THE MEASUREMENT OF VIBRATION AS A DIAGNOSTIC TOOL

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A brief introduction into the use of the vibration measurement technique as a machinery defect diagnostic tool is given. Details of different types of measuring systems and methods of application are discussed, as well as some case histories.

INTRODUCTION

The measurement and analysis of machinery vibrations is fast becoming an accepted diagnostic tool, providing the maintenance engineer with an early warning of failure. The techniques used today are primarily for detecting vibration which may be too slight for the operator to recognize. Relatively minor vibrations, which if allowed to persist, could lead to more serious trouble and eventual failure. The technique makes use of the fact that all rotating machines vibrate to some degree or other. This is so because it is prohibitively expensive to design and build machinery which is free of vibration.

Two parameters of vibration are normally used for diagnostic work: the amplitude (i.e. displacement velocity or acceleration of the particle being measured) and the frequency at which it occurs. The amplitude of vibration giving an indication of the machines condition and the frequency at which it occurs identifying its probable source.

Although vibration monitoring of machinery is a relatively new technique it is potentially one of the most cost-effective of the various non-destructive test techniques at present available. Correctly used as part of a controlled maintenance programme, together with other associated performance checks, including visual inspection, it can substantially reduce man hours spent on maintenance. The cost of machinery repairs can also be reduced as repairs can be planned to be carried out prior to catastrophic failure rather than on a time basis. Plant availability is also increased.

It can also be used as a quality control tool, reducing the number of defects built into new or refurbished machinery. A list of machinery faults which are most likely to be detected by the measurement of vibration is given in Table I. It will not usually detect worn or damaged plate valves in reciprocating air compressors or the fall-off in performance of pumping units, due to excessive sealing ring/impeller land clearances, or other similar defects.

TABLE I—LIST OF MACHINERY FAULTS WHICH IT IS CONSIDERED MAY BE DISCOVERED BY VIBRATION ANALYSIS

1) Static or dynamic unbalance or eccentricity, broken impeller or rotor blades.
2) Uneven firing.
3) Worn or damaged gears.
4) Worn or damaged bearings or bearing housings. Effects of fretting corrosion.
5) Bent shafts.
6) Mechanical slackness or insecurity.
7) Onset of cavitation.
8) Shafts becoming misaligned.
9) Presence of solid bodies in a pumps fluid.
10) Incorrect re-assembly after maintenance.
11) Absence of lubricant.
12) Damaged or misaligned drive belts.

UNITS OF MEASUREMENT

Vibration

Although the amplitude of vibrations are measured in terms of various linear units, i.e. mm, mm/s, mm/s² etc., the range required to be covered is so wide that it has been internationally agreed to use a logarithmic representation of acceleration or velocity measurement. The system which serves as best is the "Bel", which is the standard method of power level comparison.

As the Bel is a large unit, for convenience the decibel abbreviation dB is normally used and is defined as:

(1) Acceleration (adB) $= 20\ Log_{10} \frac{a1}{a2}$ decibel

where $a1$ = the rms or peak measured acceleration in mm/s²

and $a2$ = the predetermined reference level, normally $0dB = 10^{-2}$ mm/s²

(2) Velocity (vdB) $= 20\ Log_{10} \frac{v1}{v2}$ decibel

where $v1$ = the rms or peak measured velocity in mm/s

and $v2$ = the predetermined reference level, normally $0dB = 10^{-5}$ mm/s

To avoid misinterpretation of results the reference level on which the measurements are based should always be given.

AdB levels are easily converted to *VdB's* and *vice versa* should it be so required. Conversions covering the frequency range 20–10 000 Hertz in 1/3 octave steps are given in Table II.

The decibel is also used in acoustics, for sound pressure measurement the reference level in air being 2×10^{-5} N/m².

Frequency

The standard unit of frequency is the Hertz (Hz), although cycles per minute which is directly equivalent to rev/min is still used in some quarters:

$$1 \text{ c/min} = 60\ Hz$$

Frequency can be defined as "the rate of repetition of a periodic phenomenon with respect to time". Frequency (F) being the reciprocal of the period (T):

$$F = \frac{1}{T}$$

A frequency spectrum may be sub-divided into a variety of sections termed discrete: octave or fractional octave bands: discrete frequency being at intervals of one complete cyclic function.

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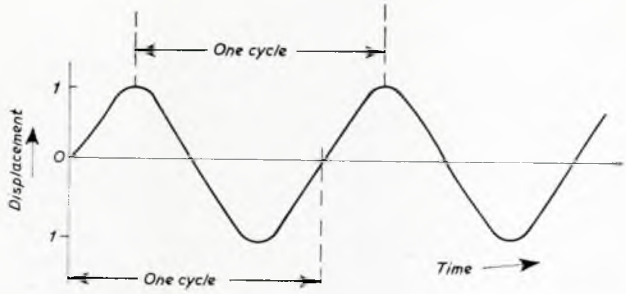


The octave is a pitch interval of ratio 2:1

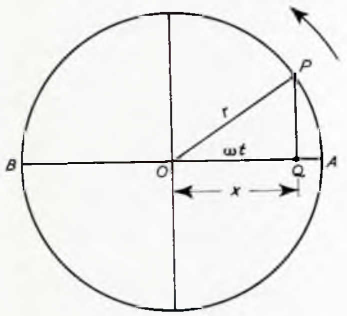
$$\frac{F1}{F2} = 2N$$

where $F2 - F1$ = the Pass Band
and $N = 1$ (for octave intervals)
$= 1/3$ (for 1/3 octave intervals) etc.

A list of octave and 1/3 octave pass bands and mid-frequencies is given in Table III.

RELATIONSHIP BETWEEN DISPLACEMENT, VELOCITY AND ACCELERATION

A particle displaced about a mean point has both a velocity and acceleration. The relationship between these is best illustrated as follows:



If P rotates at radius r with uniform angular velocity ωt then the projection of P on $A\ O\ B$ (Q) is said to move with $S\ H\ M$.

At any instant the horizontal component of vector r (displacement x) is equal to $r\ Cos\ \omega t$.

$$\text{The velocity of } Q \text{ towards } O = \frac{dx}{dt}\, r\ Cos\ \omega t$$
$$= -\omega r\ Sin\ \omega t$$
$$= \omega r\ Cos\ \omega t + \frac{\pi}{2}{}^{*}$$
$$= \omega x$$

*Velocity leads displacement by $\frac{\pi}{2}$

$$\text{Acceleration of } Q = \frac{d^2x}{dt^2}\, r\ Cos\ \omega t$$
$$= -\omega^2\, r\ Cos\ \omega t$$
$$= -\omega^2 x$$

Acceleration is minimum when velocity is maximum.

Example

A displacement of 0·001 mm rms is measured at a frequency 250 Hz (if vibration severity is measured in peak values multiply by 0·707)

Displacement x = 0·001 mm rms
Velocity ωx = $0{\cdot}001 \times 2\pi \times 250$ = 1·571 mm/s rms
Acceleration $\omega^2 x$ = $0{\cdot}001 \times (2\pi \times 250)^2$ = 2477 mm/s² rms

$$VdB \text{ rms} = 20\ Log_{10} \frac{1{\cdot}571 \text{ mm/s}}{10^{-5} \text{ mm/s}} = 84$$

$$AdB \text{ rms} = 20\ Log_{10} \frac{2477 \text{ mm/s}^2}{10^{-2} \text{ mm/s}^2} = 88$$

VIBRATION MEASURING INSTRUMENTS

Each characteristic of a machines vibration spectrum indicates some significant factor about the vibration. Amplitude, whether it be displacement, velocity or acceleration, tells us how severe the vibration is, or in terms of machinery condition how good or bad it is. The frequency at which it occurs guides us to the possible cause. A typical machine comprises of a number of parts each rotating at a different frequency and contributing a different amount of vibration to the spectrum. Instruments for diagnostic work must therefore be capable of measuring individual vibration at their respective frequency as well as the total level. A typical block diagram of a vibration measuring system is shown in Fig. 1.

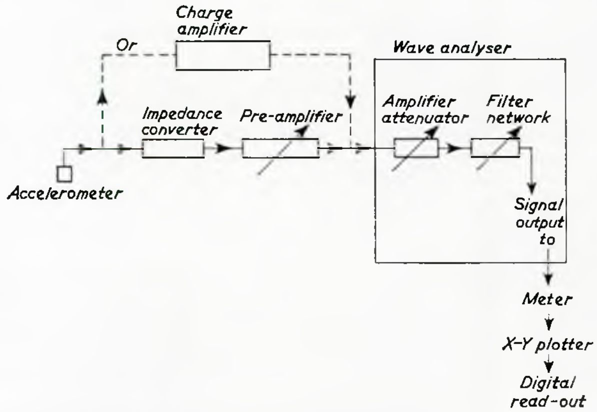


FIG. 1—*Block diagram of vibration measuring equipment*

A variety of instruments are available ranging from simple total vibration level indicators to complicated custom built, computerized monitoring/recording systems. Although each has its merits the basic requirement for use in the marine or general maintenance field is a system incorporating the simplicity of the former with the accuracy of the latter.

Although not fully meeting this requirement, a number of useful instruments are available, see Fig. 2. These can be broken down into two categories:

a) systems which measure vibration using an *accelerometer* as the transducer, together with an octave, 1/3 octave or discrete frequency analyser: the intensity of vibrations normally being measured in decibels;
b) systems which measure the displacement or velocity of a machine vibration in mm or mm/s respectively *via* a *Seismic* pick up linked to a discrete frequency analyser.

Some instruments are in fact capable of displaying the measured amplitude of vibration in each of the referred modes. The fundamental and most important difference is in the type of transducer used. The relative merits and disadvantages of using one or other of the transducers has for years been the basis of much argument.

Definition of Analysers

Discrete frequency: a tuneable wave analyser capable of measuring a vibration wave form over a specified frequency range by means of a filter network, normally has the additional ability to measure total vibration levels.

Octave/1/3 octave frequency: a wave analyser capable of measuring vibration levels at predetermined points within the

TABLE II—ACCELERATION DECIBEL/VELOCITY DECIBEL CONVERSION CHART (1/3 OCTAVE STEPS)

Frequency Hertz	Velocity dB to Acceleration dB	Acceleration dB to Velocity dB
20	−18	+18
25	−16	+16
32	−14	+14
40	−12	+12
50	−10	+10
64	− 8	+ 8
80	− 6	+ 6
100	− 4	+ 4
125	− 2	+ 2
160	Acceleration dB = Velocity dB	
200	+ 2	− 2
250	+ 4	− 4
320	+ 6	− 6
400	+ 8	− 8
500	+10	−10
640	+12	−12
800	+14	−14
1000	+16	−16
1250	+18	−18
1600	+20	−20
2000	+22	−22
2500	+24	−24
3200	+26	−26
4000	+28	−28
5000	+30	−30
6400	+32	−32
8000	+34	−34
10000	+36	−36

frequency range of the instrument filter network. Some instruments are also capable of measuring at discrete frequencies and obtaining total vibration levels.

Total level: instruments that measure the total vibration level over a predetermined frequency range—20–10 kHz, 40–500 Hz, or some other specified range.

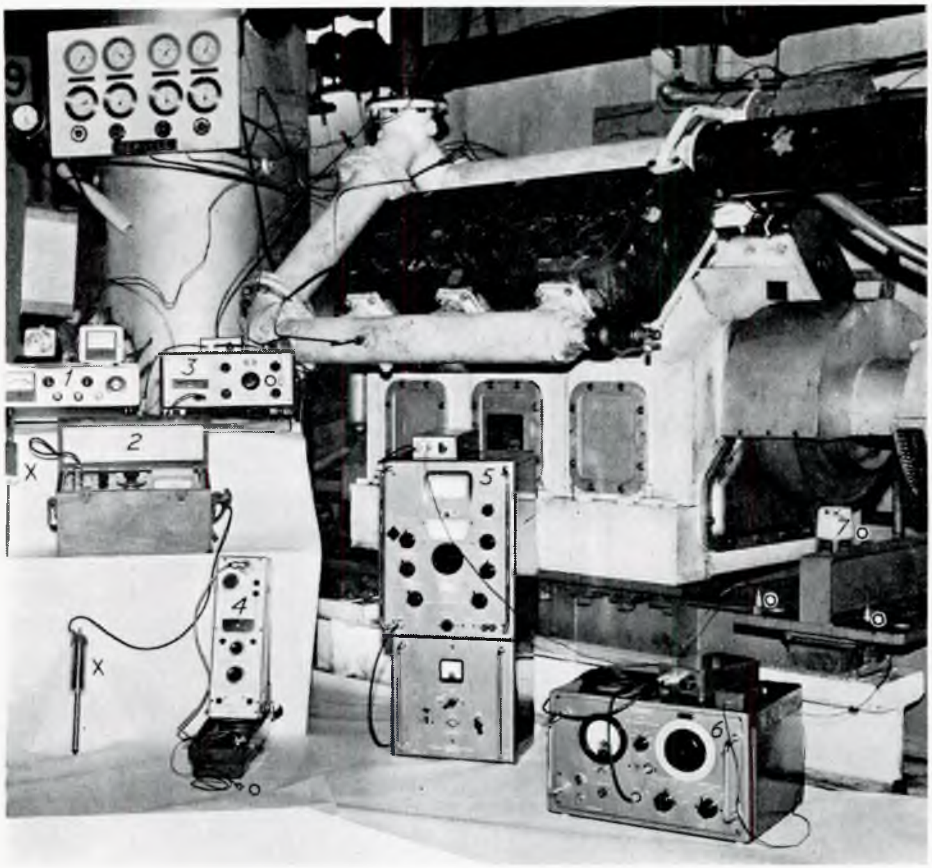
Velocity—Displacement

Portable analysers, together with velocity type transducer,

TABLE III—OCTAVE AND 1/3 OCTAVE PASS BAND AND MID-FREQUENCIES

OCTAVE	
Pass Band Hz	Mid-Frequency Hz
22·5	
	31·5
45	
	63
90	
	125
180	
	250
355	
	500
710	
	1000
1400	
	2000
2800	
	4000
5600	
	8000
11200	

1/3 OCTAVE	
Pas Band Hz	Mid-Frequency Hz
12	
	12·5
14	
	16
18	
	20
22·5	
	25
28	
	31·5
35·5	
	40
45	
	50
56	
	63
71	
	80
89	
	100
121	
	125
141	
	160
178	
	200
224	
	250
282	
	315
355	
	400
450	
	500
560	
	630
710	
	800
890	
	1000
1120	
	1250
1410	
	1600
1780	
	2000
2240	
	2500
2820	
	3150
3550	
	4000
4500	
	5000
5600	
	6300
7100	
	8000
8900	
	10000
11200	



1) and 2) Discrete frequency	×	5) $\frac{1}{3}$ Octave	O
3) Discrete frequency	O	6) Discrete low frequency	O
4) Octave	O	7) Total level indicator	O

× with velocity transducer
O with accelerometer

FIG. 2—*Vibration measuring instruments*

Fig. 3a, have been available for some years and have been used to great advantage as a machinery condition diagnostic tool. The transducer basically consists of a permanent magnet surrounded by a coil. The vibrations of a machine cause the magnet to move within the coil, thus converting the motions of the machine into an electrical signal. The signal produced is proportional to velocity; most instruments also indicate displacement, this is achieved by feeding the signal through an integrating amplifier. The usable frequency range of this type of system is in the region of 10–1000 Hz. The range is, however, considerably influenced by the method adopted to attach the transducer to the machine. A typical advertised usable range is as follows:

Method of attaching transducer	Usable range Hz
Hand held with straight probe	10–260
Hand held without probe	10–2500*
Secured by means of a stud or bolt etc.	10–2500*
Secured by vice type grips	10–120
Held in place by a magnet	10–620

* Upper limit may be high

The velocity transducer is in most cases used in conjunction with a discrete frequency analyser. An important fact to remember when considering the use of this combination is that regardless of the specified frequency range of the analyser, the limiting factor is the usable range of the transducer.

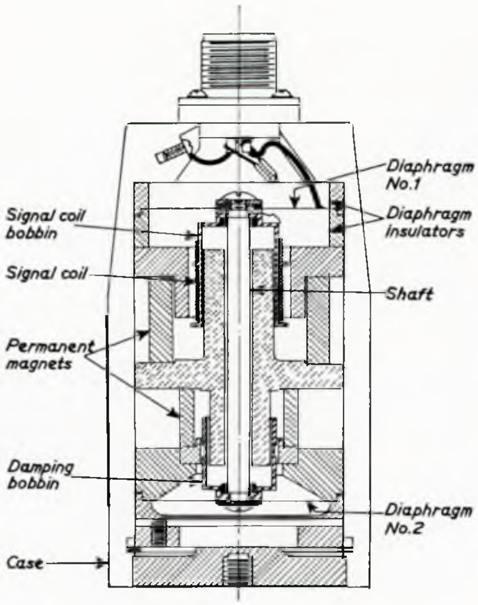
Acceleration

During recent years the piezoelectric type of accelerometer, Fig. 3b, has become the preferred type of vibration transducer. There are many different types of accelerometer available, each designed to meet some specific requirement. The type normally used in the field for machinery condition monitoring is termed "compression type". Basically these consist of a number of piezoelectric discs on which is placed a relatively heavy mass. The assembly is mounted onto a base preloaded by means of a stiff spring and sealed in a metal case.

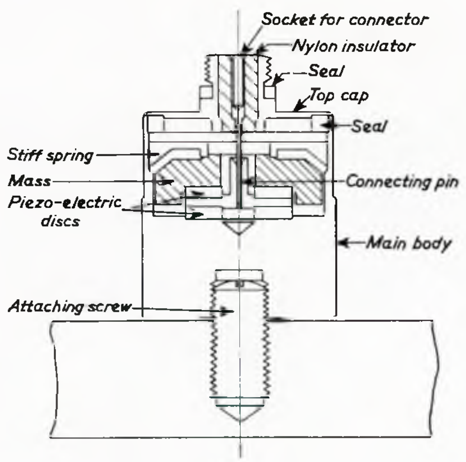
On being subjected to vibration, the mass exerts a force onto the piezoelectric discs which generate an electrical signal directly proportional to the force applied and therefore to the acceleration of the mass. The signal can also be displayed in velocity or displacement units by the use of integrating circuits which are built into some analysers.

The usable frequency range of many accelerometers is in the region of 10–40 kHz, although a 10–10 kHz range should be suitable for most machinery diagnostic work. As with the velocity type pick up, the range is considerably influenced by the method of attachment: the most efficient being attachment *via* a steel stud direct onto a rigid part of the machine.

Attachment by other methods such as special cement compounds, plasticine, magnet or hand held probe lowers the upper frequency limit to between 2k–5k Hz.

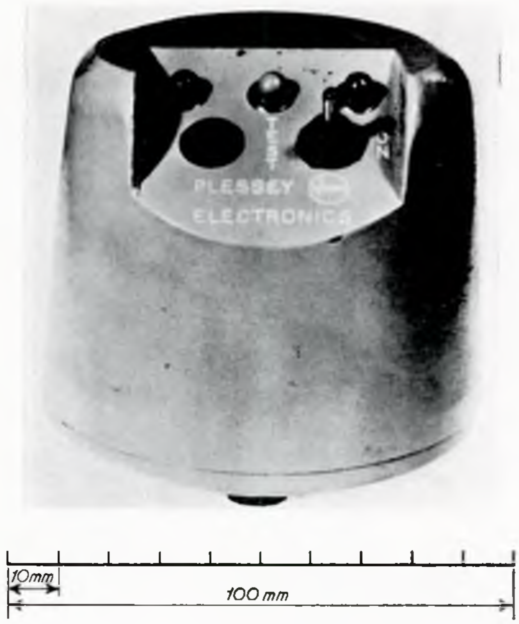


a)

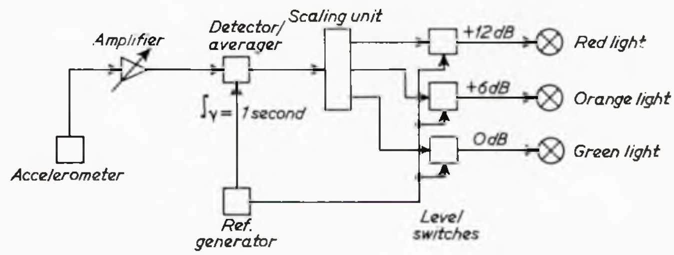


b)

FIG. 3—*a) Section through velocity transducer*
b) Section through accelerometer



a)



b)

FIG. 4—*a) Point machinery condition monitor*
b) Block schematic

TOTAL LEVEL INDICATORS

A recent innovation in this field is the introduction of point machinery condition monitors (PMCM's), Fig. 4: the generation of different levels of vibration resulting from mechanical deterioration being sensed by the PMCM *via* an accelerometer fitted to its base. The monitor sums all the vibrations in its frequency range and gives an indication of change in total level, by use of three coloured lights.

The total level of vibration of a machine under normal working conditions is termed the "base level". This level is equated to a green light condition on the PMCM. An increase in level of 6 AdB (g force doubled) causes both green and orange lamps to light. A further increase in level of 6 AdB (g force quadrupled) causes the green, orange and red lamps to light.

To establish the cause of increase in total vibration level, depending on the type of machine to which it was fitted, it would be necessary to carry out a full vibration survey after obtaining an orange or red lamp indication. Also the PMCM can only make valid comparisions of vibration if the machine operating conditions are broadly similar to those prevailing when the monitor is set up.

VIBRATION MEASUREMENT

Two basic methods of measurement have so far been adopted for monitoring machinery vibrations:

i) In three planes (vertical, horizontal and axial) in the vicinity of each principle bearing and possibly other important rotating components. Readings are taken using portable analysers, and the results compared against vibration severity charts, Fig. 5, which have been devised as a result of years of practical experience. This method is most suitable for assisting in the determination of the conditions of machinery for which no "as new" vibration information is available and for the diagnosis of suspected faults;

ii) At one or more positions on the base of a machine, above any resilient mounting arrangement that may be fitted. The principle adopted with this method is that measurement and acceptance of an "as new" vibration signature based upon the average or maximum levels obtained from one or more machines of a particular type taken over a short period after commissioning. From this information a vibration severity envelope can be produced for the machine, taking into account any special knowledge of its mechanical or performance characteristics. See Fig. 6. This system has the advantage of being very flexible in as much that the envelope can be modified without affecting the vibration severity criteria of other machinery in the programme. Also the frequency at which specific faults occur, e.g. out of balance, bearing failure, etc, can be predetermined and identified, thereby reducing the analyst's work load.

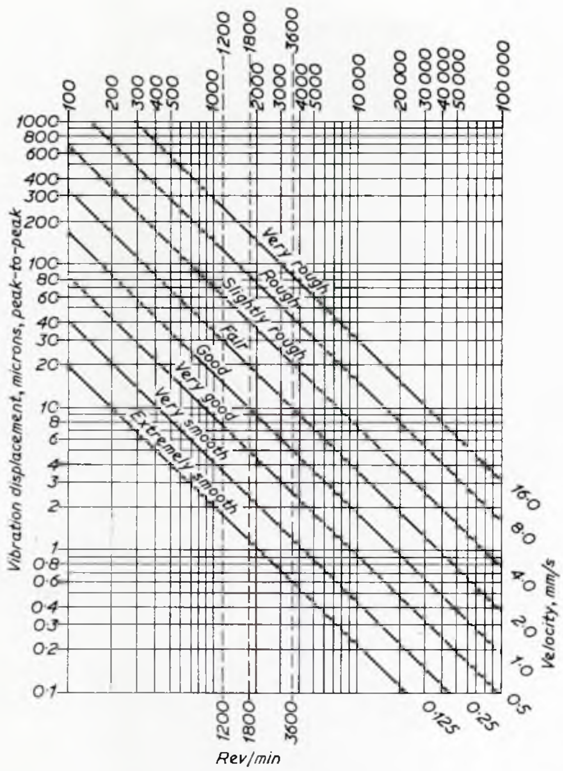


FIG. 5—*General machinery vibration severity chart*

REPEATABILITY OF VIBRATION MEASUREMENTS

For vibration measurements to be meaningful and of any value in the pursuit of machinery condition analysis, it is essential that they are taken with the machine or system, operating at steady conditions throughout the exercise. Similarly, if deterioration in mechanical state is to be identified by the recognition of change in the vibration spectrum, it is essential that subsequent measurements are taken with the equipment operating identical to, or as near as is practicable to, the original specified conditions.

This is no real problem with land based machinery, where conditions can be set to meet almost any predetermined requirement and maintained thus for as long as is necessary without being unduly influenced by external sources.

The situation with ship-borne equipment, is, however, somewhat different. Compressors and other motor driven auxiliary machinery can be set and maintained for reasonable periods at specified conditions. Other machinery, such as main feed pumps, forced draught blowers, extraction pumps, etc., are, however, considerably influenced by the particular operational requirements of the boiler at a given time. This problem can be minimized by maintaining as near as possible a set shaft speed whilst obtaining readings. The speed selected is preferably the normal ship cruising speed, thus enabling the work to be carried out with negligible interferences with day-to-day machinery discipline.

FAULT DIAGNOSIS

Major changes in vibration levels, will normally take place at frequencies associated with some mechanical or magnetic feature of the machine and its rotational speed. This being so, in general it is not too difficult for the trained analysist to interpret results and diagnose the possible cause of the change in spectrum. A list of important mechanical and electrical features which may cause vibration at various discrete frequencies is given in Table IV.

In the case of ball/roller bearings and gearing, the problem can be rather complicated. For example, the vibrational frequencies to be anticipated from ball/roller bearings are as follows:

Consider a machine bearing having:

Pitch circle radius	$= R$
Ball/roller radius	$= r$
Number of balls/rollers	$= n$
Speed of shaft	$= N$ rev/s

Then,

$$\underline{f1 = N\ Hz}$$

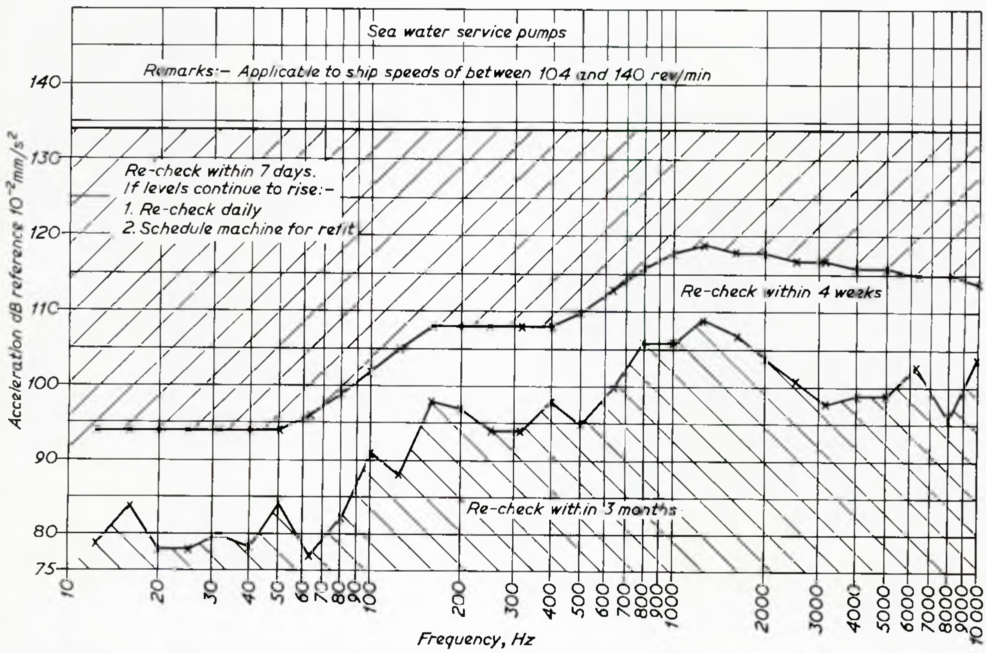


FIG. 6—*Machinery vibration severity envelope*

TABLE IV—DISCRETE FREQUENCY IDENTIFICATION

ALL MACHINES

Cause	Frequency
Oil Film Whirl	$\frac{1}{2}$ x Rotational Speed
a) Unbalance b) Eccentric Journals	1 x Rotational Speed
a) Misalignment b) Bent Shaft	1 x Rotational Speed Sometimes 2 x and 3 x
Defective Ball/Roller Bearings	Many Times Rotational Speed
Defective Gearing	Many Times Rotational Speed —gear teeth x gear revolutions/second
Bad Belt Drives	1 x, 2 x, 3 and 4 x Rotational Speed of complete belt
Reciprocating Forces	1 x, 2 x and higher x Rotational Speed
Aerodynamic and Hydrodynamic Forces	1 x Rotational Speed or number of blades on Fan or Impeller x Rotational Speed
Mechanical Looseness	2 x Rotational Speed

ELECTRICAL MACHINES

dc Machines a) Armature Slots b) Commutator Segments	No. of Slots x Rotational Speed No. of Segments x Rotational Speed
Synchronous Machines Magnetic Field	2 x Supply Frequency
Induction Motors a) Magnetic Field b) Rotor Slots	2 x Supply Frequency i) No. of Slots x Rotational Speed ii) No. of Slots x Rotational Speed $\pm$ 2 x Supply Frequency
All Unbalanced Magnetic Pull	1 x and 2 x Rotational Speed

$f1$ is the shaft rotational frequency and appears at the slightest unbalance. In a normal machine the vibrational level at this frequency is due to unbalance of the rotating section to which bearing unbalance is almost always a very minor contribution.

$$f2 = \frac{(R-r)}{2R}\ N\ Hz$$

$f2$ is due to the rotation of the rolling element train and indicates an irregularity (rough spot or indentation) of a rolling element or the cage.

The spin frequency of a rolling element is:

$$\frac{R+r}{r} \times f2\ Hz$$

and any irregularity of an element causes a vibrational frequency of:

$$f3 = 2\left(\frac{(R+r)}{r}\right) f2\ Hz$$

because the irregularity strikes the inner and outer races alternatively.

$$f4 = (f1-f2)\ n\ Hz$$

$f4$ is due to an irregularity on the inner raceway.

$$f5 = f2\ n\ Hz$$

$f5$ is due to an irregularity on the outer raceway or a variation in stiffness around the bearing housing.

$f1-f5$ are the fundamental frequencies due to the various causes and often these are accompanied by harmonics. In the case of irregularities, the more irregularities the more harmonics that are produced.

In the particular case of the upper bearing in a motor driven centrifugal pump, the bearing details were:
(*See Case History No.* 1)

Pitch circle radius	=	31·75 mm
Ball radius	=	6·35 mm
Number of balls	=	13
Speed of shaft	=	$\frac{3500}{60}$ rev/s

Thus:

$$f1 = \frac{3500}{60} = 58{\cdot}3\ Hz$$

$$f2 = \frac{(31{\cdot}75-6{\cdot}35)}{2\times 31{\cdot}75} \times \frac{3500}{60} = 23{\cdot}3\ Hz$$

$$f3 = 2\,\frac{(31{\cdot}75+6{\cdot}35)}{6{\cdot}35} \times 23{\cdot}3 = 280\ Hz$$

$$f4 = (58{\cdot}3-23{\cdot}3)\times 13 = 455\ Hz$$

$$f5 = 23{\cdot}3\times 13 = 303\ Hz$$

Experience has also shown that loss of lubricant from a bearing causes high vibrational amplitudes in the 2000–10 000 Hz frequency region.

CASE HISTORIES

(Each example given concerns an operational marine equipment)

Case 1

Type of machine—Centrifugal pumping unit driven by a 6·7 kW electric motor.

Vibration measurement equipment—Installed semi-automatic 1/3 octave analyser with x–y plot, print-out and using 50 mv/g compression type accelerometer.

Method of measurement—Measurements of vibration amplitude AdBs obtained at one position only on the base of the machine: the initial "as new" readings being accepted as the datum. Measurements were taken at 2–3 monthly intervals.

History

After a period of running under normal operational conditions the vibration signature of the machine was observed to have changed significantly. (See Fig. 7—a and b.) The motor was uncoupled, run and a further signature obtained: Fig. 7—c. In view of the reduction in dominant vibration levels it was concluded that the cause of the vibrational increase was in the pump end of the unit. Analysis of the dominant peaks in curve Fig. 7—a indicated that the probable source of the pump vibrational frequency in the region of 450 Hz was an irregularity of the inner raceway of the pump upper bearing: the peak in the 60 Hz region indicating an increase in out of balance forces. Subsequent dismantling of the pump and examination of the bearing confirmed the diagnosis. The inner track having flaked away over approximately 1/3 of the ball path and the wear debris had indented both the inner and outer tracks. (See Fig. 8.)

Case 2

Type of machine—7 kW electric motor driving a lubricating oil transfer pump.

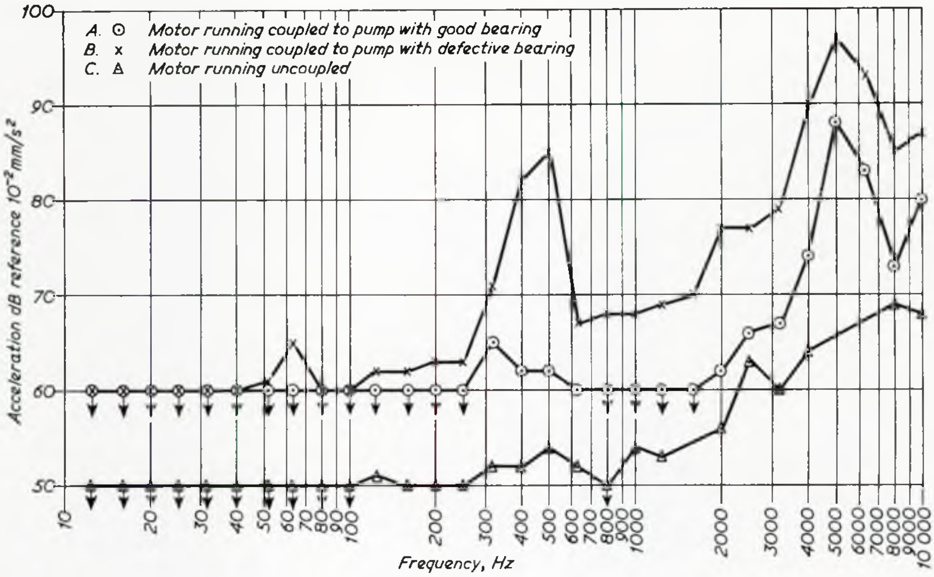


FIG. 7—*Detection of a defective bearing in a pumping unit*

Vibration measurement equipment—Installed semi-automatic 1/3 octave analysis with x–y plot, print-put and using 50 mv/g compression type accelerometer.

Method of measurement—Measurement of vibration amplitude AdB's obtained at one position only on the base of the machine: "as new" readings being used as a datum. Measurements were taken at regular intervals.

History

After monitoring the machines vibration signature over a period of 12 months, a significant increase in level was noted over the 2k–10k Hz frequency range. See Fig. 9—a and b. As the pump end of the unit was submerged in lubricating oil it was deduced that the cause of the vibrational increase was in the motor. Past experience indicated that as the increase was in the higher frequency range, the most probable cause was associated with lubrication of the motor ball bearings. Over a further period of nine months running the vibration levels recorded showed an increase over the whole spectrum. See Fig. 9—c. The pump was still operating satisfactorily and as no other indication of failure was apparent, it was decided to continue running. After a further five months a considerable increase in levels was recorded over the spectrum Fig. 9—d, and it was concluded that one or both of the motor end bearings was nearing failure. Subsequent investigation showed that the motor drive end bearing had suffered a loss of lubricant and that considerable damage to the inner and outer raceways and balls had occurred, Fig. 10. Severe fretting corrosion had also occurred on the outer diameter of the bearing and its locating face in the bearing housing.

Case 3

Type of machine—400 kW Turbo-alternator.

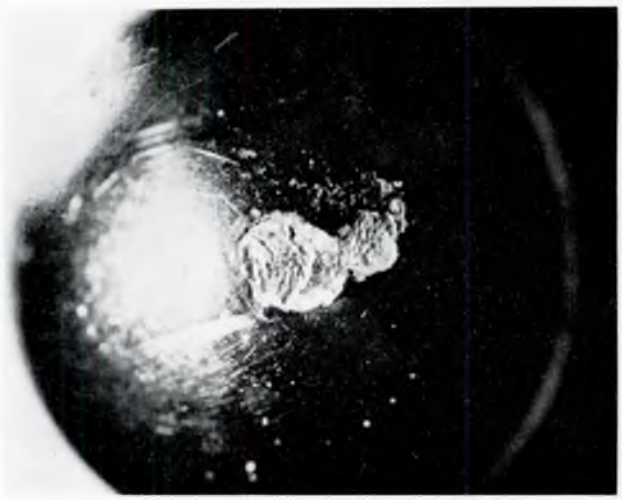
Vibration measurement equipment—Portable discrete frequency analyser with velocity transducer.

Method of measurement—Vibration amplitude displacement (mm) or velocity (mm/s) measured in three planes in the vicinity of each bearing (machine sketch Table V refers to this).

History

Subsequent to heavy priming of a main boiler, the boiler

a) Damaged area on track of internal face



b) Pitted ball bearings

FIG. 8—*Defective pump end thrust bearing*

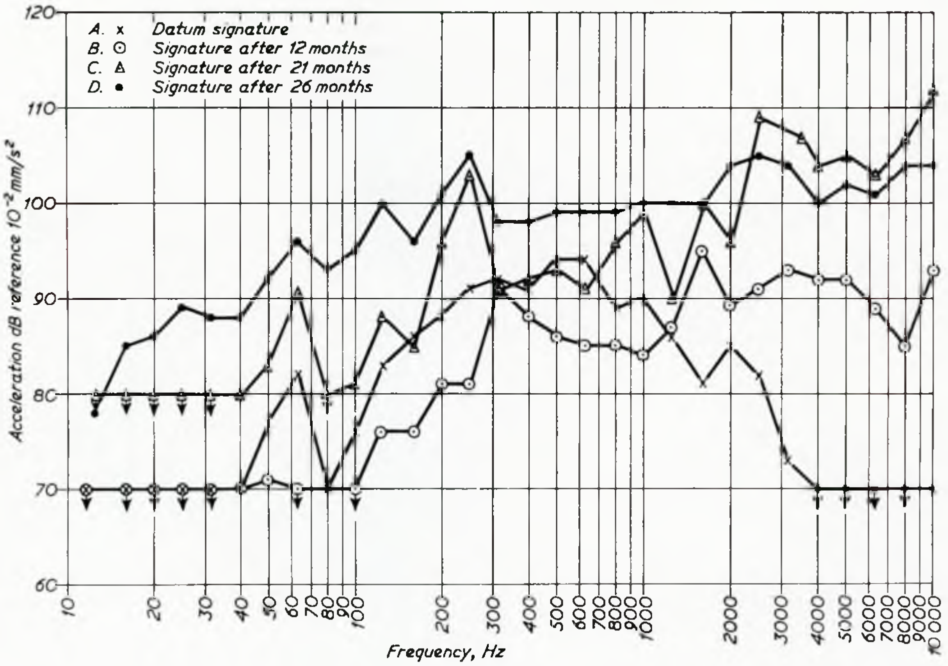


FIG. 9—*Detection of a defective bearing in a 7 kW electric motor*

room turbo-alternator axial clearances were checked. Readings indicated that the motor had moved 0·076 mm towards the steam end. After a further 21 h running the turbine shaft overheated in the vicinity of the exhaust end bearing. Axial clearances were again checked and indicated that the turbine shaft had moved 0·73 mm towards the exhaust end. New thrust pads were obtained and fitted and the T–A run up. At 12 000 rev/min there was slight noise and vibration evident, which became more pronounced as the turbine reached its normal running speed of 15 000 rev/min. The source of vibration was difficult to trace and after an unsuccessful conventional examination of the machines components, the assistance of a vibration analyst was requested.

The results of the analysis indicated that the fault lay on the turbine shaft and rotor. The worst readings were obtained at the exhaust end bearing—Table V. Further indications were a bent or misaligned shaft. The analyst recommended that the turbine rotor be lifted at the earliest convenient opportunity, and

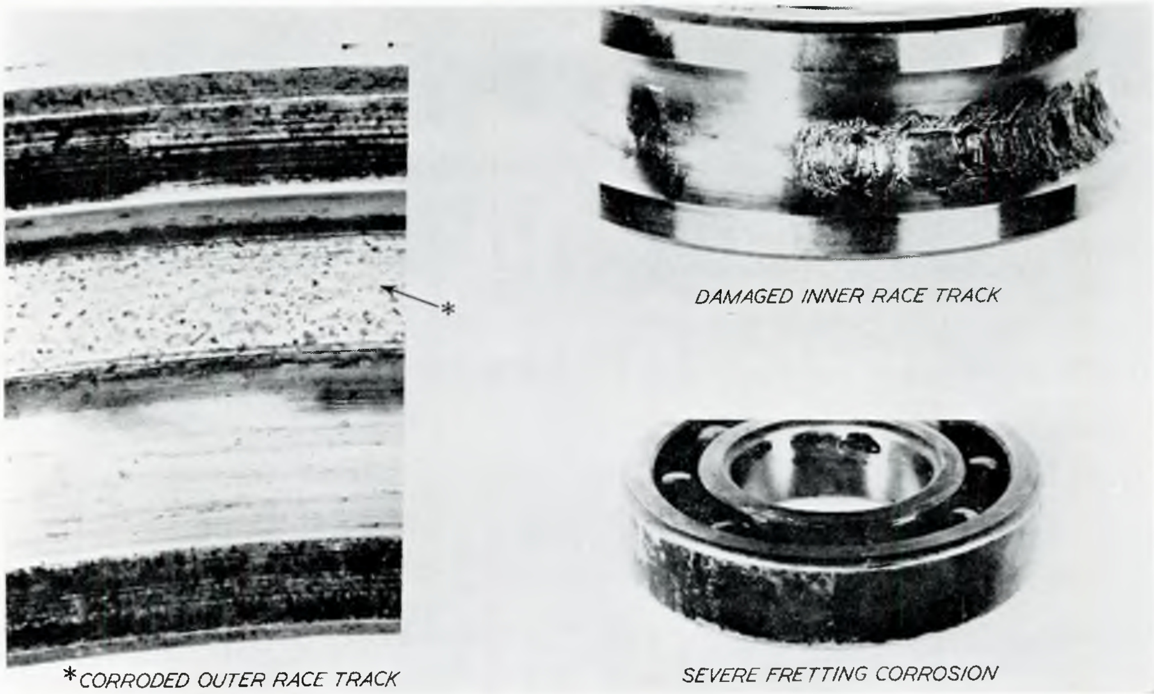


FIG. 10—*Defective motor drive end bearing*

TABLE V—DIAGNOSTIC VIBRATION SHEET

Pick Up Position		Filter Out		Filter In															
		disp.	vel.	vel.	freq.	vel.	freq.	vel.	freq.	vel.	freq.	vel.	freq.	vel.	freq.	vel.	freq.	vel.	freq.
A	H	·08	5	1	30					1	250	4	500			0·7	900		
	V	·03	5	2·5	30					1·7	,,	2·5	,,			1	910		
	A	·07	7	0·5	30	1	46			1·5	,,	1	,,			0·4	900	4	330
B	H	·05	15	1	30					6	,,	4	,,			3·5	900	5	560
	V	·02	12	1	30					4	,,	6	,,			1	,,		
	A	·02	12	0·7	30	1	46	1·5	91	4	,,			2·5	660	1·5	,,	8	400
C	H	·01	7	0·7	30					1	,,	2·5	,,	5	600	1	,,		
	V	·01	5	2	30					1	,,	2·5	,,			0·6	,,		
	A	·03	11	1·5	30					4	,,	3·5	,,	5	560	0·8	,,		
D	H	·02	10	0·6	30			1·7	83	0·7	,,	4	,,			0·8	910		
	V		5	1·5	30					0·7	,,	3	560			1	900		
	A		6	1	30			1	91	2·5	,,			2·5	800				
E	H	·02	3	—	—			1	83	0·4	,,	2·5	580			0·5	900		
	V	·02	5	1·5	30					0·6	,,	2·5	500			0·5	900		
	A	·02	6	1	30					1·7	,,			2·5	660	1·5	910		
F	H	·02	4	1·5	30					0·4	,,	1·7	530					1	330
	V	·03	5	0·5	30					0·5	,,	0·7	500						
	A	·02	4	1	30						—					0·2	900	0·5	400
G	H	·01	10	0·5	23	0·7	30				—	5	500			1·5	,,	2	350
	V	·02	12	1	30	1	48			1·7	,,	4	500	6	660	1·4	,,		
	A	·02	6	1	30						—			2·5	610	2	,,	1·7	330
H	H	·02	6	0·7	23	0·5	33			3	,,	1	510			0·8	910		
	V	·03	7	1	23					4	,,	1·7	510			0·4	900	0·6	160
	A	·02	11	0·2	23			1·7	83	6	,,	2·5	510			0·3	900		

Displacement — mm
H = Horizontal

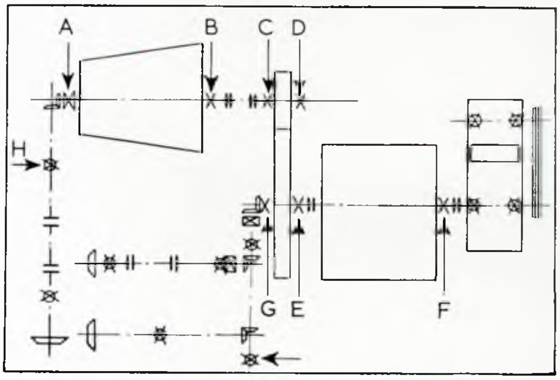
Velocity — mm/s
V = Vertical

Frequency — Hz
A = Axial

H.M.S.	
Machine Location and Type TA	Boiler RM
Serial No	
Schedule No	
Date Installed	
Operator	

Symbol	Identifies
	Pickup point
	Plain BRG
	Anti-friction BRG
	Thrust BRG
	Coupling

Date	1.10.70
Hours run	
Main eng shaft rev/min	80
Sub-assembly rev/min	1800



checked for truth, and that alignment of the turbine bearing housing be checked. It was also stated that accelerated bearing wear was to be expected until the defect was rectified.

The condition of the turbine on opening up was as follows:

a) Turbine Rotor: Several sections of shrouding on the first row of moving blades missing;
b) Hammering marks on the leading and trailing edges of the first and second stages of the curtis wheel;
c) Scoring in way of gland and diaphragm labyrinths;
d) Guide Blades: Leading and trailing edges of blades damaged with wear on the side facing towards the steam end;
e) Steam End Bearing / Exhaust End Bearing } Some wiping of white metal.

In this particular case vibration analysis identified the problem area and assisted maintenance staff in the provision of spares and the planning of the repair operation. It is considered, however, that had vibration measurements been taken sooner the expense of having a new rotor fitted to the turbo-alternator and disruption of the ships programme may well had been avoided.

CONCLUSIONS

Early detection and diagnosis of mechanical deterioration in rotating machinery can be achieved by the measurement and comparison of changes in a machines vibration signature. Considerable laboratory and field effort is still required if the full potential of the technique is to be realized. A suitable data bank will aid easy and speedy diagnosis of faults, as well as provide the designer with critical component data, to enable him to improve reliability of equipment and increase mean time between failure.

ACKNOWLEDGEMENTS

The author is grateful to the Director General Ships and the Superintendent Admiralty Marine Engineering Establishment for permission to publish this paper, and to many colleagues who have contributed to or assisted in its production.

Discussion

Mr. R. M. Stewart had found Mr. Carmody's film very interesting. Some work was being carried out, at the Institute of Standard Vibration, on vibration diagnosis at the moment and a great deal of the work was concerned with ships and power stations.

He had one or two points to make on the film. He was interested to see the equipment being used by the people concerned as it looked of quite a sophisticated type. In a great many situations it was found that one could not use that type of equipment because it was so complicated. What were Mr. Carmody's views on this?

His second question concerned analysis. If one had gearboxes with a number of meshes and gears of varying types, it was often quite difficult to distinguish between the different meshes with simple 1/3 octave analysis, and often one had to resort to much narrower band analysis, something like 20 Hz or so. That often made it possible to interpret gearbox signatures without previous knowledge of the gearbox, by simply comparing different orders of shaft levels.

His third question was concerned with the amount of advance warning one needed on gearbox failure. It had been found at Southampton that just measuring sheer overall engine vibration often did not give very advanced warning of failure and, often, on a bearing for example, one had to try to measure the shock noise. That often gave a much better indication of its condition. Even if one was measuring the overall levels of vibration, the vibration that one measured was often dominated by the machine structure and not by the bearing that one was trying to investigate. It had been found in maintaining aero-engine type down piezo-locking sets that one must often have advance warnings of the order of 100 hours if one was properly to schedule the maintenance.

Mr. W. Strassheim said that one area of vibration measurement where his company had found difficulty in obtaining measuring equipment was particularly in the lower frequencies of 5–10 Hz. Most electronic equipment was well advanced and capable of providing overall analyses in excess of 10 Hz, but in many ships it was being found, with high power, medium speed Diesels particularly, that there was severe vibration which was more difficult to measure with hand instruments. An example was the Askania vibrograph, which did not lend itself to the higher frequency B and K parameters. He would like the author's comments on the forthcoming availability, if any, of equipment to measure low frequency, and in particular propeller blade order vibration.

Mr. L. F. Moore, M.I.Mar.E., said that Mr. Carmody had stated in the paper that it had been internationally agreed to use a logarithmic representation of acceleration or velocity measurement. He agreed that a logarithmic representation should be used, but to date he had not seen an international, or even a national, standard quoting that. In B.S. 4675:1971 "Recommendations for a basis for comparative evaluation of vibration in machinery", vibration ranges were quoted in mm/s R.M.S. Those could be converted into 4dB ranges, but no reference level was quoted. He realized that 10^{-5} mm/s R.M.S., as used by Mr. Carmody, was a reference level which had been used by the Admiralty for a number of years, but he would like to see in the future 10^{-6} mm/s R.M.S. adopted as a velocity reference level and 10^{-3} mm/s^2 R.M.S. as an acceleration reference level as these were already recommended sub-multiples for SI units.

Reference was made to discrete frequency analysers. What were the author's comments on the relative merits of fixed frequency band and percentage frequency band analysers?

Again, referring to B.S.4675, that covered only a frequency range of 10 Hz to 1000 Hz. Did Mr. Carmody think that the upper end of that frequency range was adequate, especially as he stated that a 10 Hz to 10 kHz range should be suitable for most machinery diagnostic work?

In Fig. 6 a conventional logarithmic frequency scale was used. After inspection, it was apparent that a 1/3 octave analysis had been used. Was this type of analysis adequate for routine checking? He also noticed in the film that use was made of octave analysis. If the author was of the opinion that that type of analysis was adequate for routine checking, would it be easier if the results were plotted with the frequency axis divided into 1/3 octave bands?

As stated in the paper, a discrete frequency analyser was used to measure the values presented in Table V, which, incidentally, were not in dB. What bandwidth was used and why was that preferred to a 1/3 octave analysis?

With reference to Mr. Strassheim's comment about analysis down to low frequencies, he understood that B and K would modify its standard analysers to go down to 1 Hz, thus something was at the high frequency end. His organization had just been discussing this with them.

Mr. B. K. Batten, M.Sc., M.I.Mar.E., said that he had been particularly interested in the comments on total level indicators. Had the author any experience of those being used in practice, and what psychological aspects were there on personnel looking after machinery?

Obviously, there was a problem in that one had to determine the initial signature of the equipment that was being monitored before setting a figure on the vibration limits. He would imagine that in service these limits would be subject to changes of background vibration. He wondered whether the installation of such indicators on a ship might lead to machinery being stripped down because of an increase in vibration caused by some other source.

whereas the machine in question could have run perfectly well for a little longer.

With regard to bearings, he noticed that in his case histories Mr. Carmody described defects in roller bearings. He wondered whether the cost of monitoring vibration levels, so as to arrive at a point where one decided to open up and examine and change a bearing, was justified, as against establishing a general running time for bearings and changing them at the end of that time.

Professor W. Carnegie, Associate Member of Council, I.Mar.E., asked Mr. Carmody whether he could give some brief account of how he would deal with rapid failures, for those were often the ones that occurred and gave the most catastrophic failures. One might cite, for example, the classical failure that occurred in *Queen Elizabeth II.* That was some thing that gave very little warning and then happened with great rapidity.

Secondly, he drew the attention of the author—perhaps he was not aware of it—to the fact that quite a lot of work had been done in that field over many years. One might particularly mention the work of one of the Chief Engineers at Pametrada; namely, Mr. H. E. Yates, who had done some very valuable work in gear vibration.

Dr. F. Ørbeck, M.I.Mar.E., said that it was Mr. Strassheim who had prompted him to say a few words.

The analyses, which the meeting had been shown, were primarily confined to high frequency work. On the other hand, the traditional vibration measurements in the industry were on the low frequency side—shaft vibration, stern vibration and so on—and, although it was not perhaps so clearly to the point, these vibration measurements were also a means of preventing failures.

With regard to the instrumentation for low frequency vibrations, his company could give a little advice on at least tortional vibration and axial vibration. They used an electrical version of the Geiger torsionograph which they had developed themselves.

When it came to structural vibration, one had found that the best transducer spring mass system was a hand held vibrograph.

Correspondence

Mr. A. R. Hinson, A.M.I.Mar.E., wrote to ask the author if he had considered using proximity pick-ups to sense shaft eccentricity on defective rotating machines.

It was often possible on steam turbine propulsion machinery to place a proximity pick-up close to the shaft some distance from a bearing and near the point at which an anti-node occurred when the turbine rotor and shaft rotated laterally.

This method of vibration detection was probably more sensitive than using an accelerometer attached to the turbine bearing. If the support bearings were stiff, the vibratory forces must be relatively large in order to generate sufficient amplitude at the bearing. On the other hand the amplitude at an anti-node was naturally greater.

The disadvantage of measuring amplitude, or comparing amplitudes, was that it was often difficult to know whether an unusually large amplitude mattered. For example, was 2 mm transverse vibration acceptable on a Diesel engine when measured at a point, say, 10 mm above the bedplate?

Conventionally, machines were designed on a stress basis, and unless previous experience was available it was not always easy to relate amplitude to stress and hence say what was or was not acceptable.

The strain gauge bridge was more useful in this respect than devices which measured amplitude, since the vibratory stresses at the point where the gauges were applied might be accurately determined. Vibration stresses increased when many of the faults listed by the author in Table I occurred and it would be interesting to have his views on the use of strain gauges.

Mr. Hinson thought the paper was important, since it indicated methods which could be used to help make up for the absence of the watchkeeper in unattended machinery spaces.

Mr. F. A. Manning, B.Sc., A.M.I.Mar.E., said, in a written contribution that the author had shown quite clearly and accurately that a wide range of information could be gained by the simple analysis of vibration measurements described in the paper and, by these means, prevention of serious damage to machinery could often be prevented. The writer was recently concerned with an engine failure, where the engine was the gas generator of a well known aircraft type jet engine adapted for a peak load gas turbine generator. In common with plant described by the author, a basic set of vibration measurements over the speed range were taken during the factory proving test and supplied to the operator as reference data (the measurements consisted of peak to peak R.M.S. values of vibration amplitude taken on the outside of the compressor casing near the air inlet with a single transducer). During testing at site, the inlet flare (made of fibreglass) failed without warning and injestion of a broken piece severely damaged the compressor blading. Up to the instant of injestion, vibration levels were normal, followed by an immediate rise as the engine failed.

This pattern of failure was typical of steam turbine failures from bent spindles and blading failures, where no time was available to take corrective action.

The point being made, was that serious failures due to resonance often did not provide external unbalanced forces that could be detected in advance of failure by vibration measurements. Had the author considered the use of noise measurement as an alternative means of monitoring plant performance?

In common with the comments made by other speakers, the writer also looked forward to the day when failures in design could be prevented from producing a disaster by appropriate warnings from similar measurements such as those described in this paper.

Would the author care to consider what system of measurements he would recommend to advance from the present system of checking plant deterioration from wear, to design checking, say, of a geared marine steam turbine?

Author's Reply

Replying to the points raised by Mr. Stewart, Mr. Carmody explained that it had been agreed that in many cases, due to accessibility problems, it was difficult to use some of the more sophisticated equipment on board ship. At present, MOD(N) were using relatively simple portable discrete analysers or built-in 1/3 octave instruments which required only the accelerometer and load etc to be moved from machine to machine.

The major problem at present was operator fatigue, especially in the case of portable instruments using hand held probes. In many cases it took three to four hours to obtain required data. The use of a portable tape recorder and a firmly attached accelerometer was considered the most appropriate method, which enabled the data to be obtained quickly and analysis to be carried out in comfort.

The analysis of gearbox vibrations was a difficult subject. However, if the analysist was provided with gear ratios etc, it was possible to detect pending mechanical failure using 1/3 octave equipment. The use of narrow band instruments was preferable, but in many cases it was sufficient to know there was something wrong rather than know exactly what it was. The gearbox problem was frequently glossed over in diagnostic sheets by the use of the term "Many Times Rotational Speed".

In regard to Mr. Stewart's third question, experience had indicated that bearings seldom failed instantaneously. Roller bearings would run for many hours with little or no lubrication before actual failure. It was agreed that measuring overall engine vibration would not, in all cases, have been sufficient; each case must be dealt with on its own merits, taking into account the cost of installation, damage to life and other associated factors.

In reply to Mr. Strasshein's question, instruments measuring down to 3 Hz, suitable for measuring blade order frequencies, were available. The author had no advance information on new instruments. Regarding MOD(N) use, instruments measuring down to 10 Hz had been adequate to meet condition monitoring requirements. However, instruments measuring down to 3 Hz were available to meet special requirements.

In regard to the points raised by Mr. Moore, Mr. Carmody explained that the information had been received second hand, however it was understood that the ISO had recently agreed in principle that vibrations should be measured in acceleration or velocity decibels.

Regarding the reference levels 10^{-2} mm/s^2 and 10^{-5} mm/s, these were direct conversions from the 10^{-3} cm/s^2 and 10^{-6} cm/s reference levels which had been in common use for some years. It was assumed that in time internationally agreed references in SI units would be introduced.

B.S. 4675 basically covered the comparison of vibration of similar machines at the production stage and was not compatible with the requirements for machinery conditions monitoring. Regarding the use of the differing units of measurement, vibration measured in "Banana Units." would have sufficed, provided everyone used them. The problem at present was the use of differing units of measurement, reference levels and in some cases frequency bands.

With reference to the question on Fig. 6, a 1/3 octave system was considered adequate for routine checking as demonstrated by two of the case histories. It would assist if results were plotted with the frequency axis divided into 1/3 octave bands.

Regarding the discrete frequency analyser and Table V, the band width of the instrument used was about ± 5 per cent. The discrete analyser was used because it was portable and not because it was preferable.

In reply to Mr. Batten, a limited number of total level indicators had been fitted to machinery on selected ships and at shore establishments. As a result of the success achieved, a full scale trial had been put into operation. The use of indicators, together with a discrete analyser used at present for diagnostic work, was gaining the confidence of engine room staff. They quickly realized that if used sensibly, this simple vibration check could in many cases give advance warning of failure, or indicate the need for unscheduled maintenance i.e. vee belt adjustment etc.

The cardinal rule with the use of the indicator was not to assume that failure was imminent on first obtaining an orange or red light burn. Operating conditions, mechanical security and other normal checks should be carried out prior to initiating drastic action. A case in point was a recent experience on a ship fitted with indicators. A red light burn had been obtained, and the discrete analyser had confirmed the increase in vibration level so the machine had been replaced. Vibration checks at a shore base had proved that the unit was in a serviceable condition. On further investigation it had been discovered that on removal from the baseplate, the securing bolts had been found to be loose. This was obviously the cause of the increase in vibration.

In reply to the point raised by Professor Carnegie, Mr. Carmody explained that possibly the system which would serve best would consist of an auto shut-down device and associated vibration monitoring equipment incorporating tracking filters. Transducers would be placed in the region of critical areas specified by the designer who would also be required to state maximum permissible vibration levels. The author realized that valuable work had been carried out in this field; unfortunately the majority of it had only been published on a very limited scale. Also much of the scientific work had borne little relation to the practical application of the technique.

In reply to the contribution by Mr. Manning on the use of noise measurement, the author said that this had been considered, but it was thought to be unsuitable for marine application in view of the neccessity to measure individual machine noise in isolation.

The basis of non-destructive testing, of which vibration analysis was one technique, was to eliminate preventative dismantling. Most techniques were directly associated with the measurement and correlation of wear, in one form or another, and general plant deterioration. It was thought that this situation would exist for many years.

In regard to Mr. Hinson's contribution, the author replied that the relative merits of accelerometers, velocity transducers and proximity pick-ups had been considered. Bearing in mind the varying size, intricacy, accessibility and range of machinery required to be monitored, the accelerometer was considered most suitable. The cost of installing and setting up proximity pick-ups could also be prohibitive. For the purpose of monitoring one large machine, i.e. a steam generating unit, the proximity pick-up might be superior; however, he had had little experience in their use.

It must be emphasized that the use of vibration monitoring as a diagnostic tool was based on the single measurement of change in level and the eventual correlation of degree of change and acceptable mechanical deterioration.

AN EXAMINATION OF THE EFFECTS OF VARIABLE INERTIA ON THE TORSIONAL VIBRATIONS OF MARINE ENGINE SYSTEMS

W. Carnegie, Ph.D., B.Sc., C.Eng., Associate Member of Council, I.Mar.E.,* and M. S. Pasricha, M.Sc., B.Sc.†

Prof. Carnegie

Mr. Pasricha

The analysis of torsional vibrations in the running gear of reciprocating engine systems is normally carried out by neglecting the variation in inertia torques of the system arising from the motion of the reciprocating parts. When the variable inertia effect is allowed for the equation of motion taking into account the effect is non-linear. Assuming small displacements, the equation can be linearized to predict important characteristics of the motion.

Such an equation when solved by numerical methods using a digital computer predicts the regions of instability and the manner in which the amplitude and frequency vary with the speed of rotation of the engine. The responses of the system show a modulation of amplitude and frequency at definite rotational speeds. The occurrence of such a modulation in amplitude and frequency is established by use of the process given by Wentzel, Kramers, Brillouin and Jeffreys generally known as the WKBJ approximation.

The first order term in the equation and the forcing term which represents the outer impulse from the reciprocating parts, are investigated for their effect on the waveforms of the responses of the system.

Further investigations of the effect of lower order external excitations on the characteristics of the motion are given. Theoretical results are compared with solutions of the equation obtained from an analogue computer.

A discussion of some actual cases from engines in service is included.

INTRODUCTION

In recent years several cases of marine crankshaft failures are attributed to the phenomenon of secondary resonance, that is to say, the possibility of an nth order critical of small equilibrium amplitude occurring at or near resonance with the service speed being excited by large resultant engine excitations of order $(n-2)$ and $(n+2)$. Draminsky[1] in his analysis has stated that in practice secondary resonance in torsional vibration occurs only for resonance with the lower-order secondary component, $(n-2)$, and not the higher order component, $(n+2)$. The occurrence of this phenomenon of secondary resonance in multi-cylinder engines with large second order variation in inertia is explained by Draminsky[2] by the use of a non-linear theory. Archer[3] has cited typical examples of crankshaft failures in large ten-cylinder and twelve-cylinder engines from service in which measured stress values of certain orders were found to be much greater than those calculated by normal methods.

Recently a ten-cylinder, two stroke cycle engine was examined with suspected secondary resonance. It had a 2-node, ninth order vibration nearly in resonance with the service speed and the geometric resultant of the ten torque impulses of the ninth order was very small. The seventh order critical was found to have a large equilibrium amplitude which indicated the possibility of large ninth order torsional vibrations being excited. The 2-node mode ninth order measured stress obtained by harmonic analysis of stress records taken on the intermediate shaft and the corresponding theoretical stress calculated from damped-forced vibration tabulation are given in Fig. 1. The measured stress at resonance is only slightly higher than the calculated value.

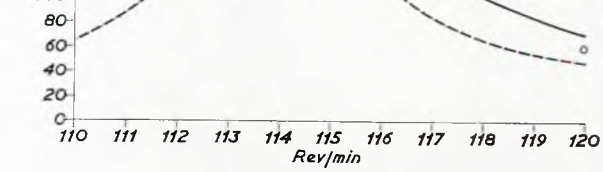


FIG. 1—*Stress in intermediate shaft plotted against rev/min for the 2-node ninth order torsional vibration*

——————— *experimental*

- - - - - - - - - - *theoretical*

The 2-node mode ninth order calculated stress in the crankshaft at resonance is ± 640 lb/in². The corresponding stress worked out from a Draminsky calculation based on non-linear

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theory would be ± 1420 lb/in². The conventional flank stress calculations carried out for 5th, 7th and 9th harmonic orders at service speed gave the following results:

Order	*Stress* ($\pm lb/in^2$)
II/9th	90
II/7th	600
II/5th	765
Arithmetical sum =	1455

The Draminsky ninth order calculated stress agrees closely in value with the arithmetical sum of the flank stresses as pointed out by Archer(3). It is obvious from Fig. 1 that in this case stress magnification due to secondary resonance fails to appear.

Draminsky in his analysis has omitted the variable part in the coefficient of the elastic term which appears in the equation derived from energy considerations. From the above observations it was concluded that further theoretical work on the basic equation representing the linear motion for a single cylinder engine is necessary.

NOTATION

A_{max}	maximum amplitude of γ
a	crank radius
I	moment of inertia of the rotating parts
I_m	$I + \frac{1}{2}Ma^2$
M	mass of reciprocating parts
r	ratio of the angular velocity ω of crankshaft to the natural frequency ω_n of the system neglecting variable inertia effects
ε	$\dfrac{\frac{1}{2}Ma^2}{I + \frac{1}{2}Ma^2}$
μ	torsional stiffness of crankshaft
γ	displacement of torsional motion
ω	angular velocity of crankshaft
ω_n	natural frequency of the system neglecting variable inertia effects $= \left[\dfrac{\mu}{I + \frac{1}{2}Ma^2}\right]^{\frac{1}{2}}$
ω_I	frequency of response at maximum amplitude

THEORETICAL CONSIDERATIONS

With the symbols defined in the notation, the basic equation which takes into account the second order variation in inertia and governs the vibratory motion in a single cylinder engine, the gas pressure in the cylinder being omitted is

$$I_m(1 - \varepsilon\cos 2\omega t)\frac{d^2\gamma}{dt^2} + 2I_m\omega\varepsilon\sin 2\omega t\frac{d\gamma}{dt} + \left(\frac{\omega^2}{r^2} + 2\omega^2\varepsilon\cos 2\omega t\right)I_m\gamma = -\tfrac{1}{2}Ma^2\omega^2\sin 2\omega t \qquad (1)$$

Equation (1) is further simplified by dividing it throughout by $\omega^2 I_m$ and changing the independent variable to $\tau = \omega t$,

$$(1 - \varepsilon\cos 2\tau)\gamma'' + 2\varepsilon\sin 2\tau\,\gamma' + \left(\frac{1}{r^2} + 2\varepsilon\cos 2\tau\right)\gamma = -\varepsilon\sin 2\tau \qquad (2)$$

where dashes represent differentiation with respect to τ.

Derivation of the theory and solutions of the equation of motion (2) at definite rotational speeds and regions of instability determined by use of numerical methods are given in a preliminary paper(4). In view of the practical importance of the subject and the experimental observations it was considered necessary to examine further the effects on the solutions of some terms in the equation and it was desirable also to confirm the numerical analysis by alternative methods. Some interesting results on the secondary resonance phenomenon and the results which follow the above analysis are given in the present paper.

SECONDARY RESONANCE

Figs 2 and 3 illustrate the waveform solutions of equation (2) at $r = 0{\cdot}1$ and $r = 0{\cdot}08$ when $\varepsilon = 0{\cdot}3$. The reference (4) gives the numerical method used for the above solutions and shows the existence of beats for all speeds defined by $r \leqslant 0{\cdot}2$ and at some other rotational speeds. There are two beats in one revolution of the shaft and according to the methods of analysis of Manley(5) for waveforms when the phenomenon of beats is exhibited it follows that:

a) the resultant waveform has the same apparent frequency as the major component (that is the component with the greater amplitude) and its amplitude varies between the sum and the difference of the component amplitudes, the beat frequency being the difference between the frequencies of the components;

b) the separation of successive peaks (crests or troughs) at the bulge and at the waist of the beat determines whether the minor component is of higher or lower frequency than the major. If the peak separation at the bulge is greater than at the waist, the frequency of the minor component is less than that of the major; and if the peak separation at the bulge is less than that at the waist, the minor component is of higher frequency than the major.

The time responses of the equation (2) at all speeds when $r \leqslant 0{\cdot}2$ and $\varepsilon = 0{\cdot}3$ show that as the amplitude and instantaneous frequency fluctuate, the maximum amplitude and the maximum

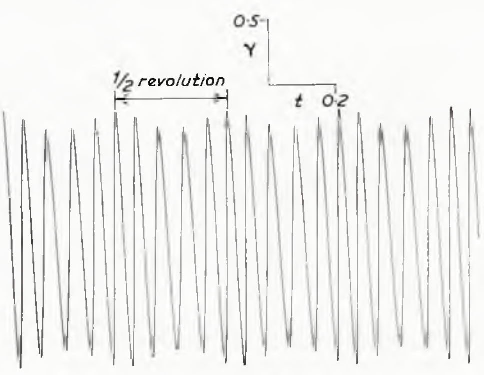

FIG. 2—*Theoretical waveform relationship of $\gamma \sim t$ for $r = 0{\cdot}1$ and $\varepsilon = 0{\cdot}3$*

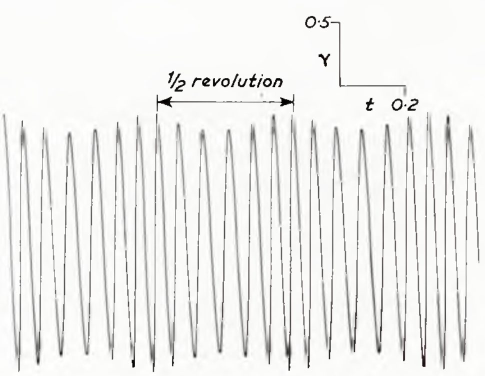

FIG. 3—*Theoretical waveform relationship of $\gamma \sim t$ for $r = 0{\cdot}08$ and $\varepsilon = 0{\cdot}3$*

apparent frequency of one oscillation of the solution occur together, as do minimum values for these quantities. Thus Fig. 2 shows the presence of 10th and 12th order harmonics, and Fig. 3 is composed of 12th and 14th order harmonics. Now if the system is considered to be acted upon by external excitations, then $r = 0{\cdot}1$ represents the condition for the 10th order direct resonance. The external excitations of both 10th and 12th order can put energy* into the system. When $r = 0{\cdot}08$, it is the case of 12th order direct resonance and the external excitations of 12th and 14th order can evoke the torsional oscillations.

In Table I of reference (4) is shown the analysis of wave forms when $\varepsilon = 0{\cdot}544$, and it can be seen that when $r = 0{\cdot}1$ the waveform is composed of 12th and 14th order harmonic components and for $r = 0{\cdot}08$ the solution contains 14th and 16th order harmonics. In the same way as above the harmonic components present in the solution determine the orders of the external excitations which can transfer energy to the system.

Draminsky(2) has stated that for the direct resonance of nth order, the energy can be transferred to the system from secondary components of $(n - 2)$ and $(n + 2)$th order. The present analysis indicates that the orders of the harmonic components through which the energy can be put into the system by external excitations of the same orders are determined from the solution of the equation (2) for the definite speed of rotation and the specific value of ε of the system. The authors also examined the solutions of equation (2) which are shown to exhibit beats for all speeds corresponding to $r \leqslant 0{\cdot}2$ where ε of the system is defined as $0{\cdot}1 \leqslant \varepsilon \leqslant 0{\cdot}544$. The solutions which correspond to the nth order direct resonance when the external excitations act on the system, do not show the existence of the $(n - 2)$th order vibrations, this being in contradiction with the theory put forward by Draminsky.

Draminsky(2) in his analysis has omitted the variable part, $2\varepsilon \cos 2\tau$, from the elastic term of equation (2) which appears when the equation is derived from energy considerations. A study of the solutions by omitting the above term from the equation indicates that, although the solutions do not change at lower values of $r \leqslant 0{\cdot}2$, the theoretical amplitudes of the waveforms at higher values of $r > 0{\cdot}2$ which are not considered extensively in the paper, are greatly affected.

EFFECT OF IMPULSE TERM $-\ \varepsilon \sin 2\tau$

The forcing term in equation (2) represents an impulse arising from the variable inertia due to the reciprocating parts. The equation,

$$(1 - \varepsilon \cos 2\tau)\gamma'' + 2\varepsilon \sin 2\tau\, \gamma' + \left(\frac{1}{r^2} + 2\varepsilon \cos 2\tau\right)\gamma = 0 \quad (3)$$

which corresponds to equation (2) with the forcing term omitted represents the free motion of the system. The responses determined from equation (3) for $r = 0{\cdot}06$ and $0{\cdot}5$ are shown in Figs 4 and 5

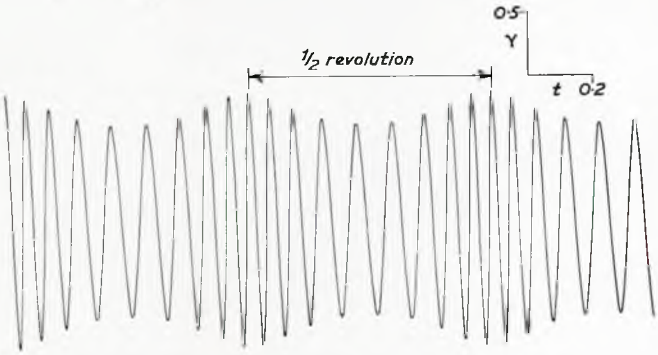

FIG. 4—*Theoretical waveform relationship of $\gamma \sim t$ for $r = 0{\cdot}06$ and $\varepsilon = 0{\cdot}544$ when the forcing term $-\ \varepsilon \sin 2\tau$ is omitted*

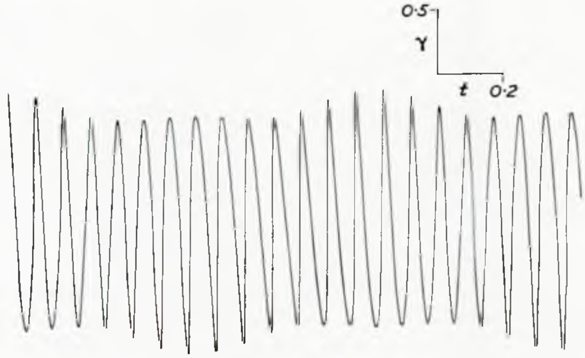

FIG. 5—*Theoretical waveform relationship of $\gamma \sim t$ for $r = 0{\cdot}5$ and $\varepsilon = 0{\cdot}544$ when the forcing term $-\varepsilon \sin 2\tau$ is omitted*

respectively. Comparing Figs 4 and 5 with corresponding solutions of equation (2), Figs 5 and 6 of reference (4), it can be seen that the response at $r = 0{\cdot}06$ shows no change whereas at the higher value of $r = 0{\cdot}5$ the response is different for the two cases and a reduction in the maximum value of the amplitude is noted. Therefore, it can be concluded that in the lower range of r, the forcing term which arises due to the incomplete balance of the reciprocating parts has no effect on the response of the system but at higher values of r, namely $r = 0{\cdot}5$, the amplitude and frequency are modified. The theoretical solutions show a modulation of amplitude and frequency which can be explained by the following method of analysis based on a process known as the WKBJ process.

THE WKBJ APPROXIMATION METHOD(7)

Rewriting the equation (3) in the form

$$\gamma'' + 2\left[\frac{\varepsilon \sin 2\tau}{1 - \varepsilon \cos 2\tau}\right]\gamma' + \left[\frac{\lambda + 2\varepsilon \cos 2\tau}{1 - \varepsilon \cos 2\tau}\right]\gamma = 0 \quad (4)$$

where $\lambda = 1/r^2$, and changing the dependent variable from γ to y through the relationship

$$\gamma = y \exp\left(-\int \frac{\varepsilon \sin 2\tau}{1 - \varepsilon \cos 2\tau}\, d\tau\right)$$

it becomes,

$$y'' + \left[\frac{\lambda + 2\varepsilon \cos 2\tau}{1 - \varepsilon \cos 2\tau} - \left(\frac{\varepsilon \sin 2\tau}{1 - \varepsilon \cos 2\tau}\right)^2 - \frac{(2\varepsilon \cos 2\tau - 2\varepsilon^2)}{(1 - \varepsilon \cos 2\tau)^2}\right] y = 0 \quad (5)$$

which on simplification reduces to

$$y'' + \left[\frac{\lambda - \lambda\, \varepsilon \cos 2\tau + \varepsilon^2 \sin^2 2\tau}{(1 - \varepsilon \cos 2\tau)^2}\right] y = 0 \quad (6)$$

Substituting

$$\frac{\lambda - \lambda\, \varepsilon \cos 2\tau + \varepsilon^2 \sin^2 2\tau}{(1 - \varepsilon \cos 2\tau)^2} = G^2(\tau),$$

equation (6) can be expressed as

$$y'' + G^2(\tau)y = 0 \quad (7)$$

If $G^2(\tau)$ has a large mean value about which only small fluctuations occur, then from the WKBJ approximation

*Energy is put into a system when a harmonic force (torque) acts on a harmonic motion of the same frequency and the work is done only with that component which is in phase with the velocity of the system(6).

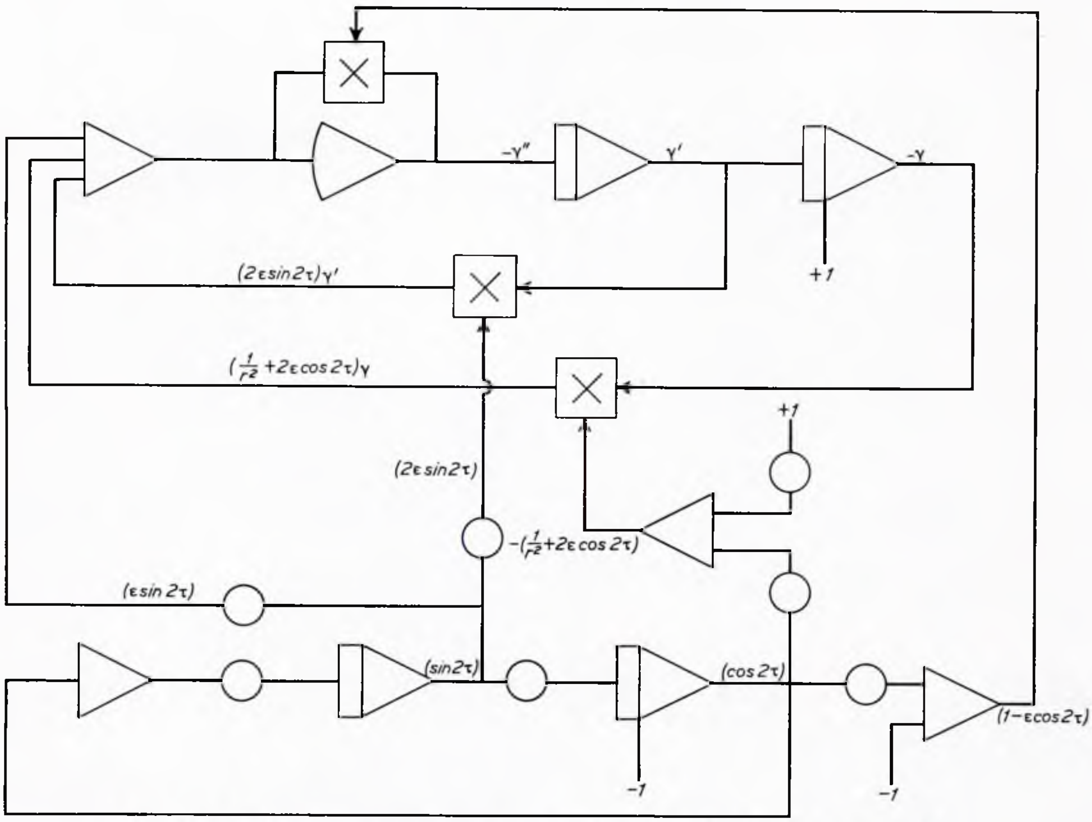

FIG. 6—*Analogue computer arrangement for the solution of equation (2)*

$$y = [G(\tau)]^{-\frac{1}{2}} A \cos [\phi(\tau) + \theta_0] \qquad (8)$$

where A and θ_0 are arbitrary constants and

$$\phi(\tau) = \int G(\tau)\, d\tau.$$

Therefore,

$$\gamma = A \left[\exp\left(-\tfrac{1}{2} \log_e\left(1 - \varepsilon \cos 2\tau\right)\right) \left\{G\left(\tau\right)\right\}^{-\frac{1}{2}}\right]$$

$$\cos \left[\int \frac{(\lambda - \lambda\varepsilon \cos 2\tau + \varepsilon^2 \sin^2 2\tau)^{\frac{1}{2}}}{(1 - \varepsilon \cos 2\tau)}\, d\tau + \theta_o \right] \qquad (9)$$

This solution represents the time response of γ in which a modulation of both amplitude and frequency is seen to occur. The region of r in which equation (9) is valid can be determined for specific values of ε.

ANALOGUE-COMPUTER ANALYSIS

The numerical analysis results are compared with the solutions obtained by use of an analogue computer. Fig. 6 shows the unscaled computer flow diagram of the arrangement for the solution of equation (2) which includes division networks. The symbols in the diagram follow conventional notation. Analogue computer arrangements involving division circuits are troublesome in practice since they exhibit a tendency towards instability. In this case it is found necessary to inhibit the instability by introducing a feedback capacitor across the high gain amplifier. Such an addition modifies the high-frequency response of the amplifier but no practical disadvantage arises in its use. The solutions at $r = 0{\cdot}06$ and $0{\cdot}5$ are shown in Figs 7 and 8 respectively. These solutions show close agreement with the corresponding numerical analysis results given in Figs 5 and 6 of reference [4].

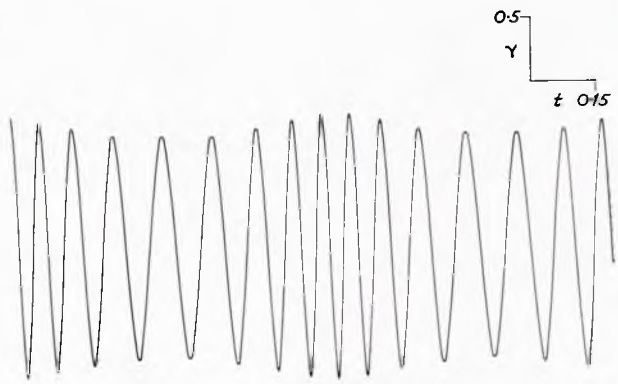

FIG. 7—*Analogue solution for the waveform relationship of* $\gamma \sim t$ *for* $r = 0{\cdot}06$ *and* $\varepsilon = 0{\cdot}544$

EFFECT OF FIRST ORDER TERM

It has been stated by Draminsky in his analysis that variation in inertia must be accompanied by a 'Coriolis' effect because the mass variation cannot take place except by moving the mass to another place in the vibrating system. This fact gives rise to the first order term in equation (2). By omitting the first order term, the equation can be written as follows, namely

$$(1 - \varepsilon \cos 2\tau)\gamma'' + \left(\frac{1}{r^2} + 2\varepsilon \cos 2\tau\right)\gamma = -\varepsilon \sin 2\tau \qquad (10)$$

Figs 9 and 10 show the time responses obtained from equation (10) corresponding to $r = 0{\cdot}06$ and $0{\cdot}5$ for $\varepsilon = 0{\cdot}544$. This shows that at the lower speeds of rotation, the maximum amplitude and minimum apparent frequency of one oscillation

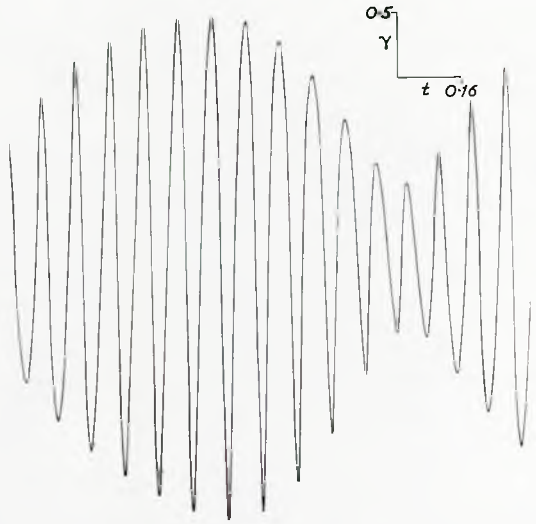

FIG. 8—*Analogue solution for the waveform relationship of $\gamma \sim t$ for $r = 0{\cdot}5$ and $\epsilon = 0{\cdot}544$*

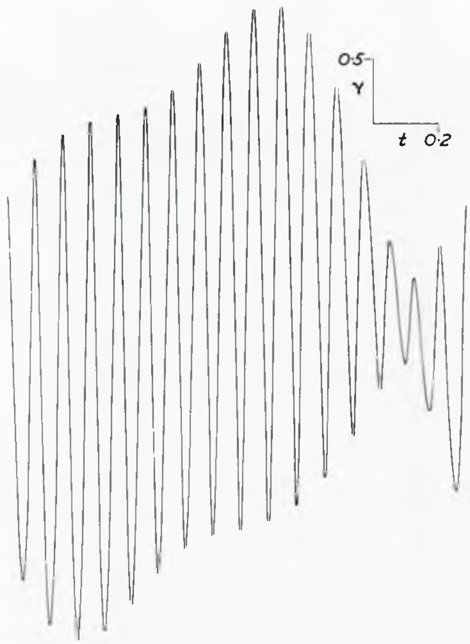

FIG. 10—*Theoretical waveform relationship of $\gamma \sim t$ for $r = 0{\cdot}5$ and $\epsilon = 0{\cdot}544$ neglecting the first order term*

of the solution occur together as do the minimum amplitude and maximum frequency. The shape of the response at $r = 0{\cdot}5$ is distorted compared to the beat form in the corresponding solution of equation (2).

Fig. 11 shows the variation of maximum amplitude against frequency ratio r. The pair of broken vertical lines bound the region of instability for $r \simeq 0{\cdot}5$. The solutions for $r > 0{\cdot}83$ are found to be unstable. The time responses are investigated in the range of $r = 0{\cdot}02$ to $r = 10$.

Fig. 12 gives the variation of the ratio of the shaft speed to the apparent frequency of response at maximum amplitude (ω/ω_I) over the range of $r = 0{\cdot}02$ to $r = 10$. Horizontal portions of the curve show the regions of instability clearly bringing out the fact that the ratio ω/ω_I remains constant at 0·5 and 1 corresponding to regions for $r \simeq 0{\cdot}5$ and $r > 0{\cdot}83$. Hence although the first order term is omitted the general behaviour of the system is modified but nevertheless still affected by the variable inertia.

EFFECT OF FORCING TERMS

Representative values for the harmonic components of the gas pressure tangential effort in internal combustion engines are given by Wilson(8). The various harmonic excitations are calculated from the corresponding tangential effort and the following single cylinder engine data.

Equivalent inertia of rotating and reciprocating parts

$(I_m = I + \frac{1}{2}Ma^2)$ = 97 697 Kg cm sec²
Bore diameter = 90 cm
Stroke = 155 cm
m.i.p. = 169·5 lb/in²
b.h.p. = 2900
$\epsilon = 0{\cdot}3$

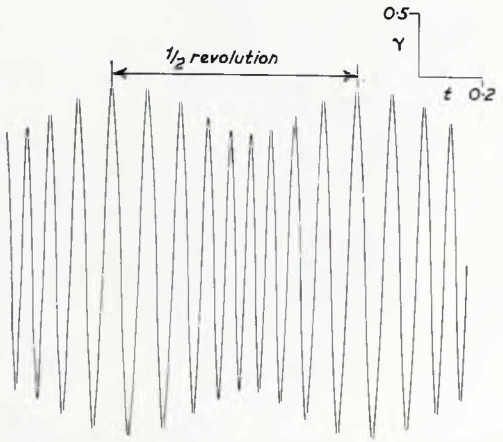

FIG. 9—*Theoretical waveform relationship of $\gamma \sim t$ for $r = 0{\cdot}06$ and $\epsilon = 0{\cdot}544$ neglecting the first order term*

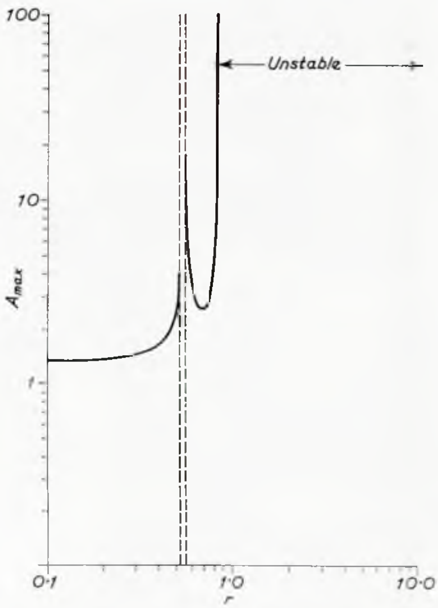

FIG. 11—*Maximum amplitude A_{max} of the response versus r for $\epsilon = 0{\cdot}544$ neglecting the first order term*

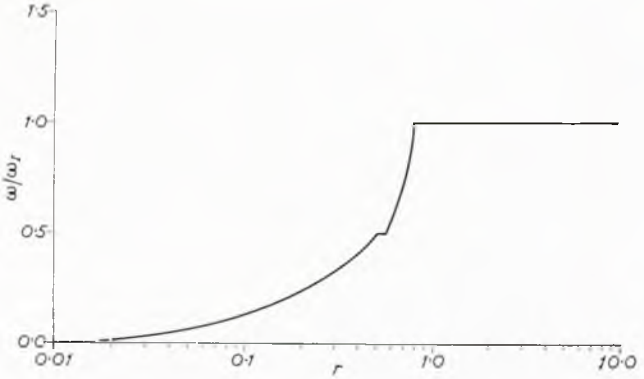

FIG. 12—*Relationship between ω/ω_I and r for $\epsilon = 0{\cdot}544$ neglecting the first order term*

The complete equation including the cylinder impulse K of order n periods per revolution is therefore

$$I_m(1-\epsilon\cos 2\omega t)\frac{d^2\gamma}{dt^2}+2I_m\omega\epsilon\sin 2\omega t\frac{d\gamma}{dt}$$
$$+\left(\frac{\omega^2}{r^2}+2\omega^2\epsilon\cos 2\omega t\right)I_m\gamma = K\sin(n\omega t+\alpha). \qquad (11)$$

Since the engine is of the two-stroke cycle type, the harmonic analysis of the tangential effort curve contains all the integer orders, that is to say $n = 1, 2, 3, 4$, etc. The responses of the system with each order of excitation from 1 to 4 are determined from equation (11). It is of interest to note that the responses are exactly similar to the case of free vibrations, that is as if no forcing term is acting on the system for $n = 1$, 3 and 4. The second order cosine component of the excitation also shows similar results, but the second order sinusoidal impulse arising from the incomplete balance of reciprocating parts has a significant effect at higher speeds for $r > 0{\cdot}2$ as shown previously. Further work is needed to study the effects of the higher order terms of the tangential effort arising from the gas pressure in the cylinder on the behaviour of the system.

CONCLUSIONS

The solutions of the equation of motion representing a single cylinder engine system, examined in the range of ϵ defined as $0{\cdot}1 \leqslant \epsilon \leqslant 0{\cdot}544$ at definite speeds of rotation corresponding to $r \leqslant 0{\cdot}2$ and at some higher rotational speeds exhibit the presence of beats. The orders of the harmonic components of motion through which the energy can be transferred to the system from external excitations of the same orders can be determined from the characteristics of the beats at the speed of rotation for a specific value of ϵ of the system.

For $0{\cdot}1 \leqslant \epsilon \leqslant 0{\cdot}544$ and the speeds corresponding to $r \leqslant 0{\cdot}2$, the solutions of the equation of motion show a modulation of amplitude and instantaneous frequency, and the maximum amplitude and maximum apparent frequency of one oscillation of the solution occur together as do the minimum values of these same quantities.

The absence of the first order term from the equation of motion increases the theoretical amplitudes of the responses in the stable regions. At lower speeds of rotation for $r \leqslant 0{\cdot}2$ the maximum amplitude and minimum apparent frequency of one oscillation occur together as do the minimum amplitude and maximum frequency which is the reverse effect to that exhibited by the complete solution.

The solutions of the equation of motion representing a single cylinder engine system with external excitations of order $n = 1$, 3, 4 and a cosine component of the second order as forcing terms show exactly similar solutions to those obtained for free vibration of the system. Further work on the effects of the higher order forcing terms will be given at a future date. The second order sinusoidal impulse arising from the variable inertia due to the reciprocating parts as a forcing term has no effect on the solutions in the lower range of $r \leqslant 0{\cdot}2$ but at higher values of $r > 0{\cdot}2$ it has a significant effect on the waveforms of the responses.

The analogue computer results confirm the accuracy of the numerical analysis.

ACKNOWLEDGEMENTS

The authors wish to express their appreciation to Lloyd's Register of Shipping for their assistance given in connexion with certain aspects of the investigation.

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Discussion

MR. W. MCCLIMONT, B.Sc., Chairman of Council, I.Mar.E., said that he was making the contribution on behalf of his colleague, Mr. D. H. L. Inns, and himself. He thought that Professor Carnegie had done an excellent job in the presentation.

As Professor Carnegie had given the meeting some additional information he craved Professor Carnegie's indulgence if he asked some questions which had been virtually answered.

As the author had mentioned, there had been a close link between Lloyd's Register and the work from the beginning, and would give the historical position as he and his colleagues at Lloyd's Register saw it, although this would to some extent go over the ground Professor Carnegie had covered in his introduction.

For many years the effects of ignoring the variable inertia characteristics of reciprocating engines on the accuracy of torsional vibration calculations were considered to be negligible; the associated secondary resonances and regions of instability tended to be dismissed by most engineers as interesting mathematical curiosities.

That situation changed in 1961, however, with the publication of Draminsky's work,* which indicated that secondary resonance effects could have contributed to a number of otherwise inexplicable crankshaft failures in large slow speed marine engines.

It was at that time that Lloyd's Register began to take an active interest in the secondary resonance phenomenon. Several cases were investigated by the society, along the Draminsky lines, and vibration measurements were taken on the actual installations. The result of two such examples were given in a paper in 1964.†

Admittedly, both examples were chosen from prior knowledge of failure, but the correlation between the practical measurements and the Draminsky calculations clearly demonstrated the existence of a phenomenon whereby an otherwise innocuous critical in the region of the service speed could be considerably magnified presumably by interaction with a powerful, but non-resonant, excitation of the $(n - 2)$ harmonic type.

The Draminsky work served to indicate, in very broad terms, the circumstances in which adverse secondary resonance effects could be anticipated.

However, since the problem was not analysed exhaustively in depth, resort being made to a number of approximations and simplifications, the work did not provide a tool of infallible precision in the prediction of the necessary conditions for resonance.

In view of this, the policy of Lloyd's Register, in its routine examination of torsional vibration characteristics, had been to apply the simple tests for the possible occurrence of the $(n - 2)$ type of harmonic interaction in the operating speed range, without going into detailed calculations.

Whenever positive indications were obtained, the engine builders were duly warned of the situation and vibration measurements were requested as a precautionary measure. That procedure was in line with the society's policy that, in the final analysis, the vibration characteristics of a marine installation must be shown to be acceptable on the basis of practical measurement. In most cases it was possible to obtain assurance of satisfactory vibratory conditions at an early stage by taking the measurements during test bed running, so avoiding delay in building.

Over the period up to the present time, very few potentially dangerous cases had been identified, presumably because the engine builders were sufficiently informed to take avoiding action. However, in the few likely cases detected by the society, subsequent measures failed to indicate any marked departure from characteristics predictable by routine conventional calculations.

The practical example provided in the present paper would seem to be a case of that nature, in which the predicted Draminsky resonance magnification failed to appear when measurements were taken.

It was against that background of some ten years' experience that the authors' paper was welcomed as the first attempt to provide a systematic and fundamental appraisal of the problem, leading, it was hoped, to the ultimate "laying" of the secondary resonance spectre.

The authors should be commended for their courage in embarking on such a lengthy research programme, involving, as it did, the critical term-by-term examination of the mathematical expressions and the exploratory evaluation of ranges of combination of several important variables.

It was fully appreciated that, relative to the overall completion of the project, the present paper must be treated as an interim report by the authors on the progress of their researches so far.

* Draminsky, P. 1961. "Secondary Resonance and Subharmonics in Reciprocating Engine Shafts." *Acta Polytechnica Scandinavica,* Me 10, Copenhagen.

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

The basic mathematical analysis of the problem had been formulated, the requisite computing tools and methods had been devised to provide solutions to the equations, and progress had been made in exploring the effects of the more important variables.

However, the stage had yet to be reached at which positive predictions regarding actual full scale marine installations could be made. That, no doubt, explained why the practical examples included in the paper, and for which measurements exhibited no noticeable resonance effect, had been passed over without comment. Possibly the authors would venture to comment now on that example—and, to an extent, Professor Carnegie had done so—in the light of their investigations, using the appropriate numerical values for the variables r and ε to get them into focus.

The conclusions to the paper also appeared to be phrased as statements of factual observation, rather than practical guidelines for the engineer. Presumably, that feature also reflected the reluctance of the authors to crystalize their findings at the present time in broad terms related to actual marine installations.

It would perhaps be constructive, therefore, for an onlooker to attempt to read between the lines and guess at the general implications of the paper.

So far, the authors had explored the effects of varying the two controlling parameters; namely, the reciprocating inertia ratio, ε, and the frequency ration r.

Unless the presentation had been misunderstood, it would appear that the following implications were deducible:

1) that the effects of secondary resonance were far more general than the work of Draminsky would indicate;
2) depending on the particular combination of ε and r, the difference in harmonic order number between the excitation and the secondary resonance response was not confined to the figure two; integer values other than two were reported in the paper; it was suggested by Lloyd's Register, therefore, that such integer values represented special cases only and that, in fact, the harmonic order number difference was a quite general characteristic ranging continuously through integer and non-integer values; as Professor Carnegie had indicated in his introduction, one presumed—and one now knew—that the wave forms chosen to illustrate the paper represented responses which were readily identified as exhibiting integer harmonic order number differences according to the Manley criteria. Therefore, he thought that Professor Carnegie agreed with what he and his colleagues would suggest, which was that response wave forms for intermediate combinations of variables, if subjected to rigorous harmonic analysis, might reveal the continuity of the difference characteristic in the non-integer ranges;
3) it should be borne in mind that the analysis so far had dealt with an idealized representation of a single cylinder engine, and, in particular, that an infinitely long connecting-rod had been assumed; that feature restricted the inertia variation to the predominant second order; however, a true representation, allowing for connecting rod obliquity, would include the higher orders of variation, although of smaller magnitude; it was suggested that, if those higher orders were included, the complete generality of the secondary resonance effects in all respects would be demonstrated;
4) further, it would be extremely helpful to the practical engineer if the authors could summarize the results of their investigations graphically in terms of harmonic order numbers or differences; it was felt that a graphical "carpet plot" showing the variables ε and r, together with the corresponding responses in harmonic order form, would clearly indicate the regions of practical interest relating to secondary resonance and would provide a particularly useful diagnostic tool for the engineer.

As to the future, having laid down a firm ground-work, much remained for the authors to accomplish before their task was complete.

As Lloyd's Register saw it, it was first essential to continue the examination of response patterns arising from the application of successively higher order external forcing excitations. The work so far had been based largely on free vibration responses, in which, in a sense, the responses had been controlled by the mass-elastic configuration of the idealized system. With external forcing, the response of the system would be more under the control of the investigator, and the results more convincing.

It was also felt that the true nature of secondary resonance response would not be completely demonstrated until two different external excitations were applied to the system simultaneously. Those could represent one harmonic excitation on resonance and the other off-resonance with a harmonic order number difference appropriate to the demonstration of secondary resonance interaction.

That having been achieved, it would then be necessary to correlate the results for the simplified analytical model with the case of a full scale multi-cylinder engine.

No doubt such considerations as damping and phase relationships, both as dictated by crank arrangement and as between different harmonics arising from the analysis of a typical gas torque curve, would have to be explored.

In those respects, the authors might care to comment on the simplifications adopted by Draminsky in his calculations. Was it justifiable to resort to vector summation and the dynamic magnifier concept of damping in the representation of a multi-cylinder engine, or could the true nature of secondary resonance be masked by those approximations?

Bearing all those points in mind, the progress of the authors' work would be followed with keen interest in the future.

As a final comment and purely as an aside, it was interesting to consider the variety of investigatory tools now available to the researcher in that field of work. Such devices ranged over instrumentation for practical measurement by amplitude pick-up and strain gauge, the digital computation of natural frequencies and forced-damped synthesized characteristics and, finally, the analogue computer simulation.

DR. F. ØRBECK, M.I.Mar.E., said that, for the anti-vibration designer, it was interesting to see that research was returning to the problem of secondary resonance. Perhaps it was better referred to as the effect of varying moments of inertia of the individual cylinder sections. In the earlier days of the study of torsional vibrations the fundamental work on the effect of varying moments of intertia was extensive, but it was found that available calculation facilities were unable to cope with that kind of complication.

In designing a shaft system one had to take a decision about what inaccuracy to consider next. For instance, one could keep the basic Holzer frequency table calculation, but use a finer subdivision of stiffnesses and inertias. During the last few years, in his company, this calculation had extended from about 10 masses to about 30 masses per system and a branched system was now used. It was, of course, quite impossible to consider variable inertia without reducing the number of masses of the system. A second development had been to go to a forced—damped type of calculation. The system was still linear, but both applied force and damping were taken into account in the torque balance. This type of calculation had been very profitable, particularly on flank stresses. With the introduction of the geared medium-speed engines one had to consider the non-linearity of couplings as an extra complication and finally, now the "ugly head" of the variable inertia was raising itself again. He had, therefore, been particularly interested in the remark, by Mr. McClimont, that Lloyd's Register had a rule which gave guidance as to when one would require to consider secondary resonance. So far the effects of secondary resonance had never been noticed in his company. For their J engine ε was, generally speaking between 0·15 and 0·23. In many cases the engines ran through major order one node resonance peaks where $v = 1$, and $v = \frac{1}{2}$ would also be a condition to be watched. In order to devise a simple rule he invited those present to refer back to the first equation of the first paper. That equation was applied to a cone-mass system and he was in full agreement with this procedure.

Let it be assumed that one considered a six-cylinder engine which ran through a sixth order resonant peak at about 50 rev/min, the flexibility of the intermediate shaft would be about five times the flexibility of the crankshaft. In those circumstances it would be very tempting to lump all the crankshaft masses together. One then noticed that if the firing order was such that the secondary out of balance force on the engine was equal to zero, there was a constant inertia for the engine. The effects of secondary resonance should consequently be very small.

He asked the authors whether there was any indication that the engines, which were suspected of having had secondary resonance, were engines with large out-of-balance second order forces? For two node vibrations it might be that they did not have a large out-of-balance over the whole engine, but an out-of-balance on the forward end balanced by the forces from the remaining cylinders.

Author's Reply

Replying to the discussion Professor Carnegie said that at the present stage he could not answer Dr. Ørbeck's question. He would have to look at the problem in much greater detail. It was his intention to study the effect in multi-cylinder engines. He had no doubt that Dr. Ørbeck might be able to get in touch with some of the people at Lloyd's Register, and an excellent person with whom to have a discussion would be Dr. Archer, for the latter had done a great deal of work on practical aspects of the problem.

Taking first the last point by Mr. McClimont, he had read Draminsky's work and, having got to the end of it, he was not a great deal wiser than when he started it; that was how he summed it up. That was one of the reasons why he and his colleagues had started the investigation. He was not criticizing Draminsky; he felt that the work which he had produced was good as far as it went as he had started people thinking about the problem. There was no doubt that people said: "Is the effect present, or is it not present?" Of course, every engine system was a variable inertia system. Whether the effect always manifested itself was quite a different point. He and his colleagues certainly hoped that they could produce some results of real practical value in due course, even if all they could say was that any unusual effects were not likely to be due to that factor. They must then be due to other additional effects present in the system. He felt, however, that in certain cases there was a strong indication that secondary resonance did have an effect. He and his colleagues would certainly not like to be dogmatic at the present stage of the investigations about their conclusions.

He took the point, of course, that there were the effects of the obliquity of the crankshaft and the number of cylinders in the engine to allow for. One had, on the other hand, to start with a relatively simple model in order to understand the basic problem. From that beginning he and his colleagues would hope to extend the work into more complex types of system.

He would also aim in the course of time to produce a summary of some type for use by practical engineers. They would also look, in fact they were currently looking, at the problem of the application of the applied external torques. They had already carried out an analysis with more than one forcing torque applied to the system simultaneously, and intended to carry that work further.

Damping was also one of the items in their programme for investigation. They intended to consider what the effects of damping on the variable inertia system were. The investigation covered a very wide field and the work could well last for a period of several years.

MEASUREMENT OF VIBRATION BY HOLOGRAPHY

B. S. Hockley, B.Sc. M.Tech.*

The application of holography to the vibrational measurement on engineering components is described together with its advantages over the usual sand pattern and contact probe methods. The time averaged fringe holographic technique has been used to study aero engine compressor blades, turbine discs and shaft assemblies. Holographic interferometric methods of real time vibration analysis, to quickly obtain some information on a number of vibration modes, are discussed.

The practical requirements and equipment for holography are studied and the development of a completely self contained holographic unit that has been designed specifically for vibration measurements on components up to 2 ft diameter.

Mr. Hockley

INTRODUCTION

As engineering components, which are subjected to vibrations, are becoming increasingly complex and are made to perform closer to their limits, the need for accurate knowledge of their vibrational characteristics becomes more critical. Experimental measurements of vibrational modes are usually made by using either Chladni sand patterns or a piezoelectric contact probe. Sand patterns are not always possible on complicatedly shaped objects because the sand tends to fall off the object or on extremely rigid components where it is difficult to generate sufficient vibrational amplitude. The contact probe overcomes most of these difficulties, although it is limited on very small vibrational amplitudes, but it is very time consuming. These limitations can be largely overcome and the accuracy of the nodal position measurement increased by the use of holography. Holography has the additional advantage of providing a contour map of lines of equal vibrational amplitude of the component. It has also proved possible to develop a self contained holographic system for application to vibrational measurement on a routine basis.

HOLOGRAPHY

Holography is a method by which both the phase and amplitude of the wavefront scattered from an object illuminated by coherent light are recorded in the form of an interference pattern on a photographic plate. The original wavefront can be reconstructed by illuminating the photographic plate with coherent light to form a complete three dimensional image of the object.

The light source coherence requirement is that the light scattered from any part of an object is capable of interfering with the light projected directly on to all parts of the plate. See Fig. 1a. This criterian is fulfilled by the gas laser, typically the He-Ne at 632·8 nm and argon ion at 514·5 nm.

A schematic diagram of a hologram recording system is shown in Fig. 1a. The object is illuminated by part of an expanded laser beam and some of this light interferes with the light directly reflected on to the plate by the mirror M and the resultant interference pattern is recorded by the photographic plate. The plate is developed and placed in the arrangement shown in Fig. 1b where it will produce a true three-dimensional image, in the position shown, when illuminated by coherent light. A full analysis of this process can be obtained.[1,2].

If the hologram plate is accurately repositioned back into the original system and illuminated with the original coherent reference beam the wavefront diffracted by the hologram is identical to the wavefront scattered from the object's surface. Since the two waves are indistinguishable they may be interchanged and hence it is possible to compare the reconstructed wavefront with the scattered wavefront originating from the object at some later time. Any difference between the object resulting, for example from an applied stress and its original

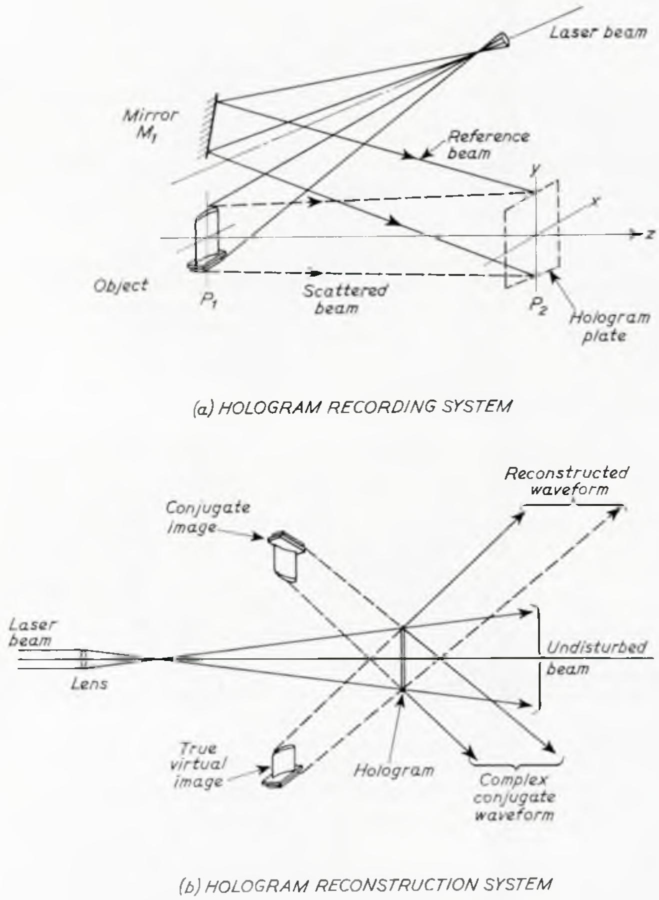

FIG. 1—*Schematic diagram of the holographic recording and reconstruction systems employed*

reconstruction is shown as interference fringes similar to those produced by standard interferometry. This is the principle upon which holographic interferometry is based[3].

In a similar manner two holograms may be recorded on the same photographic plate representing the object at different instances in time and the resultant reconstruction will show any displacement of the object that has occurred between the two exposures in terms of fringes.

Holography of Vibrating Objects

The effects of a vibrating object on a holographic system were first investigated by Powell and Stetson[4] who considered

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the time dependence of the variation of the wavefront scattered from the object, when in vibration, on a holographic recording. This technique was named the time averaged fringe method and the complete hologram was recorded while the object was in vibration.

In the case of an object vibrating with simple harmonic motion Powell and Stetson showed that the resultant image intensity at the point (x_1, y_1) on the image is in the form of a zero order Bessel function:

$$I(x_1, y_1) = J_o^2\left[\frac{2\pi}{\lambda}(\cos\theta_1 + \cos\theta_2)m(x_0, y_0)\right]I_{st}(x_1, y_1)$$
$$= J_o^2[X]\, Ist(x_1, y_1) \qquad (1)$$

when $m(x_0, y_0)$ is the vector of the maximum displacement of the point (x_0, y_0) on the object and θ_1 and θ_2 are the angles between the axis of observation from the hologram and the vector displacement and the direction of propagation of the illuminating light upon the object and the vector displacement respectively. $I_{st}(x_1, y_1)$ is the intensity of the point (x_1, y_1) on the image of the stationary object.

The $J_o^2[X]$ function is a periodic varying function whose maxima give bright fringes and minima dark fringes. The bright fringes represent a total excursion of the object of approximately a half wavelength. Hence the resultant fringe pattern on the image provides a bright zero order fringe for the nodal area and higher order fringes diminishing in brightness denoting the amplitude of vibration.

Equation (1) for the fringe formation of an object in simple harmonic vibration may be interpreted as a function of the position probability density function for simple harmonic motion which is high at the two extreme positions of vibration and gets progressively lower at points away from the two extremes to reach a minimum at the mid-point of vibration. See Fig. 2. If the vector displacement is a random function with respect to time the resultant fringe pattern is only dependent on the position probability density function.

Another and greatly simplified way of describing the formation of the vibrational fringe pattern from the probability density function of simple harmonic motion is to consider that the reconstructed image is composed of ensemble of a large number, infinite in the limiting case, of sub-images each representing a particular position in the object's vibration cycle. The intensity of each sub-image is directly proportional to the time the object spends in that position. Hence from the probability density function the object spends most of the time at the two extreme amplitude positions and so there are effectively only the two sub-images, of the extreme amplitudes position, in the final reconstructed image that interfere to produce the resultant fringes as in the case of the normal double exposed hologram.

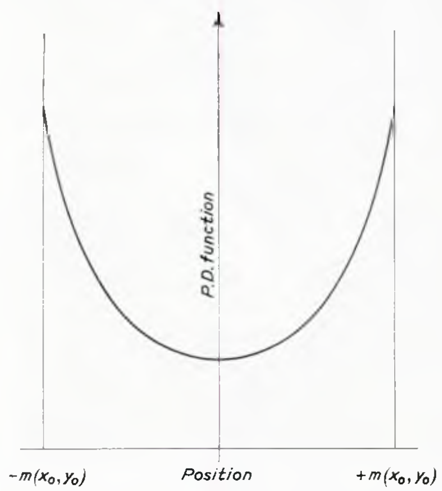


FIG. 2—*Position probability density function of simple harmonic motion*

Application of Holography to Vibration

The measurement of the vibration characteristics of a component normally involves the study of the first fundamental modes plus some of the higher modes over a particular frequency range to which the component is liable to be subjected to in service. This requires a reasonably quick scan of a number of modes followed by detailed studies of particular modes. This information can be obtained holographically by several methods utilizing the basic time averaged fringe technique.

The frequency scan involves holographic interferometry whereas the detailed information is obtained from the basic time averaged technique which has been studied in detail by the author to evaluate the process as a method of analysis of aero engine components. Some examples of the results of these tests

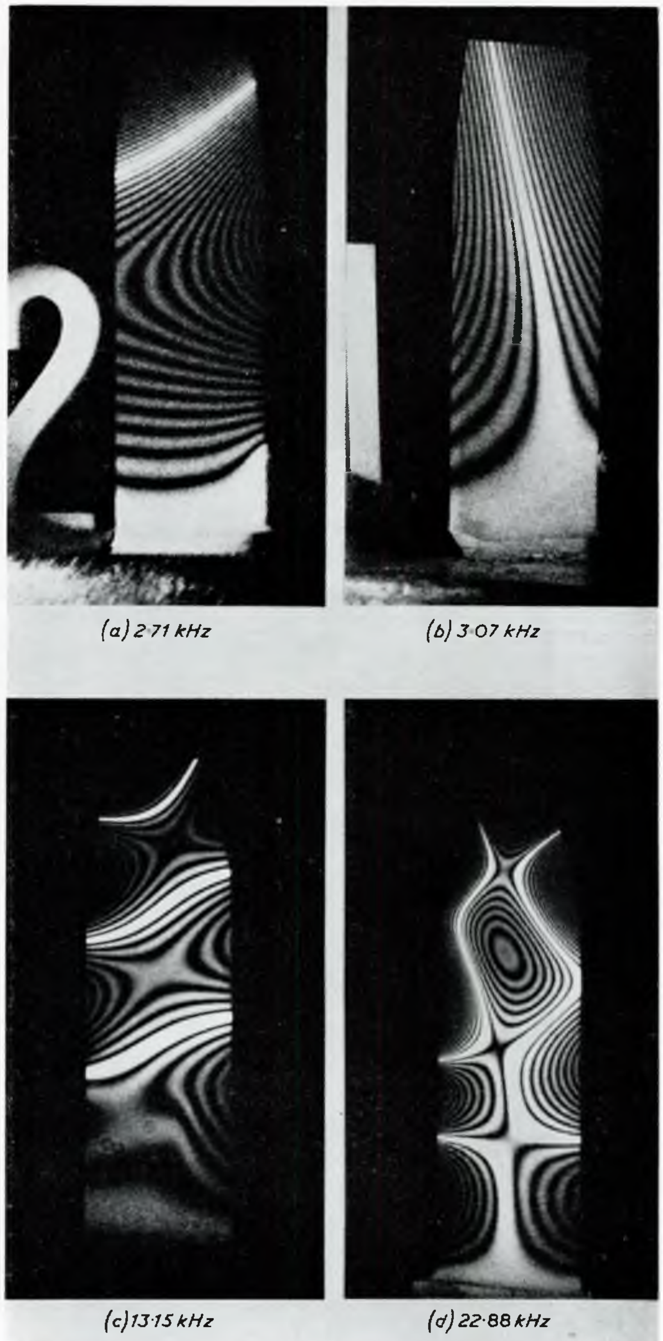
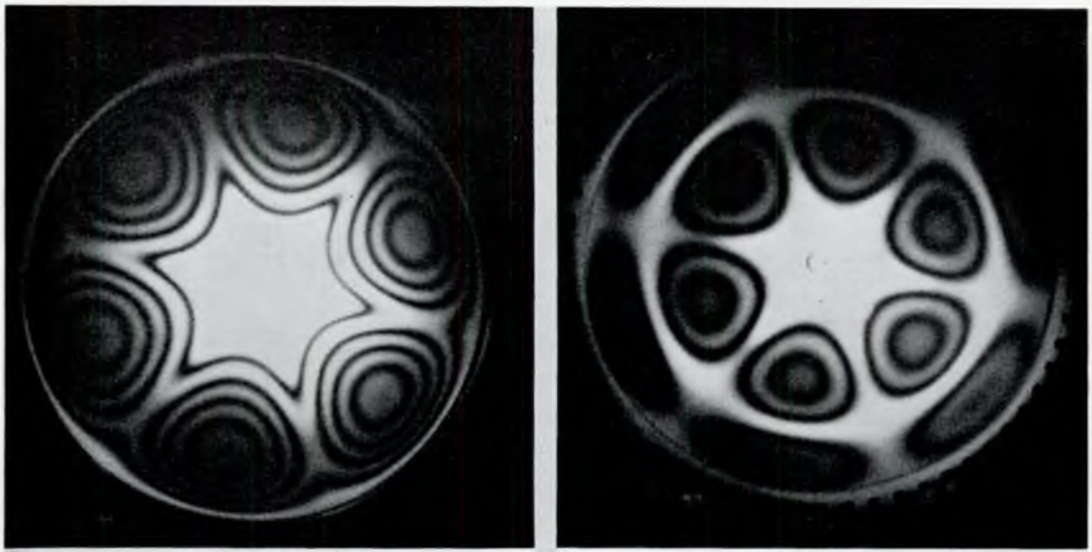


FIG. 3—*Hologram reconstruction of two compressor blades in vibration*

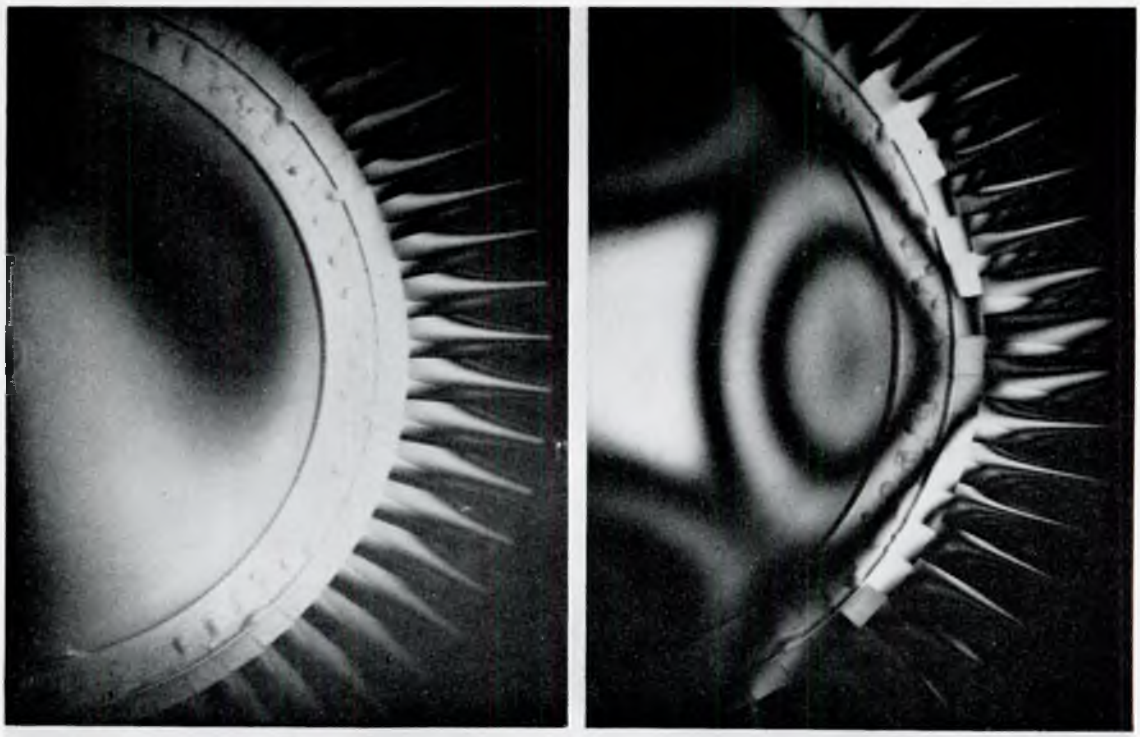
are shown in Fig. 3. The vibrational nodal lines are clearly seen as bright fringes and the relative amplitude contoured by the higher order fringes. The vibrational amplitudes are normally chosen to give a maximum of approximately 15 fringes, higher orders being avoided since the intensity function $J_0^2[X]$ drops rapidly and beyond this value the relative intensity is approximately 1 per cent of the nodal line. The hologram will readily accommodate a greater dynamic range of intensity, at least 10,000, but most photographic films and printing paper used for hologram reconstructions are limited to a dynamic range of 100. In some cases however up to 70 fringes have been observed by over exposing the film for the low order fringes.

The results in Fig. 4 show the vibration modes of a 10 in diameter disc at 11·9 and 22·9 kHz. The hologram reconstructions show the nodal line distributed symmetrically over the disc. Fig. 5 shows the 10 in diameter disc with a full set of blades cemented into place. The vibration pattern of Fig. 5b shows a vibrational node running from the disc to the blades to illustrate



(a) 11·9 kHz (b) 22·9 kHz

FIG. 4—*Hologram reconstructions of a specific turbine disc in vibration*



(a) 3·56 kHz (b) 4·6 kHz

FIG. 5—*Time averaged fringe hologram reconstructions of a 10 in diameter specimen turbine disc with a full set of blades*

the coupling effects between the disc and the blades. This would be very difficult to demonstrate by other methods.

Since a hologram has the property that it retains the optical signature of each individual point on the object it is possible to detect the movement of a surface that is purely along the plane of that surface. Other techniques such as moire do not have this property unless one of the moire grids is attracted to the objects surface.

By using this property it is possible to investigate the purely torsional vibration of a cylinder to measure the rotational amplitude of any part of the surface. Fig. 6 shows a turbine shaft assembly vibrating at 1·3 kHz and 5·3 kHz in pure torsional modes about the axis of the shaft. A bright fringe appears down the front of the shaft along the line of sight from the hologram. This does not indicate a node as the shaft is moving in the region. The movement is at right angles to the line of sight from the hologram and so must be large to produce an optical path length change of half a wavelength along the line of sight to form a dark fringe. Hence with the bright fringe the movement of the surface produces an optical path length change of less than half a wavelength. The angular rotation of the surface is inversely proportional to the fringe spacing projected onto a plane at right angles to the direction of observation, i.e. the image plane[5]. In Fig. 6b the assembly approximately half-way along the shaft is rotating at a higher amplitude than the rest of the assembly.

Another method of generating similar information is to interrupt the laser beam to form a stroboscopic illumination of the object in a holographic interferometer. The stroboscopic effect can be produced by a mechanical chopper or an electro-optical Pockel cell. In the case of the interferometer the vibrating object is frozen at one part of its vibration cycle so that the deflexion of the blade in that position relative to its stationary image is shown by the fringe pattern. The vibrational information can be permanently stored in the hologram, as in the time averaged fringe method, by exposing the hologram in two equal stages, the first when the blade is stationary and the second when the blade is vibrating and stroboscopically illuminated.

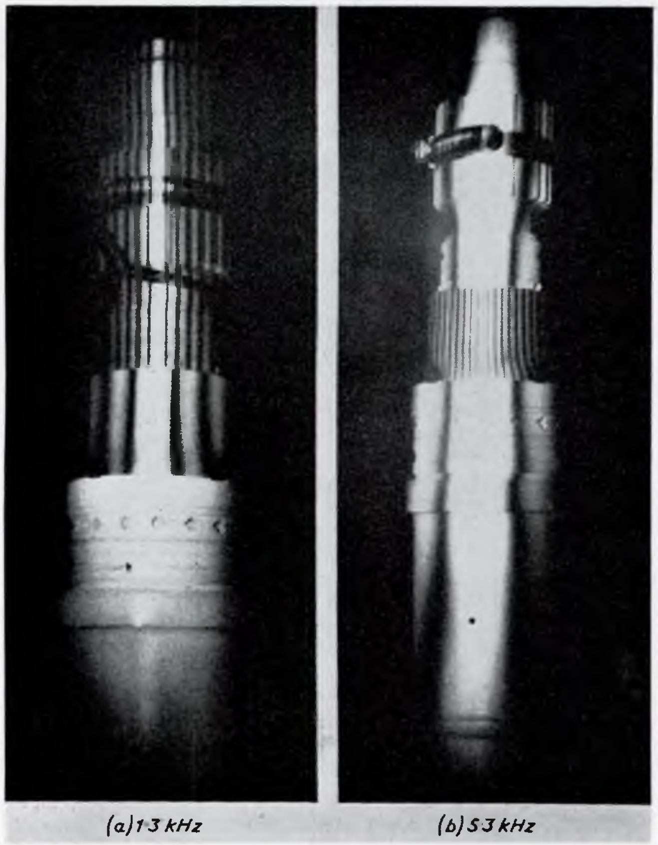


FIG. 6—*Hologram reconstructions of shaft assembly in torsional vibration*

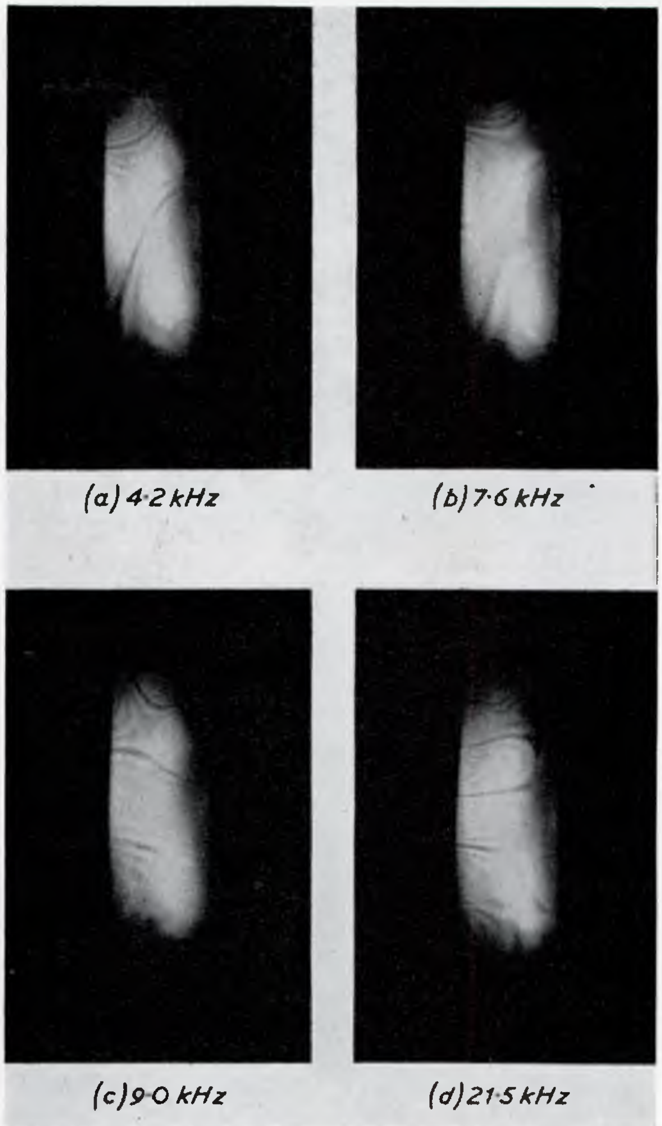


FIG. 7—*Real time averaged fringes for a* 3 *in a*l. *blade*

Real Time Vibration Measurement

There are two holographic techniques available for a scan of the resonant modes that involve holographic interferometry in the form of real time averaging fringes and fringe spoiling.

In the real time averaging technique the hologram plate in the interferometer is accurately relocated and adjusted to obtain the minimum number of fringes across the object. The theoretical ideal is one dark fringe over the whole object but this is not always possible due to emulsion shrinkage. To obtain the accurate relocation a specially designed plate holder is used that has relocation accuracy and adjustment to within 10 micro inches. An alternative method is to develop the plate *in situ* in a plate holder in which the processing chemicals flow over the plate.

When the object is vibrated the displacement produces a fringe shift that is averaged by the eye or camera in a similar manner to the standard time averaged fringes of the Powell and Stetson techniques. The fringe patterns in Fig. 7 show the real time averaged fringes of a 3 in long compressor blade at different frequencies. The nodal lines are seen as either a dark or a light fringe depending upon the adjustment of the plate holder.

The contrast of the resultant fringes is somewhat less than that of the time averaged fringes since the intensity of the image is of the form $(1 + J_0[X])$ rather than $J_0^2[X]$ together with the logarithmic intensity response of the eye. Also there are half the number of fringes for the same deflexion as in the time averaged case. The low constant limits the vibrational amplitude that can be used and the fringes are difficult to photograph, for a permanent record, but it does enable a vibrational mode scan to be made quickly.

For the fringe spoiling technique the relocated hologram plate is displaced by a controlled amount to produce a series of straight line fringes across the object. The number and direction is governed by the direction and amount the hologram is displaced.

When the object is vibrated at a resonant frequency the straight line fringes are blurred at any point that is in motion. The reason for this is that the fringes move rapidly to indicate the displacement during each vibration cycle and the eye cannot follow their rapid movement and a camera exposure time is considerably greater than a fraction of the vibration cycle. Only at the stationary points on the object, the nodal areas, will the fringes remain stationary and observable. The fringes across a stationary 3 in compressor blade are shown in Fig. 8 and Figs 8b–d show the nodal areas of the vibrating blade for three different resonant frequencies. This method only shows the nodal areas of the component, but it is a real time method as the patterns are observed as the object is actually vibrating rather than having to expose and process the hologram before seeing the results.

The use of the fringe spoiling technique can be extended by interrupting the laser beam to stroboscopically illuminate the object. The stroboscopic illumination removes the blurred averaging effect of the vibrating parts of the object and shows the distortions of the straight line fringe pattern caused by the deflexion. From the amount of distortion the amplitude of vibration is deduced and from the direction of the distortion the direction of vibration is given resulting in the vector displacement of the component[6]. The direction information is calibrated before the test where the object is deflected towards or away from the observer and the direction of the fringe shift noted.

This technique is particularly pertinent to the analysis of the relative phase of parts of a complex component or where two modes are combined at one resonant frequency with a 90 deg. phase difference. By varying the phase between the stroboscopic pulse and the components vibration the development of such a complex vibration mode can be analysed.

The various holographic vibration measurement techniques are summarized in Table I.

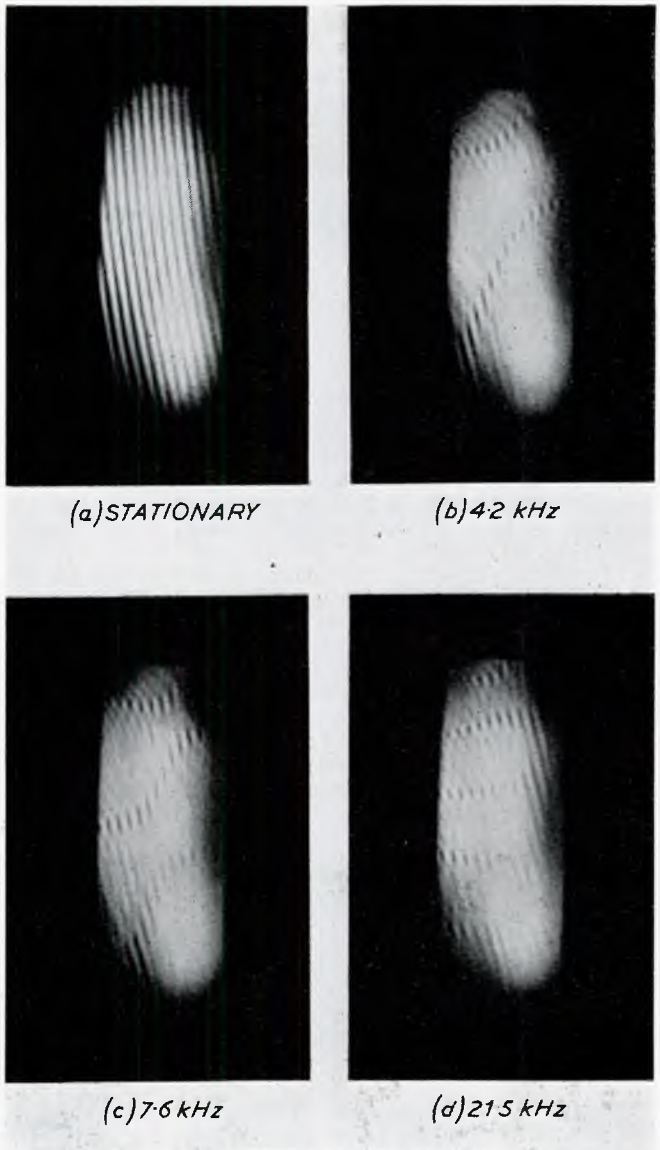


FIG. 8—*Fringe spoiling patterns of a* 3 *in* a1. *compressor blade*

The Use of Holographic Methods

The holographic methods of vibration measurement can be divided into two distinct categories, quick nodal pattern and amplitude information of a large number of vibration modes and detailed analysis of particular modes. The real time interferometric techniques provide the quick information scan and the time averaged fringes for the detailed information.

Stroboscopic holography is somewhat more difficult than the other techniques as it involves a chopping system and accurate synchronization of the strobing pulse to 'freeze' the object to within an eighth of a wavelength for the period of observation to obtain good contrast fringes. The fact that the object is only illuminated for 5 per cent, or less, of its cycle greatly reduces the averaged available light, hence requiring longer film exposures and so increasing the length of the stability requirements.

The accuracy of the nodal position measurement is better than either the sand patterns or the contact probe, particularly if a micro-densitometer is used to scan the photographic negative of the hologram reconstruction. The amplitude information provides the deflexion of the component in great detail which has not been previously attainable. This provides indications of stress concentrations for the location of strain gauges for further test e.g. full engine tests in the case of aero engines, and data from which strain measurements can be made.

The techniques are particularly useful for complex objects such as assemblies where combined modes are present and for torsional vibrations of shaft assemblies in which it is possible to reduce the analysis time from several days to a few hours.

Practical Considerations of Holography

The most critical requirement for holography, apart from the laser, is that the system must remain dimensionally stable over the period of exposing the hologram to within $\lambda/8$ such that the interference pattern produced by the reference and scattered beams at the hologram plate is not blurred. To achieve this complete vibrational isolation is required. This can be achieved by a heavy cast iron or granite table with an isolating system of sand, anti-vibration rubber matting or pneumatic, motor car inner tubes. Tests on a 6 ft 6 in $\times$ 4 ft 6 in hollow cast iron table weighing 25 cwt mounted on four inner tubes gave an indicated vibrational fringe shift of 1/40 fringes from a Michelson interferometer with a medium sized machine shop located within 6 ft of the table. Hence effective isolation can be achieved for a small cost.

Standard laboratory optical components, such as microscope objectives to expand the laser beam, simple lenses and mirrors can be used in conjunction with magnetic clamps, on a metal table, for directing and controlling light within any holo-

TABLE I—HOLOGRAPHIC TECHNIQUES SUMMARIZED

Technique	Vibration Application	Information Content	Type of System Used	Comments
Time Averaged Fringes	Detailed Measurement	Node and amplitude profile	Standard recording and reconstruction	Accurate and detailed results. Simple to operate. Can be used in a routine basis. Hologram must be processed before results seen.
Stroboscopic	Detailed Measurement	Node and amplitude	Standard recording Laser strobing mechanism	Strobing system complex. Low light levels—Long exposures. Same information as in Time Averaging. Hologram must be processed before results seen.
Real Time Averaging	Frequency Scan	Node plus some amplitude fringes	Interferometer	Measurement in real time (As objects vibrate). Fringes low contrast
Fringe Spoiling	Frequency Scan	Node	Interferometer	Accurate node position obtained quickly
Fringe Spoiling with Strobing	Frequency Scan plus detailed analysis	Node, amplitude and relative phase	Interferometer with laser strobe	Strobing system complex and difficult to set. Low light levels because of strobe. Deflection vector obtainable.
Strobing Interferometer	Frequency Scan	Node and amplitude	Interferometer with laser strobe	Strobing system complex. Low light levels.

graphic system. For holographic interferometry on accurate photographic plate relocating system is required when the plate is removed for processing. Alternatively a method of processing the plate *in situ* can be used. Relocation accuracy and adjustment can be achieved to within 10 micro inches by normal kinematic design. The design of an *in situ* processing device is less exacting as the plate is permanently clamped in position with the processing chemicals flowed over the plate. To use the *in situ* processing on a routine basis it is possible to make the development completely automatic by using a peristaltic pump and a series of solenoid operated valves to control the chemicals and the overall time required for processing can be reduced from 20 minutes to under 5 minutes. This is principally because no drying time is required for the emulsion as the photographic plate is exposed when surrounded by water or developer[7] and the interferometer observed with the plate surrounded by either water or fixer.

A range of gas laser powers are suitable for holography depending upon the size and optical finish of the object. A 5 mW He-Ne laser will accommodate objects up to 15 in diameter with either a matt white painted or a clean machined finish with an 8 sec exposure on Agfa Gevaert 10E70 photographic plates. An 8 sec exposure is approximately the maximum exposure time for which connexion currents can be easily controlled. Argon ion lasers are available with powers of 1·4W to accommodate large objects e.g. 4 ft diameter with exposures less than 8 sec.

The components under vibration must be mounted by heavy metal clamps or blocks to ensure that they are rigidly clamped when vibrated and that none of the vibration is transmitted in to the table to vibrate the optical components and destroy the hologram. Aero engine blades are clamped with the appropriate root fixing clamp and then mounted in a heavy vice. Alternatively they are cast into blocks of a low melting point metal. Correct clamping is extremely important as this governs both the 'Q' of the system and the mode shapes. Discs are clamped at their centre as in the majority of vibrational modes is a nodal region, as seen in Fig. 4. More irregular shaped components are clamped in the expected region of a node.

Routine Application of Holography to Vibration

The standard system on a flat bench is more complex and requires a higher degree of skill to operate than is needed for routine use.

It is possible to simplify the apparatus and tailor it to fit particular types of components hence reducing the overall cost and required operator skill. A holographic unit capable of being operated on a routine basis has been developed by the author.

The holographic system is designed to be operated with a 20 mW He-Ne laser and to obtain holograms of objects up to 24 in diameter. This is completely self-contained as it records the holograms and photographs the resultant reconstructions and can be completely blocked out for use in the normal laboratory environment. The whole unit is shown in Fig. 9 and is a box-like structure with the laser in the top plate and the object placed flat on the base plate. The position of the laser in the unit is shown in Fig. 10a. The thin pencil-like laser beam is reflected down through the top plate by mirror M_1 and then reflected parallel to the underside of the plate by mirror M_2. The beam is then divided into two portions, one containing 90 per cent of the total laser light and the other containing 10 per cent. This is accomplished using a dielectrically layered glass plate, called a beamsplitter, placed before the lenses which spread the laser light on to the object and hologram plate respectively.

The beam reflected from the beamsplitter containing 90 per cent of the laser light is projected vertically down towards the centre of the base by mirror M_3 to illuminate the object situated at the centre of the base. This beam is expanded from the 0·050 in diameter pencil beam to a diameter sufficient to cover the object by a microscope objective lens. The lens focuses the light through a small aperture which acts as a stop to filter out stray light components, such as light diffracted by dust particles on the lens, and spurious reflected components allowing only the direct laser beam contribution through.

The diameter of the illuminating beams, at the object, is dependent upon the semi-angle of the divergent conical beam, which in time is a function of the focal length of the lens, and the distance of the lens from the object. To accommodate the varying size of objects from 1 in to 24 in diameter the illuminating beam is adjusted by changing the lens and altering the distance between the lens and the object. The latter is achieved by mounting the lens on to an optical bench saddle which is adjustable along a vertical rail system.

The 10 per cent beam transmitted by the beamsplitter is

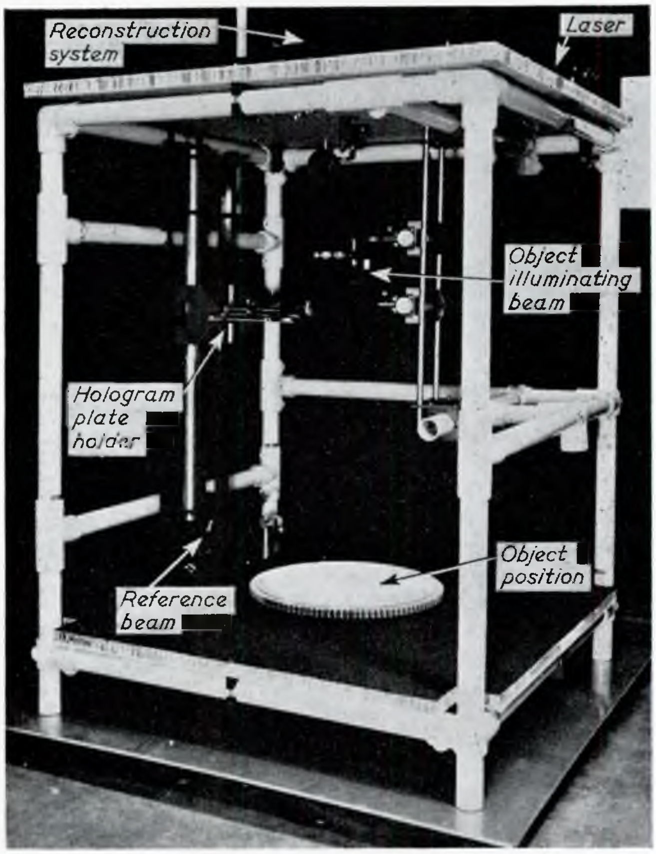


FIG. 9—*A complete hologram unit for routine analysis*

reflected down on to the base by mirror M_1 and firstly reflected on to a lens by mirror M_5 to expand the beam directly on to the hologram plate to form the reference beam.

The hologram plate is adjustable on a rail system within the limited positions HA to HB, shown in Fig. 10b to keep the hologram interference pattern within the resolution of the photographic plate. The photographic plate used for the system is Agfa Gevaert 10670 size 5 in × 4 in with a resolution of 2 800 cycles/mm and since the unit is intended for continuous use the plates are used economically by only exposing a quarter of the plate in any one test.

The whole unit is made extremely rigid and is anti-vibration mounted from the floor by means of a pneumatic system of inflated rubber inner tubes which provide adequate vibration damping for operation in an inspection by environment.

After exposure of the photographic plate for the hologram the plate is removed and processed in the usual manner and rapidly dried with alcohol such that it can be ready for reconstructing the image within a few minutes[8].

The processed hologram can be inspected for immediate observation or alternatively the information can be converted into a permanent photographic record by the system shown in Fig. 10c that may be used as required for future reference.

CONCLUSIONS

Holography has been shown to be an efficient tool for vibration measurement on engineering components and can be developed into a system capable of routine applications. It is more accurate and versatile than the Chladni sand pattern or contact probe and provides vibrational amplitude information that has not been previously possible. The measurements can be taken quickly and permanently stored in the form of photographs. The significance of the amplitude information and the ease of obtaining it has placed holography in the forefront as an engineering diagnostic tool and as the technique becomes more streamlined and engineered to the users requirements its use and that of similar optical techniques will increase.

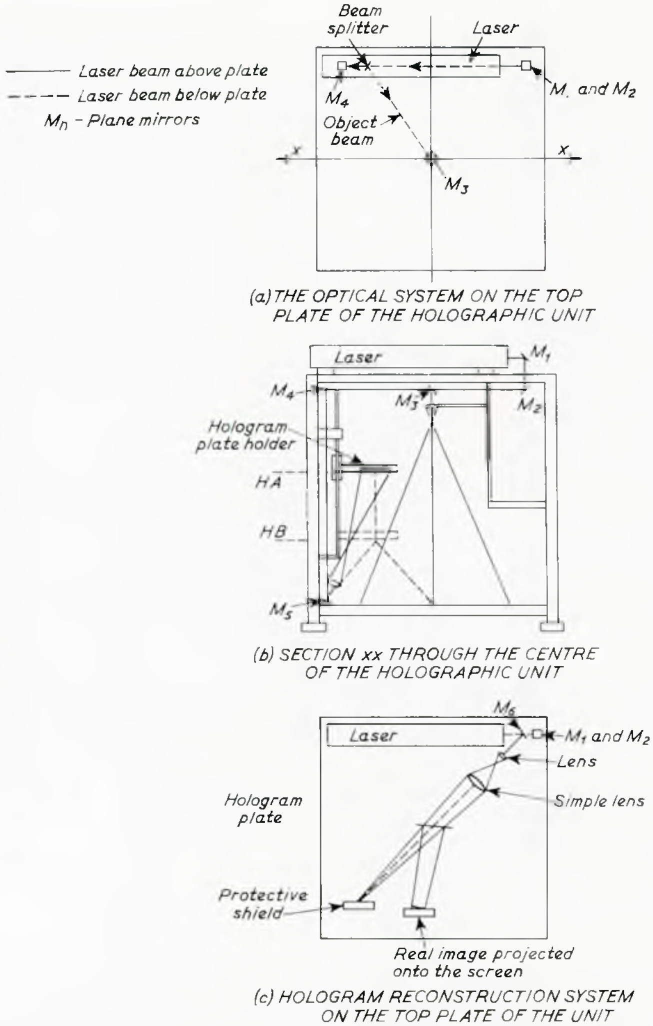


FIG. 10—*Holographic system designed to be operated with a 20mW He-Ne laser*

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Discussion

PROFESSOR W. CARNEGIE Associate Member of Council, I.Mar.E., said that, apart from the problems that he had already mentioned at the meeting, he and his colleagues also had a big programme of research being undertaken at the University of Surrey into the vibration of turbine blading. The method of examining vibrations by holography interested him greatly.

He wondered whether the author saw the possibility of that method being used on a rotating system, because if one had a rotating system with a disc carrying a series of blades, there was evidence to show that the behaviour of the blade on the disc under rotation was quite different from that when it was stationary. Yet at the present stage there was, to his knowledge, no known method of being able to examine the nodal patterns of a blade rotating or mounted on a rotating disc. That was something he felt was badly needed. Could the author suggest whether it was possible for that technique to be applied in that way?

The CHAIRMAN DR. A. J. JOHNSON asked whether one could use closed-circuit television video instead of photographic plates in order to obtain real time results. The processes involving the development of photographic plates appeared to be excessively time consuming.

Also, he had not quite appreciated how the normal out-of-balance displacement of the specimen was separated from the in-balance displacement. He presumed that there must be some technique for being able to separate the two parts.

That brought him to the splendid pictures in Fig. 6 which looked rather like space rockets. He was not clear from those how the nodal positions could be fixed and why one part of them could apparently be rotating at a much greater amplitude than the top and bottom parts. Perhaps Mr. Hockley could elucidate that for him.

He had also intended to raise the question that Professor Carnegie had posed, regarding the possibility of using the technique on running machinery, but he imagined that the kind of limitations Mr. Hockley was restricted to on dimensional tolerances were absolutely critical and would be the main difficulty.

Author's Reply

Mr. Hockley said that with respect to Professor Carnegie's question concerning the application of holography to visualize the vibration mode patterns of rotating turbine discs and blades, this was an important engineering application and would be extremely valuable to both the aerospace and marine industries. However, a viable engineering system capable of this application had not been developed at present.

There were two approaches to this problem, either by the use of continuous wave, gas lasers in conjunction with a stroboscopic illuminating system, or by pulsed lasers, such as ruby.

With a continuous wave laser, a stroboscopic hologram interferometer was utilized with the rotating component illuminated when it was in one particular position during each or every *n*th rotation. Because of the inherent sensitivity of holography the object had to be accurately superimposed on its image so that the stroboscopic pulse required accurate timing and its duration was an extremely short portion of the rotation. Hence the percentage of the effective use of the light compared to the time the light was shut off was very small and resulted in low average light levels and long exposure times which radically increased the overall stability requirements.

The pulsed laser however, concentrated sufficient energy to expose a hologram plate in to a single pulse of the order of 20 ns duration, greatly reducing the stability requirements. This technique used a double laser pulse, where the laser was pulsed while the object was in a particular part of its rotation and then pulsed again at the same part of the next or *n*th rotation to form a double exposed hologram to show the vibration pattern. The accuracy of the pulse separation was required to a few ns which made the triggering system complex, but the overall accuracy and stability requirements were less than those of the continuous wave laser techniques which required synchronization over several minutes.

Research workers both in the United Kingdom and United States had been studying the former technique which was proving difficult. The double pulsed method appeared to be attractive although it was dependent upon the availability of a double pulsed laser of suitable performance.

Referring to Dr. Johnson's questions on the possibility of a real time method of vibration visualization using a television recording mechanism rather than a photographic plate, the feasibility of such a technique had been demonstrated for small objects by Professor J. N. Butters and J. A. Leendertz, of Loughborough University of Technology. In their system, the object was illuminated with laser light and observed by means of a television camera with a very small lens aperture, f120. This produced an image that had a speckled appearance in which the speckles were large enough to be resolved by the television tube. A reference beam was also projected on to the television tube to produce an interference effect with the speckled image, as in a hologram. When the object was vibrated, the observer saw an averaged interference pattern, as in the real time holographic interferometer which showed the nodal pattern and some amplitude fringes. The technique was in an early stage of development but appeared to be very attractive.

It was possible to analyse an out-of-balance vibration by stroboscopic holography where the object was illuminated at any one point in the vibration cycle and compared with the image of the stationary of component. Thus it was possible to obtain a separate measurement of the vibrational amplitude either side of the static position. By comparing the two resultant fringe patterns, the differences indicated the out-of-balance amplitude. The stroboscopic technique had to be used, as it was not possible to use the simple time averaged fringe method which only measured the total excursion between the two extreme positions of vibration.

With reference to the question concerning the examples of the torsional vibration of a shaft assembly (Fig. 6), the assembly consisted of three separate concentric shafts which, for this particular test, were rigidly fixed together at the base of the assembly, at the bottom of the photographs. The torsional couple used to vibrate the assembly was applied to the inner shaft; a clip connecting the vibration to the shaft could be seen at the top of Fig. 6.

It must be noted that although the fringe pattern on the shaft that was under purely rotational motion may have appeared to be similar to vibration patterns of other components such as a blade, its analysis was quite different. A hologram system measured the vibration deflexion vector projected along the angular bisector of the angle of the light illuminating the object and the direction of observation, so that if the deflexion of the object was at right angles to this bisector, the system would not detect the movement, i.e., zero sensitivity. The bright fringe along the centre of the object was a result of the phenomenon and this would always appear, irrespective of the vibrational amplitude. The vibration amplitude information was obtained by the fringe spacings projected on to a plane at right angles to the direction of observation, i.e., the image plane, which was inversely proportional to the amplitude of rotation.

Therefore at the resonance at 1·3 kHz (Fig. 6a), the outer shaft had a very small amplitude, the middle shaft a slightly greater amplitude and the inner shaft the greatest amplitude, whereas in the resonance at 5·3 kHz, the inner and outer shafts had approximately equal amplitudes and the middle shaft had a much greater amplitude. The relative amplitudes of the three shafts at each resonance was a function of the "Q" of each shaft together with the coupling effects between the shafts.

WHIRLING OF LINE SHAFTING

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The long standing problem of whirling due to first order (unbalance) and propeller blade order excitations is discussed. The source of the blade-order harmonic components and the close correlation with alignments and the resulting bearing loadings are traced. The possible damages caused by the phenomenon are illustrated by examples. The problems associated with frequency calculations are outlined and a computer program is described.

INTRODUCTION

In its broadest sense the term whirling is used to describe the dynamic response of a shaft running in its bearings. It was probably the first form of shaft vibration to beset engineers with problems, the earliest paper on the subject dating back to 1869, Rankine[1].

Unfortunately some early investigators were misled in their appraisal of the phenomenon and their confused thought has succeeded in misleading many later generations. These pioneers treated whirling as something apart from a normal forced vibration problem. It was considered to be a state of instability reached at certain rotative speeds which were designated "critical" or "whirling" speeds. The fact that the apparent "instability" could be proved to occur at speeds equivalent to the natural frequency of transverse vibration of the shafting was regarded as confirmation of their attitude rather than as a pointer to forced vibration.

It was not until 50 years after Rankine's paper that Jeffcott[2] in 1919, interpreted the phenomenon as we know it today. He brought to the forefront the fact that whirling is a vibration arising from the excitation of natural frequencies of a shaft by a periodic source.

Whirling is thus a problem of forced vibration and as such two factors emerge immediately. Firstly, the resultant vibration will depend, amongst other things, upon the severity of the forcing—a fact used to advantage in dynamic balancing and, in particular, modal balancing of rotors. Secondly, the former definition of instability becomes one of resonance which can appear at any shaft speed depending upon the cyclic frequency per shaft revolution, i.e. the order of the excitation.

The problems confronting the early engineers had occurred with shafting subject to forced vibration due to unbalance. As unbalance manifests itself as a first order excitation this explains why at that time "instability" arose at speeds equal to the natural frequency. Thus the terminology developed was descriptive of a particular form of vibration and not of the phenomenon in general.

The terminology has persisted and, in some respects, can be considered adequate since, in general, problems due to unbalance will remain important. However in both the aero and marine engineering fields, a form of shaft whirl can occur which is caused by propeller excitation, not unbalance. The excitation arises as the result of the propeller working in a non-uniform flow and the frequency of the vibration is of blade order, i.e. the number of blades × shaft rev/min.

In marine shafting the tailshaft is most liable to suffer excessive vibration through such propeller excitation. For this reason the phenomenon has often been referred to as tailshaft whirl although, in the absence of a general terminology, other expressions including transverse vibrations, lateral vibrations and blade-order whirl have come into common usage. This phenomenon is covered in this paper and the term lateral vibration will be used.

PROPELLER EXCITATION

Since the problem of lateral vibration begins with the propeller it is worth while to examine its function.

If a propeller was a perfectly symmetrical body rotating in a homogeneous flow it would experience a constant thrust and torque. In practice it works in a non-uniform wake where the inflow velocity of the water to the propeller varies over the disc in both magnitude and direction. The non-uniformity of the wake is mainly a function of hull size and design—particularly of the afterbody—the location and diameter of the propeller and the design and positioning of appendages. The problem is more acute, generally, for single-screw vessels than for twin-screw vessels.

Each propeller blade experiences periodic forces of complex wave form. However, when the wave-form is harmonically analysed it will be apparent that some wake components act on the blades to produce a bending moment about the propeller i.e. the thrust forces on the propeller as a whole are unbalanced. Such bending moments produce the excitation for lateral vibration of the shafting.

For a propeller with n blades these bending moments will emanate from the $(kn \pm 1)$ wake components, k being any integer. The most significant components are those where $k = 1$. It can be shown that the $(n-1)$ component gives rise to a bending moment, of frequency $n \times$ propeller speed, relative to the propeller, which rotates in the same direction as the propeller. The $(n+1)$ component will give rise to a bending moment of the same frequency but rotating in the opposite sense to the propeller. The two forms of lateral vibration which can ensue are termed forward whirl and counter, reverse or backward whirl respectively.

To an observer on a ship measuring the relative motion of the shaft, both forms of whirl produce a vibration of the nth order. However, strain variation measurements would reveal the $(n-1)$ and/or $(n+1)$ orders.

By analysing measurements of wake velocities over the propeller disc from models certain generalizations can be made

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as to the relative merits of propellers with different blade configurations. Such measurements reveal that a ship's wake is, to a large extent, symmetrical about a mid-line resulting in the even order components predominating over those of odd order. Thus propellers with an odd number of blades, taking their bending moment variation from the even order components, give greater excitation than propellers with an even number of blades. The reverse applies for torque and thrust variations as shown in the fuller treatise by Archer[3].

Whilst it is true, in general, that 5-bladed propellers are greater offenders with regard to lateral vibration, 4-bladed propellers have also given rise to excessive vibration in certain cases.

Table 1, compiled using the designs of a large sample of ships currently building to Lloyd's Register Class, illustrates trends in propeller choice.

TABLE I—PROPELLER CHOICE FOR VESSELS BUILDING TO LLOYD'S REGISTER CLASS

Number of Propeller Blades	Percentage of Vessels Examined in each Power Range	
	0–20 000 hp	20 000–40 000 hp
3	10	0
4	86	26
5	4	37
6	0	37

Two surprising facts emerge. In the higher power range, which includes the fast container ships and VLCC's, choice is fairly evenly balanced between 4, 5 and 6-bladed propellers. However, below 20 000 hp the 4-bladed propeller predominates exclusively. Firm reasons cannot be given for this but the results in the higher power range are probably indicative of careful consideration being given to the vibration aspect. In the past many problems have resulted in vessels in the lower power range equipped with 5-bladed propellers.

PART I—LATERAL VIBRATION—ITS EFFECTS AND SOME CASE HISTORIES

In 1950, Panagopulos[4] propounded his theory for lateral vibration in relation to the tailshaft failures of liberty ships and focused the attention of marine engineers on the effect of the higher orders of propeller excitation. Although outmoded now for general types of vessels, the theory led to a number of investigations into the problem. Probably the best-known work in this sphere is that due to Jasper[5].

Jasper, being primarily concerned with naval shafting, considered propeller excitation of small significance. With naval shafting systems, the bearing spans are limited generally from considerations of shaft whirl—the source of excitation being unbalance. The fundamental frequency is sometimes not far removed from the maximum operating speed and thus propeller excitation will give rise to resonant speeds well down in the speed range with consequent low levels of excitation. To the authors' knowledge, the only case of lateral vibration occurring with naval shafting involved an intermediate shaft. The resonance occurred at a low shaft speed and the amplitudes of vibration were not of significance.

With merchant shafting systems and their more conservative bearing spans the problems due to excessive lateral vibration—almost exclusively of tailshafts—occur at the service speeds.

Since the time of Panagopulos and Jasper, numerous independent investigations, including many by Lloyd's Register of Shipping, have been conducted. However, there has remained a degree of unpredictability about lateral vibrations for two major reasons *viz* the effect of the dynamic response of the structural supporting system and the effect of changes in the shaft alignments.

EFFECT OF STRUCTURAL RESPONSE OF SUPPORTING SYSTEM

The behaviour of a shaft can be greatly influenced by the flexibility of the bearings and the adjacent hull structure and this is one area in which there remains scant knowledge. The effects of support are likely to be more pronounced in twin or multi-screw vessels where the shafting is supported in bossings and/or 'A' brackets. However some instances of coupled vibration of shafting and hull have been discovered in fast single-screw ships with fine hull form. Here the hull vibration has been confined to the after part mainly below the tunnel deck-head and reaching forward as far as the aft end of the deep tanks either side of the tunnel.

The importance of bearing flexibility in regard to lateral vibration is described more fully in Part 2.

EFFECT OF CHANGES IN SHAFT ALIGNMENT

At first sight the effect of changes in shaft alignment may seem of lesser significance than the effects of bearing and hull flexibility. However, in the past, had ship's shafting been adequately supported in its bearings at all times very few cases of lateral vibration would have arisen. The majority of problems have occurred particularly as a result of the forward stern tube bearing and, at times, also the aftermost plummer bearing becoming unloaded.

Whilst the shafting may take up a different attitude from the designed arrangement this unloading may also arise from the initial alignment, particularly with a conventional shaft arrangement as shown in Fig. 1a. This arrangement comprises a tailshaft supported by two lignum vitae stern tube bearings connected to intermediate shafting with two plummer bearings per shaft section. When all bearings are in a straight line the forward

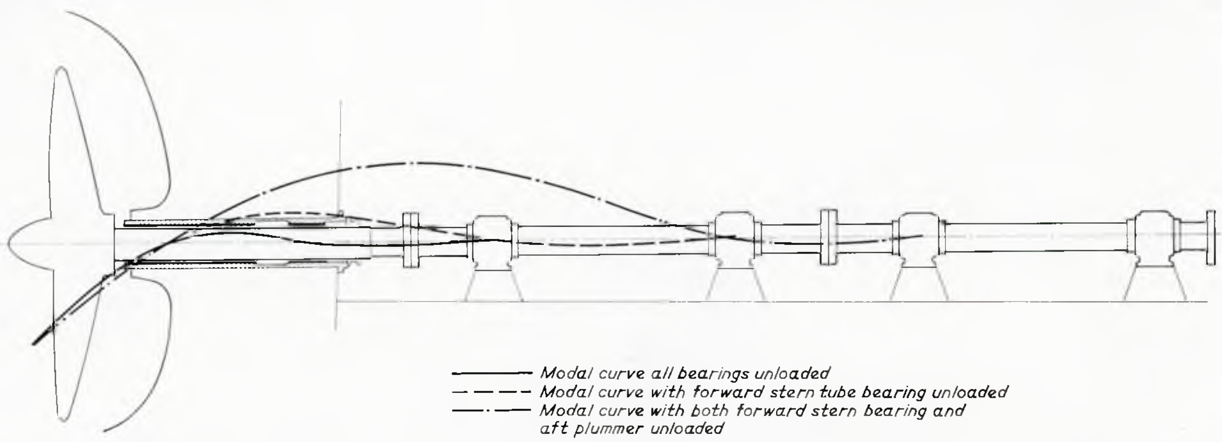

FIG. 1a—*Conventional shaft arrangement for aft-engined oil tanker and depicting modal curves for lateral vibration*

stern tube bearing will be unloaded when the shaft is at rest. The bending moment from the overhung propeller will accentuate this and also tend to offload the aftermost plummer bearing. The computed modal forms for resonant lateral vibration of the shaft considered supported (i) at all bearings (ii) with the forward stern tube bearing unloaded (iii) with the forward stern tube bearing and aftermost plummer bearing unloaded are shown. Fig. 1b illustrates the relevance of tailshaft support to the resonant shaft speed.

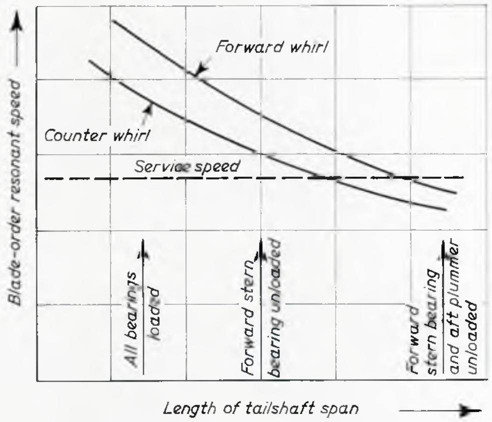

FIG. 1b—*Relationship between forward and counter whirl frequencies and tailshaft span length*

Many cases of tailshaft lateral vibration have arisen following either initial or subsequent relining of stern tube bushes—thus indicating incorrect alignment—but many vessels have developed excessive vibration after varying intervals of satisfactory operation. The basic reasons why a change in alignment will occur other than being brought about by working in a sea way are stern bearing weardown, eccentric thrust and ship deflexion.

With a lignum vitae or phenolic resin type bearing the relatively large amount of weardown has been one of the major reasons for excessive tailshaft vibration. The weardown in normal operation can cause a loss in load at the forward stern-tube bearing through the shaft "bedding in" and the effective support point moving forward. With white metal stern tube bearings the weardown is generally negligible by comparison so that the shafting is not subjected to significant changes of alignment which accords with their better performance with regard to lateral vibrations.

When a vessel is at loaded draught the propeller thrust will act above the shaft axis tending to load up the forward stern tube bearing. In a very light ballast condition with the propeller near the surface the thrust may act below the shaft axis tending to unload the forward stern tube bearing. In addition the increased non-uniformity of the wake in the ballast condition generally results in increased excitation.

As a ship is a flexible platform, differences in loading and disposition of cargo result in changes in bearing height and therefore in shaft alignment but, generally, the influence of ship deflexion has been of small significance for vessels with lignum vitae or phenolic resin type linings.

The current desire for shafting systems to possess adequate flexibility is in acknowledgement of the fact that changes in alignment will occur with varying operating conditions. In general, shafting systems have had too many bearings rendering them sensitive to the aforementioned effects.

Attempts to cure lateral vibration by adjusting the alignment are often thwarted because, where there are too many bearings, it is unlikely that any alignment recommended, following calculation, could be both achieved initially and maintained subsequently in service. Sometimes, success comes only after drastic changes to increase the flexibility are made to the shaft arrangement.

NATURE OF LATERAL VIBRATION

Measurements of lateral vibration amplitudes taken from an oil tanker fitted with a 5-bladed propeller are shown in Fig. 2a.

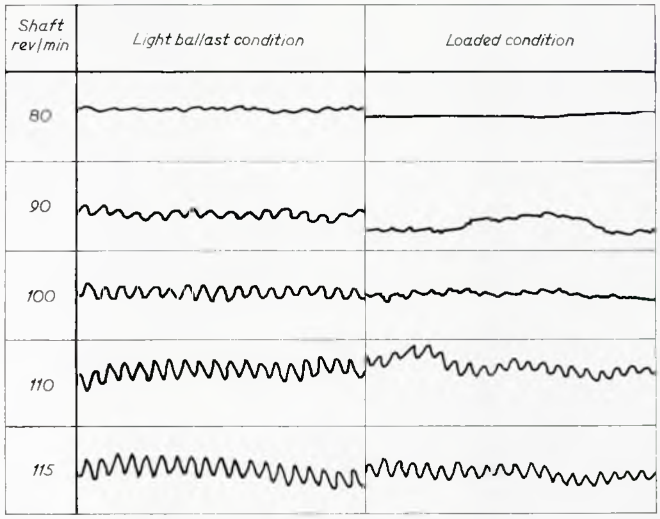

FIG. 2a—*Relative movement of tailshaft to stern tube—oil tanker with 5-bladed propeller—ballast and loaded condition*

The traces showing a very distinct propeller blade-order frequency vibration were obtained from a displacement transducer inserted at the mid-length of a sterntube and bearing onto the tailshaft. The stern tube had a forward and after bronze bush each being lined with lignum vitae. The two sets of traces represent measurements taken in the loaded and light ballast ship condition.

Plotting these measurements against shaft speed as in Fig. 2b indicates the flattened type of resonance curve so typical

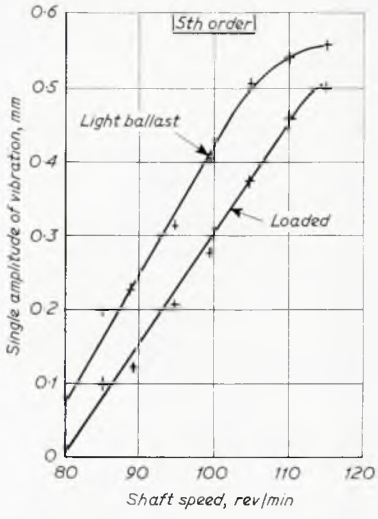

FIG. 2b—*Measurements from Fig. 2a plotted against shaft speed*

of that found in many vessels suffering from excessive tail shaft vibration. The main reason put forward for this phenomenon is that the tailshaft is vibrating in its clearance. It is not possible to judge whether the vibration is a result of a high level of forcing or a resonant condition in such a case.

Since lateral vibration generally involves a relative motion of the tailshaft in its bearings damage to both the rotating and structural elements of the stern gear can be sustained.

DAMAGE TO STERNGEAR

In addition to heavy weardown in lignum vitae or phenolic resin linings in a short period of time, a sign of excessive vibration is hammering of the bush linings, particularly at the aft end of the aft bush. On occasion the staves have been found forced aft towards the retaining ring. Fracture of the aft bush and of the stern tube itself can also occur.

Often the first signs of excessive tailshaft vibration developing are difficulty in maintaining proper water tightness at the stern gland and vibration of the aft plummer bearing. Damage to the aft plummer bearing usually takes the form of the bearing becoming loose on its seating and sometimes fracture of the seating itself.

With white metal stern tube bearings, failure can take place very quickly and it is not always possible to assess whether the cause is vibration or simply mal-alignment. However, the white metal in the aft bush has been hammered in some cases.

The shaft liner has been found eroded due to lateral vibration of a tailshaft in a lignum vitae stern tube. The erosion has been found to exist in way of both the aft and forward bush linings taking the form of longitudinal bands equally spaced around the liner either in-line with the propeller blades or between them. In many cases the erosion is so deep that water has penetrated to the shaft. Instances have also occurred with sand cast liners in which the core plugs have been shaken out and water has penetrated to the shaft in this way.

Both the leaded and non-leaded 88/10/2 gunmetal liners have suffered from erosion. Whilst a suitable material may be found highly resistant to erosion and compatible with the lignum vitae or phenolic resin, remembering some quite alarming wear rates of bearing material and liner have resulted from the use of aluminium bronze, it is preferable to remove the cause than change the liner material.

Less frequent than liner erosion but brought about in some cases by lateral vibration is the loosening of the propeller nut, fretting of the propeller on the cone, fracture of the tailshaft and fracture of the tailshaft coupling bolts.

EXAMPLES OF LATERAL VIBRATION

The majority of the investigations carried out by Lloyd's Register of Shipping have been on installations with lignum vitae or phenolic resin stern tube bearings. The following two cases typify the problems encountered.

Case 1

Two identical single screw dry-cargo vessels fitted with 5-bladed propellers and having the tailshaft supported in two stern tube bushes had suffered repeated and excessive weardown of the aft sterntube bearing material and tailshaft liner erosion.

Measurements were taken of the dynamic bending strains at the inboard end of each tailshaft together with the movement of each shaft relative to the stern gland in both the vertical and horizontal plane.

Vessel A required relining of the aft stern tube bush approximately once a year. Failures of the stern gland packing and vibration of the aftermost plummer bearing, culminating in a fracture of the seating, had also occurred. The first investigation carried out following relining with a new tailshaft, the previous one having been found cracked and the liner badly eroded, indicated negligible blade-order vibration.

Vessel B had required one relining of the aft bush and a tailshaft replacement due to liner erosion. The bending strain measured showed both 4th and 6th order variations consistent with large blade order excitation. Fig. 3 shows the relative movement of the shaft at blade frequency. At the next survey the aft stern tube bearing was found badly worn, the shaft fractured at the cone and the propeller nut loose. Fig. 4 shows the results obtained following relining of the aft bush and a complete realignment. This vessel now appears satisfactory.

The results of the two investigations indicate that whereas in Vessel A, when newly relined, the forward stern bearing was loaded, it was unloaded in Vessel B and lateral vibration occurred. The problems with Vessel A have continued and radical changes to the shafting arrangement have been carried out to achieve a satisfactory alignment to load the forward stern tube bearing.

Case 2

An oil tanker fitted with a 5-bladed propeller and having a

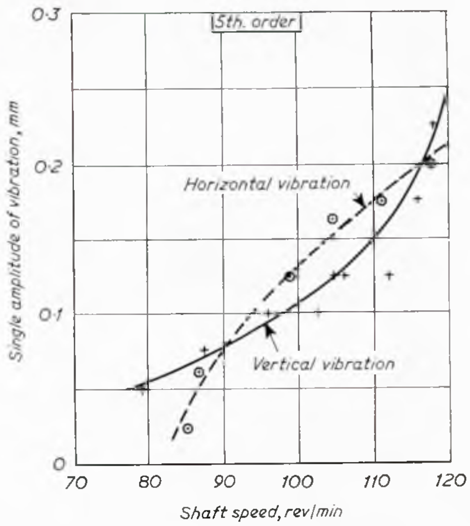

FIG. 3—*Case 1—dry cargo vessel (B)—blade order movement of tailshaft relative to stern gland—stern bearings worn*

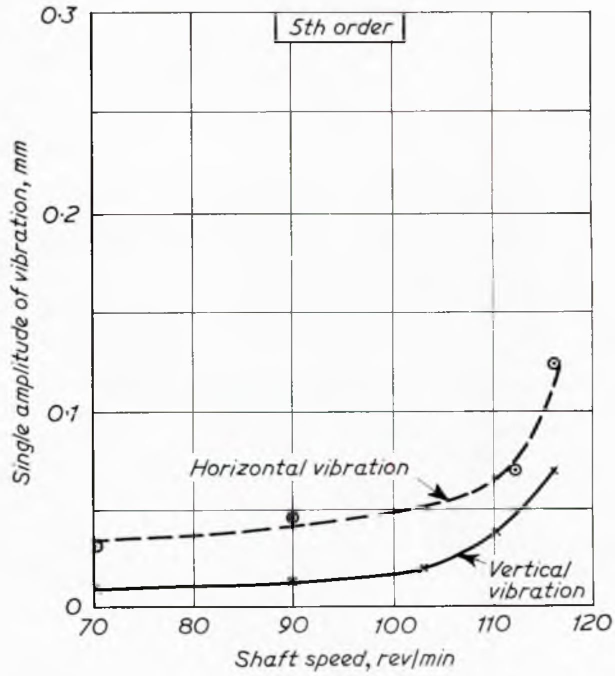

FIG. 4—*Case 1—dry cargo vessel (B)—blade order movement of tailshaft relative to stern gland—stern bearings new relined*

shaft arrangement as shown in Fig. 1, suffered from repeated failure of the stern gland packing and vibration of the aftermost plummer bearing. The conventional soft packing was being shredded and passed out between shaft and gland.

Measurements of the blade-order vertical movement of the tailshaft relative to the stern gland are shown in Fig. 5 for both conditions—with new packing and after 24 hours operation. Attempts have been made to prevent the development of lateral vibration by changes in the shaft arrangement to load the forward stern tube bearing adequately.

CURRENT DESIGN

Today the white metal stern tube bearing predominates and there has been an apparent decrease in problems due to lateral vibrations. No doubt this can be attributed to the restricted clearances and smaller weardown.

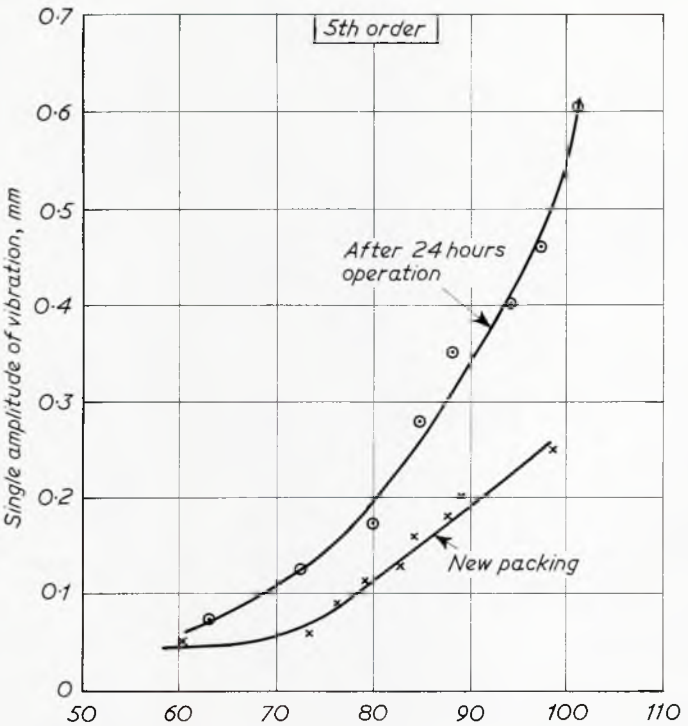

FIG. 5—*Case 2—oil tanker—blade order movement of tailshaft relative to stern gland for new packing and after 24 h operation*

Where a white metal forward stern tube bearing is used it is still important to ensure that it is loaded at all times. Some owners have dispensed with this bearing but, in this case, due consideration has to be given to the construction of the aftermost plummer bearing pedestal, particularly in the athwart ship direction. The designer should be wary of the possibility of lateral vibration resonance in view of the increased bearing spans.

With container ships the fine hull lines necessitate a long tailshaft span and the resonant speeds may approach the service speed. With a 6-bladed propeller, the resonance may lie within the operating speed range.

If the larger powers revive the interest in contra-rotating propellers these will have some interesting aspects with regard to lateral vibrations. Model tests have indicated the possibilities of combining a 4-bladed forward propeller with a 5-bladed aft propeller. It is claimed the forward propeller will smooth out the irregularities of flow to the aft propeller so that the excitation for the 5-bladed propeller could be reduced to no more than that for a conventional 4-bladed propeller.

There may be coupling between the frequencies of the two shafts if the speeds of rotation are the same and the bearings of the inner shaft are not in line with those of the outer shaft. The complete system requires investigation, noting that since the shafts rotate in opposite directions forward precession of one propeller can occur with the backward precession of the other. What vibration will result from the use of propellers with different numbers of blades can only be decided after extensive investigations.

PART II—FREQUENCY CALCULATIONS AND MODAL SHAPES

The work involved in calculating the natural frequencies for the line shafting is complicated and the earlier works used approximate methods, generally considering the aft end only. The problem was treated as that of the case of a single thin disc, the propeller, on a "massless" shaft and combined the effect of the shaft mass either by rough approximations added to the disc or by such formulae as Dunkerley's[6]. The design charts, Fig. 6, based on Jasper's method and appropriate to a 5-bladed propeller, may be used for guidance as to the necessity for more detailed investigation.

With the advent of computers the whole length of the line shafting may be considered. This also allows the higher critical speeds to be determined as well as the fundamental to which the former methods generally limited the investigations.

One fallacious notion that should be disposed of at this stage is the effect of gravity. Because elementary methods, such as the Rayleigh energy method, use the gravity form of the bent shaft as a first approximation to the fundamental whirling shape there has arisen an idea that whirling is directly influenced by gravity. In fact, whirling is a form of bending vibrations occurring about the bent form of the shaft, in its bearings, whilst rotating in the steady state.

The mathematical model deals with the shaft loaded at any point by the mass multiplied by the shaft deflexion at that point. From elementary beam theory, the normal differential equation of the deflexion curve for a shaft, of variable section, whirling with its central axis in a bowed shape is:

$$\frac{d^2}{dx^2}\left(EI\,\frac{d^2y}{dx^2}\right) = u\,\omega^2 y \qquad (1)$$

where x = distance along the shaft
y = deflexion of shaft
E = modulus of elasticity
I = diametral moment of inertia of shaft cross-section
u = mass per unit length
ω = frequency of vibration rads/sec
all being in consistent units

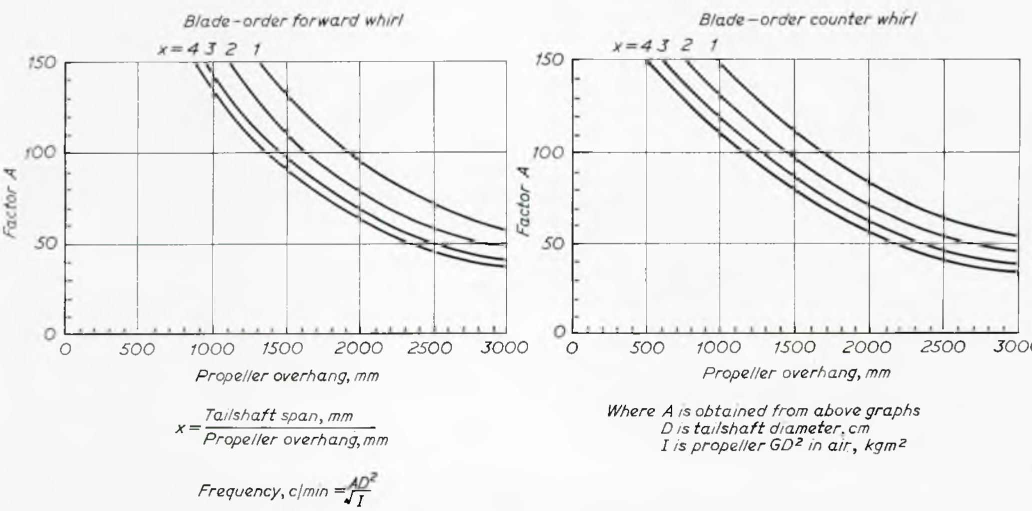

FIG. 6—*Guide charts for frequency calculations and modal shapes*

The slope of the deflexion curve depends not only on the rotation of the cross-sections of the shaft but also on shear. The shear will be important if the cross-sectional dimensions are not small in comparison with the length. This may be the case for vibrations at higher frequencies when the shaft is subdivided by modal sections into comparatively short portions. The variable angle of rotation of the cross-sections is equal to the slope of the deflexion curve and gives rise to a moment of the inertia forces termed the "rotatory inertia". The inclusion of these effects for the shaft leads to a complicated differential equation. Fortunately, for the whirling of line shafting the scantlings and frequencies do not warrant their inclusion.

Integrating equation (1) twice gives the well-known form:

$$EI\frac{d^2y}{dx^2} = M \tag{2}$$

where M = bending moment

equation (1) may be rewritten as

$$\frac{d^2M}{dx^2} = u\,\omega^2 y \tag{3}$$

Equation (1) is a fourth order differential equation and there are four boundary conditions to be satisfied. Any speed satisfying this equation and also the four boundary conditions is a natural frequency of the system. A solution by constructing the bending moment and deflexion diagrams using a step-by-step integration process can be found in conjunction with equations (2) and (3).

In general, these diagrams can be constructed to satisfy three of the boundary conditions. By plotting the discrepancy in the fourth boundary condition as a function of the assumed speed and noting where this becomes zero, the natural frequency can be determined. In plotting such a curve it is necessary, of course, to select some arbitrary boundary condition and hold it constant.

A numerical method, analogous to the well-known Holzer method for torsional vibrations may be used for calculating whirling frequencies. Four integrations are involved instead of the two for torsional vibrations and additional complications arise in dealing with the boundary conditions. Such a method, described by Prohl[7], is outlined in the Appendix.

THE PROPELLER AS A DISC

The whirling of a disc on the shaft involves a linear and a tilting oscillation giving rise to four critical speeds. Two of these will occur when the centre of the shaft oscillates in the direction of shaft rotation (commonly called forward whirl) and two of backward precession (reverse whirl).

For a disc, such as a propeller, where there is an harmonic order of excitation and the shaft speed is not the same as the natural frequency of whirling, the couple tending to centralize the disc on the shaft, commonly called the "gyroscopic couple", as shown in the Appendix, is given by:

$$M_g = -\left(I_d \pm \frac{I_p}{n}\right)\omega^2\varphi \tag{4}$$

where I_p = polar mass moment of inertia

I_d = mass moment of inertia about a diameter

n = order number of the vibration = ω/Ω

ω = the angular velocity of the vibration

Ω = angular velocity of the shaft

φ = slope of the shaft at the disc position

Since the propeller is overhung the slope of the shaft at the equivalent disc may be significant, particularly in the fundamental mode of vibration.

Entrained Water

The water being churned round by the propeller increases the effective mass and moments of inertia of the propeller for all types of vibration. For a long time it has been the practice to add a fixed percentage of the moment of inertia of the dry propeller in torsional vibration calculations to simulate the effect of entrained water. The valid argument against this procedure is that for two propellers of identical shape but manufactured in, say, manganese bronze and aluminium bronze the same quantity of water will be displaced. However, the aluminium bronze propeller, being lighter, will be allotted a smaller entrained water allowance. The true amount of entrained water will depend upon the effective radius of gyration of the immersed propeller compared with that of the dry propeller.

Despite the fact that many investigators have produced formulae for entrained water, some of which give very high values, correlation between calculations and measurements have indicated the need to add only some 25 to 30 per cent to the polar moment of inertia in torsional vibration calculations. In view of the great use of lower density aluminium bronze propellers today, it appears sufficient to use the top of this range in lateral vibration calculations.

Assuming the fore and aft motions of the blades will be similar to that for axial vibrations of the shafting, it appears reasonable to use the normal entrained water allowance of 60 per cent on mass in those investigations for the diametral moment of inertia in whirling calculations. It is agreed that doubts regarding the true stiffness of the thrust block make an accurate assessment based on measured axial frequencies difficult. However, for "blade order" vibrations a slight discrepancy, in the absence of reliable test results, is of secondary importance.

For the propeller, the mass is the most important parameter in the calculation of the lateral vibrations of the line shafting. Unfortunately there is no value for the entrained water allowance which can be determined by analogy with other types of vibration. However, since the motion of the propeller mass is in a plane perpendicular to the shaft axis and assuming the movement of the blades to be similar to that for torsional vibrations it would seem reasonable to increase the mass of present day dry propellers by some 15 per cent.

Bearings and Points of Support

Ideally all bearings should be downward loaded. Panagopulos assumed there was no support at the forward sterntube and, as pointed out in the first part of the paper, due to the short span between the two stern tube bearings it is often difficult to obtain a downward loading with a straight alignment. It is worthwhile noting that some shafts with appreciable negative loadings at the forward sterntube have run well but where the natural frequency is close to the running range a degree of instability could result. Modern methods of alignment can overcome this and it is recommended that the alignment and whirling investigations be carried out early in the design stage.

With long bearings it is always difficult to estimate the true point of support. With the overhung propeller its distance from the effective point of support in the aft sternbush has a considerable effect on the natural frequency. Examination of the wear conditions of lignum vitae bearings of a length four times the shaft diameter would indicate an initial point of support approximately one diameter from the aft-end. However, since the effective point of support moves forward as the shaft "beds in" it is preferable to calculate for the support at a quarter to a third of the length from the aft end according to whether the estimated critical speed occurs below or above the service speed.

For the shorter white metal aft sterntube bearings the effective point of support should be considered to be at a third to a half of the length from the aft end. The other bearings in the line shafting are generally about one shaft diameter in length and it is sufficient to take the point of support at the mid-point.

FLEXIBILITIES AT THE SUPPORTS

The natural frequency of the line shafting is also dependent on the flexibility of the support bearings. Except where the length between bearings may result in local vibrations for a particular span, the relevance of flexibility decreases the further forward the bearing. In general, it is sufficient to consider only the aftermost three bearings within the ship as flexible. In any case, if the natural frequency of the bearing support is at least $3n\times$service speed, where n is the number of blades, it can be considered rigid.

Without a comprehensive investigation carried out by the 'finite element" method it is difficult to estimate the proportion of the surrounding structure which contributes to the general flexibility. Since the dynamic movements may differ from those calculated by statical considerations the justification for employing such a complicated procedure will depend upon the proximity of the critical speed to the service speed.

In general, for a single screw vessel, even for the lightest structure, the bearing stiffness will exceed 50 000 kg/mm (1250 tons/in).

For twin screw vessels, the flexibility of the bracket is very important. The influence coefficients can be calculated by the normal methods but there will also be an effect due to the bending moment acting on the side plates of the ship. Where it is desired to achieve conditions for the propeller as near open water as possible, as in some high speed vessels, the flexibility of the ship may be commensurate with that of the bracket. Although the effect of the tube shaft mass may be neglected, the flexibility it imparts to the bearings within it needs consideration.

When very flexible bearings are incorporated into the system the modal shape based on the straight line may not indicate the modes with which many engineers are familiar. As Fig. 7 shows, the base line has to be altered to pass through the equilibrium positions of the loaded bearings.

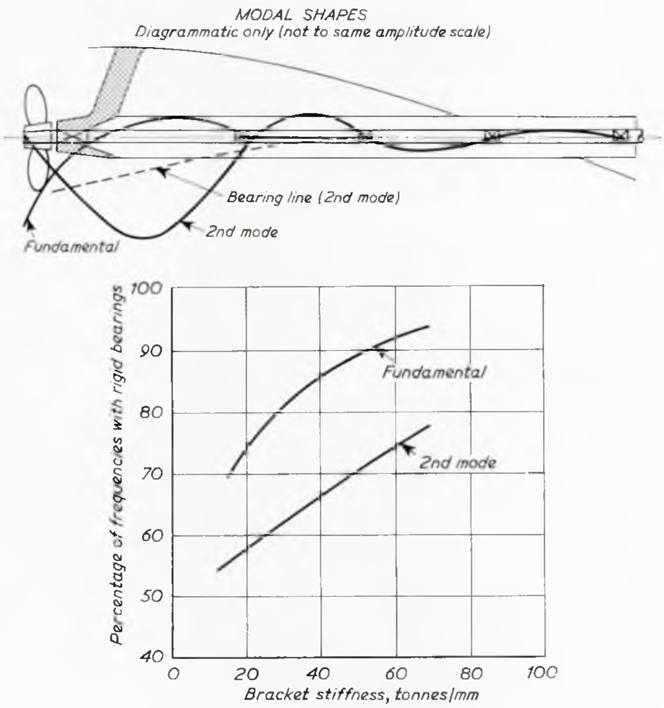

FIG. 7—*Twin-screw vessel—effect of A—bracket stiffness*

GENERAL REMARKS

The resonance for whirling particularly for the first order, is sharply tuned, in general, but it should be noted all shafts are subject to vibration in the presence of the appropriate excitation and it is not necessary to be at resonance to suffer excessive vibration.

The accuracy of the calculation will depend upon the assumptions made. In former times, where the critical speed was normally calculated to be above the service speed, a margin of 25 per cent between the calculated critical speed and the running range was considered safe. With the more sophisticated methods of today, this margin may be reduced to say, 15 per cent remembering it still has to cater for the accuracy of the assumptions and the width of the resonance band which is dependent upon the magnitude of the excitation and the damping in the system.

COMPUTER PROGRAM

A program for the calculation of lateral vibration characteristics and the associated whirling speeds of shafting systems has been compiled by Lloyd's Register of Shipping. The natural frequencies and modal shapes, for both forward and reverse whirl, due to either first order or "blade order" excitations are computed.

The essential features include the ability to deal in detail with shafting having variable cross-section and supported at many points each of which may be considered structurally as rigid or linearly flexible.

Provision is made for the attachment of concentrated mass and the application of inertia and gyroscopic bending effects at any point in the system, in addition to accounting for the distributed mass of the shafting.

The mass-elastic system is considered to consist of a number of discrete masses connected by lengths of uniformly elastic massless shaft. The distributed mass of the shaft is disposed at the mass stations, where attached concentrated masses together with any bending moments arising from inertia and gyroscopic effects may also be applied. This finite element treatment simulates the flexural properties of the system in terms of "thin rod" bending theory and the calculation procedure adopted is broadly in line with the method described in the Appendix.

The frequency-dependent relationships for dynamic bending are set up to allow the calculation procedure to pass along the system continuously element-by-element from one end to the other. The associated variable functions are suitably modified as each point of support is encountered.

The problem is examined over a prescribed range of exciting frequencies. For each value of frequency, a solution is obtained at the end of the system which satisfies the boundary conditions and is expressed in terms of the starting condition variables. At the same time a residual function is evaluated and natural frequencies for the whole system are identified by the annihilation of this residual.

For each natural frequency so determined, the corresponding modal deflexions are calculated throughout the system, together with the relative shear force, bending moment and slope distributions. The print-out gives these values at each mass station.

The distribution of shaft mass is approximated and thus the accuracy of the results increases with increasing numbers of sections. Where a long uniform span of shafting without externally applied loads is involved this can be subdivided into any number of sections within the computer. Thus it is necessary only to insert, in the input data, particulars of the complete section and the number of subdivisions required.

Where large disc-like masses, such as the propeller form part of the dynamic system, the associated gyroscopic bending moments are related both to the harmonic order of excitation and its direction of rotation. The allowances for entrained water on the propeller are variable input data allowing the designer to use his own experience regarding mass, diametral moment of inertia and polar moment of inertia.

The boundary conditions at each extreme end of the shafting system can be arranged to suit one of the possibilities including fully encastre, simply supported, whether linearly flexible or rigid, or completely free and unsupported. Each intermediate bearing support may be treated as either completely rigid or as having a linear radial spring stiffness.

ACKNOWLEDGEMENTS

The authors thank the committee of Lloyd's Register of Shipping and the technical director, Mr. B. Hildrew for permission to publish this paper.

They are specially indebted to Mr. D. H. L. Inns for the preparation of the computer program and to their colleagues in the technical investigation department for their advice in the preparation of the paper.

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Appendix

Nomenclature

E = modulus of elasticity
H = angular momentum
I = diametral moment of inertia of shaft cross-section
I_d = mass moment of inertia of disc about a diameter
I_p = polar mass moment of inertia of disc
k = bearing stiffness factor
l = length of shaft section
m = mass of equivalent disc
M = bending moment
M_g = gyroscopic couple of disc
n = $\frac{\omega}{\Omega}$ = order of excitation
t = time
V = shearing force
x = distance along shaft
y = deflexion of shaft
θ = slope of shaft
φ = angle
ω = frequency of vibration, radians per sec.
Ω = rotational speed, radians per sec.

The actual system is transformed into an idealized form of a series of discs connected by sections of elastic but massless shaft. The mass of the discs and their spacing is chosen so as to approximate the mass distribution of the actual system. The bending flexibility of the connecting sections is taken to correspond to the actual flexibility of the system.

Where the disc has significant mass moments of inertia the effect of these need to be considered. For the disc shown in Fig. 8, let the centre of gravity whirl in a path of radius y at a speed ω with the axis of the disc making a constant angle φ with the axis of rotation. The moment, Mg, which must act on the disc about the centre of gravity to sustain the prescribed motion is equal to the time rate of change of the resultant angular momentum.

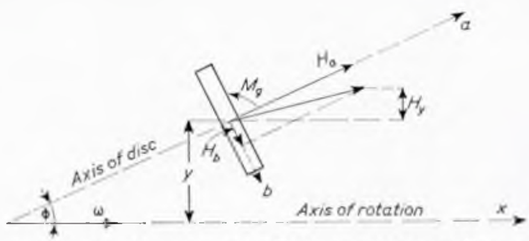


Fig. 8—*Gyroscopic effect on rotating disc*

The components of angular momentum about axis a and b of the disc are given by:

$$H_a = I_p \omega \cos \varphi \qquad H_b = I_d \omega \sin \varphi \tag{1}$$

The component of angular momentum normal to the axis of rotation is given by:

$$H_y = H_a \sin \varphi - H_b \cos \varphi = (I_p - I_d)\, \omega \sin\varphi \cos \varphi \tag{2}$$

Then for small angles of φ:

$$M_g = \frac{dH}{dt} = H_y \omega = (I_p - I_d)\, \omega^2 \varphi \tag{3}$$

For a propeller, with its harmonic orders of excitation and where the shaft speed is not the same as the natural frequency of whirling, the moment for forward and reverse whirl is given by:

$$M_g = -\left(I_d \pm \frac{I_p}{n}\right) \omega^2 \varphi \tag{4}$$

The moment which the propeller exerts on the shaft will be equal in magnitude but opposite in sign.

Fig. 9 shows an idealized system together with the shearing force, bending moment and deflexion diagrams. Since the shaft sections are massless the shearing force is constant between any two masses and a finite range in shearing force occurs at each mass.

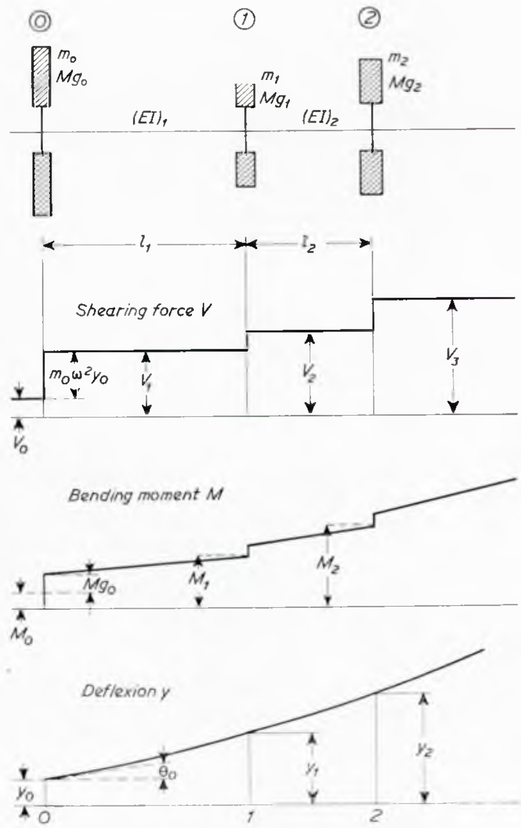


Fig. 9—*Idealized system*

Thus:

$$\frac{dM}{dx} = V \tag{5}$$

$$\Delta V = m\omega^2 y \tag{6}$$

Assume that V_0, M_0, θ_0, and y_0 at the left-hand end of the system are known. Let:

$$M^g_n = M_n + M_{gn}$$

Then an infinitesimal distance to the left of point 1:

$$V_1 = V_0 + m_0\omega^2 y_0 \tag{7}$$

$$M_1 = M^g_o + V_1 l_1 \tag{8}$$

At any distance X from the left-hand end of section 1, the bending moment is given by:

$$M = M^g_o + \frac{[M_1 - M^g_o]}{l_1} x \tag{9}$$

the slope at point 1 is given by:

$$\theta_1 = \frac{1}{(EI)_1}\int_0^{l_1} Mdx + c$$

$$= \frac{l_1}{(EI)_1}\left[M^g_o + M_1\right] + \theta_0 \tag{10}$$

Similarly by integrating the slope function, it can be shown:

$$y_l = \frac{l_1^2}{(EI)_1}\left[M^g_o + \frac{M_1}{2}\right] + \theta_0 l_1 + y_0 \tag{11}$$

For the nth section and point the equations are:

$$V_n = V_{n-1} + m_{n-1}\omega^2 y_{n-1} \tag{12}$$

$$M_n = V_n l_n + M_{n-1} \tag{13}$$

$$\theta_n = V_n \frac{l_n^2}{2(EI)_n} + M_{n-1}\frac{l_n}{(EI)_n} + \theta_{n-1} \tag{14}$$

$$y_n = V_n \frac{l_n^3}{6(EI)_n} + M_{n-1}\frac{l_n^2}{2(EI)_n} + \theta_{n-1} l_n + y_{n-1} \tag{15}$$

The values at the right-hand end of a section may be derived from their values at the left-hand end. By applying equations (12) to (15) for successive sections the values at any point in the shaft may be expressed in terms of V_0, M_0, θ_0 and y_0. In all cases, two of these will be known from the boundary conditions. The other two can be determined if the boundary conditions at the other end of the system are expressed in terms of these with coefficients depending on the stiffness parameters, masses and speed of rotation. The solution of the resulting simultaneous equations will give the frequency.

It should be noted that if there is a bearing at point $(_{r-1})$ with support stiffness given by k $(_{r-1})$, equation (12) becomes for point r

$$V_r = V_{r-1} + \left(m_{r-1}\omega^2 - k_{r-1}\right) y_{r-1} \tag{16}$$

Discussion

Mr. G. C. Volcy, M.Sc., M.I.Mar.E., noted that the author's conclusions were parallel to his of a couple of years ago, and that they had encountered a lot of confusion concerning the phenomena relating to such a relatively simple mechanism as line shafting. He regretted that several of his considerations on shafting and ship behaviour, had escaped their attention, and also that in his opinion, the confusion concerning dynamic behaviour of line shafting had not yet been cleared up in the paper and recent publications.

Mr. Volcy had several points relating to: the virtual mass and moment of inertia of water in propeller and shaft vibrations or hydrodynamic coefficients; stiffness (static or dynamic) of supports; hydrodynamic excitations; contra-rotating propellers; continuing presence of stern tube (the basis of numerous trouble with stern gears); location or displacement of supporting points; number of shafting supports necessary for vibration calculations; and wear down of stern tube bushing material.

He would concentrate on certain differences of opinion between the authors and himself and present some recent results of research carried out by the Special Study and Research Section of his organization.

There were two different types of propeller shaft vibration: lateral, i.e., either vertical or horizontal vibration; and precession (positive or negative) also called forward or counter whirl (not to mention supplementary nutation).

He considered it was hazardous to combine the two types of vibration and analyse "the lateral vibration characteristics and the associated whirling speeds of shafting systems", as both phenomena, although in essence similar, appeared in different ways.

Lateral vibration occurred on a vertical or horizontal surface, but precession was due to the influence of gyroscopic coupling (appearing as the vector of coupling, maintaining its position in space as constant) and occurred on a circle or cone; i.e., only at extreme points, at 90 degree intervals, were these phenomena encountered: in upper and lower vertical positions the precession movement was equal to vertical vibration; in extreme starboard and port positions the precession was equal to horizontal vibration; if, of course, precession or whirling occurred.

However, in reality, neither of these phenomena occurred. Due to non-linearity of support stiffnesses in each direction, the presence of clearance and non-linearity of film oil and displacement of support points as a function of rev/min, thrust and its eccentricity, the propeller shaft moved in an oscillatory semi-circular or elliptic, down and up, and port and starboard movement.

How many cases of whirling, either positive or negative, had the authors encountered? Whirling had been experienced on a series of small fishing vessels, when the shafting had relatively high revolutions and a relatively heavy propeller. In this situation whirling occurred, especially with wear of the aft *lignum vitae* bushing.

Lateral, vertical and horizontal vibration of the tailshaft was encountered several times before the introduction of rational curved line shafting alignment, about ten years ago. An example of this vibration was shown in Fig. 10. It was eliminated by applying curved alignment and rational distribution of shafting bearing reaction. The results are shown on the lower part of Fig. 8.

The conditions described concerned initial dynamic behaviour of the shafting, and in his organization's experience, before further analysing dynamic behaviour, it was essential to know the actual static conditions in which the shafting was situated. The increase in tonnage and building of huge oil and ore carriers made for incompatibility between the flexibility of the steelwork and the stiffness I/l of the shafting. Deformation of steel parts in ships would influence the distribution of the reaction of the shafting.

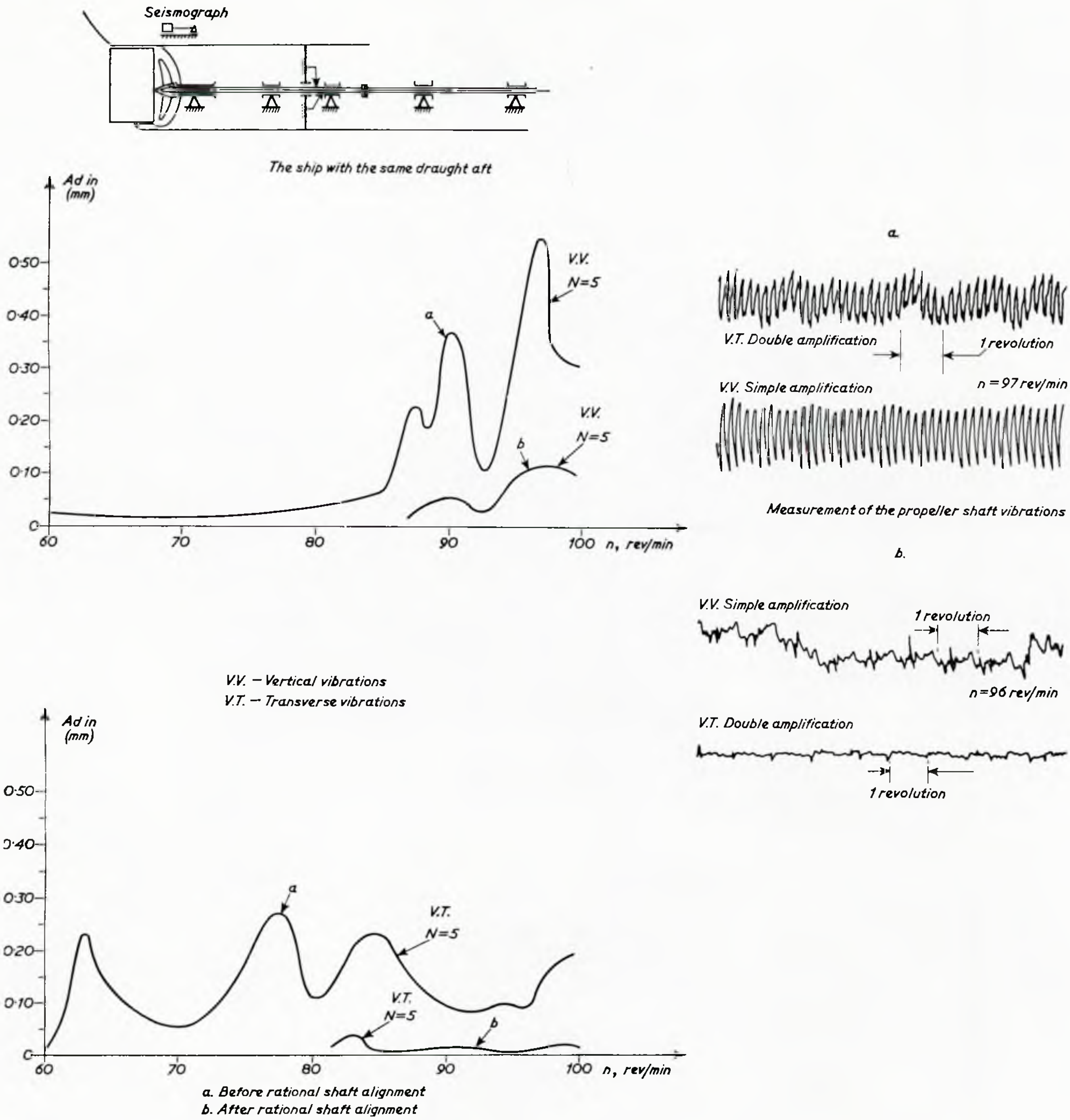


FIG. 10—*Forced hull vibrations engendered by propeller shaft vibrations of a 58 000 dwt tanker—Five-bladed propeller*

Fig. 2a which showed the influence of propeller thrust eccentricity, could also be explained by the presence of these deformations, which resulted from:

1) deformation of the assembly of the hull girder as shown in Fig. 11 (N.B. non-conventional deformation of the hull aft part, in way of the engine room, was mainly due more to shearing deformation than flexural);
2) deformation of double-bottom steel work (hogging for loaded ships);
3) importance of outside shell steel work stiffness in relationship to that for double-bottom steel work.

He cited two examples: one concerned tankers with full aft body; when the relationship between the forces acting on long double-bottom floors was greater than that acting on the outside shell plating; such a force distribution instigated the hogging deformation of the double-bottom as shown in Fig. 12. The other case concerned the big tanker with slender aft body shown in Fig. 13. As could be seen, the force acting on the outside shell was more important than that due to the increase of hydrostatic pressure acting on short double-bottom floors. Due to loading conditions, sagging deformation of the double-bottom floors resulted.

Their experience, from simultaneous calculation and measurement, had shown that correlation could only be achieved if the whole elastic system of the hull girder was considered and the double-bottom structure and internal steel work of the engine room, also taking into account the steel work of superstructures. Computer calculations for finite element techniques had been

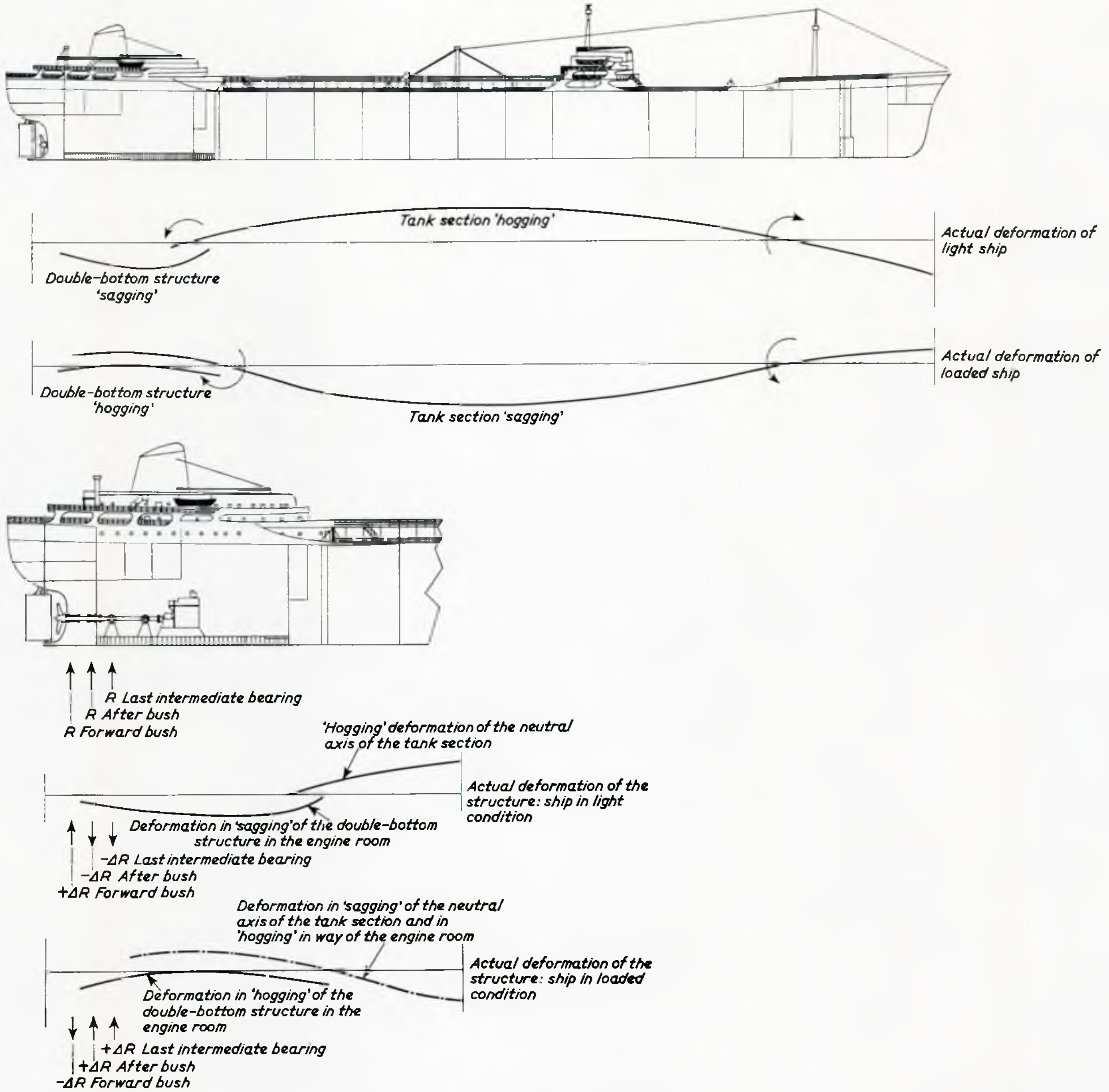


FIG. 11—*Influence of hull structure deformations of a tanker, due to loading conditions, on the distribution of reactions in way of propeller shaft supports*

carried out by his organization. Fig. 14 showed an elastic system of the aft part of the hull girder and an engine room of a 250 000 dwt ship.

With the above information and the stiffness and mass matrices, response of the system to all excitations could be analysed, including those from the propeller influencing the tailshaft behaviour. In present studies these excitations were obtained either from model tests and measurements or from calculations. Research on resonators of forced vibrations and their detuning was carried out and the response of the assembly of the considered steel work determined.

MR. W. STRASSHEIM asked whether the authors could advise on the availability of the data accumulated on the effect of bearing and seating stiffnesses particularly with reference to twin-screw ships and the effect that had been found on whirl frequencies when included with other data, i.e., bearing stiffness, shaft seating stiffnesses etc.

In a short case history outlining his company's conflicting experiences when applying these relevant points of data, he said that it was the practice of his company to calculate tailshaft whirl by the Jasper method in the earliest design stages, because it was considered essential to arrive at the number and disposition of bearings at an early stage, at which they were not able to supply the data required for the Prohl method. If a problem was anticipated, a full calculation for the entire line shafting was carried out by Prohl—at the moment with an outside agent at Newcastle Polytechnic.

When using the various stiffnesses obtained in practice, unfortunately the methods usually varied in frequency prediction by large amounts. Usually Prohl had given a higher frequency for a given shafting system. This was understandable with the Jasper method, involving just a short shaft and disc system, whereas Prohl was quite different, and one would not expect the same. They were then placed in a dilemma, whereby they must present owners, if they so requested, with cases in which the Jasper

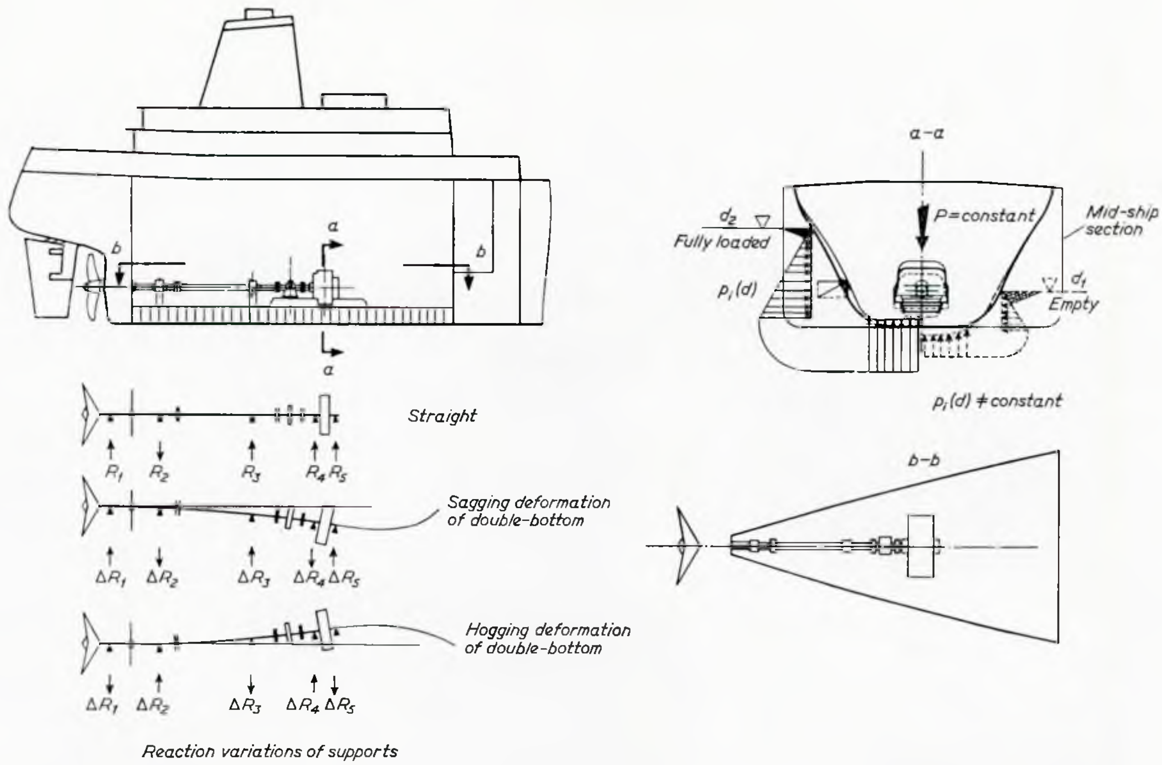


FIG. 12—*Local deformation of engine room steelwork due to loading conditions—Tankers with full aft-body*

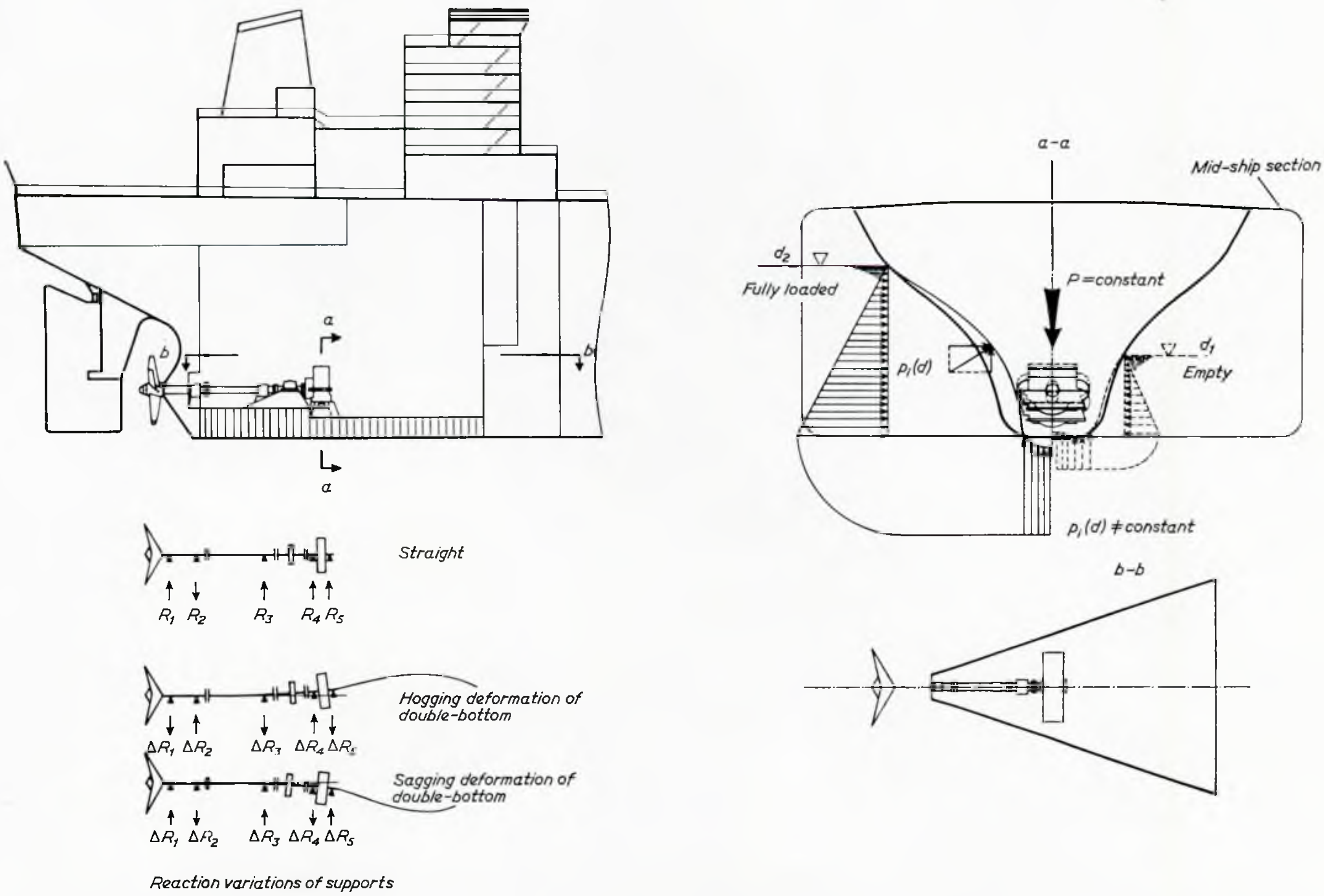


FIG. 13—*Local deformations of engine room steelwork due to loading conditions—Large tankers with slender aft-body*

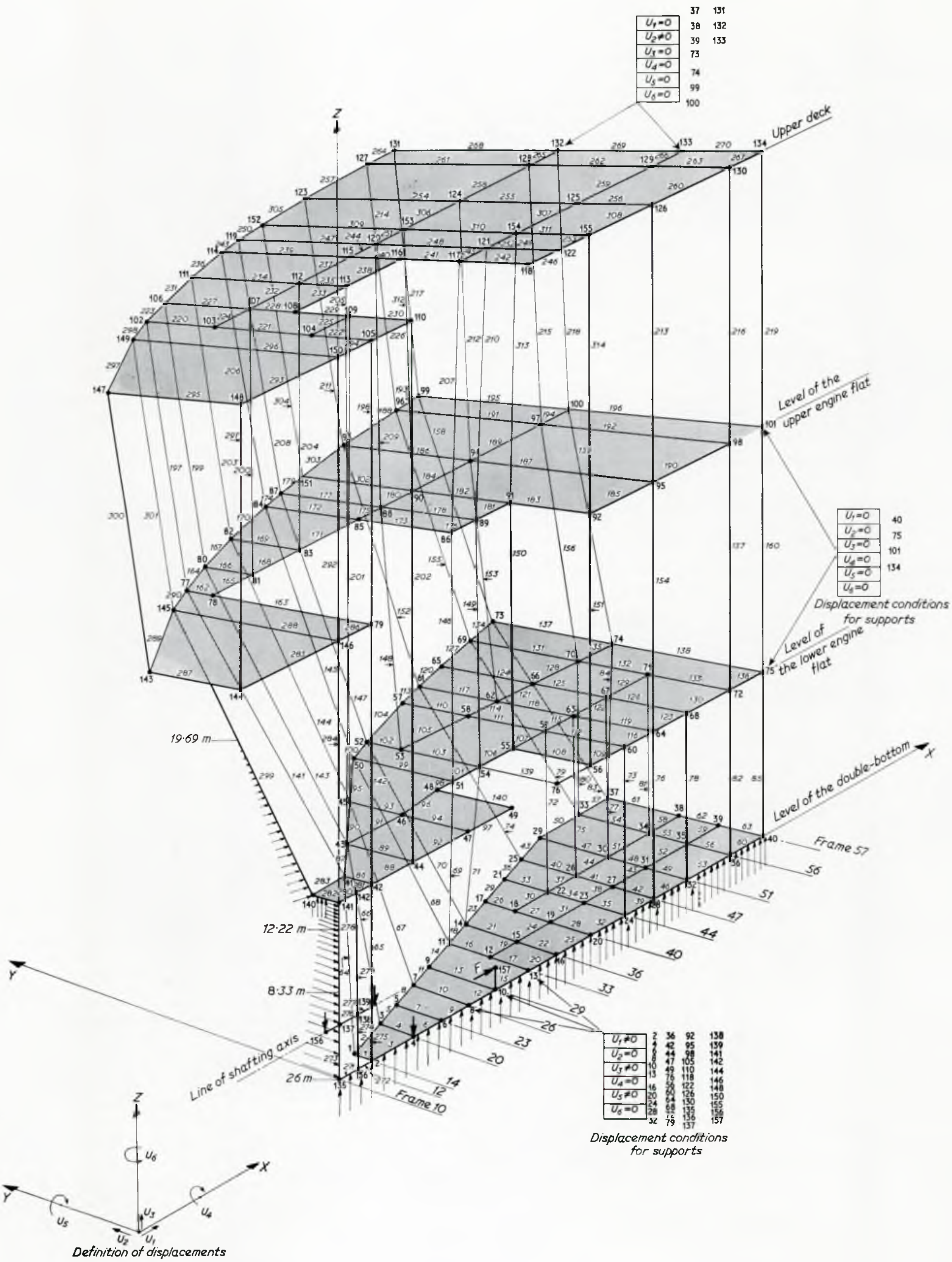


FIG. 14—*Deformation calculation of the after part of the hull—Equivalent elastic model*

method indicated the ship with a whirl frequency perhaps of only half operating speed and quite satisfactory, whereas Prohl might indicate 25 per cent above running speed. Invariably the question arose: what happened if it was somewhere between the two?

It had been noted that on installations where critical speeds had been predicted by both, no sign of either had occurred. In large tankers, which they had investigated most closely, the lowest critical, according to Jasper—blade order—was just above the maximum rating of the engine in any case, this preventing any confirmation of their design calculations. They applied a 25 per cent margin, and not a 15 per cent margin, because of these conflicting examples.

An example had occurred about 18 months ago on a British Rail ferry, when tailshaft whirl was apparently well clear of operating speed, calculated by Jasper, and the ship developed very severe blade order whirl in all shafts when running astern at speeds of over 200 rev/min. The ship had an m.c.r. of 280 rev/min. It was found that the worst vibration occurred in the intermediate shafts where the maximum recorded was in excess of $\pm$ 1 mm; the operators not feeling inclined to increase revolutions to find out whether they were actually on the critical level and could not get through it. There was also a severe hull vibration, in way of the shafting, which was most alarming, in engine room spaces. The system was checked by the Prohl method and it indicated blade order whirl at 203 rev/min., exactly at the speed measured, but no significant vibration at all occurred during ahead running. Astern vibration commenced after the ship had developed a wake, when forces acting on the hull appeared to excite shafting to whirl. This also occurred with the port machinery developing near full power and the starboard machinery shut down. A sympathetic vibration appeared in the stationary shaft. So much was vibrating that it was very difficult to measure amplitudes within the scope of the instrumentation.

The speaker demonstrated some amplitudes measured in each section of shafting with a maximum of 1 mm in the intermediate shafting. This ship had c.p. propellers and arrangements for including an extra bearing in the system were made for the next drydocking. This apparently completely cured the astern whirling vibration. Putting an extra bearing in the system on both shafts involved considerable work on the hull. Also some work was carried out on stiffening the hull plating. Therefore, it was very difficult to tell whether the introduction of an extra bearing had achieved a cure, or whether it was attributable to the hull stiffening.

A second case had arisen recently concerning a single-shaft vessel. It also had a c.p. propeller and developed extreme apparent lateral vibration, although so far none of his company's engineers had measured it on board. The ship had a critical speed calculated at only six revolutions above running speed, running speed being 150 rev/min whirl at 156 by the Jasper method. Prohl would indicate it at well above running speed, with no possibility of occurrence.

The single-screw ship to which the authors referred highlighted the contradiction which they were getting between the Prohl and the Jasper method. He hoped the authors would comment on what they believed was the best to apply in the early design stages, bearing in mind the restricted information.

Commenting on seating stiffness, the speaker recalled that this had been mentioned on the twin-screw ferry which had the severe vibration, on which seating stiffnesses were well under half the minimum value quoted by Mr. Toms for single-screw ships. They had also measured very low seating stiffnesses on single-screw ships such as bulk carriers, stiffnesses of the order of 800 to 1000 ton/in being measured, in some cases.

He asked the authors to comment on the phenomenon which occurred in twin-screw ships, where there was a difficulty in differentiating between the bearing seat and the ship hull structure, where the ship hull structure was so near to the shaft that it was difficult to determine where the seat ended and the shell began. He was thinking particularly of ships with twin bossings and very low seats of the order of only 18 in high virtually embedded in the bossing, being very susceptible to wake forces.

The authors were also asked to comment on the bearing which c.p. propellers seemed to have on those two incidents, and the fact that the speaker had experienced no whirl vibration in other ships with conventional propellers. His company had done all their alignment work very carefully in the normal fair curve method to ensure bearing loading with optimum spans.

Finally, the authors were asked whether they thought that propeller manufacturers should provide more information, even if it was typical information only, on thrust variation and offset thrust effect; the speaker's company had had great difficulty in getting this information about propellers of continental manufacture, particularly controllable pitch propellers. One was left with a dilemma on the assumptions to make for the Prohl programme on entrained water. Did it include the entrained water allowance on fine pitch, or the entrained water allowance on maximum pitch, the variation of which was considerable?

Mr. I. S. McGill commented that the paper was a natural follow-on from Professor Carnegie's and Mr. Pashricha's paper in that it illustrated the emergence of a new generation of vibration problems. While resonant torsional vibration remained one of the major vibration problems associated with marine propulsion systems there was a growing number of vibration problems which, while they might not lead to short term system failure, could nevertheless significantly impair the performance or endurance of the system. That a torsional vibration analysis might require to be supplemented by axial vibration analysis was demonstrated by a work in 1960.* The present paper demonstrated a requirement to consider, in addition, the whirling characteristics of the system.

The authors were concerned in the main with blade rate whirling. If this had been a serious problem with naval ships, few such ships would now be sailing, since the blade rate critical speed was almost bound to be within the running range, and probably near or below half speed.

The problem most likely to be encountered was synchronous whirl, particularly with the increasing use of c.p. propellers which had a relatively high mass and a centre of gravity located some way aft of the blades. The speaker quoted one instance where, with the bearings assumed rigid, a conventional Prohl calculation, predicted a critical speed which was, in fact, within the operating speed range. The ship had been to sea. Some measurements had been taken on it which showed evidence of growth in the vibration level towards full speed, and the semblance of a critical speed in this region. The vibration levels recorded were not, however, sufficient to cause worry. He wondered how much this was due to water damping. The resonant vibration considered was associated with the outboard span and propeller both of which were in the water. Very little appeared to be known about the influence of water damping. It was known that for synchronous whirl, damping would lift the critical speed, but one did not know by how much. Was it possible to run comfortably at a critical speed in the presence of water damping and was one therefore, worrying about a problem which did not exist?

The speaker said that as his principle interest was in synchronous shaft whirl, he would not consider further the subject of blade rate excited whirl. In synchronous whirl, one was basically concerned with the solution of the shaft in some fixed set generated by out-of-balance forces. There was no flexure of the shaft and hence no internal sources of damping. Thus, in the absence of external sources of damping one had, as stated in the paper, a very sharp resonance. Significant external damping would both reduce and broaden the resonance peak.

Turning now to the problem which existed in many minds regarding the analogy between transverse vibration and whirling, the speaker noted that, for a simple single-span rotor in transverse vibration, damping reduced the critical speed, or the resonant frequency. For whirling it had exactly the opposite effect, i.e., it raised the critical speed. The same was true of the gyroscopic effects. The gyroscopic moment reduced the transverse natural frequency but raised the critical whirling speed. In the Pana-

* Goodwin, A. J. H. 1960. "The Design of a Resonance Changer to Overcome Excessive Axial Vibration of Propeller Shafting." *Trans.I.Mar.E.* Vol. 72, p37.

gopulos formula the effect of the gyroscopic moment was to reduce the critical whirling speed. Therefore, this was a transverse vibration formula and not a whirling formula and could be very pessimistic when used to predict critical whirling speeds. He thought that the formula should be reviewed with caution.

To conclude, whirling was becoming of increasing importance. There was a shortage of information. What was the significance of damping? What were the entrained water effects as opposed to the set of "magic numbers" which were currently used in whirling calculations? He agreed with the authors that there was a case for some experimental work to give a reliable and meaningful design procedure.

Authors' Reply

The authors, replying to the discussion, thanked those who had participated. It was clear from the questions asked that considerable attention was being paid to this form of shaft vibration.

In reply to Mr. Volcy, the authors confirmed that they were aware of his paper of 1967, which was available both in full and condensed version as B.S.R.A. translations.

He had commented on the motion of the shafting and had sought to distinguish between the single-plane, or truly lateral vibration and blade order whirl.

If both the bearing stiffnesses and excitation were completely symmetrical in the plane of vibration, the motion of the geometric centre of the shaft would be a circle. In practice, it was rare for complete symmetry to be achieved. In general both the bearings and their supporting structure possessed different stiffness characteristics in the horizontal and vertical planes, and the excitation also varied in these directions. When calculations were carried out for each of the stiffnesses, two distinct critical speeds would result. When the shaft rotated at one of these speeds, the path of the geometric centre was a narrow ellipse and not a circle. If the stiffness characteristics in the two mutually perpendicular planes were sufficiently different so that the two distinct critical speeds were separated by more than 20 per cent, the motion of the geometric centre would degenerate into a straight line. The direction of this motion would depend upon which of the critical speeds coincided with the running speed. This single-plane, or truly lateral, vibration was the type that was usually noted and had led people to doubt the existence of what has been termed "blade order whirling". In fact, it was simply a particular manifestation of the general phenomenon which had been described in literature on the subject.

The motion of tailshafting was extremely difficult to measure. Lloyd's Register had been involved in many experiments of this nature, one of the first being in 1951, when Dr. Dorey* reported on the results of tests carried out on a T2 tanker. Spring-loaded probes riding on the shaft liner had electronically recorded the motion of the shaft centre in the vertical and horizontal planes. With the vessel in the lightly ballasted condition, resonances of linear displacements in both planes at blade frequency were recorded and these had the characteristic that in both planes the peaks occurred at two distinct speeds. With a service speed of 86 rev/min and a four-bladed propeller, the smaller peak occurred between 72 and 75 rev/min and a larger peak at 82 to 88 rev/min. In the latter case the lower figure referred to the vertical plane and the higher figure to the horizontal plane. At each fourth order peak there was a corresponding eighth order peak and where the fourth order peak was small the eighth order peak was large and *vice versa*. The order of magnitude of the deflexions was $\pm$ 0·010 in maximum for the fourth order. The fourth order movements at each resonance plotted as four repeated elliptical orbits.

These experiments were carried out at a time when tailshaft whirling was first under study and, since then, Lloyd's Register had completed many further measurements of this particular type—the last being some two years previously on a tanker equipped with a six-bladed propeller and a white metal stern tube. However, one could not have expected shipowners to go along with this type of experiment very often.

* Dorey, S. F. 1951. "Stresses in Propellers and Propeller Shafting under Service Conditions." *Trans.N.E.C.I.E.S.* Vol. 67, p 389.

Mr. Volcy had commented further on the effect of hull flexibility on shafting alignment and had correctly stressed the importance of good alignment in avoiding tailshaft lateral vibration. The authors would certainly agree, that with the present-day large tankers, there could be problems of shaft alignment associated with hull flexibility. This was particularly so with vessels of very wide beam due to the effect of the hydrostatic forces on the underside of the hull. Such a phenomenon could not be investigated theoretically with the representation of the hull by a simple beam. Fortunately, at Lloyd's Register, they had fully comprehensive finite element computer programmes which enabled them to explore theoretically the effect of variations in loading on hull deflexion. However, the time and cost entailed prohibited the use of this form of analysis for routine production calculations of alignment and shafting vibration.

In reply to Mr. Strassheim's question concerning stiffness particularly in twin-screw vessels, the effect of hull flexibility on lateral vibration would be experienced most where there were "A" brackets or bossings. Unfortunately, there was very little published data available. Lloyd's Register had partaken in static stiffness tests of bossings and, whilst it was difficult to give a realistic answer to the question regarding values of stiffness, the authors felt that for bossings, one would expect stiffnesses of 30 to 40 ton/mm.

The assumption that a bossing could be represented by a linear spring was generally adequate, provided that the shafting natural frequency was well removed from the natural frequency of the bossing.

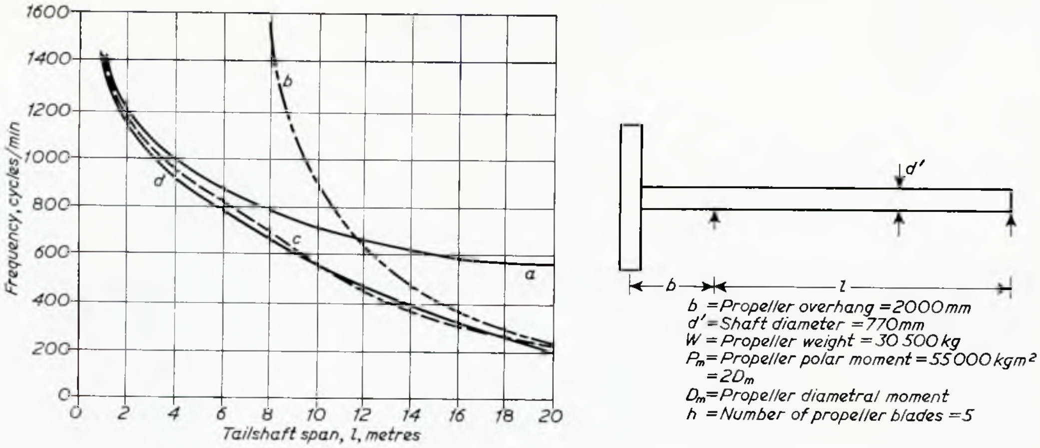
For twin-screw container ship installations, where the bearing seatings—as described by Mr. Strassheim—were of low height and virtually integral with the bossing or shafting tube, some firms had investigated the combined response of both the bossing and the shafting. Unfortunately, such an exercise still remained largely academic.

In regard to Mr. Strassheim's problems concerning the calculation of natural frequencies of lateral vibration and the use of either Jasper or Prohl, the authors had prepared Fig. 15. There should be little difference between the results of Jasper and Prohl provided that for the former, the mass of the shafting was taken into account. As shown in detail in the paper, the method adopted by Lloyd's Register was analogous to that of Prohl which the authors had recommended.

In regard to Fig. 15, the authors had calculated the fundamental natural frequencies of blade order, forward whirl for the simplified system shown, using both Jasper and the Lloyd's Register computer programme. In applying the Jasper formula, the forward end of the shafting was taken as simply supported.

The figure showed that with increase of tailshaft bearing span, the discrepancy in frequency between the Jasper and Prohl-type method increased, with Jasper understandably predicting the higher of the two. However, by taking into account the natural frequency of the tailshaft bearing span by simple beam formula and combining the results with those of Jasper by Dunkerley's method, the resulting natural frequencies fell closely in line with those predicted by Prohl's method.

Mr. Strassheim's experience in regard to lateral vibration of intermediate shafting on a ferry was of considerable interest and was the second case known to the authors. Each time the shafting had been supported by roller bearings. The authors believed that the excitation from the propeller had found its way to the shafting via the hull and shaft bearings. The likelihood of this happening



Curve a is derived from Jasper formula:

$$\Omega_1{}^2 = \frac{(m\delta_p + G\theta_m) \pm \sqrt{(m\delta_p + G\theta_m)^2 - 4G\,(\delta_p\theta_m - \delta_m\theta_p)}}{2mG\,(\delta_p\theta_m - \delta_m\theta_p)}$$

where:

Ω_1 = natural frequency, rad/s
m = propeller mass plus entrained water
G = $I_d\,(1 - K/n)$
K = I_p/I_d
I_p = propeller polar moment plus entrained water
I_d = propeller diametral moment plus entrained water

$$\delta_p = \frac{b^2\,(b + l)}{3EI}$$

$$\delta_m = \theta_p = \frac{b\,(b/2 + l/3)}{EI}$$

$$\theta_m = \frac{(b + l/3)}{EI}$$

Curve b is derived from simple beam formula:

$$\Omega_2{}^2 = \frac{\pi EI}{lA\rho}\,g$$

where:

I = second moment of area of shaft section
A = area of shaft section
ρ = density of material

Curve c is derived from curves a and b using Dunkerley's Method

Curve d is derived from Lloyd's Register computer programme employing method analagous to Prohl

FIG. 15—*A comparison of Jasper and "Prohl-type" calculations*

to a shafting natural frequency which was otherwise virtually unexcited could not reasonably be determined at the design stage.

The authors agreed with Mr. Strassheim that it would be useful to have more information from propeller manufacturers. Calculations of lateral vibration and alignment should be carried out at the design stage and the offset thrust was one quantity which was not often available and one was sometimes put to great trouble in trying to formulate a reasonable value.

In regard to the point raised by Mr. McGill, the authors agreed that with naval shafting arrangements the problems were somewhat different from those of merchant ships. The considerable flexibility of naval shafting, coupled with high speeds of rotation, meant that care had to be taken to avoid first order whirling arising from unbalance. However, the blade order whirling speeds would lie within the operating shaft speed range —a condition which also existed on some present-day ferry designs.

With first order whirling one could expect only external damping to exercise any control on amplitudes of vibration, whereas with blade order whirling there would be a far greater degree of internal damping arising from shaft hysteresis. Unfortunately, there was still scant information in regard to damping. The authors could not give a qualitative answer to the degree of damping to be expected, but progress was being made and the more use that could be made of advanced computing techniques, the better able they were to put the available data to good use.

Mr. McGill's comments in regard to the mass centre of gravity of controllable pitch propellers were of interest. This would vary somewhat with differing designs and was likely to be more pronounced with propellers of naval design. The authors had examined the relative weights of fixed and controllable pitch propellers which was a further factor leading to a reduction in natural frequency and they had found one instance where for the same propeller diameter, the weights varied by a factor of three.

SOME SHIPOWNER VIBRATION PROBLEMS AND ANALYSIS

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Three cases of serious vibration which have occurred in new vessels within the past six years are described in some detail together with the steps taken to understand the basic technical problems and provide economic solutions to each.

INTRODUCTION

To shipowners the words vibration and trouble are generally synonymous. There must be very few modern ships which have entered service and which have not been afflicted with vibration to some degree resulting in failures ranging from serious hull and machinery defects, to disintegration of small components. It would also appear that in recent years the incidence of vibration troubles has increased considerably. The basic task of the shipowner when faced with such problems is to keep the ship in service in the best way possible pending a long term solution, and to ensure that in future ships the fault cannot recur. It is not necessary that the solution for existing ships be the best technical one, although generally this is desirable, but it could be a palliative which overcomes the immediate difficulty at minimal cost, and which is free from serious side effects and long term hazards. Solution by palliative must include a full knowledge of the troubles, the fundamental reasons for them and the ideal solution. This paper gives three examples of this approach to overcoming difficulties with vibration.

CASE I—DIESEL ALTERNATORS—UNSTABLE PARALLEL OPERATION

The Problem

This case is a problem of the electro-mechanical frequency of oscillation being in resonance with the natural frequency of the mechanical control system with resulting electrical and mechanical manifestation. It is included here for the vibration rather than strict electrical context, although inevitably good electrical as well as mechanical design is essential to the correct operation of the system.

When direct current generators are operated in parallel, the prime movers are independent of each other and apart from the bus-bar connexion and the field equalizing circuit, the electrical ends are also relatively independent, each producing its own electrical output to the bus-bars. With alternating current alternators connected in parallel, the position is different in that the electrical ends are locked in phase together with their prime movers. Thus any disturbance in one set produces an immediate reaction throughout the system. Although all this is fully described in any elementary treatment of a.c. alternators, it is considered necessary for this concept to be appreciated to enable a reasonable understanding of electro-mechanical frequency of oscillation and the problems which can occur when this corresponds with the natural frequency of another part of the prime mover/alternator system. The following is an attempt to describe electro-mechanical frequency of oscillation.

The characteristics of low frequency power oscillations which can occur in Diesel alternator systems may be considered analogous to a mechanical system consisting of two equal large lumped inertias connected by a torsionally elastic shaft. These masses may oscillate torsionally anti-phase at a resonant frequency equivalent to one of the inertias and the torsional rigidity of one half the length of the connecting shaft.

For the case of Diesel alternators in parallel, in its simplest form the inertias are those of the engine and alternator, and when the machines are connected on load *via* the bus-bars, the elastic coupling is provided by the restoring torque that a synchronous machine experiences if it is displaced from its quasi-steady state position relative to a steadily rotating flux wave. The electro-mechanical frequency is equivalent to one of the inertias and the synchronizing power.

Inherently synchronous machines tend to remain in phase, but cyclic disturbances originating in the engine, for example, may cause one alternator to be out of phase with the others, resulting in one machine as a generator supplying power to the other as a synchronous motor in an effort to pull it back into step. If the input cyclic disturbances were at the electro-mechanical frequency of the sets, the anti-phase oscillations could build up to a resonant condition with consequent power transfer to and from the machines at the resonant frequency.

The first experience of this was in the author's company in a large passenger vessel with four 1500 kW turbo alternators. The evidence of trouble was occasional rapid shuttling of load from one alternator to another without any perceptible change in voltage.

Investigation immediately revealed that the steam control valves operated by the governors were partially seizing under working conditions and that the seizure was due to error in working clearances. Rectification of this fault reduced considerably the incidence of unstable operation, but did not eliminate it as occasional severe instability recurred. The basic cause was traced to resonance between the electro-mechanical frequency of oscillation and the natural frequency of the governor system of the turbines. When the springs in the governors were changed and the sensing point for the pressure governor of the pass-out steam was moved to a new position, the problem was solved. That was about ten years ago and there has been no recurrence.

The difficulties in the case of the Diesel alternators were not so easily analysed or overcome.

In new vessels the fitting of small compact high speed Diesel alternator sets has attractions, but their introduction in a.c. installations has presented some operational problems. The most difficult to understand and resolve concerned these large electrical power interchanges between sets operating in parallel. The worst conditions resulted in 100 per cent power oscillations which were self sustaining and produced sympathetic and violent transverse oscillations of each set on its resilient mountings. In some instances the reverse power protection relays were

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caused to operate, tripping a set, In the long term mechanical failures occurred; for example, the flexible coupling between Diesel and alternator. Trouble indications in the control room were represented initially by small oscillations in the kilowatt, ampere and power factor meters while voltages were fairly constant, giving little indication of the surging at the sets which increased violently until it could be heard in the control room.

The instability in service was first reported when the second vessel of a class of four was six months old and became serious when the third vessel of the class was twelve months old and the sets on the third vessel had run about 3000 h. Eventually, all vessels of this class were affected and two of a different class with the same type of Diesel alternator. Eventually a total of 28–250 kW sets operating at 1800 rev/min were displaying instability to some extent.

Test bed records showed that the first set to be built had severe single set instability due to torsionals and to the specified flexible coupling being unsuitable for this particular Diesel engine to alternator combination. New couplings resulted in single set stability, although there was some flickering of the kW meter with sets in parallel.

The Investigation

The manufacturers of the component parts of the sets embarked on a series of investigations which extended over a period of almost two years. The 28 Diesel alternators affected were distributed as follows: four ships, each with four sets plus one turbo alternator; two ships each with six sets plus one turbo alternator. Each machine had an output of 250 kW, operated at 1800 rev/min, and was flexibly mounted in the ship. At the outset little was known regarding the true causes and the only reasonable explanation advanced was that the electro-mechanical frequency for the sets operating in parallel was co-incident with a mechanical frequency of the engine or some external forcing frequency. This idea was arrived at after elimination of any mechanical defects and fuel supply problems as possible causes.

Investigations were segregated into two areas; the first concentrating on the sets and mainly conducted by the suppliers and the second examining external influences and conducted by the shipowner. The former series of trials, made with electronic instrumentation, sought to establish instability conditions and measure variables, power, rotational speed of sets, fuel pump rack movements, exciter field volts and amperes, main field volts, line volts and amperes and transverse movement of the sets. Efforts were concentrated on one vessel having six sets, with occasional visits to others, including two which were fitted with component variations of the standard sets being discussed. Other trials, undertaken at sea and in port, were aimed at discounting the possibility of external forcing frequencies having effect. In this two approaches were employed: on one twin screw vessel by using the main engines and varying the revolutions on both over the required frequency bands, and secondly by using an unbalance generator bolted to the bedplate of selected sets. Neither of these approaches produced any worthwhile clues, except to confirm that external cyclic forces apparently had no influence on the instability problems.

From the work of the suppliers and from observations in service it was soon established that the frequency of load transfer was 5·0 to 5·6 Hz, applicable to any of the six vessels, and further that the only readily available remedy to the problem was to change the fuel pump and governor on an unstable set or sets with a new unit. This practice gave freedom from instability for a period until wear occurred again in the mechanical governor working parts at which stage instability recurred. This remedy was expensive in reconditioned governor units, some lasting only 600–800 h. A lasting solution was needed which could only be provided following a full understanding of the problem.

Factors which emerged early in the investigation were that no set operating singly on load could be made unstable, and furthermore, operating any single Diesel set in parallel with the turbo alternator gave stability. Instability was most likely to occur when paralleling one Diesel set with another, although in service sudden changes of load, for example starting a main air compressor, could initiate unstable conditions. It also became evident that lightly loaded sets were more susceptible, and efforts were made to maintain loadings in excess of 200 kW.

In the process of elimination, regular checks were continued for blocked fuel filters and crankcase air breather pipes, inefficient fuel injectors, air in fuel pipes, slack or damaged couplings between Diesel engine and alternator, and to ensure that these and other mechanical features had no effect on producing instability during service. Although some of these could aggravate the condition, none was the primary cause. The torsional characteristics of the sets together with the performance of the automatic voltage regulators was also examined, but although, for the latter, badly adjusted sets could have some influence both were eventually discounted.

At this stage the efforts of the prime suppliers began to give some clear indications of causes and a better understanding of the problem. It became evident that the single set surge frequency was changing over a period of time; the frequency increasing from between 2 and 3 Hz up to approximately 5 Hz as wear developed in the mechanical governor. Examination of a number of 'worn' governors showed that there were no apparent differences from the new governor which replaced it, excepting for wear or polishing of an amount which was difficult to measure. The gradual wear, however, was sufficient to reduce the damping in the control system, and raise the surge frequency of the single set sufficiently for it to coincide with the now known electro-mechanical frequency at 5·0 to 5·6 Hz and induce instability.

Calculations had established that using the following standard equation that the electro-mechanical frequency would be 3·5 Hz:

$$\text{Electro-mechanical frequency } F = 3220\sqrt{\frac{Ps \times p}{N \times WR^2}} \text{ c/min}$$

where Ps = synchronizing power per radian (i.e. kW per electrical radian)
p = number of alternator poles
N = synchronous speed (rev/min)
WR^2 = inertia of alternator, flywheel, engine etc. (lb/ft^2)

The present theoretical study and observations in practice showed that this equation was inadequate for this application when considering high speed low inertia sets, as it was insufficiently refined to take account of the various system component transfer functions and the engine speed control and electrical damping factors which were needed to predict the true electro-mechanical resonant frequency.

The understanding of the problem permitted an examination of changes in these factors which would give separation of the two resonant frequencies. Increased damping could be introduced at the governor, or in the alternator, but the latter would have meant the complete redesign of the alternator. Frequencies could be modified by increasing inertia and by altering the resilient mountings of the sets.

Examination of the frequency characteristics of the flexible mountings showed these for dynamic conditions to have transverse, longitudinal and vertical resonant frequencies of 4·7, 19·3 and 6·6 Hz respectively. During checks for instability it could be demonstrated that these mountings had influence: by making mechanical adjustments to the mountings the sets could be made stable or unstable as required. With the transverse frequency of 4·7 Hz so close to instability frequencies there was no doubt that these amplified the condition when it existed. On selected sets at first, and later on all sets, the fore-and-aft pairs of mountings were turned through 90 degrees while the centre pair remained unchanged. Service performance of sets improved by increasing the intervals between instability, without achieving prevention.

The governor droop, that is the allowable set speed change with load change from zero to full load, was also modified experimentally from 4 per cent to 6 per cent. This gave temporary improvements, by increasing damping, as did also increasing compression on the governor buffer spring, but eventually unstable conditions once again enforced the changing of the governor and fuel pump assembly.

Two further series of tests were carried out, firstly on the three emergency generator sets of a large passenger vessel and on the three main generating sets of a dry cargo vessel. The Diesel alternators fitted in both were basically similar to the unstable sets, but in the former vessel the output was 175 kW. The inertias were almost double, 197·9 lb/in s² compared with 95·4 lb/in s² and the alternators had virtually full damping windings compared with an alleged 80 per cent damping. The sets in the passenger vessel were fitted with mechanical governors but in the dry cargo vessel the principal difference was the separate inclusion of an hydraulic amplified governor. Inertias in the cargo vessel were similar to the unstable set at 104·37 lb/in s² compared with 95·4 lb/in s².

The machines on the passenger vessel exhibited speed damping which was almost "dead-beat", making it impossible to measure the single set surge frequency, while the electrical decay factor was 0·18 compared with the worst value in an affected vessel of 0·035. The electrical mechanical frequency was again 5·0 to 5·6 Hz, but these sets could not be made unstable even though fitted with the same mechanical governor.

The tests in the dry cargo vessel were even more informative, in that the single set surge frequencies were all approximately at 1·2 Hz, with the electro-mechanical frequency at 3·8 Hz and the electrical decay factor at 0·06–0·08. Again the speed traces or rev/min of the sets showed "dead-beat" type damping, and considering inertias in these and unstable sets were closely similar, while in the sets of the passenger vessel inertias were double, emphasized the importance of system damping. Sets could be designed with inertias aimed at giving separation between surge and electro-mechanical frequencies but if the system lost damping due to wear or any other cause, they could become so modified as to be co-incident. The design of Diesel in the unstable sets did not permit any increase in inertias say by a larger flywheel and thus any increase would have to be made at the alternator end.

An interesting extension of the dry cargo ship tests lends further support. Two of the hydraulic governors were replaced by two mechanical governors which had previously been fitted on unstable sets. The modified sets immediately became unstable on paralleling with characteristics almost identical with those given for previous unstable sets. The hydraulic governors were refitted and the sets again became stable. The dry cargo ship with sets fitted with hydraulically amplified governors, has now operated for over four years without the instability troubles experienced in the previous six vessels.

The Solution

The knowledge now available was applied with the objective of obtaining the optimum solution. This meant increasing the effective damping and the best solution was to incorporate this

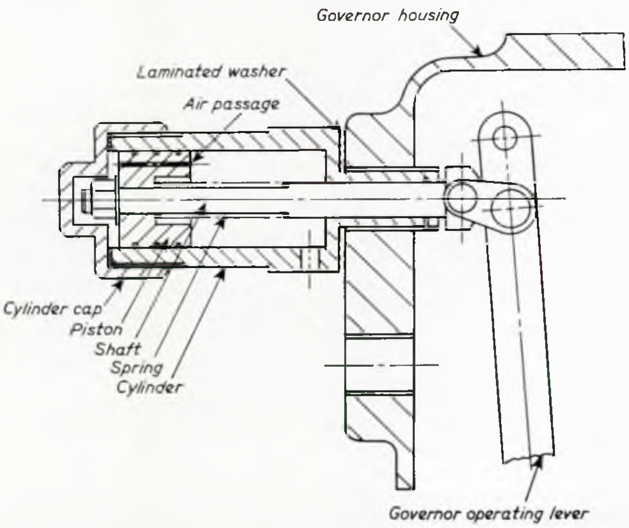

FIG. 1—*Mechanical governor pneumatic damper assembly*

at the governor. During the period of difficulty, chief engineers had demonstrated this very well by fitting rubber bands to the fuel pump racks of unstable sets to stabilize them temporarily. Changing the governor itself for the hydraulic type would have been very costly, particularly with 28 sets requiring modification. Air and electrical damping devices were designed and the former, consisting merely of a piston operating in a cylinder as in Fig. 1, was adopted and fitted to the free end of the governor fuel rack system. This palliative has operated successfully for 2½ years on all 28 sets.

CASE II—MAIN ENGINE LUBRICATING OIL PUMPS—VIBRATION IN PIPING SYSTEMS

In July 1966 when the second of a class of two twin screw Diesel vessels was six months old, a report indicated that there was severe vibration occurring at the lubricating oil pumps and the piping of the main engines, which had produced failures in components on the discharge side. The starboard pump was examined and the body and screws were found to be badly scored and the shaft worn, warranting the substitution of the spares. The vibration level on the return voyage from Cape Town was not improved.

Examination of records for both vessels showed that the top and bottom bearings of the motor of the port L.O. pump had been replaced once on the second vessel, while in the first, five changes had occurred in the port, starboard and stand-by pumps and the motor on the port pump had been removed ashore for balancing checks.

These events initiated investigations which continued for approximately 18 months.

Details of the Installation

The port and starboard eight cylinder main engines are each fitted with screw type lubricating oil pumps operating at 1760 rev/min and delivering 152 ton/h at a maximum discharge pressure of 77 lb/in² g.

Details of the pump systems are presented in Fig. 2 and those for the drain tank suction arrangements in Fig. 3.

Lubricating oil from each drain tank is drawn through a magnetic filter to the pump, and discharge through a valve, motorized main filters, filter isolating valve and the main lubricating oil coolers to the main engine. The inlet piping to the main filters consists of a "T" piece giving an adjoining pipe connexion to port and starboard systems from the stand-by pump which is common to both engines. A control valve is fitted on each pipe close to the stand-by pump and this pump is connected to both main engine drain tanks.

Vibration Investigations

During the first exploratory visit to the second ship, questioning of the engineer officers established that vibration had steadily increased since the vessel entered service but the greatest levels had recently been experienced at sea.

Subjective examination with both pump systems operating showed little vibration at the pumps, but some in the discharge lines at the main filters and adjacent to the stand-by pump in piping connected with port and starboard systems.

Vibration was assessed to be equally bad on both port and starboard systems and operation of the stand-by pump system with both suctions and discharges open gave no improvement. Further variations in the tests included the pressing up to capacity of the starboard drain tank to reduce suction head, and the increase and decrease of the discharge pressure on the system from the normal 68 lb/in² g; neither gave improvement.

Examination of the surface of the oil in the drain tanks with the systems operating, showed no visible movement of oil, but the surface was highly aerated to a depth of ½ to 1 in. Oil flowing down the No. 8 connecting rod also appeared to be aerated. Air cocks in the discharge system evidenced air in quantity.

It was acknowledged that the sump drain pipes were insufficiently submerged in oil and, that from No. 8 cylinder was located only 7 ft from the forward suction pipes and was not fitted with deflectors. The quantities of air in the systems could produce rough operation at the pumps and the two preceding

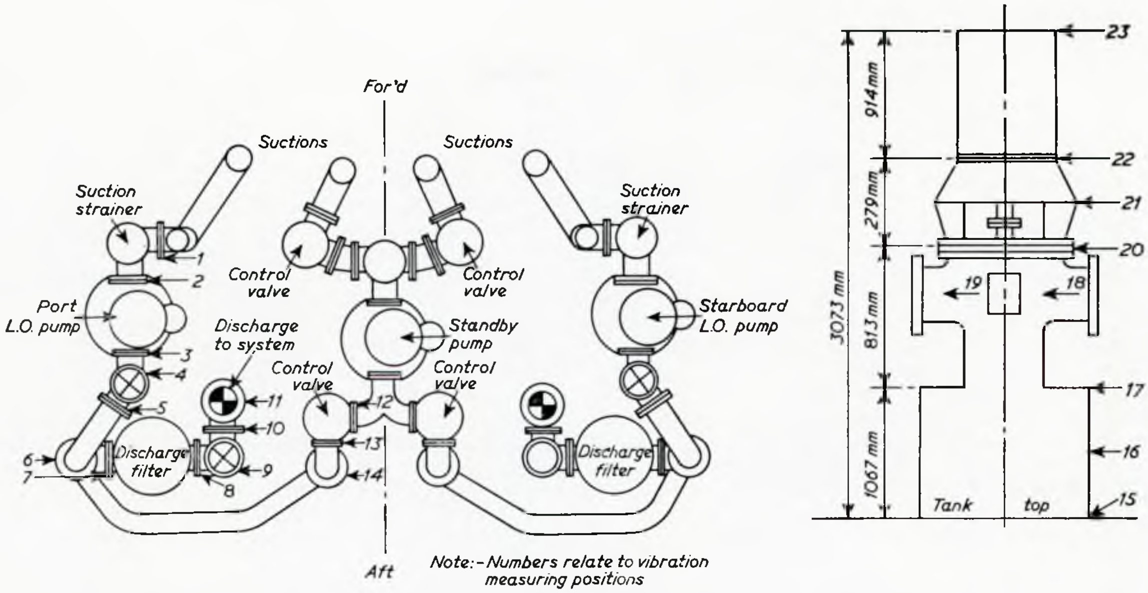

FIG. 2—*Diagrammatic arrangement of main engine lubricating oil pumps*

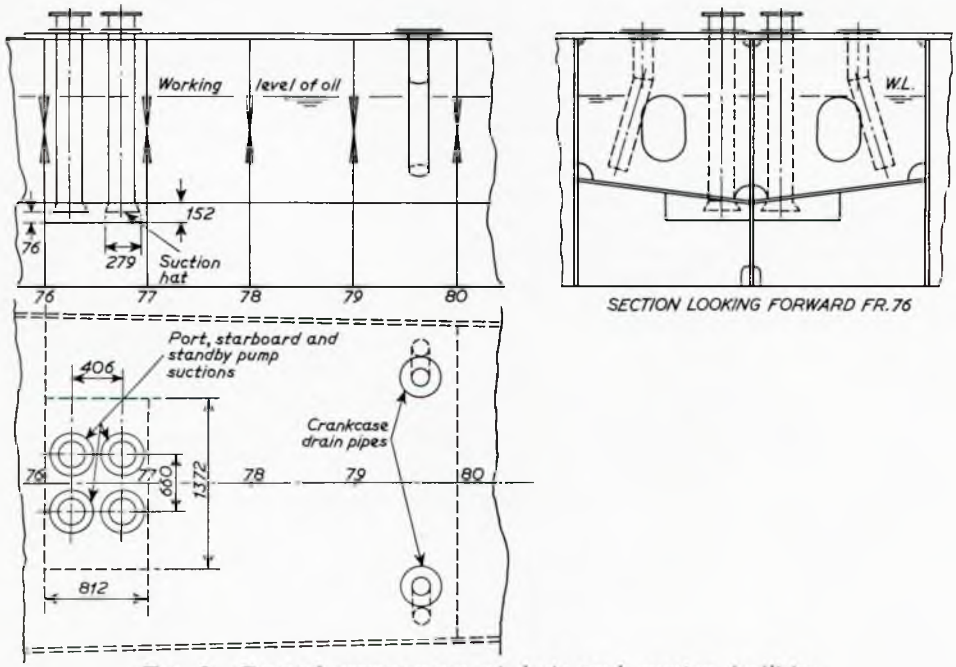

FIG. 3—*General arrangement of drain tank suction facilities*

factors could contribute to system aeration and warranted further investigation.

Consideration of the pump suction lift conditions gave the following figures.

Manufacturers stated suction lift	= 16·0 ft
Calculated maximum suction lift	= 15·0 ft
Observed normal suction lift	= 13·9 ft

During the next voyage of the second vessel the No. 8 cylinder drain pipe was blanked off, and to investigate further the pump suction lift conditions the suction filter basket was removed, the latter being equivalent to reducing the lift by over 6 ft. The results of both these trials were negative.

Following the next visit to the second ship it was decided to undertake the following modifications.

a) fit a splitter to the port lubricating oil suction pipe;

b) fit perforated plates and cruciforms to the aft and forward main engine drain pipes respectively;

c) increase the clearance of the bottom screws on the starboard pump on the basis of,

Pump capacity as ordered	— 152 t/h
3 per cent reduction of capacity	— 147·4 t/h
Main engine requirement	— 141·7 t/h

The work was completed in November 1966, but again tests showed no improvement in the vibration levels, both in port and at sea, but detailed examination of the operating systems lead to the opinion that air entrainment was considerably reduced.

Earlier provisional running of the plants on the first ship suggested that the same problems existed, but were less severe. However, just prior to the preceding modifications in the second vessel, the first serious troubles with cracks in the discharge piping of port and starboard systems, failure of valve support columns and pipe clips and mal-operation of stand-by system control valves were reported. The latter defect was serious and caused inadvertent transfer of lubricating oil between the two engines.

Reconsideration of the problem accentuated these facts; that there were many similar installations which were operating satisfactorily and that system aeration and pump suction lift if influencing the vibration were but secondary factors. The investigations so far undertaken had failed to clarify the true cause or causes.

Further comprehensive vibration trials were needed and these were carried out on the first ship, using an Anson vibrometer and a Pamatrada microwave instrument the latter also permitting the measurement of noise level.

The programme was aimed at establishing vibration levels throughout the system and of isolating the system beyond the pump and steadily introducing sections of the discharge piping. It was hoped that this procedure would isolate the true cause.

A long flexible pipe, of almost equivalent diameter to the system, and a sluice valve, were employed and the oil discharge led from the appropriate position in the discharge piping back to the crankcase. A considerable number of tests were run, the more important indicated below and the results presented in Fig. 4. For each position, vertical, athwartship, and fore and aft, vibration measurements were taken but only complete athwartship results are presented. Vertical levels, maximum points are presented, and these were generally lower, while fore-and-aft levels were of the same order as those given.

Measuring positions in the pump system are indicated in Fig. 2. Some tests run, were:

a) preliminary trial with the standard system;
b) the flexible pipe and sluice valve fitted directly to the pump discharge;
c) the flexible pipe fitted to the normal pump discharge valve;
d) the flexible pipe connected to the T piece in place of the discharge filter, T piece connexion to the stand-by pump in position;
e) as (d) but with stand-by pump connexion blanked-off;
f) T piece removed and a 90 degree bend fitted to the discharge piping from the pump to the discharge filter;
g) original system with the pump seating and pump motor shored up with 4 in $\times$ 4 in timbers;
h) as (g) but with the timbers on the motor removed.

The destructive vibration frequency was established at 29·4 Hz, the pump running speed and the records were remarkably similar to earlier less comprehensive results for the second vessel, confirming the problem was the same for both vessels. Vibration acceleration levels in some tests and parts of the system, namely the pump motor, the bends at the discharge filter and the adjoining bends at the stand-by pump approached 32 ft/s² (Fig. 4) for the normal system, which accounted for the failures experienced. The noise levels measured 2 ft from the pump exceeded 100 Db.

Tests (b) and (c) above with the pump and motor system operating independently of the discharge piping gave acceptable acceleration levels of 3·0 to 6·0 ft/s² (Fig. 4) but there was deterioration as the discharge piping was reintroduced into the system. These tests also showed that the balance of the pump and motor was acceptable.

Disconnecting the discharge filter together with piping to the coolers (test c) and piping to the stand-by pump (test d)

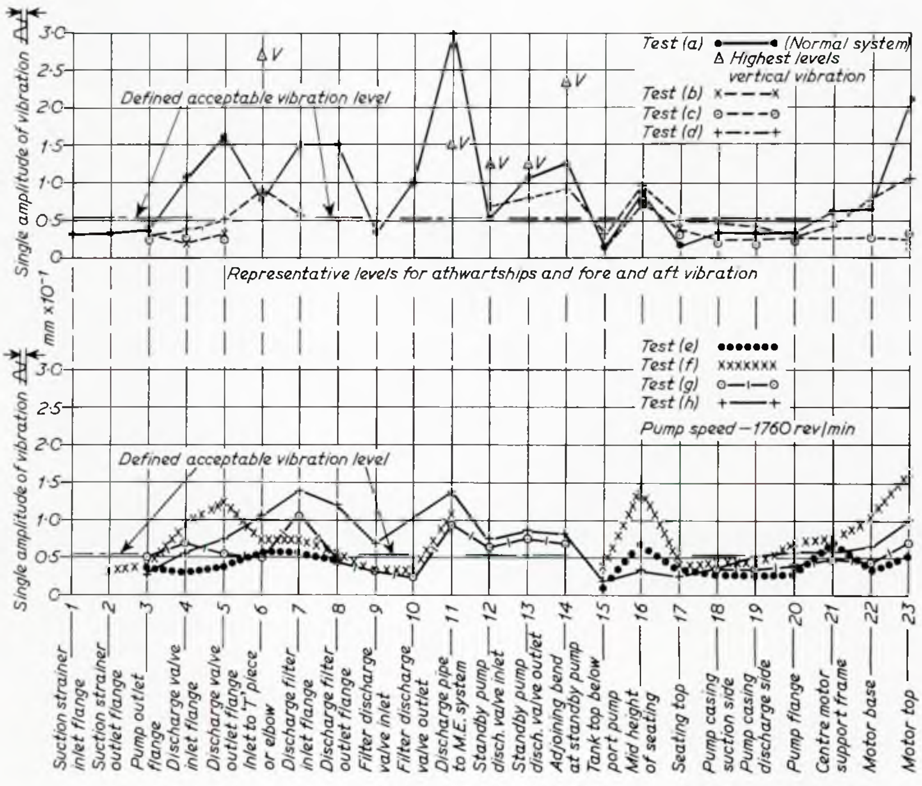

FIG. 4—*Main engine lubricating oil pump vibration—port system*

both gave some reduction in the acceleration levels but neither could be considered acceptable. It was apparent that vibrations originating at the pump were being magnified in the piping system in particular by the bourdon tube effect of the piping to the stand-by pump and also as a result of vibration interaction between the systems of the two pumps; vibration levels were always increased when both systems were in operation.

Unaccountably the test with the 90 degree smooth bends (test f) fitted in the system produced comparatively high acceleration levels.

Shoring up the pump seating and motor gave improvements throughout the system, but this was not maintained when the motor shoring was removed.

For the normally operating system (test a) accelerations of approximately 30 ft/s^2 at the motor top were reduced to 6·0 ft/s^2 at the pump, and this was applicable to both the vessels. Both the pump motor and the seating were considered suspect as it seemed possible that the vibrating cantilever could so affect the operation of the motor as to produce rotational fluctuations which would be transmitted to the pump. The regular failure of motor bearings fitted with this hypothesis. The considered cure for this was the strengthening of the seatings either by stiffening or by appreciable lowering with the consequent costly pipe modifications.

The influence of air leaking into the system *via* the pump gland during these tests, again clearly demonstrated this factor adversely affected the system.

Even with the present knowledge and data available it could not be said that the solution could be clearly defined. The vibrations or fluid pressure fluctuations appeared to originate in the pump and these were being magnified in the piping system. The necessary modifications that this suggested could be extensive and result in high costs. Consequently the following programme of changes was agreed and as an acceptance basis it was considered that the acceleration levels should be reduced to 6·0 ft/s^2 throughout the system:

a) the pumps would be fitted with flexibox type seals and the pump-motor flexible drives and flange gap checked;
b) the motors would be examined by the manufacturers, with particular reference to the motor bearings. The balance of these motors had been previously examined;
c) the pump seatings would be stiffened;
d) the system T pieces would be replaced by Y pieces. A non-return reflux valve would be fitted in the adjoining discharge pipes from the stand-by pump, together with 1 in thick rubber joints at the end flanges;
e) should the foregoing innovations fail, a slower running pump-motor combination would be tried to modify the system frequencies.

Undertaking the service items (a) and (b) revealed the fact that the motors had been supplied originally with incorrect roller bearings for the particular duty.

The items (c) and (d) were progressively introduced on the second vessel with the surprising result that on the starboard pump system the vibrations were reduced to an acceptable level throughout the system, but for the port side no matter what changes or combination were made the high vibration levels were maintained.

A process of elimination of the trials made on the starboard system suggested the stiffening of the pump seating to be the most influential factor, but did not account for the vibration differences between the two systems. In this the only major variation was the reduced diameter pump screws fitted earlier to the starboard pump.

During the next voyage of the second vessel the starboard screws were fitted in the port pump and trials run with the reflux valve in the discharge system. As this combination gave no improvement, the reflux valve was removed and replaced by a blank flange; this combination gave sufficient reduction to permit temporary safe operation. It was considered that the unsatisfactory port system results could arise because the screws and pump were not matched and clearances could be at variance with those intended. Further, it would be advantageous to reduce both upper and lower screws.

The Solution

The drydocking of the second vessel afforded opportunity to introduce the correctly reduced screws and also a slower running pump-motor unit. The former, although giving promising results, still left the system vibration above the defined limit. Introduction of the replacement pump, running at 1150 rev/min, necessitated lowering the seating by 4 in and modifications to take the larger pump, all of which further strengthened this structure. Operation of the pump, however, produced vibrations which exceeded those previously experienced. Athwartships movements were excessive which led to the tests being curtailed and the replacement of the original pump. Running this on the newly strengthened seating gave a marked improvement, sufficiently so to warrant the decision to reduce the seating heights in both vessels by the maximum practical amount of 18 in.

The final results confirmed the earlier tendency when the systems vibration was reduced to, or below, the agreed acceptance level excepting for isolated positions.

In conclusion there seemed little doubt that the earlier prognoses and solution regarding the cantilever vibration of the seating/pump/motor resulting in rotary disturbances and the magnification of the consequent fluid fluctuations in the piping system was correct, but air ingress to the systems undoubtedly worsened the condition.

CASE III—MAIN ENGINE VIBRATION AND RESULTING INSTRUMENTATION FAILURES

Introduction

From delivery, an 80 000 ton bulk carrier, fitted with an 8-cylinder main engine of 16 800 bhp placed aft, and a 4-bladed propeller experienced vibration which affected the bridge equipment, steering gear electrical equipment, main switchboard, engine room instrumentation, data logger and engine room ancillary equipment.

Some of the resulting failures could be corrected without incurring operational complications, but others were more serious and the two extracts, below, taken from the chief engineer's reports, give examples:

"For each alternator or circuit breaker on the main switchboard, isolating links are fitted between the circuit breaker and the main bus-bars. These links take the form of knife switches, there being one for each phase, but unlike conventional knife switches, the fixed blade is a single piece and the moving blade is made up with two pieces, these two pieces being spring loaded to provide firm contact. The three links are not strapped together thus each phase can be isolated separately.

During the recent passage to this port at 33 ft draught and full service power there has been considerable vibration and one of the many effects has been that the above isolating links have tended to creep out. The danger of single phasing will be apparent. Fortunately this occurrence had no serious consequences, and was noticed in good time. As an immediate remedy these links have been tied in position but a more permanent solution is definitely required."

A second incident concerned the steering gear electrical equipment:

"We are presently concerned for the well-being of the electrical equipment placed in the area of the aft peak. A recent incident involved failure of fuses for the steering gear Ward Leonard converter and although the automatic change-over functioned to switch in the alternative feeders and fuses, these had also failed. Matters were very quickly put right but for some minutes the vessel was without steering and at a time that caused the master considerable concern. As there is no evidence of electrical unloading, it is not unreasonable to attribute such a breakdown to vibration.

As a consequence of the above, we have need to make frequent examinations of this control equipment and the steering gear units are now changed over daily as a further check that we have full steering facilities at all times."

Further examination of the problem later showed that the contactors were vibrating, causing a short circuit and failure of the fuses.

Further reports indicated inadvertent slow down and shut down of the main engine *via* the bridge control system, initiation of spurious alarm conditions believed to result from vibration at the transducers, collapse of the auxiliary boiler brickwork, collapse of radar mast and complete inability to use the work shop lathe at sea due to transverse vibration.

Early subjective appraisal by ship's staff tended to indicate that the propeller was the main source of the trouble, but the temporary expedients of maintaining the aft peak full of water and providing stiffening for selected units gave no improvement. Further, the opinions were that since the vessel had entered service the vibration had steadily worsened to such an extent that both shipbuilders and owners agreed to undertake investigations.

Vibration Investigations

Initial vibration trials, conducted in the ballast condition, suggested that both main engine and propeller were contributory factors, but the former was the principal cause of the instrumentation faults, a great deal of which was mounted in way of the engine on 'tween decks port and starboard.

Attention centred on the transverse engine vibration and the shafting axial vibration characteristics.

Using an ultra-violet recorder and associated transducers, surveys through the engine speed range with measurements at ends of the engine and cylinder tops followed by further surveys at the fixed speed of 110 shaft rev/min gave sufficient information to establish that the main engine exhibited a fifth order transverse vibration with the critical positioned at 9·167 Hz (110 rev/min). At the ends of the engine the maximum measured amplitudes were ±0·56 mm and the engine was moving in a skew mode with a nodal position at its centre. These results are presented in Figs 5 and 6, together with vertical and fore and aft engine movements through the speed range (Fig. 5) for the same measurement positions. Vertical vibration records indicated that the engine also heaved vertically about its centre with amplitudes of ±0·375 mm at its ends at 110 rev/min.

The main engine is connected to the hull structure by ten horizontal tie girders which are equally spaced over the length of the engine; these also form the supports for the upper platform on the port side. It was suspected that the ties were not fully effective, particularly as there had previously been failures of the securing bolts. Measurements on each girder, also presented in Fig. 6, gave clear evidence that of the more important end members only one was truly functional.

The transverse engine vibration effects were obviously being transmitted directly to the 'tween decks in the engine room, but to clarify the mechanisms further, surveys of the double bottom and shipside were conducted. Both fifth order transverse and eighth order cylinder vibration was being transmitted to each 'tween deck vibration amplitudes reaching maxima and minima midway between and at each deck level respectively. Consequently, vibration on the engine room 'tween decks generally increased with distance inboard from the shipside.

The axial vibration characteristics presented in Fig. 7, were obtained with transducers mounted on the top of the separate thrust block. The system critical frequency was established at 9·1 Hz, excited within the engine speed range by eleventh order engine forces and eighth order propeller and engine forces and, additionally at 109 shaft rev/min by the fifth order transverse forces. At the latter engine speed the vibration amplitude at the thrust block was ±0·313 mm.

In addition to the measurements described, many surveys were made on decks in way of and on equipment influenced by vibration. The principal purpose was to ascertain effective methods of providing local stiffening. For the majority of units analysis gave clear solutions, but for the switchboard, due to its length, the vibration mode was complex. The main body adopted a sinuous form with maximum amplitudes at its centre and two ends and minimum values at quarter lengths. This mode was further complicated by a large panel vibration at the centre of the length.

For all the ancillary equipment investigated the main vibration source was the main engine.

Discussion and Corrective Measures

Discussions with the shipbuilders ensued immediately on conclusion of the vibration investigations. For local stiffening such as in after peak tank, switchboard platform etc. agreement was reached and the measures were immediately instituted with considerable success. The main switchboard isolator links had pins fitted which prevented further creep.

The inherent engine vibrations, however, needed more careful appraisal and with the vessel in port for less than 72 h at any time other than dry docking, major corrective measures could only be introduced gradually.

There seemed little doubt that the rocking of the thrust block, which was sufficient to cause concern with respect to the holding down arrangements, was transmitted forward and, also

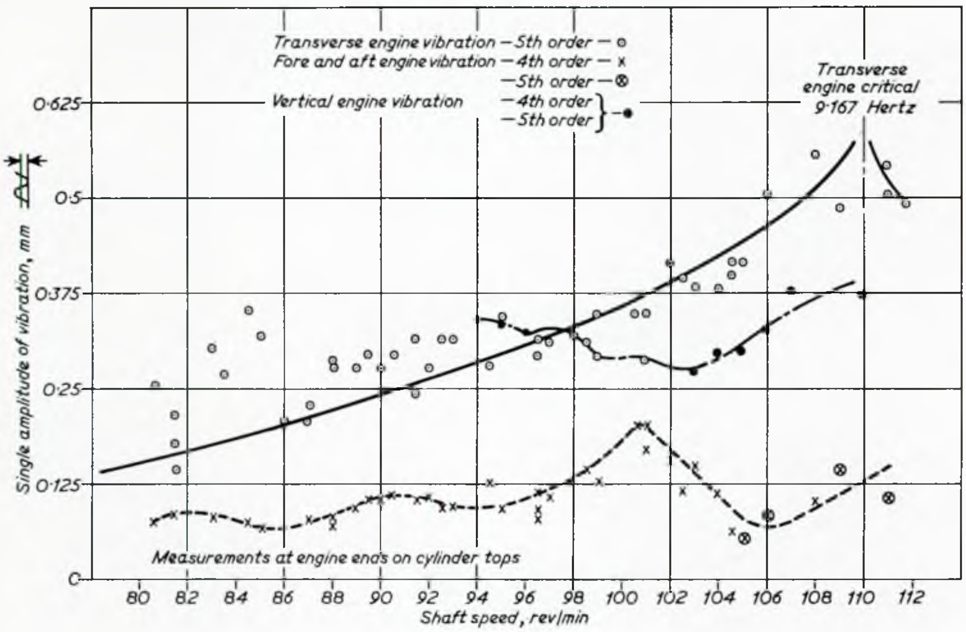

FIG. 5—*Transverse, vertical and fore and aft main engine vibration*

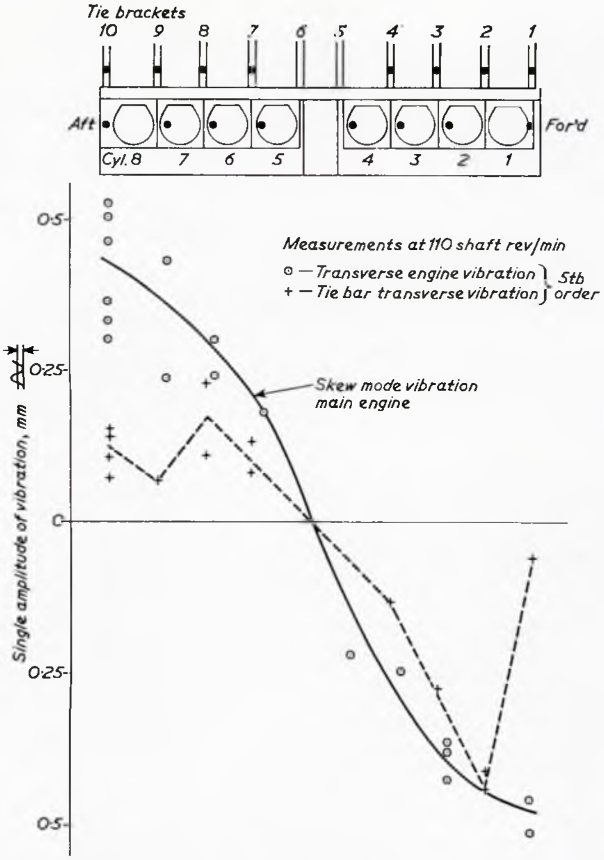

FIG. 6—*Survey of main engine cylinder tops and main engine tie-bars transverse engine vibration*

induced the rocking of the main engine, although this had not been proved. Discussion concluded that the elimination of this axial vibration would also reduce engine induced vibration at both forward and aft bulkheads and consequently some unacceptable levels experienced in the accommodation and on the bridge. It was, therefore, decided with the shipbuilders, that an axial vibration detuner be fitted at the forward end of the crankshaft.

The rocking motions of the main engine and the thrust block had undoubtedly influenced the security of the holding down arrangements. A full programme was introduced to progressively check these and to recheck at regular intervals. This programme has continued to the present with the shipbuilder's representative visiting the vessel frequently.

One complication encountered was that some bolts which were apparently tight were later found to have nuts which were seized on the bolts. Until recognized, attempted tightening, ringing the bolts and checking the chocks gave the impression that their state was satisfactory. This condition predominated at the ends of the main engine and the bolts concerned had to be burnt off. The practice now adopted is first to unscrew the nut and then retighten it to instructions.

Following the taking of the vibration measurements, a detailed examination of the main engine to hull tie girders showed these, in some instances, not to be in intimate contact with the engine frame. For the engine end girders in particular, the bolts were subjected to a fluctuating stress condition coupled with bending effects resulting from the vertical vibration of the main engine. This combination clearly accounted for the earlier failures of bolts.

Releasing the tie girder securing bolts along the length of the engine confirmed former impressions; gaps of 3·25 mm existed on the girders between face plates and the engine, but more important the faces were not parallel.

Immediately following the correct reinstallation of the main engine tie girders, the axial vibration damper was fitted and a further series of vibration measurements obtained. The amplitude and frequency characteristics with the axial damper inoperative remained substantially as presented in Fig. 5; the tie girder improvements making no appreciable change to these results. Inclusion of the damper, however, practically eliminated the vertical heaving of the main engine with amplitudes at number 1 and 8 cylinder tops (110 shaft rev/min) being reduced to less than 0·001 mm. This was coupled with reductions in the vibration levels experienced in the accommodation and on the bridge. Unfortunately, transverse engine vibration increased from $\pm$0·56 mm with the axial damper inoperative to levels of $\pm$1·125 mm with this unit in use. It can only be assumed that the interaction of the two forms of vibration limited the transverse amplitudes. Subsequent reports from chief engineers, however, stated that there were no further tie girder bolt failures and in this both the careful reinstallation of the girders and the removal of the fluctuating bending effects on the bolts contributed.

The increased levels of transverse engine vibration and possible cures were discussed with the shipbuilder. In the latter context the feasibility of fitting hydraulic damping was considered but thought to be impracticable. In the event, no changes were made.

It was the intention to release all main engine to hull tie girders and observe the effects on vibration at main engine

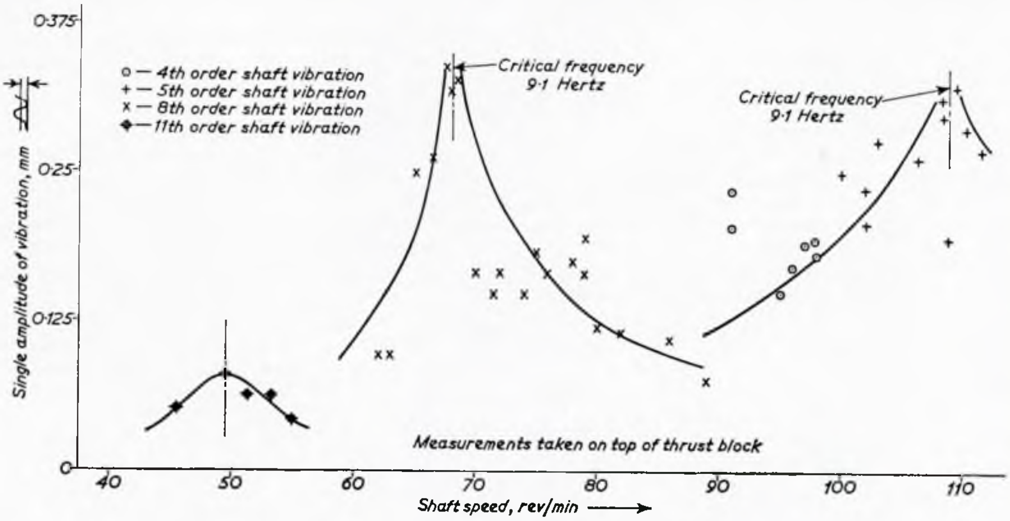

FIG. 7—*Axial shaft vibration characteristics*

rev/min up to 115. This has not been realized due to operational reasons. However, with the deterioration in hull smoothness common in this class of ship, the main engine rev/min are lower than they were originally with consequent reduction in transverse engine vibration.

The investigation did indicate that the design of the main engine seatings and attachments did not have the rigidity necessary for the forces encountered in service.

ACKNOWLEDGEMENTS

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Discussion

Mr. L. W. Tew-Cragg, A.I.Mar.E., enquired whether the authors had any experience of leakage across seals, principally due to vibration of shafts.

Mr. C. Archer extended his congratulations to the authors on their preparation of a very interesting paper and commiserated with them on having had the problems in the first place. If it was any consolation, the troubles they had described were not unusual. In Lloyd's Register Investigation Department they had had experience of two instances of unstable parallel operation of Diesel alternators. In both cases the symptoms were the same as those described. The remedy, however was much more drastic in that fitting damping bars on the alternator gave the only satisfactory solution, although considerable time was spent in trying to achieve a cure by adjustment of the engine governors.

The type of problem described in Case III which he had placed in the general category of engine or propeller vibration, was quite commonplace, as Mr. McNaught was obviously aware. The speaker said that he had been involved in an almost identical case. They had also had the red herring of engine holding-down bolts which were supposed to be tight. The bolts were tightened by means of a hydraulic nut. However, it was apparent, because of the movement of the chocks, that something was very wrong. The bolts were strain gauged and it was found that a large percentage of the tension in the bolt disappeared when the hydraulic pressure was removed. The cure was simply to apply more torque to the retaining nut while the hydraulic pressure was applied. Also, on that engine, an axial damper was fitted and proved to be effective. As in the paper, the horizontal tie bars were found to be a poor fit.

Referring to Case II, the speaker said that he was not familiar with the instruments used for measuring vibration, namely the Anson vibrometer and the Pamatrada microwave instrument. It had been found at Lloyd's Register that it was always important to take a trace of deflexion or displacement, as this gave a complete picture. Analysis of the wave form invariably gave a clue to the source of excitation, but it also gave the peak value of the deflexion, no matter how complex the wave form, and this was usually the value that caused the damage.

Recently he had been involved in an investigation where an instrument was used which only gave a direct reading of the RMS value of the wave form. This could be very misleading because the wave form was complex and the r.m.s. value was quite small compared to the peak value.

He also noted in Case II that acceptable levels of vibration had been specified. Lloyd's Register were frequently asked to do this, but it was always difficult to give a simple answer. Manufacturers' limits on components varied so greatly. A bearing manufacturer would quote an acceptable acceleration of 7*g* on large thrust bearings.

On many structural members where failure might be due to bending fatigue and where stress measurement by direct methods was not practicable, he felt that amplitudes of deflexion would be a much better criterion than accelerations. Admittedly this could be relatively difficult in that a more careful appraisal of each structure would be necessary before measuring positions and limits could be decided.

Mr. L. F. Moore, M.I.Mar.E., agreed that, as stated in the paper, the equation for calculating the electro-mechanical frequency was inadequate and wondered if the authors had had any success in refining the equation and, if so, what was its form?

Later, three resonant frequencies were quoted for the flexibility mounted systems. He was curious to know how these figures were calculated, as with most conventional mounting systems there were two coupled transverse and rotational frequencies, two coupled longitudinal and rotational frequencies, one vertical frequency and one further rotational frequency which was about a vertical axis, making six all told. These coupled frequencies assumed particular importance when there were unbalanced couples or transverse vibration of the machines.

Referring to Fig. 4, he asked the authors to state how they arrived at their figure for a "defined acceptable vibration level" and what were the vibration frequencies measured—or were they total levels? Also with this pump problem, the authors had stated that a noise level of 100 dB was measured. He wondered whether this was a flat response or whether the A weighting was used, and whether there were any discrete frequencies in the noise spectrum, such as the 29·4 Hz established from vibration measurements.

Case III was an interesting example of the necessary detective work required to sort out primary and secondary effects. In Fig. 5 the authors showed a transverse engine critical at 9·167 Hz. From the nature of the experimental points, he would have been very dubious about the suggestion that this was a resonant condition and not a forced vibration. Perhaps the authors had further information to substantiate this statement?

The Chairman, Mr. B. K. Batten, M.Sc., M.I.Mar.E., commented that in Case III the authors referred to the problems of holding-down arrangements. He wondered whether, as shipowners, they felt that engineers gave sufficient instructions to naval architects on the nature of the seatings required both for propulsion machinery and thrust-bearings.

Correspondence

Mr. F. A. Manning, B.Sc., A.M.I.Mar.E., said the paper seemed to provoke some adverse comment from the floor criticizing the elementary type of equipment that was used in one of the investigations. He was continually being advised by non-practical scientists that no vibration problem could be tackled without a mathematician, a computer and a load of electronic measuring equipment as a minimum "engineer's vibration kit". It came as a welcome surprise to find allies in the authors, who could, by intelligent use of simple equipment, economically solve the run of the mill vibration problems. The writer was fully conscious of the danger in applauding a paper mainly because it was what he would like to hear, but in this case Mr. McNaught had stated plainly that shipowners stayed in business by the application of "profit motive" and his own cost conscious approach to technical investigation revealed what he himself believed was the true engineer's function in society, i.e., the conservation of resources.

With an apology for blowing one's own trumpet, he would like to add the following case history to promote and encourage more "simple" vibration investigation. The problem as presented to him some years ago was qualitative only. The plant in question was a three-cylinder, inline reheat turbo-generator of 120 MW (160 000 shp) running at 3000 rev/min. Symptoms were described as a high level of vibration around the H.P. turbine such that control oil pipelines connected to integral governor valve chests had failed. No vibration measurements had been systematically recorded other than bearing pedestal amplitudes. A number of attempts to improve rotor balance and shafting alignment had been carried out without reducing the overall vibration levels.

The turbine unit was connected to the national grid and the plant operators claimed that during the winter on isolated occasions when the grid frequency was either higher or lower than 50 Hz vibration seemed, to them, to be reduced.

Using equivalent equipment to the authors, with the exception that the reed vibrometer was calibrated for Q in the range 25 to 100 Hz a one hour survey of measurements throughout the H.P. turbine, carried out by the writer, revealed that all vibration was principally at engine order 50 Hz and that the steam inlet assembly of the H.P. turbine consisting of six radial steam control valves vibrated in excess of 6*g* at the valve chest extremities. Nodes were found to exist at the outer circumference of the cylinder between the steam inlet belts. Bearing vibration levels at this time, were less than 0·001 in p.t.p.

A preliminary theory, at this stage by himself, was that the problem appeared to be H.P. cylinder resonance being excited from rotor unbalance with energy being fed in via the cylinder palm supports from the bearing pedestal.

Two proposals were made and adopted as follows to test the theory:

a) timber struts 3 in $\times$ 4 in section were fitted between the antinodes (the extremities of the six valve chests);
b) A vibration survey was carried out at no load from 45–55 Hz as soon as the machine could be released from the grid to check for resonance.

The struts reduced vibration to one-sixth and the survey revealed resonance in the node previously identified at a close value to 50 Hz. A u.v. recorder was used for the latter exercise since one was readily available. A permanent cure was later affected by fitting steel struts in place of timber which further reduced vibration levels to minimal values.

All investigation was carried out from start to successful completion in three weeks without planned loss of generation from the plant.

Authors' Reply

In reply, Mr. McNaught and Mr. Couchman thanked the participants in the discussion for their contributions to the paper.

Mr. Tew-Cragg had raised the subject of stern tube oil sea leakage due to vibration. The authors' company had only one experience of seal leakage not apparently caused by vibration. The vessel concerned had been in service for two years before sudden serious leakage occurred via the forward outer seal, from the secondary oil system supplying the space between the two forward seals, but no damage had resulted. An oil loss had continued from voyage to voyage and had varied from a maximum of 60 to a minimum of five litres per day. The variation was thought to have been due to influences of ship loading and vibration. The condition existed in a controlled state until the vessel was drydocked for propeller shaft withdrawal two years after its initiation. The cause of the leakage had been shaft grooving at the seals.

The discussion would have been enriched had Mr. Archer been able to expand his comments on solving Diesel alternator instability by increasing damping in the alternator windings. Theoretical studies often suggested solutions, but results from practical modification were limited and invaluable. The authors had referred to such a solution in the paper and their doubts of it in the particular case, and whether it offered a complete cure by disposing resonant frequencies, or was only an expensive palliative solution of damping the vibrations to an acceptable degree.

The equation for calculating electro-mechanical frequency (Case I), referred to by Mr. Moore was, the authors agreed, totally inadequate for a detailed analysis of the problem. It neither took account of system transfer functions nor of system damping. One of the suppliers had undertaken detailed theoretical studies but this was not available for publication. However, they could refer Mr. Moore to the paper by Brodin[2] given in the bibliography.

The authors agreed with the point made by Mr. Moore concerning flexible mount coupled frequencies, but values supplied by the manufacturers were only for those vibration modes expressed in the paper. The manufacturer's quoted frequencies were for static conditions while those presented in Case I were for dynamic conditions and were somewhat higher.

Messrs. Archer, Moore and Manning had referred to the type of instrumentation used in the investigations. The authors' approach here could not be better expressed than by the opening remarks in Mr. Manning's contribution. It was pleasing to find one who shared their philosophy—perhaps best described as obtaining the most from the least. This had to be the approach of any good management system, whether applicable to the more important company business or a vibration investigation. In part, this was the basic aim of all modern management techniques.

The Anson vibrometer was a simple extending reed instrument giving only knowledge of the principle frequencies, while the Pametrada microwave instrument gave a measure of vibration levels in decibels r.m.s. and also total noise level in decibels r.m.s. The advantage of both instruments was that they were pocket size. Where extensive investigation was needed, as with Case III, which involved both ship hull and main engine vibration, ultra violet trace recording coupled with integrating amplifiers and C.E.C. transducers were considered to be appropriate instrumentation.

The specification of acceptable vibration levels was discussed by both Mr. Archer and Mr. Moore. This was always difficult.

Realistic practical safe levels were the aims, not limits, which were rigorously imposing. Lack of information and experience often prevented such assessments, a state which could only be corrected by undertaking considerable research work into a particular sphere of vibration. With respect to Mr. Archer's reference to the use of amplitude as a criterion of acceptability, the authors felt that this had application, for example, for large plate panels where specific knowledge of stress was required, but invariably in a ship investigation, time was a factor and prior clear assessment of the value and justification for the approach was essential.

For the problem of the lubricating oil pumps (Case II), with a destructive frequency of 29·4 Hz, the pump running speed, the safe practical vibration level was assessed at 6 ft/s^2. This was calculated from a knowledge of satisfactory operating levels on similar installations and from the practical experience of both owners and suppliers. The acceptance level specified was realistic, as a recent vibration survey showed amplitudes below this throughout the pump and piping systems.

Mr. Manning's use of wooden struts as an aid towards solution of his turbine problem had close parallel with the actions taken by the authors on the lubricating oil pumps. Here again, their use had reduced the destructive vibration levels and in fact the vessel had operated for at least one voyage with these in place.

Main engine seatings and holding-down bolts, referred to by both Mr. Archer and Mr. Batten, were extremely important and, unless the shipowner paid particular attention to these areas, suspect design could result. Over a period of years the authors' company had specified in detail their requirements in the ships they built and few troubles had resulted. They had been spared many of the difficulties encountered with some pre-war vessels, where chocks had bedded into the tank top and caused serious misalignment in the engine. For ships purchased at the near complete stage, faults in seating design and holding down bolt arrangement could be inherited. The lack of stiffness in the main engine seating and double-bottom structure of Case III was an example, of which the shipbuilders were aware, and it was considered that improvements had been made in subsequent vessels. Holding-down bolts had also been troublesome and the vessel had been used by the shipbuilder as a research test bed. The most interesting aspect of these investigations was the surprising rate of extension of new tensioned bolts, which accounted for the rapid slackening experienced under dynamic conditions. The advised techniques for tightening now allowed for this condition by appreciably increasing the initial tightening pressures.

The authors could assure Mr. Moore that the vibration measured on the main engine in Case III was an engine skew mode. This appeared to be a feature of the large eight-cylinder slow speed marine Diesel engine which, in their fleet, they had now encountered twice and on differing engine types. Fortunately in the second instance, there were no detrimental effects. The authors recognized Mr. Moore's particular concern in that there was always the possibility that initiation resulted via the hull structure and hull/engine ties. The authors had taken great care to clarify this condition during the investigation.

THE DESIGN AND OPERATION OF SUBMERSIBLES

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In this paper the author discusses the design and operation of submersibles as a means of servicing underwater oil and gas wells on the U.K. continental shelf in water depths of more than 300 ft. He relates his discussion to design studies, undertaken at University College, London, of a manned submersible of some 60 tons submerged displacement, which he calls the "Pobble". The concept of the "Pobble" is that it should be capable of operating from a land base independently of a mother ship, for which purpose the author advocates the use of a recycle Diesel engine for propulsion. The UCL studies indicate that with this type of propulsion plant it would be possible for the "Pobble" to transit between the land base and a well area 150 miles or more apart at a speed of at least eight knots, and to achieve an endurance of five days or so.

The author suggests that the production of oil and gas from wells in deep water will necessitate the development of an underwater station including a pressure-tight house in which personnel can work when necessary. Although the design is based on this idea, so that it can carry passengers and equipment and transfer them to and from the house, it is also capable of undertaking light engineering and surveying work. The author has not yet been able to evaluate initial and operating costs, but considers that the "Pobble" could be made an economic proposition.

Professor Rydill

INTRODUCTION

In this paper the author gives a very personal view of the design and operation of submersibles partly because he considers that it is impossible in a brief paper to do justice to the subject in a general way – the submersible field today being so wide and lacking in any clear direction of development – and also because, by not being tied to a strictly factual approach, he can, he hopes, provoke some controversy.

The author believes that there is a need for controversy about submersibles. A review of any of the many survey papers on the types and uses of submersibles(1) demonstrates that they have received – and are receiving – a great deal of attention which is rather out of proportion to the role that they have up to the present played in ocean engineering and research. Most of the submersibles available today are limited in their capabilities and, due in part to their need for the support of a mother ship, costly to deploy. In consequence potential industrial users have had insufficient incentive to employ their services and, because the engineering work to be done has up till now been within the capability of commercial diving, have been able to forgo those services. Would the money that has been spent on the development of submersibles and their support facilities have been more wisely used in other areas of ocean engineering in which the same investment could have yielded a better return in the short term?

Nevertheless, there will in the future be a place for the submersible in ocean engineering as an indispensible and not necessarily costly work boat for carrying out engineering and other tasks all the year round in depths beyond the reach of commercial divers. Divers will no doubt be used whenever practicable, but it is difficult to conceive that the depths at which day-to-day diving operations can be effectively, economically, safely and reliably carried out are likely to be much in excess of 300 ft.(2) Perhaps for this country the day of the submersible will arrive when it becomes necessary for strategic or political or international shortage reasons to face up to the formidable problems of producing oil and gas from wells in deep waters on our continental shelf. (In this paper the word deep is used to mean water depths in excess of 300 ft.)

The author accepts that submersibles also have a role for oceanographic investigations in the sea and on the sea floor, but is not convinced that the considerable sums of money involved in conducting extensive undersea research programmes might not be better invested in other less expensive forms of oceanographic research. It is fully appreciated that submersibles can provide information that cannot be obtained in any other way, but deciding on the priorities between the many choices that are made possible by modern technology is a problem not particular to oceanography.

There are other roles in which submersibles can be used, including: marine geological work; marine archaeological work; salvage; fishery investigations(3). But in this paper attention is confined to the use of submersibles as work boats for servicing deep undersea oil and gas wells on the U.K. continental shelf.

USE OF SUBMERSIBLES

The customary approach to the design of a new product is to start with a set of requirements and by working through a fairly standard design process to deduce the capabilities that the product should provide and the features that should be incorporated in it, so as to meet those requirements. Often today the requirements are derived from market research, operations research or needs analysis. Where the product is to be a submersible for use in servicing wells in deep water, however, the customary design approach is unhelpful because techniques for the production of gas and oil from deep water wells are still being developed and could depend on the capabilities offered by the submersible and the cost and dependability of its support. In this situation one has to make a back-to-front approach by starting with an assessment of what submersibles can do and deducing the requirements particular to the application under consideration.

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The general capabilities that submersibles can offer are:

a) independence of sea surface conditions;
b) mobility;
c) deep diving;
d) submerged endurance;
e) detection;
f) manipulation;
g) communication.

Not all present-day submersibles provide all these capabilities adequately, but they are possible to achieve.

None of these capabilities requires men inside the submersible, though some are undoubtedly improved by their presence, and there is a good case for the use of unmanned submersibles. The author, however, intends to exclude the unmanned submersible from consideration in this paper on the grounds that it necessitates the use of a mother ship and introduces the launching and recovery problems which, in their turn, impose limitations on the sea surface conditions in which the system could be used. In fact, the unmanned submersible cannot exploit the first of the capabilities listed above, and yet this is in a way the most important characteristic of the submersible. Certainly for a well-servicing system for use in the North Sea, for example, dependence on sea surface conditions would rule out all-the-year-round use. Users would be justified in doubting the worth of a system which could not be relied on to provide the service whatever the weather conditions. There is also the point that a mother ship is expensive – probably much more so than the unmanned submersible – so that the cost of hiring both would be much higher than if the submersible could be used on its own.

It will be seen that the same arguments apply against the use of a manned submersible which requires a mother ship to support it. Further, since the mother ship has to launch and recover the submersible, its size and weight have to be limited, severely constraining the design. The operations of launching and recovering the boat in rough seas also present a formidable problem and, although the considerable ingenuity being applied to the problem might well enable safe launching and recovery to be carried out in more severe sea conditions than at present, there will still be times when these operations will be quite impossible. Yet all the time such storms rage at the surface, the water at the bottom will be undisturbed.

To exploit the calm of the sea bed, it would be necessary to develop the potential of the submersible for prolonged endurance submerged, and this in turn entails finding an energy source of sufficient capacity. Clearly exotic energy sources such as the nuclear reactor and the fuel cell can be ruled out. Whatever their ultimate development might be, they are at the moment too expensive for the submersible application and are likely to remain so for many years. On the other hand the various types of battery used in almost all submersibles up till now are so heavy and bulky as to severely limit the capacity of the installation that can be accommodated. Most present-day submersibles have in consequence been able to achieve endurances of only ten hours or so, and these only at low speeds, so that the distances which can be covered have been limited to about 20 to 30 miles. The submersibles have then had to surface for battery recharging or replacement, a time-consuming business that also throws away the advantages of completely submerged operation. From the user's point of view, this sort of performance is not very attractive.

It might seem that a worthwhile improvement in submerged endurance could be obtained by equipping the submersible with a Diesel engine fitted with a snort mast so that the vehicle could transit at shallow depth from a land base to the well area using the engine and then dive on battery power to carry out its task. However, experience with military submarines so fitted is that this capability is limited by sea surface conditions. The submarine has to keep near to the surface and depth control becomes more difficult as the sea gets rougher. In any case, at some stage in rough seas the head valve on the snort mast (the function of which is to keep to a minimum the water drawn into the submarine) closes so frequently that snorting has to be abandoned. Since the submersible would be even more at the mercy of surface conditions than the much larger military submarine, this alternative cannot be regarded as a suitable means of increasing submerged endurance.

Nevertheless, it seems clear that the way to increase submerged endurance for submersibles lies in the direction of utilizing the Diesel engine and it is encouraging that Ricardo and Company Ltd. have been carrying out research on the use of the re-cycle Diesel engine as an underwater power source (not particularly for the submersible application, but quite suitable for the purpose)[4]. So far the work has been on a Perkins 4.108M Diesel engine continuously rated at 36 hp at 3000 rev/min, and although this is too small for the submersible in mind, the work could be applied to larger engines if the need were established. Basically, the Ricardo system involves the use of oxygen (from oxygen gas bottles) which is diluted with some of the exhaust gas, appropriately cooled and dried, and fed into the engine intake manifold; the remainder of the exhaust gas is compressed and pumped overboard. The main problem in the re-cycle system design is to maintain engine efficiency while avoiding abnormal mechanical or thermal stresses, and the main problem in its application to submersibles would be to reduce or isolate airborne noise. Clearly, therefore, in the submersible application there would have to be a separate engine room well insulated, acoustically and thermally, from the rest of the boat.

The trend of the author's thoughts on submersible design is, thus, towards a work boat powered by a re-cycle Diesel engine for transiting entirely submerged from a land base to the well area, where the required operations would be undertaken, after which the boat would return entirely submerged to the land base. Thinking in terms of the U.K. continental shelf as the area in which the submersible would be employed, the distances to be traversed in reaching the well area from the land base might be 150 nautical miles or even more. Thus another problem is encountered: transit speed. The maximum underwater speed of the battery powered submersibles used until now has been about five knots. Evidently this would be much too low for the proposed submersible, as it would take at least 30 hours to get to a well area 150 miles from base and at least 30 hours to get back. (It is necessary to say "at least" because currents would reduce the speed relative to the ground to less than five knots). Even with ten knots maximum speed submerged there would be a total of at least 30 hours transit time there and back, and ten knots transit speed demands eight times the power needed for five knots though, other things being equal, only four times the fuel. However, by careful attention to hull shape and streamlining, minimum excrescences and good propeller design, the power/fuel requirements can be kept within practicable bounds.

Returning to the list of capabilities, mobility has been touched on in talking about transiting to and from the well area. Once there, the submersible would need to be able to position and orientate itself relative to its work with precision. The capability to inch up and down forward and back, and also to angle the boat, can be readily provided using the propeller in conjunction with transverse thrusters or similar devices.

As regards diving depth, there is no reason for aiming for more than about 750 ft capability in a boat intended for operation on the U.K. continental shelf. The emphasis, particularly in the U.S.A., on much greater depths has in the author's view been misguided as it has driven many designers to use spherical pressure hulls, and there could hardly be a more inconvenient shape from the aspects of space utilization, ergonomics and crew comfort (crew comfort is a far from negligible consideration if one is thinking in terms of working periods of several days). The author's approach would therefore be to use a cylindrical pressure hull with domed ends enveloping everything except the essential minimum of external items. With the relatively modest maximum diving depth of 750 ft envisaged, this shape of hull is entirely practicable. One other point in the context of diving depth: because the concept of the submersible is to work in depths beyond the reach of commercial diving, there is no point in providing diver lock in/lock out facilities.

To finish this section the detection, manipulation and communication capabilities have been grouped together as in many ways they are the most important features of any submersible since they determine its ability to do useful work in the almost impenetrable darkness of the sea bed in deep waters on the U.K. continental shelf. The other capabilities discussed are mostly concerned with getting to the work area and back again and while they are important in deciding on the overall system requirements, ultimately it is the ability of the submersible to do useful work that will most influence decision on its worth to potential users. This group of capabilities encompasses sophisticated electronic equipment (needed for: navigation instruments; position and attitude sensors; television; search sonar; high discrimination sonar; underwater telephone) and equally sophisticated electro-hydraulic equipment (needed for the manipulators). These are bound to be costly but there is no escaping the fact that the investment would have to be made to achieve sufficient worth.

It is in considering this group of capabilities that one returns again to the chicken and egg nature of the overall problem. Up till now the direct question: what work should the submersible be able to do? has not been faced. The author has no expertise on the problems faced by industry in exploiting oil and gas wells in deep water, and so hopes that he will be excused if, in trying to answer the questions, he misrepresents these problems. Now while it is a remarkable feat to discover oil and gas wells under the sea floor in deep water, the really difficult problems arise when exploration ceases and production starts. It is easy to see how oil is recovered in water which is shallow enough to allow the production platform to be erected with its feet firmly planted in the sea floor, but it seems unlikely that the fixed platform can be used when the water depth is much in excess of 300 ft. If this is so, there appears to be only two possible solutions:

1) to use a floating production platform;
2) to use a production station on the sea floor.

In both, the oil or gas can either be brought direct to the surface or, if the well area is not too far from land, piped ashore. Since the U.K. continental shelf is being considered, the latter will be assumed.

If a floating platform were used, it would have to be moored more or less directly over the well area. There would, however, always be a risk that in very bad storms the platform might be so displaced that its connexions to the wellheads would be broken and control of the wells lost, with the consequent danger of pollution. There would also be difficulties with routine servicing and maintenance, and particularly with repair.

With the underwater station, all bar the essential minimum of equipment would be housed inside a pressure-tight envelope. Although the aim would be to make operation fully automatic, with control effected from the shore, means would have to be provided to enable personnel periodically to work within the envelope to undertake maintenance and repair. Since the station would be independent of sea surface conditions and its equipment isolated from the sea, it should be safe and dependable and introduce little risk of pollution.

Although it might be rash to predict how things will develop in ocean engineering, the author is prepared to pin his faith on the guess that the underwater station will be used for deep water oil and gas production. The concept of the underwater station is not original[5,6], and also the scheme is likely to be very costly. However, to produce oil or gas from wells in deep water is bound to be costly – so much so that, as things stand at present, it is hardly conceivable that production could be economic. But, in time, it may become politic or necessary to exploit the U.K.'s deep water resources, and this requires that development of the necessary techniques should start now.

In what follows the author proceeds on the basis that the underwater station would provide the main source of work for the submersible being considered. With this approach the primary role of the submersible would be periodically to transport men, equipment, supplies, etc., between the land base and the pressure-tight "house". It could also carry out preliminary surveys and preparation of the site for the house and, subsequently, maintenance of the few fittings outside the house and inspection of the collecting pipelines and those connecting to the shore. Since the concept depends completely on the safe and timely delivery of men to and from the house, no interruption by bad weather could be permitted, and this points directly to the use of a submersible able to operate without mother ship support.

Before going on to the main purpose of this paper and describing the concept of operations and design of the proposed submersible, the author has given it the name "Pobble", after Edward Lear's character who lost his toes when swimming at sea.

CONCEPT OF OPERATIONS

In this section the various suggestions from the preceding sections are collected together and discussed. They are:

i) to operate from a land base independent of a mother ship;
ii) to have minimal surface performance consistent with the need to operate safely on the surface only when leaving and returning to base;
iii) to transit submerged to and from a deep water well area up to about 150 miles or so from base;
iv) to transport men to and from a submerged pressure-tight house at the well area and transfer them to and from the house – similarly for stores and equipment;
v) to undertake light engineering work on underwater equipment external to the house and in the well area;
vi) to inspect and survey pipelines, sea bottom conditions etc.

To what extent the light engineering work and/or inspection and surveying functions could be combined with the passenger/stores transportation function would depend on how far the well area was from the land base, on the number of crew and passengers and amount of stores to be carried and on the features incorporated in the "Pobble". In deciding on these features there would be a trade-off between the initial and operating costs of the boat on the one hand and its maximum transiting speed and total submerged endurance on the other. Since the periods occupied in transiting to and from the well area will mostly be dead time, there is considerable pressure to make the transiting speed of the "Pobble" as high as possible, but because the power required for propulsion varies nearly as the cube of the speed, the price of increase in speed mounts very steeply, both as regards the size of the propulsion unit and the fuel stowage required. (The term fuel is used here to embrace Diesel oil and oxygen). The corresponding price of total submerged endurance is not so clearly defined since to carry the men at all is an important factor in determining boat size: clearly the longer they have to stay on board the larger is the provision that has to be made for life support, but this is less important from the aspect of hotel load in a re-cycle Diesel-powered boat than in one powered by batteries. Nevertheless the dead time spent in transiting determines the minimum submerged endurance that has to be catered for, which, by the standards of most present-day submersibles, is already considerable. To enable the boat to undertake a reasonable amount of other work (having to spend up to two days just getting to the work area and back obviously makes this at least desirable), one is obliged to think in terms of a total submerged endurance of about five days.

To decide on how the balance should be struck between these various considerations, it would be necessary to undertake design studies in some detail. Preliminary studies by the author's group at University College London (where a postgraduate course is run which includes lectures and project work on the design of submersibles) indicate that to provide the capabilities discussed, the "Pobble" would have a submerged displacement of about 60 tons. Much more work would have to be carried out in order to validate this prediction and to yield an order of cost estimate to give some idea of the economics of the concept.

Although transport and transfer of men and equipment to an underwater oil or gas production station in deep water is central to the concept of the "Pobble", the UCL studies demonstrate that the approach of using a re-cycle Diesel engine for propulsion in conjunction with a well streamlined form is of value in its own right. It leads to a much higher degree of mobility than has been achieved by any submersible to date, a feature that is worth having whatever activities the boat is intended to undertake.

DESIGN ASPECTS

It is not the intention of this section to provide a guide to the design of submersibles, but rather to comment on particular aspects of their design, illustrated by reference to the UCL studies and drawing on lecture material given at UCL, in the hope that this might be of general interest. Despite the considerable activity in submersible design and operation in the last decade, there are few really useful textbooks or technical papers on submersible design and it is to be hoped that someone will come forward to fill this vacuum. Some basic guidance on safety and operational requirements has been given[7], but this is not much help as regards submersible design proper.

Weight and space

Unlike the deep diving submersible, the design of which is weight-governed because of the considerable proportion of the total buoyancy that has to be invested in supporting the weight of the pressure hull, the relatively shallow diving "Pobble" is space-governed. This means that its size is determined by the space that has to be provided for personnel, stores, operations room, ballast and trim tanks, propulsion and auxiliary machinery, fuel, etc.; the buoyancy of the envelope yielding the required volume is more than enough to support the weight of the pressure hull, its contents and external structure and fittings, so that there is adequate weight in hand to provide sufficient solid ballast to achieve adequate hydrostatic stability and longitudinal balance. The difference between the two types is important because in the deep diving one emphasis has to be placed on making the pressure hull as small as possible so as to achieve minimum pressure hull weight, whereas in the shallow diving one there is no need for this emphasis. The designer of the deep diving submersible is driven to using a spherical pressure hull with its poor space utilization characteristics; to locating as much machinery and equipment as possible outside the pressure hull; and often to the use of buoyant materials to achieve weight/buoyancy balance and/or adequate hydrostatic stability. In contrast, the designer of the shallow diving submersible can use a cylindrical pressure hull and thereby achieve a reasonable utilization of the available space, can locate the majority of machinery and equipment inside the pressure hull, and has no need to use costly and unproductive buoyant materials.

Flotation and hydrostatic stability

A difficult problem in the design of submersibles is deciding on an adequate reserve of buoyancy in the surface condition, i.e. how to provide main ballast tanks of sufficient capacity to:

1) give enough freeboard to avoid water being shipped through the access hatch;
2) provide enough hydrostatic stability for safety and to avoid heavy rolling in waves;
3) reduce risk of collision due to the very small silhouette (this is particularly relevant to the "Pobble", which might have to make its way on its own when on the surface near its land base).

Here is one of the many conflicts that arise in submersible design. To provide large main ballast tanks is to increase the size of the submersible, and thus propulsive power and fuel requirements; if the tanks were located outside the pressure hull and shaped to give a good surface waterline, drag would be increased and propulsive power and fuel requirements further increased; if the tanks were located inside the pressure hull the demands on internal space would be increased; if a conning tower were fitted to reduce the risk of shipping water, it would have to be faired by means of a "sail", and while this would increase the silhouette it would also increase both top weight and propulsive power requirements.

The need in the "Pobble" to achieve the highest transit speed practicable puts heavy emphasis on achieving low drag. For this reason the main ballast tanks should be located inside the submersible and a retractable conning tower should be provided, so that the "sail" can be dispensed with. The latter step is of course a complication, but can be justified by the reduction in propulsive power and fuel stowage thereby made possible. There is also the advantage that by dispensing with the "sail" the heeling moment applied to the boat in a cross-current would be much reduced.

A reserve of buoyancy of at least ten per cent has been advocated for submersibles[7] but in a boat like the "Pobble" (which will have to surface on approaching harbour entrances, where the water can be quite rough) 15 per cent should be provided. To work to this proportion with internal main ballast tanks would eat even further into internal space, since it entails providing a tank capacity of about eight tons against about $5\frac{1}{2}$ tons with ten per cent reserve of buoyancy, but it is a prudent investment.

As regards the hydrostatic stability of the "Pobble" when submerged, one important consideration is the effect of movement of personnel on the attitude of the boat when it is stationary or nearly so and, because the levers are so much larger, fore and aft movement is more likely to be the determinant than transverse movement. Without quantitative investigation it is not possible to say whether this aspect would determine an acceptable lower limit for BG, or whether it might be determined by the forces involved in manipulative work under water or by the transient phases of surfacing/submerging. These phases have led to difficulty in some Diesel-propelled military submarines owing, amongst other things, to the free surface effects of water in the main ballast tanks. Surfacing is more important from the stability aspect than submerging because the former operation tends to be slower and during it the boat becomes increasingly exposed to wind and waves.

To achieve adequate hydrostatic stability both on the surface and when submerged and to effect longitudinal balance, it would be necessary to provide solid ballast capable of being disposed as required. A suitable form of ballast is plastic coated lead, since it could then be stowed inside the internal tanks right down at keel level.

Another aspect which must be considered is the provision of the means for the submersible to effect an emergency surfacing. It has been specified[7] that arrangements should be provided for solid ballast to be jettisoned manually or for tanks to be blown so as to make the boat positively buoyant. The author prefers the latter, and in the "Pobble" would arrange for the foremost main ballast tank to be capable of being blown in an emergency at full diving depth with high pressure air. In this way about two tons of positive buoyancy could be provided well forward which would bring the boat to the surface with a bow up angle and with the access hatch well clear of the water surface.

Manoeuvring and control

Both hydrostatic and hydrodynamic aspects are treated under this heading because in the submersible the forces from both these sources can be invoked to achieve the desirable handling qualities under water.

Considering hydrostatic aspects first, trim control in the "Pobble" (in the submarine sense of keeping weight and buoyancy equal and the LCG and LCB in the same longitudinal position) would be provided in the conventional way by using:

a) a compensating tank near amidships which could be filled or emptied as necessary by pumping to and from the sea;
b) a forward and an after trim tank between which and the compensating tank water could be transferred by pumping, but with no connexion to the sea.

The function of the compensating tank is to cater for changes in weight, e.g. due to fuel consumption or to unloading of personnel and stores, and in buoyancy, e.g. due to variation in density of the sea water. The function of the trim tanks is mainly to cater for changes in LCG (changes in LCB are negligible). As these various changes will be relatively slow, the pump capacity needed is small. To conserve space inside the boat the compensating tank can be combined with the after main ballast tank, which would be sized so that it was empty at the start of a mission when the boat would be relatively heavy because of its full load of personnel, stores and fuel.

Turning now to the hydrodynamic aspects, and considering first manoeuvring and control of course and depth when transiting, there is no reason for designing the "Pobble" to be very manoeuvrable at speed, since the transit run can mostly take place at constant depth on a constant course. To ease the task of the pilot, the boat would be fitted at the after end with stabilizer fins sufficient in size to give good course and depth stability. For depth and course changing the stabilizer fins would be fitted with flap rudders and hydroplanes and a suitable configuration would be an inverted "Y", since this would give more force in turning than for depth changing and also would avoid projection below the keel line. Because of the extra drag involved, the author would not advocate fitting forward hydroplanes even though this omission would (in the absence of other control devices) result in poor depth control at slow speeds, when after hydroplanes cease to be effective[8].

In any case, it is necessary in a boat required to mate with the house at the underwater station to provide special means for position and attitude control at very slow speeds relative to the water. (Because of currents, the boat will have a velocity relative to the water even when stationary relative to the ground.) The mating problem has been investigated in the U.S.A. in connexion with the design of the DSRV[9] but clearly one would wish for a less expensive solution than that reached in those very costly vehicles. The author's solution would be to use a combination of jet flaps on the after control surfaces and vertical and horizontal ducted thrusters forward, which would be brought into use when there was a need to control the position and attitude of the "Pobble" precisely and to keep it stationary relative to the ground. This capability would be useful not only for the mating operation but also for inspection and light engineering work underwater. The ability of the proposed arrangement to maintain the submersible in position and attitude in the current speeds likely to be encountered in deep water on the continental shelf would determine the size of the installation.

The jet flap concept[10] involves ejecting water from the trailing edge of a control surface at an angle to the chord so as to increase lift. It is relatively more effective at slow speeds of advance and thus is a good way of tackling the problem of position and attitude control. The control arrangements for the jet flaps aft and transverse thrusters forward would be designed so as to apply to the boat a couple or a force or any combination of these. The system could cope with a certain amount of out-of-trim of the boat, but clearly it would be desirable for it to be kept in trim as nearly as possible using the compensating and trim tanks.

Because of the importance of minimizing the crew and the demands made on them, it would be necessary to bring together into one position the controls for the main ballast tanks, compensating and trim tanks, propulsion, rudders and hydroplanes, and transverse thrusters and jet flaps. The control panel would also have to include displays of speed, course, depth, distance from bottom, heel, trim, angle in azimuth and sonar information. Adjacent to this panel would be another displaying tank contents (including fuel), state of H.P. air and oxygen bottles, condition of battery, hydraulic system pressure, and so on. All this adds up to quite a formidable specification, and yet one which must be faced up to in order to achieve a boat which would be safe, effective, convenient and reliable in operation.

Propulsion

Since it is fundamental to the concept of the "Pobble" that as high an underwater speed as possible should be attained, its development would entail breaking new ground as far as submersibles are concerned because few have a maximum speed in excess of five knots and most are much slower. Now although the use of the Ricardo re-cycle Diesel engine technique makes available higher powers and endurances than are feasible with batteries, the need to keep down initial and operating costs of the "Pobble" makes it important that the size of the propulsion machinery and the amount of fuel to be carried should be no larger than absolutely necessary. These in turn necessitate not only that appendages and excrescences should be kept to a minimum, but also that the hull form/propeller combination should be selected so as to give maximum propulsive efficiency. At first sight this might seem an obvious observation, but one does come across the suggestion that the aim should be to select the hull form on the basis of minimum drag alone. The minimum drag approach leads to a choice of a form with a length/diameter ratio of about five to six, and a relatively fine after end. To achieve good propulsive efficiency it is necessary to use a single propeller located at the after end in the wake of the hull, where it can recover some of the energy lost in the wake. Subject to certain limitations, the higher the wake can be made over the diameter of the propeller, the higher the propulsive efficiency will be. It follows that it can be worthwhile to adopt a hull form which is fuller at the after end than is optimal from the drag aspect. The suction field ahead of the propeller will help to avoid flow separation.

To establish what can be achieved with the maximum propulsive efficiency approach, and how the choice of hull form/propeller combination can be made, it is necessary to undertake numerical studies. In principle these cannot be based wholly on hydrodynamic considerations since, for example, the propeller speed for high efficiency is relatively low and thus tends either to cause the size and weight of the propulsion motor to be larger than would otherwise be necessary, or to require the extra cost complication, weight and space of a reduction gearbox. In the case of the "Pobble", as it happens, Ricardo's development of the re-cycle Diesel engine technique includes the use of an hydraulic pump for power transmission and this feature lends itself to the use of an hydraulic motor for driving the propeller which would be smaller than an electric motor of the same power and could provide that power at low rev/min without needing a reduction gearbox.

Studies undertaken at UCL indicate that an efficient propulsion arrangement would be to use a single 3 ft diameter propeller with a hull form of 5·5 to 1 length/diameter ratio and a very full after-body. Bearing in mind the need for a transit speed relative to the ground of at least eight knots, and the incidence on the U.K. continental shelf of appreciable tidal currents, there is a case for providing a capability for a maximum speed relative to the water of ten knots. The UCL studies indicate that the "Pobble" could achieve this speed, with ample power in hand for the "hotel" load, using a Ricardo 4.236 M Diesel engine which delivers 62 hp at 2250 rev/min at its maximum continuous rating. This engine is about 20 per cent larger in linear dimensions and at 1000 lb dry weight is about twice as heavy as the Perkins 4.108 M Diesel engine used by Ricardos in their re-cycle development. Nevertheless there would be no difficulty at all in accommodating the larger engine in the "Pobble's" engine room. The Diesel oil consumption rate (using air) is about 50 per cent greater than that for the 4.108 M engine and if this relativity were maintained under re-cycle conditions the UCL studies indicate that the "Pobble" could undertake a five day mission, including transits between land base and well area 150 miles apart, on less than a ton of Diesel oil.

In fact, the investigations show that it is the oxygen requirements which dominate the problem of achieving such a protracted submerged endurance. The "Pobble" would need to carry about 2½ tons of oxygen. This quite rules out the use of oxygen gas bottles and there is no alternative but to use liquid high test peroxide, H.T.P., of which about 5½ tons would be necessary. The H.T.P. can be stowed in PVC bags in an internal tank into which sea water can be pumped as the H.T.P. is consumed. H.T.P. is less convenient logistically than bottled oxygen gas, but it is available commercially. It is also dearer than bottled oxygen gas but the total fuel cost component would still be a relatively small proportion of the total operating costs.

With the re-cycle Diesel engine plant and H.T.P. as the oxygen source, the risk of fire could not be ruled out. Since the engine room would be quite separate from the rest of the boat and operated unmanned, means could be provided for automatically extinguishing a fire without immediate danger to personnel or to the boat, except in that the main propulsion and energy source would then be unavailable. For this reason it would be necessary to fit at the forward end of the boat an emergency battery of sufficient capacity to enable personnel to survive for up to, say, 48 hours while awaiting the arrival of a rescue ship, when the boat would surface using its emergency H.P. air blow.

Structure

With the form thus described and a submerged displacement of 60 tons, the "Pobble" would have a maximum diameter of 9 ft and an overall length of about 50 ft. The pressure hull would be a cylinder of constant circular section 9 ft in diameter closed at each end by a dome to give a total length of 36 ft. The ends outside the pressure hull would be free-flooding spaces enclosed by light plating to provide the required end shapes.

Although the design would be space-governed, there would still be a need to keep weight, including that of the pressure hull, under tight control, and particularly to ensure that enough weight could be kept in hand to provide a substantial amount of disposable solid ballast. However, there is less need than in deep-diving submersibles to aim for minimal pressure hull weight and there is, in consequence, some latitude in the choice of the type of steel used for the pressure hull plating and framing (so that priority can be given to ease and cheapness of fabrication) and in the choice of frame spacing (which can be suited to the arrangement of the boat).

Studies of the pressure hull structural design undertaken at UCL, based on the theory presented in reference 11, indicate that with the two main transverse bulkheads envisaged, overall collapse should not be a problem, and that from the local collapse aspect there would be no need to use high strength steels which are costly and difficult to weld. The studies show that it would be advantageous with the configuration chosen to use relatively low strength steel such as the Navy Department's B quality steel. This would not only give an increased margin against failure by local buckling of the plating (to which the relatively lighter scantlings possible with a high strength steel would make the structure more sensitive) but its use would substantially reduce the risk of fatigue cracking (to which a pressure hull of high strength steel would be more prone because of the closer approach to the stresses at full diving depth to the ultimate tensile strength).

The UCL studies assumed a design collapse pressure of 500 lb/in^2, corresponding to a factor of safety of 1·5 at the 750 ft full diving depth proposed. The main features of the structural design adopted for the pressure hull were:

Plating thickness: 0·75 in

Frame size: 6 in × 3 in standard T bar

Frame spacing: 30 in

Pressure hull weight: 16 tons, i.e. 27 per cent of the buoyancy.

The various tanks which occupy the full length of the boat are another major component of the structure. The forward main ballast tank selected for the emergency H.P. air blow would, in the interests of safety, be designed as a hard tank to withstand the collapse pressure. Although the tank structure is below the axis of the boat, so that its weight contributes to hydrostatic stability, its fore and aft disposition is not at the disposal of the designer and thus cannot be utilized as a means of achieving longitudinal balance. There is in consequence a need to keep the weight of tank structure, other than that of the hard tank, to a minimum and to this end the tank operating arrangements would be designed to ensure that they could never be subjected to sea pressure. The UCL studies show that on this basis the weight of secondary structure would be about eight tons, so that the total structural weight would be about 24 tons, i.e. 40 per cent of the buoyancy. The balance of 36 tons should be more than enough to support the weight of machinery, equipment, fuel, water in tanks, etc., leaving ample weight in hand for solid ballast.

General arrangement

The general arrangement considered appropriate is shown in Fig. 1. The forward transverse bulkhead is a structural bulkhead and its main function is to support the pressure hull against overall collapse. A 3 ft diameter trunk is provided just forward of amidships at the top and bottom of which are the access hatches to the boat. The purpose of the trunk is to limit the amount of water that would be admitted to the boat in the event of an accident in the operation of either of the hatches. The after transverse bulkhead, which would also be worked structurally, isolates the unmanned engine room from the rest of the boat. The foremost compartment is the control space and is arranged for two operators. The adjacent compartment is arranged for six passengers (an arbitrary choice) plus two crew members off watch, and also contains the galley and heads. The next com-

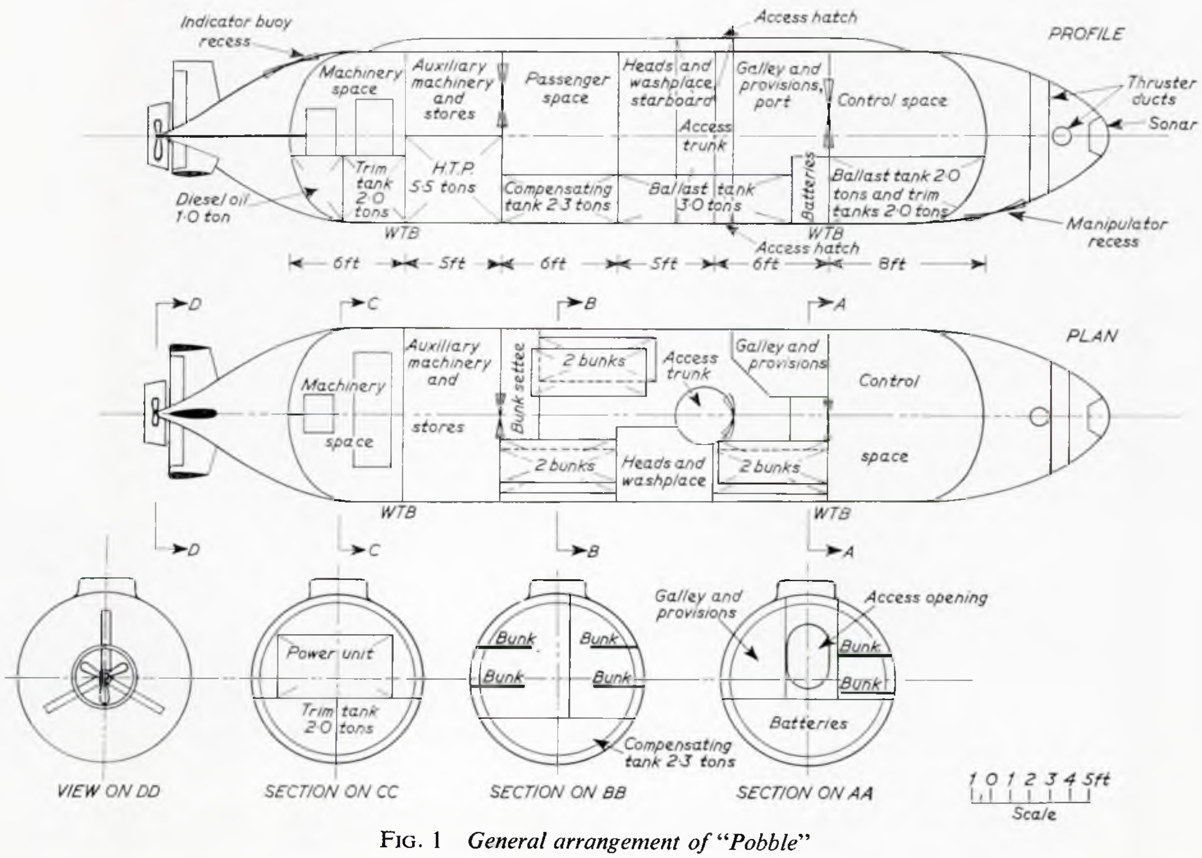

FIG. 1 *General arrangement of "Pobble"*

partment is arranged as stowage for stores and equipment and also contains auxiliary machinery for, amongst other things, ventilation, air conditioning and life support.

The arrangement is compact but by general submersible standards almost spacious, and thus should be conducive to crew comfort and efficiency.

CONCLUDING REMARKS

The author hopes he has shown that there are sound technical reasons for considering a submersible workboat like the "Pobble", that is, a relatively large land based boat capable, by means of its re-cycle Diesel engine propulsion plant, of transiting at a speed of at least eight knots to a well area 150 miles or more out on the U.K. continental shelf, transferring passengers and stores to and from a pressure tight house there, undertaking light engineering and/or surveying work, and then returning to base within a time limit of about five days or so. Although this concept has been used to develop arguments relevant to the design and operation of submersibles generally, it is not essential to the case. The great advantage that the "Pobble" offers is high mobility, which is worth having whatever activities the boat is intended to undertake.

At this stage in the UCL investigations the author is not in a position to be at all specific about the economics of the "Pobble", although he is in no doubt about the strength of the economic case for the approach of making the boat independent of a mother ship. By this means operating costs will be drastically reduced and the design problem is to minimize production and maintenance costs. It would be quite wrong to suppose that because the "Pobble" is relatively large it would also be relatively expensive to build. Quite the contrary, since a small cramped boat makes for difficulty, and in consequence high cost, of both production and maintenance. The policy advocated, of keeping most of the machinery and equipment inside the pressure hull, facilitates the use of off-the-shelf components wherever possible. The cylindrical pressure hull in B quality steel is simple to fabricate and easy to weld. The Ricardo re-cycle system applied to a standard Perkins Diesel engine is remarkably inexpensive. The need to use H.T.P. as the oxygen source, with the associated logistics problem, entails appreciably higher costs than if bottled oxygen gas could be used, but a brief analysis of fuel costs indicates that these would still be modest. The cost of the electronic equipment and associated displays needed would be high, but these would be needed in any submersible intended to carry out useful work on the U.K. continental shelf.

All in all, there is every reason to believe that the "Pobble" could be made an economic proposition whose services the oil and gas industries would find it worthwhile to hire.

ACKNOWLEDGEMENTS

The author would like to express his gratitude to his colleagues at UCL, R.K. Burcher, D.J. Low and R.J.H. Todd, for their comments and help in preparing this paper.

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Discussion

Mr. K. R. Haigh said that Professor Rydill had said in his introduction that he hoped that he could provoke controversy. Mr. Haigh accepted the challenge by saying that "Pobble" was not a submersible but a small submarine designed strictly on naval architectural principles of military submarines; she did not satisfy either the military or civilian definition of the term "submersible". To be absolutely clear about the matter—a submersible was not a small submarine.

In the era of the development of manned underwater vehicles late in the last century, the boats such as Brun's "Le Plongeur", Nordenfelt's designs and the first six designs of Holland, had all had one method of propulsion for surface and submerged, and a single hull form of construction; these designs had been known as submarines. Holland's "Plunger" of 1895 had been the first vessel with dual method of propulsion, steam for the surface and an electric motor for submerged; the French "Narval" had been the first European boat so equipped. These dual power system boats had been termed submersibles due to long surface endurance and strictly limited submerged capabilities. The majority of manned underwater military boats from this time until the era of nuclear propulsion had been submersibles and not submarines, although the former term had gradually declined in use except by the purist. With the coming of "Nautilus" in 1954 and the return of the single hull it had been necessary to differentiate between the old submarine and the new concept of boats with their strictly limited underwater endurance. Boats with nuclear propulsion were strictly submarines, whilst boats propelled by HTP with an extended underwater endurance became "intermediate submarines".

In the civilian field the American Bureau of Shipping defined a submersible as "any vessel or craft capable of operating underwater, submerging, surfacing and remaining afloat under weather conditions not less severe than sea state 3 without endangering the life and safety of the crew and passengers". The MTS "Safety and Operational Guidelines for Undersea Vehicles" defined a true submarine as a "vehicle that is primarily a submerged craft and secondarily a surface craft and is stable in both modes of operation". "Pobble" satisfied neither the military nor civilian definition of a submersible, but fitted both descriptions of a submarine.

Mr. Haigh then examined the principal features of what was generally accepted as a civilian submersible, as compared to a submarine:

Table I

	Civilian	*Military*
Design limitation	weight	space
Hull compressibility compared to water	less	more
Neutral buoyancy at depth	yes	no
Hovering capability	yes	practically nil
Compensating tank	no	yes
Submerge on an even keel	yes	no
Ballast tanks fitted	not necessarily	yes
Forward speed for depth keeping	no	essential
Speed	not important	vital
Operating area	seabed	mid-water

"Pobble" met all the military features of Table I. A similar design was shown in Fig 2: length 53 ft 10 in, diameter 10 ft 3 in, L/B ratio 5·26, submerged displacement 75 tons, propelled by a 50 hp internal combustion engine, body of revolution hull,

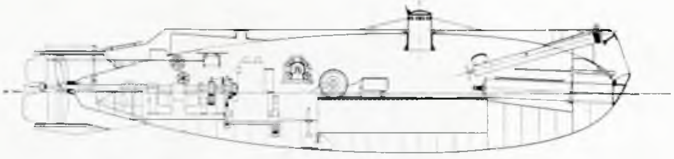

FIG. 2

minimum of reserve buoyancy, very little superstructure, telescopic conning tower, large diameter slow turning propeller located on the axis of revolution for enhanced propulsive efficiency, stern planes only; she had all the features of "Pobble". The boat was "Holland" of 1897; built in modern steel and fitted with HTP fuel, there was not much to choose between her and "Pobble".

The shape of the boat was not new. It had been rediscovered with "Albacore" and the shape was now used almost exclusively in nuclear submarines.

These were the main points, but Mr. Haigh could not agree with the author's philosophy on the use and application of submersibles; his design was strictly a work boat for the North Sea, and its application to oceanographic research was extremely limited. The design would cover less than one tenth of the ocean floors; oceanographers did not want to travel about the sea floor at five to eight knots, as even at one knot the data could not be assimilated.

He would have liked to have seen a lot more detail on the instrumentation and life support equipment which could amount to a very large proportion of the payload. This boat was going to require an inertial navigation system and sonar equipments of a sophisticated nature and consequent high cost—even as high as the cost of the hull and propulsion system.

The closed cycle Diesel was very important as a propulsion unit. It was developed from work done during the last war and, at the end of the war, in 1945, the German "Seal" and "Beaver" class had been fitted with automatic closed circuit cycle. A proposed design working on HTP had a submerged speed of 30 knots and a range of 1000 nautical miles.

The closed cycle Diesel was very attractive as a propulsion unit, but it brought other problems with it and the author mentioned one. This was the airborne noise which the crew would find tiresome and distracting. There was also the very big problem of self and radiated noise and their effects on the acoustic instrumentation. He therefore thought that although "Pobble" was good as a delivery vehicle for divers and rig crews, the boat had very limited application in oceanography. The closed cycle Diesel and associated pumps were likely to produce noise levels in excess of sea state 6, which would render the operation of sonar and echo sounders to a marginal level; instrumentation such as sub-bottom profilers and sonar would have been rendered practically useless. Unfortunately no data from the running of the Ricardo closed cycle Diesel in Shoreham Harbour was yet available.

Nevertheless, he welcomed Professor Rydill's design and only hoped it did not join the other British designs which seemed to have been tucked away in someone's bottom drawer.

MR. G. S. HENSON said that he disagreed with a lot that Mr. Haigh had had to say, but he wished to concentrate his comments on Professor Rydill's paper. There was much in the paper with which Mr. Henson agreed, but he wanted to argue about the design process which had been suggested, before going on to costs and economics, where he could supplement what had been said.

As to design, Professor Rydill had suggested that the customary approach was to start from requirements and then work through a fairly standard sequence. He then suggested that for submersibles used for servicing oil and gas installations in areas such as the North Sea, a reverse or back-to-front process was necessary.

But in the science based industries, wherever the state of the art was advancing rapidly, the approach that Professor Rydill adopted was normal; for example cryogenic computing and the world appetite for high speed arithmetic. These things got together by what Mr. Henson would call the "systems approach". System design involved an iterative or recircling process, each time going into more detail and each time answering the two questions; "What could we achieve and what would it cost?" and "What was wanted and what was it worth?" Professor Rydill answered the first part of the first question, but one needed to go round the whole circle.

Fortunately, a couple of years before, Mr. Henson and his colleagues had done some work in comparing submersibles' costs. He could give rough costs and other data for alternative submarines and submersibles which might help them to arrive at firm if somewhat broad conclusions.

Fig. 3 was part of the story given to N.R.D.C. a couple of years ago, when Mr. Henson and his colleagues had been suggesting that they might help to support a diver lockout submersible line of development. Mr. Henson had had experience of a submersible operating with a mother ship as illustrated in Fig. 3 on the right and on the left (shaded) was the submarine experience. The question had been whether, in the longer term, e.g. by 1975, they would have had an autonomous boat or a supported one.

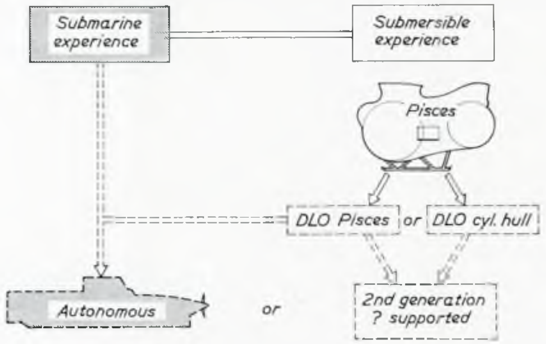

FIG. 3

They had tried to answer the question by defining tasks such as surveillance, and alternatively a small amount of work with divers or manipulators working on the bottom for a relatively short time, and alternatively again the transporting of men and equipment to sites, where the task might have required a couple of days on the seabed. They had compared the economics of alternative ways of doing such tasks.

In Fig. 4 they had assumed that the weather had been perfect, the boats completely reliable and that there had been plenty of profitable work to be done. Later on he would try to correct this gross falsification of real life. They had compared the submersible plus mother ship, with the autonomous submarine. They had been studying utilization, looking at the effect of distance from the shore base to the work site, for various work periods such as half a day, one day, three days. It was clear that one only got improved utilization from the autonomous boat, in terms of work at site, for the short distance and the longer times at site. It would have been kind to the autonomous boat if they had said that, on average, the two alternatives were equal in merit.

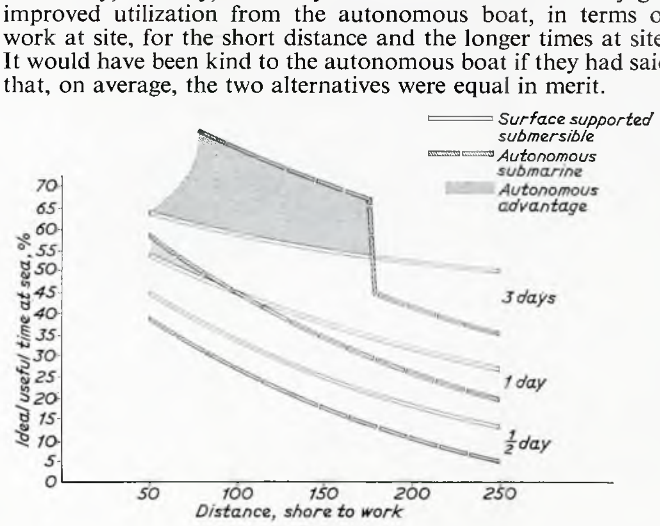

FIG. 4

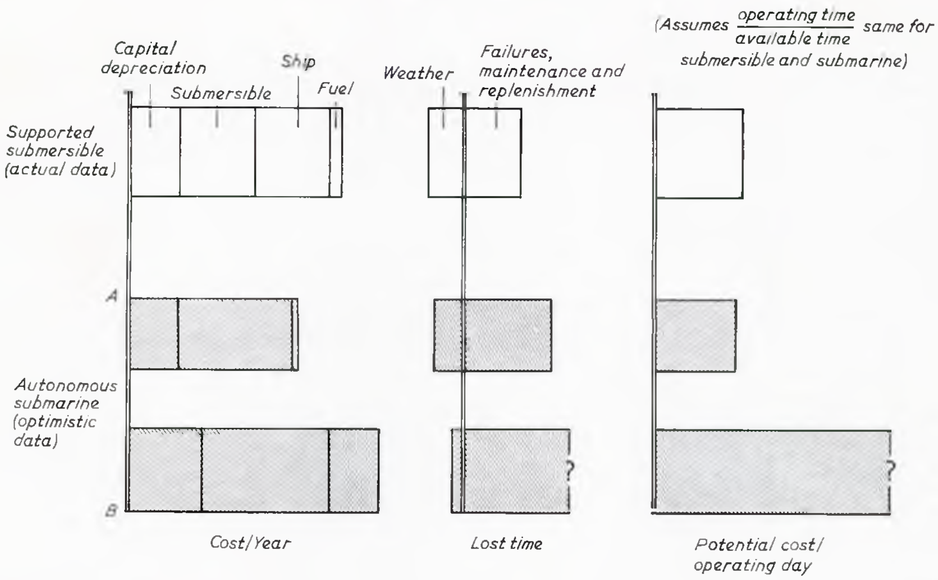

FIG. 5

In Fig. 5 he tried to bring in a bit more of real life. Like real life, it was rather complicated. He would deal first with the two top lines, for the submersible plus mother ship, and for the autonomous submarine. On the left he had given annual costs. These were made up of initial spending depreciated as capital over the life of the system; operating costs, and fuel costs. The lower cost for the autonomous system was attractive. In the middle one had the lost time, due to weather, which was the point Professor Rydill made, but Mr. Henson had also added lost time due to equipment failures, maintenance and replenishment. Failures were important particularly where there was need for a return to base, and there was a marked difference in annual overhaul times etc. These were all taken as an average over many years. On the right, derived from the previous two, was the cost per working day. He had assumed that the basic utilization, as shown in Fig. 3, was equal for the two boats.

The data for the top lines was based on real life. The more recent data for "Pisces", and what was now believed could be achieved from a rather larger mother ship than *Venturer*, made the top line seem pessimistic rather than optimistic. The second line for the autonomous submarine was optimistic. Mr. Henson concluded that the two alternatives were very similar. He believed that one day there would be a place for the autonomous boat, particularly when there was enough work for it to do, but it must be a reliable and maintainable boat. Looking at the third line—autonomous submarine "B"—a set of data had been put together for lost time, etc. The main difference was that submarine "B" was a closed cycle Diesel engine using HTP instead of Diesel-electric. About fifteen years ago a lot of work had been done on these systems at Barrow, so figures were available. Leaning over backwards to be fair to HTP, while the time lost due to weather was small, the other times for maintenance, overhauls etc. was very large. One was in a cleft stick; as one tried to get down these figures for lost times by testing and by a programme of work on improving reliability, so the costs went up. The costs per working day for the autonomous boat "B" looked as if they had jumped right out of the picture.

Mr. Henson said that he was not being dogmatic about this. Perhaps at some time Professor Rydill and his colleagues could get together with Barrow people to set target figures for maintenance, availability, etc., and costs for getting that capability out of the equipment, to see if it was worth pursuing this "Pobble" idea further?

In conclusion, he thought that the back-to-front design method—the systems approach—was better for underwater vehicles. It showed how the alternative design solutions could be hammered out. Under the present circumstances, the best submersible strategy was to have a flexible system to meet the rapidly changing market and the rapidly changing technology. As he saw it at present, the autonomous boat was rather a brittle system, and could be the wrong strategy. When someone else's money had produced a reliable closed cycle system and when there was a lot more work for a commercial boat to do, perhaps in another five years, we might want to specialize in the way that Professor Rydill suggested. Until then, he did not think the HTP game was worth a candle.

CAPTAIN J. R. PARDOE, O.B.E., R.N. said that during the last few months, his job had enabled him to go to sea in the DSRV to look at all sorts of other small submersibles and to talk to various nations about their development. Captain Pardoe found himself in agreement with a great deal that the last speaker had said, and at variance with almost everything shown in Table 1 by the first speaker.

First, he would like to make a comment on the future of deep diving. No less than three nations felt that they would get to 1000 ft with saturation diving, in the next year. In the light of the equipment which they had—especially the French—he believed that this would be done. But this did not detract from the arguments in favour of submersibles.

As regards "Pobble" the author seemed to be arguing that if you did away with the support ship, you saved a lot of cost. But what he was developing would be pretty complex, if it was to do the various tasks covered at the moment by the support ship and the submersible. By the time you had included these in a streamlined craft such as "Pobble" you would have a craft which would need a good deal of maintenance, and therefore there would have to be some support somewhere. One would probably put it in a ship, so as to be as near as possible to the scene of the task. Even if one did not launch and recover the craft at sea, having this ship might well allow one to keep in the support ship many of the complexities that one would otherwise have had to have squeezed into the submarine. Therefore Captain Pardoe thought that there was a way for the two to act together, as an intermediate step, before going over to the entirely autonomous submarine.

MR. J. R. PUTTICK added some further comments on the choice of power system for the proposed submersible. The author had opted for the use of a recycle Diesel engine and it did appear at the time, that this system offered the only practical means of increasing the power and endurance of the small submersible.

To illustrate the point one might compare the specific weights of the recycle Diesel with a typical lead acid battery, the values being typically 58 Wh/lb as against 11 Wh/lb respectively. Both these values were essentially practical results obtained from existing equipment and in the case of the recycle unit it was known that specific weight could be further improved by, for example, utilizing the heat rejected during the decomposition of HTP to oxygen and steam.

The author did, however, mention one particular drawback of the recycle system, namely noise, and there could be few people who would disagree with the statement that Diesel engines were objectionably noisy. This was a problem to which the automotive industry generally was very alive, particularly with legislation being introduced throughout the world aimed at reducing vehicle noise.

The problem could be approached in two ways, firstly by attention to the design phase to provide sufficient stiffness etc., in the engine components and secondly by providing palliative treatments to the finished product such as enclosure by a noise absorbing material. The first approach was not of particular interest in this context as one of the advantages of the recycle unit was that it used essentially standard production components and one would not have wished to have become involved in special designs. The second approach was relevant however, as the unit, being intended for use underwater, would inevitably be enclosed by a pressure hull and the ¾ in plating thickness which had been suggested by the author would in itself have brought about a theoretical noise reduction of some 40 dbA. This reduction in noise transmitted would have been further reduced by the use of an acoustic lining within the hull.

The actual reduction that could be achieved depended to a large extent on the degree of isolation that could be obtained between the noise source and the enclosure. This was substantial in the case of the recycle system where the unit was completely self contained and required no direct mechanical connexion to the surroundings.

The author had mentioned the test bed development work which has been carried out by Mr. Puttick's company; this work had in fact been extended over the past year to build a complete recycle Diesel power pack capable of underwater operation. This project had been undertaken mainly to demonstrate the feasibility of the system in its operational environment.

During the underwater test operation of this unit the opportunity had been taken to measure noise transmission levels and although the results were still being analysed, subjective impressions gained at that time had been favourable.

Commander R. Mack D.S.C., R.N., M.I.Mar.E. said that he had been interested in large and small submersibles for some time. His first reaction in reading this paper had been that the big difference the author had been striving for had been in the main propulsion, which gave the autonomy and unusually long range. It was a combination of closed cycle Diesel and HTP. Closed cycle Diesels had been tried in submarines by the Germans, at the end of the war, and also by the British. He understood that in the middle 1950s the Russians had built some "Q" class coastal submarines which had closed cycle Diesel propulsion from oxygen in bottles. They had become known as "cigarette lighters" among the Russian crews, and had been all laid up within two or three years. HTP also added to the difficulties and dangers. The Germans had tried that, and we had built two "E" class submarines, but did not keep them very long. With these two elements in a very small submersible, within close proximity to the crewspace, what a hope there would be from a practical point of view, although on paper it did look attractive.

Miss E. J. Macnair said that assuming that the author had made a case for the autonomous shore-based submersible for certain applications on the continental shelf—all-weather underwater transit and search at reasonable speed, without the need for handling through the surface—she would like to endorse his choice of power plant and oxidant.

The table compared the weight and volume for energy storage using different oxidants in standard containers. It would be seen that though oxygen was most effectively stored in bulk as liquid, it suffered from high proportional container weight on this relatively small scale: moreover in a small submersible compensating tanks had to be provided for the full weight of O_2 consumed. HTP could be carried in light flexible PVC bags and displaced by water, either in an internal tank as in "Pobble", or in a free-flooding space outside the pressure hull, so that compensation was only needed for the difference of density between HTP and water. It was considerably more expensive than oxygen (£100/ton) but the capital cost of containers was lower, and the running costs of fuelling were not a large proportion of submersible operating costs. HTP must be treated with care, and stored and handled under clean conditions with compatible materials; however although it was inherently a more hazardous material than liquid or compressed oxygen it could achieve a safer and simpler engine system. The fact that the oxygen from HTP decomposition was hot and ready diluted with steam resulted in an easy engine starting procedure, whereas starting an engine on cold pure oxygen was potentially hazardous and needed a carefully controlled routine. The Russians might have had more success if they had used HTP instead of oxygen in their "cigar lighter" submarines referred to by Commander Mack.

Table II

	Compressed O_2 (4000 lb/in²)	*LOX*	*HTP (85 per cent)*	*Fuel (gas oil)*
Unit container	9 ft³ standard submarine air bottle	100 gall vacuum-jacketed tank	2-ton PVC bag	
Empty weight, lb	1100	3360	200	
Weight of O_2, lb	225	1140	(1780)	
Weight of HTP, lb			4480	
Net volume of container, ft³	12	41·5	54	
lb/bhp h, recycle Diesel	1·8	1·8	4·5	0·5
lb/kW h (net) (a)	3	3	7·5	0·83
kW h per container	75	380	600	
lb/100 kW h including containers	1760	1190	780	87
ft³/100 kW h, including containers (b)	27	18·5	9·5	1·6

Notes
(a) 93 per cent generator efficiency; 15 per cent auxiliary power.
(b) 60 per cent utilization for cylinders; 93 per cent for flexible bags.

Commander Mack had stated that the R.N. had not kept their HTP submarines long, but this had been because the cost in money, manpower and resources of developing and operating nuclear submarines had left no effort available to continue with development of alternative propulsion systems. HMS *Explorer* and *Excalibur* had successfully met the need for high speed submarine targets in the pre-nuclear generation. However their Walter turbines had been essentially boost plants, with poor part load economy, unable to operate on air, and noisy underwater because of their geared transmission. The recycle Diesel engine could easily be switched to normal air-breathing operation on the surface when weather and sea-state permitted (thus conserving submerged energy storage), and a Diesel-electric or Diesel-hydraulic drive on closed cycle could be effectively noise-isolated at the cost of some additional weight and space. Mr. Henson had referred to reliability and maintenance problems: a Diesel on recycle certainly suffered from corrosion and wear more severely than when operating on air, and special materials would probably have been needed for a long-life sea-bed unit. However in a mobile submersible the economics of the operation would probably have been little affected even if the engine had been scrapped after a few hundred hours, assuming that a standard vehicle engine costing a few hundred pounds had been used.

Mr G. J. R. Maclusky said that he had two points on the use of the recycle Diesel engine. One point was that, quite apart from the direct acoustic noise in the engine itself, he wondered if that proportion of the exhaust gases which was pumped out of the vehicle at depth would contribute very much to the water-borne noise and thereby interfere with the sonar equipment.

The other point was in regard to the oxygen source. Could liquid oxygen stored in croygenic conditions be used? In order to produce two tons of oxygen, one needed five tons of HTP. This suggested that up to three tons could be allocated to the Dewar system, and to evaporation loss during five days.

Author's Reply

In his reply, Professor Rydill said that he had to comment adversely on Mr. Haigh's contribution, because his statements had been made with such apparent authority that there was a risk that he had misled those who were not experts in the field.

The table of differences that Mr. Haigh had shown had been of the consequences and not the causes. He was reading into the observed differences influences which were not real. Professor Rydill was aware that "Pobble" worked very much like the "Holland" boat, but there was no reason for regret in this; it was a splendid tribute to Holland himself.

Everything that Mr. Henson had said was welcomed. He was very aware of the deficiency in the paper regarding the cost and economics side and would welcome the opportunity for a get-together with Vickers, Barrow.

In reply to Captain Pardoe, the author agreed with many of his points. He had thought that the discussion would have brought out what he considered to be the serious flaw in the "Pobble" concept, but that had not come out, except by implication, in Captain Pardoe's contribution. It was that the concept of an autonomous boat operating without anything on the surface was a potentially dangerous one, simply because if anything at all went wrong, the submersible was in some difficulty in bringing to the attention of its land base the fact that it was in trouble, and so rescue could be quite difficult in this situation. He had always been aware of the Achilles heel in the argument, and he thought that it would need a support ship—not one to carry it, as he had pointed out, but a very simple one without the complicated and heavy lifting gear, etc.

Mr. Puttick, unlike some of the other speakers, contributed from a position of authority and detailed knowledge of the Ricardo recycle Diesel, and particular attention should be paid to his remarks. Miss Macnair had dealt very succinctly with Commander Mack's points, to which he would add that it was not a good thing to go forward looking backwards so much.

Professor Rydill had little to add to what he had said in reply to the discussion at the meeting but, in a written comment, said that he should like to thank Miss Macnair for the helpful table which accompanied her contribution. It would be kindest for him to refrain from further comment on Mr. Haigh's contribution.

His colleagues at U.C.L. had since had the opportunity for further consideration of the Vickers, Barrow, investigation reported by Mr. Henson. It would not be unfair to say that although the Vickers team had given a lot of thought to the cost aspects of the issue, autonomous versus mother-ship-supported submersibles, their findings did seem to be unduly heavily weighted by the assumed unreliability of the former.

It might be apposite to conclude by reporting that the "Pobble" concept had been further explored by another group of postgraduate students at U.C.L. The students had largely confirmed the earlier estimates of size and dimensions. So the crux of the concept remained its economic viability and its safety.

Mr. Walker

THE POTENTIAL USE OF UNDERWATER VEHICLES IN THE OIL AND GAS INDUSTRIES

R. I. Walker, T.D., M.A., M.I.C.E.*

The present oil and gas production of the free world is 40 million barrels per day. The offshore production accounts for 6·5 million barrels per day, or 16 per cent of the total production and it has been estimated that by 1980 more than 20 million barrels per day will be produced offshore, amounting to 33 per cent of the total production.

Essentially, at present all the offshore production is controlled and subjected to primary processing in the same way as on land, by marinized conventional equipment mounted above water on platforms. From these it is piped ashore for storage and transfer or stored in floating storage before loading into tankers for bulk transportation. Except for the platforms to support the production equipment and the pipelines themselves, no equipment is placed on the seabed.

With increasing depths of water and increasingly severe weather in areas in which oil and gas finds are made, the first tentative steps are being taken to place equipment on the seabed in order to reduce the number, size and complexity of equipment support platforms. This is discussed in greater detail later in the paper.

Commercial production of oil and gas is currently limited to water depths not exceeding 350 feet. This is well within practical commercial diving depths for temperate and tropical waters and has been so for some considerable time. The present depth record for diving is 1500 feet. This was a simulated dive carried out at the Royal Navy Physiological Laboratories in a chamber. The open sea depth working record stands at 780 feet for extended periods under water, when some typical work was carried out on a dummy wellhead off the coast of Corsica.

Exploration drilling operations can and do take place in much greater depths. Without making an exhaustive search, the deepest commercial drilling known to the author at the time of writing was 1200 feet. On a recent scientific mission a core was recovered from the mid-Atlantic by drilling, after re-entering the hole in a water depth of 16 000 feet. It should not be inferred that oil and gas wells can be drilled commercially at these depths, but it is a beginning. This does, however, highlight the fact that production techniques rather than drilling at present limit, and for the foreseeable future will limit, the water depths in which the production of hydrocarbons can be commercially undertaken.

In the following paragraphs, the author has assumed the widest possible definition of the word 'submersible'—any unit controlled directly or indirectly by man to carry out specific tasks or observation under water. On the basis of this definition, submersibles can be split into the following main types:

1) stationary work chambers,
2) towed submersibles,
3) tethered submersibles,
4) free swimming submersibles;

and where manned, the above categories are sub-divided into wet (diver transport) and dry (with a life supporting atmosphere) versions.

No matter what the form of the submersible, all are dependent to a greater or lesser extent on a surface support vessel—generally to a greater extent. In the past, and to some degree now, owners and operators of submersibles have turned a blind eye to the need for adequate surface support with their submersibles, or have at the eleventh hour improvised one. One can well appreciate that reasons for doing so are financial. An effective surface support unit is likely to exceed in value the submersible which it services. The handling system for recovering and launching a submersible is at the heart of the surface support vessel and one does not have to be a mariner to appreciate and envisage the problem of recovering a submersible, weighing in all probability five tons or more, in any sort of sea, and of landing it securely on the deck of a vessel.

The extent to which a submarine vehicle can be used, and hence its cost, is dictated by the number of effective working hours that such a vehicle can be used. Looking at sea states in the North Sea, admittedly a bad area, weather data collected over the past decade shows the following significant wave heights to be expected throughout the year:

Less than three feet	16 per cent
Less than five feet	64 per cent
Less than eight feet	86 per cent
Less than twelve feet	95 per cent

In the writer's experience, there are few vehicles that can be operated in seas of wave height more than four feet, and none that can be operated in seas of six feet wave height. Operators, not unnaturally, claim appreciably higher figures and may be on the way to increasing these limits with novel single purpose vessels designed purely as a mother vessel.

It is pertinent to review the range of jobs to be performed under water by the industry. First comes seabed survey for exploration drilling operations. This, by the very nature of the work, can take place in water depths beyond existing commercial production depths. At this stage information is only required on the anchor holding ability of the seabed in the case of floating drilling units and on seabed bearing capacity in the case of bottom supported barges. Additionally, both types require a general topographical survey of the seabed prior to positioning for drilling, for the presence or otherwise of obstructions, including wrecks. This data can be obtained more economically by indirect methods of echo sounding, side scan sonar, and shallow penetration high resolution continuous seismic profiling. Shallow-scrape samples of seabed material can also be more easily obtained from other than submersibles with their present range of sampling devices. The extent of seabed and soils mechanics information required for permanent structures on the seabed is so great in detail that the current range of submersibles and accessories has little part to play.

* Technical Manager, Seal Petroleum Ltd.

In the Gulf of Mexico and the North Sea and other areas heavy in traffic, an attempt has been made to bury a pipeline, or rather to dig a trench around the line. The trench subsequently fills and buries the line by the normal processes of deposition. The submersible has no part to play in these areas, other than to check that the line is covered and remains covered. In some areas pipelines are laid directly on the hard seabed without any attempt at covering. It is probably only in these areas, on the rare occasions that a pipeline inspection is judged to be necessary, that a submersible truly becomes competitive with diving and indirect methods of survey.

It is in the area of pipeline repair, maintenance and installation that the first category of submersible—the dry ambient pressure work chamber—has already found a use in the oil industry. A number of pipelines have been repaired by welding T-junctions cut in and made to pipelines previously laid. Similar static chambers have been used for the repair and inspection of platforms already installed. Access to these work chambers is made by working divers, who enter the chamber through a water lock. The chambers are initially installed from a surface support vessel, and throughout their utilization on the bottom, they are powered and life-support maintained from the surface.

One or two partially successful attempts were made in the early sixties to visually survey unburied pipelines by divers towed along from a bottom supported or swimming towed vehicle. Due to the problems of communication between divers and the surface these were only partially successful, and the author knows of no recent attempts to make use of towed submersibles for pipeline inspection. This paper does not discuss towed mid-water or bottom supported instrumented "fish" used in indirect survey methods, such as side scan sonar, magnetometer, etc. These methods are extensively used by the industry.

With the exception of tethered vehicles, which are discussed in the paragraph dealing with the future trends in the industry, free swimming vehicles have found limited use in the industry and in the author's view will continue to do so although in number they form the majority of vehicles. They have been used, however, in emergencies, where conventional means have failed. Their prime virtues are in the search and observation role where, hopefully, the industry has few needs for such operations which are associated with emergencies and mal-functions. One or two cases of the successful recovery by submersibles of high cost equipment lost or dropped from a drilling rig have been recorded.

Turning next to the future of the industry, the author believes purpose designed, special purpose tethered submersibles will have a part to play. With the move of the industry into deeper water, there will come a time when it is more economic to carry out the simplest control and production functions directly on the seabed. It is for servicing these units and carrying out low frequency control functions that tethered submersibles will be required. These submersibles are unlikely to look like the conventional fish-shaped vehicle. They will be handled from the surface support unit, guided to a specific location on the seabed where they will be required to perform some pre-planned specific task such as operating valves designed for this purpose as opposed to those designed to be operated by man, replacing whole subsystems and assemblies for maintenance on the surface, thus transferring men to a dry controlled environment where tasks can be performed below the water. The precise water depth in which it becomes economic to transfer these functions below water is at present the big question mark in the industry. It is, however, significant to record that there are five commercial organizations working to this end. It is the author's view that for some functions this critical water depth has been reached.

From the foregoing paragraphs, it should be clear why subsea vehicles have been little used in the past by the oil industry. Firstly, the industry generally requires to "do rather than see". Submersibles to date have been built to observe rather than to perform a specific task.

Except for exploration drilling, as opposed to exploitation drilling, all work for the oil and gas industry has taken place in water depths in which commercial diving services are available.

Even so, all commercial production at present relies on platform mounted surface production units. There are a few quasi-commercial subsea production exercises being conducted, such as the ADMA Subsea Production Scheme. These, however, are information gathering exercises directed at future production systems.

The chief virtue of conventional submersibles are their mobility. With the exception of pipelines, the oil and gas industry requires to work at specific locations rather than over a wide area.

ACKNOWLEDGEMENTS

The author would like to acknowledge the assistance in writing this paper from his present colleagues and his former ones, with the British Petroleum Company Limited.

Discussion

CAPTAIN K. A. GOUDGE, R.N., said that it seemed that Mr. Walker had been thinking very much of the possibilities of using submersibles in the offshore oil and gas industry in the state that these vessels had been some two years ago. Since then Captain Goudge's company and their rivals had improved the "Pisces" class a lot. His company had had the benefit of buying tools and charters both from the M.o.D. (N) and the D.o.T.I. They had improved regarding speed and control. They used to travel at about 1¼ knots, so there had been severe restrictions in operating in cross-currents which were very strong around Britain, and it was nonsense to say that there were no currents on the bottom. They now had an indicated 4·1 knots on the log, and had operated at a 2½ knot cross-current, so there had been an increase in both speed and power.

Fig. 1 gave some idea of the change in tooling, the sort of things they had now got and had operated, and this would go on into the future. He hoped that this sort of thing would make a difference to the ideas of Mr. Walker, with regard to operations on the seabed. Not only did one need to inspect pipelines, but also do ultrasonic testing of pipe where required. With such a picture of a whole system on the seabed at, say, the Persian Gulf, one might want to move from one place to another, with diver lockout, locking them in, and so on. To do it from the surface with a bell from the mother ship meant that the mother ship must be moved with great precision because the bell itself could not move, so four anchors were needed which must be relocated, or an expensive position-keeping device which required a thruster as well.

The other aspect they had tackled had been navigation and search. See Figs. 2 and 3. In the beginning, everything had been very primitive. They had seen some very attractive 17th Century muskets in Loch Ness, but they had slipped from the grab and

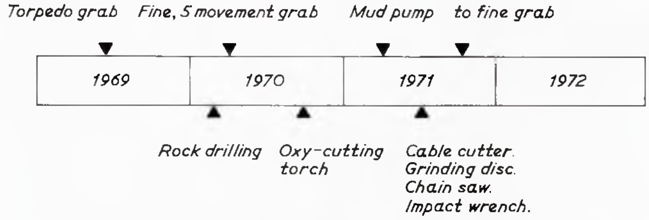

FIG. 1—"*Pisces*" *Manipulators tools.*

Underwater telephone.

C.C.T.V.

T.V. buoy transmission from Pisces.

Side scan sonar.

Stereo photography.

FIG. 2—*Data supply systems.*

could not be found. But they had since got on better, and had now a considerable capability of doing a close search, and had rapidly found quite small objects which people had lost, on the seabed, which had had to be searched for in a close search.

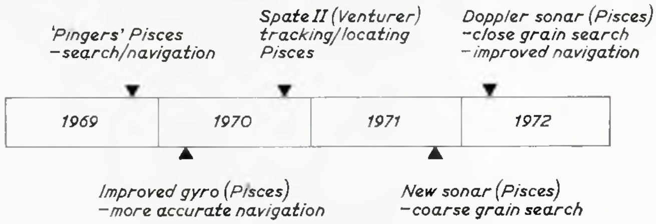

FIG. 3—*Navigation and search.*

Doppler sonar was the next step. They could already, without it, see their position on the bottom to about $\pm 1\frac{1}{2}°$ and perhaps $\pm 1°$, and perhaps 10 ft from the surface end, and could lay bench marks on the bottom and do an independent close search. This was quite good, and it was getting better.

Fig. 4 gave an idea of the increase in capability and they hoped that 1972 would increase it still further. They could not always see into the future, but it was building up; Fig. 4 gave some idea of the changes. He did not know how far these carried conviction, but they had been demonstrated in the charters mentioned, and reports would be forthcoming to show prospective customers what they could get. The one for the last year was already out.

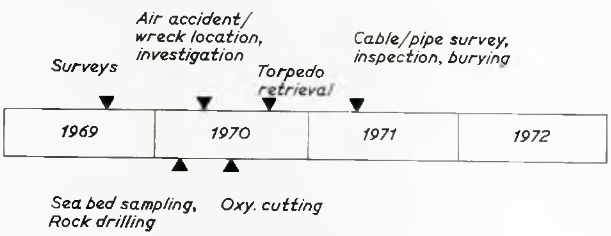

FIG. 4—*Capabilities.*

He agreed that the all-round vehicle was not a possibility. But equally, if one only had a certain amount of cash and had to earn enough on it to keep it going, one could not afford to specialize too far. He believed that there was a solution in modular construction which kept a lot of the basics and changed particular modules for particular tasks. He thought that this was the way for the future.

MR. R. DUNBAR said that Captain Goudge's comments on the paper and the paper itself had been of particular interest to him because they lined up with the thinking at the Heriot-Watt University in Edinburgh, where they had an Underwater Technology Group. Over the last 18 months this group had been developing an unmanned cable-controlled underwater vehicle. They actually had the hardware and the basic shell had had preliminary sea trials. They had yet to produce the prototype propulsion system, although they had plans for this and had sufficient knowledge in practical terms to have confidence that it would be successful.

The vehicle, in its initial concept, would carry closed circuit television equipment and was primarily a survey vehicle. Therefore they had called it "Angus", which stood for A Navigable General-purpose Underwater Surveyor—as well as being a good Scots name. This vehicle would incorporate variable buoyancy control to allow it to hover in mid-water or to settle on the seabed. They were aware of the principle of hydro-dynamically stabilized vehicles, but for close-bottom work the rather slow speed of operation would, they thought, make it difficult to utilize that technique. They had taken the view that what was needed was a low cost vehicle, partly because their university funds would not have stretched to a more expensive one, and in line with this, they were designing systems round modified mass-produced equipment where possible. Using commercial motors costing of the order of £2000 seemed unnecessary to them and they were aiming at something which would cost about one-tenth of that.

They were aware that of necessity, oil companies were only interested in gear which was fully developed and reliable, and not so much in the R and D stage. They had therefore not contacted the oil companies at this early stage in their programme to find out how the vehicle could be developed to suit their task, although they had been in consultation with many potential users. In view of the comments about cable-controlled vehicles, particularly the type attached to an oil rig for inspection purposes, he would have been interested in Mr. Goodfellow's comments on their ideas of a cable deadweight—to cater for the surface/seabed tidal cable drag—as compared with permanent attachment to the base of the rig. The submersible's umbilical cable could be attached to the base of the oil rig, or one could have a deadweighted main cable a 100 metres off, allowing the vehicle to move in.

Their vehicle would carry closed circuit television as its prime function, but they were extending its capability to include simple manipulators, and they would like to design around a real need. If one aspect was of more importance in oil rig inspection than any other, they would have liked to have engineered their design in that direction. They were out to produce hardware for immediate application and not purely for academic consideration.

He and his colleague, Mr. Holmes, were grateful to those who had encouraged them in their step of faith in going for the hardware rather than involving themselves strictly with the paperwork. They believed that it was the right step, and in the months and years to come they would like to share the developments with others.

Author's Reply

Mr. R. Goodfellow, replying for the author, said that having been involved, in presenting papers on diving, in the past, and now presenting underwater vehicles, he did not know whether the complexity was beyond anyone's concept and defeated discussion, or whether it was so astounding that it numbed people. The complexity of operations and the physical environment was far removed from an academic approach. The question of research and development allied to what was considered a fairly affluent industry, was not easy. The main concern, within his company, was to produce oil as economically as possible, and they endeavoured to do this. They were constantly looking at economics; every day of the week, every operation was monitored and recorded and the cost controlled very tightly. Any research attitude had therefore to be looked at very carefully. Mr. Goodfellow's company possibly used more helicopters than the average person in the room, but they had not put any money into that research as it was a specialized area where there were other people who were more capable of putting in money and providing a service. This was not intended to put anyone off long term research, or to say that they had not assisted in prodding that type of research when a shot in the arm had been needed.

Obviously they had a need to locate objects fallen overboard, although they endeavoured not to drop too many, but this was usually at the location of a rig where there was a sophisticated form of diving service. Obviously a lot of time had been spent on location and recovery of equipment, but this was on specific locations; they were not free-ranging the seas and trying to find their way home. They generally knew where they were, and tried to be as economical as possible.

This was no reflection on the ability of an underwater vehicle to perform a specific task, but it showed the other factors. The ability to operate in over 1000 ft of water would keep them busy for years. Ideas of when the potential sources of oil and power would dry up was open to conjecture, but he certainly felt that supplies would see him to his retirement.

The aspect of modular concept was being dealt with. Re Seal Petroleum, the two aspects which they had considered from the beginning were, firstly, an APD system, a development at atmospheric pressure, in which the idea had been to transfer the man to the well head at atmospheric pressure to work directly, and secondly, the other extreme of all operations being carried out remotely from the surface, without men going below water. Between these two standpoints technology had not bridged the gap. The gap was being closed between the first idea and the point where ironmongery had been produced to lower a man in a capsule to the seabed at a maintained atmospheric pressure, and the concept where the unit was remotely controlled and maintenance free and the work was just that of recovering a module which had failed to perform the required task, and replacing it. However, we were a long way from that. Development by module was clearly effective.

In reply to Mr. Dunbar, the aspects of educational requirements considered and the support which the industry gave to academic bodies were considerable. This was also true of requirements for specific and specialized engineering skills. There was a greater feeling in this country for the oil and gas industries, possibly due to the fact that the North Sea was on the doorstep, and there should be more encouragement for them to be associated with these developments.

With regard to the umbilical cable, while the rig was standing it was very securely anchored and represented a very fixed location. All the concentration was on the wellhead, which was a specific location, so it was advisable to locate as much as possible in the dry, on the rig, and it would be better to develop ideas from that aspect, rather than re-working remotely from the rig.

A lot of work on manipulators had, of course, been carried out by the A.E.A., and this was very far advanced and with very sophisticated means of control. He doubted if in the near future that degree of sophistication would be needed for more ordinary jobs. He could not think of any use for this—although they were an aid—but in the arguments about underwater vehicles and submersibles, it would be an aid to a submersible. In future working he hoped that remote control would be brought to the seabed as a complete operation, or that an actual operator, who would physically handle the normal arrangements, would be on hand, so that a manipulator was not needed.

Written Reply

Mr. Walker wrote that he had been surprised that Captain Goudge had found a different attitude in the written paper to that in the presentation by Mr. Goodfellow; Mr. Goodfellow and himself shared the same views on the use of submersibles in offshore oil and gas operations. These views are re-iterated as follows:

Even with the increased operating characteristics of the present family of vessel (of which he had been aware when writing the paper), the submersibles were largely dependent on their mother vessels at the surface. Using a submarine also meant paying for a support vessel; down time, arising from the inability to launch and recover the submersible in other than good sea and weather conditions, gave a low utilization. Taken as a system, using a submersible was a costly exercise. Secondly, most subsea jobs in the offshore oil and gas industry centre around a fixed location; there was no use for the principle virtue of a submersible—mobility—except in pipe line surveys. It therefore had a limited role to play; pipe line surveys were required infrequently.

Captain Goudge had shown slides of general purpose tools, but they did not cover the sort of operations that the oil industry required to carry out underwater.

He could not, for the above reasons, see that a free-swimming submersible had a role to play at the present time in offshore operations other than pipeline inspection. Tethered diving bells with limited manoeuvrability did, however, have an immediate part to play both for transfer of personnel to dry working enclosures and for transfer of divers, or as a means of carrying out pre-planned operations below water with purpose-made devices.

Mr. Foster Mr. Cameron Dr. Hemmings Mr. Parrish

THE CASE FOR A TOWED UNDERWATER VEHICLE FOR FISHERIES RESEARCH

J. J. Foster, B.Sc.*, G. Cameron*, C. C. Hemmings, PhD.* and B. B. Parrish, B.Sc.*

INTRODUCTION

Direct underwater observation is the only way to obtain some information that is relevant to research projects being undertaken by the Marine Laboratory of the Department of Agriculture and Fisheries for Scotland. This is particularly true of research into fish capture which involves studies of the performance of fishing gear and the behaviour of fish in the vicinity of the gear. Development of techniques and instruments for monitoring the engineering performance of trawl fishing gear is now at an advanced stage involving ship-borne computers and already a great deal of data are being acquired. Sophisticated sonar equipment is being introduced to study certain aspects of fish behaviour. And yet with all this technology there are certain parts of the fish capture processes the full understanding of which can only be realized by direct observation.

The value of direct observation by free divers of fishing gear and the reaction of fish to the gear has been made abundantly clear in the case of gears operated in shallow water. Unfortunately, the physiological limitations on the working depth, endurance and speed of free divers so limits their operation range that their useful employment for investigating fish capture processes is restricted to depths less than 20 fathoms and then only for gears moving at speeds slower than is normal for trawling.

In the past the Marine Laboratory's free divers have sometimes operated from open towed sledges and a wet towed vehicle (Mobel). Whereas these vehicles reduce some of the physiological limitations, they have little or no effect on the divers' depth endurance. If full elucidation of fish capture processes by trawling, both for pelagic and demersal species, is to be realized it will be necessary to extend direct observation to greater fishing depths and for longer duration than has hitherto been possible. One way of achieving these objectives is to use a manned dry vehicle.

Self-propelled vehicles are heavy due to the exclusive use of batteries and electric motors for propulsion. This would be a particularly severe problem in a fisheries submersible because of the necessity to travel for long periods of time at cruising speeds of three to five knots. The high weight involved would increase the problem of handling at the surface. Even in the clearest water, visual range is so restricted that an accurate and reliable navigation system would be essential, capable in the case of the fisheries submersible of allowing the pilot to locate and manoeuvre around moving fishing gear with absolute safety. All these factors would combine to result in very high capital and running costs of a powered vehicle especially when the additional costs of a specially rigged support ship are taken into account.

Experience with towed wet vehicles suggests that a towed dry vehicle is a feasible proposition. Subsequent performance trials of a quarter scale prototype towed vehicle and handling trials of a full-scale version to investigate problems of launch and recovery from a research vessel operating fishing gear have now demonstrated that a towed dry vehicle is a practical proposition.

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APPLICATION OF TOWED UNDERWATER VEHICLE

The otter and other types of demersal (bottom) trawl, together with midwater trawls are the most important commercial fishing methods. Fig. 1 gives the range of sizes of typical trawling gear, which would be towed through the water at speeds of between three and five knots (the higher speeds being more normally used in midwater fishing). The total drag of these fishing systems can be up to ten tons, with shock loads, caused by snagging on the sea bed of two or even three times that value.

The operation and performance of the gear has been monitored by instrumentation. Measurements of geometries and tensions in wires are obtained from self-recording instruments after recovery of the gear or telemetered via acoustic links and/or through electrical cable links to the ship during towing. The gear, particularly the net, is not rigid during fishing and with the limited number and resolution of instruments full information about its shape is not obtained.

The behaviour of fish near fishing gear is even more difficult to assess by use of traditional instrumentation. Advanced sonar equipment, especially sector scanning sonar, is proving to be a very useful tool for studying fish behaviour away from the sea-bed, but it has serious limitations for this purpose close to the sea-bed.

In the case of the trawl, considerable water disturbance is caused by the passage through the water of the otterboards which leave behind them a trail of turbulence and suspended bottom sediment. Little is known of the degree to which the orientation of fish is dependent upon these turbulence trails. These conditions make indirect observation by sonar very difficult and will inevitably restrict the distances over which direct observation is possible. The facility to observe from above and immediately in front of the gear is obviously of considerable importance.

In studies of the seine net, where maximum speeds rarely exceed three knots, direct observation by divers in shallow water has provided important information about the reaction of fish to the wings and other parts of the net. However, the depth limitation imposed on free-divers (about 20 fathoms) can be a serious one due to the lack at these depths of suitable quantities of the main commercially important species of fish.

In studies of the natural behaviour of fish in many situations, as for example, spawning and feeding, it is desirable to station the observer on the sea-bed or in midwater for extended periods of several hours, which clearly would not be feasible with free-swimming divers. In this context the towed underwater vehicle could be used as an observation chamber analogous to the bird-watchers' hide.

In summary, some aspects of fisheries research for which a manned underwater vehicle is required are:—

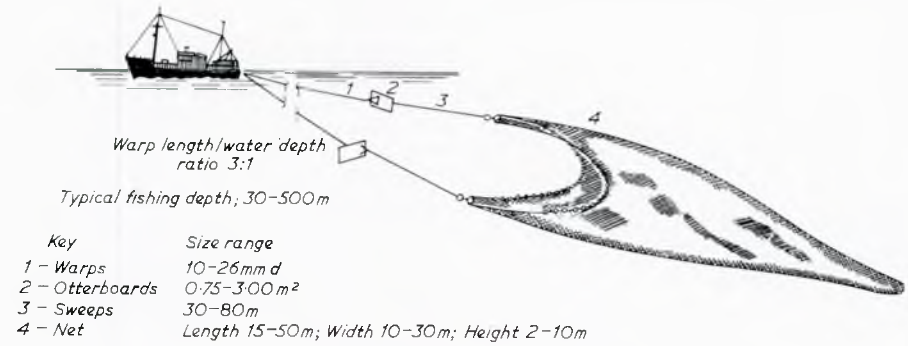

FIG. 1—*Simplified diagram of fishing gear*

1) Direct observation at towing speeds of:
   a) net shape and subsidiary movement characteristics;
   b) otterboard performance;
   c) behaviour of wires connecting the parts of the fishing gear;
   d) reaction of fish to:
      i) the various components of the fishing gear
      ii) mud clouds and turbulence
      iii) noise
2) Extended direct observations of various stages of the life cycle and habits of important marine species.

OPERATIONAL REQUIREMENTS AND DESIGN OF TOWED VEHICLES

Operational requirements are based primarily upon previous experience of the Marine Laboratory's free-divers, when operating around fishing gear, and on the use of wet towed vehicles. First, it is considered that a high speed observation capability is required of the vehicle whilst retaining precise control of position and orientation when working in close proximity to the gear under observation. Second, it is considered that the dry towed vehicle must be a two-man machine to allow pilot and observer to concentrate fully upon their jobs. Third is the question of working depth. Most commercial Scottish fisheries take place in fairly shallow, continental shelf waters, up to about 600 ft deep. In the clearest of these waters the maximum depth to which natural light penetrates with sufficient intensity for unaided vision is about 300 ft. Whereas the working depth will therefore not normally be greater than 300 ft, the vehicle is required to have a full endurance depth of 600 ft for safety. Further, it is desirable to keep the weight of the vehicle as low as possible. It is interesting to note that a one-man towed vehicle for use in fisheries research in the U.S.S.R. has a weight of just under two tons and a working depth of 100 m.

At all depths down to the maximum working depth, the vehicle must achieve a depression angle at the ship of at least 20 degrees and a yaw angle of 15 degrees each way. Towing speeds must be up to six knots, to allow for the occasions when it is necessary to advance the vehicle forwards past the net. A life-support system capable of supplying two men for at least 12 hours' normal operation is essential, although it is unlikely that towed trawl watching experiments would be longer than one hour.

Certain additional safety equipment and facilities are essential, and where possible these should be duplicated. The towing cable must be releasable from inside the vehicle, and the vehicle should have two independent flotation systems. Almost certainly the most important safety and operational feature is a good communication system with the support vessel whereby early warning and tactical instructions can be given should potentially dangerous situations arise. Normal communication would be most conveniently conducted down the towing cable in a telephone lead, but this should be duplicated by a through-water system. This would allow communication with the support vessel to be maintained should the telephone system fail and, in the event of an emergency, provide a means of contact with surface divers. All these requirements have to be combined with the necessity to operate from a 110 ft research ship with limited special modification.

Hull design and controls

The requirements suggested above could be met using a hull comprising two horizontal cylinders mounted side by side with transparent hemispherical end cap and conical tails carrying diving and steering control surfaces (see Figs. 2 and 3). The

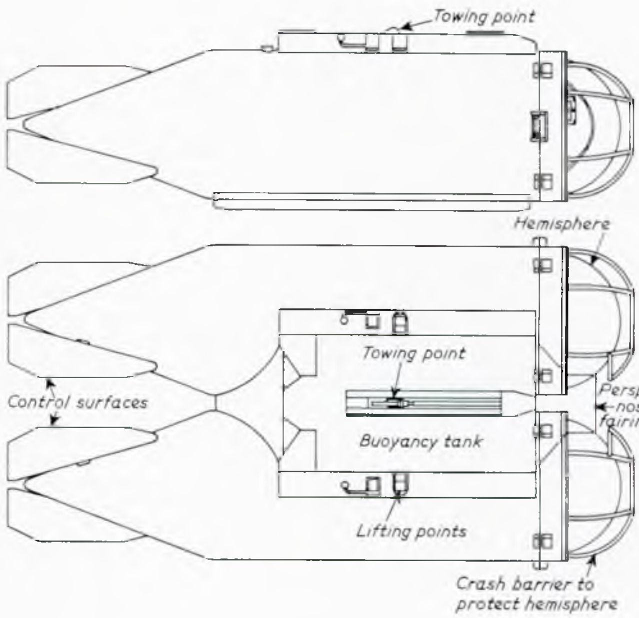

FIG. 2

occupants would lie face down with their heads in the centre of the hemispheres which would give them an ideal viewing position. The control surfaces mounted on the tails would be balanced and controlled by a joystick in each compartment. It is envisaged that the hemispherical viewing ports would be on hinged mounts that serve as entrance hatches on deck. Buoyancy controls are necessary and it is envisaged that they would be achieved in the following ways:

1) Internal ballast which would be adjusted before launch.
2) A dropweight keel which can be discarded situated in the space below and between the two cylinders operated by a mechanical linkage from inside the vehicle.
3) A buoyancy tank situated in the space above and between the two cylinders and in series with a compressed air bottle and a suitable valve system. An important design requirement which might need to be

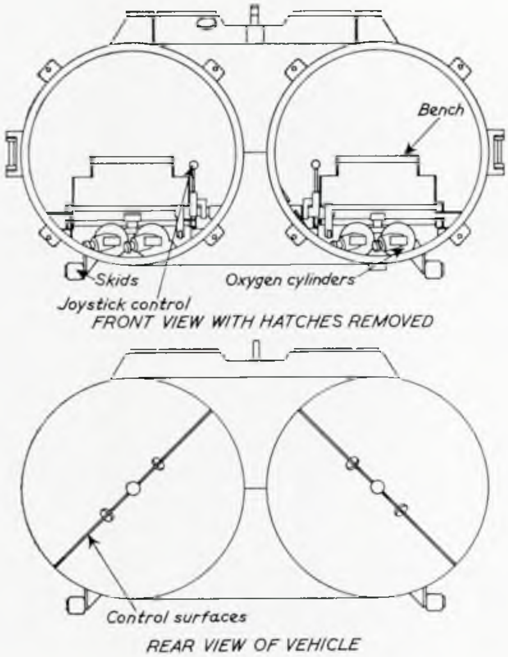

FIG. 3

used in conjunction with the buoyancy control would be a cable release mechanism, possibly operating by an explosive guillotine. This would enable the pilot to release his vehicle from the towing cable and surface independently.

Power and instrumentation

The anticipated power capacity of a vehicle of this sort would be in the order of 1 kW of which about 10 per cent would be required for internal lighting, circulation fans and instrumentation, leaving some 900 W for use in external lighting. The towing cable would carry power and telephone cores.

Visual observation facilities

As mentioned previously, a suitable all-round viewing window would be a transparent hemisphere but here lies one of the greatest technical problems anticipated in engineering. Acrylic materials (perspex) are almost exclusively used in the aircraft industry for windscreens but only to a limited extent in underwater vehicles. Some work carried out in the U.S.A. is reported in this field (Ref. "Windows and transparent hulls for man in hydrospace" by M. R. Snoey and J. D. Stachiav) but there is little experience of acrylic hemispheres of the size envisaged (28 in dia.). There are two main problems with acrylic hemispheres under pressure. The first is that acrylic materials in general are produced to a very high standard in sheet form while components cast in alternative shapes are usually of a variable standard and therefore unsuited for use as part of a pressure hull. This forces the designer to free blow hemispheres from sheet with the consequent problem of varying thickness. A free blown hemisphere made from $2\frac{1}{4}$ in thick sheet will end up with a $\frac{3}{4}$ in thick apex and a $2\frac{1}{4}$ in thick base. The other problem using acrylic hemispheres arises at the metal to acrylic interface, where the difference in the stiffness of the two materials makes it difficult to clamp the window in position. Any fixing which does not allow the plastic to flex under pressure sets up destructive stresses in the window over and around the area of contact.

MODEL TESTS

A $\frac{1}{4}$-scale model has been tank tested at the National Physical Laboratory. This model was basically the shape shown in Fig. 2. These model tests showed that in general the stability of the vehicle was satisfactory over the complete range of speed, yaw and depression angles. The exception was a slight lateral instability at four knots (full scale equivalent) with 20 degree yaw which disappeared when the speed was increased to five knots or reduced below four knots.

Analysis of the results, partly by extrapolation, showed that the maximum cable depression achieved was approximately 40 degrees in the speed range three to five knots, but considerably less (just below 30 degrees) at two knots. These values were considered satisfactory even allowing for errors arising through scale effects; depression angles of greater than 20 degrees will not normally be necessary (see Fig. 1 for relative lengths of warp at various depths). In the case of yaw measurements the maximum cable yaw angle was 20 degrees in the speed range three to five knots. This is almost twice the total yaw required to manoeuvre a vehicle across the whole frontal area of trawls.

HANDLING TRIALS

Mock-up trials—surface vessel handling systems

The handling system for a vehicle must be simple and reliable because launch and recovery of the vehicle is conducted during fishing operations. Use of the wet towed vehicle showed that the sequence of operations was most conveniently:

a) shoot trawl and commence towing;
b) lower towed vehicle;
c) after the experiments recover vehicle;
d) haul trawl.

It is estimated that a suitable vehicle would weigh about 3700 lb and a launch and recovery system has been developed to handle weights rather larger than that value. The support

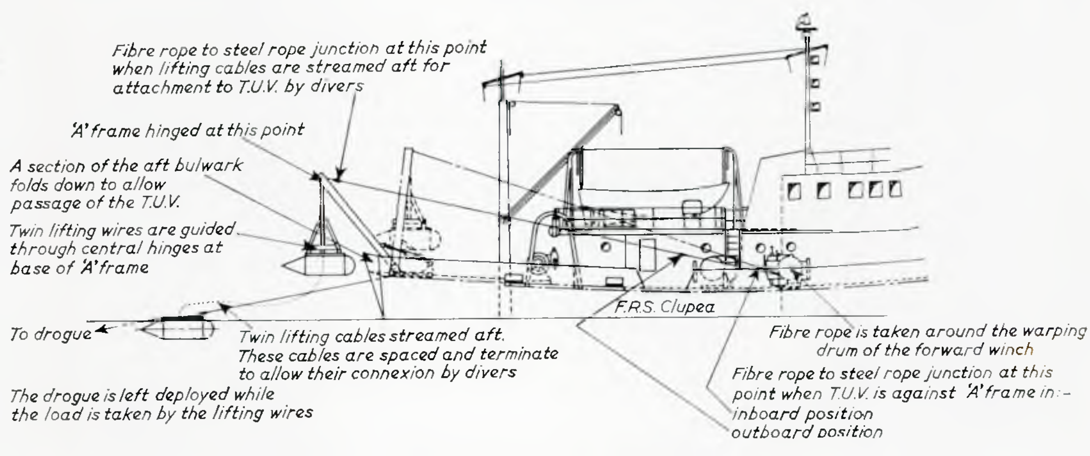

FIG. 4

craft would initially be a DAFS fisheries research vessel (110 ft) *Clupea* and the vehicle will be launched and recovered over the stern using a hydraulically-operated gantry and a swinging 'A' frame as shown in Fig. 4. The vehicle is lifted off the deck (after pilot and observer are installed within) by two lifting cables and held hard against the rubber base of the 'A' frame while two hydraulic actuated rams locate into special lifting cable terminations on the vehicle. This retains the vehicle on the 'A' frame and eliminates all lateral and pitching movement relative to the gantry until the gantry has been moved to the outboard position when the weight of the vehicle is again taken on the lifting cables and the hydraulic rams retracted. Fig. 5 shows details of the 'A' frame. At this stage a small drogue is deployed to keep the vehicle from swinging towards the ship's stern while it is lowered into the water. The lifting cables are paid out and the tension is taken on the towing cable, at which time swimmers release the lifting cables and drogue cable, thus completing the launching operation. Recovery procedure is the exact reverse of launching, the swimmers being needed only for surface work in connecting the lifting cables using easy fix attachments.

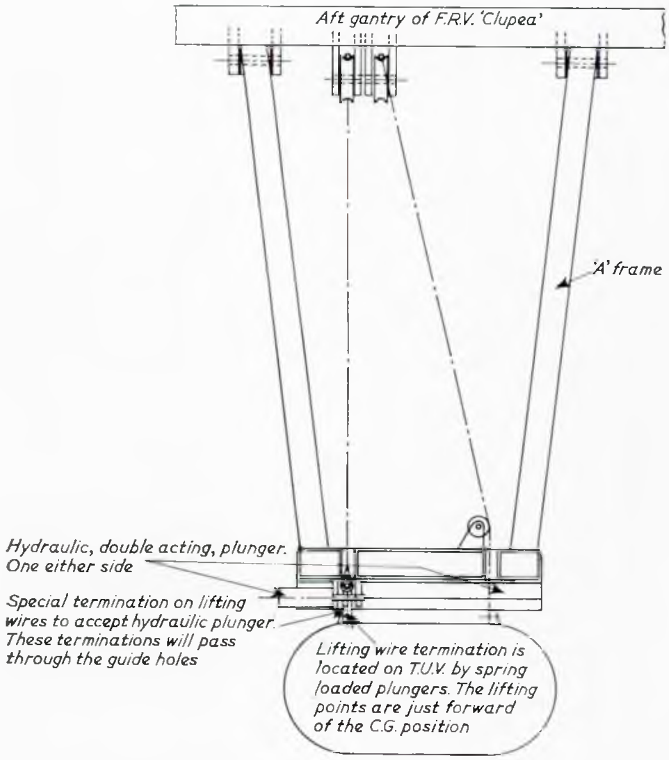

FIG. 5—*'A' frame (lifting wire system, locking method) for T.U.V.*

Trials

A full-scale mock-up of a TUV with identical weight, C of G, size, lifting attachments and top profile to that of a proposed vehicle was produced and used to carry out a series of handling trials during a short cruise on FRV *Clupea*. The system worked very well and one exercise was carried out while the ship was fishing. It proved to be easier to carry out the operation with the fishing gear deployed than without owing to the stabilizing effect on the ship of towing the gear. During these exercises a few minor design errors came to light (eg the fact that the lifting bolts on the vehicle were fitted the wrong way round for easy operation by a diver who was holding onto a moving 9/10 submerged vehicle) but in general the system worked easily in sea states up to 3+ with a typical time for complete launch, disconnexion, reconnexion, and recovery of six minutes. The total staff required to handle the vehicle was seven. Of these, three would normally be part of the ship's crew. They included:

i) diver for connexion of cable etc;
ii) diver as safety back up;
iii) boatman for divers' boat;
iv) winch operator for towing cable;
v) winch operator for lifting cables;
vi) aft gantry operator;
vii) deck operations' supervisor.

Future applications may extend the use of the vehicle to a 250 ft research vessel of a stern trawler type. As yet, it has not been decided how a vehicle would be handled there but an underwater vehicle would certainly have to be handled over the side by some special equipment as the fishing method on the new ship requires the whole of the aft deck area.

Discussion

MR. G. C. EDDIE, M.I.Mar.E. said that this paper brought out the fact that there was another classification possible: to divide submersibles into those intended to carry people who would make their living underwater and therefore could be specially trained and inured to the application, and those which would carry scientists and engineers who only wanted to go underwater occasionally as observers. This vehicle came into the latter class.

They had heard a lot that day about the technical differences between submarines and submersibles, but here was another type of vehicle, something rather more like a kite or a glider. On p. 219 there was mention of advancing the vehicle forward past the net, so there was a problem of manoeuvre along the line of advance. He gathered that the authors were not thinking of controlling such manoeuvres by varying the actual weight of the vehicle during its flight, but only by varying its lift-to-drag ratio and presumably also the length of cable. How was the cable length to be controlled—at the vehicle end or the ship end? Would there be a communications problem? No doubt Mr. Foster had been into response times and the useful range of the angles of incidence of the entire vehicle, etc. One could envisage some danger of the vehicle hitting the bottom when it plunged forward, unless there was some co-ordination of attitude and cable control, or did things happen slowly enough for this not to be a problem?

The paper mentioned the large transparent acrylic hemispheres through which the pilot and observer would look, and the problem of sealing these to the steelwork of the vehicle. He knew that the authors did not necessarily want to give details of the design of the seals, but had they made any progress towards the solution of this problem? They were clearly somewhat worried about the material used for these hemispheres. In the past, in the aircraft industry, it had been said that the aeroplane of the future would be made of glass. More recently this had been said about the submersibles of the future. Had the authors any comments on this?

COMMANDER R. MACK said that the first speaker had referred to the Russians as having stolen a march on us in this field but having published no reports. He had been right about the first thing, but not the second. As in the case of most research and development for commercial use (i.e. non military) they had published very fully.

Commander Mack then showed a press photograph published about eight years ago, of a small Soviet towed vehicle for observing trawls, constructed about fifteen years ago, and called "Atlanta—1" which looked rather like a small Messerschmitt fighter. It had been built in 1952, tested, and taken to the Mediterranean where it had done some work in the Gulf of Tunis, and at Hammamet Bay, two or three years later. Its mother ship was a trawler, and there was a considerable amount of published information about its layout. "Atlanta—1" was four and a half metres long and weighed one and a half tons. It was towed by a kilometre long cable and carried two men. It seemed to be very successful.

He said that he would draw the attention of the people concerned with research and development in this country to the fact that some people in Whitehall had information on what had been done by the Russians in this rather unique field, which could be useful to them.

The Soviet towed vehicle was rather different from the British one, but if one had had in mind the idea for putting forward a case for funds to build something of this sort, it must strengthen one's case to show that the Russians had been operating such vehicles for the past nine years—and most satisfactorily.

Authors' Reply

In answer to Mr. Eddie's question about the control of the vehicle, Mr. Foster said that he was right in assuming that the main method of control of depth during flight would be by adjustment to the length of the towing cable. This adjustment would be made by winching at the ship end. Fine control of depth, as well as yaw, could also be achieved hydrodynamically through the control surfaces on the vehicle. It was necessary to be able to disconnect the towing cable at the vehicle end from inside to allow for surfacing if the cable was caught up.

Continuous communication should have been satisfactory with the use of a slipring winch for the cable system together with a back up through-water communications set. Incidentally, the towing cable system had required special design consideration. Cable drag could be estimated (laboratory tank test results were available) but the results of interaction between cable and vehicle were difficult to predict with precision.

On the question of control surfaces it was recognized that the fin areas were small and undoubtedly the extent of hydrodynamic control would be affected by the incidence angle of the vehicle to the stream. It was hoped that a balanced system would be achieved in practice alleviating the needs for complex gearing or power assisted controls.

He thanked Commander Mack for his offer of help which they would certainly follow up. They had of course been in contact with several government and university departments as well as industry in connexion with towed vehicles, but really had found difficulty in obtaining information about Russian vehicles. A film which had been presented by Russian research workers at a recent international meeting had shown "Atlant" in operation. Special attention had been given to demonstrating its suitability for fishing gear research by showing fishing gear in action in midwater and on the seabed. He thought that that vehicle had been a version of the one which had been shown in Commander Mack's slide.

Mr. Cameron, in his reply, said that there were considerable problems in producing a seal for the hemispheres on the proposed TUV, e.g. the form of interface between two materials as dissimilar as perspex and steel, also the method of retaining the hemispheres (and thus the seal) under low pressure and yet enabling the perspex to flow under high pressure.

The normal "O" ring type of face seal was not practical in this case as the edge of the groove would have acted as a stress raiser. The proposed seal was a face seal with a rubber gasket covering the complete interface. The hemisphere was located by a large section "O" ring which would allow the hemisphere to be compressed and a clamp ring with a soft seat was used to retain the hemisphere under low pressure conditions. Two model hemispheres had been obtained which were roughly $\frac{1}{3}$ scale, and had the same DI/T ratio, and the sealing system would be tried out on these model hemispheres. As could be seen from Fig. 2, the hemispheres were in the most vulnerable position (as he was sure the pilot would agree) and for this reason a crash barrier had been included which should give some protection during launch, recovery and operation.

In his reply Dr. Hemmings said that he had been concerned with the more psychological aspects, from the user's point of view. From their diving operations, where they had been free-swimming on a gear which moved more slowly, and from looking at trawls at normal speeds, it was quite clear that all-round visibility—downwards upwards and sideways—in order to align the vehicle with respect to the two towing warps leading up to the ship, was important. They had shown that two people were essential; one purely as a pilot and the other as an observer. He had spoken to one of the Russians who had used "Atlant" (VN Martyshevsky) and he had conveyed to him the fact that one of their problems using it as a one-man towed vehicle had been that too much time had been spent in piloting the vehicle, leaving inadequate time for making useful observations. Therefore they had chosen the two-man vehicle, with dual control, in order to exchange roles under certain circumstances. In addition, with two people, there was the psychological advantage of having someone to talk to.

He felt that communication was the most important safety aspect of this towed underwater vehicle, because a problem could be prevented from arising if there was communication between the two people in the vehicle and those on the ship. There was a telephone line in the towing cable, duplicated by a through-water system. This allowed for communication with divers, where there was an exercise involving the simultaneous employment of free-swimming divers and the towed vehicle.

OPERATIONAL REQUIREMENTS IN THE DESIGN OF SUBMERSIBLES FOR UNDERWATER WORK TASKS

M. B. F. Ranken, C.Eng., M.I.Mar.E*

Commander Ranken

INTRODUCTION

Following the great enthusiasm for building submersible vehicles in the past few years, especially in the United States and often by aerospace companies with little real understanding of the sea, there is a growing realization that a large proportion of the seventy or so which have been built are little more than experimental, with almost no capability for doing useful work; many can only have been designed to give experience to their builders. There has been a tendency to build submersibles for the sake of building them, rather than with any particular underwater task in mind. In the same way as we talk about a diver without designating his special skill, these vehicles are usually called submersibles without any qualification, but it is important to stress that the "all-purpose submersible is a delusion"[57]. "Submersibles are highly mission-oriented in design and capability."[69]

"A submersible is a diving vehicle that is dependent upon surface support, whereas a submarine can operate independently of any other vessel."[3]

The word submersible covers both manned and unmanned vehicles of all kinds, and it is primarily with these that we are concerned in this paper. Conventional submarines have however been used for research and survey work, and at least one design of submersible comes near to being a true submarine, which is clearly the ideal for most kinds of work, on account of the vulnerability of submersibles to the elements during launching and recovery at the surface, as well as to their generally low endurance.[62, 68]

Although the naval submarine developed more or less directly as a vehicle which is intrinsically heavier than water, the manned submersible for peaceful purposes followed the bathysphere or bathyscaphe like Beebe's in 1934, the French FRNS 2 and 3 and the "Archimède", and Piccard's "Trieste" which descended to 35 800 ft in the Challenger Deep of the Marianas Trench in 1960. These vehicles are the nautical equivalent of balloons, and similarly unmanoeuvrable and slow underwater. Today's manned submersibles are at about the equivalent stage of development as aircraft were before World War I. On the other hand the technology is certainly available to develop these vehicles fairly quickly into useful, essential and versatile tools for the conquest and exploitation of "inner space".

The primary role of every submarine and submersible is, like diving, to take men or equipment to the work site, there to do various kinds of useful work. Their secondary role may also be to provide support in the form of power, lighting, buoyancy, tools, etc., as well as safe refuge, to facilitate the work to be done. Many variations are possible in the design of submersibles and underwater work systems, and it is the purpose of this paper to examine these in relation to the work tasks which are likely to be performed underwater.

The designer needs to know exactly what any particular vehicle is required to do, but it is the potential operator's responsibility to thoroughly brief him. The submersible must be able to do useful work at a more economical cost than any alternative method.

Man is Essential Underwater

"Man truly is only 'emerging' in his capacity to perform useful work on the ocean floor,"[76] but, whatever the present difficulties, he must go underwater if our knowledge of the seabed and of what goes on in the water volume is to progress at any reasonable rate or with any degree of accuracy. "Almost all deep submersible dives are made on ocean bottom never before seen by man . . . each dive reveals something new . . . a true indication of how little we know . . . Using the best, most up-to-date charts available, it takes only one submergence to readily recognize that the accuracy of surface-vessel recorded 'facts' degrades rapidly with increasing depth . . . Bathymetric charts of today are not exactly true representations . . . 'Man at the viewport' can see sufficient distance to perform man-machine tasks of medium complexity."[37]

Underwater Work

It is now established beyond reasonable argument that, however ingenious or sophisticated they may be, it is not sufficient to rely on instruments and equipment mounted in a surface vessel, or operated from one, to carry out efficiently a large number of underwater tasks. It is absolutely essential to take the best instrumentation or equipment underwater under proper direction by man, whether or not he too goes below. In the present state of development, many tasks cannot be undertaken satisfactorily without a man at or near the site, but it is also necessary to remember that present diving techniques do not yet permit a diver to perform useful work economically much below 600 ft, or possibly 800 ft†, since the equipment he needs to wear to safeguard him against the environment at this depth or lower, absorbs most of his work capability. Comex may have achieved the record depth of 1706 ft in the tank at Marseilles, and the Royal Naval Physiological Laboratory 1500 ft at Alverstoke, but there is a long way to go in developing the necessary diving systems and suits to permit divers to work as deep as this in the sea itself.

It is therefore clear that divers can only be employed on the bottom of about 7 per cent of the ocean bed, though it is true that so far there are few commercial requirements for them to go any deeper. Also, although a diver can go to this depth to work, he needs extensive support and refuge, and the amounts increase rapidly as he goes below 180 ft, the

* Managing Director Aquamarine International (Fisheries & Ocean Development) Ltd.

† Comex made a successful experimental working dive to 850 ft off Corsica in November 1970.

safe limit for continuous breathing of air under pressure.

Various complementary or alternative systems and vehicles become increasingly important below say 150 ft, and from about 1000 ft downwards submersible vehicles of one kind or another are the only means of conducting any work whatever beyond the limited capacity of instruments and surface-operated or towed dredges, trawls and grabs. For continental shelf conditions in Europe, and especially in the North Sea, of which the greater part is less than 300 ft deep, it could be argued that submersible vehicles have little application, since even when their existing limitations are overcome, there is the ever-present problem of low visibility. However there are many tasks now where this handicap is not decisive, and sonar, laser or other methods of overcoming turbidity will surely emerge. Where a lock-out is available, the value of the submersible is beyond any argument for diver transport and support in carrying out many tasks, and its role becomes decisive in deep and especially in saturation diving, as a means of extending and augmenting the diver's limited physical power, endurance and range. As in space, the diver needs his support vehicle.[68]

TABLE I—PRINCIPAL UNDERWATER ACTIVITIES—ACTUAL AND POTENTIAL

Exploiting Sea Assets	
Fisheries and aquaculture	fish, crustacea, molluscs, weed. Attraction, herding, catching, harvesting, processing
Mineral mining	coal salt, sulphur
Mineral dredging	sand and gravel tin iron sands light heavy minerals gold nodules
Oil and gas extraction	exploration, drilling completion, wellheads, storage, terminals, pipelines repair and maintenance
Desalination	
Chemical/mineral extraction from seawater	salt bromine magnesium and magnesium compounds
Harnessing physical/thermal energy of the sea	tides currents upwellings
Defence	
Submarines	
Anti-submarine warfare. Bottom listening devices	
Clearance of beach obstructions	
Frogmen and midget submarines	
Monitoring—instrumentation, habitats	
Highspeed submarine vehicles and weapons	
Submarine rescue and salvage	
Recovery of equipment—vessels, wrecks, torpedoes, bombs, instruments, etc.	
Recreation and Tourism	
Diving	
Habitats	
Submarine observations vehicles	

Table I lists the principal underwater activities, and Table II the underwater work involved in using the sea and exploiting its resources. Table III attempts to reduce all the work to a number of common tasks, together with the various actions needed to perform these tasks and the supporting functions required to enable the actions to be performed.

Oceanographers work from what they call "platforms" and Table IV lists the considerable number of these which are available to us, together with the approximate maximum depth to which each can presently go, and the kinds of equipment and personnel which they can carry.

One author has classified these into two groups as manned "remote" and manned "syntopic", a word which he has coined from the Greek ("syn", together; "topos", a place) to signify man directly on the job.[76] It is noteworthy that the majority of the available systems place man close to the work, on account of the complexity and unreliability of remote control underwater, though, as will be discussed later, few provide him with adequate tools, accessories and support with which to facilitate his work. Furthermore, far too few vehicles exist which have been designed as part of the system, rather than in isolation, and many do not meet life support and safety requirements.

In deciding how to tackle any underwater task, it is important to use a systematic approach. Our aim should be whenever possible to choose or devise the simplest and most economical way of carrying out the work involved. Experience has proved that submersibles "are only one part of a system, all parts of which must work just as well".[50] Their use or otherwise depends on whether they are cost-effective in comparison with other systems. They are most useful on or near the bottom, and for some work are the only method which can be used effectively. "There is already a satisfactory commercial case for a number of 1000 ft submersibles with a diver lock-out capability."[3, 63]

Perhaps in view of the high cost of working underwater the other major recommendation in tackling any particular task is to make the best possible use of information already existing on charts, in records, from authorities and people with local knowledge, and obtained by surveys from the surface, using echo sounders, sonar, gravity meters, magnetometers, galvanic potential devices, photography or television, sampling, divers, observation chambers, etc. This research should be done and its shortcomings assessed before committing major equipment or personnel underwater. "Time spent in reconnaisance is never wasted."[17]

LIMITATIONS OF UNDERWATER SYSTEMS

The limitations of many observation and work tasks conducted from surface vessels and platforms have been indicated earlier, but it is thought desirable to analyse some of the pros and cons of the various underwater systems from the work point of view.

(1) *Weather in Relation to Launch and Recovery*

Since every underwater work system except the full-scale submarine is dependent largely or entirely on surface support from a vessel or platform of some kind, all these systems are fundamentally affected by the prevailing weather. Prince Bernhard said in Den Helder in June 1970 that "fear of the sea is the beginning of maritime wisdom", and satisfactory surface conditions for the launch and recovery of personnel and equipment are essential for safety.

Seaworthiness is a vital necessity, coupled with simplicity and ruggedness. Submersibles must have adequate stability both underwater and on the surface; some have neither, and many are vulnerable on the surface through lack of freeboard; they lack an adequate sail, fixed or retractable, to protect the entry hatch while it is open.

In the operation of the submersible "Alvin" and her support vessel *Lulu* during 1967, it was found that 30 per cent of potential diving time on station was lost due to weather, and 20 per cent in transit to and from the working area. The

TABLE II—PRINCIPAL UNDERWATER WORK

Observation	
Exploration —	scientific archaeological specific
Inspection —	ships and rigs structures pipelines and cables instrumentation
Observations, surveying and charting —	hydrographic topographical (stereo-photography, side-scan sonar) cable and pipeline routes, sites in detail geological, geophysical; bottom, sub-bottom (sonar, magnetometer, gravimeter, seismometer) fisheries, biological resources, behaviour, fishing methods, fauna, reefs physical soil mechanics, sediment transport, engineering properties, site conditions
Measurement/monitoring, oceanographic data —	tides, currents, waves, stratification, motion, salinity, temperatures, upwellings, chemistry, radioactivity, light propagation, deep scattering layer, thermocline meteorological pollution engineering
Data collection and processing —	scientific, ecological acoustics (sound propagation, voice) engineering
On-site assay of minerals	
Search, Rescue, Recovery	
Location	wrecks, ships, submarines, aircraft, structures equipment, flight recorders, torpedoes, bombs, engines bodies, divers
Rescue	divers, submarine crews
Recovery/lifting	sunken vessels, wrecks, aircraft, structures hardware valuable cargoes
Prepare for/direct	lifting
Search, Rescue, Recovery—cont.	
Wreck disposal/removal	
sampling	sediment, outcrops, cores, nodules—grabs, trawls deep ocean drilling
Work	
Transport	men, material, equipment cargo—oil, bulk materials
Cleaning and preservation	underwater portions of ships, rigs, structures, moorings, etc.
Repair and maintenance	
Construction and operation, maintenance and repair of bottom-sited installations	mining aquaculture power stations—nuclear, tidal terminals, artificial islands storage tanks—oil, bulk materials, goods tunnels pipelines, outfalls, cooling water culverts submarine cables burying pipelines and cables
Emplacement of moorings, cables	navigation/survey aids, reference points (transponders), systems, traffic control systems, etc.
Oil field completions	valves, risers, blow-out controls, platforms, crude treatment plant, flow controls, etc.
Installation of foundations for bridges, lighthouses, towers, structures, harbours, locks, breakwaters, etc.	
Excavation and levelling of sites	dredging and reclamation, piling
Demolition and blasting	placement of explosives
Provision of diver support and work base	
Evaluation	equipment—optics, sonar navigation, tracker, communications, tools, instruments, etc. personnel

"Alvin's" overall utilization covering 55 dives, averaging 5·86 hours each to depths between 26 and 6250 ft, was only 45 per cent out of an expected total of 120 dives, with weather and its prediction being the most significant factor.[59] In diving, the time required for descent and ascent, especially the latter because of decompression stops, is a considerable and often a major additional loss of available working time. Time lost in reaching the working area is naturally greater the further the site is from base, but this is due not only to the distance, but also to the frequency of weather deterioration in transit and "it must be emphasized that the underwater aspects of diving operations cannot be divorced from the surface functions".[17]

In general the launch and recovery of submersibles is

TABLE III—SUMMARY OF WORK TASKS AND MEANS FOR THEIR EXECUTION

Tasks	*Actions*	*Support Functions*
Observation/direct	Sample/core	Viewing
Inspection	Measure	Lighting
Survey	Drilling	Sensing
Exploration	Clean	Power
Search	Grab/release	Heating
Emplacement	Attach/anchor	Endurance/speed
Transport	Assemble	Life support
Replenishment/refuelling	Cut/weld	Photography/T.V.
	Hammer/chip	Communications
Rescue	Torque	Manoeuvre/hover
Recovery	Mating	Navigation/position fixing
Operate	Injest/dig	
Maintain	Lifting	Logistics
Service/repair	Moving	
Salvage	Towing	
Trenching	Alignment	
Mining	Lock-out—Wet/dry	
Construction		

seldom attempted except in emergency at a wave height greater than about six feet, corresponding to sea state 3/4 (Table V), although it is reported that Vickers' "Pisces" has been recovered in sea state 5/6.[24] Handling any small submersible, tools, or equipment in the sea is always difficult, as has been demonstrated so graphically on television during every recovery of astronauts from their space capsules.

An additional cause of lost time is the preference for daylight launching and recovery, which automatically reduces the available time by at least half; it may in fact reduce it even more drastically if transit times are also partly in daylight.

The overall weather factors are probably worse in United Kingdom waters than in any other important offshore area, and this accounts for the under-utilization of "Pisces" and other submersibles, as well as for the serious losses and failures which have occurred. It is stated that a relative motion of 14 ft can be accommodated between "Pisces" and her support vessel *Vickers Venturer* which corresponds only to sea state 3/4. The limitation of launch and recovery to this sea state results in a drastic reduction of the operating times available in many areas. Thus during the winter months, only 44 per

TABLE IV—PLATFORMS FOR SURFACE AND UNDERWATER WORK

Medium	*Platform*	*Approx. Max. Submergence Depth ft (to date)*	*Personnel/Equipment*
space	satellites	0	cameras, sensors, data collection
atmosphere	balloons aircraft kites	0	observers cameras sensors, data collection cargo (drops)
	helicopters sea planes	0	observers (remote) cameras sensors (airborne and dipping) data collection passengers equipment, transport
air/sea interface	ships and vessels (all types)	0	observers (remote) cameras and t.v.
	hovercraft, etc.	0	sensors, data collection and processing
	drilling rigs, ships	0	samples
	floating structures	0	drilling and coring equipment
	towing, fixed structures (bottom supported)		dredge, trawl, grab passengers and workers (surface) equipment-transport and use (surface) surface support for underwater vehicles/work
underwater	diving bell	160	observers (underwater)
	skin divers (free)	90	workers
	scuba divers	200	cameras and t.v.
	diver transport vehicles	300	sensors, data collection
	oxy/helium divers	800*	samplers
	standard diving dress	600	drilling and coring equipment, underwater tools, manipulators
	armoured diving dress	800	dredge, trawl, grab
	personnel transfer chamber (SDC)	600	passengers and workers (underwater)
	towed underwater vehicles manned or unmanned	600	
	submersibles with diver lock-out	1000	equipment transport and use underwater
	submarines with diver lock-out	2000	
	observation chamber	1500	
	rescue submarine with dry lock-out	3500	
	telechiric- cable-controlled	7000	
	untethered	20 000	
	submersibles	20 000	
	bathyspheres	35 800	
	free buoys	15 000	
seabed	seabed vehicles/machines — manned and unmanned	600	observers, operators/workers
	habitats with diver lock-out	600	cameras and t.v.
	self-contained power packs	600	
	one atmosphere chambers (building)	1200	sensors, data collection
	submerged structures	1500	samplers drilling and coring equipment underwater tools welding and cutting equipment work base, living quarters, support

* World record simulated dive at Marseilles two men at 1706 ft for one hour (19 November 1970). Experimental working dive off Corsica, three men at 850 ft for $3\frac{1}{2}$ hours per day for 8 days (September 1970).

TABLE V—WAVE CHARACTERISTICS AT DIFFERENT SEA STATES AND CORRESPONDING WIND STRENGTHS FOR FULLY ARISEN SEA

(From Myers, J. J., Holm, C. H., and MacAllister, R. F., "Handbook of Ocean and Underwater Engineering". McGraw Hill Book Co. 1969, Table 11–10)

Sea state	Description	Wind				Sea								
		Beaufort wind force	Description	Range, knots	Wind velocity, knots†	Wave height, ft: Average	Wave height, ft: Significant	Wave height, ft: Average 1/10 highest	Significant range of periods, sec	T_{max}, period of maximum energy of spectrum	T average period	l average wave-length	Minimum fetch, nmi	Minimum duration, hr
0	Sea like a mirror.	0	Calm	Less than 1	0	0	0	0						
	Ripples with the appearance of scales are formed, but without foam crests.	1	Light airs	1–3	**2**	0·05	0·08	0·10	Up to 1·2	0·7	0·5	10 in	5	18 min
1	Small wavelets, still short but more pronounced; crests have a glassy appearance, but do not break.	2	Light breeze	4–6	**5**	0·18	0·29	0·37	0·4–2·8	2·0	1·4	6·7 ft	8	39 min
2	Large wavelets, crests begin to break. Foam of glassy appearance. Perhaps scattered white horses.	3	Gentle breeze	7–10	8·5 10	0·6 0·88	1·0 1·4	1·2 1·8	0·8–5·0 1·0–6·0	3·4 4	2·4 2·9	20 27	9·8 10	1·7 2·4
3	Small waves, becoming larger; fairly frequent white horses.	4	Moderate breeze	11–16	12 **13·5** 14 16	1·4 1·8 2·0 2·9	2·2 2·9 3·3 4·6	2·8 3·7 4·2 5·8	1·0–7·0 1·4–7·6 1·5–7·8 2·0–8·8	4·8 5·4 5·6 6·5	3·4 3·9 4·0 4·6	40 52 59 71	18 24 28 40	3·8 4·8 5·2 6·6
4 5	Moderate waves, taking a more pronounced long form; many white horses are formed (chance of some spray).	5	Fresh breeze	17–21	18 **19** 20	3·8 4·3 5·0	6·1 6·9 8·0	7·8 8·7 10	2·5–10·0 2·8–10·6 3·0–11·1	7·2 7·7 8·1	5·1 5·4 5·7	90 99 111	55 65 75	8·3 9·2 10
6	Large waves begin to form; the white foam crests are more extensive everywhere (probably some spray).	6	Strong breeze	22–27	22 24 **24·5** 26	6·4 7·9 8·2 9·6	10 12 13 15	13 16 17 20	3·4–12·2 3·7–13·5 3·8–13·6 4·0–14·5	8·9 9·7 9·9 10·5	6·3 6·8 7·0 74	134 160 164 188	100 130 140 180	12 14 15 17
	Sea heaps up and white foam from breaking waves begins to be blown in streaks along the direction of the wind (spindrift begins to be seen).	7	Moderate gale	28–33	28 30 **30·5** 32	11 14 14 16	18 22 23 26	23 28 29 33	4·5–15·5 4·7–16·7 4·8–17·0 5·0–17·5	11·3 12·1 12·4 12·9	7·9 8·6 8·7 9·1	212 250 258 285	230 280 290 340	20 23 24 27
7	Moderately high waves of greater length; edges of crests break into spindrift. The foam is blown in well-marked streaks along the direction of the wind. Spray affects visibility.	8	Fresh gale	34–40	34 36 **37** 38 40	19 21 23 25 28	30 35 37 40 45	38 44 46·7 50 58	5·5–18·5 5·8–19·7 6–20·5 6·2–20·8 6·5–21·7	13·6 10·3 14·9 15·4 16·1	9·7 10·3 10·5 10·7 11·4	322 363 376 392 444	420 500 530 600 710	30 34 37 38 42
8	High waves. Dense streaks of foam along the direction of the wind. Sea begins to roll. Visibility affected.	9	Strong gale	41–47	42 **44** 46	31 36 40	50 58 64	64 73 81	7–23 7–24·2 7–25	17·0 17·7 18·6	12·0 12·5 13·1	492 534 590	830 960 1110	47 52 57
	Very high waves with long overhanging crests. The resulting foam is in great patches and is blown in dense white streaks along the direction of the wind. On the whole, the surface of the sea takes a white appearance. The rolling of the sea becomes heavy and shock-like. Visibility is affected.	10	Whole gale*	48–55	48 50 **51·5** 52 54	44 49 52 54 59	71 78 83 87 95	90 99 106 110 121	7·5–26 7·5–27 8–28·2 8–28·5 8–29·5	19·4 20·2 20·8 21·0 21·8	13·8 14·3 14·7 14·8 15·4	650 700 736 750 810	1250 1420 1560 1610 1800	63 69 73 75 81
9	Exceptionally high waves (small and medium-sized ships might for a long time be lost to view behind the waves). The sea is completely covered with long white patches of foam lying along the direction of the wind. Everywhere the edges of the wave crests are blown into froth. Visibility affected.	11	Storm*	56–63	56 **59·5**	64 73	103 116	130 148	8·5–31 10–32	22·6 24	16·3 17·0	910 985	2100 2500	88 101
	Air filled with foam and spray. Sea completely white with driving spray; visibility very seriously affected.	12	Hurricane*	64–71	>64	>80‡	>128‡	>164‡	10–(35)	(26)	(18)	~	~	~

* For hurricane winds (and often whole gale and storm winds) required durations and fetches are rarely attained. Seas are therefore not fully arisen.
† A bold figure means that the values tabulated are at the centre of the Beaufort range.
‡ For such high winds, the seas are confused. The wave crests blow off, and the water and the air mix.
SOURCE: W. A. McEwen and A. H. Lewis, "Encyclopedia of Nautical Knowledge", p. 483, Cornell Maritime Press, Cambridge, Md., 1953. "Manual of Seamanship," pp. 717–718, vol. II, Admiralty, London, H.M. Stationery Office, 1952. Pierson, Neumann, James, "Practical Methods for Observing and Forecasting Ocean Waves," New York University College of Engineering, 1953.

cent of the available working days on the oil rigs are better than this (Table VI), and it is not surprising that most work is done during the summer period when the available days rise to 66 per cent.

It is well known that wave movement diminishes rapidly as we submerge, so that in sea state 6 the average 1/10th highest wave reduces from 20 ft on the surface to only 10 in at 90 ft. It is therefore clear that the best way of reducing or eliminating the effects of weather is to go below the surface. The following are some of the methods which have been tried or proposed:

(i) The twin-hulled offshore work vessel *Duplus* was built for Netherlands Offshore Company in The Hague. Apart from her many other novel features, *Duplus* comprises two submerged submarine hulls supporting a 131 ft long by 56 ft wide working platform some 20 ft above the sea surface, through two thin veins of very small waterplane area. The submarine hulls are well below the main wave action and therefore remain extremely stable in very rough sea conditions. They are connected together below water by two fixed hydroplanes, in each of which are situated two Voith Schneider propellers. When working it has been found that the enclosed area of sea between the two hulls remains extremely calm and ideal for diving operations, or with a larger vessel of similar design, for launching and recovering submersibles or other underwater work vehicles.

(ii) Launch and recovery of submersible vehicles and perhaps divers could be done by means of a submerged platform or

TABLE VI—SEA CONDITIONS AT SELECTED UNITED KINGDOM OFFSHORE STATIONS

(Based on Putman, J. L., April 1970, "Submersible Vehicles for the U.K. Continental Shelf", Society for Underwater Technology)

	Percentage time below sea state indicated							
	April–Sept.				Oct.–March			
Sea State: up to	2	3	4	5	2	3	4	5
Morecambe Bay, Irish Sea	81	92	98	99·5	58	77	90	99·7
Seven Stones, Land's End	17	40	67	88	95	26	49	83
Dover Strait	66	83	94	99·7	45	69	87	98·7
Smith's Knoll	64	88	97	100	42	71	87	99
Gas Rigs, North Sea*	40	53	66	82	21	31	44	62
Corresponding Rig Operating Days*	73½	97	121	150½	38	57	79½	113½

* Average over seven areas for three years, and corresponding number of days.

drydock towed behind the support vessel.[61] Makai Range Inc., a component organization of the Makapuu Oceanic Centre in Hawaii, has recently built a launch recovery and transport vehicle (LRT), 33 × 17 × 2·5 ft deep of catamaran configuration, and used it successfully off Oahu in sea state 5 conditions. The first unit has a lifting capacity of 10 000 lb but a larger one of 50 000 lb capacity is at the planning stage. The LRT is one of a series of developments employing controlled buoyancy techniques for work in the ocean. It is towed to the work site with the submersible secured to its deck; the crew man the submersible and scuba divers open the flood valves on the LRT, check that everything is ready, and then open the vent valves. Both hulls then flood and the complete assembly submerges to an appropriate depth below the wave action, around 50 to 60 ft, at which it hovers. After releasing the submersible, the LRT returns to the surface until required. When recovering, the procedure is reversed.[21] This technique obviously points the way to many possibilities for underwater work, including its use for diving operations with a submersible workbase/habitat, and possibly a personnel transfer chamber (PTC), hovering at a convenient depth and suitably anchored to the bottom. The construction assistance vehicle (CAV), recently tested off the Californian coast by the U.S. Naval Civil Engineering Laboratory, and the buoyancy transport vehicle (BTV) developed by the U.S. Naval Undersea Research and Development Centre, with a buoyancy of 1000 lb at 850 ft depth provided by a hydrazine fuel cell, are two vehicles in this category; both are electro-hydraulically propelled from batteries. This type of vehicle should be more secure for many saturation diving operations than any surface craft or ship. The possibility also exists of combining such a submerged base with a specially designed surface vessel to which the base could return, perhaps from below, and from which power and other services were obtained by umbilical.

(iii) A mother submarine or a bottom-sited work-base could be used, from which supplies and fuel could be obtained and where batteries could be charged. This is the principle to be used by the U.S. Navy's air-transportable deep submergence rescue vessel (DSRV), which can be carried to sea pig-a-back on a submarine until it is close to the search area. It can then be released to go and locate the stricken vessel, mate with it, and rescue the crew for transfer to the first submarine.

Cammell-Laird's "Crawler" (seabed vehicle) and various habitats come close to the requirements for a base on the bottom, although these are all designed primarily for diver operations, and therefore have wet lock-outs and chambers at ambient pressure.

(iv) The final answer is the full-scale submarine, completely autonomous, returning to harbour on completion of its work, or with sufficient endurance to await better weather before surfacing in the open sea. Various naval submarines have been used from time to time for scientific and other observations, and the U.S.S.R.'s *Severyanka* was modified and used for fisheries research. Recently the requirement to make extensive surveys of the seabed under the ice of the Arctic Ocean has led Marine Resources Consultants Inc. of California to buy from Sweden three 1000 ton *Neptune*-class submarines built in 1943 and convert them for seismic survey work.

The U.S. Navy's two oceanographic submarines, the conventional *Dolphin* and the nuclear *NR1* with a pressurized water reactor, are believed to be the only two research submarines so far built for the purpose, although France's very advanced *Argyronète,* with a crew of ten, a diver lock-out, seven days endurance on the surface and three to eight days submerged at 1970 ft, will perhaps be the first genuine commercial work submarine independent of surface support. It is noteworthy that this vessel has a displacement of 260 tons on the surface, compared with the majority of existing submersibles which are between five and fifteen tons, with a few as low as one ton; it is nowadays considered that the optimum minimum size is something over 100 tons for most operations in deep water.

(2) *Underwater Tools*

As indicated in Table III many of the tasks and actions to be performed underwater are exactly the same as on dry land, but this does not mean that tools designed to work in the atmosphere are likely to work in the entirely different water environment. This is a problem common to all underwater equipment, and a large proportion of it does not work initially. Some of it never does. These failures are not only due to problems of corrosion and the entry of water, but also to pressure and in some cases to weightlessness. The essential lesson is therefore that all equipment for use underwater must be properly designed for it.

Apart from many types of instruments, lighting and cameras required for observation, surveying and recording, a variety of tools are required to perform the actions listed in Table III. Submersibles require in particular remote-controlled manipulators to take the place of a man's arms and hands, and it must be said that none of these have so far been satisfactory. They have been clumsy and difficult to control, and their successful use on some vehicles has been solely due to the dexterity of the operator. "The manipulator design must be developed from a particular mission requirement, and designers must provide a total system solution, which integrates the boat and the manipulator, maximizing their combined capabilities.[76]

The versatility and value of the submersible and diver alike depend greatly on the utility of the tools and accessories fitted to or provided for them. These include all types of manually and power-operated hand tools, welding and cutting gear, drills, hammers, cutters, chisels, saws, grabs and tackle, lifting capacity, cleaning and grinding units, corers and samplers, etc. Power tools must be reactionless on account of weightlessness, and hydraulic and electric drive seem best for deeper water, since air tools are subject to the back pressure due to the water depth and are also vulnerable to corrosion from water entering the power units.

Submersibles can carry power packs to assist divers, although these can also be independent and dropped onto the seabed; much development remains to be done, but usable fuel cells and closed-cycle Diesel-generators are already available. Lock-out submersibles can also provide life support and heating through umbilicals.

Far too little design effort has been devoted to underwater tools and accessories and the cost in lost time and unsatisfactory work must far exceed what it would cost to develop really good tools, properly tailored to the job and the environment.

Underwater systems and submersibles need to have a satisfactory work capability; few have yet achieved this.

(3) *Logistic Support*

The limiting factor in all underwater systems tends to be the logistic support from the surface, and far too little thought has been devoted to this essential part of the system. The underwater vehicle needs to be designed as an extension of the surface or other vessel concerned, but it is equally important that the surface vessel is properly integrated into the overall system, since normally it is the most expensive part to operate in terms of man-power and steaming and waiting times.

(4) *Endurance*

Perhaps the weakest feature of every underwater work system at present in service is its endurance and this involves personnel as well as material. The limitations on the diver are imposed by the system and equipment in use, the depth and his ability to maintain body temperature in relation to the ambient water temperature. The time underwater determines the facilities required.

Endurance is closely related to surface support, weather conditions, sea temperature, the capacity of the life support system, and above all to the availability of sufficient power. No existing underwater power source provides any margin for heavy or long power consumption, such as is needed for welding or heavy tools, not to mention high submersible speeds, so that when these are needed power must be obtained by umbilical cable from the surface, introducing yet another weak link in the system. Electric batteries appear to have reached their limit and as stated earlier much work remains to be done in developing large-capacity fuel cells, closed-cycle engines, or best of all a really cheap, compact and lightweight nuclear reactor.

The endurance of a submersible or submarine also depends on the degree of crew comfort provided, and this in turn dictates the size of the vessel. Sanitary, cooking and sleeping facilities begin to loom larger the longer the endurance required. Thus *Ben Franklin,* which had to drift for 30 days with a crew of nine men in the Gulf Stream displaces 130 tons.

(5) *Navigation and Tracking Underwater*

Since visibility underwater seldom exceeds 150 ft, and in many areas around our costs, especially in ports and estuaries, may be down to a few inches, we cannot rely on vision, even assisted by powerful lights, for safe navigation or tracking; nor is dead reckoning adequate or safe underwater with three dimensions involved. Radio frequency waves are not usable underwater and we are left with acoustic signalling. But this too has limited range and is subject to unreliability due to bending and attenuation of the waves by thermal and other layers in the water body. Thus for controlling an untethered telechiric (Greek "tele", far; "cheir", hand) vehicle like "Sea Drone I", with a depth capability of 20 000 ft and a range of 36 miles at six knots, the acoustic range of the equipment is only six miles; there must therefore be a fear of losing such unmanned vehicles, quite apart from any fault in the control and other systems.

"Navigational accuracy is fundamental to all oceanographic research"[30] and this is equally so in all search and survey work, and in many engineering operations. Accuracy needs to be within a very few feet, and sometimes less, in three dimensions. Accurate depth measurement below the vehicle and real time tracking from the surface support vessel have also been found important.[59] Pinger and transponder arrays on the seabed are also of great use in many cases to act as local reference points, and Döppler sonar is available for velocity measurement. Very elaborate requirements are laid down for the U.S. Navy's deep search and rescue vessel (DSRV) but would not be justified in most, if any, commercial vehicles.

This is a field of considerable difficulty and much remains to be done to improve the methods used in relation to the work and to safety requirements. It is as important for shallow water diver transport and recreation craft like the Rebikoff *Pegasus* used for hull and pipeline surveys, the French *Totalsub 01* and the Perry *Shark Hunter,* as it is for operation in the deep ocean.

(6) *Communications and Telemetry*

So far as untethered submersibles are concerned, both communications and telemetry must rely on underwater acoustics, and communications between the surface support vessel and the vehicle are vital for safety. Telemetry is needed to control untethered telechirics, and for receiving information from these and from such units as fishing vessel trawls towed a long way astern of the trawler. Divers also use telemetric methods of maintaining communications and location, but range is limited.

Where an umbilical is used, there is no particular difficulty in transmission, but divers involved in deep saturation diving are breathing helium, and there is still a great need for reliable unscrambler systems. These are needed not only for divers and personnel transfer capsules, but also for habitats and lock-out submersibles maintained at ambient pressure.

(7) *Life Support Systems*

One of the limitations on the endurance of the small lock-out submersible is the volume of breathing gas which can be carried. Closed-circuit breathing apparatus helps to conserve available supplies, but for long endurance the size of the vehicle must go up to provide more space and greater gas storage capability. "The next step in diving evolution—the marriage of the diver, the submarine and the habitat."[68] The diver's safety and comfort will be increased at the same time as his range and work capability.

(8) *Cost*

One of the major handicaps to the use of submersibles is the high capital, and particularly operational cost of nearly all of them. In consequence, other less appropriate and often more dangerous methods continue to be used. For many tasks surface vessels are preferred because they appear cheaper and can often work 24 hours a day, compared with eight to ten, nearly always in daylight, by the submersible. They are also far less affected by weather, though this is not true of any alternative underwater system, except the fully autonomous submarine.

The main reason for the high operating costs is the need for surface support, which usually involves a full-scale ship and always a large number of personnel, at least ten times the number in most small submersibles, or for that matter deep diving operation. There is also the high cost of transit and waiting times.

Complete independence of surface support, perhaps with a submerged base or submarine mother ship, would be the ideal for many purposes, but a system like the U.S. Navy's DSRV, which comes near to this, is totally uneconomic for commercial work, since, with the attendant nuclear submarine, the system costs around $100m to build. No civilian alternative is yet in sight.

Proper attention to the design of the whole support and logistic system in conjunction with the submersible and the work requirements, from the original conception of the project, can certainly help to reduce overall capital and operational costs. Already "economic and design analyses have demonstrated the cost-effectiveness of and logic for the presence of man in offshore oil operations on the ocean floor". This refers to concepts involving well control installations and wellhead chambers on the seabed, with men transferring to them at one atmosphere from submersible transfer vehicles.[16]

The submersible and submarine come increasingly into their own for seabed operations the deeper we go, and irrespective of the cost, but costs must come down as in all other offshore work. It is salutory to remember that the cost of a personnel transfer chamber for deep diving operations is only about one sixth that of a small submersible, and can perform many of the same tasks at diver depths, where mobility is not essential. Of course operational costs will not come down to that extent, because the surface support for a personnel transfer chamber, including deck compression chamber, compressors, gas purification plant and the like, will probably cost more than the equivalent needs of a dry submersible, though not necessarily those of a lock-out submersible.

SAFETY, REGISTRATION, INSURANCE, LAW

The death in September 1970 off California of one member of the crew of the small submersible "Nekton Beta" was the first recorded fatality in U.S. commercial submersibles. It is noteworthy that this vehicle was not insured at Lloyds.

Other parts of the offshore industry do not have such a good record, and insurance premiums today on oil rigs are four times what they were twenty years ago. Diving also has a dubious record, though many accidents are kept quiet to avoid rises in insurance premiums.

The rapid development of the commercial submersible over the past decade has been largely based for structural design purposes on naval submarines; it has also resulted in a recent statement that "as soon as they are launched they are already out of date". Many are "laid up as being too expensive and sophisticated to operate economically".[11]

In an effort to stabilize the situation, the Marine Technology Society (MTS), in the United States, published in 1968 *Safety and Operational Guidelines for Undersea Vehicles*. These lay down sensible standards for the design and operation of submersibles, and based on them a standard MTS insurance contract has been drawn up to cover this class of vehicle and permit owners and operators who comply with the guidelines to obtain sensible rates for insurance. A similar scheme is under consideration for divers. The American Bureau of Shipping, Lloyd's Register of Shipping and Germanischer Lloyd are now able to classify submersible vehicles and other underwater equipment, and such classification is virtually essential to obtain insurance cover. Insurance stands in a regulatory role; if design and operational standards are raised, the premiums can be brought down.[11]

Since most submersibles are not autonomous at any commercially viable size, surface support is essential, and the MTS guidelines cover the whole system. Furthermore all work, equipment and personnel are likely to be registered or licensed under legislation now before the U.S. Congress. Similar requirements are believed to exist in some other countries, and are needed wherever these vehicles are operated.

The guidelines cover all aspects of design, construction, quality control, pre-dive checks and records, and the rest. While in no way unreasonable, they are based on aircraft principles, with which there are many parallels. A substantial section deals with the selection of personnel, and points out that "in many respects the scope of the pilot's attention and tasks are similar to those of an aviator or spacecraft pilot, yet he does not have the sophisticated computers or automated equipment to assist him".[38]

A Deep Submersible Pilots Association has been founded in the United States, which is dedicated to improve training and operational standards. It encourages the keeping of qualification, experience and dive logs. A similar organization, the International Association of Professional Divers, has been formed with similar aims for the diving fraternity.

The guidelines for operations are all concerned with sensible and prudent precautions, and pre-dive checks, in fact with good seamanship. One important recommendation is that naval and coastguard authorities and any emergency search and rescue organization should be informed of every dive and dive plan. A Mutual Assistance Rescue and Salvage Plan (MARSAP) was established in 1967, on the initiative of Lockheed, to provide immediate help between civilian operators in case of emergency. The keys to success of MARSAP, as demonstrated in October 1969 when "Nekton" released "Deep Quest" from a rope entanglement at 430 ft, were standardized communications frequencies, the fact that the support vessel *Transquest* knew exactly where "Deep Quest" was trapped from her track record, and the ability of "Nekton" to home on "Deep Quest's" sonar. Some similar organization to MARSAP is urgently needed in European waters and standard frequencies need to be agreed, as at present there is "dreadful chaos".[18]

CONCLUSION

The present stage of development of the submersible is not far beyond that of the aircraft about 1914. Few people are prepared to back it with money, despite some considerable successes, like the recovery of the nuclear bomb from 2850 ft in 1966, of the submersible "Alvin's" manipulator arm from 4500 ft in 1967 of the "Alvin" herself from 5100 ft in 1969, and of the 95 ton *Emerald Straits* from 670 ft in 1969. The traditions of the marine industry outside defence do not encourage such heavy spending on development as is commonplace in aerospace, and in consequence the rate of development of economical underwater vehicles and systems has been slow, and this is likely to continue until there is a demand to work in depths beyond any possible reach of divers. This is most likely to happen first in the offshore oil industry. Progress has also been retarded by a lack of adequate market research on a worldwide scale.

There is certainly a place for both the manned and the unmanned submersible system, and even some existing telechiric systems have been successful, though probably not economic, e.g. "Curv I" on the hydrogen bomb, and her later versions on torpedo recovery at depths of 2500 ft and more, well below a diver's capability.

The development of better tools and work systems for use with submersible vehicles would certainly help to silence the argument over whether or not to use a submersible. The main valid criticism to date is that they have mostly been unable to perform useful work. Probably shallow water diver transport vehicles with equipment for the underwater cleaning, inspection and survey of the hulls of large tankers and bulk carriers, as well as of offshore structures and oil rigs, will be the first to emerge as viable propositions. The spectacular photographic surveys of a damaged 250 000 ton tanker, performed by the Rebikoff "Pegasus" diver-controlled vehicle are an encouraging example of this.

A great advantage of the dry submersible is its ability to take engineers and scientists underwater with a minimum of difficulty and training to familiarize them with the hostile environment. The observation chamber has the same advantage, and this is why the Japanese have built special barges with these chambers for use on offshore construction sites in 50 ft and more of water; the intention is to build at least one for some 200 ft, which covers much of their continental shelf. Low visibility is of course a handicap, but the vehicles can carry powerful lighting systems and high-resolution cameras, far better than the human eye for record purposes.

Our detailed knowledge of the underwater world is extremely limited, and our ability to do useful work in it is even more so. The development of efficient cost-effective systems for working in the ocean and on the sea floor are pre-requisites to ocean exploitation. They should also serve to bring together the many disciplines involved in the sea and break down their present divergence. This is what oceanology and ocean industry are all about.

The submersible has an important part to play in the future development of ocean space, but it must be properly designed around specific tasks, and provided with the tools, facilities, logistic support and above all competent men to permit it to form part of an effective and economic system capable of performing useful work.

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Discussion

MR. G. W. R. COOKE said that the author of the paper had covered the general aspects of the subject comprehensively and informatively but it was significant that only eight lines were devoted to a vague mention of Diver Transport Vehicles.

The DTV, using the author's own definitions, had been used either as a submersible or as a submarine. It was probable that the latter applications would mostly be military except where DTV's were operating in such good conditions of sea and visibility that they could operate from a shore base.

The launch and recovery problems of a DTV were similar to those of a dry submersible but with the distinct advantage of lower weight. The risk of losing a DTV by a surface accident was very slight because flooding through the entry hatch could not occur.

The main advantages of a DTV over a dry vehicle were those of speed, handling and, most important, the ability of the crew to leave the craft at any time without problems of pressure differential.

A streamlined DTV might travel faster than ten knots in applications where speed was important. Normally, such a speed would only be used in mid-water. When near the bottom, slower speeds were essential for safety reasons and to allow time for the collation of data.

Data might be gathered in the forms of visual observation, photography, photogrammetry, metal detection and magnetometry in addition to measurements of temperature, salinity and turbidity.

It was Mr. Cooke's opinion that the DTV would begin to play a major part in the development of the continental shelf by acting as a mobile underwater workshop. In this mode, power sources for tools would be carried in the vessel. The crew would have suit heating facilities and every modern technological aid to their work.

Nine years of development work on DTV's by the present author had shown that the craft could best be operated by analogy

with aircraft. The similarities between the two types of machine were most marked.

The technology existed for the design and construction of high-performance DTV's. The only really doubtful factor at present was navigation. Equipment such as was used with dry submersibles was not generally applicable to wet submersibles. Everything in a wet vehicle, must, of course, be waterproof and either pressure-proof to the rated working depth or pressure equalized.

More development work was needed by the electronics engineers in the fields of underwater sonar navigation systems and diver communications systems with particular reference to the wet submersible.

The present author's prototype two-man side-by-side DTV, X2, had completed eighty-six sorties to date during which time methods of control had been experimented with in addition to power, propulsion and ballast systems The machine was now fully operational with a present speed of five knots and a range of five miles.

Other uses foreseen for DTV's were—sport, fish hunting, habitat support vehicles, fish farming, fish studies; and observation and photography of towed bodies.

Correspondence

Captain J. R. Pardoe, O.B.E., R.N. wrote that Commander Ranken had given an excellent view of the capabilities of submersibles for various tasks but Captain Pardoe had hoped that he or some potential user in the audience might have gone on to discuss user approach to the highly important matter of defining the operational requirement for a specific craft.

An example of just how easily this process could get out of hand could be seen in the American DSRV programme Not even its staunchest supporters, and he counted himself among them, would deny that it was expensive—the two craft at about 40 million dollars each, or thereabouts—why was it so expensive?

He suggested that it was largely because the task had been and was inevitably complex. Submarine rescue, like salvage, called for a problem solving operation where problems were legion and not easily foreseen. The more one added in operational requirements in terms such as: diving depth, manoeuvrability in currents, ability to heel, to fly away, to piggy-back on a mother submarine, to support large (24) numbers of services, to have good endurance, to navigate etc, the more inevitably one spiralled the costs.

All too often, in all walks of life, the operational requirement was not defined and confined closely enough. Could not this pitfall be avoided at the outset?

Author's Reply

In reply to Captain Pardoe, it was quite true that the paper did not in the end attempt to define the operational requirements for a specific craft to suit a particular or a number of potential users, and this was obviously absolutely essential if we were to develop the submersible into a viable and credible alternative to other underwater work systems. An attempt had been, however, made in Tables II and III to classify the various classes of underwater work and to reduce the likely operations to a number of common tasks and actions which needed to be executed underwater, together with the support functions required to make this possible. No one submersible could possibly have incorporated the capability to undertake all the tasks, nor the tools and attachments required to carry out all the actions; nor would it have been able to provide all the support functions. Any underwater work system must be chosen and designed around the particular range of tasks to be executed.

It was probably not too sweeping to say that, with the exception of a few specialized sea-bed machines, not one of the existing commercial submersibles had been designed to undertake a particular task or group of tasks, apart perhaps from observation, inspection and possibly search; most also had some facility for recovering objects from the sea-bed, e.g. torpedoes, missiles, or other relatively small items. Part of the reason was the high cost of any submersible system, as exemplified in such an aggravated form by the DSRV, perhaps the only project with a single mission. A consequence of the high system cost was the natural desire to make each submersible as versatile as possible, but not only did this lead to an often dubious and over-complicated compromise, it also raised the cost still further.

Underwater systems were only a means of achieving, or making possible specific work tasks; they were not an end in themselves. People usually talked loosely about diving, habitats, hyperbaric chambers and submersibles, not about specific tasks and how they could be done as cheaply, efficiently and safely as possible.

The major advantage of a submersible over any other system should be its mobility, and this had value for large-area searches in clear water, for pipeline and cable inspections, and for some scientific observations, e.g. certain fish behavioural patterns; with a look-out or mating hatch, it might also be useful for personnel transport or transfer, e.g. of divers to certain work sites, or of personnel to a one-atmosphere chamber. But there appeared to be very few actual work tasks where diver transport was necessary in the horizontal plane, and so far no system had been built requiring personnel transfer by this method, as opposed to a hyperbaric or one-atmosphere chamber lowered vertically.

However, with its short endurance and limited speed, as well as its vulnerability at launch and recovery, and almost equal need for surface support, the submersible had not so far achieved really decisive advantages over other systems for manipulative work on the continental shelf, especially within the depth capability of divers, and this latter was constantly being extended well beyond the maximum depth required for most existing commercial operations. Other solutions were already emerging for really deep-water work, e.g. the Lockheed wellhead cellar system for oil exploitation.

Where greater speeds or continuous working were needed, towed vehicles had advantages for certain types of scientific observations (fishing gear studies), and for cable and pipeline trenching or burying; unmanned towed instrument-carrying vehicles were commonplace for surveying, and the National Institute of Oceanography's deep-water high power sonar "Gloria" was perhaps the most elaborate so far.

Most of the other disadvantages of submersibles were discussed in the paper. Probably the most decisive break-through which could have been made would have been in the development of a cheap, compact and economical underwater power source of adequate output and long endurance. Much longer autonomy would have gone far to solving the vulnerability at the air/sea interface, and might have eliminated or at least reduced the most costly operational item, the surface-support vessel. Adequate power would have allowed powerful tools to have been provided, and higher speed would have been an advantage both for transiting to and from work sites and for some tasks requiring high-speed mobility.

Mr. Cooke had felt that justice had not been done to the wet submersible, either in the paper or in its presentation. This was simply because significant references to its value for commercial work had not been found during the Author's extensive review of the literature. "Pegasus" with its high-resolution cameras had greatly extended the value and reliability of under-water surveys, and it had been hoped at one time that the similar British vehicle built at the Admiralty Materials Laboratory would be developed into a commercial alternative. On the other hand, the "Total Sub 01" had not been found to fulfil any important need during the Zakum project in the Persian Gulf.

The wet submersible must normally be limited to shallow depths for decompression reasons, and generally speaking diver lock-out submersibles would have appeared to have been more appropriate beyond say 180 ft. It was quite true that the diver badly needed a work base with power sources and tools, but so far the untethered wet or dry vehicle did not seem to meet this need adequately, due to lack of power; hence the personnel transfer chamber or a surface-connected habitat seemed preferable.

In general the wet submersible did not yet seem to have had applications outside the military and recreational fields, and even here, as indicated by Mr. Cooke, there were still limitations on its use and safety, especially in turbid conditions.

What has been said here must imply little confidence in the value of present submersibles. This was inevitable at the present stage of their development, which had been compared to the Model T Ford of the 1920s and the Wright Brothers' aircraft. But some spectacular successes had been recorded, several of which would probably have been impossible by any other means. Further development would certainly increase the versatility of submersible systems, but the message of the paper and indeed of the whole symposium must surely have been that future development needed to be oriented towards the execution of underwater work tasks, not simply towards going underwater.

THE "PISCES" SUBMERSIBLE SYSTEM

G. G. Mott, B.Mech.E., C.Eng., M.R.I.N.A.*

This paper describes the experience of operating the submersible "Vickers Pisces" together with the support ship *Vickers Venturer*. Designed jointly by British and Canadian companies, "Pisces" is capable of operating at depths to 1100 metres.

To be a commercially viable proposition in the U.K. setting, the submersible is seen as part of an overall system capable of operating in sea conditions normally expected in the European continental shelf areas.

Some of the demonstration tasks are described and from these the types of task at which the Pisces/Venturer System might be successful can be deduced. Descriptions are given of the major studies carried out to improve the performance of "Pisces". Attention is also given to the development of an acceptable framework of safety for underwater vehicles and their operating procedures, arising from current experience. A proposal is made for formalized procedures of approving aspects for construction, testing and operating undersea vehicles in Europe.

It is concluded that the calculated risk taken in introducing the submersible system to this country has been worthwhile. The existence of a national policy on development of underwater engineering capability is questioned.

Mr. Mott

INTRODUCTION

In 1968, when seeking an avenue of direct engagement in commercially viable underwater activities, the author's company investigated the range of submersibles then operating and concluded that one particular small submersible showed the most promise of being developed for the European continental shelf sea areas.

Agreement was reached with the small Canadian company International Hydrodynamics Ltd., designers and builders of "Pisces I" submersible (600 metres), for the joint design and operation of a new "Pisces" class of submersible having a depth capability of 1100 metres. Two submersibles of this new class have been built; one is operating in the U.K. and is the subject of this paper, the other is being operated in Canadian waters by IHL.

The acquisition of "Pisces" was in the nature of a research and development exercise aimed at gaining first hand knowledge of the behaviour of systems, materials and components in a deep diving environment. It was also an important requirement that a serious attempt should be made to operate "Pisces" under commercial conditions.

Against the background of a large number of submersibles built in the U.S.A., not making a commercial living, this proposal did not appear to be very enterprising. However, after careful analysis the conclusion was reached that either most of the existent submersibles had not been designed to operate commercially, or that they were too expensive. Presumably this was because of the aerospace industry environment in which many of them were developed. Many were designed for specific tasks for which there was little or no commercial demand.

In the main it was recognized that, at least in the initial phase, the potential customers of a U.K. submersible system would be government departments, the needs of which would fall into two categories:

1) research bodies—seeking an improvement in the gathering of information concerning the underwater environment;
2) emergency—recovery of valuable objects from the sea bed.

It was considered that the more obvious commercial applications such as surveying pipelines, wellhead completions, bottom sampling etc., were less likely to be commercially rewarding until the country's demands for sea bed materials and minerals reached such a level that would necessitate work in water depths at which divers operating alone, or in conjunction with "wet" vehicles, were seriously handicapped. Such applications would necessitate considerable investment in highly specialized equipments for which the submersible would play only a supplementary role. The commercial groups holding rights to the minerals would no doubt determine the direction and pace of these developments. An attempt to exploit this market would only be feasible after the submersible proved to be a practical proposition in the environmental conditions prevailing in the intended operational areas. Recent events in the North Sea suggest that the need for these services might well be closer than was previously supposed.

It was also considered that the problems to be solved to make submersible operations on the European continental shelf viable would rank in the following order:

i) launching and recovery in average sea states;
ii) ability to locate key objects, the positions of which were precisely known;
iii) locating objects, the positions of which were unknown;
iv) accomplishment of particular tasks to the standard at least equal in cost and quality to that of a diver, taking into account tides and water turbidity;
v) stimulation of a supply of U.K. components suitable for the deep ocean environment.

"Pisces" was completed early in 1969 by a combined British/Canadian team and proving trials were undertaken in 800 metres of water to meet the certification requirements of the American Bureau of Shipping (the first full scale certification of such a submersible to these rules).

Subsequently "Pisces" was air transported to the U.K. and entered service in a training mode in June 1969. Since then it has completed 250 dives, totalling 600 hours of underwater operations in depths of water varying from 10 to 400 metres.

PROGRAMME OF OPERATIONS

During this period, a variety of tasks was undertaken and these are summarized in Table I. For obvious reasons, work undertaken for U.K. defence departments is omitted.

* Vickers Ltd., Shipbuilding Group

TABLE I—PROGRAMME OF UNDERWATER ACTIVITIES 1969–70

Tasks	Location	No. dives	Depths, ft	Remarks
Fresh water trials	Loch Ness	18	50–820	During these trials "Pisces" achieved a depth greater than any previously recorded depths in Loch Ness
Demonstration	Loch Ness	6	10–600	For news and TV media
Film work	Loch Ness	4	480–720	This was "Pisces" first commercial venture in U.K. and was undertaken on behalf of a U.K. film company
Demonstration	Loch Ness	20	120–765	These demonstrations were given to introduce "Pisces" to possible customers
Demonstration	Loch Linnhe	5	20–165	Various tasks were performed for MAFFS
Demonstrations and sea water trials	Gare Loch	13	105–165	Evaluation trials with MOD(N)
Variety of tasks and demonstrations	Oban, Sound of Mull, Loch Morven, Firth of Lorne, Loch Fynne	13	90–630	A series of dives under the auspices of NERC to allow "Pisces" to be assessed by various government depts., i.e. IGS, SMBA, NIO, MAFFS, DAFFS
News demonstration	Ramsay Bay	1	100	Undertaken for a north west newspaper
Demonstration	Portland Harbour	12	50–100	Undertaken by MOD(N) and SITV
Search and salvage	North Sea	3	130–205	Operations over wreck of rig *Constellation*
Aircraft wreck-search	Lyme Bay	3	60–100	
Fish research	Lyme Bay, Plymouth	3	100–160	A series of trials under the sponsorship of Mintech for Dept. Agriculture and Fish for Scotland, White Fish Authority, Board of Trade, Air Accident Investigation and Atomic Weapons Research Establishment.
Aircraft wreck-search	Tuskar Rock	6	260–280	
U/W light tests	Smerwich	2	96	
Filming and U/W burning etc.	Killany Bay	2	30	
Scampi investigations	Jura Sound	3	330–360	
Pipeline survey	Seascale	2	68–90	
A series of geological surveys and samples were taken during the course of these operations	Firth of Lorne	2	535–670	NERC charter during which the deepest dive by "Pisces" in U.K. waters was undertaken
	Barra Head	5	300–1300	
	Stanton Banks	1	270	
	Skerryvore	2	240	
	Canna Head	2	200–270	
	Rhum	2	55–300	
	Malog	2	370	
	Rona Deeps	1	1000	
	Hawes Bank	3	560	
	Isle of Muck	3	165–310	
Sampling and sea bottom investigation	Looe	5	180	NIO operations with "Pisces"
	Mevagissey	1	120	
	Eddystone	3	175–230	
TV link	Luce Bay	2	150	Continuation of MinTech operations

The depths of dives undertaken in European waters were well within the limits of "Pisces"; on many occasions they were even within that of free diving. There is a considerable appeal in the use of "Pisces" by qualified people who wish to observe the conditions for themselves, rather than obtain information secondhand from persons with limited observation capability. The merit of this facility has been emphasised on a number of occasions by salvage and scientific experts.

In the past there have been several examples, involving both civil and military aircraft, where considerable sums of money have been expended in recovering wreckage. Notable amongst these are the Comets of the 1950's, the Comet which disappeared off Crete in 1966 and, more recently, the Aer Lingus Viscount off Tuskar Rock.

The prime objective of such recoveries is to determine what went wrong, and in this context the most significant component to recover is the flight recorder. It is also important to analyse the geometrical position of all pieces of wreckage to ascertain if structural damage occurred before the actual crash.

Under the present methods of using trawlers to drag or grab for wreckage, the relative location of parts is lost; some parts are not recovered, others are seriously damaged.

It has been suggested that the submersible could be adapted to introduce a small TV camera into the flight deck or cockpit to scan the state of the instruments before a decision is made to move the wreckage. A further suggestion is that flight recorders should be fitted with "pingers" which automatically operate on immersion in water. Some progress is believed to have been made to include a pinger in prototypes being tested over the sea.

With this kind of application in mind, an exercise was carried out in association with the Ministry of Technology and the BBC for "Pisces" to locate aircraft wreckage and to relay the pictures by a surface retransmission unit to a control headquarters for live analysis by a group of experts (Fig. 1). An outline of this operation was demonstrated in the "Tomorrow's World" programme on BBC television.

During 1970 a military aircraft crashed in the general area in which the submersible was operating and the system was temporarily attached to the naval unit carrying out the search. Although not fully equipped for the role, "Pisces" made contact with the wreckage and recorded video tapes, which were subsequently replayed at the accident investigation.

Also, during 1970 a survey was undertaken of the wreck of the oil rig "Constellation" which was lost during a storm in the North Sea in October 1969, whilst under tow to Rotterdam. This particular operation was undertaken to determine the

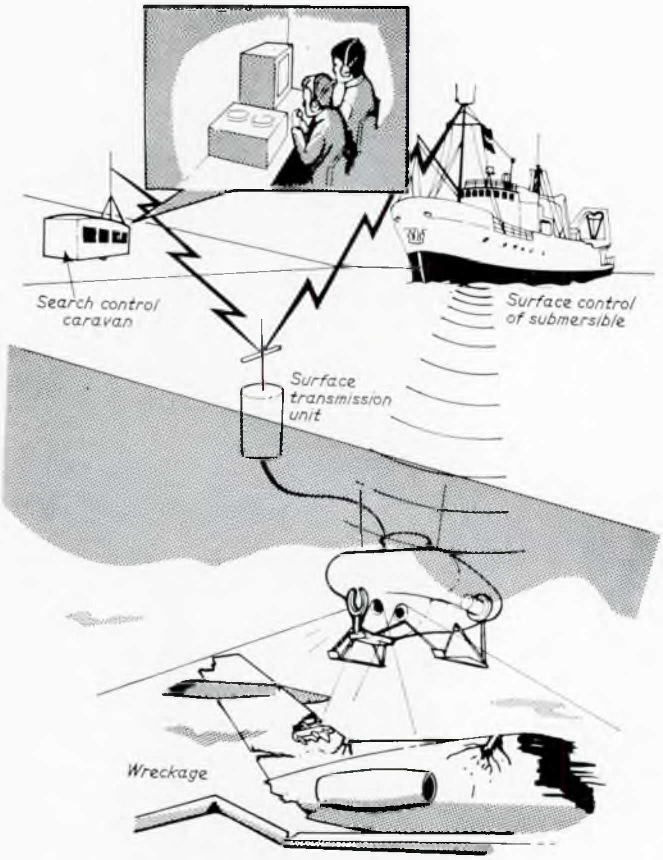


FIG. 1—*Transmission of underwater t.v. from sea bed*

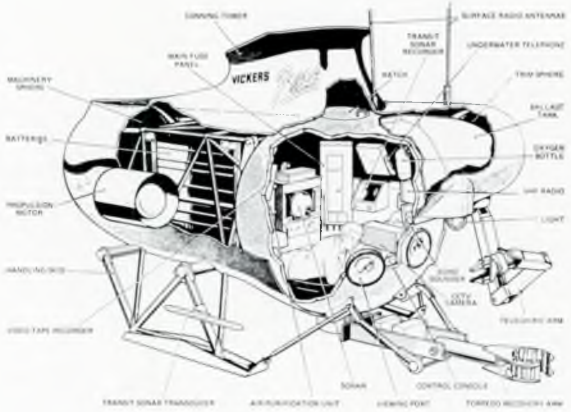


FIG. 2—*"Pisces II" submersible*

feasibility of salvage. The wreck and associated wreckage was located by the support ship *Venturer* using an excellent side scan sonar of British manufacture. This sonar unit can also be fitted to "Pisces" since the recorder is of a chemical type (the sparking type would cause atmospheric difficulties within the confines of the submersible hull).

Although this operation took place at the time of the equinoctial gales, four dives were made in sea states 3/4 and it was found that the rig was inverted. A record was made of parts of the wreck but, because of the severe tidal conditions and the hazardous state of the wreckage, an overall inspection was not considered possible until the weather conditions were more suitable. Whilst the exercise was not classed as a fully successful one, it did give valuable information on the future requirements for submersible vehicles suitable for working in such conditions.

"Pisces" is fitted with two grabs (Fig. 2): one is large and simple, capable of movement in one plane and designed for lifting cylindrical objects; the other is small and versatile with five degrees of freedom and capable of lifting 230 lb. Using the two grabs together, a wide variety of operations are feasible. It has been found possible to make *ad hoc* arrangements for gathering geological samples such as manganese modules and the taking of water samples and plankton at various depths using these grabs. The stage has now been reached where, having demonstrated a general capability, more effective use could be made of "Pisces" if earlier consideration could be given by potential customers to special tool fits appropriate to their purposes.

For one operation a small rock drill was fitted and satisfactory samples were obtained. One advantage of the submersible mounted drilling device is that samples can be taken in various planes at the discretion of the operator.

In each of the scientific dives one or two scientists have been included as observers. They carried out their own operations apart from piloting and navigating the submersible. Judging by the general tenor of comment, the availability of the free swimming submersible as a scientific tool has considerably enhanced the capabilities of the marine scientists in their task of gathering information concerning the continental shelf.

Some further experiments which have been carried out include the underwater firing of percussion studs to assess the feasibility of fitting patches to wrecked ships and possibly as a means of getting food and air to a wrecked submarine. The ignition and manipulation of an oxycutting torch was successfully tried, although in this case the gases were supplied from the surface. An ultrasonic examination of an underwater pipeline was also carried out.

These experiments, although rather crude, were aimed at evaluating the prospects of coping with the more demanding applications of underwater techniques from the confines of the submersible pressure hull. Enough information was gained to indicate that feasible solutions to these problems can be developed if the need is great enough. It should be noted that during 1969–70 another "Pisces" submersible carried out, single-handed, the salvage of *Emerald Straits*, a 30 metre tug, from a depth of 200 metres in Vancouver harbour.

LAUNCHING AND RECOVERY

It was considered that the greatest impact on the use of submersibles would be made by solving the problem of the launching and recovery of the submersible in sea states normally found in the European continental shelf area. The problem of handling submersibles at sea has not generally been considered as part of the overall operational system, a factor which it is believed contributes to the significant low usage of many American submersibles.

From the outset an answer was sought which would be commercially viable, the basis being a support ship small enough to be economical and yet large enough to be modified to carry the personnel associated with the submersible and customer scientists.

A former fisheries research vessel of 40 metres length was purchased and extensive alterations were made to it, including conversion of the fish hold into an additional accommodation block for 26 persons. Several concepts of the handling gear were considered, including travelling cranes, portal cranes, pantographs and derricks. Eventually the design preference settled on the present "A" frame design (Fig. 3).

The new requirements of the Board of Trade Freeboard Regulations applicable to small ships, together with an unexpected increase in the weight of the "Pisces" submersible (now 11 tons), imposed serious limitations on the overall weight of the handling gear. To save weight it has proved necessary to manufacture the "A" frame in aluminium and to exercise great care in the design of other components.

The concept of the handling gear is to use, as far as possible, the potential energy of the sea itself to assist in the recovery of the submersible; the winch being provided with an automatic

FIG. 3—*Stern of "Venturer" showing handling gear*

overhauling gear which keeps the hoist rope taut. An early decision was made to use a large nylon braided rope as a single part hoist with a soft eye at the end. This decision was made on two separate accounts:

a) to avoid the necessity for a heavy block to be handled by the skin diver in the water;
b) the nylon rope provides a means for absorbing the variable energy of the load without leading to uncontrollable situations.

In the event it proved necessary to use a shackle for connecting the hoist rope to the submersible.

Fig. 4 shows diagrammatically the current version of the handling gear which has been successfully used in the launching and recovery of the submersible in sea state 4. Conditions on some occasions were reported to be closer to sea state 5 but, as the estimates are somewhat suspect (tending to be based on the feelings of the individuals at the time in question) claims are being reserved until corroborative measurements are obtained about the true state of the sea during operations.

A pendant 'A' frame hangs from the main 'A' frame and this limits "Pisces" developing an uncontrollable athwartships oscillation whilst it is being lifted clear of the water. It is intended to fit a device on the base of the inverted 'A' to lock automatically with the "Pisces" hook, so that the load could be taken off the hoist rope prior to ramming the submersible on board. A suitable design has not yet been evolved but, in the interim, the requirement has been met by using a steel wire strop fitted by the skin diver.

An hydraulic system provides the power for lifting the load and ramming the 'A' frame in and out, energy being stored in accumulators to provide for peak periods during the operating cycle. The geometry of this system is so arranged that "Pisces" is landed on a seating welded to the deck on which it is secured for passage. The current apparatus is designed for a safe working load of 12 tons, for 1·5 m freeboard and an overhang of 6 m over the stern of the vessel.

As would be expected, technique plays a large part in the recovery system and many alternatives have been investigated. The current technique is as follows:

1) "Pisces" surfaces and turns into wind;
2) *Venturer* (stern to wind) takes position about 60 ft from "Pisces" and wire towing line is attached to eye on the stern of "Pisces";
3) two guide ropes are attached to "Pisces" and *Venturer* moves ahead at about two knots—to ensure manoeuvrability of herself and "Pisces";
4) "Pisces" is drawn in by a capstan until her hoist hook is 30 ft aft of *Venturer* and a skin diver attaches rope;
5) lifting sequences start, the skin diver taking up a suitable position on "Pisces".

By this method the whole system is stable and under control from *Venturer* despite the relative movements of herself and "Pisces". Attention is now being given to new developments aimed at enabling the pick up to be made without the need for a swimmer.

The concept of the "Pisces" system consisting of the submersible, the mothership and the handling gear is that it should give mobility and make available a submersible at reasonable commercial rates of hire. The present mother ship *Venturer*, whilst adequate for the purpose, has some notable faults which would be avoided in a new vessel. These are:

i) a limited capacity for deck loads in meeting MOT requirements for stability;
ii) a machinery arrangement consisting of high speed engines driving a reversible pitch screw which is both noisy and dangerous to the submersible and swimmers;
iii) unsatisfactory handling characteristics at low speed;
iv) main engines which are obsolescent with consequent maintenance problems.

From current experience it is considered that a 12 ton submersible requires a support ship approximately 180 ft long with electric main drive, for the European continental shelf operation. Transverse propulsion units would also help such a vessel to keep correct station during diving operations.

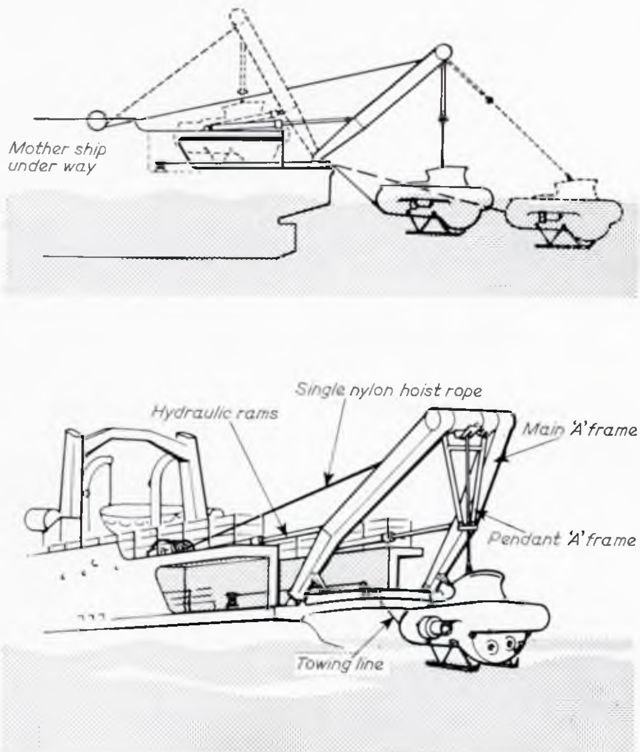


FIG. 4—*The procedure of recovery*

NAVIGATION AND SEARCH

Operations during 1970 have revealed the overall inadequacies of available equipments to meet the needs of underwater navigation and search. With a limited market for submersible services it has been a deliberate policy to seek low cost solutions to these problems rather than some of the more exotic systems said to be available—but for which we cannot find evidence of successful operation. The three main problems for which solutions have been sought are:

a) to know where the submersible is in relation to the mother ship at all times;
b) to enable the submersible to search a defined area with

the minimum possibility of missing targets within its range of visibility;

c) to locate objects, the position of which are known approximately.

"Pisces" is equipped with its own scanning sonar set which gives a completely satisfactory performance for the location of obstructions in its path and, to a certain extent, is helpful in locating targets within the search area. Operating at 155 kHz, it has various ranges up to 3000 ft. A directional hydrophone can be fitted for missions in which it has to search for lost objects fitted with a pinger.

A programme was undertaken to develop two navigational systems which became known as Submarine Position and Tracking Equipment (SPATE) and Pisces Interim Navigational System (PINS). Fig. 5 illustrates the systems and problems arising from interference.

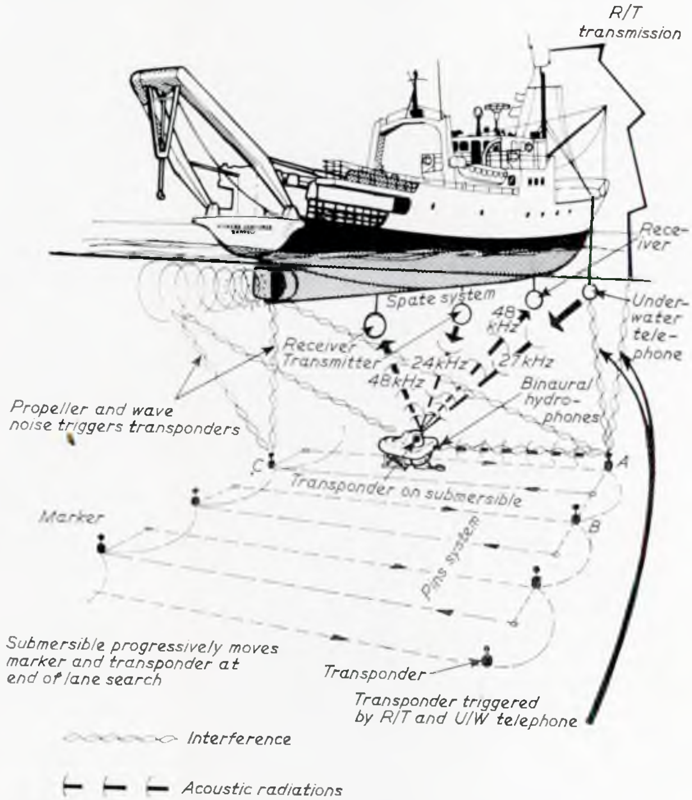


FIG. 5—*Showing SPATE and PINS systems and sources of interference*

SPATE relies upon three accurately positioned transducers carried in the support ship, where the centre transducer emits 24 kHz pulses at fixed time intervals. This produces an automatic 48 kHz reply from a transponder on the submersible. The reply is picked up by the forward and after transducers on the support ship and, from difference in time of receipt together with a knowledge of the submersible's depth (provided by submersible over UQC), bearings can be calculated. Slant range is provided as a function of the elapsed time between propagation and receipt. The initial concept was for the calculations to be done by slide rule and tables, and approximately seven minutes was required to determine the submersible's position relative to the support ship. If successful it was planned to link the system to a small computer with graphical output.

The SPATE system allows the support ship to know the whereabouts of the submersible relative to herself and "slave" her movements by instruction over the UQC. Bearing in mind that the ship herself is not a fixed platform, and that sound speed in water varies, SPATE is not an exact means of guiding the submersible in a bottom search.

PINS provides an accurate means of conducting an area square search using planted reference points on the sea bed. These take the form of two transponder buoys with sinker, each secured by a length of chain to a marker weight. They are dropped by the support ship at known co-ordinates. The markers and transponders are layed in such a manner that a line drawn between them is parallel to the tidal flow; thus reducing submersible drift error.

The submersible interrogates the transponder at 'A' (Fig. 5) and receives a return signal on binaural hydrophones. The crew read the results of the interrogation from a bearing meter and this, together with a gyro heading, enables the submersible to home on to the transponder. At the end of the leg, the submersible advances the transponder position by two lane spaces **B** in preparation for the next run on the heading toward transponder C.

In a purely visual search, where it is essential to ensure that the maximum possibility exists of covering the area, visibility determines lane width and a combination of visibility and system inaccuracy determines lane length. If the object of the search is expected to "stand proud" of the sea bed and provide a reflection on the scanning sonar, then lane length and width can be increased.

Whilst both SPATE and PINS systems were perfectly sound in principle and complementary in function (and on some occasions even worked), the transponders of both were subject to triggering by acoustics from the propeller of the mother ship, by communications with the submersible and by wave and bottom noises. This false triggering caused blanking out of sonar responses and early collapse of transponder batteries. In addition both systems have given problems which were primarily due to the failure of electronic components. Both systems have been shelved without determining whether a change of frequency or redesign of the electronics would improve matters.

Current and preferred thoughts may be summarized as follows:

i) For location of the submersible relative to the mother ship, the system consists of a sonar automatically scanning desired sectors of any width operating at 180 kHz. The submersible is fitted with a transponder operating at a frequency of 39 kHz which responds to the beam resulting in the submersible appearing on the tracking screen as an illuminated spot.

ii) The search system consists of transponders suspended above the sea bed on buoys which respond to the frequency of the sonar of the submersible. As a result, the transponders show up as an illuminated spot on the submersible's sonar screen. The submersible pilot can undertake search patterns by either maintaining equidistant circles around a single transponder or by steering in straight lines, keeping the transponder spot at equidistant 'X' distances on his sonar screen (suitably overlaid for his guidance).

PROPULSION

Experience during 1970 has indicated that whilst the propulsion system of "Pisces" is adequate for still water conditions, it is inadequate for the tidal conditions prevalent in certain areas of the North Sea. Accordingly a means was sought of improving the propulsion efficiency of the existing "Pisces" and with it consideration of the philosophy to be used in the design of a new submersible more suitable for operating in the North Sea. Propulsion efficiency emerged as two separate problems:

1) reduction of hull resistance;

2) improvements in the thrust of the propulsion units without increased power consumption.

Model tests using a 1/12 scale model (Fig. 6 (a)) conducted in a cavitation tunnel indicated the following make-up of the resistance:

	Resistance*
Naked hull	100
Undercarriage	140
Grab	55
Manipulator	48
	343

FIG. 6a—*Model in cavitation tunnel showing present hull configuration*

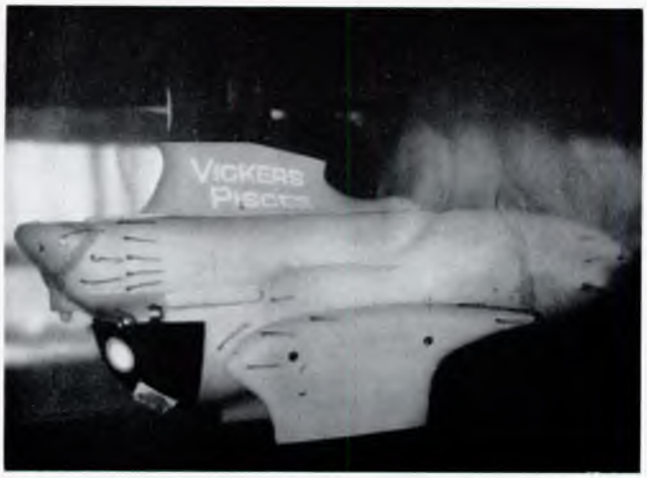


FIG. 6b—*Model in cavitation tunnel showing various improvements to hull streamlining*

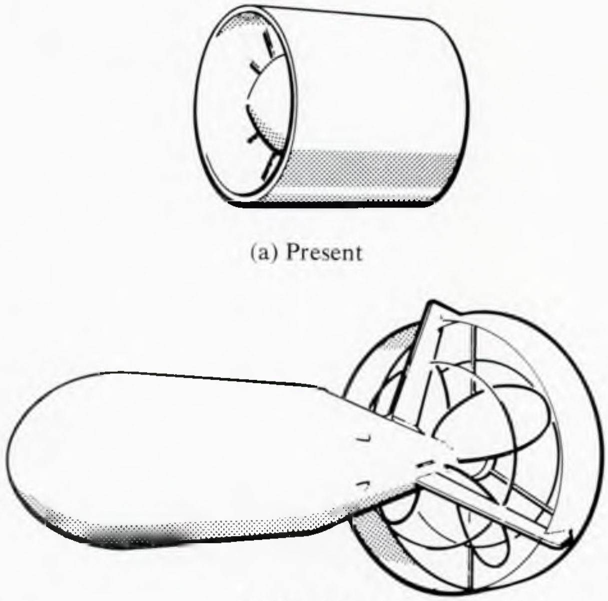
Various configurations of appendages were tried and that shown in Fig. 6(b) indicates an ultimate form of improved hull configuration giving a total resistance close to that of the naked hull in the previous table. The improvements which give rise to this result are:

	Resistance*
Tail fairing	40
Faired undercarriage enclosing motors and manipulators	110
Removal of grab required for special purposes only	60
Nose fairing	15
Removal of fin	20

* Based on resistance of naked hull = 100.

This series of tests indicated a line of action which could be taken to reduce the resistance of "Pisces", some aspects of which depend on the type of mission being undertaken at the time.

Parallel with this investigation a further set of experiments was undertaken to evaluate the efficiency of the ducted fan type of propulsion unit Fig. 7(a). Since "Pisces" has no rudder, steering is effected by the speed control of the port and starboard motors. The stepped control fitted to these motors proved unsatisfactory in maintaining "Pisces" on a particular heading. A new propulsion unit (Fig. 7(b)) was designed having an open



(a) Present

(b) Modified

FIG. 7—*Propulsion units*

high efficiency propeller driven by a series wound d.c. motor in which stepless speed control is achieved by using thyristor controllers.

Trials with these new propulsion units have shown that, without any further streamlining of the hull, speed has markedly improved. Pending a complete redesign of the fairings, which may not be done to "Pisces", an intermediate step of fitting a tail cone and fairing in the undercarriage will be undertaken in the near future. The former raises some difficulties of clearance during handling over the stern as "Pisces" is now faced stern forward under the present launching and recovery procedures. The fairing of the skids has further attractions in reducing the chance of entanglement by ropes or nets whilst on the bottom.

To meet the special conditions of the North Sea it was considered that it might be necessary to design a submersible with a low frontal area with the distinct disadvantages of having the pilot and observer in a prone position, but propulsion trials indicate that a design is possible based on "Pisces". Providing attention is given to high efficiency propulsion pods and to the streamlining of all apendages, a submerged speed of four knots is possible with propulsion units not exceeding 12 shp. A design based on this is shown in Fig. 8.

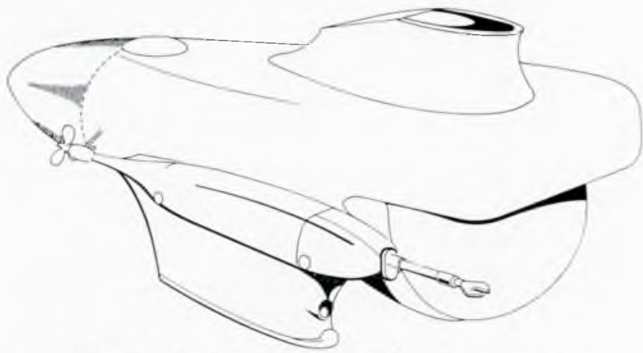


FIG. 8—*Design concept for North Sea service*

SAFETY OF OPERATIONS

Apart from the technical aspects described previously, considerable progress has been made towards formulating a philosophy on the safety of underwater operations with submersibles. As with many aspects of life in the technological age, safety is never absolute and one is left with the problem of defining what is an acceptable risk and what is not.

In this context the matter has been actively pursued in the U.S.A. where the coast guards have taken a lead in sponsoring attempts to get underwater operators to join in preparing arrangements for mutual assistance in cases of emergency. In addition, the American Bureau of Shipping has instituted a special committee on submersibles which meets every six months to review technical aspects of safety.

The "Pisces" submersibles have had a good record of safety. Nevertheless, in Canadian waters an exercise has been carried out to demonstrate the practicability of one submersible rescuing another from a depth of 600 m.

The low level of submersible activity in Europe at present deprives an operator of any potential source of help should his submersible get into trouble during a diving operation. There is no second submersible in the U.K. to call upon and the only form of rescue feasible, should "Pisces" fail to surface from below 300 ft, is to drag the bottom.

It is therefore vital to have a means of continuously tracking the submersible from the surface. This is done by having a transponder which is triggered from the surface and indicates the position of the submersible by a definte spot on the PPI of the sonar tracking system.

The broad requirements for the safety of a submersible are:

i) avoidance of projections which can become ensnared in nets or ropes which are too heavy for the submersible to lift. If such projections are necessary they must be ejectable;
ii) ability to discharge sufficient weight to ensure an ability to surface under any damaged conditions in which the occupants remain alive;
iii) independent means of surfacing which does not depend on the electrical power from the main battery;
iv) a life support system to enable the personnel to survive for a sufficient period for rescue facilities to be effectively mobilized;
v) a capability to escape from the submersible in water shallow enough so to do;
vi) an ability for the passenger to surface the submersible in the event of incapacity of the pilot.

The risks to, and the standards of, safety of the submersible should be judged against those which are acceptable in normal diving operations. Apart from the lack of an alternative vehicle of equal performance (and this we expect is a transitory phase in the U.K.) the risks are considerably less for the "Pisces" submersibles. An alternative comparison could be made with light aircraft capable of carrying one or two passengers where a similar sort of risk is involved:

a) mechanical failure destroying the ability to maintain height;
b) no external means of assisting in safe descent;
c) no external means of safely abandoning the machine;
d) failure of communications which would prevent warning of hazards;
e) sudden indisposition of the pilot.

Does anyone seriously suggest abandoning flying because of these risks?

A device is being sought with a minimal weight in water for providing supplementary buoyancy by chemical means in an emergency, acting like a balloon in the atmosphere. There is a possible chemical which seems worthy of closer investigation for the development of a rescue pack of this type. If successful, it could also be used for other applications such as a "strap on lifter" for raising weights from the sea bottom.

CERTIFICATION AND APPROVAL

The introduction of the "Pisces" submersible to this country and its immediate use in commercial charters has stimulated interest in the legal issues of underwater operations. It had been anticipated that a form of continuous certification of the safety aspects of the submersible itself would be required and an application was made to the American Bureau of Shipping for their certification. "Pisces" was the first submersible to be built and tested to the certification of ABS.

It is interesting to recount that prior applications to the Board of Trade and Lloyd's Register revealed an interest in the special requirements of submersibles, but a total unpreparedness for certification. Clearly, with the prospect of a number of submersibles, undersea habitats, sea bed vehicles and sea sledges operating in the European continental shelf area in the next few years, there is an urgent need for the setting of acceptable standards of engineering applicable to them and a means of receiving official approval of the overall parameters of design.

An analogy can be drawn by comparison with the surface ships where Lloyd's Register sets down the standards of engineering covering materials, scantlings and proving tests; whereas the Board of Trade rules govern such things as free board, prevention of fire at sea, bulkhead sub-division, damage stability, etc.—parameters of basic design which naval architects are required to observe. Likewise, builders of submersibles should be expected to work to rules governing the integrity of the pressure hulls, penetrations, windows, electrics, hydraulics, as well as observing good practice in respect of the provisions for life support systems, reserves of buoyancy and means of coping with emergencies, etc.

As indicated in the early part of this paper, most of the work undertaken by "Pisces" has been with one government sponsored organization or another. This has stimulated the submarine branch of the Royal Navy to take an interest in the operational development of commercial submarining in this country, at least in the context of the use of such vessels by government departments. The Royal Navy has taken the view that the Submarine Service is the only professional submarine body in the country able to advise on the standards of operational procedure and competence, and has accordingly sought official appointment in that role.

A near parallel can be drawn with the situation in the U.S.A. where the U.S. Navy has its own submersible capability and issues special rules for the qualification of submersibles and operating crews for the carrying of U.S. Naval Personnel.

Until now the employment of submersible vehicles in the U.K. in a private capacity has been in the nature of wishful thinking, but it is now apparent that the U.K.'s industrial survival may depend increasingly on them for the exploitation of the resources of sea areas adjacent to it. Accordingly, the emergence of an authoritive control body is to be encouraged; whether or not it should be a supplementary activity of the Naval Submarine Service is open to some doubt for the following reasons:

1) submersibles operate either on the surface or on the bottom, almost never in mid-water in depths greater than the collapsed depth which is normal for submarines;
2) submersibles operate as part of the surface ship system, and therefore the philosophy of operation is entirely different;
3) submariner's tend to be technical or navigational specialists and rarely achieve an all round engineering and operating capability. (In the present state of the art, the submersible pilot must be both a competent operator and maintainer of his vehicle and equipment.)
4) the scale of the operational requirements of submersibles is so different to that of military submarines that there is a danger of requirements being set from an entirely incorrect standpoint.

Despite the reasons for doubt, there is obviously a significant degree of commonality with submarine operations and one hopes that a common point of view can be reached which will enable the submersible system to be developed as an industrial tool without having to meet crippling and unrealistic requirements.

This matter is of vital interest to all bodies in the U.K. with ideas of utilizing submersible vehicles in extending their present capabilities as well as to those who have direct interest in supplying the future market needs for equipment. As is underlined in the U.S.A. experience, submersible services will not prosper if the construction of equipment is undertaken by, or is forced into, an unnecessarily expensive environment.

CONCLUSIONS

It is considered that the calculated risk in introducing a submersible system to this country as a commercial operation has been successful. The encouragement which has been forthcoming from various marine authorities, which see in the "Pisces" system a means of augmenting their respective tasks, augurs well for its future development.

A submersible system cannot in any way be considered a cheap facility and commercial development should therefore be cautious. More constructive progress will be made by short steps, building progressively on existing knowledge and expertise; learning from experience what is commercially acceptable and what is not, rather than by attempting dramatic projects which have doubtful commercial value.

The U.K. with its depleted land resources may more and more be dependent on the successful exploitation of new resources found within its immediate subsea areas. Continental countries such as France and Germany appear to have recognized the importance of this source of new materials and are taking steps on a national basis to ensure that their interests in it are not dependent on equipment supplied from the U.S.A. It is still questionable whether we in this country have yet formulated a clear policy of development for serious work on the sea bed.

Discussion

MR. K. R. HAIGH said that one of the main "raisons d'être" of the manned submersible was the ability to transport experienced observers to an underwater site because either the observer was untrained in diving techniques or the site was beyond diver reach. The manned submersible had enabled scientists and engineers to observe first hand in real time with the benefit of three dimensional viewing backed by the human mind. Before this time the tools required had been cameras, photographic and television, carried by divers or unmanned vehicles. Therefore he could not understand the philosophy behind the development of the transmission of underwater television pictures from the sea bed to the shore involving a towed radio buoy; the only purpose he could think of was for mass media presentation. The system did little to advance the technique established at the time of the "Affray" disaster in 1951. Many unmanned vehicles could carry out this task at a fraction of the cost of the "Pisces" system. He was completely ignorant of the economics of the system but it could not be cost effective. The network obviously needed time to establish and collect the party of experts ashore, during which time one supreme expert could have been riding the vehicle for a first hand view and, if not sure of his own convictions, consecutively taking video and ciné film of much superior quality. What were the advantages claimed for this new system?

He fully supported the author on his views on safety and certification. It was generally accepted in the U.S.A. that a diver made a better submersible pilot than a military submariner. A diver had a much better feel for the environment than a person who was trained to avoid the sea bed and who never saw where he was going, once below the surface. Two operational aspects of "Pisces" frightened him a little—the towed radio buoy and the marker buoy sometimes used—both of which could become snagged. He had been once involved in an escapade where they had been trapped on the bottom by snagging a buoyant line. Also, "Pisces" did not have the facility to release the personnel capsule in an emergency, as called for in the MTS Safety Guidelines. This seemed to govern Vickers' philosophy of having a second boat for emergency purposes.

CAPTAIN J. R. PARDOE, O.B.E., R.N. said that, as he was from the Navy, he thought that it might be of some interest to some of those present to know what the Navy was thinking concerning what might be done in this very important area regarding certification and standardization of practices. That morning they had seen the slide of a nuclear submarine. These has cost Mr. Mott a lot of blood and toil to build, and then a lot of money to buy; resources in manpower and money were both overstretched in meeting the military requirements the Navy had in running such big submarines. He could not see them being able to afford the people they needed for the small submersible task—he wished they could. It was a task with which some of them could cope, e.g. those who had run small submarines after the war, craft about the same size as "Pobble". For future development he was confident that he would hear from Commander Messervy that you needed the diver and the submariner. The latter was the fellow who understood the problems involved in shifting an inanimate object through the water, because he has been doing just that for many years.

He explained that the Navy would like to help. If anyone there could persuade the M.O.D. to let them have the cash and the necessary extra manpower, they would be most grateful.

MR. G. S. HENSON, in following up the point about certification, said that it was important that all underwater legislation and certification of underwater craft was considered as a single problem. This was becoming critically important commercially and was neglected at present in Britain. Many other countries had moved to give their own nationals an advantage in exploiting their underwater wealth. As a minimum, it would be necessary to move on three fronts. Firstly classification, certification standards and permission to work underwater could have been laid down quite easily on a north European basis. There was not much competition between submersibles and the difficulties would be minimal. If it were left one more year it would be difficult, and in two years it would be impossible.

At the same time there should be standard communication systems, so that everyone working on the surface and underwater could talk together, especially in an emergency for rescue purposes, and could work together as a single unit.

Finally a rescue operation plan should have been produced on a north European basis, where people would be working increasingly under water in conditions much more difficult than in the Mediterranean or elsewhere. Unfortunately there was a shortage of money as had been said. Even more important at this time; there was a shortage of a lead.

LT. CDR. C. A. HELY-HUTCHINSON R.N., A.M.I.Mar.E. said that of all the papers that day, Mr. Mott's was the only one from a commercial company which had decided to develop a submersible system for purely commercial reasons, and it was a pity that so little had been said about the commercial side of the operation. He was quite sure that Mr. Mott and his men had been working to a rigidly controlled budget. How close was "Pisces" coming to recovering at least its variable costs? When would they expect to show a reasonable annual profit on the operation? Of the types of work they had experimented with, which offered the best market in the short term?

MR. K. C. HALES, A.M.I.Mar.E. said that having been challenged on the position of Lloyd's Register, with regard to the classification of submersibles, he was in a position to answer officially. Two years ago they had been asked to try to find a method of classifying these craft, and after a year they had produced a provisional draft of some rules and guidance notes. These had been vetted within the industry in this country, including the company of the author of this paper, and Lloyd's had prepared a draft ready to be presented to their technical committee.

They were now in a position to classify all types of submersibles and diving systems. They had gone further than the actual vehicle itself and had examined the requirements of the life support systems, etc., so that they were able to cover the whole conception in toto, under the title of Classification.

When one came to describe what a submersible system was, this did create a certain number of problems. In doing the work, they had been asked to look beyond the description given by the manufacturers and the users, and they had had to look at the type of access and even the question of deck decompression and recompression chambers, so that they could cover the whole field of the submersibles today.

COMMANDER P. J. MESSERVY, in answer to Mr. Haigh's criticisms as to why a real time television system was required, said that he tended to agree with him. This particular system had been forced on them by engineers and technicians who could not —or would not—dive in a submersible and had wanted immediate information on what had been seen. This had been sponsored by the Government.

Secondly, Mr. Haigh had asked why they towed a bouy. They did it at depths down to 600 ft because they were very much "belt and braces" when running the submersible. For "Pisces" they had an operational procedure which more or less had been forced on them in the beginning by their own ad hoc methods and F.O.S.M./S intervention had been gratefully received. Towing the buoy was not difficult. They did not particularly want to tow a radio buoy, and below 200 ft they also relied on the SPATE II system.

As a simple sailor, he preferred manual seamanship to show where the submersible was, bearing in mind that it was never more than 1500 ft from the support vessel, which was why they did not object to the buoy.

Mr. Haigh's third point had been on capsule release. This was purely a matter of cost; they would have very much liked to have had an "all-singing all-taking" submersible doing everything itself. There were basic reasons why they could not. They had a submarine service. They dived and underwent the same risks without being able to release their main submarine if they got stuck on the bottom. They had a chance to get away with it, just as he himself did.

He endorsed what Captain Pardoe had said; they had asked for help and were most grateful for F.O.S.M/S work on their behalf. He also endorsed his statement that a submariner should be used for this.

Correspondence

CDR. M. B. F. RANKEN, M.I.Mar.E. wrote that in his presentation of the paper the author had mentioned the purchase of the fish factory vessel *Fairtry II*. He had explained the better characteristics of this larger vessel in a sea way from the point of view of launching and recovering submersibles. However, he had given a figure of £500 000 as the operating cost, and it would have been interesting to have had a breakdown of this figure, since it seemed extremely high for the vessel alone, especially as she would presumably have had a much smaller crew than during her service as a fish factory vessel.

MR. J. B. WILSON AND MR. N. C. FLEMMING wrote that the increased manoeuvrability given by the new Vickers propulsion motors and the new SPATE II navigation system had enabled a detailed survey of the Eddystone shell gravels and of a series of sand ribbon S.E. of the Scillies to be made. Fixes of the submersible's position taken at frequent intervals on *Vickers Venturer* had been plotted in the ship and converted into X Y co-ordinates for transmission to "Pisces" where the observer could plot his position and track easily and accurately on a corresponding X Y grid.

The use of "Pisces" by marine geologists had enabled data and samples to be collected in a way hitherto hardly possible offshore even with the use of experienced divers and certainly the short timescale involved using "Pisces" could not even remotely be matched by any other method.

Author's Reply

In reply to Mr. Haigh, about capsule release, it would have been desirable to have had such a facility, but at what penalty? Did one have it in aircraft? The answer was "no". The technical decisions had to be made based on the risks and the costs of trying to reduce them.

They were very grateful for the assistance they had received from F/O Submarines and his staff but they were somewhat surprised that they had not had many enquiries about this operation from ex-submariners. They were building up their operating staff now and so felt that there was a chance that there would be more interest from that sector. He agreed with Captain Pardoe on cash support from the government agencies for commercial submarine activities so that in this country a presence in that international field could be maintained. Even for a large firm it was difficult to get the sums of money which were required.

As he did not wish to go deeply into the questions of Mr. Hely-Hutchinson and Commander Ranken on cost breakdowns, because they affected commercial strategy, he thought it suffice to say that it took some £250 000 per annum to keep the *Venturer*/"Pisces" unit on station. They had originally thought that they could economize by standing down the ship's crew when they were not required, but it did not work, as one needed a completely trained crew available all the time. On the new deep sea unit they estimated that operating costs would be of the order of £0·5m so they needed to ensure that there was work to justify it. They thought that it was viable with certain activities in which they were engaged. The *Venturer*/"Pisces" unit was engaged almost entirely on military support work, and it was economically viable.

Mr. Wilson and Mr. Flemming made an interesting point in the use of submersibles which confirmed the view that they could be very productive if the user took the trouble to study the potential capabilities and then plan well beforehand how he intended to use them. The submersible operator's main task was to provide the capability and they generally expected the user to have a deeper knowledge of the performance of instruments peculiar to his particular task.

Mr. Henson had spoken about legislation. A word of warning was in order; it was easy to make rules, but one must ensure that commercial viability was not destroyed. There always seemed to them to be two distinct versions of safety rules, one for aircraft and one for marine. As an example, for aircraft carrying passengers the rules about fire protection were minimal whereas for carrying passengers at sea, ship economy was penalized very heavily in this respect. In the marine world they thought that the rules were designed specifically to make ships non competitive with aircraft. There was some danger that the same thing could apply to submersibles and they must therefore ensure that legislation for underwater activity was so designed as to help rather than hinder development, whilst at the same time accepting that control was needed.

REPORT OF EXPERIENCES WITH THE UNDERWATER LABORATORY "HELGOLAND"

G. Haux*

Mr. Haux

There can be no doubt that scientific, economic and military considerations necessitate increased activities and long-term underwater operations for exploitation of the ocean. One possibility of solving this task is the use of so-called underwater laboratories. These habitats enable one or several persons to stay underwater for periods of weeks or even months. The first underwater laboratories, "Man in Sea" and "Precontinent 1", began to operate in 1962, in the Mediterranean. About fifty different habitats have been tried so far and the depth record for saturation diving with underwater laboratories is 158 metres.

Most of these enterprises, however, were undertaken in clear and warm waters with low currents and favourable weather conditions on the surface. The initial operation of the underwater laboratory "Helgoland" in July 1969, however, was performed under more disagreeable conditions. No depth-records had been set up and, due to the prevailing conditions, personnel as well as material involved were subjected to utmost stress. At the site of the operation of the UWL "Helgoland", water temperatures hardly rise above 14°C during summer and the visibility is often less than two metres. Even storms are liable to occur in this season and the currents are often unpredictable.

All equipment had to be adapted because of these unfavourable environmental conditions. A particular difficulty was the long distance to the shore station and for the very first time, power supply had to be taken over by an unmanned supply buoy.

INTRODUCTION

Investigation into the ecological conditions of the ocean is at present one of the most important scientific tasks. The major concern of marine biological research is the highly complex system of relations between environment and organism. We are therefore trying to advance and increase our knowledge about synthesis, transformation and degradation processes in the ocean.

These great scientific tasks however, can only be solved if new technical methods are developed. Until now, ocean research has been primarily performed from the surface but the method of scuba-diving is now gaining more and more importance in the field of marine biological research and has finally led to the designing, construction and operation of underwater laboratories all over the world.

For a number of years scientist-divers of the Biologische Anstalt Helgoland have dived in the rough waters of the North Sea, where surge and low water temperatures, as well as poor visibility conditions, present difficulties for them. Nevertheless, these diving operations have yielded some very interesting results. Based on the experiences of these divers, German marine biologists were encouraged to start on a very challenging project: the Unterwasserlaboratorium "Helgoland". This underwater laboratory was designed in co-operation with the Institut für Flugmedizin of the Deutsche Forschungs- und Versuchsanstalt für Luft- und Raumfahrt, and was intended for operation near the coast of Helgoland. After the required funds of one million DM had been provided by the Federal Ministry for Scientific Research, and after two years of preliminary planning, the project was finally commissioned to Drägerwerk Lübeck, in November 1968.

In August 1968, the Biologische Anstalt Helgoland had already been provided with a small underwater test station which was constructed by another German company. This was a steel container of cylindrical shape, six metres long, two metres in diameter, standing on four straddled legs. This comparatively simple underwater station was submerged in the Baltic Sea in the autumn of 1968, to a water depth of ten metres. A crew of two divers had been working in the station for a period of eleven days when it was realized that this underwater venture had a severe handicap: the surface power support from a supply vessel. The position of this vessel had to be changed due to weather and wind conditions, thereby interrupting the energy supply and telephone communication to the underwater station for several hours. It became evident that an underwater station operating in the waters of the northern areas would have to be equipped with a maximum of self-contained life support, to render the station independent from surface supply. This experience had already been taken into consideration when the Unterwasserlaboratorium "Helgoland" was under design and construction in Lübeck.

TECHNICAL DESCRIPTION

This underwater laboratory is a cylindrical pressure container of 9 metres in length and 2·5 metres in diameter. The pressurization capacity of the underwater complex is designed for an equivalent water depth of 100 metres, when it is fully closed. The total volume amounts to approximately 43 m^3 and the total weight, including the ballast, amounts to approximately 75 tons. The laboratory consists of two separate compartments, one wet room with manhole for escape and one work room. The ends of the laboratory are equipped with two bunks each. The work room, which can be entered through a door, is also intended for use as a decompression chamber so that the aquanauts can be decompressed during their stay. For this purpose the room can stand an additional outer positive pressure of 10 kp/cm^2. An electric switchboard plant with measuring instruments, switches

* Chief Designer and Constructor, Department Diving Technique, Drägerwerk AG Lübeck.

and indicator devices for control of the entire electrical installations and a life-support-system have been installed in the front of this chamber.

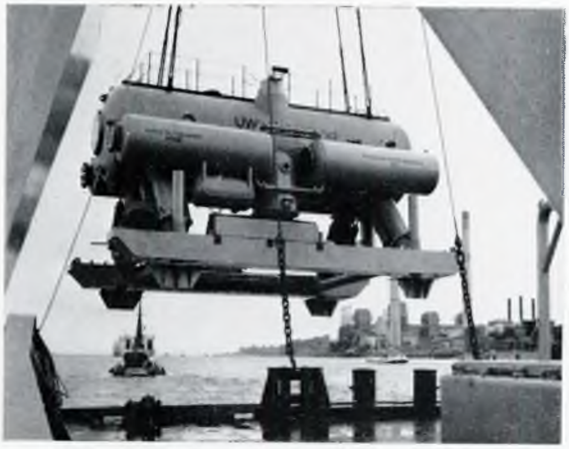


FIG. 1—*Underwater laboratory "Helgoland" being transported to Helgoland by way of a Magnus-Crane*

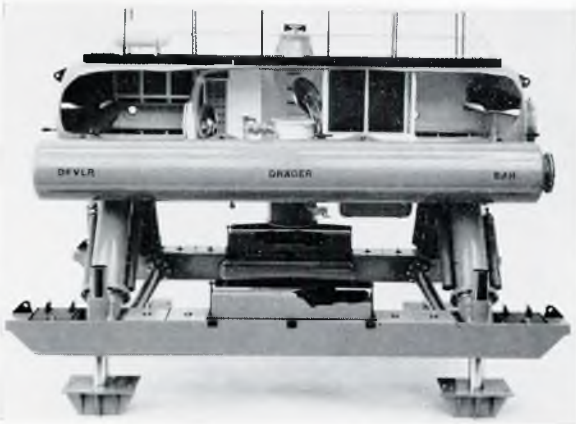


FIG. 2—*Interior spaces of the UWL "Helgoland"*

The atmosphere within the underwater laboratory is continuously regenerated in the closed-circuit system; exhaled carbon dioxide is absorbed by soda-lime, the exhausted oxygen is constantly replenished and impurities are retained by charcoal filters. The oxygen partial pressure amounted to 0·3 to 0·4 ata.

The manhole of the complex on the lower part of the vessel has a length of about 2·5 metres in order that the water level, which is affected by the tides, can be compensated for without any loss of gas. It can be closed to be pressuretight and can be opened hydraulically from inside as well as from outside. Behind the two doors there is a hot water shower, and next to it the WC. The excrements are led through a removing device into an excrement tank. The kitchen is on the opposite side and is equipped with a deep freezer, refrigerator, an infra-red heated stove and a sink. The atmosphere within the laboratory is controlled by numerous metering units. CO_2, O_2, relative atmospheric humidity, pressure and temperature values are constantly measured and recorded. An additional warning device controls the atmospheric CO and CO_2 values, and the water level in the escape tower.

Opposite the manhole is a control panel which is equipped with all the necessary valves and, with regard to ergonomic aspects, the switchboard has been divided into four parts. According to the available gases (air, nitrogen, oxygen and helium) the manometers for supply pressure and working pressure, the control valves, the distributing valves and the pressure reducers were combined to form clearly arranged blocks. This switchboard is equipped with connectors to supply divers by way of a hose. It can also be used as filling station for compressed air cylinders.

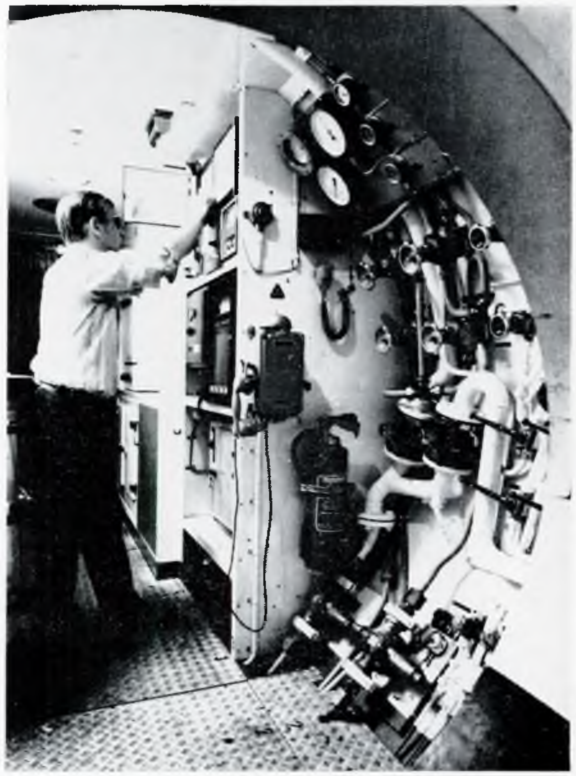


FIG. 3—*Gas switch board and control panel*

The surroundings of the underwater laboratory can be watched through seven small windows. A technical particular of the UWL "Helgoland" is the submerging system. The construction enables submerging and surfacing without assistance from outside. For this procedure the entire interior room has to be pressurized up to the pressure value which is expected at the point of use. Then two of the four straddled legs have to be flooded, until the UWL begins to sink. A negative buoyancy of a few kp is sufficient for this manoeuvre. When the laboratory reaches the seabed, the touching of the ground is softened by an arch-shaped anchor chain fastened to the ballast tank. After checking that the laboratory has reached the expected point, the trimming tanks in the straddled legs are completely flooded and the lateral ballast tanks are filled with water so that a total negative buoyancy of 16 tons is obtained.

Together with the solid ballast within the lower ballast boxes, which contain iron concrete blocks of 1·2 tons each, the centre of gravity has been placed so low that the whole system functions like a cork tumbler. Unevenness of the seabed is balanced by means of telescopic legs, which can be lowered mechanically, so that the laboratory is always kept in a horizontal position. The lower stand surfaces are additionally equipped with long vertical blades to penetrate into the ground and avoid transpositioning of the laboratory. The final effect of all these measures is a safe position for the UWL "Helgoland", even under the most unfavourable conditions.

SELF-CONTAINED SUPPORT BUOY

It is well known that the supply of an UWL with energy and gas is a particularly difficult problem. Nearly all former enterprises have been performed either with a supply line from the land or with a supply vessel or pontoon anchored above the laboratory. In case of the UWL "Helgoland" the first method could not be practised because of a three kilometre distance to the

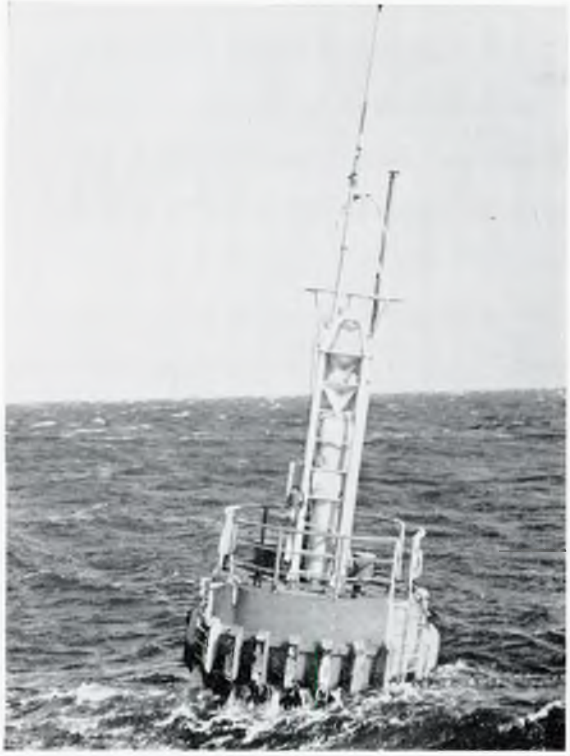


FIG. 4—*Supply buoy*

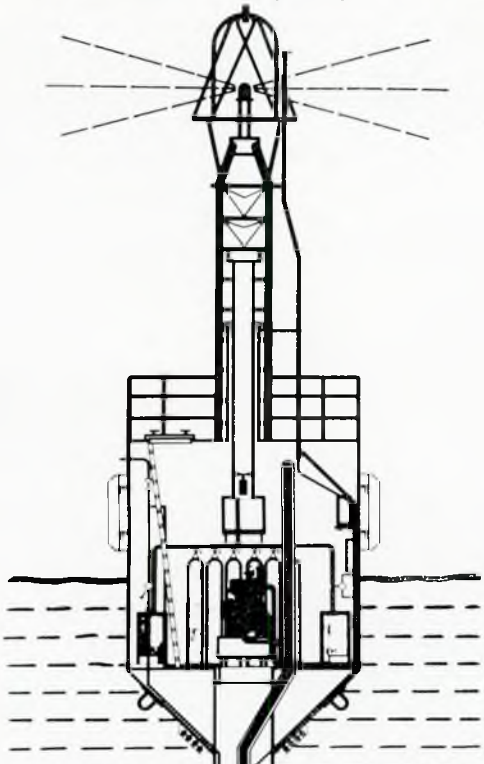


FIG. 5—*Cross-section of the simple buoy for UWL "Helgoland"*

land, making it too expensive. The second method also had to be excluded because of the stormy weather conditions which must be expected in this area, even during summer. It was therefore necessary to find a new solution, which led to the very first employment of a self-contained support buoy, which is moored adjacent to the laboratory by three tetrapod anchors with a weight of 16 tons each. The buoy comprises a Diesel aggregate with a capacity of 25 kVA, two high-pressure compressors, oil tanks and gas cylinders for storage of oxygen, nitrogen and helium.

The buoy carries a radio link and provides for radio and television communication to the central shore station. The actual supply of power, compressed air, nitrogen, oxygen, helium and fresh water to the laboratory is made by umbilical cords. The entire aggregate has been so dimensioned that, without attendance, a supply is guaranteed for at least 14 days. All aggregates and particularly the Diesel aggregate and the compressors work satisfactorily, even if the supply buoy is in an inclined position of 45 degrees. This is achieved by installation of special oil pan lubricating systems.

SAFETY DEVICES

Safety of the divers and particularly of the aquanauts is one of the most important principles for performance of operations. Therefore all preparations for UWL "Helgoland" took into account these safety problems and the lives of the aquanauts are safeguarded by numerous rescue means. A one-man rescue chamber is immediately attached to the decompression chamber of the laboratory. This enables a diver to be rapidly conveyed to the surface in case of emergency without entering the water. The rescue chamber can be recovered by helicopter or boat and will be flanged to a shore-mounted treatment-decompression chamber. Another diving bell is to act as an elevator which will convey the divers to and from the underwater laboratory, especially in case of bad weather and in great operating depths. For weight reasons a very compact design of this lift was needed and it therefore has a maximum capacity of only three persons.

The laboratory is also equipped with a life raft which can be released after the crew have been decompressed. This life raft however should be reserved for those cases of emergency when no help can be expected from the surface. A further and highly

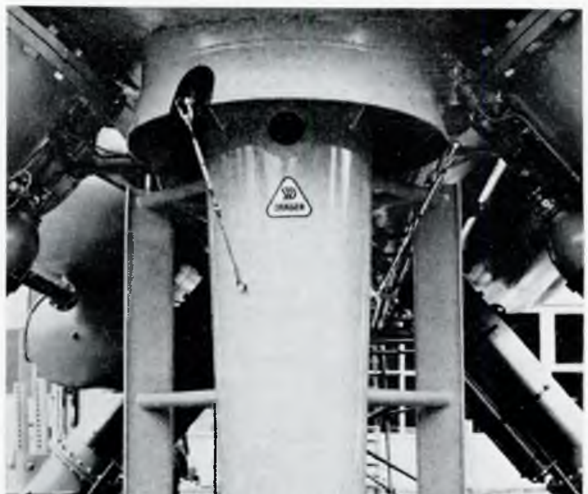


FIG. 6—*One-man rescue chamber at UWL "Helgoland"*

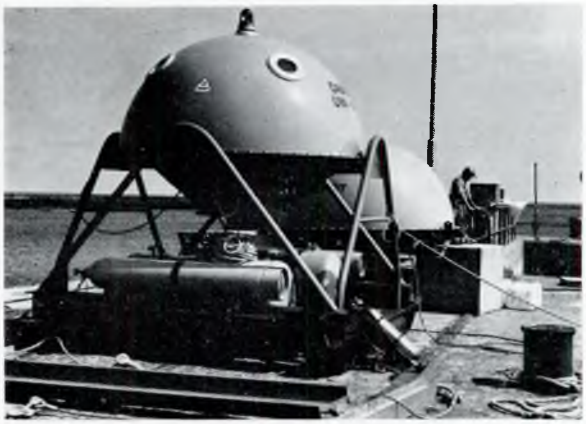


FIG. 7—*Underwater igloo and depot prior to operation in the North Sea*

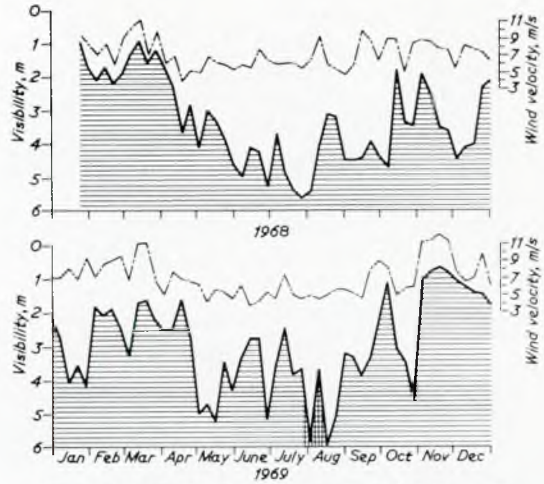


FIG. 8—*Wind velocities and visibility scale*

esteemed safety device for the aquanauts is the underwater igloo which has been projected and designed by Draegerwerk. This "little brother" of the UWL with a diameter of 2 m was equipped with a complete life-support-system, with a reserve apparatus and with a ballast and trimming system. It was intended for short-term stays of two divers and should serve on a distant working place as a shelter and as a safe place of refuge. Prior to initial operation all the equipment and interior fittings were thoroughly scrutinized by Draegerwerk and the other co-operating firms. This is the reason why the entire operation could be performed without complications, thus giving the aquanauts a feeling of security.

FIRST LAUNCHING

On 28 July 1969—only eight months after receipt of a firm order—the Unterwasserlaboratorium "Helgoland" was towed to its actual position by two boats of the Biologische Anstalt Helgoland. The self-contained support buoy had already been moored to its actual position eleven weeks before this event. The laboratory was submerged with a velocity of 7 m/min and had a "dead on target landing" after exactly three minutes. Two engineers (another designing-engineer and the author) and two aquanauts from the Biologische Anstalt Helgoland were on board.

The results of the first 22 days' operation are based on data received by the kindness of Dr. Gotram Uhlig, the chief aquanaut of the Biologische Anstalt Helgoland. The author also calls attention to the fact that he is not a marine biologist, but the chief of the project and construction department for diving techniques of Draegerwerk.

The results obtained during the initial operation of the "Helgoland" laboratory can only be fully appreciated by taking into consideration the local environmental conditions. As has been mentioned before, the North Sea near the Isle of Helgoland is a most dangerous water. One year prior to the initial operation, extensive *in situ* measurements of the tidal current on the seabed near the actual stationing position of the underwater laboratory were performed. The highest current measured was 80 cm/sec, while the average was between 20 and 40 cm/sec. Only a skilled diver is able to overcome such velocities. The comparatively low water temperatures are not at all agreeable. Even during summer the water temperatures near Helgoland never rise above 14°C (57°F), in a depth of 23 metres and the insulation capacity of the commercial type neoprene diving-suits is reduced due to compression in such a depth. (Unfortunately there was no money for heated suits.) Another point which creates considerable difficulties for the scientist under water is the poor visibility.

Fig. 8 shows the average weekly visibility values obtained by means of a Secchi-disc during the years 1968 and 1969. The upper curves indicate the average wind velocities, whereby the relation between wind velocity and visibility can be clearly observed. The most favourable conditions, a view over a distance of three to six metres, are met between the months of May and September. Thus the initial operation of "Helgoland" took place at a time when visibility conditions were most agreeable. It does not seem expedient to carry out investigation tasks during the winter period because, apart from the poor visibility of only a half to three metres average, the low water temperature of almost 0°C limits all diving procedures to a minimum. Apart from that, the wind conditions near the coast of Helgoland limit the periods of manned underwater laboratory operations. Fig. 9 indicates all the wind currents of the last ten years that have exceeded a daily average of 8 m/sec, which corresponds to a wind force of 4 on the Beaufort scale, and which lasted longer than three days without interruption. This shows that there was a minimum of wind during the months from May to September, while during the winter season there were periods of storm which lasted from about three to six weeks, sometimes even more.

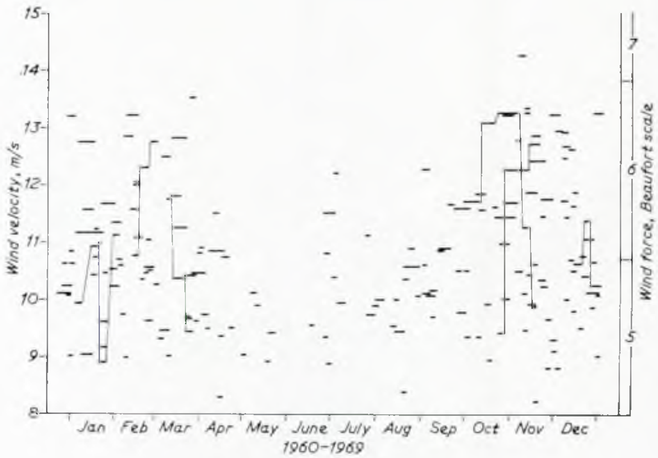


FIG. 9—*Wind velocities*

The Unterwasserlaboratorium "Helgoland" has been equipped so that, in the event of being cut off from surface support, the crew can be sustained for a period of fourteen days. The laboratory carries all emergency supplies such as electric power, gas, food, fresh water, soda-lime, etc. Considerably longer interruption periods, however, are liable to occur during the winter season, and these would involve too many risks for the underwater team.

One day after the initial submergence and after a decompression period of nine hours in the laboratory, the two engineers were replaced by two aquanauts so that the first team was complete. The daily work was primarily devoted to technical tasks as the crew had to be acquainted with the

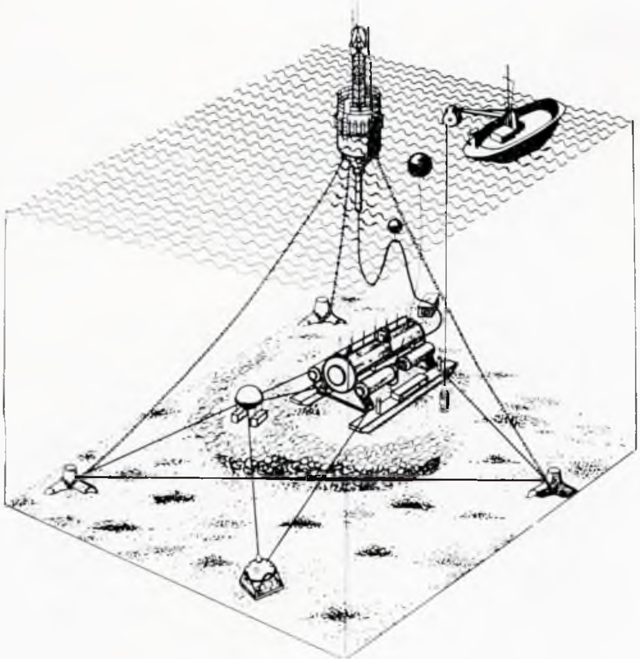


FIG. 10—*View of the panorama at the site of operation*

technology and new conditions. After professional and sport divers had installed all auxiliary equipment, the underwater depot, which has a similar design to the underwater igloo without a life-support-system, is charged with soda-lime and silicagel containers and situated eight metres away from the laboratory. The underwater "igloo" is set up approximately 43 metres away from the laboratory. The triangular area between the tetrapodes, which is marked by guide ropes, was the actual biological investigation area. The excrements, which are first caught in a sewage tank, are let off with the tidal current by way of a plastic tube 100 metres long. Stock replenishment of every-day supplies was by means of a "stew-pot" from a supply vessel. This "stew-pot" was airtight and had to be carried to the laboratory by supply divers. As this operation soon proved to be difficult and did not work very satisfactorily, an automatic submersion procedure is planned for the future.

The diving periods which were daily performed by the aquanauts indicate that the divers became gradually acquainted with environmental conditions and technical systems. After a period of ten days the first team of four divers performed an average diving period of $1\frac{1}{2}$ hours per day, while the second team performed two hours daily and finally the third team of three divers performed three hours daily. Individual diving periods of even four hours daily have been performed by divers, which is doubtless a very remarkable performance considering the unpleasant conditions mentioned above. It is therefore not at all surprising that a consumption of 5000 calories has been recorded by the physicians. Apart from the cold food, the divers prepared deep frozen ready cooked meals of high quality. It was interesting to note that all divers had the feeling of a considerable increase of weight, while in fact a loss of weight of between three to five kilos in ten days was registered after they had returned to the shore.

It cannot be denied that a certain degree of inertia has to be overcome prior to any physical labour in the underwater laboratory. Due to the physiological exertion, the divers had a tremendous demand for sleep. In general they had about nine to ten hours' sleep. According to experience, the working capacity of the aquanaut seems to be dependent on three conditions:

1) on the fitness of the diver;
2) on adaptability to the special environmental conditions;
3) on a certain degree of living comfort.

The combination of a team is of great psychological importance. On the other hand, it should be noticed that interdependence had a very stabilizing effect on the crew.

The fit condition of the aquanauts was proved in very thorough medical examinations which were carried out prior to the dives. The adaptation period of a new diving team takes about one or two days. When an aquanaut is fully saturated in the depth of 23 m he will need a 24-hour decompression period in the underwater laboratory. Therefore it would not be at all economical to assess an operation period in the laboratory of four to six days only. The team should instead be working under water for a period of at least eight to ten days. After all, the labour which is performed under water takes no more time than labour which is performed ashore.

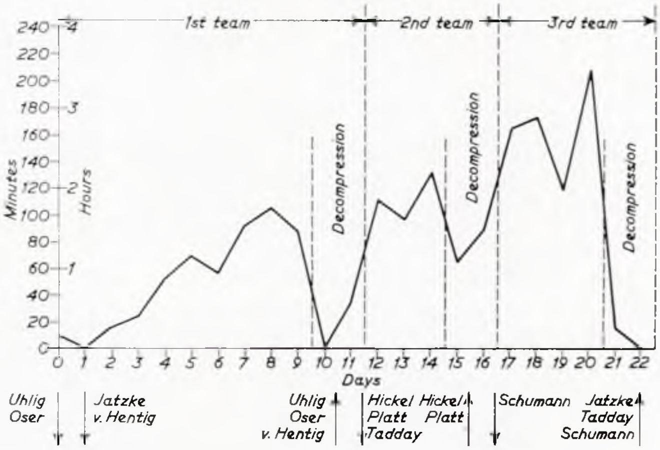


FIG. 11—*Diving periods of the aquanauts*

Uncomfortable living conditions no doubt have a negative effect on the working capacity of the divers. About fifty different underwater habitats have been tested and operated at different water depths by numerous nations during the last eight years, beginning with the French Precontinent enterprises and the American Sealab tests, and up to Tektite II and the Aegir-record operation in Hawaii where, last summer, a depth of 158 m, that is approximately 500 ft, was reached.

Now that we have manned underwater projects it is very important to increase the efficiency of such underwater laboratories with regard to special fields of operation like the living conditions mentioned above. This requirement has in fact been met by the Unterwasserlaboratorium "Helgoland" which provides a considerable amount of comfort in spite of its comparatively small dimensions. The freshwater shower was particularly appreciated, also the hygienic WC with its underwater flush system, as well as the almost noiseless operating air-conditioning plant and last, but not least, the perfect kitchen. The air temperature of the underwater laboratory was set from about 18 to 21°C overnight, and between 20 to 23°C during the day. As the atmosphere did not contain any helium, these temperatures were sufficient. There was a temperature gradient of about 5°C between the floor and the ceiling. In future better air circulation will be required.

The increase of humidity within the laboratory was rather disagreeable. Considerable quantities of water were brought in after the numerous diving tasks and the four silica gel dehumidifiers which were driven by ventilators could not eliminate the humidity. In spite of the fact that the third team of aquanauts consisted of three men only, the situation did not improve very much, since more water was brought in due to the more frequent dives. The humidity value amounted to 83 per cent on the first day of operation and values of about 90 per cent were measured after 13 days. Unfortunately there was no possibility of de-humidifying the laboratory, due to energy insufficiencies. It will therefore be necessary in future operations to separate the wet rooms from the dry rooms or to obtain the water separation by means of condensing systems, provided that the energy supply can be improved.

The regeneration capacity of the life-support-system has been so designed that the CO_2 values range between 0·1 per cent and 0·3 per cent, when the soda-lime containers exchange in a 36-hour cycle. The O_2 values ranged between 15 per cent and 21 per cent, but in general were below 20 per cent.

During initial operations in the Baltic Sea in 1968, all divers had been complaining about irritation of the external auditory meatus, which was occasionally accompanied by pains. In Helgoland the medical doctors have tried to take prophylactic measures, unfortunately with little success, because five of the eight divers had ear complaints. A smear which was made from the clear discharge of the ear showed that it was an infection caused by pyoceanic bacilli.

PROSPECT

The experiences which have been gained with the Unterwasserlaboratorium "Helgoland" until now can be summarized as follows:

even in our rough northern area the operation of underwater habitats is absolutely profitable, both technically and scientifically. Underwater habitats, which are very efficient and important in the field of marine biological research, will have to be further developed to help marine biologists solve the manifold problems which are involved in the exploitation of the oceans.

An official statement with the first evaluation contained the following particulars:

"The initial operation period in the 'Helgoland' underwater laboratory was above all intended to test the various kinds of technical equipment under the special conditions prevailing at the bottom of the sea. Its purpose was to provide information for the detailed planning of a long-term marine biology research programme. To this end initial research work was devoted to simple investigations of marine organisms living in the water and at the bottom, and to systematic examination of the medical and psychological effects on the reactions and performances of the aquanauts."

A full evaluation of the technical and scientific results of these experiments will take months to complete. However, a few fundamental conclusions can already be drawn:

1) There were no psychological problems. All the aquanauts felt very well. Their stay at the bottom of the sea produced a remarkably positive emotional stabilization.
2) There were no fundamental medical problems.
3) The technical equipment worked perfectly.
4) The significance of the UWL for the future of marine biological research at the bottom of the North Sea was viewed with optimism by all aquanauts and by all those in charge of the project. A new dimension has been opened for marine research.

Thus it can be stated with certainty at this early point that the public funds devoted to this project have proved a very useful investment. The question now remains as to the future of this project. The Ministry's report is also very satisfactory from this point of view. It states:

"The 'Helgoland' UWL is expected to remain in its present position for one year—at first unmanned, but under permanent supervision by diving teams working for short periods. During the autumn and winter storms experience can be gained about safety factors and working possibilities under extraordinary conditions."

For the first year of operation the following research projects are planned in the Marine Biology Research Programme.

1) Measurements of biologically vital environment factors, particularly: the use of thermistors to determine temperature gradients in water close to the sea bed and the sea bed itself; measurements of current velocities especially very close to the sea bed; recording of water disturbance caused by tidal currents; measurements of light close to the sea bed using underwater photometers.
2) Collection of plankton for specific purposes and particle counts during the various phases of tidal movements.
3) Collection of sediments (over a period of several successive days) and their analysis.
4) Investigations of micro-biological organisms and of micro-fauna, experiments to determine the qualitative and quantitative parameters for the movement of sediments displaced by and carried in tidal currents and the resultant variations in distribution patterns of micro-organisms; continuous sampling of sediments in sterile conditions to determine changes in micro-flora (cell count factors) under the influence of the environment; studies of the resettlement of sterilized areas.
5) Experiments to determine physiological behaviour, such as setting out lobster grubs, young lobsters and mature lobsters on beds of pebbles, in artificial holes and *in situ* breeding chambers; studies of migration, social, environmental and sexual behaviour as well as daily cycles of activity; parallel studies are also to be made of hermit crabs.
6) It is also planned to investigate the possibility of breeding and feeding ecologically and economically important species of fish crabs and shellfish in large sea-bed "aquariums" and other special underwater units.

FIG. 12—*UWL "Helgoland" after nine months exposure*

The Underwater Laboratory "Helgoland" was being thoroughly overhauled and partially reconstructed at the time of writing. Preparations are being made for supplementary mounting of a special wet room, so that the problem of humidity would be solved for future operations.

Discussion

LT. CDR. C. A. HELY-HUTCHINSON, R.N., A.M.I.Mar.E. said that in discussing the life support system, Mr. Haux mentioned control of oxygen and carbon dioxide. What action had been necessary to control other gases and aerosols present in trace form, which were important when living in an enclosed space, such as aromatic hydrocarbons which resulted in gaseous or aerosol form from human waste?

CDR. M. B. F. RANKEN, M.I.Mar.E. said that, as usual, Mr. Haux had given them an excellent description of his habitat, and a lot of the things he was doing to improve on what went wrong before regarding humidity etc. When he had spoken at the Society of Underwater Technology in September 1971, he had had one of his colleagues on the marine biological side present. Could Mr. Haux say what had been possible with the habitat when it was off Helgoland in 1969/70, and also about the problems of communications and cables which did not survive. Although not serious, these were of operational interest.

Author's Reply

In reply to Mr. Hely-Hutchinson, the author said that, in the paper something had been said about the measurement of CO_2 and oxygen only, but they might have heard about the famous Dräger gas detecting tubes which could be used for about a hundred different gases and gas mixtures. One can be sure that they did their best and measured all these things in the underwater habitat, and that they had been recorded. However, he was not a physicist or chemist, so could not say to what range they had arrived. But after the twenty-two day experience, they had still been within the safety limits.

In reply to Commander Ranken, Mr. Haux said that they had only heard about positive experiences with equipment, but he thought that more could be learnt in speaking about the negative experiences which they had had. He would therefore say something about those. In Helgoland their first experience had been with a marine biologist. The first thing that they had found was that it was not very easy to bring biologists in to do a mechanical function; if they had to turn a lever to the right, they would do it wrong and turn it to the left. They had tried to put them into the picture in three days, but they had had a lot of trouble. It was about twenty days before the biologists had been trained in the way that they liked.

An underwater laboratory such as theirs was a very difficult system. They had had visitors who had turned on a tap and had expected to have warm water, without thinking where it came from, and this had been very bad for the technicians. Thus they had had a relatively bad time at the beginning.

Concerning the cables, they had had four communication ways from the underwater laboratory to the shore base, two of them wireless by the buoy incorporated below the level of the water, and there had also been telephones to the normal base. They had planned to go by cable, over the water, to the shore base. Those things had never worked well because, after two hours, communication had been broken in the surf on the beach area. They had not been able to repair the cable during the first four weeks. Afterwards they had had the supply buoy working as a supply buoy for gas and electrical energy quite satisfactorily. They had bought the Diesel oil in a container, to the buoy, after the first two weeks, and the container had been full of dirt, so there had been something wrong in the logistics. The Diesel did not like the dirt and the filter had become blocked very quickly. They had had several unfortunate problems with the water supply in the first period. However, all in all, the system worked quite satisfactorily. In the second operation, most of the alterations they had had to make were basically to put on a new wet room because they had found that during the first run the humidity in the laboratory had been too high. It had been about 90/95° relative humidity with no wet room.

To some extent they had expected this, but they had been very short of funds, and the whole system under water level including the buoy, the decompression chamber, and the land-based control station, as well as the driving unit, all had had to come out of about £100 000, which was not very much for such a big system. Therefore at that time they did not have the money to enlarge the system and to add the wet room, but they were now at that position, and he thought that they would get the money within the next few months, so that next year they would be better placed. On the other hand, the underwater laboratory had been successfully used by marine biologists, and he considered this a very satisfactory thing because if one read the history of underwater laboratories built since 1962, one would know that there were now nearly fifty of them; some of them would never be used, and a lot of them would only be used on one occasion. There were very few which had been used two or three times. This particular laboratory would be used for years, for basic development and basic research, not only for technical reasons but mainly for pure marine biology.

Related Abstracts

Atmospheric control in the hyperbaric environment

Careful control of the atmosphere in a hyperbaric facility is necessary to protect the health and well-being of the occupants. In this paper total environmental control in the hyperbaric chamber is broken down into eight sub-areas of control: total pressure, oxygen partial pressure, CO_2 partial pressure, fire prevention, temperature, humility, trace contaminants, bacteria, and noise. For each sub-area an assessment is given of the difficulty and degree of control required and its importance to the divers' well-being. Also discussed for each sub-area are the state of the art, and the areas of greatest technological need.

Control of oxygen and CO_2 partial pressure in breathing circuits of diving equipment is described. Methods of heating the diver and his inspired gas are also briefly discussed.—*Reimers, S. D., Naval Engineers Journal, June 1971, Vol. 83, pp. 50–65.*

New oil tank construction

A unique project for underwater oil tanks has been developed in Oslo, representing a radical break with conventional construction methods and performance of oil tanks.

Submerged to the bottom of the North Sea, the installation will hold one million barrels.

The tanks are composed of cylindrical containers of prestressed concrete. They are cast on land, then set afloat and joined together whilst afloat in the harbour. In this way the tanks can be made completely tight and all inside concrete walls can be given coatings which prevent the oil from penetrating into or through the concrete.

Whilst the tank volume is of exactly the same size as that of the biggest underwater oil tank in the world, i.e. Phillips' Ekofisk tank now under construction at Jattavagen in Rogaland, Southwest Norway, considerably less concrete will be required for the construction, and the entire installation will be of far smaller dimensions.

The tanks will be about 45 m high, standing about 25 m below the surface, at the bottom of the sea. They are planned to be placed in an area where the North Sea is about 70 m deep.

Six 45 m steel cylinders erected on the tanks will carry a rig with a total area of about 3000 m^2, lying about 20 m above the sea surface. The rig will be equipped with all gear required

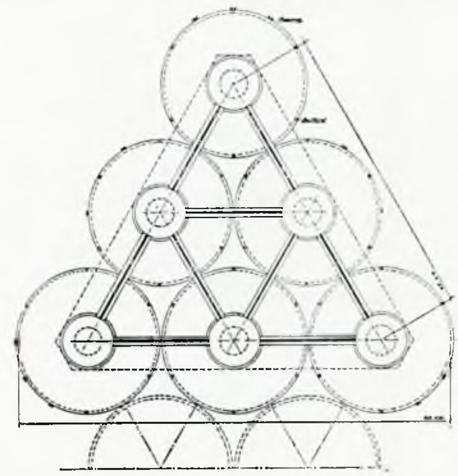


Diagram of the tank installation in position on the sea bed. The steel cylinder and the top platform are shown with dotted lines. The two tank sections at the bottom illustrate the possibilities of the system to increase its capacity by installing additional tanks.

for safe operation. The base of the installation is almost triangular, the sides measuring about 93 m each. Each tank section has seven compartments. Twelve such sections are necessary for a storing capacity of one million barrels of crude oil.—*Norwegian Journal of Commerce, 25 November 1971, p. 10.*