed to fit the strain gauges • Arrangements are needed to turn the shaft • Relies on a theoretical calculation • Does not detect a bent shaft
Jacking of Bearings	• Shafting is fully coupled • Uses simple equipment • Gives information that can be used to estimate the bearing adjustments • Can detect a bent shaft	• Can be used only for accessible bearings • Is impractical for transverse bearing loads • Support conditions at adjacent bearings are affected as the shaft is raised. • Hysteresis can cause significant uncertainty in the results: either from the friction in the jack seals or between the journal and bearing surface. Can be partly counteracted by using a load cell rather than the jack hydraulic pressure. • Interpretation of the results can easily be erroneous if adjacent bearings become unloaded as the shaft is raised. • Theoretical calculations are needed to derive correction factors if the jack is not positioned close to the bearing, or the shaft span to an adjacent bearing is small.
Gaps and Sags	• The most convenient way of establishing the alignment of the machinery relative to the sterntube bearings during newbuilding • Simple measurements. • Addresses vertical and transverse alignment.	• Shafting must be uncoupled. • Temporary supports are often needed. • Coupling flanges must be in good condition—uncorroded. • Shaft alignment is sensitive to measuring tolerances and small errors.

Gear tooth strain measurement

Within a typical marine gearbox there are many mechanical and thermal interactions; for example, between the gearbox housing and hull structure, the relative attitude of the various components within the gearbox, thermal effects due to loading or lubrication systems, manufacturing deficiencies and the power transmitted. The traditional method of examining the contact markings across gear teeth is by the use of Engineers' Blue or hard lacquer. The disadvantage of these methods is in the variations of load distribution that may be encountered as the transmission torque load is increased throughout the power range or, alternatively, during transient conditions such as propeller ice contact. To overcome these types of problems the techniques and advantages of measuring the root strains in the teeth of the pinions or wheels of gear trains were further developed from the original work of PINNEKAMP.[5]

To monitor the operating strain distribution across the teeth would require many channels of strain gauge instrumentation. To address this consideration an elegant use of the Wheatstone bridge has been evolved, FIG. 8. Strain gauges are mounted in the roots of successive teeth, outside of their respective areas of contact, and connected into the bridge circuit. The signature which results as the wheel rotates takes the form shown in FIG. 8. Thus four, or even eight, strain values depicting the general tooth root stress distribution can be obtained from one channel and, if non integer tooth ratios are present, the differences between the meshing of individual teeth can be distinguished. Most importantly, the effects of load sharing, alignment and helix mis-match can be monitored throughout the speed range. The installation of strain gauges in the roots of gear teeth requires the use of miniature gauges, of about 1.0 mm gauge length, together with high levels of expertise; particularly as the work is often done with the gears in situ. The measured strains are transmitted from the rotating gear component by means of radio telemetry and (FIG. 9) shows an installation of such a system.

To interpret the large quantities of measured data satisfactorily, computer based statistical techniques are employed in order to separate the mean load distributions from the tooth to tooth variations inherent in the meshing process. In these trials the absolute strain levels are of less importance than their qualitative variation with loading and running conditions.

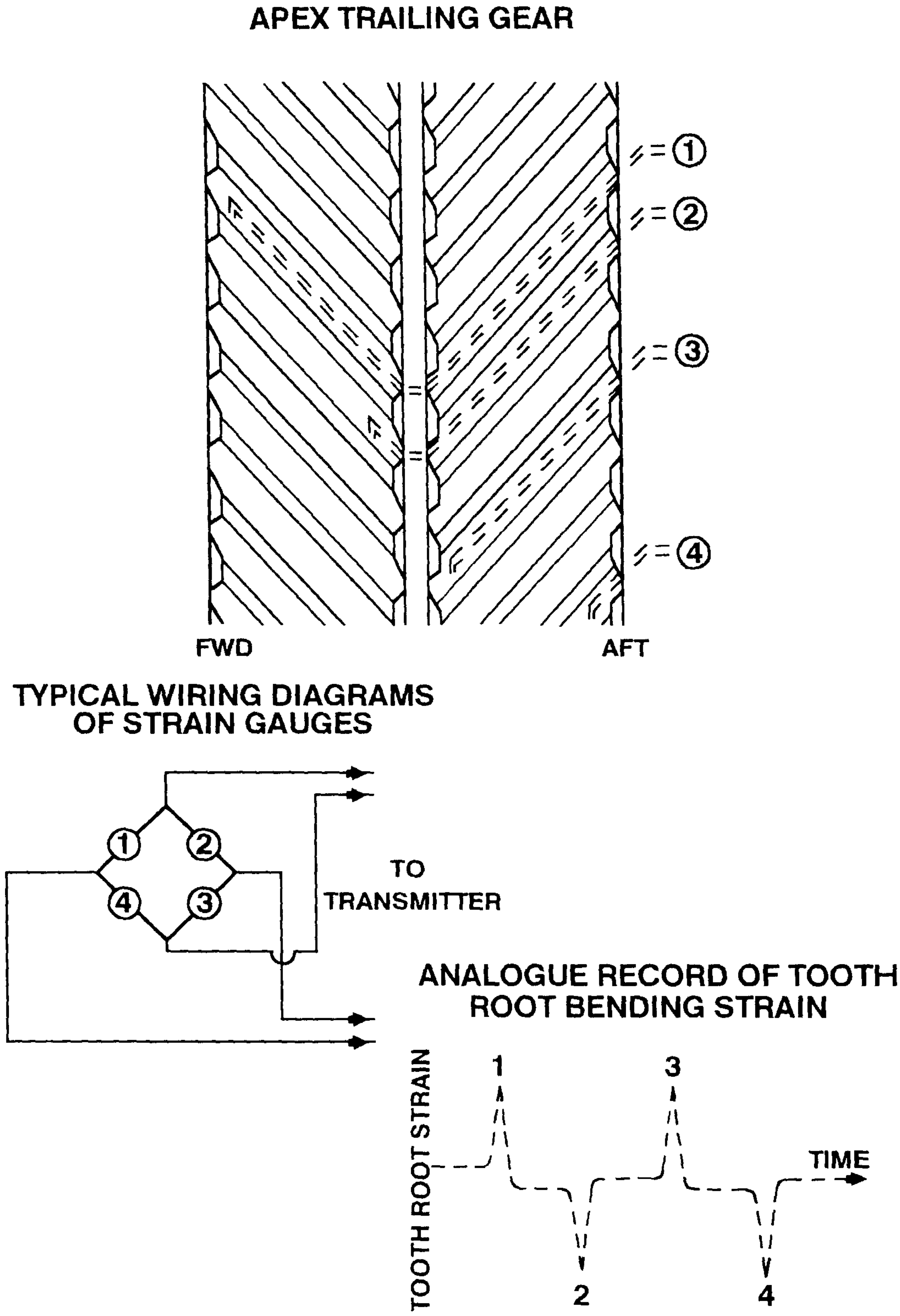


FIG. 8—GEAR TOOTH STRAIN MEASUREMENTS

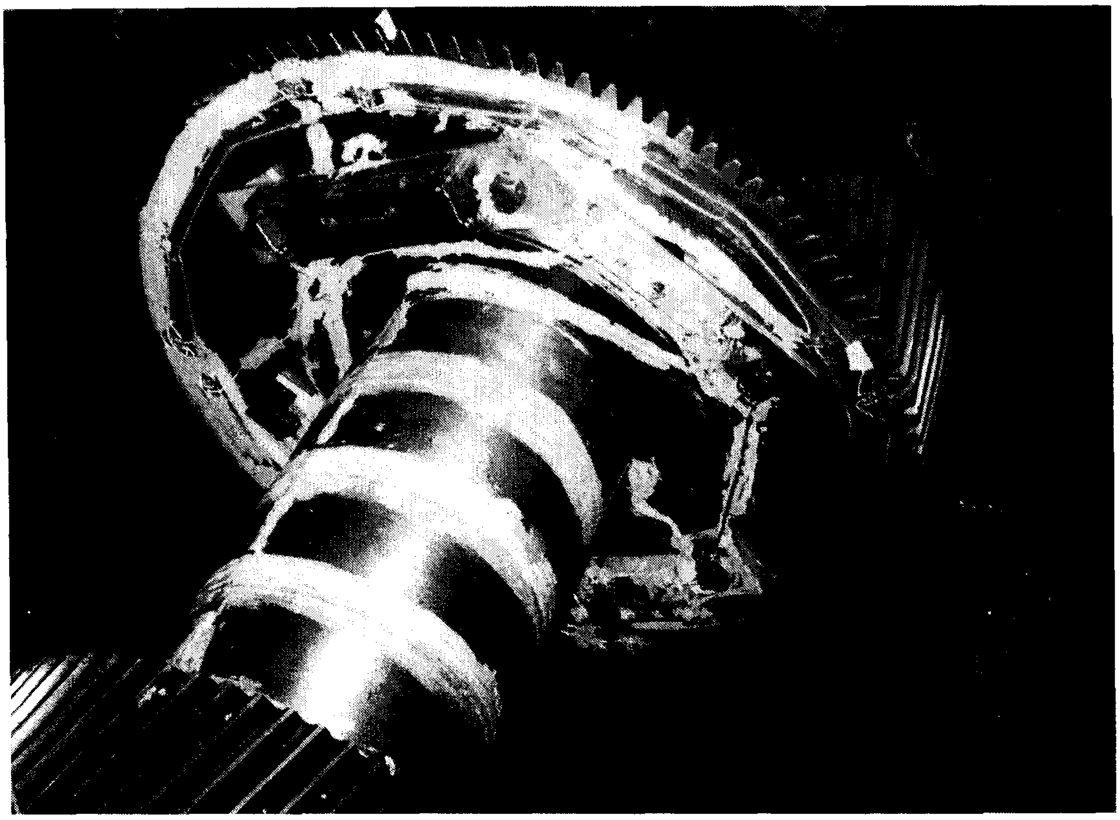
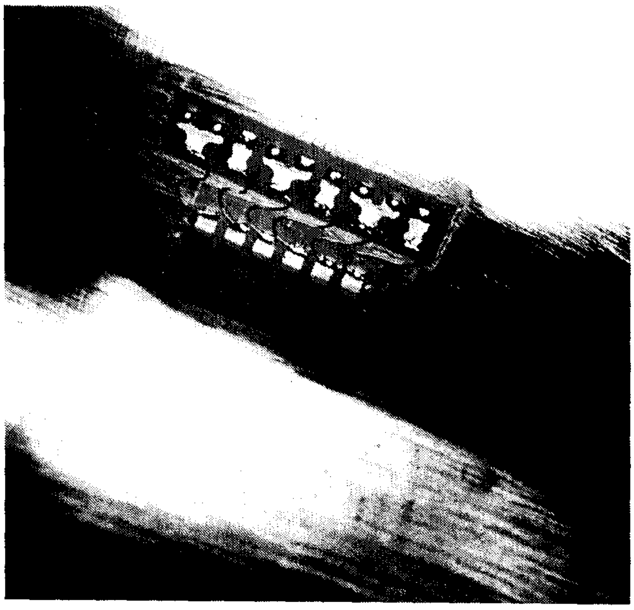


FIG. 9—STAGE IN THE INSTRUMENTATION OF GEAR WHEEL

Propeller blade strains

The measurement of propeller blade strain has long been considered a difficult problem. The traditional method has been to require the centre boring of the tail shaft in order to bring the instrumentation signal leads inboard from the propeller so that they can be transmitted from the rotating shaft by means of telemetry or slip rings. In the case of TID the last use of slip ring technology was in the early 1970's. To simplify the measurement procedure in overcoming the centre boring problem and also increase the reliability of the strain gauge installation, a three year programme of work was commissioned in the TID laboratory. This involved the development of a suitable underwater telemetry technique and the extensive testing of waterproofing, protective compounds and installation procedures in order to protect the strain gauges and their associated leads from the ingress of sea water over a period of months and attack from cavitation.

To carry out blade strain measurements the propeller is normally marked out and gauged in a dry dock, preferably with the propeller removed from the shaft although this is not a constraint on the use of the method. Following the gauging process the waterproofing and cavitation resistant coatings are applied. This centres around a two-coat protective system for the strain gauges and their signal leads. The signal leads from the gauges are connected to individual FM transmitters housed in a purpose designed casing mounted on the propeller boss at either the forward or after end, depending on the space available. From there the signals are led to a common transmitting aerial mounted on the forward end of the boss under the rope guard. Facing it, is a receiving aerial which can be mounted up to 20 mm distant. The FM signals are transmitted across the water gap, and then conducted through leads, protected by metal conduit, around the stern of the ship to the recording instrumentation. The collage of pictures in (FIG. 10) shows the various



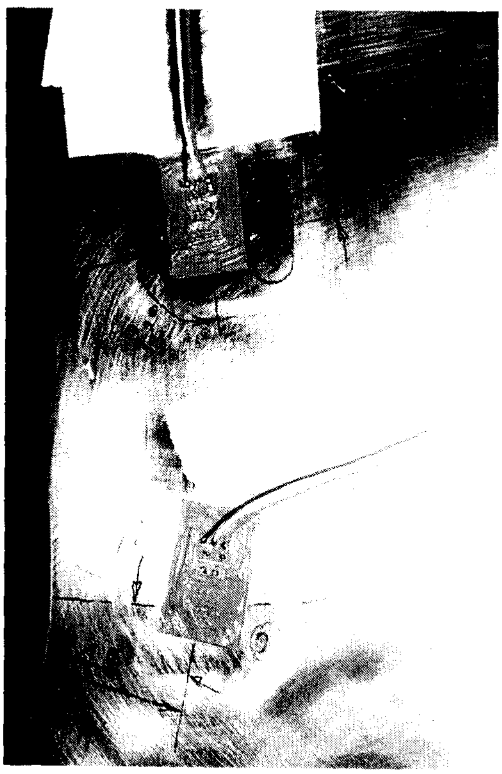


FIG. 10—STAGES IN THE INSTRUMENTATION OF PROPELLER BLADES

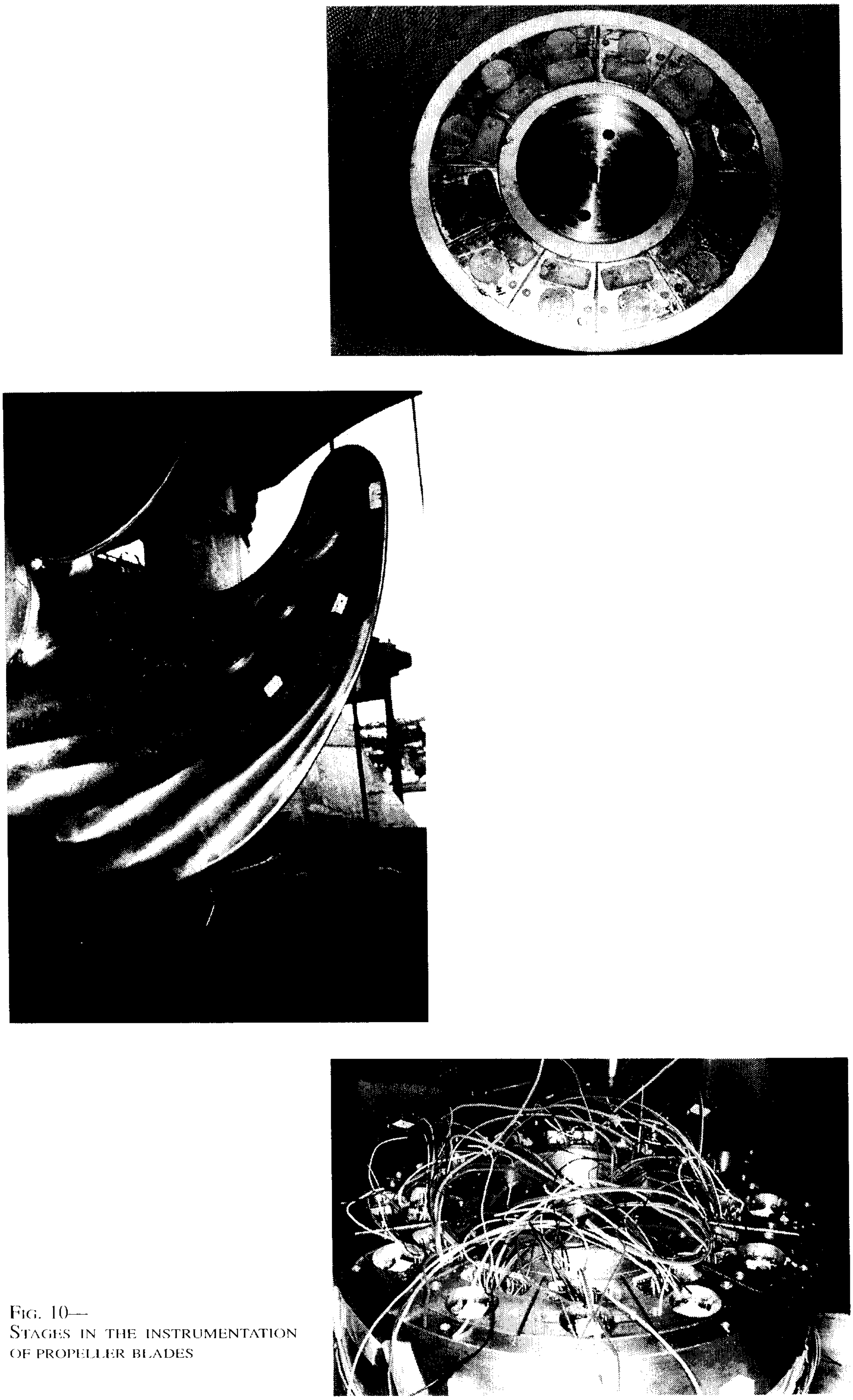


FIG. 10—
STAGES IN THE INSTRUMENTATION OF PROPELLER BLADES

instrumentation stages. Although subjected to hostile conditions and installed under less than ideal conditions, the protective encapsulation on the propeller blades generally survives for several months at sea. Notwithstanding its successful use in measuring propeller blade strains, the method has also been used for measuring bending strains in the shafting aft of the stern seal and can be applied to other similar situations in hostile environments.

Data Transmission

Long term measurement exercises can gather large amounts of data but in today's economic climate can be expensive if continual attendance by a TID Surveyor is required. In recent years much work has been undertaken within TID to harness the power of modern communication, data logging and computational systems in order to provide effective remote investigation capabilities. Typical of an application of the results of this initiative is the measurement of the response of a marine structure to environmental and operational loading in another European country, but controlled and monitored from the Department's laboratory in the UK. This type of approach offers distinct advantages in the context of minimal attendance on site for certain kinds of investigation and monitoring activities at sea and on land. Furthermore, the reliability of these kinds of method, when used in conjunction with mixed mode instrumentation, has been shown to be satisfactory. (FIG. 11) shows a typical installation using this approach.

Further developments in the related field of digital image transmission have also taken place. These now permit an investigation surveyor at a remote location in the world to consult, if he so wishes, an expert colleague at the Department's base in Croydon with the aid of pictorial images of the failure being considered. This is achieved using a real time, digital image link with two way communication by means of any convenient telephone link. (FIG. 12) shows just such a case relating to a crankshaft failure with the inset to the Figure illustrating the level of detail that can be transmitted with the appropriate equipment.

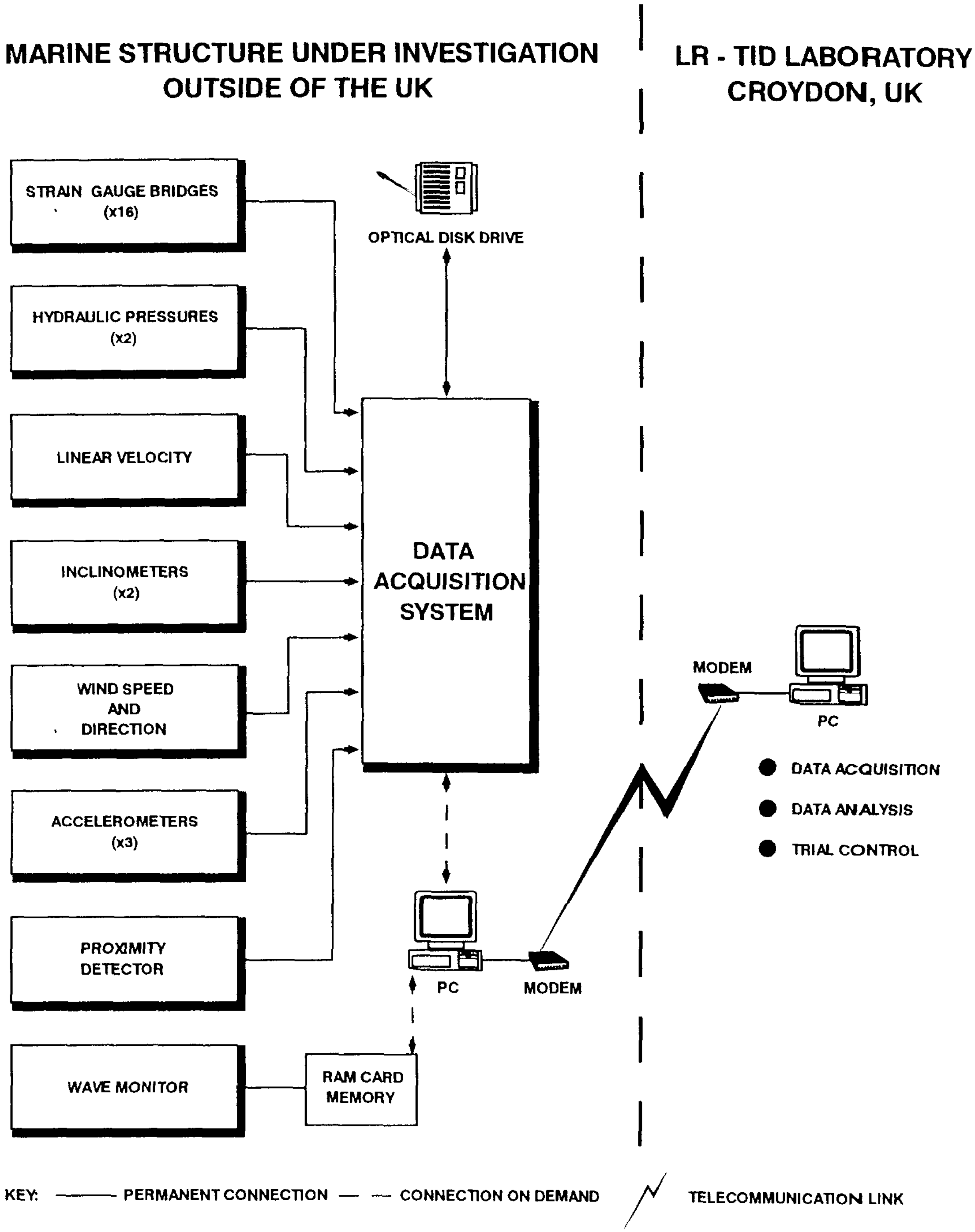


FIG. 11—REMOTE DATA ACQUISITION SYSTEM

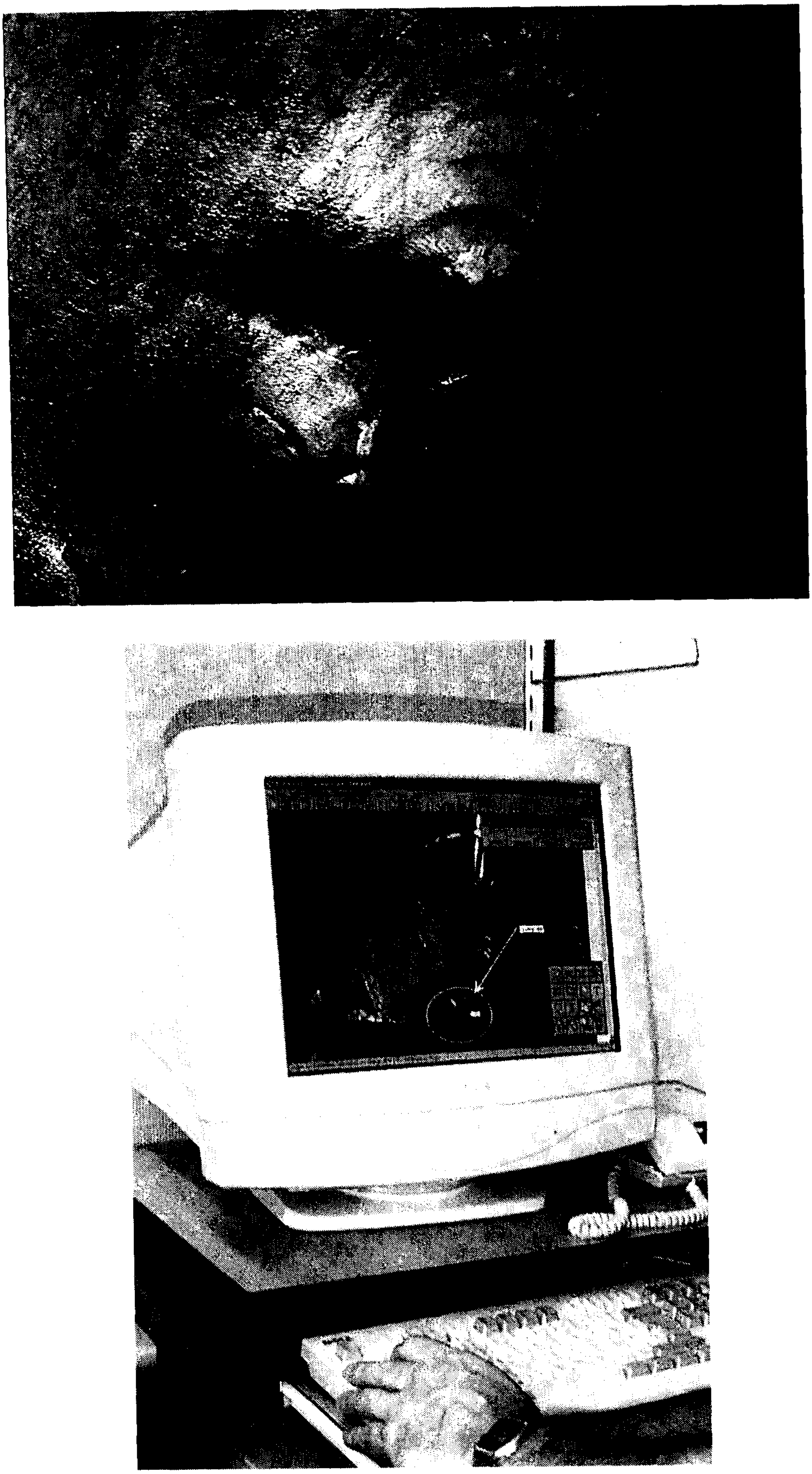


FIG. 12—DIGITAL TRANSMISSION OF VISUAL INFORMATION FROM A REMOTE LOCATION

Development of computational methods

To complement its measurement capabilities the Department maintains a strong engineering analysis capability to satisfactorily undertake the range of investigations presented to it. The essential feature, however, of any computational method is that it has been properly validated and correlated with the results of model or full scale experience. By way of example of the divergence that can occur between the results of computations obtained from reputable organisations, (FIG. 13) shows the propeller blade stresses calculated using finite element methods for a highly skewed propeller blade:[6] the experimental values for which were obtained from work done in TID using a 254 mm model of a 72 degree biased skew propeller under the action of point loading. As a consequence, it is of the utmost importance that correctly validated codes are used as a basis for engineering analysis.

Computational capabilities within the Department include the normal range of linear and non-linear finite element, fracture mechanics, modal analysis and computational fluid dynamics capabilities and the group is continually active in extending the applicability and validity of these approaches to new engineering problems. However, apart from the standard commercial computational packages, TID has from its earliest days been deeply involved in the development of analysis capabilities both for its own use and also for the wider use within LR. Typical of this has been the standard alignment and torsional, axial and lateral vibration capabilities for LR; indeed this is an ongoing process into which the developing methods of analysis and the advances in computer science are regularly being incorporated.

Specific computational developments include a propulsion hydrodynamic analysis suite, which embraces simple regression based methods through wake scaling procedures and surface panel methods to a vortex lattice propeller analysis capability which is able to estimate blade cavity extent, hull surface induced pressures and blade stresses under normal operation and in ice; a shipboard and offshore accommodation module noise prediction procedure;[7] the MERLIN thermodynamic system steady state and transient simulation suite[8] and a waterjet analysis method to determine the hydrodynamic and strength characteristics of the propulsion unit and its auxiliaries.[9]

SYMBOL	ELEMENT TYPE	No. OF ELEMENTS	SYMBOL	ELEMENT TYPE	No. OF ELEMENTS
■	3 - DIMENSIONAL HEXAHEDRA	70	■	2 - DIMENSIONAL TRIANGULAR PLANE	161
▲	SUPERPARAMETRIC THICK SHELL	97	□	2 - DIMENSIONAL TRIANGULAR PLANE	161
◇	3 - DIMENSIONAL HEXAHEDRA	160	●	3 - DIMENSIONAL HEXAHEDRA	300
△	ISOPARAMETRIC SHELL	161			

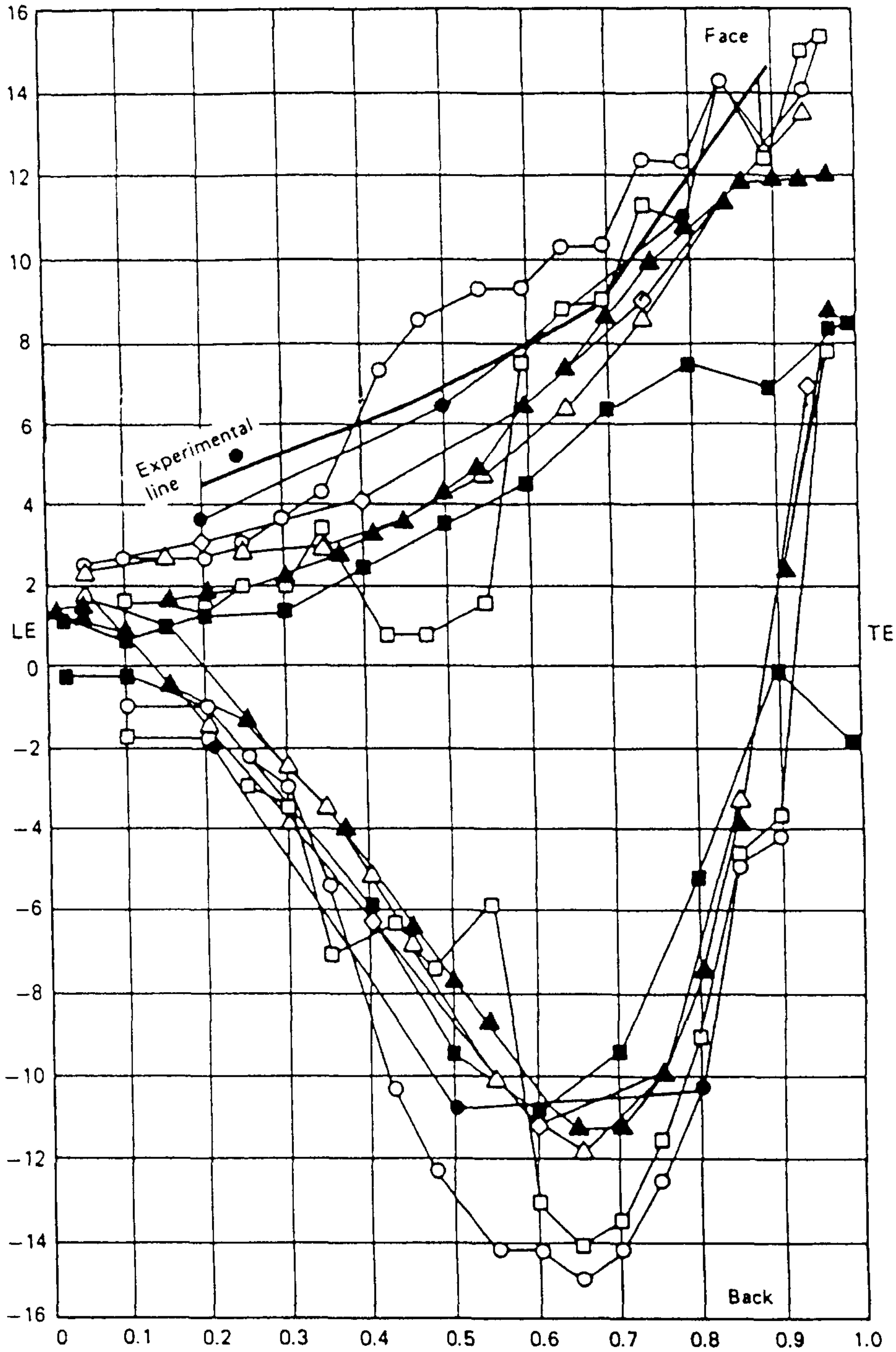


FIG. 13—COMPARISON OF DIFFERENT PROPELLER BLADE STRESS CALCULATIONS WITH EXPERIMENT

Noise Prediction

TID's noise prediction software has been continually developed over the last twenty years and is used for the majority of the prediction calculations. Noise abatement measures are widely applied in a ship to control the noise and, as such, the calculation also considers the noise reductions achieved. It comprises three modules covering airborne, structureborne and HVAC noise calculations. (FIG. 14) shows the flow chart of the program.

The airborne noise calculation module is based on the assumption that the airborne overall sound pressure level in the receiving space is equal to the combined effect of the sound power levels emanating from all significant noise sources. The partition insertion loss in way of the noise transmission path and the acoustic properties of the receiving space are considered in the process.

The major assumption in the structureborne noise calculation module is that vibratory energy transmits into the structure from noise sources in the form of acoustic frequency vibration. The attenuations are considered to be concentrated at structural discontinuities, such as deck plate stiffening, junctions of deep frames and bulkheads, and junctions of shell plating with decks, along the path between the source and the receiving end. Studies of data collected through measurements provide nominal values, and as a rough guide for merchant ships, structureborne intensity reduces on average by about half a decibel per frame. Structureborne noise also falls with vertical distance from the source; for example, the noise reduction on the first two decks is approximately 5 decibels per deck and on subsequent decks is around 2 decibels per deck.

The calculation of HVAC noise depends on a knowledge of the sound power output of the fans in the air conditioning system. The noise emitted from air outlets in the receiving space will depend on the amount of noise attenuation due to the:

- Length of ducting,
- Types of duct branches,
- Air flow rates and
- Characteristics of any silencers fitted.

The sound pressure level at the receiver is calculated in the frequency range 63Hz to 8kHz from the results of these individual calculations. It has been demonstrated that the results of the analysis procedure when compared to full scale measurements made by the Department on a range of ships and platforms generally lie within a range of around 3 dB(A).

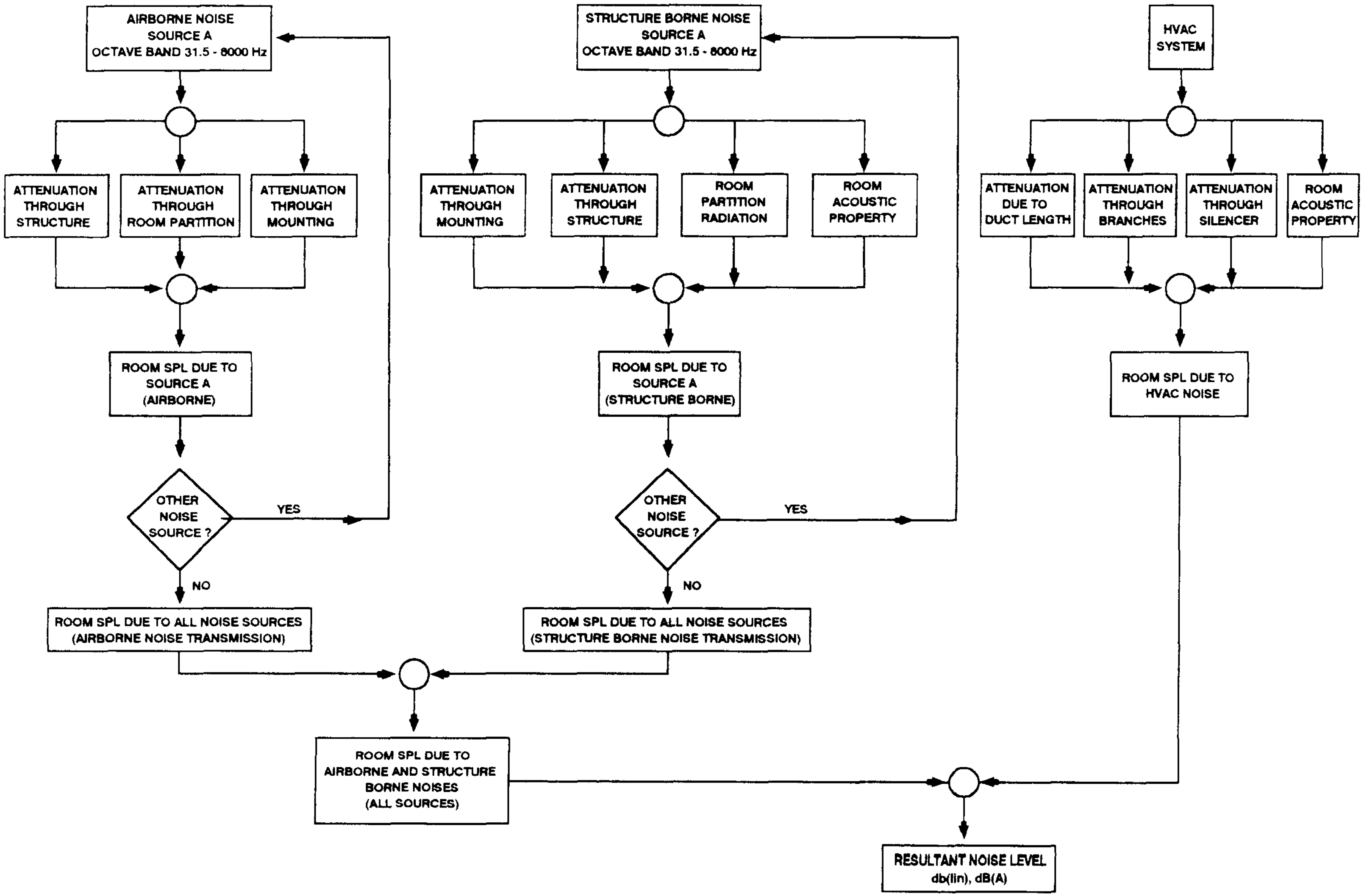
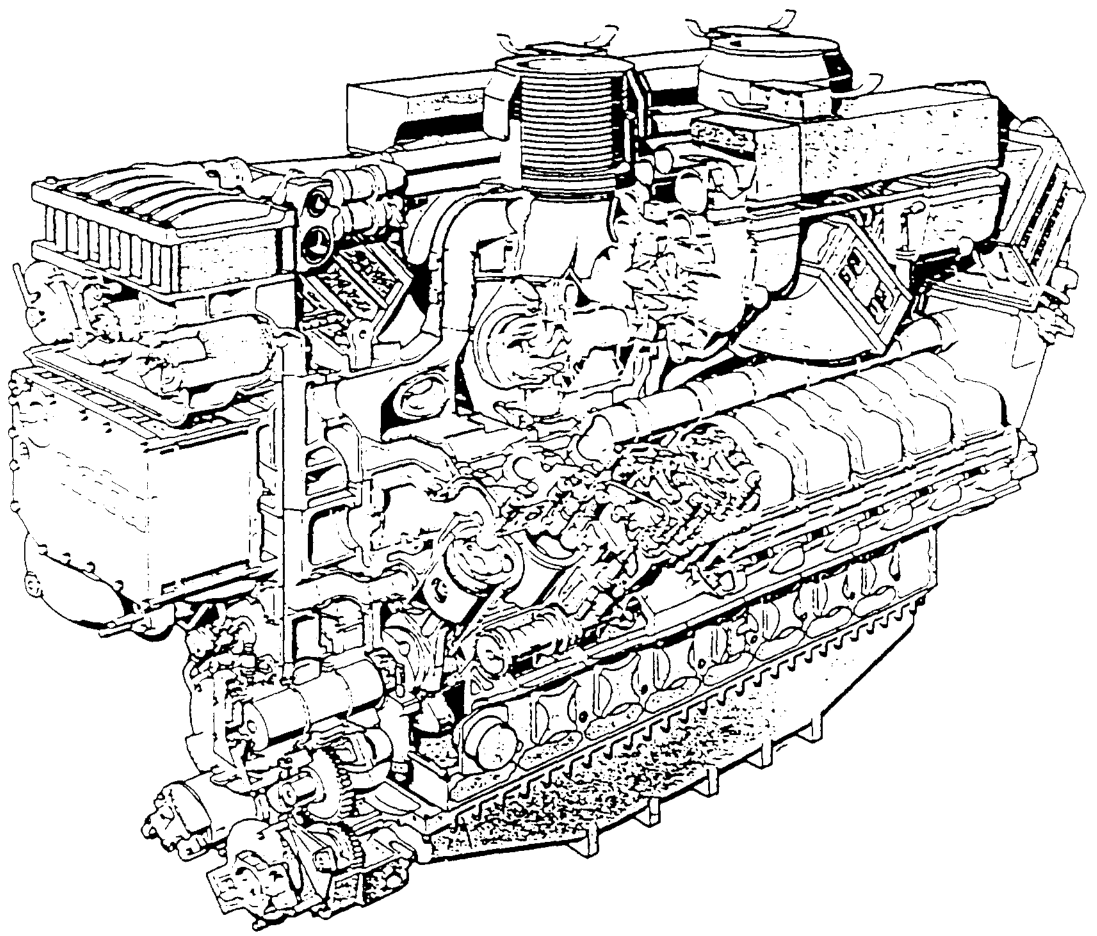


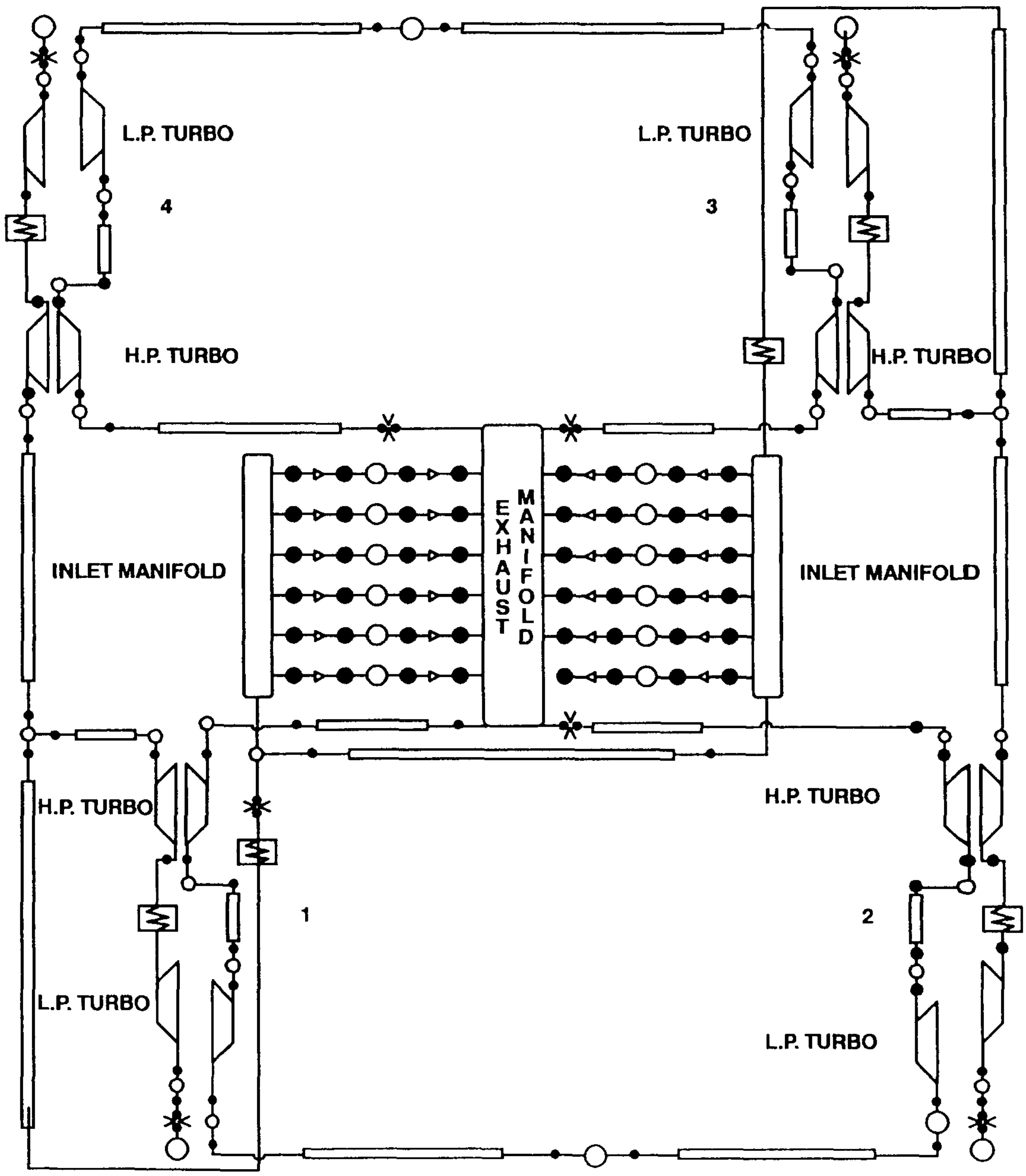
FIG. 14—NOISE PREDICTION CALCULATION PROCEDURE

MERLIN Suite

The MERLIN-DEEPC software is a one dimensional, transient capability primarily developed for engine performance evaluation and exhaust gas emissions prediction from a wide range of diesel engine and pressure charging systems. The range of engine types that can be modelled embraces high, medium and slow speed engines together with flow path modifications such as, for example, exhaust gas recirculation. (FIG. 15) shows a typical MERLIN idealization of a diesel engine while (FIG. 16) shows a comparison of the experimentally measured and predicted results for a two-stroke marine diesel engine.

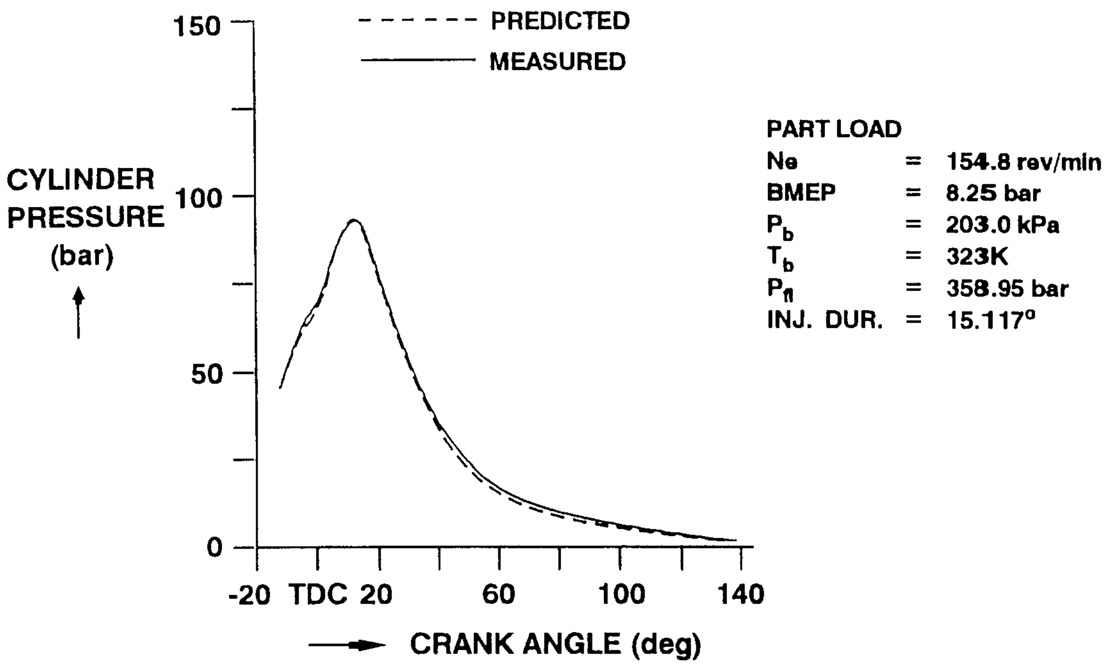


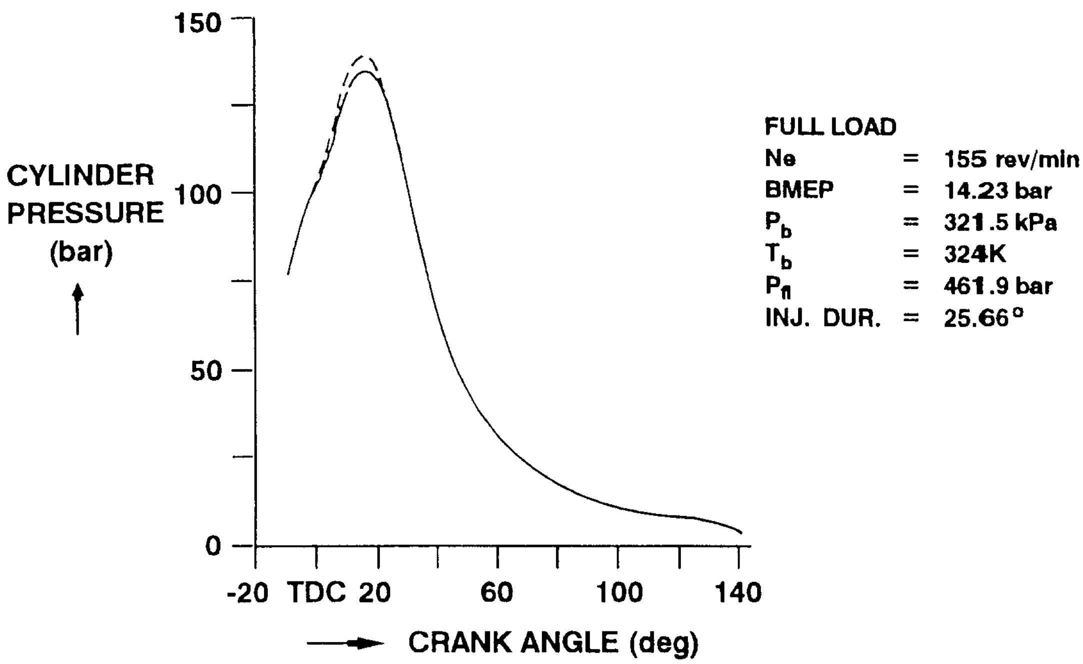
THE TWO STAGE MTU SERIES 595 SEQUENTIALLY TURBOCHARGED DIESEL ENGINE (16 CYLINDER)



SCHEMATIC REPRESENTATION OF THE MTU SERIES 595 ENGINE (12 CYLINDER)

FIG. 15—TYPICAL IDEALIZATION OF A MARINE DIESEL ENGINE


PREDICTED
MEASURED
CYLINDER PRESSURE (bar)
150
100
50
0
-20 TDC 20 60 100 140
CRANK ANGLE (deg)
PART LOAD
Ne = 154.8 rev/min
BMEP = 8.25 bar
Pb = 203.0 kPa
Tb = 323K
Pfl = 358.95 bar
INJ. DUR. = 15.117°


CYLINDER PRESSURE (bar)
150
100
50
0
-20 TDC 20 60 100 140
CRANK ANGLE (deg)
FULL LOAD
Ne = 155 rev/min
BMEP = 14.23 bar
Pb = 321.5 kPa
Tb = 324K
Pfl = 461.9 bar
INJ. DUR. = 25.66°

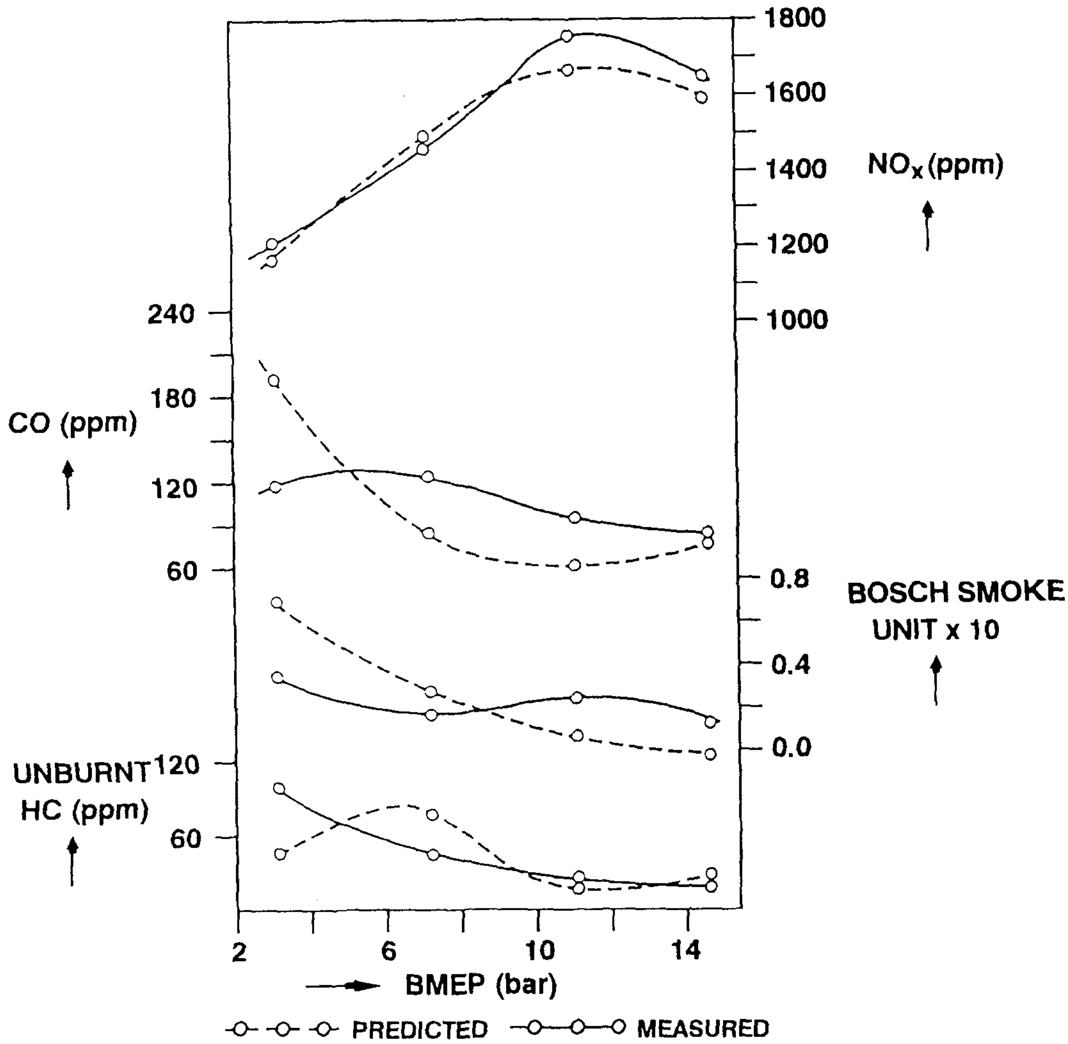


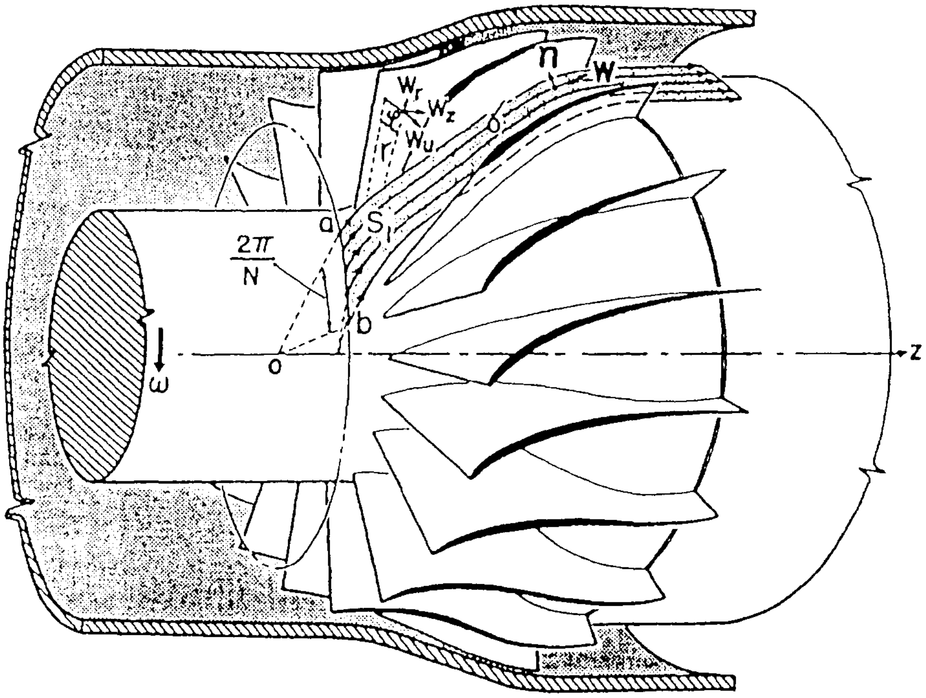
FIG. 16—COMPARISON OF MEASURED AND PREDICTED CYLINDER PRESSURE AND EXHAUST EMISSIONS FOR THE TWO-STROKE ENGINE AT FULL SPEED, EXPRESSED ON VOLUMETRIC, DRY BASIS AND CORRECTED TO 15% OXYGEN

The MERLIN capability was not written for any specific engine configuration or thermo-fluid system. Instead, it is based on the philosophy that the user is allowed to build-up the system of interest, no matter how complex, by specifying a unique connectivity from a series of primary engineering building blocks representing commonly used components such as compressors, turbines, cylinders, valves, pipes, volumes, orifices, and so on. An appropriate combination of these building blocks can be selected to model either internal combustion engines or other thermo-fluid systems. The selected combination of the building blocks can be linked together by specifying node numbers at the connection of one building block to another. MERLIN understands this information and automatically couples the required mathematical models to solve the system specified. Within the suite of programs is a pipe wave dynamics capability which can be used to investigate the properties of single or branched systems incorporating uniform or tapered members.

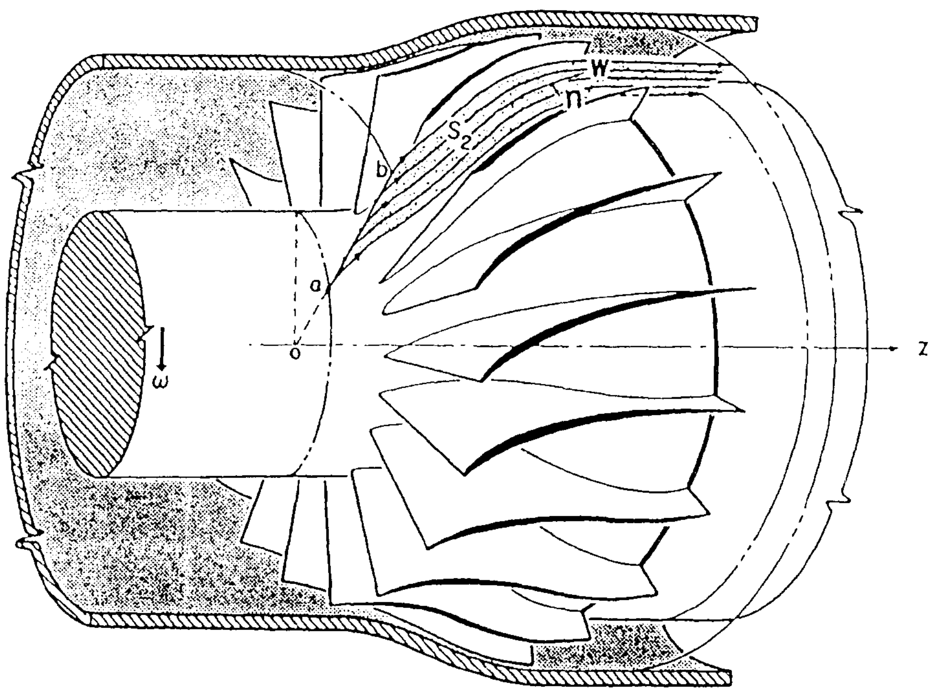
Waterjet Analysis

Due to the recent increased usage of waterjet units and the associated increase in installation powers and, hence, size, a method for the hydrodynamic and strength analysis of waterjet units has also been recently developed in TID. The method was based on a modification of WU's original comprehensive theoretical formulations, including the through-flow analysis model in which turbo-machinery flows are specified in terms of stream functions on orthogonal meridional stream surfaces, designated S1 and S2, as shown in (FIG. 17).

Within the code, the waterjet's axi-symmetric channel may be of arbitrary shape, for which a finite difference mesh is automatically generated and the flow velocities are determined from first derivatives of the stream function and the stream surfaces are determined by interpolation between the finite difference grid points. Offsets of the camber surface are then obtained at intersections between the co-axial stream surfaces and the rotor and stator blades. During successive iterations new stream surface locations and energy transfers are determined by including the influence of the rotor and stator rows in the principal equations. When a converged solution is obtained from the finite difference scheme and camber surface model, the full aerofoil sections are determined from intersections of the final stream surfaces with the rotor and stator blades. A cascade flow analysis is then performed on each stream surface using an approach based on Martensen's surface vorticity formulations which leads to a more accurate assessment of the boundary layer development, cavitation performance and tendencies for flow separation, particularly in the stator row.



THROUGH-FLOW MERIDIONAL STREAM SURFACES S1



THROUGH-FLOW MERIDIONAL STREAM SURFACES S2

FIG. 17—FLOW SURFACES OF A WATERJET ANALYSIS

During the development of the present hydrodynamic model, the kernel finite difference schemes were validated against published case studies and specific CFD studies. The surface vorticity used in the cascade model was verified against published aerofoil and cascade data, while the torque characteristics of a waterjet unit were verified by comparison with data from sea trials undertaken by the Department. (FIG. 18) shows the predicted and measured power consumption of a three bladed waterjet unit fitted to a high speed catamaran.

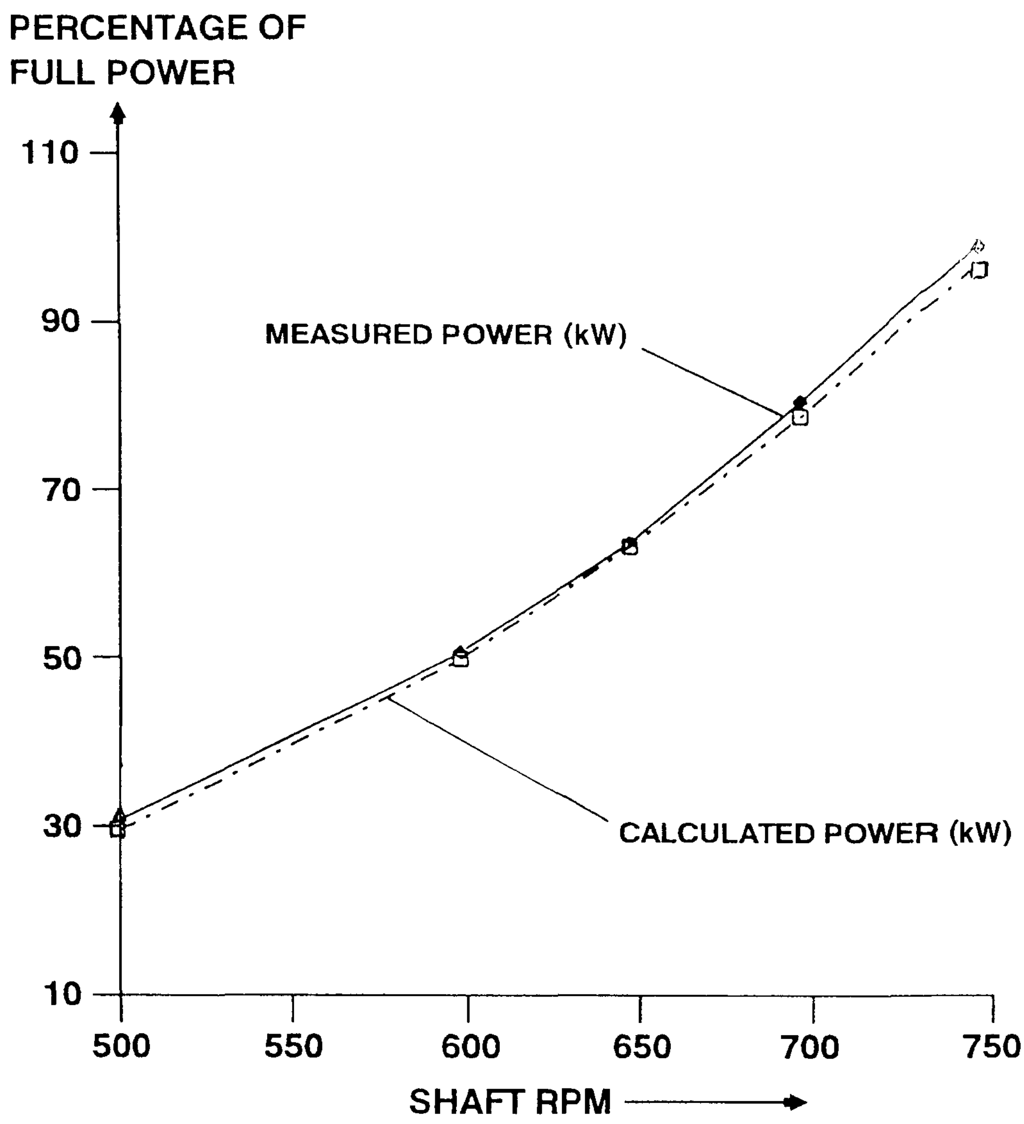


FIG. 18—COMPARISON OF MEASURED AND PREDICTED POWER ABSORPTION

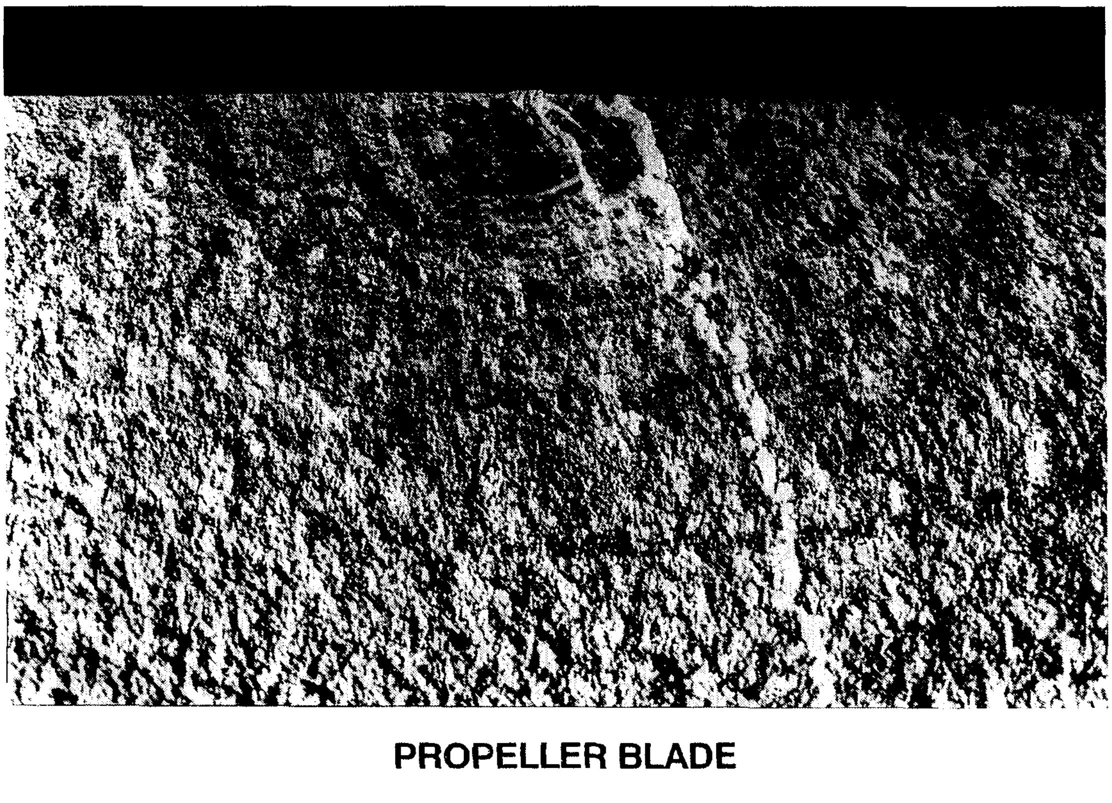
PRINCIPAL FAILURE MODES

The majority of failure investigations encountered today involve aspects of fatigue. Other failure modes such as brittle failure and creep are less frequently encountered with pure ductile failures being a rare event.

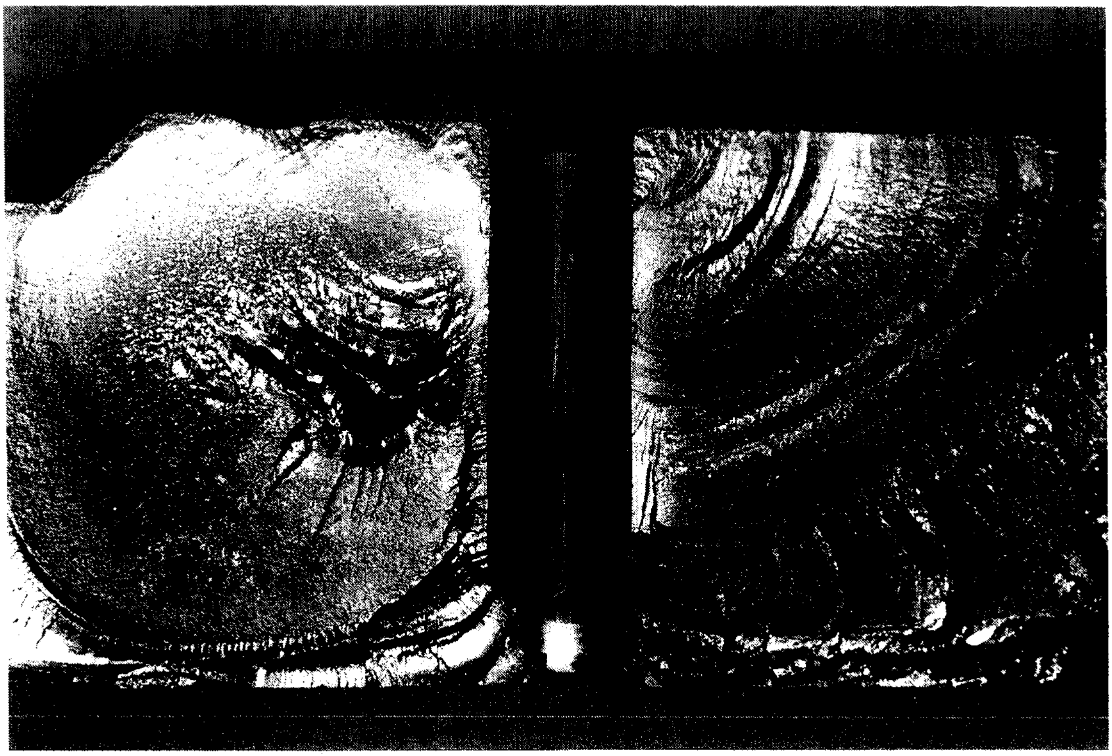
Fatigue failures

Fatigue failures require cyclic stresses to be experienced by a discontinuity in the material. Furthermore, the fatigue life of the component will be influenced by the magnitude and nature, compressive or tensile, of any steady mean stresses. The material type, local geometry, mode of manufacture and the environment in which a component works also affect the fatigue strength.

Fatigue failures are normally characterised by being transcrystalline and occur without significant plastic deformation. Moreover, the general appearance of a fatigue failure, as distinct from its orientation, will be similar for most modes of loading, for example, whether the failure has been induced by axial, bending or torsional loading or a combination of these. (FIG. 19) shows two typical fatigue failures, one on a propeller blade and the other on a diesel engine gudgeon pin. Fatigue failures are normally considered to progress as three distinct phases in ductile materials:



PROPELLER BLADE



GUDGEON PIN

FIG. 19—TYPICAL FATIGUE FAILURES

The first phase

Stage 1, is strongly influenced by the slip characteristics of the material, the applied stress level, the extent of crack tip plasticity and the characteristic microstructural dimensions of the material. Where cracks are initiated in ductile materials it is generally considered that the cracks grow cyclically by the deformation in the slip bands near the crack tip which leads to the creation of new crack surfaces by the mechanism of shear de-cohesion. In components with relatively small flaws it has been found that cracks can spend a considerable time in this mode of the development. Typically, for a propeller blade this stage might account for some 80 to 90% of the crack life.

The second phase

Stage 2, is where the plastic zone at the crack tip extends over many grains due to the higher stress intensities. This is in contrast to the **Stage 1** mechanism which embraces only a few grains at a time and is essentially a single shear type of mechanism in the direction of the primary slip system. For **Stage 2** crack growth the process involves simultaneous or alternating flow along two slip systems in the material. This duplex slip mechanism results in a planar crack path which is normal to the far-field tensile strain direction and hence defines the orientation of the crack face. For components or structures where there are significant initial flaws the major part of the fatigue life is spent while the crack is growing in this mode. Consequently, for welded structures **Stage 2** crack growth laws and codes are especially applicable. One frequent characteristic of a **Stage 2** fatigue failure is the presence of striations in the fracture surface the spacing of which, within the applicability of the Paris Law regime of crack growth, has been shown to correlate with the measured average rate of crack growth per cycle. (FIG. 20) shows a set of striations relating to a fatigue failure in the tooth of a gear wheel of an ice breaker. Of particular interest is the relatively coarse nature of some of the striations in relation to the normally observed magnifications of around 1000 to 4000. However, it is important to note that striations are not always present and it has been shown that environmental effects can influence their development. In air, for example, striations are clearly seen in pure metals, some ductile alloys and in many engineering polymers, but this is not always the case in steels and they can often be indistinct in cold worked alloys. In a vacuum, striations are not seen in a number of alloys which would normally be expected to exhibit them in air, furthermore, crack growth rates can be an order of magnitude slower.

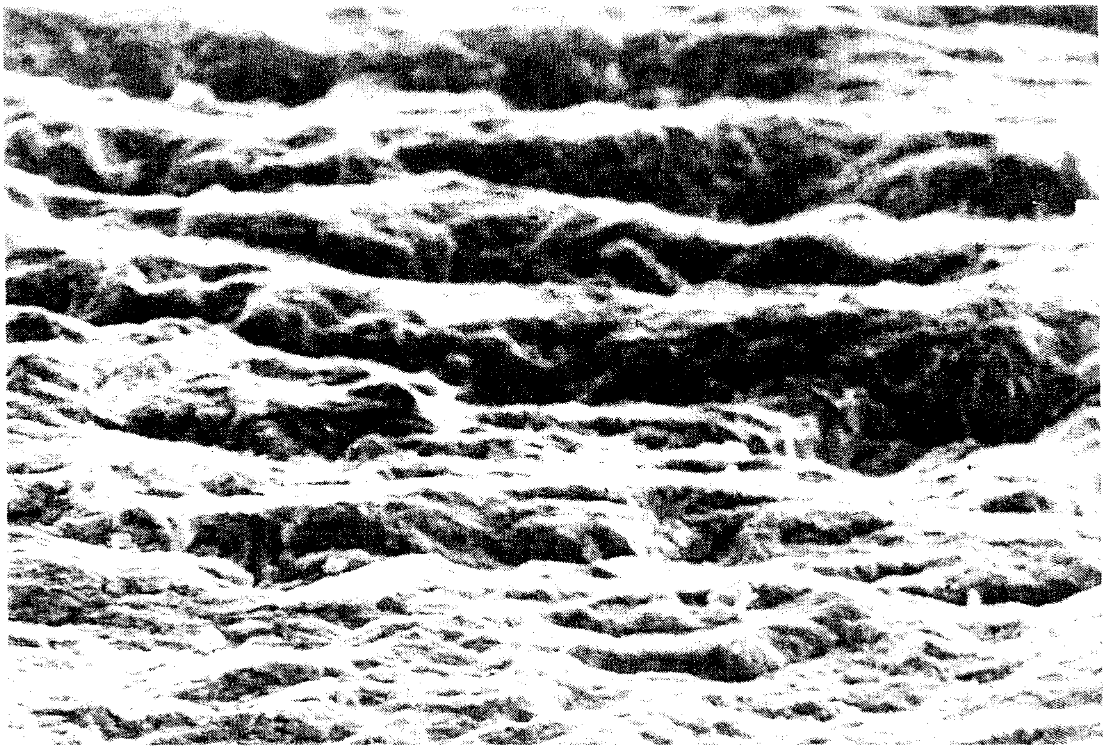


FIG. 20—FATIGUE CRACK STRIATIONS ON A GEAR TOOTH FAILURE (x500)

The final phase

Stage 3 and this is where the crack has grown to a sufficient size where the component can no longer withstand the mechanical loads imposed upon it and, therefore, it fails by another mode. This final phase in the fatigue crack life is normally very short, of the order of microseconds.

Striations should not be confused with beach marks which are often clearly visible on fatigue fracture surfaces, (FIG. 19). Beach marks are normally associated with periods of crack arrest whereas striations can be thought of as being due to the plastic blunting of the crack tip upon each cyclic application of tensile stress.

The propensity for crack growth is influenced by the environment due to the deleterious effect of corrosion when combined with stress. If the stresses are sensibly constant, as for example in a pressure vessel, the corresponding failure mechanism is usually termed stress corrosion cracking whereas the term corrosion fatigue is applicable when cyclic stresses are predominant. In practice, such clear distinctions are unusual because all components experience some stress fluctuations and significant residual tensile stresses can be induced by manufacturing processes. All environments are corrosive to some degree, even air and pure water, and it is interesting to note that the fatigue strength of samples is increased if they are tested in a vacuum. The crucial function of the corrosive action is its contribution in overcoming the stronger microstructural barriers during the early stages of crack growth. Such action is a chemical function (fluid composition, pH value, electropotential) and is consequently time dependent which helps explain why there is no conventional cyclic stress limit for corrosion fatigue conditions. Furthermore, fatigue life is not related solely to the number of stress cycles and the results of corrosion fatigue tests are influenced by the frequency of the applied loads. Corrosive agents include both acidic and alkaline embrittlement mediums: chlorine, sodium and sulphur are commonly encountered in marine and industrial failures which are caused by corrosion assisted cracking.

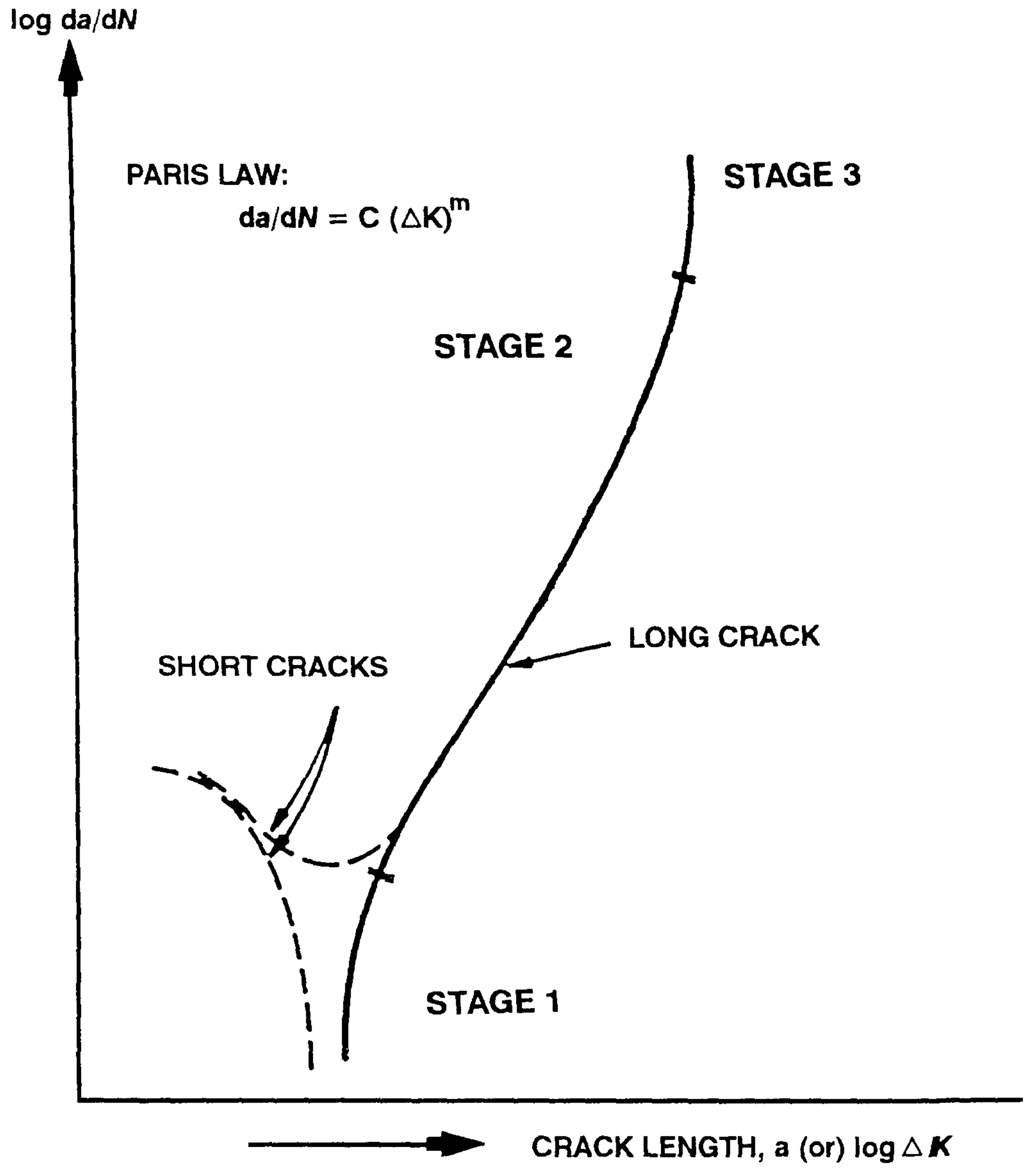


FIG. 21—FATIGUE CRACK GROWTH

Fatigue research has more recently concentrated on short cracks and it has been shown that the growth rates of small flaws can be significantly greater than those for long flaws when considered in terms of the same nominal driving force. (Fig. 21) shows in schematic form the typical fatigue crack growth behaviour of long and small cracks at constant values of imposed cyclic range and load ratio. In this context small flaws are considered in terms of one of four classifications:[10]

- Microstructurally small

  Those where the crack size is comparable to characteristic microstructural dimensions.
- Mechanically small

  Cracks where the near-tip plasticity is comparable to the crack size or which is encompassed by the plastic strain field of a notch.
- Physically small

  Cracks typically less than about 2 mm in length.
- Chemically small

  Cracks which would normally be amenable to linear elastic fracture mechanics but exhibit anomalies in growth rates below a certain size as a consequence of corrosion fatigue effects.

Indeed, while each of these classifications present some difficulties from an engineering standpoint, the physically small flaws can be the most difficult to quantify their behaviour because it has been demonstrated that these types of cracks can grow appreciably faster than long cracks subjected to the same nominal stress intensity range. The studies of short crack behaviour and non-propagating cracks mean that fatigue failures can now be explained in terms of crack growth throughout the process from the first strain cycle to final rupture using fracture mechanics principles.

Additionally, it seems likely that many familiar classical techniques and empirical factors used to estimate fatigue strength will be superseded; for example, the treatment of multiaxial stress states, notch sensitivity, stress-strain cycle counting and strain hardening or softening characteristics. A good illustration of this change of thinking is that the fatigue limit was considered previously to be identified only by a stress value. A better concept, however, is:

'the ability of a crack, whatever its length, to propagate to failure'

because the fatigue limit refers to the stress level required to overcome the strongest barrier to crack growth which will be represented by a microstructural distance.

Apart from the greater susceptibility of soft whitemetal bearing materials to crack, the effects of temperature on the fatigue strength of most engineering materials are not serious until about 400°C when all mechanical properties are progressively impaired. Temperature fluctuations can, however, cause severe thermally induced cyclic stress gradients which in turn induce fatigue damage which is manifested as crazed cracking. Typical examples are the overheating of diesel engine bearings and consequent damage to crankpins and journals and the impingement of water on the bore surfaces of superheated steam pipes.

Fretting Fatigue

Fretting fatigue is a special case of fatigue action which results from a combined mechanical and chemical action and in which three fundamental conditions are necessary for the failure mechanism to develop. These are, the ability of two surfaces to move relative to each other, albeit by a small amount; points of asperity on the surfaces which make contact and the presence of sufficiently high stresses in the vicinity of the contacting points to cause surface cracking. Typical engineering situations giving rise to the conditions promoting fretting action are flat contacting faces such as flanges where shear loads across the faces may exist and the normal forces may permit some degree of slippage between them. Alternatively, holes which, for example, may house bolt shanks or rivets and be subjected to interface sliding and varying interface pressures. Further situations include key and keyway interfaces, leaf springs, splines and contacting strands in wire ropes.

When surfaces are permitted to rub together under the conditions described, scars on the surface tend to form relatively rapidly and these often have a roughened appearance. In cases of fretting in steel the scars contain a reddish-brown oxide, often in the form of a powdery deposit although in some cases this may form a glaze. Alternatively, if the action is between aluminium surfaces then the deposit is black in colour. Research into the fretting fatigue mechanism has suggested that the nucleation of fatigue can result from one of a number of causes; typically these are an abrasive pit-digging mechanism, asperity-contact microcrack initiation, friction generated cyclic stresses that lead to the formation of microcracks and subsurface cyclic stresses that lead to surface de-lamination in the fretted region. It has also been shown that compressive stress between the members can have a beneficial effect on the suppression of fretting fatigue.

(FIG. 22) shows two views of an example of fretting torsional fatigue which caused the failure of a shaft. This failure occurred on the cone end of the screwshaft and not in the keyway and was induced by an inadequate interference fit between the shaft and the bore of the mating part.

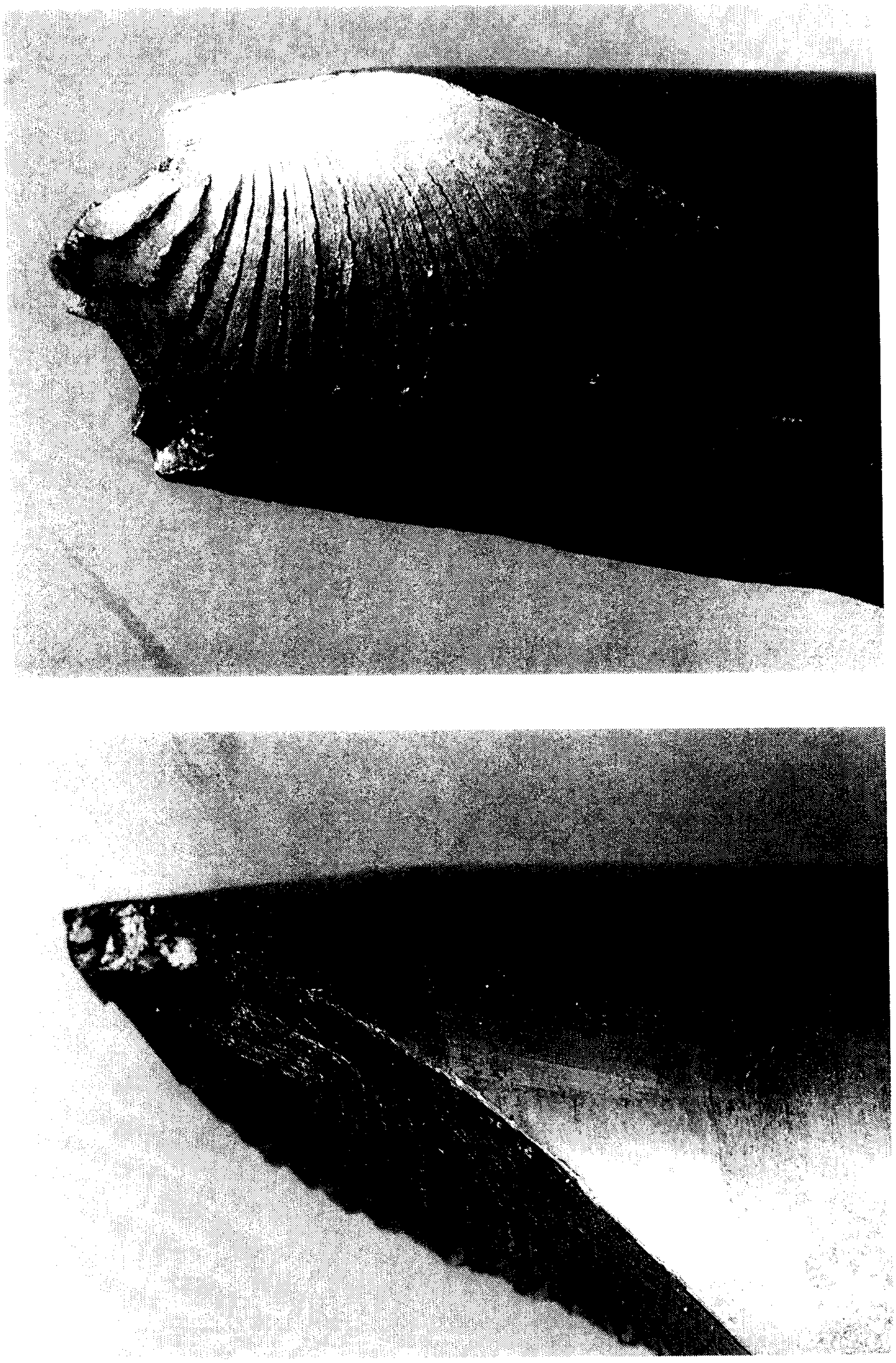


FIG. 22—FRETTING TORSIONAL FATIGUE OF A SHAFT

Brittle Failure

True brittle fracture occurs without significant gross deformation. This fracture mechanism manifests itself in a plane normal to the applied stress and is essentially a fracture mechanism which occurs through the grains of the material. When viewed under the microscope the fracture face is seen to contain a large number of facets together with a branching pattern of cracks. In plate sections a pattern of chevron markings is seen, the direction of which points to the origin of the failure, (FIG. 23). While this is not a commonly encountered type of failure today, a recent example is the failure of some 38 mm bolts from a steering gear hydraulic ram stopper pedestal, shown in (FIG. 24). These bolts were manufactured from a 0.35% C, 1.48% Mn steel. Material tests showed them to have a ratio of the 0.2% proof to ultimate strength of 98% and an elongation of only 9%. The bolts had failed at the underhead position in way of a 0.45 mm radius by brittle fracture. Also around the circumference of the failed section was a small, 5 mm deep region of fatigue propagation from multiple origins.

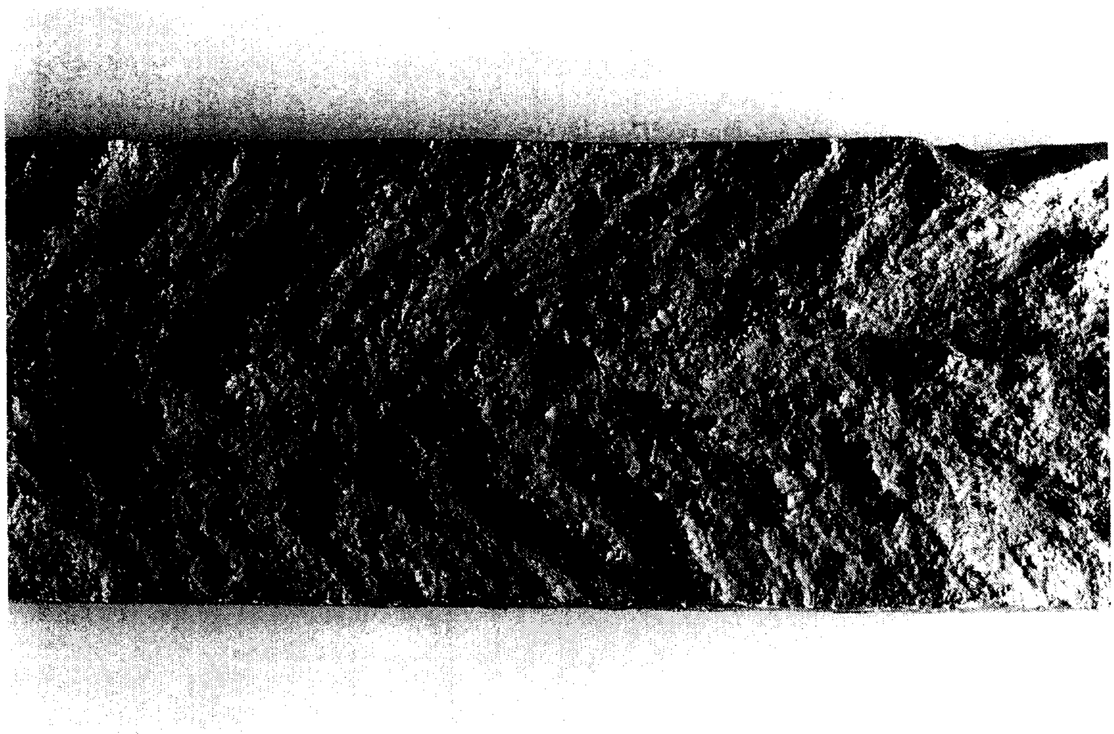


FIG. 23—CHEVRON MARKINGS ON A BRITTLE FRACTURE OF A PLATE

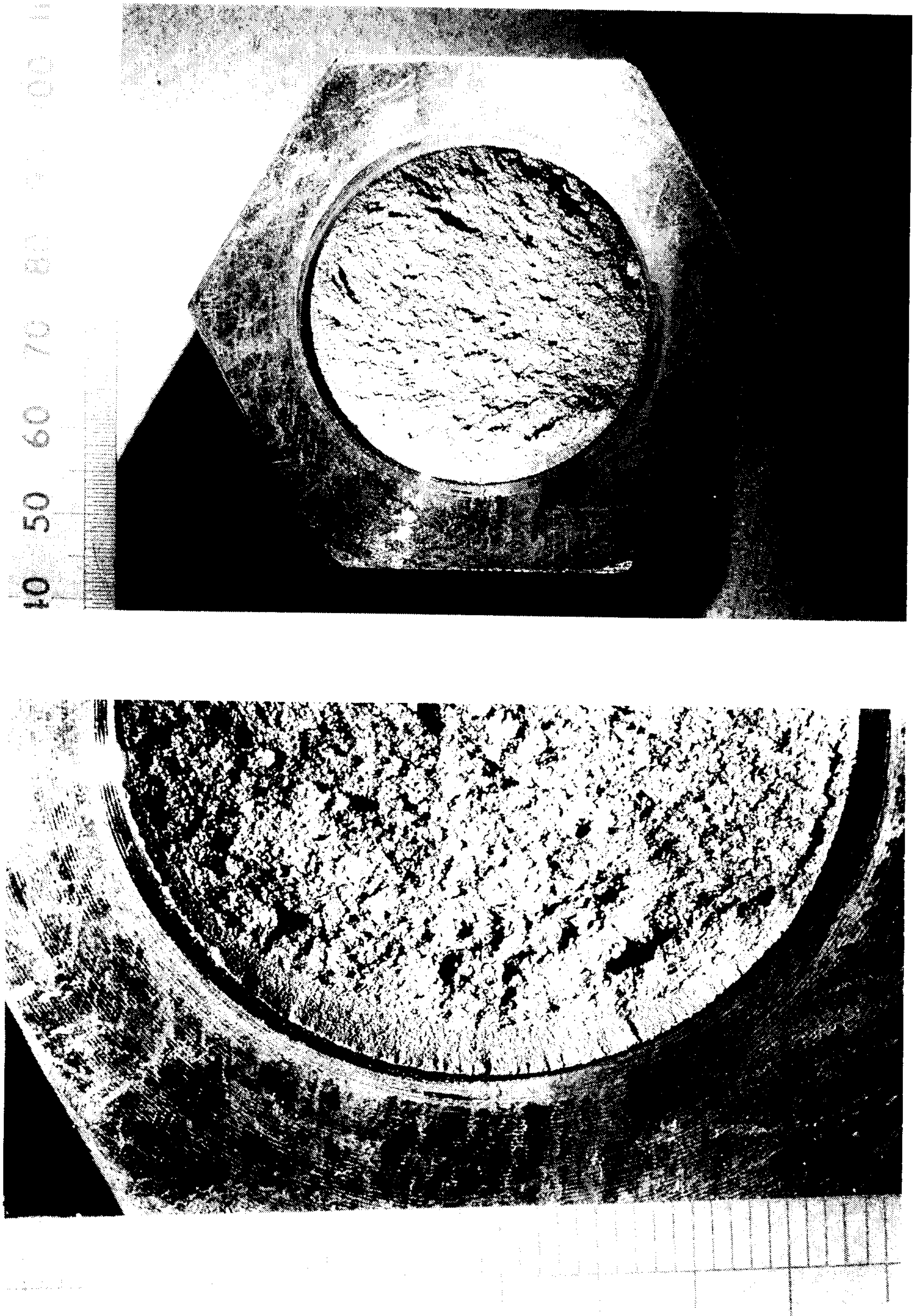
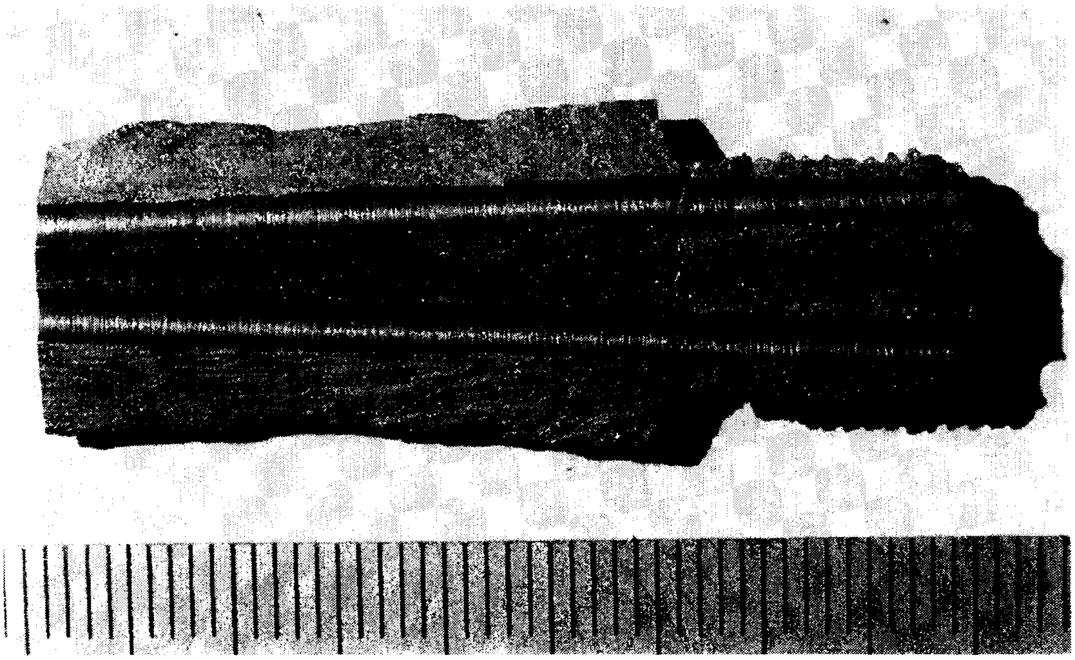


FIG. 24—BRITTLE FRACTURE OF A BOLT

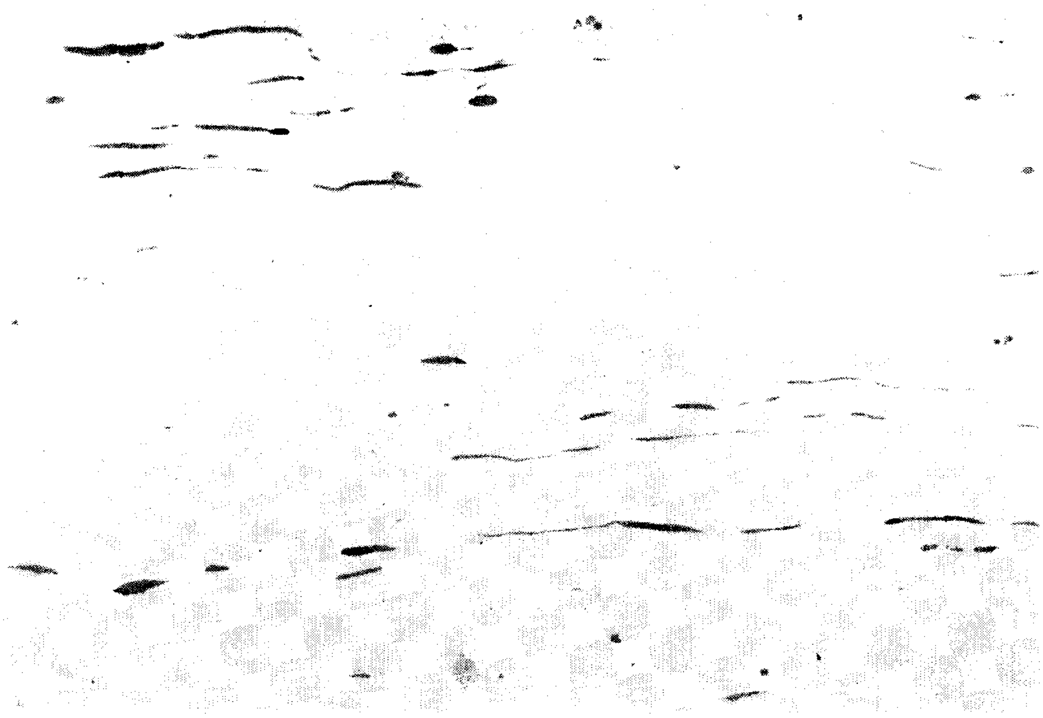
Ductile failures types

Pure ductile failures are rarely encountered today, however, a variant of this mechanism is occasionally seen. This is the lamellar tearing mechanism which is known to sometimes occur in the parent plate beneath welds in the through thickness direction. (Fig. 25) shows a classical example in a cruciform joint. In this type of tearing mechanism the presence of non-metallic inclusions has a significant effect on its development. Another example has recently been found in a fractured fuel injector. The injector was made from a medium carbon sulphur bearing steel with a significant quantity of manganese sulphide inclusions elongated in the axial direction. The holder had cracked longitudinally due to the action of lamellar tearing caused by high hoop stresses and an unfavourable microstructure of the material, (Fig. 26).

Fig. 25—Lamellar tear on a cruciform joint

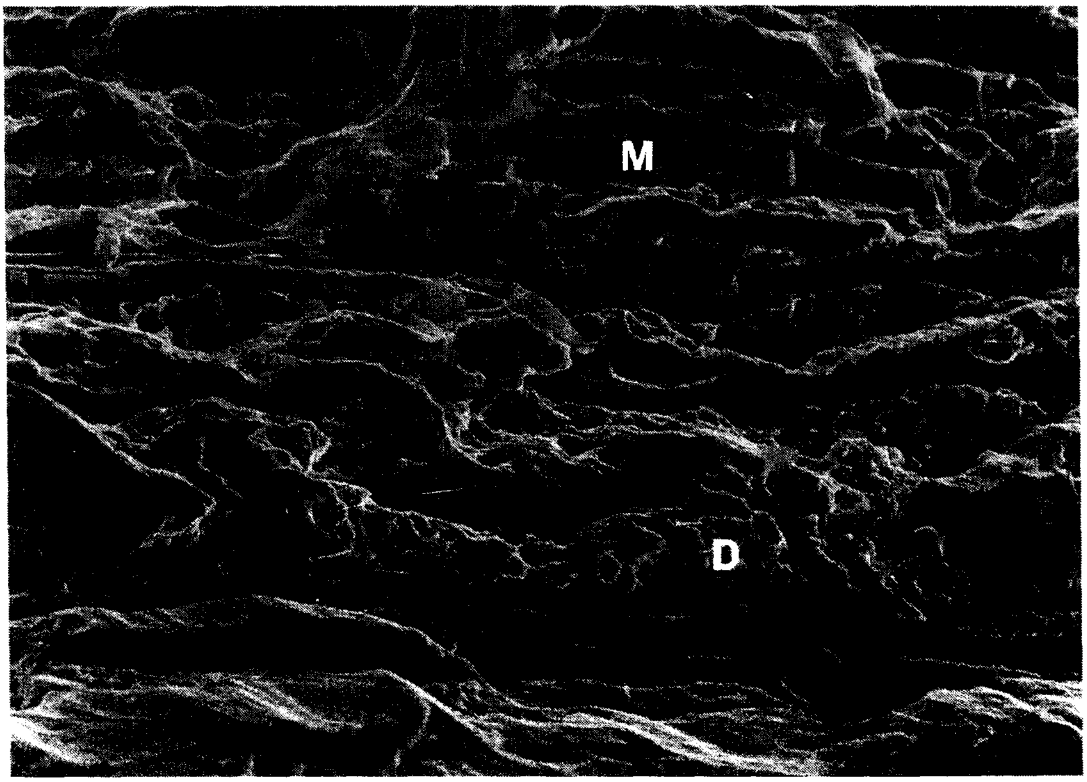


AXIAL FRACTURE FACE IN BROKEN INJECTOR



MANGANESE SULPHIDE INCLUSIONS IN FRACTURED INJECTOR

FIG. 26—LAMELLAR TEAR IN A FUEL INJECTOR BODY



FRACTURE FACE SHOWING:
(D) DUCTILE TEARING
(M) ADHERING MANGANESE SULPHIDE INCLUSIONS

FIG. 26.—LAMELLAR TEAR IN A FUEL INJECTOR BODY

Vibration excitation

Throughout TID's existence vibration investigations as seen in Table 1 have featured prominently. Experience has shown that the most effective way of dealing with these problems is to reduce the excitation forces. In the marine environment excitation normally originates from one or more of propulsor-hull interaction, main or auxiliary machinery and wave action. Various considerations apply to these excitation sources as follows.

Propulsor-hull interaction

The propulsor-hull interaction problem is intimately related to the representation of the three dimensional inflow field and to the complex nature of cavitation dynamics on the propulsor. Both of these aspects are far from completely understood at this time.

In terms of analysis effective wake is a relative concept, since it embodies the induction effects of the propeller model, and hence there are as many effective wakes as there are numerical propeller models. In the context of hull pressure the spatial, nominal wake field, normally measured at model scale, is the traditional starting point for the effective propeller inflow. The simplest approach is to scale the nominal velocities in the ratio of the disc mean nominal velocity and the effective velocity obtained from propulsion tests. In practice, there is a tendency to use methods such as the V-shaped Segment Method (VSM) and the Force Field Method (FFM) in which axi-symmetric contraction and actuator disc induction effects are applied to the nominal flow field. Such methods additionally require estimates of the propeller thrust, but do not generally appear to be unduly sensitive to errors in this parameter. Within TID adaptations of both methods have been used in the course of investigation and research activities; neither method has been found to be significantly more accurate than the other for all ship types. HOEKSTRA's wake scaling method[11] has been in use at LR for more than 15 years, principally in providing wake harmonics for use in modelling cavity dynamics. In recent years alternative scaling procedures due to TANAKA[12] and SASAJIMA[13] have been used in conjunction with the above effective wake methods. None of these combined methods has adequately predicted mean thrust, power and shaft speed, although the VSM-TANAKA and FFM-SASAJIMA combinations compared favourably with a limited amount of ship trials results.

Cavitation dynamics are complex and in their general form do not readily lend themselves to mathematical analysis. Blade cavitation has been one dominant source of many of the forced vibration problems encountered by the Department, and is characterised by vibration and surface pressure signals which normally contain significant harmonic content at frequencies which are multiples of the blade passage frequency. As the ship's power is increased to service or maximum power, the blade rate and, to some extent, the twice blade rate amplitude will normally increase continuously throughout the power range. The higher harmonic content of the spectrum may also increase in strength but in well behaved cavitation situations it should remain less than the blade rate component and reduce in strength monotonically as the harmonic number increases. Where cavitation is a problem it is not uncommon to observe that some of the higher harmonics, in the range of twice to the fourth or fifth harmonic, will exceed or be of similar magnitude to the first harmonic in strength. It is likely, however, that a significant proportion of the higher harmonic content of the measured pressure spectra originates from the tip vortex following on from the cavities over the blades and the interaction between these two forms of cavitation. Typically should significant dynamic activity in these regions occur, then troublesome higher order or broadband excitation on the hull may result. Much yet needs to be learnt about the behaviour of the tip vortex both in terms of its prediction from theoretical methods and of its inception and scaling from model tests. Theoretical methods of cavitating propeller analysis, however, can predict blade rate hull surface pressures reasonably well; at least to an equivalent error bound to the normal range of model test accuracy.

From a practical viewpoint it is difficult to give definitive guidance on the acceptable levels of hull surface pressure for ships as the dynamic structural response of the hull and superstructure influences the resulting vibratory levels considerably. However, for the blade rate component of hull surface pressure, current practice suggests a reasonable probability of achieving an acceptable vibratory behaviour for the ship due to hydrodynamic excitation may be obtained if the values given in Table 3 are not exceeded.

TABLE 3—*Typical modern values of blade rate hull surface pressures*

General Ship Type	**Typical blade rate hull surface pressure range**
Cruise Liner	1–2 kPa
Ro/Ro Ferry	2–4 kPa
Container Ships and Fast Cargo Ships	3–6 kPa
Slow Bulk Trade Ships	4–7 kPa

With regard to the higher blade rate harmonics these should always descend in a monotonic fashion with, as a guide, twice blade rate around half of the first and the third being of very small proportions. The harmonics beyond the third should generally be negligible and significant broad band excitation in the first decade of the blade rate harmonics should be avoided as this has the potential to excite a range of structural frequencies.

When considering the acceptable levels of hull surface pressure it must be recognised that some variability will inevitably exist in the actual pressure signature. This will result from spacial and temporal variations in the wake field including the effects of variable gas content in the water and the motion of the ship and sea. These, among the other factors involved, induce significant variations in cavity structure which may result in a wide variation of induced hull surface pressure. (FIG. 27) shows the level of variation that can occur in the pressure maxima measured on a ship which had considerable dynamic activity in the tip vortex structure adjacent and behind the blade trailing edge.

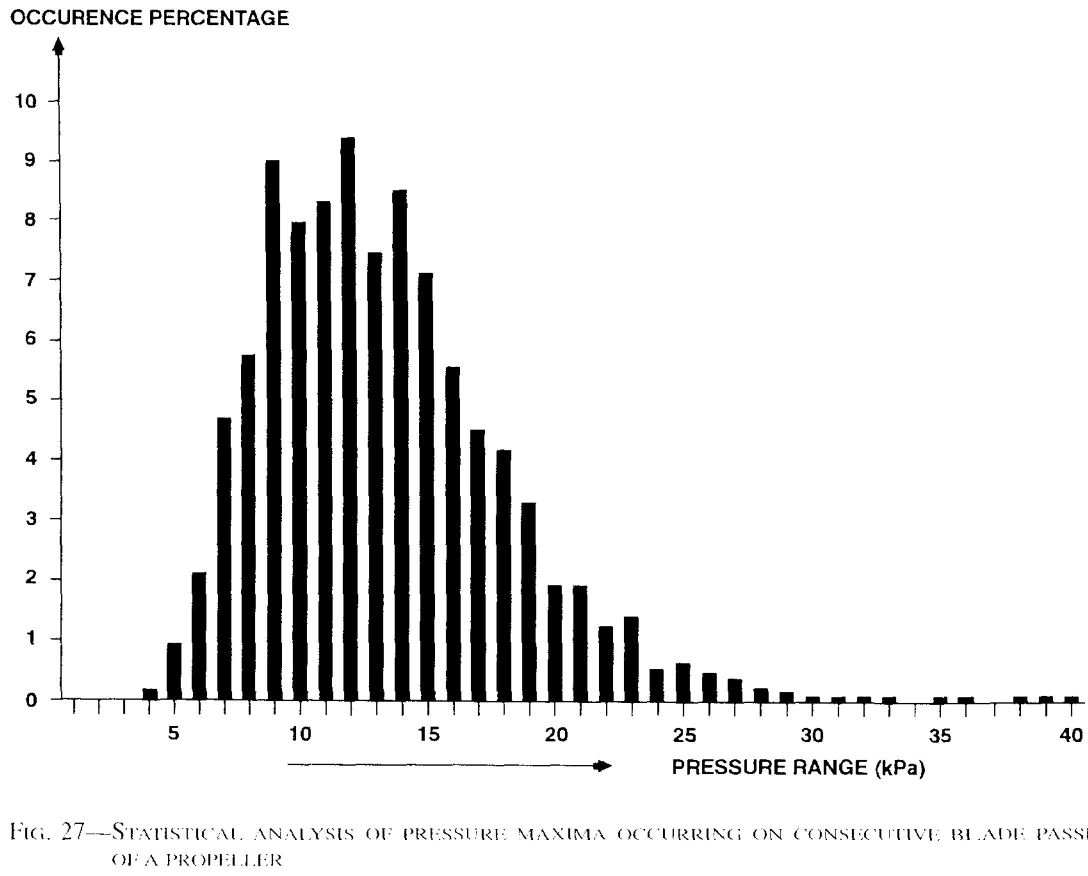


FIG. 27—STATISTICAL ANALYSIS OF PRESSURE MAXIMA OCCURRING ON CONSECUTIVE BLADE PASSES OF A PROPELLER

Propeller-hull vortices are generated as standing vortices in the flow field between the propeller and the hull as a result of a three-dimensional stagnation region in the hull flow field. In some cases these vortices may also cavitate. Conditions leading to the promotion of this type of vortex are a full hull form ahead of the propeller - particularly in the upper region of the propeller disc, a low advance coefficient and a flat afterbody above the propeller. The excitation and response signals resulting from the dynamic activity of this vortex show an intermittent character in which very high localised excitation may be observed.

Main or auxiliary machinery excitation

The vibration excitation forces generated by the main and auxiliary machinery on ships can be generally categorized in relation to two distinct types of machines.

- Those with only rotating parts
- Those with a combination of rotating and reciprocating parts.

For rotating machines the principal excitation results from mass unbalance which generates forces and moments.

These are rotating vectors and the corresponding vibration response of the machine bearings and structure which react the unbalance is a vibratory motion with a frequency equal to the machine speed; 1st order. Misalignment can also excite vibrations at this fundamental frequency together with higher harmonics and in particular a 2nd order component. Second order excitation is also associated with the universal Hooke's joints used in cardan drive shaft arrangements. When such designs incorporate an intermediate shaft or shafts, for example the propulsion shafts commonly found in double-ended ferries and Z-drives, the natural frequencies of the lateral vibration modes must be carefully considered to avoid coincidence with the excitation from the cardan couplings.

Certain purely rotating parts can induce excitation forces at frequencies with higher multiples of the shaft speed; gears, fans, pump impellers and rolling element bearings where the resultant vibrations are related to the passing frequencies of the gear teeth, vanes and so on. If, however, there are manufacturing errors such as tooth pitch deviations or physical damage to a tooth or roller bearing, the excitation is more complex due to the impulsive characteristics of the forces. In such circumstances the vibration behaviour is usually influenced more by local structural resonances than the impact frequency of the excitation forces.

At the other extremity of the frequency spectrum, hydrodynamically lubricated bearings can generate journal reaction forces which cause rotor vibrations at a sub-harmonic frequency. This phenomenon occurs in high speed bearings and is known as half speed whirl or oil-film whip.

Diesel engines and other similar machines have inertia forces induced by the motion of their rotating, reciprocating, and oscillating masses: crankshaft, balance weights, pistons, piston rods, crossheads and connecting rods. The development of long stroke engines has been accompanied by a reduction of the ratio between the connecting rod length and crank radius in order to minimise the engine height. Greater angularity of the connecting rod affects the piston acceleration and increases the higher order inertia forces. Cylinder gas pressure variations are a further source of excitation forces.

The inertia and gas forces for each cylinder line are reacted by the crankshaft bearings and cylinder liner wall or crosshead bearing guides in the case of slow-speed two-stroke engines. The resultant external forces and moments, torque fluctuations and guide force moments are then a function of the engine firing order; that is, the fixed phase relationship between the respective crankthrows. These external forces are transmitted to the engine seating by the holding down bolts when the engine is rigidly supported on solid chocks.

The inertia forces from individual cylinder lines can also combine to impose internal moments on the engine frame. Unlike the better known external unbalanced force and moment effects, internal moments are usually ignored as a source of vibration excitation because in theory there should be no transmission through the holding down bolts, if it is assumed that they are reacted entirely by the engine frame. A small number of cases have been encountered when engine internal moments have excited hull girder vibrations, which must be attributed to an insufficiently stiff engine frame. Similar underlying causes have been identified on engines supported by resilient mountings. Here, the absence of any reinforcement from the seating stiffness has increased the cyclic deflections of the engine frame which in turn has excited excessive vibrations of the attached ancillaries.

Wave excitation

The sea is a further source of excitation of ship structures by virtue of the ship passing at some angle of incidence to a wave sequence. In poor weather conditions and at speed, impact or slamming loads can be induced in many ship types.

For normal trial conditions a relationship between the wave characteristics, ship speed and heading can be used to derive wave encounter frequencies which in turn provide an excitation frequency. In general such frequencies are below about 2 Hz.

TYPICAL CASE STUDIES

To illustrate some of the specific lessons learnt from investigation activities a series of recent examples have been drawn from the Department's records. Clearly, within the confines of a paper such as this only a small portion of the available experience can be presented, however, throughout the paper aspects of the more general experience have been summarised.

Ship based investigations

Since its inception TID has been predominantly involved with technical problems and failures on ships. In particular, main propulsion machinery investigations have remained as an almost continuous activity. As demonstrated by the following case studies, many assignments deal with recurrent subjects and failure mechanisms which in themselves are well understood.

Propulsion Shaft Alignment

Detailed predictions of hull deflections and their effect on the shaft alignment are rarely included in design calculations. Additionally, the designers of direct drive diesel engines specify permissible limits for the static forces and moments that can be imposed on the engine crankshaft by the propulsion shafting in order to achieve positive downward static loads on the engine bearings. The span between the engine main bearings is short and these alignment tolerances represent stringent requirements in the same way as the more familiar differential load limits specified for the output gearwheel bearings of main propulsion gearboxes.

Many of the above aspects were evident during the investigation of the propulsion shaft alignment on a large bulk carrier. The design bearing loads and engine alignment are given in Table 4 and the installation procedure was intended to achieve the values in the left hand column. It was assumed that the alignment would be changed only by the relative thermal rise of the engine.

Inherently, shaft systems with a single plummer bearing require an especially careful consideration of the initial alignment to accommodate the changes that can be expected on vessels with a large displacement variation between the ballast and loaded conditions. In this instance there was insufficient load on the sterntube forward bearing and the engine alignment was too close to the hogging limit. Correction of a hogging moment at the engine flange could not be achieved by raising the plummer bearing without unloading the stern-

TABLE 4—*Design bearing loads and engine alignment conditions for a bulk carrier*

Bearing	**Bearing reaction (kN)**	
	Cold engine half immersed propeller	**Hot engine fully immersed propeller**
Sterntube Aft	639	606
Sterntube Forward	48	67
Intermediate Shaft	201	180
Number 8 Engine	170	234
Engine Flange		
Shear Force kN	167.6	182.1
Bending Moment kNm	−77.0	−25.5*

*—ve sign means sagging

tube forward bearing. The scope for adjustment was further exacerbated by installation errors which resulted in a 250kN static load on the plummer bearing.

As the vessel's draught increased the respective acceptance criteria were progressively compromised in that the:

- Sterntube forward bearing became unloaded
- Plummer bearing became overloaded
- Hogging moment imposed on the engine crankshaft increased beyond the design limit
- Number 7 main bearing was unloaded.

A fundamental re-alignment of the engine was required to restore realistic practical alignment margins. It is, however, a salutary lesson to report that these margins were subsequently eroded in part by the permanent alignment changes that occurred due to the relaxation of hull structure residual stresses during the vessel's maiden voyage and an initial encounter with adverse weather.

The static shear force and bending moment at the engine output flange can be verified only by strain gauge measurements and it is suspected that many installations are operating satisfactorily with shaft alignments that may be out with the permissible values because the limiting parameters are not necessarily reflected by unsatisfactory crankweb deflections.

The execution of the engine re-alignment would have been an especially interesting experience for those former TID surveyors who struggled to optimise engine bedplate and crankweb deflections by the selective adjustment of chock thicknesses. Their work was based on micrometer measurements taken from a taut piano wire judiciously threaded through the crankcase above the crankshaft in a way which avoided the connecting rods. They may have been surprised to find that the structure of the six cylinder engine installed in the bulk carrier was sufficiently stiff that it could be supported only at its four corners without jeopardising the crankweb deflections.

Rudder Investigations

A wide range of rudder or steerable duct actuating force, cavitation and integrity problems occur from time to time. In the case of rudders, these may take the form of bearing failures or erosion of the rudder plating; the latter being due either to cavitation developed by the rudder itself or from the cavitation entrained in the helical slipstream from the propeller. Steerable ducts also can have accentuated vertical seaway loadings and, in common with their fixed counterparts, erosion on the inner plating. Actuating force problems can occur with either form of steering device.

Typical of such an investigation was the full scale measurement of rudder stock moments on the semi-balanced spade rudder of a container vessel by the use of strain gauges. This assignment was undertaken in order to establish the reasons and to provide a basis for remedial action for both an under powering of the steering gear and a poor response to the helm from the ship. Rudder torque and stock bending strain measurements were made and the results of this trial's programme were compared with computed loadings. These were derived from the surface pressure distribution over the rudder horn and blade by means of one of LR's lifting body vortex panel codes. For this computation the incident flow to the rudder was evaluated from a lifting line propeller analysis procedure. The rudder model was verified initially using the model test data and, subsequently, calculations were performed for the container ship rudder in the slipstream of the propeller for comparison with the measured full scale data. It was found that good agreement between the measured and predicted loading was obtained and that the form of the rudder stock torque relationship for a 35–35 degree rudder angle zigzag manoeuvre was well represented.

The same modelling procedure was then used to predict the effect of increasing the rudder chord length by 20%. The predicted maximum torque increased by more than 150%, while the side force increased by approximately 10%. A similar percentage increase in rudder torque was subsequently re-measured on board the ship when the rudder extension was fitted.

Gearing Investigations

Despite improvements made in the design and manufacture of propulsion gears and an almost universal adoption of case hardened teeth, infrequent but persistent failures continue to occur. Throughout TID's existence the common

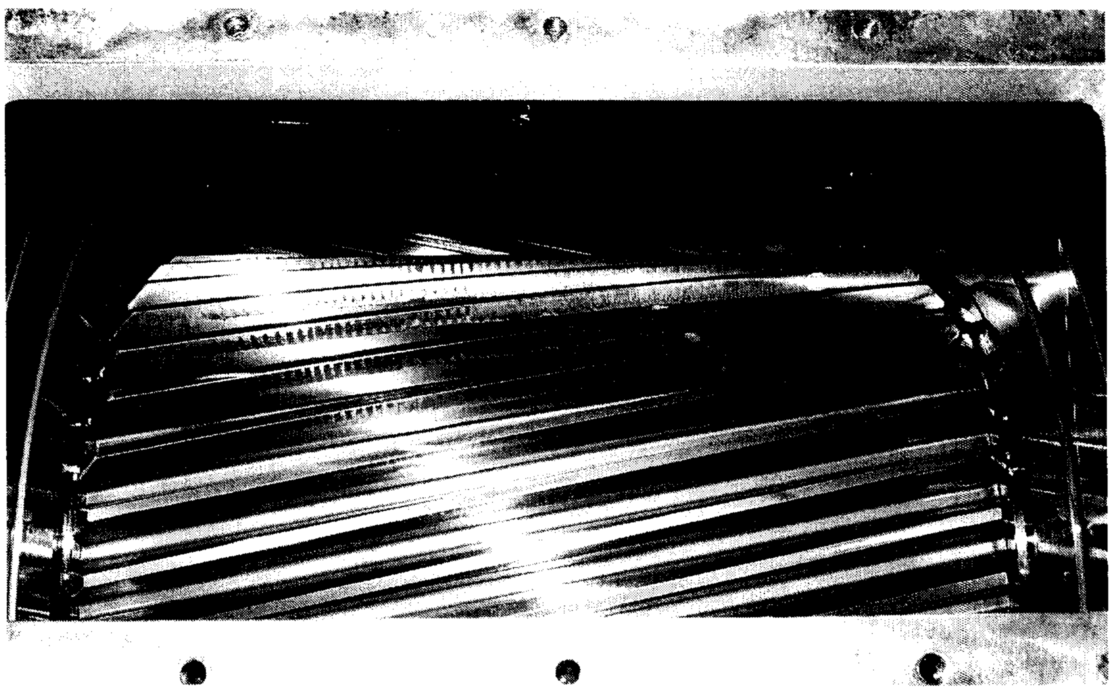


Fig. 28—Scuffing of case carburised pinion

failure modes, tooth fractures and surface damage caused by either scuffing or fatigue, have been reported. Such repetition must be attributable to the mechanical complexity associated with tooth meshing conditions and the interrelated influence of elasto-hydrodynamic lubrication.

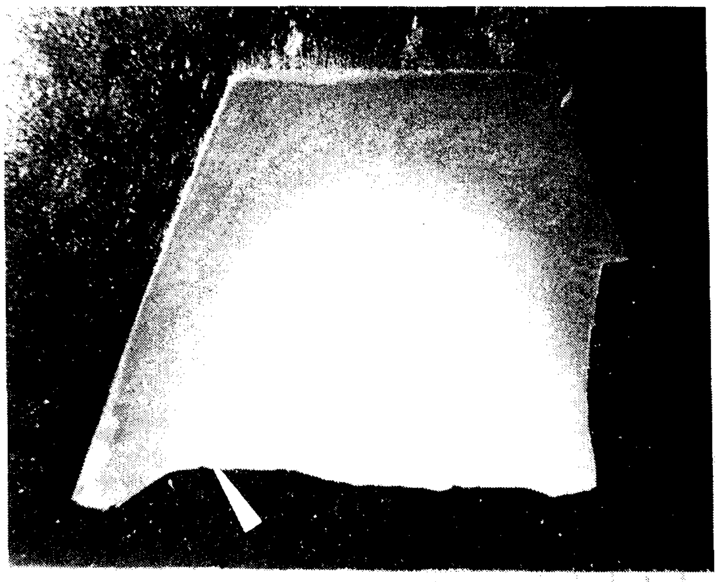
The following examples illustrate the importance of the findings derived from metallurgical examinations and the intractable nature of some problems when compared to an obvious immediate cause of a similar tooth fracture.

(Fig. 28) shows scuffing damage on the case-carburised pinion of a single reduction main propulsion gear train. The characteristic tiger-stripe pattern is thought to reflect small undulations generated by the grinding process, and the extent of the damage suggested a tooth load distribution which was biased more heavily towards the forward end of the helix. A reason for the observed tooth misalignment was confirmed by measurements of the gear wheel attitude in its bearings taken under running conditions which indicated a tilt induced by deflection of the integral main thrust bearing on its seating.

Nevertheless, this behaviour was similar for three other identical gearboxes which ran without a problem in the same service application. The most plausible explanation of the scuffing was thought to be a temporary deterioration of the lubrication, such as an ingress of water, but as in many cases where operational shortcomings are suspected as being a principal contributory factor, the failure hypothesis can be difficult to substantiate.

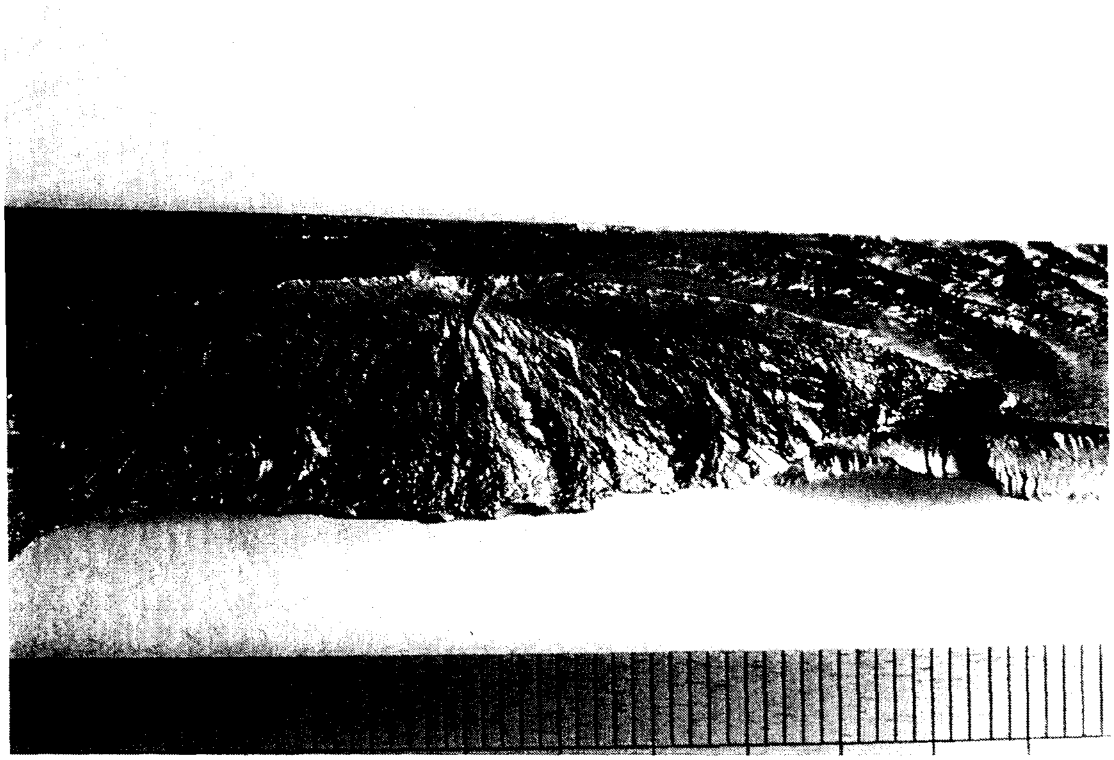
It may be of interest that subsequent careful running-in of the scuffed tooth surfaces re-established satisfactory operation of these gears.

Fatigue cracks which lead to tooth breakages invariably originate on the tooth root surface or on the flank in areas of pitting and exfoliation damage. The isolated single tooth failures shown in (Figs 29a&b) are less common where



VIEW OF BROKEN TOOTH PROFILE
NOTE: ARROW IDENTIFIES THE ORIGIN OF THE FATIGUE CRACK

Fig. 29a—Gear tooth fatigue cracks with sub-surface origins fatigue cracks have grown from sub-surface origins. In both cases cracks were initiated close to the transition between the core material and the carburised case; a position where the expected residual stresses would be tensile. Other similarities included an absence of metallurgical defects and a heat treatment procedure which resulted in a satisfactory case depth hardness profile and distribution of carbides.

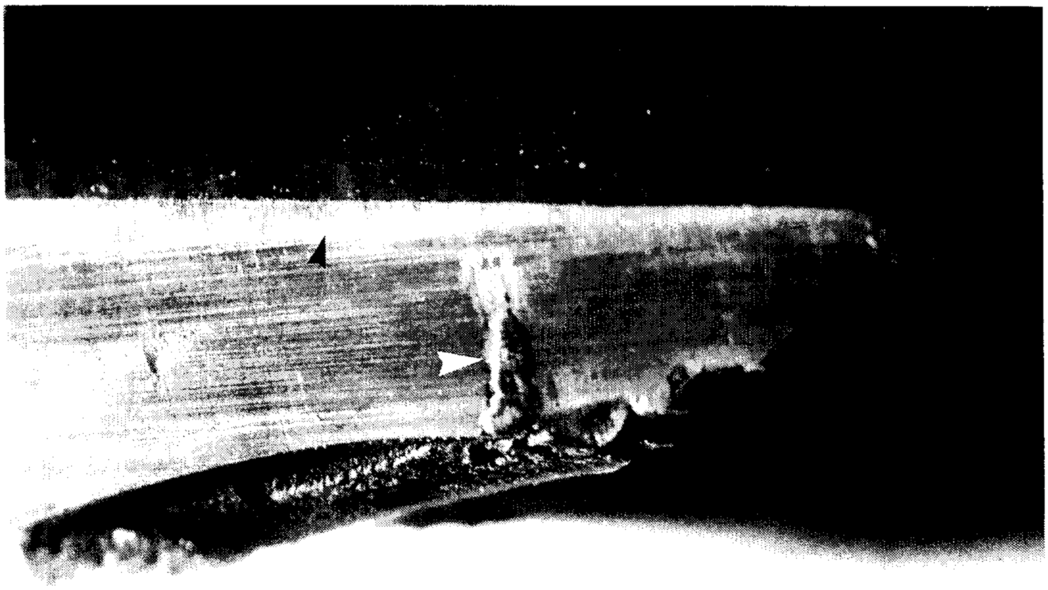


TOOTH FRACTURE SURFACE

FIG. 29B—GEAR TOOTH FATIGUE CRACKS WITH SUB-SURFACE ORIGINS

The failure shown in (FIG. 29C) could be easily explained by the remaining evidence of debris that had been entrained in the mesh. This was found as a small accretion of steel cold welded to the tooth flank in way of the sub-surface crack origin. The chemical composition of the alloy confirmed that it was a foreign body. Conversely, the similar single tooth fracture of a different propulsion pinion remains unexplained; all the more so because the crack origins were some one third of the face width distant from the end of the tooth, (FIG. 29B).

Problems have also arisen on less precisely manufactured gears. The rack and pinion jacking mechanism on a self-elevating offshore drilling platform suffered from excessive plastic deformation of the tooth surfaces. Strain gauge measurements taken in the tooth roots and an analysis of the contact stresses led to new pinions being fitted with a greater number of teeth.



DEBRIS ENTRAINED IN THE MESH
NOTE: ARROW SHOWS ACCRETION COLD WELDED TO THE TOOTH SURFACE

FIG. 29C—GEAR TOOTH FATIGUE CRACKS WITH SUB-SURFACE ORIGINS

Ship superstructure investigation

A twin screw, diesel-electric cruise liner reported noticeable vibration on the bridge wings from the time the ship entered service. The propulsion system comprised four medium speed diesel generator sets supplying power to two three phase synchronous and frequency controlled electric motors driving controllable pitch propellers.

The vibration investigation was carried out while the ship was in normal cruising service. It was found that while the global vibration characteristics of the ship under both port and sea-going conditions were satisfactory, the bridge wing's vertical natural frequency was excited at main engine first order excitation frequency. The bridge wing vibration varied in a complex way with the port wing suffering to a greater extent. High to severe vibrations were experienced when the ship was:

- Manoeuvring,
- At full speed with all four engines running,
- Operating with No. 4 engine running.

It was found that the main engine's resilient mounting system was reasonably effective, however, the residual vibration energy transmitted to the ship at engine first order frequency was sufficient to dominate the ship's vibration and to excite the bridge wing resonance. While the vibration severity was considered unlikely to cause structural damage in the short term, the effects on the operation of the local control console and the compass repeater were of concern. It was, therefore, recommended that both bridge wing structures were stiffened in order to raise their natural frequency above the excitation range. As with all problems involving resonant conditions care in implementing the remedial action was taken so as to avoid an undue response from other excitation frequencies.

Diesel engine bearing failures:

Typical of the periodic problems that arise is the fatigue cracking of soft whitemetal or babbitt bearing materials. These lead or tin based alloys have a poor inherent fatigue strength which is further impaired by modest temperature rises. Their ability to tolerate lubricating oil contamination and imperfection between the journal and bearing surfaces is however an indispensable attribute. In medium speed engines such materials are used as a thin surface overlay to protect the stronger copper-lead or aluminium tin alloy bearing material from incipient pick-up and potential seizure.

Removal of the overlay is detrimental and the manufacturers provide well documented guidance regarding the extent of permissible wear.

The Rillenlager type bearing was designed to counteract overlay wear problems for engines burning residual fuels, (FIG. 30). Fatigue cracking and consequent premature loss of the overlay from the grooves of such bearings have been found in a connecting rod application, (FIG. 31), which required a stronger but harder overlay material.

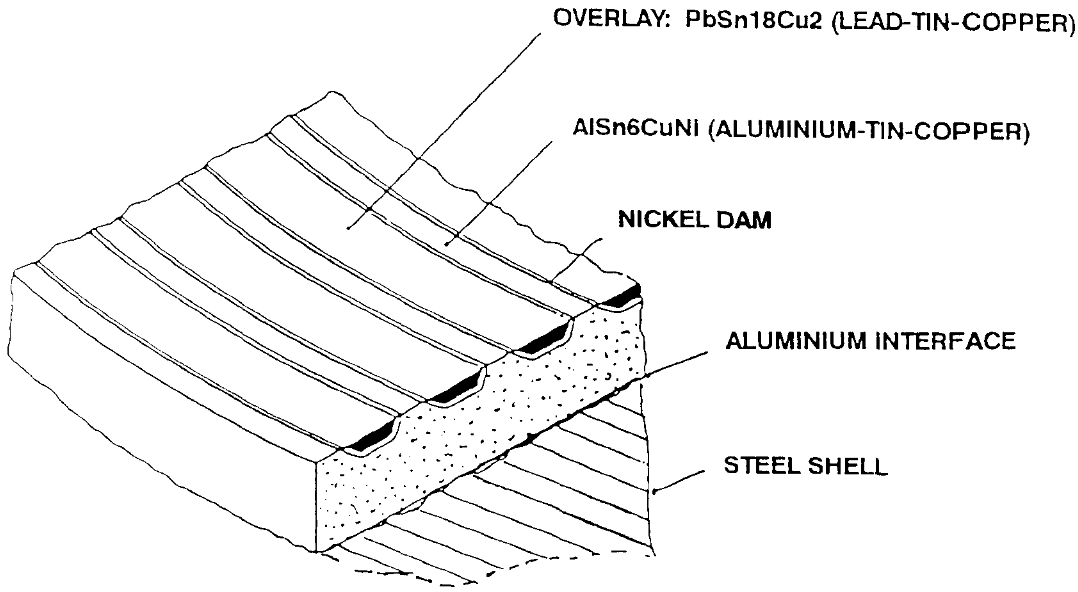


FIG. 30—CONSTRUCTION OF RILLENLAGER BEARING

SURFACE DAMAGE CAUSED BY REMOVAL OF OVERLAY

SECTION THROUGH RILLENLAGER GROOVE SHOWING LOSS OF OVERLAY

FIG. 31—FATIGUE CRACKING OF BEARING OVERLAY

Similar cracking also occurs in the thicker tin based alloy layers used in slow speed engines where the same material is used to provide load bearing and embeddability functions. Experience of fatigue damage to these bearings suggests that the cracking is influenced by the oil film thickness. The cracks originate on the surface, although their growth and the detachment of fragments can lead to an initial impression that the metallurgical bonding between the bearing material and the steel backing shell has failed. Engineers should also be aware of the importance of the material microstructure in terms of the distribution and grain size of the respective alloy phases. In addition to fatigue cracks, (FIG. 32) illustrates a mottled surface appearance caused by thermal ratcheting of a poorly cast tin based alloy.

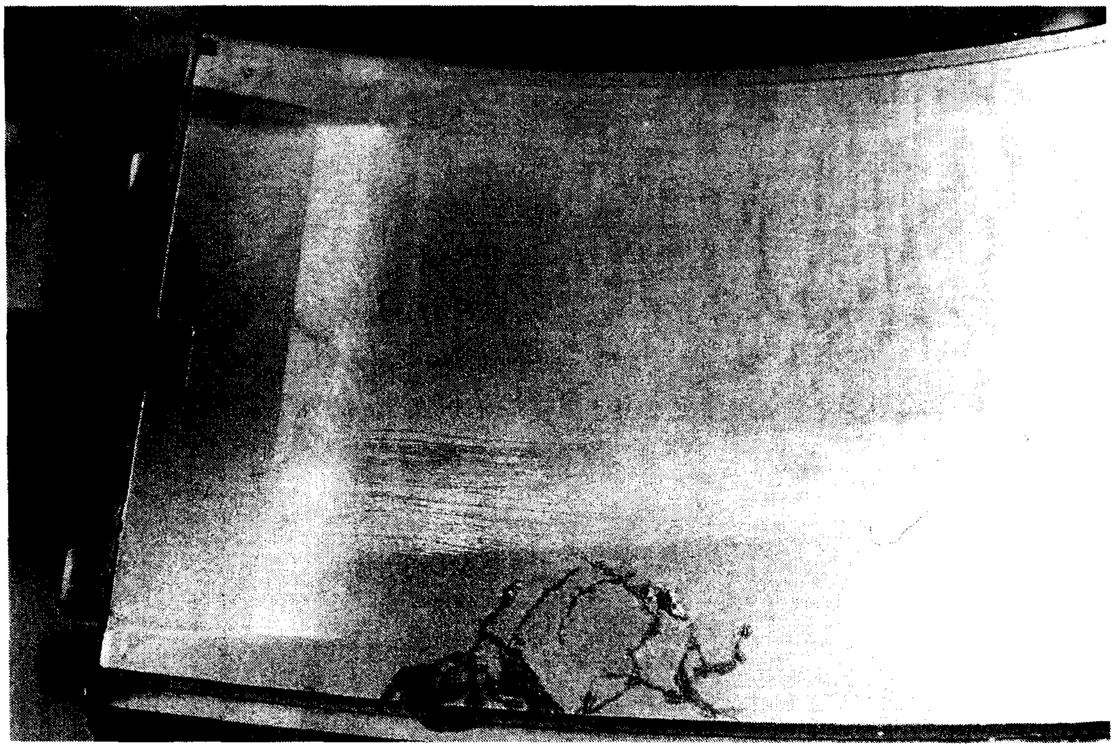


FIG. 32—FATIGUE CRACKING AND THERMAL RATCHETING OF TIN BASED BEARING ALLOY

Shipboard Vibration Investigation

A shipowner reported that, on board one of their new ships, there were excessive levels of vibration in the accommodation and localised cracking of various tanks and support brackets due to vibration. The vessel's power was supplied by a five cylinder, two stroke diesel engine, developing 2942 kW at 210 rpm with the power being directly transmitted to a five bladed fixed pitch propeller. An investigation, prior to the ship's guarantee dry docking, in which the linear vibration characteristics of the hull, superstructure and engine were quantified, in addition to torsional characteristics of the main engine shafting system, showed that fifth order vibrations measured in the accommodation spaces at some operating conditions were at a level where complaints regarding habitability were considered likely to occur. Furthermore, vibratory torques which were measured in the intermediate shaft exceeded the mean torque at higher shaft speeds, (FIG. 33). Associated with these vibratory torques was a corresponding increase in axial vibration of the intermediate shaft.

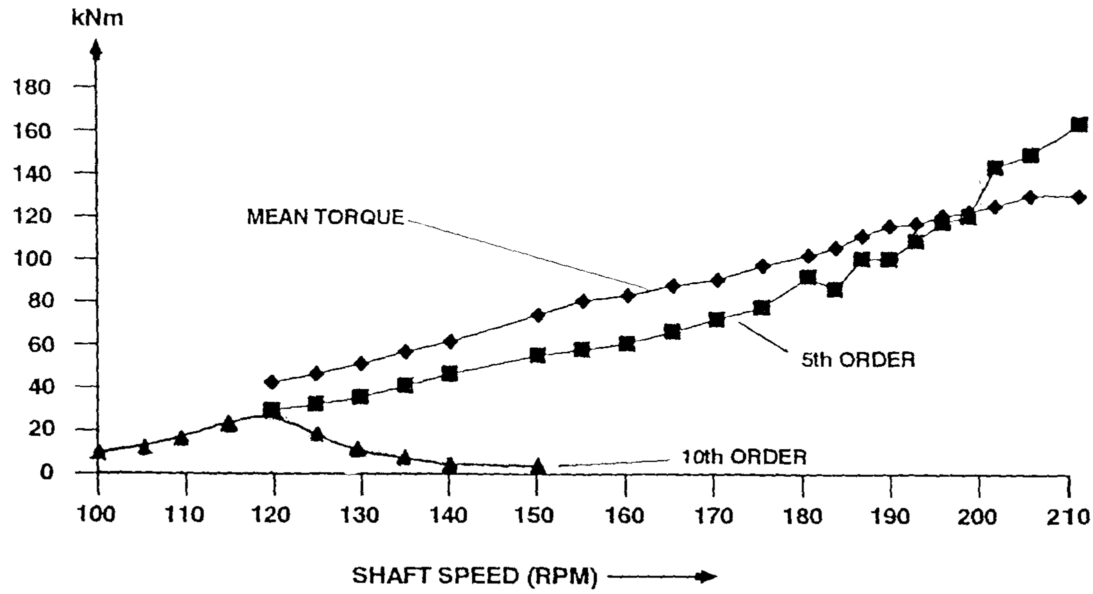


FIG. 33—INTERMEDIATE SHAFT MEAN AND VIBRATORY TORQUE

A one node, tenth order torsional vibration critical speed was measured at 130 rpm, indicating that the corresponding one node fifth order critical frequency would occur at 260 rpm. This speed was only 24% above the main engine speed of 210 rpm and compared with a critical speed of 284 rpm calculated by the engine builders. Consequently, vibratory stresses and torques were larger than predicted as the operating point was further up the flank of the resonant response curve.

It was recommended that global vibration of the ship could be reduced by either a re-tuning of the shafting torsional vibration characteristics which would entail the installation of a torsional damper, a change of shafting and possibly a barred speed range or, alternatively, the installation of an electrically driven, independently mounted, fifth order thrust compensator. The vessel's owners elected to adopt the latter recommendation and the modification was implemented. Subsequent vibration measurements indicated that vibration levels throughout the vessel had fallen to a level below those where complaints regarding habitability were likely as seen by (FIG. 34).

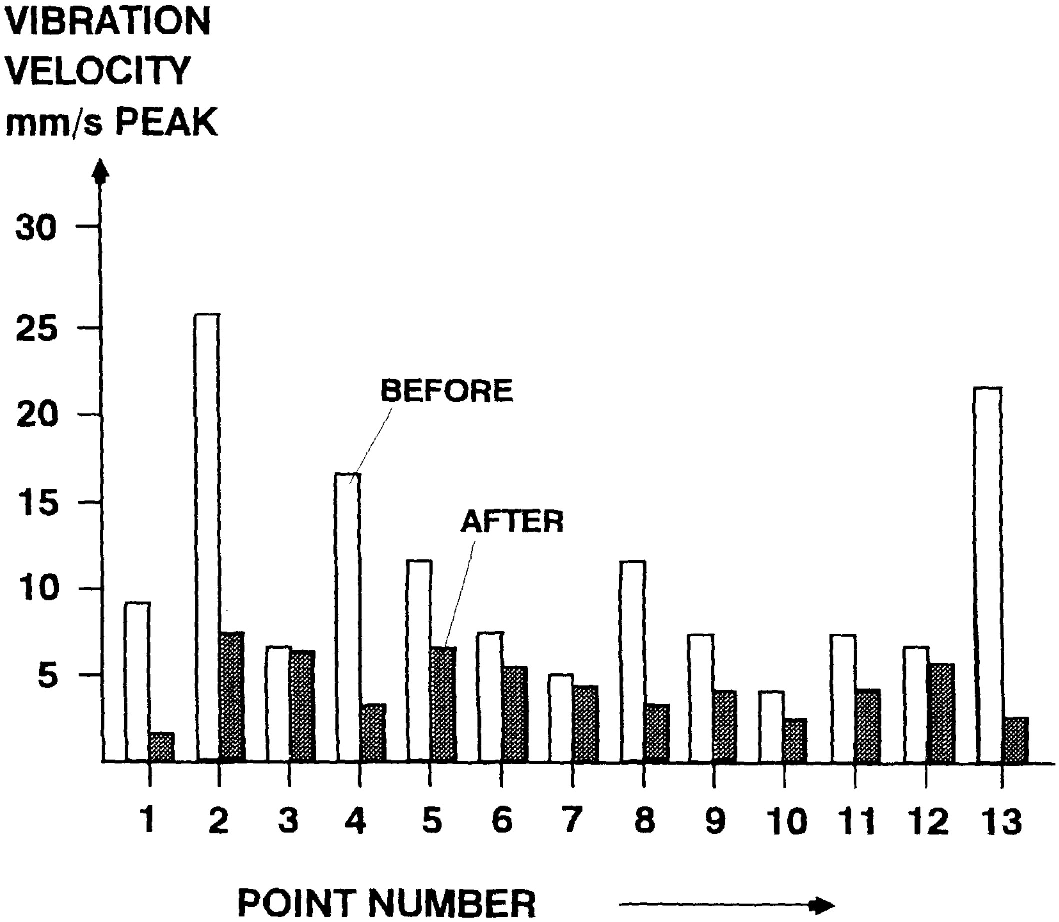


FIG. 34—COMPARISON OF 5TH ORDER VIBRATION MEASUREMENTS AT SELECTED POINTS BEFORE AND AFTER FITTING OF COMPENSATOR

Shaft Bracket Flows

Hull surface pressure and vibration measurements carried out during sea trials on a small twin-screw coastal ferry indicated that the shape and positioning of the A-brackets together with the design of the propeller were likely to be the main causes of unwelcome cavitation activity on the propeller blades and consequent levels of structural vibration; in this case reaching levels of 125 mm/s on the after deck. A redesigned highly skewed propeller replaced the original propeller and the vibration levels reduced considerably, but not to an acceptable level. The cavitation extent was observed, through windows cut in the hull, to significantly increase locally as the blades passed behind the outboard arm of the A-brackets.

Because of the scaling complexities of using model tests to understand the flow field around the A-brackets, the flow field was analysed using CFD codes in combination with the Department's propeller analysis codes. (FIG. 35) shows a view of the computational grid used in the simulation model which has 49, 59 and 54 cells in the x, y and z directions respectively.

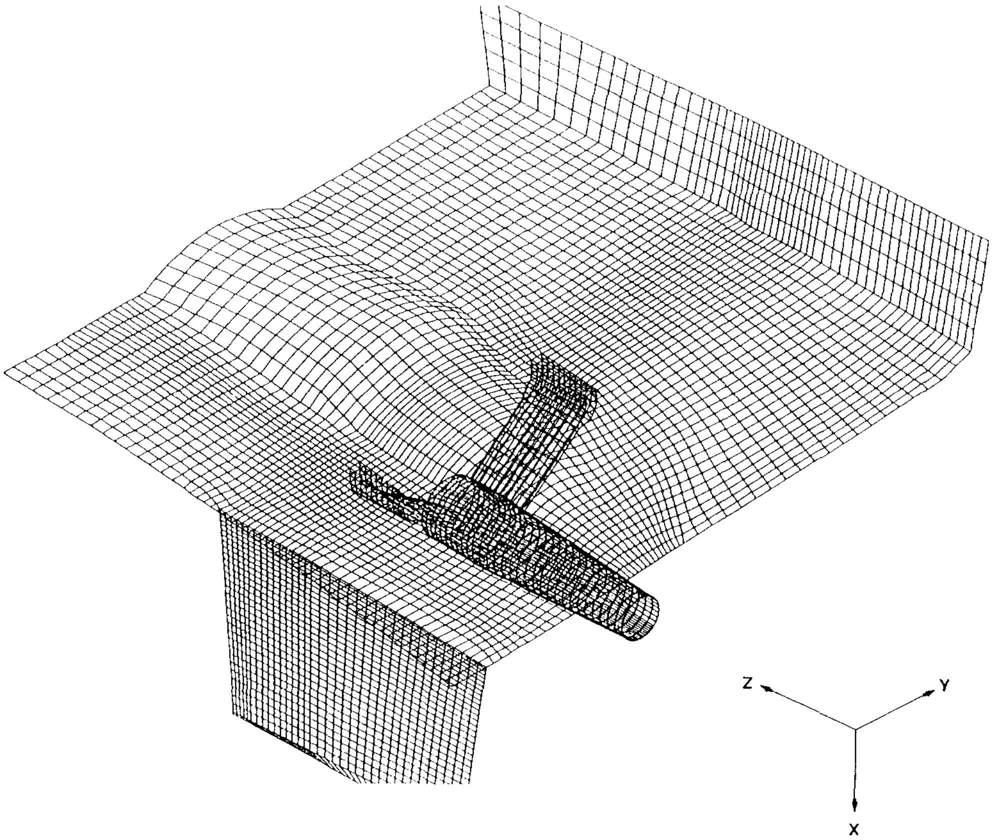
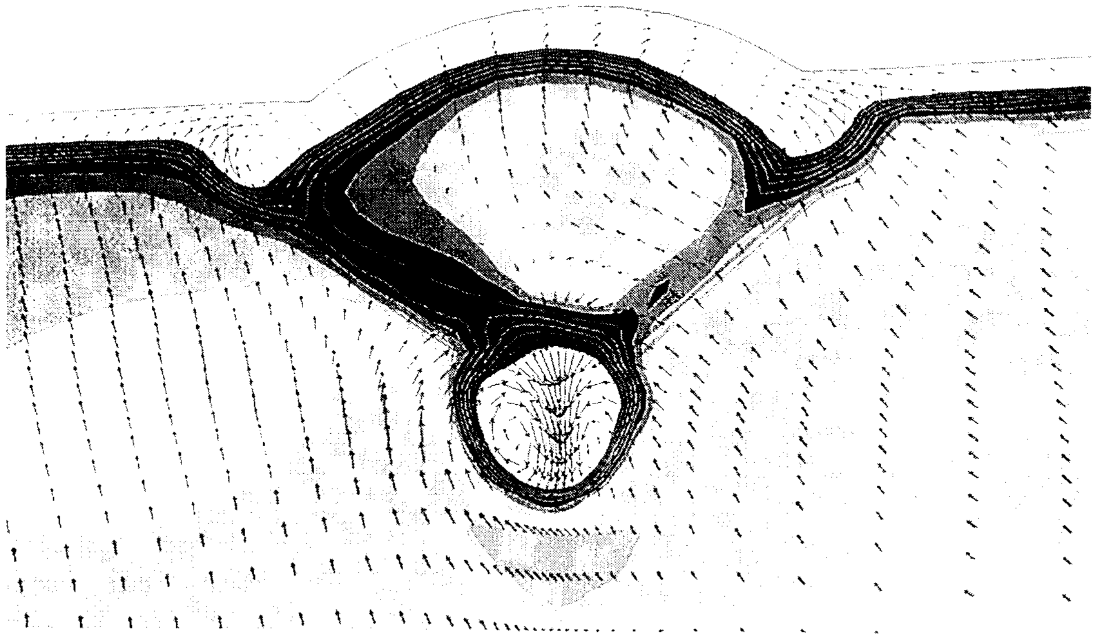


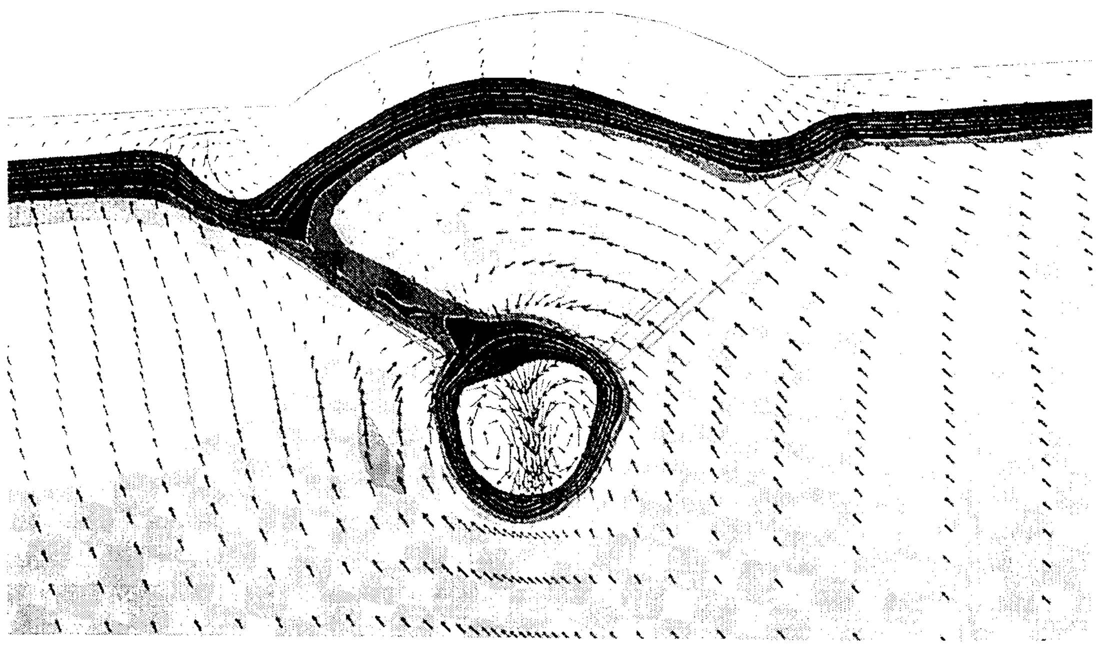
FIG. 35—SHAFT A-BRACKET CFD MODEL.

The simulations, using a ke turbulence model and by specifying zero velocity on the hull surface and employing the wall-function approach, showed that the A-brackets were contributing significantly to the wake field disturbance and could be better aligned to the local flow directions. However, to achieve such an alignment would have required the vessel's twin screw shafting system to be mechanically re-aligned which the shipbuilder was reluctant to undertake if other methods could be developed to solve the flow problem. A compromise solution was achieved by re-adjusting the leading and trailing edge shape to form an S-shaped cross-section of the A-bracket. The system was then re-analysed with the modified bracket geometry, in order to confirm improvements. Contours of velocity ratio at several planes downstream from the brackets allowed the quality of the axial wake entering the propeller plane to be assessed. (FIG. 36A) shows velocity contours and transverse components at a location half a bracket length behind the trailing edge of the original brackets. Results for the re-aligned bracket geometry, (FIG. 36B), showed a marked reduction in the axial wake peak and hence indicated that vibration and noise levels might be reduced by adopting such a design. The diffusive effect of the partial propeller tunnel in conjunction with the blockage effect of the bracket cross-section were also considered to have caused the formation of a separation pocket on and behind the upper bracket arms. The CFD calculations further showed that this latter tendency could be reduced by rounding both the leading and trailing edges of the upper bracket arms; this design aspect not having featured in the builder's original A-bracket design.

Subsequent vibration measurements following the modification to the A-brackets confirmed that an acceptable vibration characteristic had been achieved.



COMPUTED VELOCITY CONTOURS AFT OF ORIGINAL BRACKET



COMPUTED VELOCITY CONTOURS AFT OF MODIFIED BRACKET

FIG. 36—SHAFT BRACKET FLOW INVESTIGATION

Propeller Root Cavitation Erosion

Propeller root cavitation problems can be among the most complex to solve. In this region, along with tip vortex dynamics, rigorous mathematical analysis is currently of least assistance for design or analysis purposes.

In some cases, when either scant attention has been paid to the propeller design environment or, alternatively, where ship layout difficulties have arisen and high shaft angles or a poor choice of advance angle have resulted, erosion problems can occur which are difficult to resolve. One such case is that of a fast displacement craft where deep cavitation erosion in the blade root regions was produced after only 30 minutes running at full power. The problem was identified as originating in the angle of attack variations at the blade roots due to the propeller shaft's inclination. (FIG. 37) shows the full scale cavitation on the propeller blades when operating at high speed and observed through windows in the hull. Of interest is the cavitating structure at the trailing edge of the root cavity which was in the root erosion region.

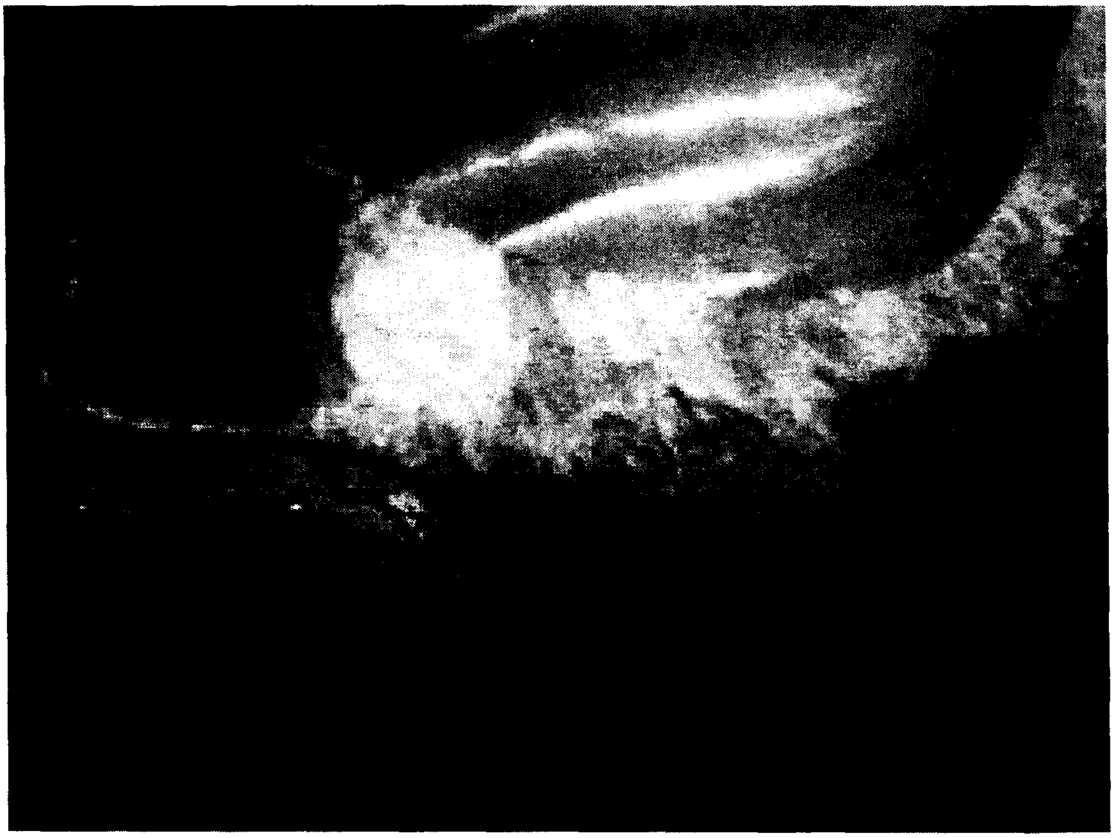


FIG. 37—FULL SCALE ROOT CAVITATION ON A HIGH SPEED PROPELLER

An air injection system was devised based on previous successful experience by TID.[14] This proved to be moderately successful in that it allowed the vessel to enter restricted service while a more permanent solution was found. Full scale observation of the air injection process on the ship provided key information for a subsequent series of model tests. The video images showed that due to the high pitch ratio of the blades, in excess of 1.7, this conventional approach did not allow sufficient air to be held in the cavities and hence could not fully cushion the cavity collapse process producing the erosion damage.

Model tests were used to develop an alternative air supply configuration and machined holes in the blade roots at approximately the one third and two third chord positions at the 0.3R radius. By an iteration process a suitable air flow rate was determined such that a minimum speed loss was achieved while maintaining, intact, stencil ink coatings on the model blades; these being used to assess the tendency to cavitation erosion. The machined holes in the blade roots appeared to delay the onset of erosion, and hence increased the speed at which air injection would need to be started. In addition, the holes provided a mechanism by which air could transfer from the pressure side to the suction side and, hence, ensured that air was resident in the root cavities for a substantial part of each blade revolution.

The vessel is now in normal service and after the equivalent of one full year of high speed running some minor erosion damage is still present but it appears to have stabilised.

Noise Emissions from a Cruise Ship

Residents living near a cruise ship terminal complained about the noise from one particular ship. Noise measurements were taken at a number of locations in the residential area, both with the ship at its berth and with the berth unoccupied. Overall A-Weighted sound pressure levels were considered acceptable but analysis of the noise signals when the ship was berthed showed a distinct peak at 37.5 Hz. This gave the noise a tonal characteristic and was the feature of the noise spectrum which the residents found annoying.

The source of the tonal noise was identified as exhaust gas pulses from the four diesel generators exiting the funnel and the frequency of the pulses corresponded to the firing frequency of the six cylinder, 4 stroke diesel engines. Pressure measurements on the inlet and outlet sides of the installed exhaust gas silencers showed that frequency components less than 50 Hz. were not attenuated effectively by the silencers.

An assessment of the generator exhaust gas system dynamic behaviour was made using Lloyd's Register's 'MERLIN' diesel engine simulation program. The system, as described earlier, allows the thermo-fluid dynamics of a particular system to be simulated against time, which permits a functional analysis of the design details to be undertaken. The predicted results showed good correlation with measurements and further simulations were run to investigate the effect of fitting an additional plenum downstream of the existing silencer to act as a buffer to dampen the pressure pulses. These calculations demonstrated that such a solution would be effective.

The measurement and calculation results were then used by a silencer manufacturer to design a secondary silencer and these were fitted in each of the four exhaust trunkings. Follow-up measurements confirmed a significant reduction in low frequency noise components, (FIG. 38).

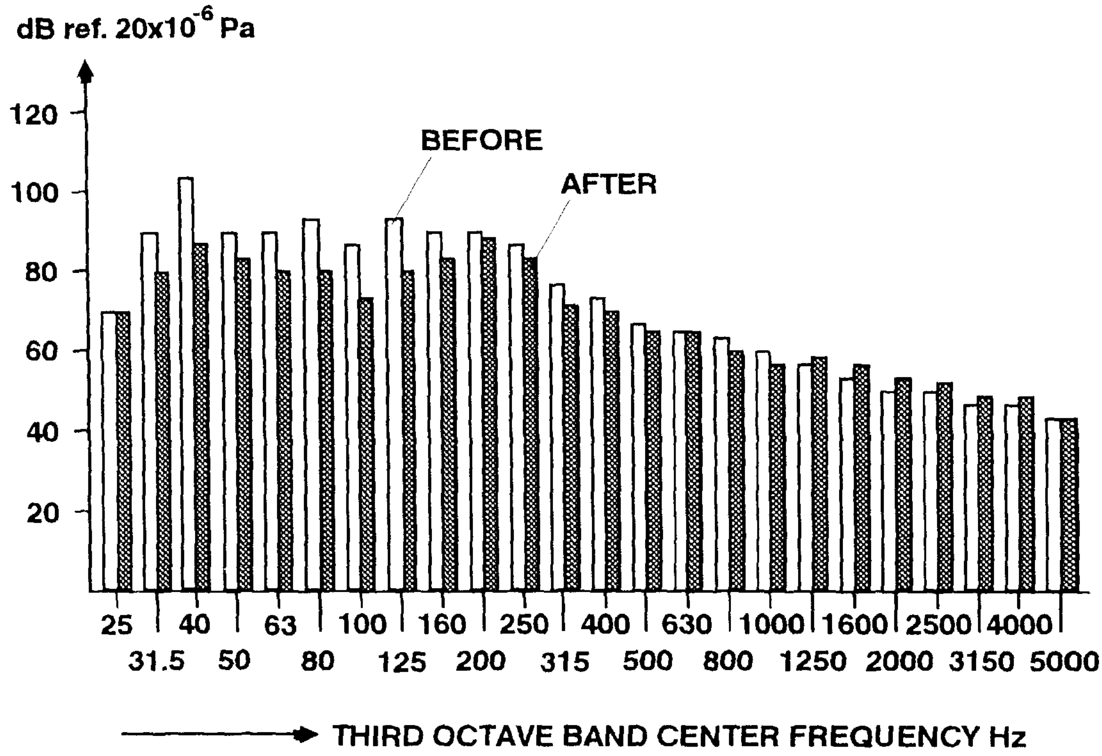


FIG. 38—EXHAUST GAS SOUND PRESSURE LEVELS AT 4M FROM FUNNEL OUTLETS

Land Based Investigations

Loose crank bearing bolt

TID surveyors are often called upon to attend the site of a mechanical mishap to conduct an investigation into the cause. Sometimes the cause is surprisingly simple, as in this case of the loose crank bearing bolt.

The engine which suffered damage was an eight cylinder two stroke diesel engine arranged to drive a large alternator. The first indications that it had a problem were loud knocking sounds. Unfortunately, before the engine could be brought to a stop, one of its connecting rods abandoned its relationship with the crank pin and emerged from the crankcase, causing substantial damage. The final state of the connecting rod can be judged from (FIG. 39).

As the various damaged parts became available, possible alternative causes were continually tested for credibility. By this process, attention was increasingly drawn to the need to find the crank bearing bolts, which were not initially accessible. When they were retrieved, one of them was found to be virtually intact, but without its nut. This nut, its locking device and locking screw were all found separately. The nut threads for about one and a half turns nearest to the lower face were seen to have been damaged by axial shear. About one quarter of the first turn, although present, had been separated from the nut around its circumference. The other bolt had broken under severe combined tensile and bending load and its nut with locking device and screw were still present on the broken end.

Furthermore, a bolt in a different connecting rod was found to be 1 mm slack with its nut locking device and locking screws correctly fitted.

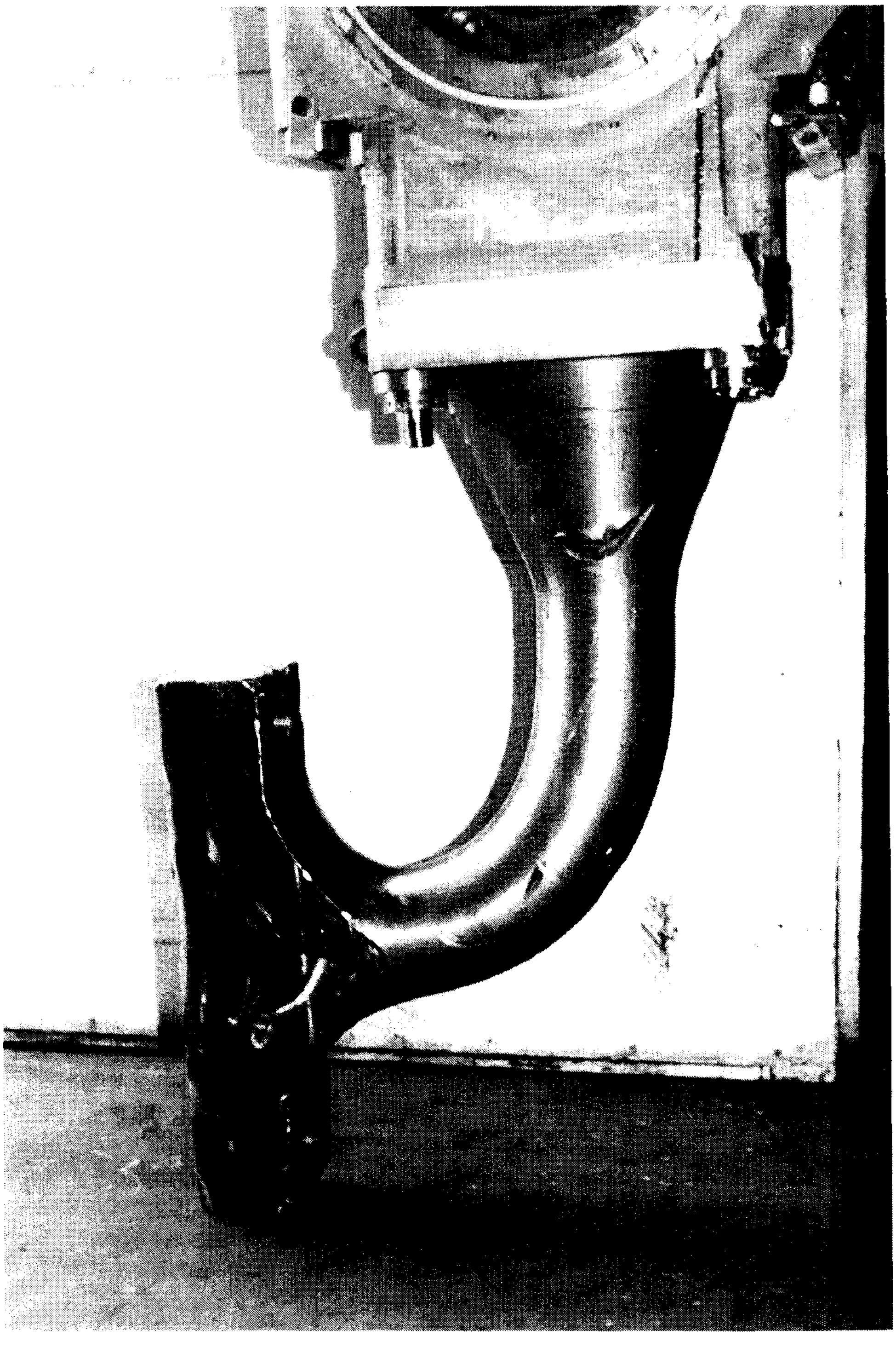


Fig. 39—Damaged connecting rod

This evidence clearly showed that the slackening and loss of the intact bolt was the initiating cause of the engine damage. In searching for the reason why the bolt became slack, the effects on the bolt of possible cylinder malfunctions were calculated and physical tests of similar bolts were conducted. The key finding from this work was that if the bolt had been properly tightened, the probability that sufficient torque could have been generated to loosen the nut against the frictional resistance of the threads and of the nut face was minimal. The torque due to tensile load, which normally acts to unscrew a nut, was found to be one eighth of the frictional torque resisting such movement. It was concluded, therefore, that the damage resulted because the bolt had not been properly tightened to the correct tensile load.

The routine maintenance procedures adopted for this installation involved periodic checks of the bolt tension. The method used was to slacken and re-tension the bolts and, as such, this procedure had the potential to introduce error.

Hot oil pump failure

In this case a fractionation process was used to distil highly flammable products such as ethane, propane and butane from natural gas liquids pumped from oil fields by pumping hot oil through distillation columns. At one facility, the failure of a hot oil pump resulted in a fire which was extinguished by a water spray deluge system.

The investigation commenced with an examination of the damaged components and debris collected from the pump vicinity. Selected components were subjected to non-destructive testing, dimensional checks and metallurgical analysis. Additionally, records of the maintenance undertaken, operational process data and condition monitoring data were examined while, simultaneously, a number of experienced maintenance personnel were interviewed to determine whether the pump maintenance practices had changed since the plant was commissioned thirteen years earlier.

It was concluded that failure of the pump was caused by high temperatures generated at the pump thrust bearing, followed by bearing collapse, bearing seizure and fracture of the pump shaft due to excessive torsional shearing loads. The high bearing operating temperatures were attributed either to the use of defective bearings or to incorrect bearing assembly procedures. Inadequate lubrication or misalignment of the pump and drive motor were discounted. Recommendations included a review of the procedures for hot oil pump maintenance, procurement and condition monitoring.

Scaffold Structure Integrity

The aircraft maintenance structures, (FIG. 40), are large aluminium scaffold towers and are used by the owner to provide access to the tail area of aircraft to permit servicing and repairs. As these structures were over 25 years old and were proving increasingly difficult to maintain, the Client invited LR to undertake an independent structural assessment to advise on their continued use.

The project comprised three main phases, namely the:

- Strength assessment via structural analysis
- The strength assessment via load testing
- Ongoing condition monitoring of the structure.

In the first phase a series of structural analyses were performed to investigate the effects of the current operational loadings. The structural analysis model was based on the fabrication drawings with material properties being the design values quoted on the drawings and the section sizes those measured on site. Assessment loads were developed in conjunction with the client on the basis of the operational history of the structures. The strength assessment was carried out with the structures having to satisfy the assessment criteria of stability, stiffness and strength. The second phase comprised a series of load tests which were performed on key member connections, deck platforms and the towers to confirm the results of the strength assessment via structural

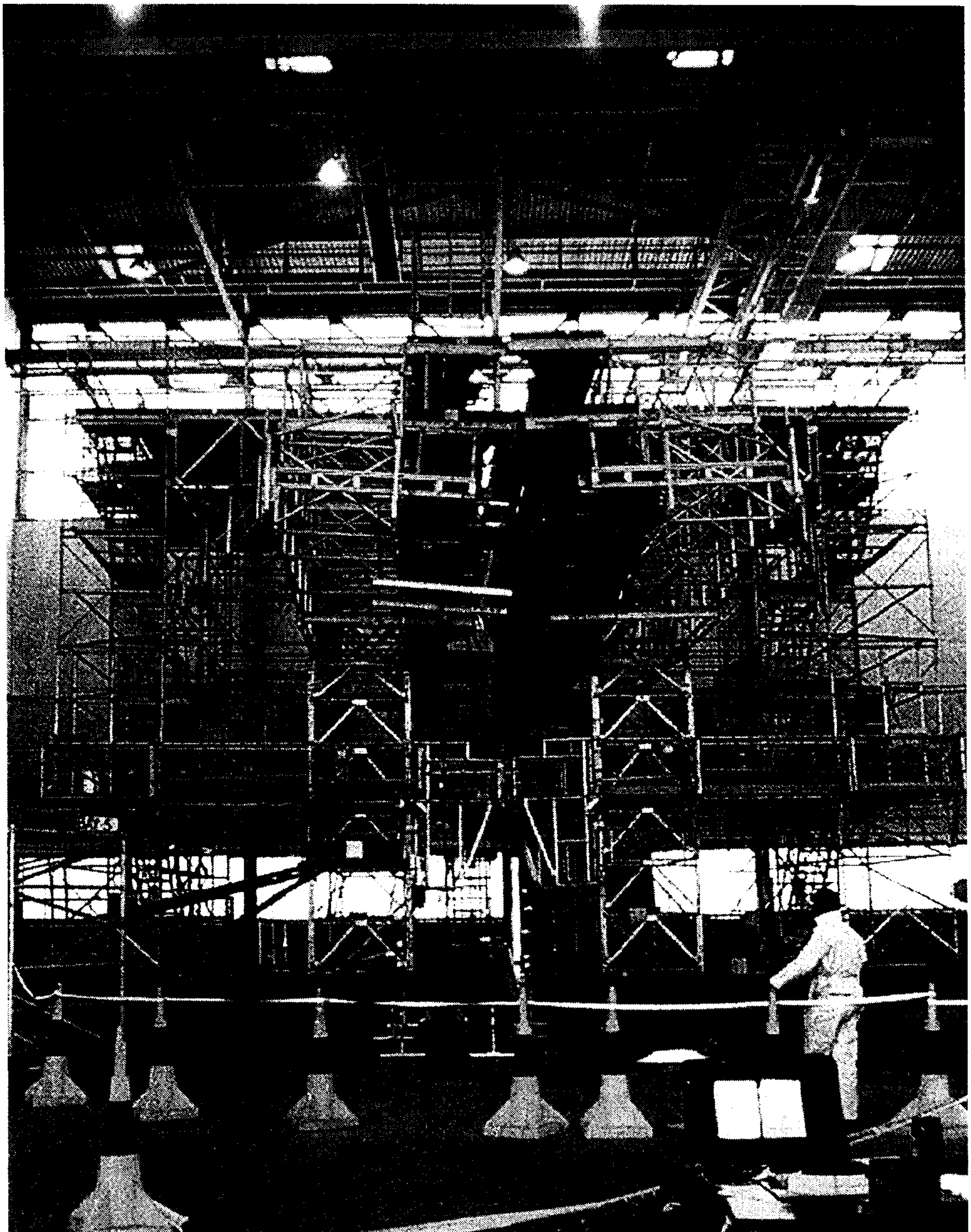


FIG. 40—SCAFFOLD TOWER

analysis. The member connections were load tested to destruction to determine if the likely failure mode was the ultimate strength of the connections. In the case of the deck platforms and the towers, the static load tests were intended to show whether the structures could carry the unfactored (nominal) live loads without exceeding the serviceability limit state and also whether they could carry the factored live loads without exceeding the ultimate limit state. In the third phase, TID developed and implemented a programme of visual inspections and non-destructive examinations of the structural members and joints to monitor the structural condition of the tail docks. On a biennial basis, visual inspections and non-destructive examinations were undertaken to assess the current condition of the tail docks. The purpose of this ongoing condition monitoring was to:

- Confirm that the framed members of the inner tower, outer tower and cantilever beams were as shown on the fabrication drawings.
- Inspect all framed members of the tail dock to ascertain whether they displayed signs of structural or fatigue damage.
- Inspect all joints to the framed members of the tail docks to ascertain whether they displayed signs of structural or fatigue damage.

The structural analysis and load testing of the tail docks supported the view that the tail docks were capable of withstanding the assessment loads provided that they were constructed in accordance with the fabrication drawings. TID recommended that the tail docks continued to be inspected by the client prior to each use to ensure that all members were present and that the ongoing condition monitoring programme be maintained.

Offshore based investigations

'Pig Hunting' in offshore pipelines

The furring of flow lines by heavy molecular weight paraffins is becoming more common as a result of the increasing use of satellite wells to extend the recoverable resource from existing offshore tower structures. To avoid this problem, a utility line is normally provided to allow periodic flushing with solvents and the use of cleaning pigs to remove hard and soft deposits. However, variability in the contents and flow can still lead to blockage of the line by, for example, the build-up of sludge and wax ahead of a hydrate or a cleaning pig.

In a recent investigation TID was invited to locate three cleaning pigs which had become trapped in a flow line running between two satellite wells. The task was complicated by a number of factors: a persistent strong current of 5 to 6 knots caused by a river outflow, shallow water of around 6 to 15 m depth, almost nil visibility, a large Atlantic swell and the pipeline being buried up to 1m in places.

The device used was a high sensitivity calliper strain-gauge, designed to measure the radial extension and compression of the pipe due to changes in internal pressure. The normal test procedure is to pressurise the pipe from each end in turn, to establish communication with the measurement position, initially at the centre of the line. The next measurement is made at the mid point of the side identified with the blockage. This procedure is repeated until the blockage is located to the required accuracy. For example, to locate a blockage to ± 0.5m in a 1000 km flow line requires just 20 measurements. The opposite end of the blockage is located by repeating the measurements on the opposite side starting at a point where the pipe is expected to be clear.

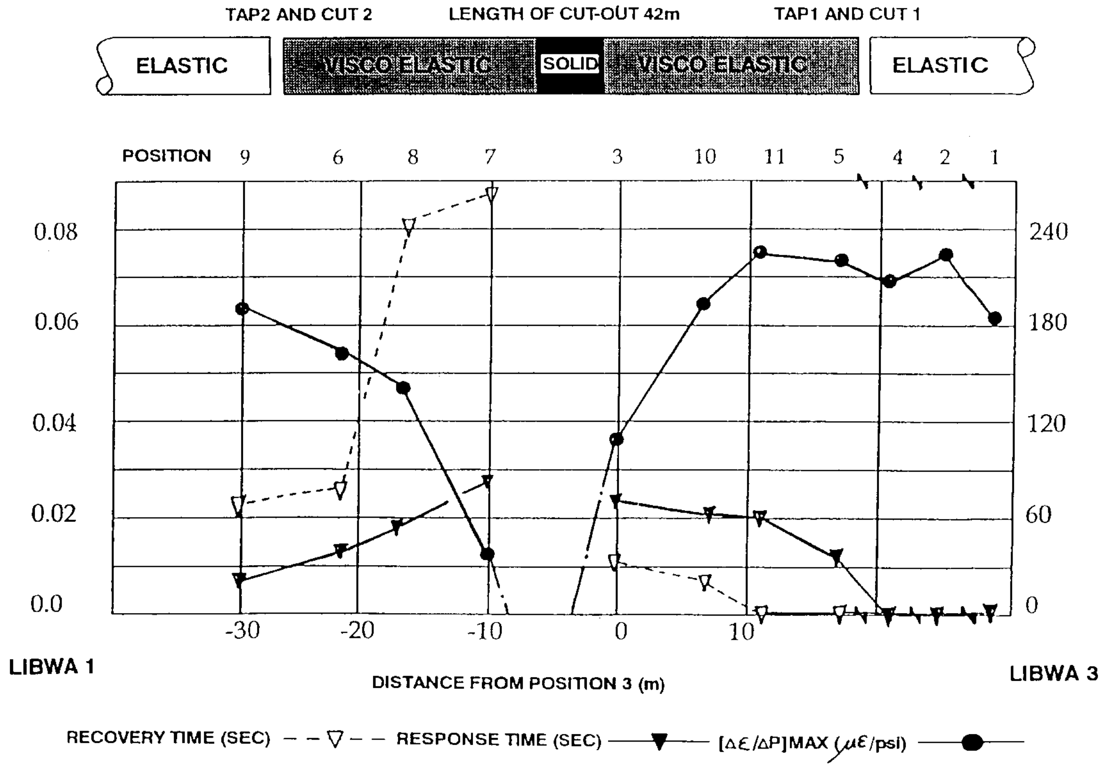


FIG. 41—VALUES OF |Δε/ΔP| MAX AND THE CORRESPONDING RESPONSE TIME AND RECOVERY TIME AS A FUNCTION OF MEASUREMENT POSITION ON THE PIPE

In the case in question the trapped pigs and the full extent of the blockage, which included the build up of wax on each side, were determined from just 11 measurements, (FIG. 41). A hot-tap was then made in the clear parts of the pipe on each side of the blockage and water flushed back to the respective ends of the pipe. The blocked section was then removed and the ends raised to the surface for the fitting of connecting flanges to receive a replacement spool-piece.

Leak testing by acoustic emission methods

The Department maintains a general capability for acoustic emission based crack detection and monitoring in structures and dynamic engineering components. Recently TID was requested to consider the feasibility of acoustic emission methods for leak monitoring in an underwater oil production manifold. The numerous control valves and interconnecting spool-pieces in the manifold presented numerous possibilities for leakage, which may not be evident until appreciable loss has occurred. A demonstration of leak tightness at start-up and during early operational service was considered desirable. Access to an identical manifold, under repair at the time, provided the opportunity to study the propagation of sound in the structure to determine the most economical arrangement of sensors for the detection of leaks at the high risk locations. It was concluded that eight sensors distributed around the central valve block and one sensor on each of the connecting hubs with the manifold, would provide the required sensitivity to leaks at the valve seals and hub-to-valve block spool-piece connections, depending on the level of background noise. The high hydrostatic pressure existing when the manifold is on the sea bed would affect the leak velocity and, consequently, the conditions for turbulence and cavitation. Acoustic emission sensors attached to the structure were favoured over the use of hydrophones, which are less sensitive to leaks internal to a structure and would be affected more by water-borne noise interference.

It was proposed that the acoustic monitoring system would sample the root mean square signal level within octave frequency bands covering the range 10 kHz to 160 kHz. This information would be relayed to the floating production, storage and off-loading vessel via an existing hard-wire digital communications link, which could also be used to supply power to the system. Alternatively, the acoustic monitoring system could be operated as an autonomous unit with battery power replenishable annually. A sonar beacon would be actuated to indicate the existence of a possible leak. At this point a service vessel in the field could be mobilised to the location of the manifold and the raw data history recovered via an acoustic telemetry link. These data would then be used to help verify the existence of the leak before deploying a remotely operated vehicle for visual confirmation.

Condition monitoring

As a natural extension to its engineering measurement and analysis capabilities, TID maintains a strong interest in machinery and structural condition monitoring.

Marine machinery condition monitoring

The condition monitoring of marine machinery is a topic which in recent years has excited considerable interest in the marine industry. This interest, which can be gauged from (FIG. 42), has, in part, been stimulated by the considerable use of the various techniques in the aviation and land based industries and the potential for technology transfer to the marine and offshore industries. This condition monitoring activity, dealing as it does with the detection of developing failure, follows on from TID's failure and vibration investigation work and has formed a significant extension of the Department's work for the last eight years. Furthermore, this activity complements Lloyd's Register's primary interest in machinery condition monitoring which is in using the techniques to enhance survey procedures to obtain the required levels of safety in the most cost effective way to ship operators.

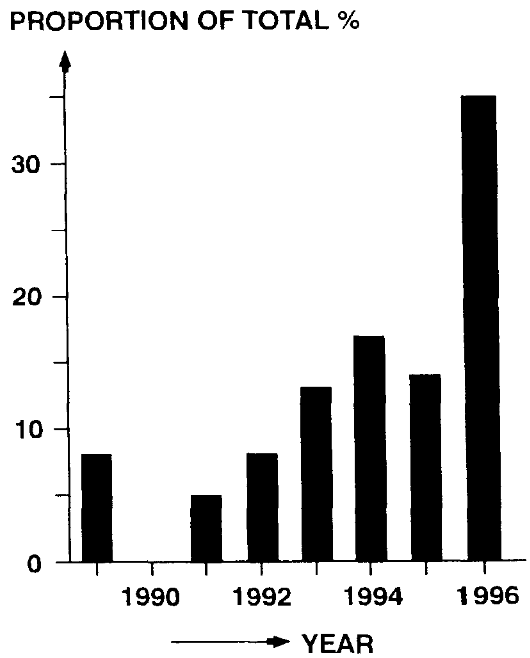


FIG. 42—NUMBER OF MACHINERY CONDITION MONITORING UNITS INSTALLED BY TID

The first system was supplied in 1989 and the development of that system has now been installed on many vessels and is known as the Vibration Monitoring System (VMS). Recently the developments in Lloyd's Register's ongoing research programme have led to a second system which embraces both lubricating oil and machinery performance monitoring in addition to vibration monitoring. This new system is called the Integrated Condition Monitoring System (ICMS). For LR classed ships the *ShipRight* procedures include several condition monitoring facilities. Among those directed towards machinery condition monitoring, the application, for example, of either the

VMS or ICMS systems together with an approved planned maintenance system will lead to the award of the PMS(CM) notation in the Register Book. Alternatively, either of the LR systems can be used on their own without recourse to the *ShipRight* procedures and some are used on ships classed with other classification societies. In developing these two systems TID's underlying focus has always been on installations which are robust, relatively cheap and, where possible, in making the fullest use of instrumentation which the ship has on board as standard items.

In the application of the VMS System a list of all the rotating machines contained in the master list of surveyable items is compiled together with other machines of particular concern to the ship operator. The desirability of applying vibration monitoring to each of the machines on the list is then carefully considered. Each vibration or performance parameter is assigned a level at which an alarm will be generated, indicating to the user that action is required. The alarm levels used are based on TID's experience, coupled with the recommendations given in various International Standards, and these are reviewed periodically. The machine details, measurement parameters, locations and limits then form a database created using the software supplied with the system.

The VMS System hardware comprises a lightweight battery powered portable data collector connected to a hand held vibration transducer and a desktop computer. The data collector is connected to the host computer and loaded with information about the machines to be monitored. The operator will be informed by the display screen of the data collector which machine is to be monitored and the sequence in which the measurements are to be taken. If any parameter exceeds a pre-set level, a warning is given at the time of the measurement. The system also includes a facility to input manually the value of parameters associated with the performance of a machine, such as pressures or motor current. After the measurements have been taken the data collector is again connected to the host computer and the measured data downloaded on to the database. The associated computer program supports various functions, including trend analysis, anomaly reporting, guidance on the possible nature of faults and spectral analysis. In addition to these functions a record history of the machines will eventually be created. The software can provide an interface with an engine room data logger which will allow performance parameters and running hours to be entered automatically to the machinery database. An additional facility is communication with a computerised planned maintenance system so that vibration readings exceeding alarm levels can be used to initiate maintenance recommendations.

In contrast the underlying philosophy of the ICMS system is that the greatest potential benefit from a condition monitoring system can be gained from combining the features of lubricating oil analysis, vibration monitoring and machinery data logging into an integrated system. As a consequence all of the features from the earlier VMS system are retained and are augmented with the data from regular lubricating oil analysis, maintenance and operating environment histories and machinery performance data. The underlying structure of the ICMS system is illustrated in (FIG. 43).

With regard to lubricating oil analysis the results of regular tests can be recorded either manually or automatically, depending on the facilities available from the particular testing organisation. Furthermore, any commentary that is provided on the analysis, such as that given by LR's LQS service, can also be added. These data may then be recalled and plotted to examine trends in particular components or compared with other performance or condition parameters. This facility reduces the need, in some cases, for significant manual intervention in the data analysis and interpretation.

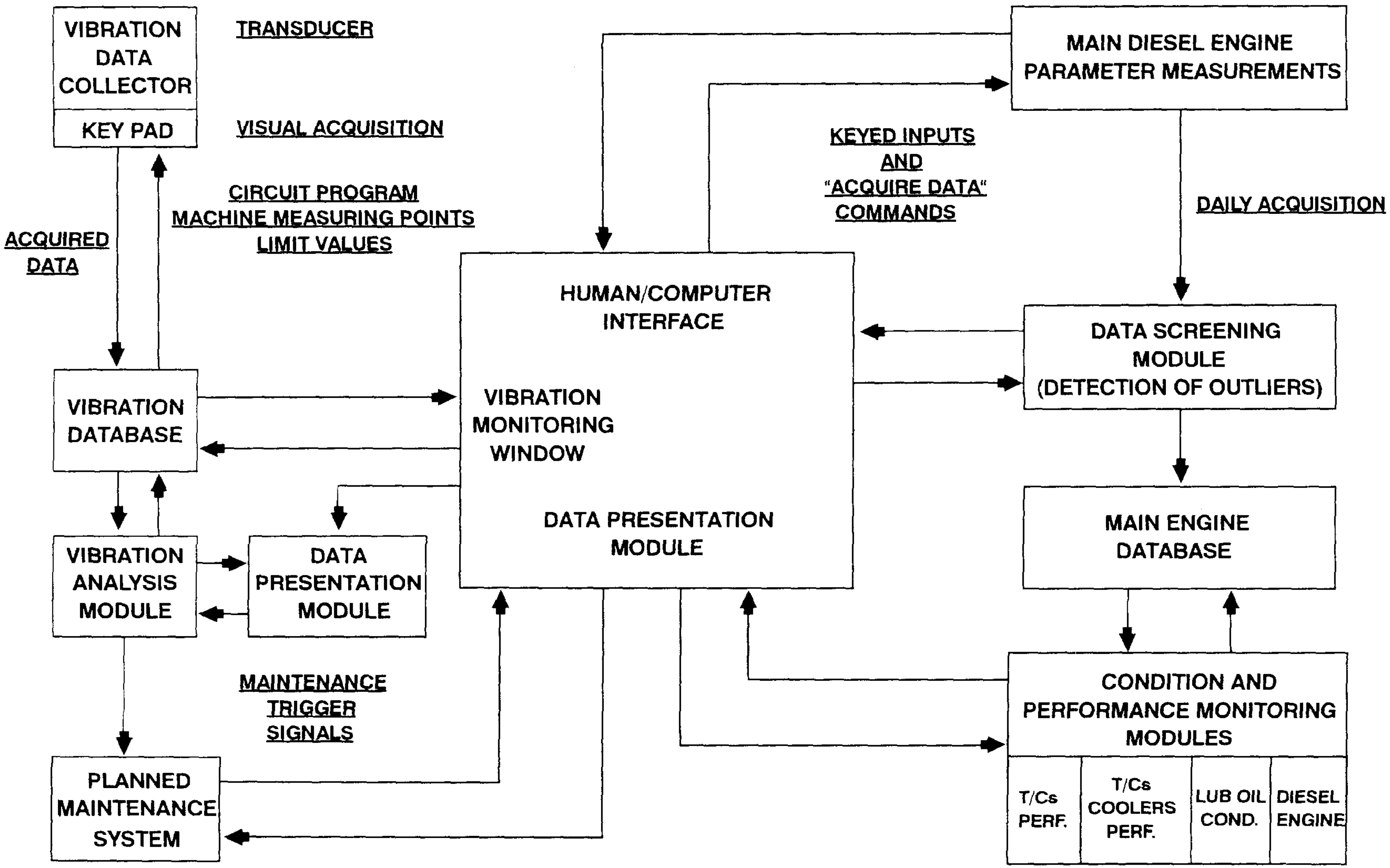


FIG. 43—STRUCTURE OF ICMS SYSTEM

An ability to include machinery performance data from the ship's normal data logging system provides a means of augmenting the other vibration based condition monitoring information. Typically, in a direct drive slow speed diesel marine application some one to two hundred different parameters might be available for analysis in terms of trending, cross plotting and to facilitate use of the system guidance is given as to the more suitable combinations of data.

Analysis within the ICMS system is directed towards providing an aid to the ship operator's shore and sea staff rather than endeavouring to replace their skills: condition monitoring technology as applied to the diesel engine is still a long way from being able to achieve this. To help in this supplementary capacity a feature has been incorporated into the ICMS system which allows the input of relevant operational and environmental information into the database.

Lloyd's Register's research into machinery condition monitoring systems commenced in the early 1980's when it was clear that systems with imbedded levels of intelligence would need to be classed in the foreseeable future. At that time much work had been achieved with rotating machinery within the aviation industry; chiefly with pumps and gas turbines. As a consequence LR chose to extend the field of then available knowledge into reciprocating machinery.[15]

Experience has shown that condition monitoring systems need to be designed with the requirements of the end user firmly in view. As such, in order to gain acceptance from the crew and derive the maximum benefit from the implementation of the system, the data presented and the operation of the system must be consistent with the operator's experience and training. The achievement of this ergonomic balance is seen as critical. In the case of VMS type systems, ship's crews generally become comfortable with the level of complexity very quickly. The greater degree of flexibility offered by next generation systems, such as the ICMS system, normally require a longer familiarisation period and an extended training period is beneficial if the full potential of the system is to be realised.

Currently some machinery condition monitoring systems are successfully used to provide data for seeking the deferral of opening up of machinery for routine survey purposes. This trend is likely to increase as confidence grows in the intrinsic reliability of condition monitoring systems and their ability to predict failure. Furthermore, these trends will be further enhanced by the introduction of the emerging ISO STEP standards such as that for the AP226, Ship Mechanical Systems, which defines through life data exchange protocols. Upmost importance, therefore, needs to be given to ensuring the traceability and reliability of the condition monitoring data used for survey purposes. In particular, matters such as authenticity of the data, the conditions under which they were measured and the operator qualification, the validity of the analysis technique together with sensor reliability considerations, defect visibility, sufficiency of information and the ship's operating regime must all be addressed and satisfied.

Structural Condition Monitoring

In addition to its work in machinery condition monitoring, the Department is called upon, by virtue of its wide experience in condition monitoring and measurement, to consider structural condition monitoring in the land based and offshore industries. By way of example of land based condition monitoring is some ongoing work on a family of glued segmental prestressed concrete motorway bridges, (FIG. 44). The project comprised a number of phases, the most important of which were the analysis of historic data to study trends and diagnosis; the devising of a strategy for ongoing data collection, reduction and diagnosis; and ongoing condition monitoring.

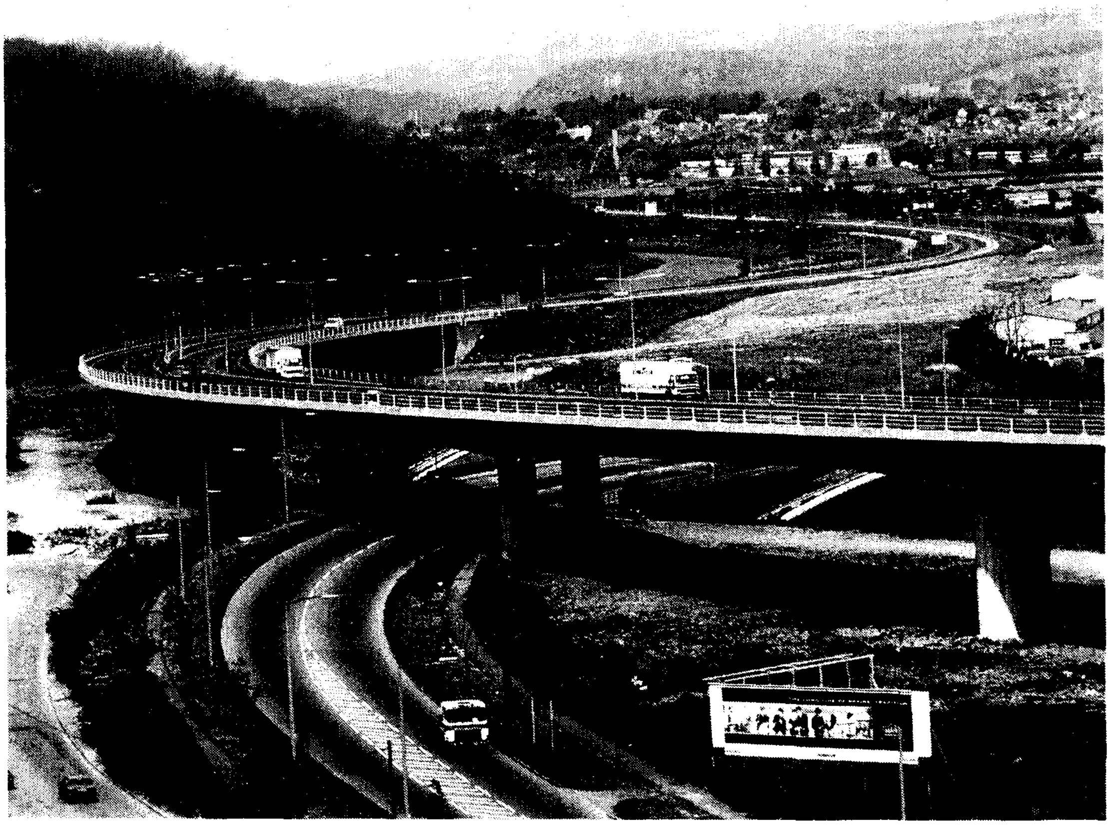


FIG. 44—SEGMENTAL PRE-STRESSED CONCRETE VIADUCT

Ninety two vibrating wire strain gauges were originally installed in 1987. Data logging was on a daily basis in the early days but later opened out to weekly, monthly and eventually yearly intervals covering the historic period 1987–1995. Over this period plots of microstrain against time showed a reasonable correlation with predictions from an analytical model. The scatter of the strains measured by individual gauges remained almost unchanged over the life of the gauges. Furthermore, the most significant data collected showed strain versus time relationships broadly similar to those reported for other prestressed concrete structures; strains during the casting and erection period rising relatively rapidly while strains during service rise relatively slowly and steadily. (FIG. 45) shows a typical time history recorded for one location in the bridge from which it can be seen that a good functional relationship to the in-service phase can be achieved using an exponential formulation.

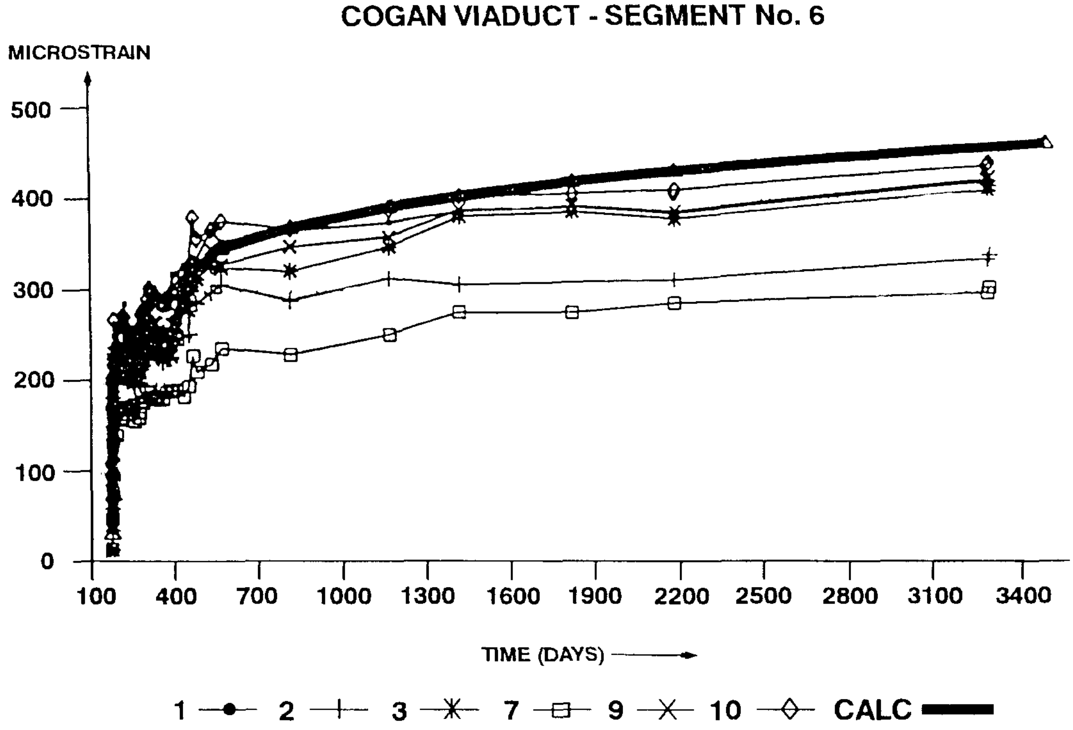


FIG. 45—STRAIN HISTORY OF A LOCATION ON A VIADUCT

The vibrating wire sensors only give information relating directly to the location at which they are installed. As such, it is unlikely that the locations of the gauges will correlate exactly with specific areas of deterioration, such as localised corrosion of a prestressing tendon. Consequently, the strain gauges are unlikely to identify deterioration of the structure remote from them. To overcome this problem, TID established a set of supplementary levelling points at regular intervals across the bridge which can be used to monitor the overall profile of the structure as well as any settlement of the supports.

Offshore installation structural condition monitoring can be considered using acoustic emission methods. It is known that the greater part of the fatigue life of welded tubular joints in offshore structures is taken up by **Stage 2** crack growth and the remnant fatigue life of a defective node joint can be significantly greater than the required safe working life. This is particularly important if a repair is being considered following NDT, since the savings resulting from the avoidance or the safe deferment of such repairs can be considerable.

Over the past ten years, operational experience with acoustic emission systems subsea and topside has provided data on different size fatigue cracks growing under service loading conditions, which have been verified independently by NDT.

Procedures for the detection of fatigue cracks by acoustic emission methods are now available for a range of applications, however, the potential for total volumetric surveillance has yet to be fully realised and the method continues to be used primarily for monitoring critical parts of a structure, for example:

- To distinguish between NDT indications and growing cracks.
- To detect and size cracks in inaccessible locations or hazardous working environment.
- Where the cost of repair or replacement of a defective structural element is high and may be avoidable.

- When approval to continue operating a defective structure is being sought while mobilising for a repair.

Floating production installations and poor fault tolerant structures stand to gain most in the short term. A recent study undertaken by TID of the leg connections with the operating deck of a typical three legged jackup platform showed that a 16 sensor system will be capable of monitoring all primary structural elements in a 9 m length of one leg. Monitoring would be continuous throughout a winter period to ensure that any crack growth was monitored under representative loading conditions and for a statistically significant number of cycles. The same coverage of the remaining legs to detect more advanced cracking, would be possible by multiplexing.

Data are normally retrieved to the shore base for analysis at weekly intervals via a modem. The location of 'hot spot' acoustic emission activity is shown on a CAD representation of the structure, similar to the finite element mapping of stress concentration and ultrasonic C-scan imaging. Defect significance is determined from the acoustic emission source energy and its relationship to wave loading.

Further successful applications of acoustic emission methods by TID have been in the fields of crack monitoring in hydraulic turbine volutes and large journal bearings.

Concluding remarks and some underlying lessons from failure investigations

There are a number of lessons that can be learnt from the failure investigation activities of the Technical Investigation Department of Lloyd's Register. Some are specific to certain areas of engineering application while others are more general in their nature. In the case of specific lessons, the case studies presented throughout the paper have been chosen to highlight some of the more common types of failure. The underlying general lessons can be summarised as follows:

- When undertaking engineering design it is essential to **stand back** from the detail of the design and look at the whole engineering problem. From such an exercise it is possible, for suitably experienced engineers, to identify the weak points in the design.
- Today the investigator, or designer for that matter, has a range of analytical and experimental capabilities at his disposal. In most engineering situations these capabilities only give a partial picture of the problem and, as such, they are aids and not a substitute for sound engineering judgement.
- The Department's data base shows that in a great many instances problems continue to reoccur. Whether this is due to technology moving on in other areas which then reintroduces problems of a similar type in a later age or the lessons of the past not being effectively passed on to new generations of engineers is unclear.
- A great many failures are caused by a lack of attention to the detail design of individual components. Moreover, problems frequently arise when similar care is not exercised in the integration of components into an engineering product or system; especially when different manufacturing sources are involved.
- Similarly, installation, maintenance and operational procedures are critical if failures are to be avoided.

- Investigators must always attempt to identify and understand the prime cause of the problem so that proper judgement can be exercised in deciding whether fundamental changes are needed or if palliative countermeasures will be sufficient.
- The training of engineers in the detailed practical and theoretical skills of engineering is fundamentally important, but more critical is the combining of these skills into a unified professional engineering knowledge and instinct.

Acknowledgements

The Authors wish to thank the Committee of Lloyd's Register for permission to publish this article. Additionally, the experience and knowledge of a group such as TID can only be built up by much careful and hard work from the technical and administrative staff over the years. Consequently, thanks are due to the Authors' many colleagues in the Technical Investigation Department, both past and present, who have contributed to and developed the various techniques outlined in this paper and without whom, the Department would not exist with its present capabilities.

References

1. Santayana, G. 'The Life of Reason; or The Phases of Human Progress'. Vol.1, Ch.12, 1905.
2. 'Marine Exhaust Emissions Research Programme.' *Lloyd's Register*, London, 1995.
3. Carlton, J.S. 'Propeller Service Experience'. *7th Lips Symposium*, 1989.
4. McClimmont, W and McKinlay, D. 'Effect of Hull Movements on the Propulsion Shaft System'. *MER* June 1979.
5. Pinnekamp, W., 'Performance of Marine Reduction Gears with Modern Design Features'. Trans. *I.Mar.E.*, Vol 86, 1974.
6. Proc. 18th ITTC—Report of the Propulsor Committee, Kobe 1987.
7. Holland, C.G.; Wong, S.F., 'Noise Prediction and Correlation with Full Scale Measurements in Ships'. Trans. *I.Mar.E.*, Vol 107, 1995.
8. Banisoleiman, K., Smith, L.A., Bazari, Z. and Mathieson, N., 'Simulation of Diesel Engine Performance'. Trans. *I.Mar.E.*, Vol 105, Part 3, pp 117–135, May 1993.
9. Fitzsimmons, P.A. 'The Analysis of Waterjet Units'. Trans. *I.Mar.E.*, 1997.
10. Suresh, S. and Ritchie, R.O. 'Propagation of Short Fatigue Cracks'. *Int. Metals Rev. 29*, pp 445–476., 1984.
11. Hoekstra, M., 'Prediction of Full Scale Wake Characteristics based on Model Wake Survey'. *ISP*, Vol 22, 1975.
12. Tanaka, I. 'Scale Effects on Wake Distribution and Viscous Pressure Resistance of Ships'. *JSNA*, Vol 146, Dec. 1979.
13. Sasajima, H. and Tanaka, I. 'On the Estimation of Wake of Ships'. *11th ITTC*, Tokyo, 1966.
14. Carlton, J.S. 'Marine Propellers and Propulsion'. *Butterworth-Heinemann*, 1994.
15. Carlton, J.S., Holland, C.G., Rhoden, D., Newbury, M. and Triner, A. 'Recent Operational Experience and Research Relating to Marine Machinery Condition Monitoring'. Trans. *I.Mar.E.*, 1996.