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TRANSACTIONS (TM)

BP's PERFORMANCE-MONITORING SYSTEM FOR MARINE DIESEL ENGINES

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BP's Performance-monitoring System for Marine Diesel Engines

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SYNOPSIS

As part of a comprehensive energy-conservation project, methods of optimizing marine diesel-engine performance were examined; this was particularly relevant because of the effects of deteriorating fuel specific energy. About three years ago, the author's company evaluated a mean indicated pressure calculator, with good results; however, because ships' staff would need guidance in its use, a comprehensive standard of application was developed: the performance monitoring system for marine diesel engines. The author describes the system's objectives, background and development, together with a full explanation of how flow charts, fault-finding tables and trend graphs are used to monitor and optimize engine efficiency. Some results taken from the 12 ship-years' operating experience already accumulated with the system are shown and, in conclusion, some ideas are developed on future computer-aided fault-analysis.

The views expressed in this paper do not necessarily reflect the policy of the Author's Company, although they are based on data extracted from project work and shipboard evaluations carried out over the past few years.

INTRODUCTION

Several years ago, like most other owners, BP Shipping Limited embarked on a carefully planned energy-conservation project. The objectives, in terms of fuel savings, were clearly defined, as were the broad areas in which these savings could be made. Because the company operated a large number of diesel-powered vessels, it is not surprising that marine propulsion diesel engines received immediate attention.

The author's duties at that time included a detailed appraisal and subsequent shipboard evaluation of diesel-engine performance and condition-monitoring techniques. This paper deals with the performance aspect and describes a system aimed at optimizing the specific fuel consumption of the main engine.

PERFORMANCE MONITORING

Historically, marine engineers have monitored the performance of main diesel engines in the following ways:

- By reading data from fairly basic local instrumentation fitted to the machinery and subsequently recording these in an engine room log-book, on a watch-by-watch basis. Even up to comparatively recent times, the degree of sophistication introduced into this system has generally been limited to:
  - Centralization by use of remote-reading instruments;
  - Data logging on demand or automatically;
  - Control of temperatures and pressures, within preset limits, by suitable control systems;
  - Limitation of load/speed, by means of governors;
  - Incorporation of various degrees of alarm/shutdown facilities.
- By sound, sight, smell and touch, coupled with a general awareness of the engine room environment (a 'feel' for how the machinery is running—a 'sixth sense').

Performance-monitoring of engines by the above means has obvious disadvantages:

- The log-book data are not corrected for changes in ambient conditions in the engine room and fuel specific energy.
- Trends in performance are difficult to detect by scanning the log-book entries page by page.
- The accuracy and repeatability of measuring devices are often outside acceptable limits.
- Performance data published by engine manufacturers are based on different ambient conditions and fuel specific energy to those experienced in service.

However, with the introduction of devices that monitor engine combustion and fuel system condition (mean indicated pressure (MIP) calculators), the engineer can be more acutely aware of how his engine is performing. These devices are a great improvement on mechanical indicators.

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OBJECTIVES AND BACKGROUND

Objectives of the system

These are as follows:

- To carry out a critical examination of the combustion process and then, through step-by-step corrective actions, to tune the engine so that an acceptable balance is obtained.
- To carry out a critical examination of the engine's fuel system and make any adjustments required to ensure correct operation.
- To provide long-term monitoring of engine performance by plotting relevant performance data on 'trend graphs'.
- To record engine conditions, at preset running-hour intervals, for reference purposes.
- To establish that the engine is operating within design limits and that bearing loads are safe.
- To determine specific fuel consumption and thus monitor engine performance against a known standard.

An overriding objective during its development was that the system should be simple, containing as little operational paperwork as possible, and be presented in such a way that it could easily be consulted and, therefore, used to best advantage.

Background to the development

Some years ago, the author's company commissioned the building of several 25 000 tdwt sister ships, all to be fitted with the same make and type of diesel propulsion machinery. Neither the vessels nor the engines were all built at the same yards, so this presented an ideal opportunity for comparing engine performance across the class.

The comparison was made by constructing graphs of the following parameters, the values of which were taken from shop and sea trial data.

1. Absolute compression pressure (P_{CA}) to a base of absolute scavenge pressure (P_{SCA}).
2. Maximum cylinder pressure (P_{MAX}) to a base of mean indicated pressure (P_{MI}).
3. Fuel pump index (FPI) to a base of mean indicated pressure (P_{MI}).
4. Cylinder exhaust temperature (T_{EXH}) to a base of shaft kilowatts (SkW).
5. Main engine revolutions (N_{ME}) to a base of mean indicated pressure (P_{MI}).
6. Scavenge pressure (P_{SC}) to a base of mean indicated pressure (P_{MI}).
7. Pressure drop across waste heat unit (ΔP_{WHU}) to a base of shaft kilowatts (SkW).
8. Pressure drop across air filters (ΔP_{AF}) to a base of scavenge pressure (P_{SC}).
9. Pressure drop across air coolers (ΔP_{AC}) to a base of scavenge pressure (P_{SC}).
10. Specific fuel consumption (SFC) to a base of shaft kilowatts (SkW).
11. Turbo-charger revolutions (N_{TB}) to a base of pressure at turbo-charger outlet (P_{TB}).

Since the ships had identical hull forms and engines, and the same conditions (e.g. ballast) prevailed during their sea and shop trials, it seemed logical to assume that, across the class, there would be very little difference in the results as shown on the above graphs. This was not the case and the spread of results between vessels was quite considerable. This could only be explained by:

- Variations in ambient conditions from one shop/sea trial to the next.
- Variation in the specific energy of the fuel used on the trials.
- Incorrect ignition timing on all, or some, of the engines.
- Inaccurate measurement of trial data.

The next, major, task was to establish the effects of outside influences on the families of 'curves' which had been drawn up for the class of ships under investigation. A mathematical model was used to determine which factors had a significant effect on the position of each curve.

It was found, for example, that the following factors had a significant effect.

Sulphur content of fuel.
Specific gravity of fuel.
Engine room ambient temperature.
Scavenge air temperature.

The following factors had comparatively little effect:

Water content of fuel (after treatment).
Ash content of fuel.
Ambient pressure in engine room.

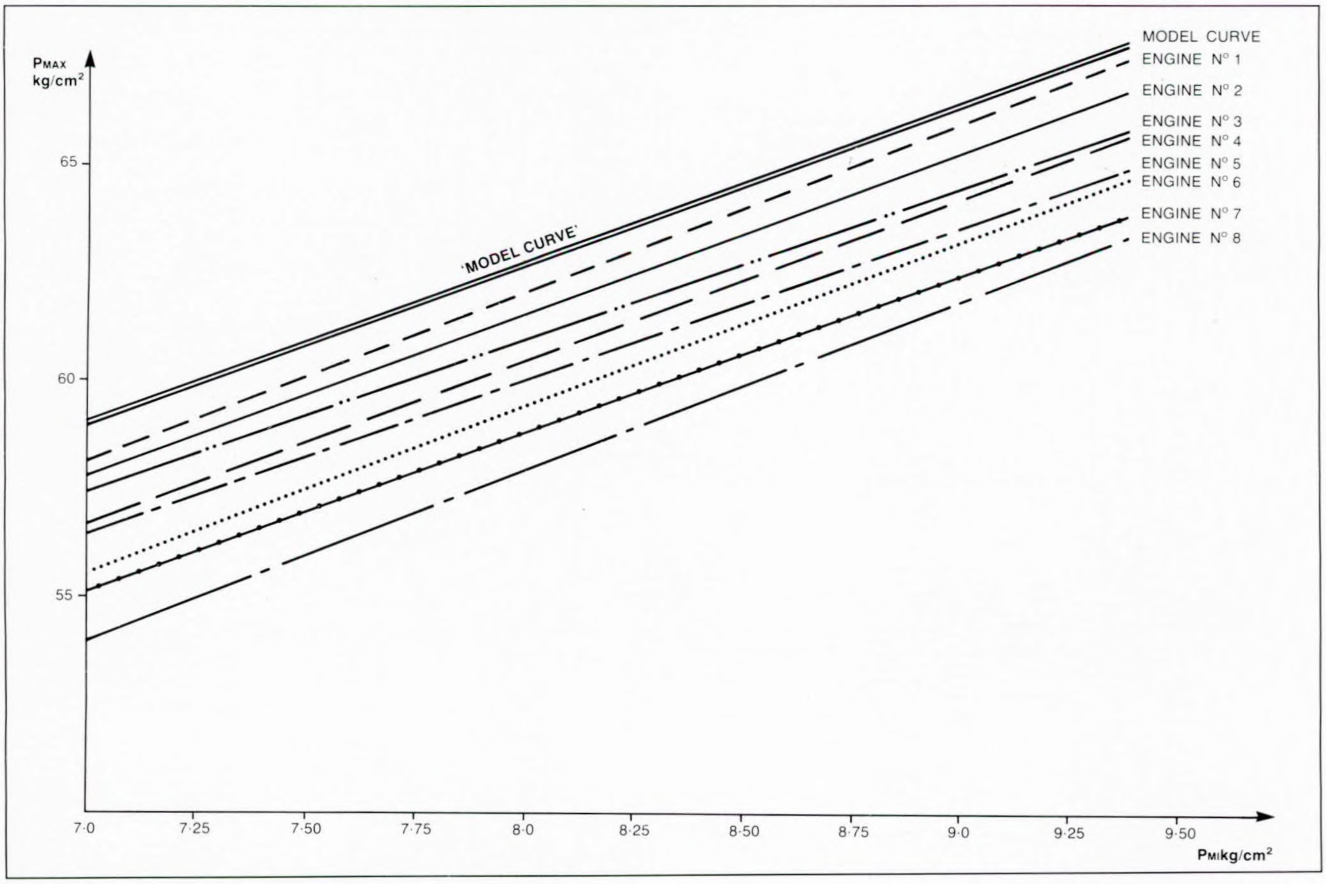
It was then decided that the system should be based upon the ISO standard conditions of ambient temperature (27°C), scavenge air temperature (45°C) and net specific energy of fuel (42.0 MJ/kg). Then, ignoring the factors having comparatively little effect, each curve was corrected to the standard conditions and redrawn, using the mathematical model. A significant scatter was still evident between individual vessels. Since all engines were identical, within normal manufacturing tolerances, it could be assumed that the remaining differences were mainly due to variations in adjustment of timing and fuel-system components.

Figure 1 shows a typical set of corrected curves (P_{MAX}/P_{MI}) for eight engines. It will be noted that, for the power range chosen (70–100%), the performance characteristics have been approximated so that they appear as straight lines.

For validation purposes, further data from engine manufacturers' test-bed trials were studied and compared with the results obtained; and, in some cases, used to increase the sample size. It was then possible, for each set of curves, to position a 'model curve' depicting the line of optimum performance.

In Fig. 1, for example, the model curve depicts the ideal relationship between P_{MAX} and P_{MI}, as defined by manufacturer's data, taking into consideration any down-rating which may have been applied by the owner.

FIG. 1 Corrected performance curves with model curve



Some of the relationships under consideration were affected by parameters that depend upon mechanical condition or degree of fouling: e.g. compression pressure and air cooler cleanliness. In such cases, the model curve was placed in a slightly more 'efficient' position than that of the best engine, but with the same slope as the bulk of the samples under consideration.

The basis for the performance monitoring system was then established in the form of the model curves. It remained only to determine how data were to be collected, stored, evaluated and presented so that corrective actions could be identified.

MIP CALCULATOR

Concurrent with the development of the performance monitoring system, a MIP calculator was being evaluated by the author's company. This proved to be an effective monitoring device.

The output from the MIP calculator provided measurements of the parameters required for the majority of the model curves.

Figure 2 is a block diagram of the MIP calculator. The device is operated by signals from transducers which are fitted to the appropriate parts of the engine. The signals are proportional to:

Scavenge pressure;
Crank angle;
Cylinder pressure;
Fuel oil pressure before the injector.

MIP is derived by integration over the compression and expansion strokes, averaged over eight cycles. It is then displayed digitally, together with engine revolutions; maximum cylinder pressure; compression pressure; expansion pressure 36 deg after top dead centre (TDC); scavenge pressure; angle between TDC and maximum pressure; fuel-valve opening pressure; fuel-pump discharge pressure, and crank angle between TDC and fuel-valve opening.

An oscilloscope is provided with the system. It displays curves of cylinder pressure and fuel injection pressure, from which permanent records may be taken by Polaroid camera.

THE PERFORMANCE MONITORING SYSTEM

The system itself is contained in two documents—an instruction manual and a data-recording file. These documents show, in a simple, step-by-step manner using diagrams and tables, how the MIP calculator is used to collect data from the engine and how the model curves are used to monitor performance. Finally, the documents cover fault-finding and logging of trends in engine performance.

The instruction manual is divided into four main sections:

1. Introduction and engine data.
2. Pre-performance check on the fuel system.
3. Performance check on the combustion process, engine balance and load.
4. Optimization of performance, using recorded data to compile trend graphs and identify final adjustments.

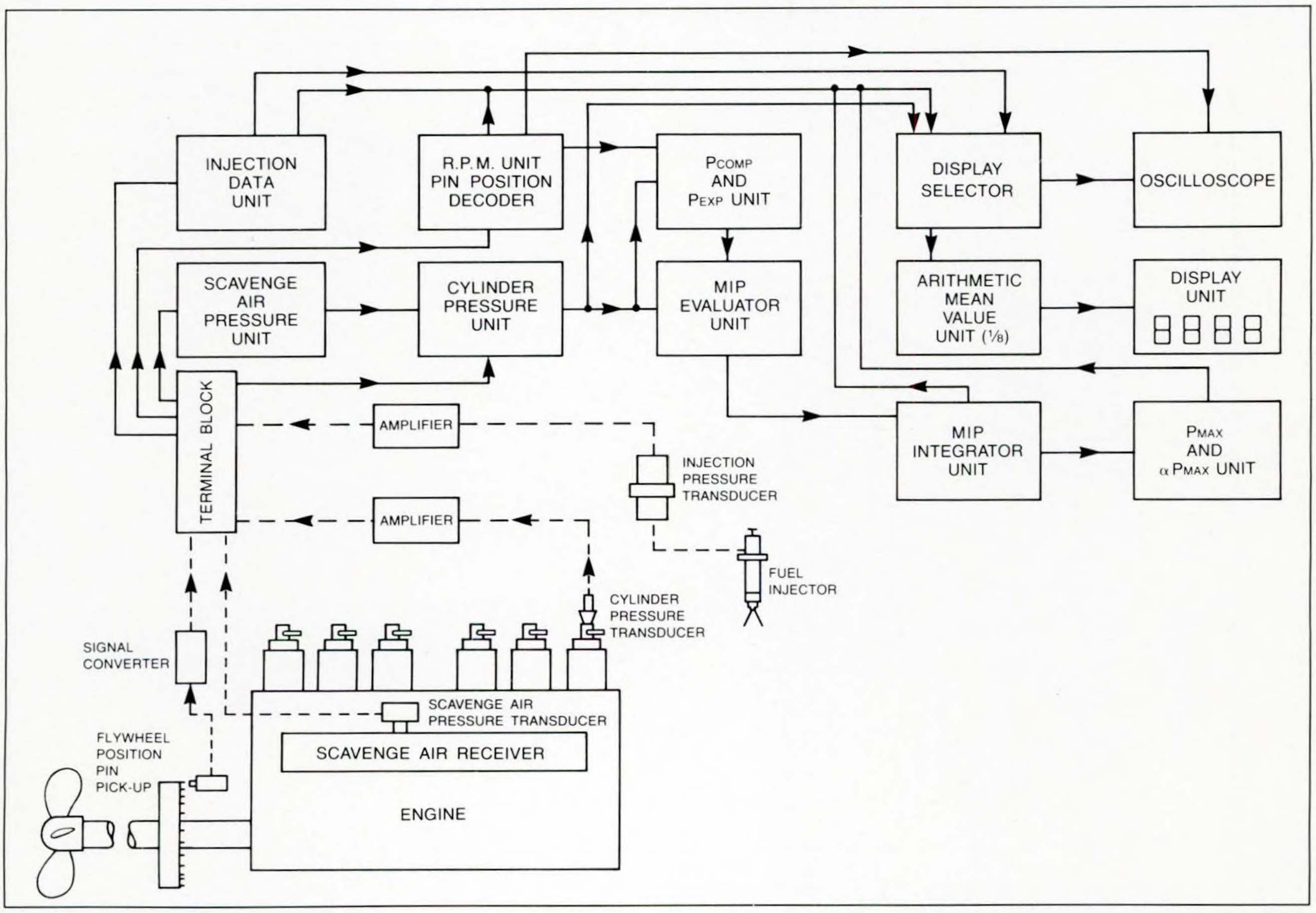
The actions in sections (2), (3) and (4) should be carried out at intervals of approximately 200 running hours.

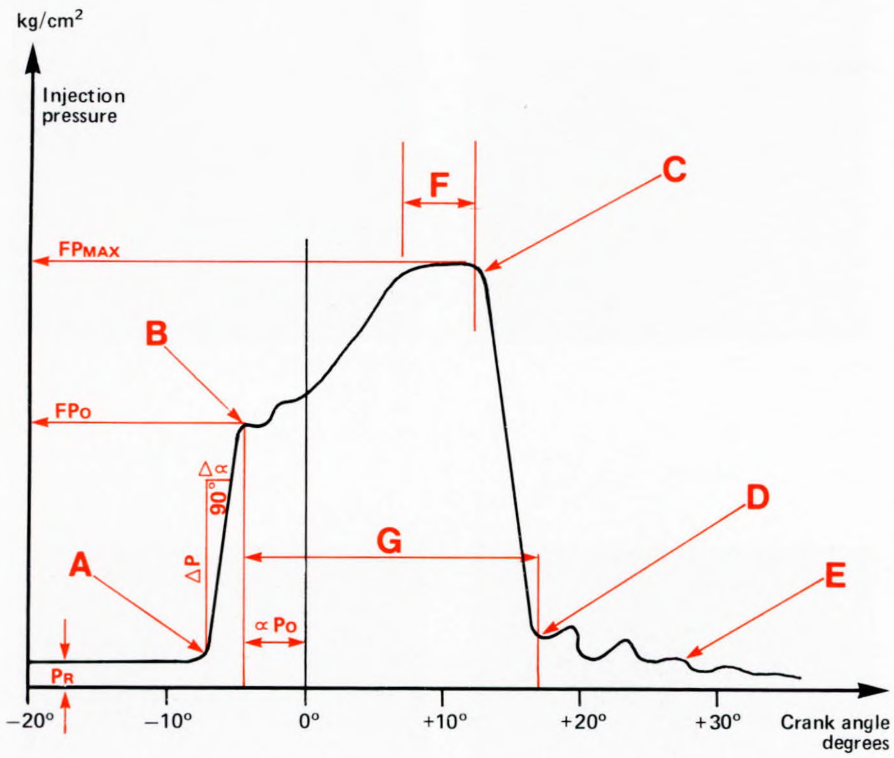
Introduction and engine data

The basic concepts and operation of the performance monitoring system can be applied to any marine diesel engine. It was necessary, however, to design certain parts of the system specifically for a particular make and type. The introduction and engine-data section therefore contains information relevant only to the engine on a particular ship. Subsequent sections apply to any diesel installation. Engine data are presented in the following format:

- Timing and dynamic operating conditions. Although these data are initially supplied to the vessel, they are often not readily available, therefore this section acts as a handy reference. Additional operating limits have also been suggested; for example, the allowable deviation in compression pressure is ± 1.0 kg/cm^2.
- The model curves and their equations.
- Typical traces of fuel pressure and cylinder pressure for an engine at full load, depicting (to scale) the shapes to be expected and identifying the salient points (see Figs 3 and 4).

FIG. 2 Block diagram of MIP calculator





A	Pump spill closes	8 deg approx. BTDC
B	Fuel valve opens	4 deg approx. BTDC
C	Spill opens	12 deg approx. before TDC
D	Fuel valve closes	16 deg approx. ATDC
E	Reflected pressure wave due to fuel valve closing	
F	Partial equilibrium	20 deg approx.
G	Injection period	
P_R	Residual fuel pressure	
$\Delta P/\Delta\alpha$	Rate of fuel pressure rise before fuel valve opens	2.5 approx.
FP_O	Fuel valve opening pressure	350 kg/cm²
αP_O	Angle at which fuel valve opens relative to TDC	4 deg approx. BTDC
FP_{MAX}	Maximum fuel pump discharge pressure	650 kg/cm²

FIG. 3 Typical trace of fuel pressure at approximately full power

- Some background theory on the correct procedure for down-rating the engine.
(Most engines are operated below the manufacturer's nominal maximum continuous rating (MCR). In this down-rated condition, the ignition timing requires some adjustment in order to raise P_{MAX} back to its MCR value commensurate with safe bearing loads. This will improve the thermal efficiency and specific fuel consumption. The information is presented to show a safe operating zone, defined by engine revolutions and mean effective pressure, in which the MCR value of P_{MAX} may be used.)
- A load diagram for the engine, clearly defining the safe operating zone.

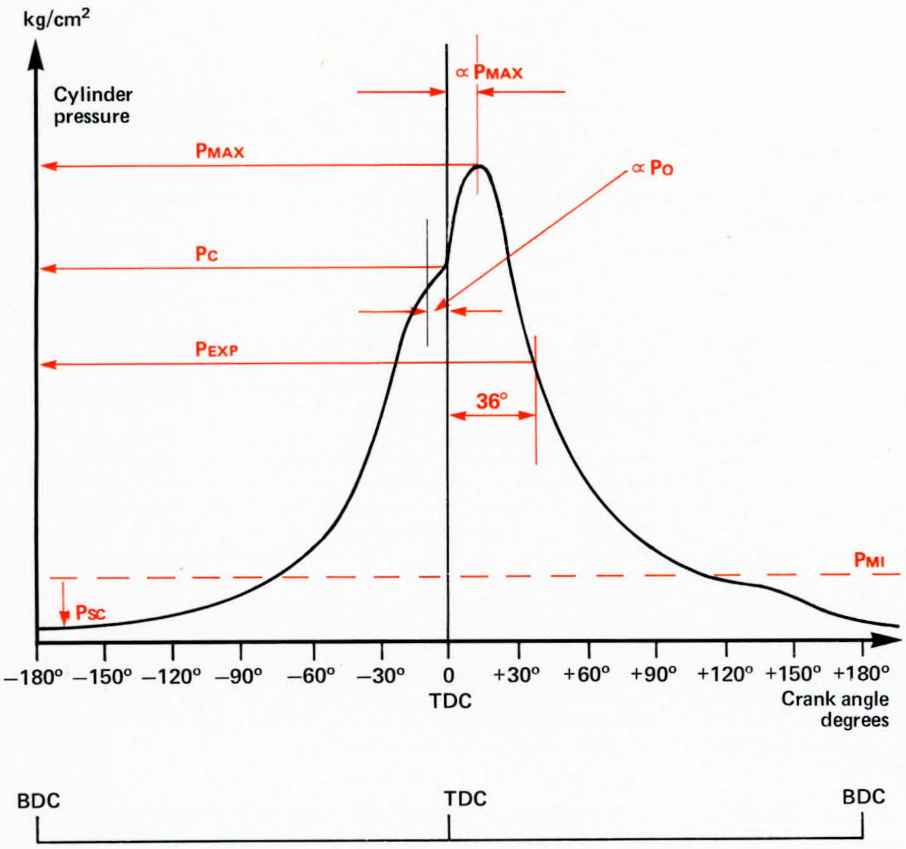
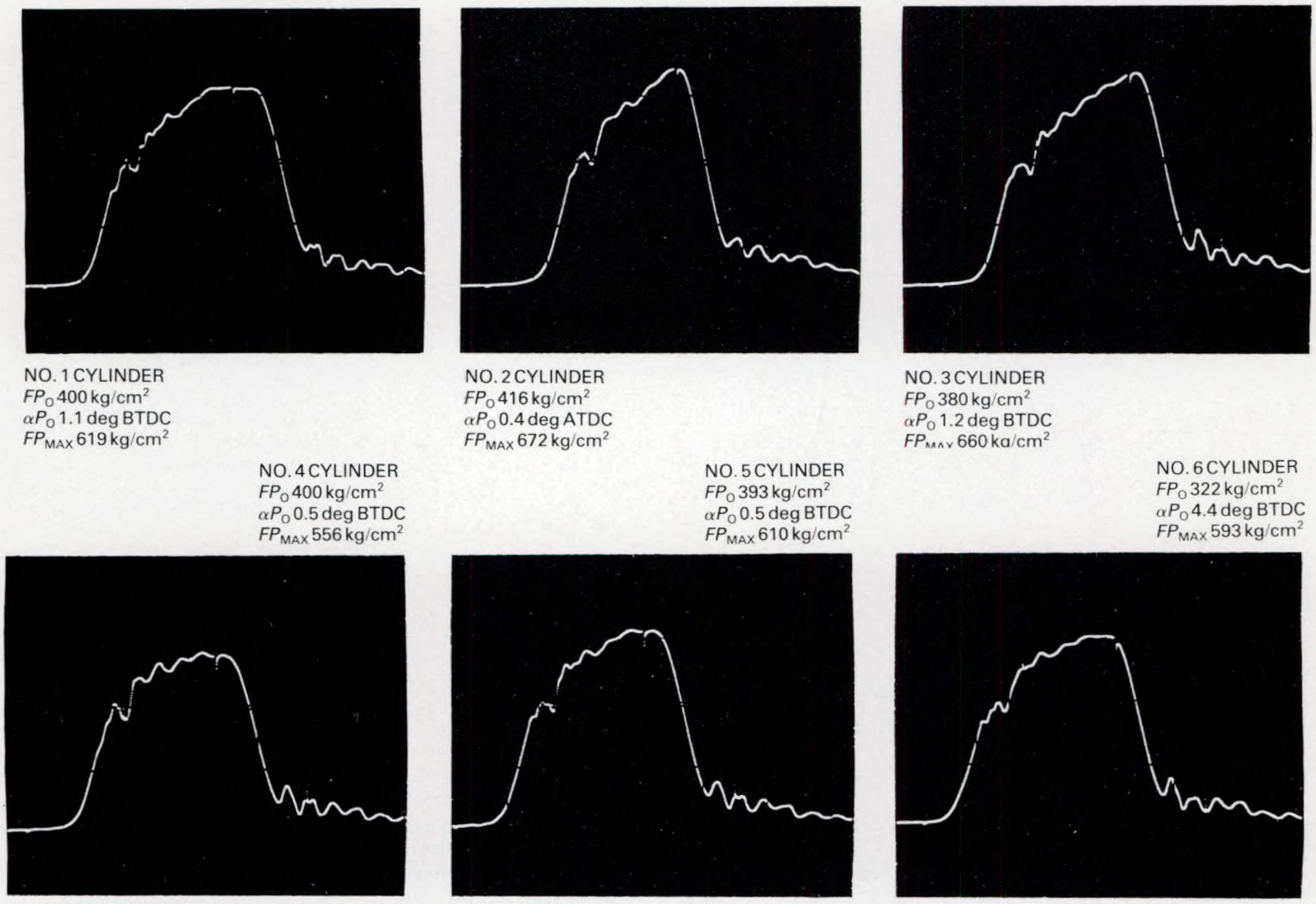


FIG. 4 Typical trace of cylinder pressure at approximately full power

BDC	Bottom dead centre	
TDC	Top dead centre	
SO	Scavenge ports open	142.5 deg approx. ATDC
SC	Scavenge ports close	142.5 deg approx. BTDC
FVO	Fuel valve opens	4 deg approx. BTDC
EO	Exhaust opens	. . . deg ATDC
EC	Exhaust closes	. . . deg BTDC



FP_O Fuel-valve opening pressure (normal value 350 kg/cm^2)
αP_O Angle at which fuel valve opens relative to TDC (normal value 4 deg BTDC)
FP_{MAX} Maximum fuel-pump discharge pressure (normal value 650 kg/cm^2)

FIG. 5 Polaroid photographs of fuel-pressure traces

Pre-performance check

Checks on performance and optimization are described in the manual by a series of flow charts showing green and red 'go' and 'no-go' paths, respectively.

As a diesel engine will operate most efficiently when the fuel system is in perfect order, the pre-performance check concentrates on this area—from storage of fuel to its injection into the cylinder. It commences with some simple, but often forgotten, tasks, such as checking fuel tank conditions, surcharge pump operation, filter condition and fuel viscosity.

The MIP calculator is then used to measure fuel pressure distribution at each fuel pump and display a set of traces; if required, a permanent record can be obtained in the form of Polaroid photographs (Fig. 5).

The results are recorded on a 'performance monitoring log sheet' (PMLS). The data table from the PMLS for fuel system readings is shown in Fig. 6, together with the relevant flow chart lines. Log sheets are 'customized' for the type of engine and therefore contain normal values and expected limits. Two references can now be used to commence fault-finding:

- Comparison of the values of fuel pressure distribution at the individual fuel pumps (obtained from the MIP calculator) with the normal values and the limits shown on the log sheet.
- Comparison of the photographs of the actual fuel traces with the 'ideal' trace (Fig. 3).

Tuning of the fuel system to achieve 'normal' readings across all cylinders, within the prescribed limits, is assisted by sketches that show 'faulty' fuel traces, typical of various engine defects, compared with 'ideal' traces. An example, showing the difference between an 'ideal' trace and that resulting from a worn or leaking fuel pump, is shown in Fig. 7.

A fault-finding matrix (Fig. 8) is also provided, to assist in identifying worn engine parts or incorrect settings of fuel-system components. If a ship's staff decide to make adjustments at this stage, without completing the whole 'performance check', the 'initial engine condition' is not known and there is no basis on which to judge any improvements. It is therefore recommended that a complete 'performance check'—described below—is carried out before corrective action is taken.

Performance check

This concerns the measurement of combustion conditions, engine balance and load. The relevant flow-diagram (Fig. 9) includes the recording section from the PMLS, entitled 'cylinder pressure trace'. The 'performance check' commences by using the MIP calculator to measure cylinder parameters; the oscilloscope displays are photographed, and the data are recorded on the log sheet.

For each parameter, the mean engine value is calculated for all cylinders and compared with the individual cylinder value, to establish whether an 'allowable deviation' has been exceeded. If there is an unacceptable deviation, the reason is investigated, in much the same way as pre-performance checking, by the use of a fault-finding matrix and comparative sketches of 'faulty' and 'ideal' cylinder traces.

Figure 10 shows typical photographs of combustion traces. Figure 11 gives an example from the fault-finding system, showing the effects on the pressure curve of a worn liner, defective piston rings or a burnt piston crown.

Corrective actions can now be taken to achieve engine balance, but they must be carried out in a logical sequence. Since MIP depends on all the other parameters measured, balance can only be achieved by correcting values in the following order:

P_C; αP_{MAX}; P_{MAX}; P_{EXP}; FPI; T_{EXH}; P_{MI}

Therefore, after corrective action has been taken to improve compression pressure (P_C), another complete set of readings should be

* While taking traces, the fuel system should be suitably 'locked' to remove governor influence and so achieve steady running conditions.

** The dynamic opening pressure of a fuel valve in service will be nearly 100 kg/cm^2 higher than the static opening pressure on the test bench. This is due to inertia in the moving parts of the fuel valve.

*** $\Delta P/\Delta X$ (rate of injection-pressure rise) and injection period will have to be measured by scale drawing from a Polaroid picture of trace.

**** Where faults are corrected immediately, two PMLS will be required: one showing the 'before-correction' data, the second showing the 'after-correction' data.

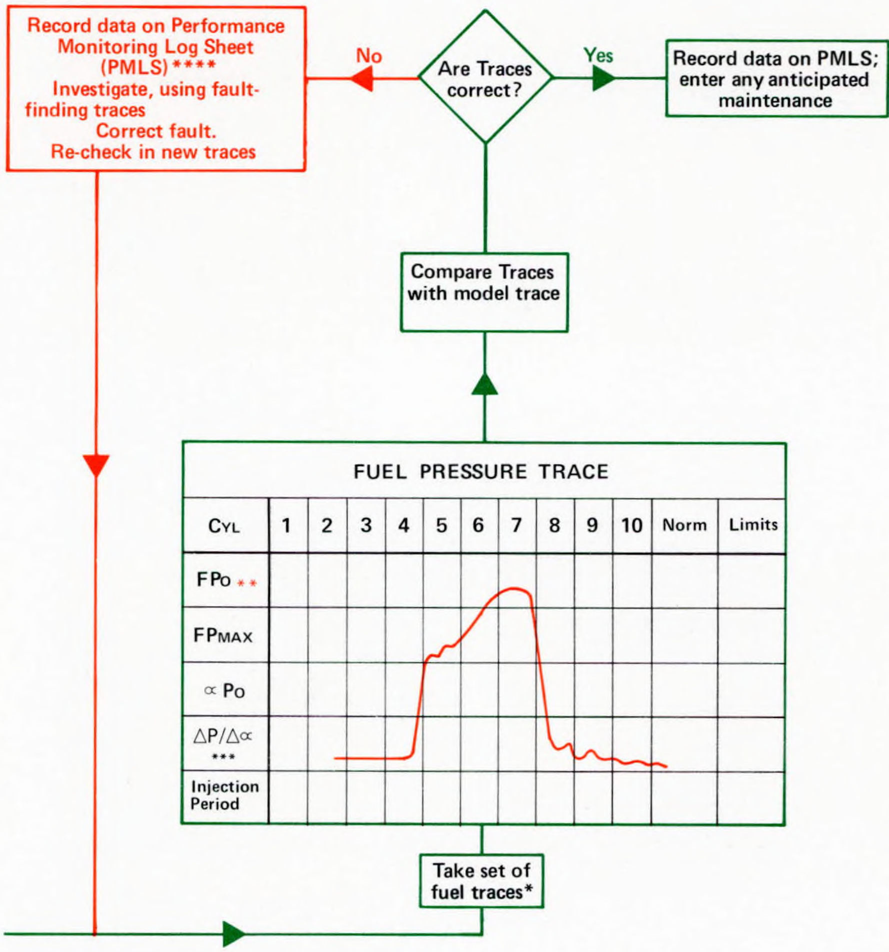


FIG. 6 Pre-performance check

obtained before the next adjustments are identified and subsequently carried out. Having dealt with P_C, balancing can be progressed by moving to αP_{MAX} (angle between TDC and P_{MAX}) until, finally, the exhaust temperatures are compared; P_{MI} should now be uniform across the engine. At each stage, the fault-finding charts and sample traces should be consulted.

Depending upon the engine's condition when the monitoring system is installed, it may take several months to complete the adjustments required to achieve engine balance. In fact, the process is continuous, because moving parts are subject to wear and possible failure.

Engine loading should be checked frequently, by reference to the load diagram provided with the system, and the main engine controls

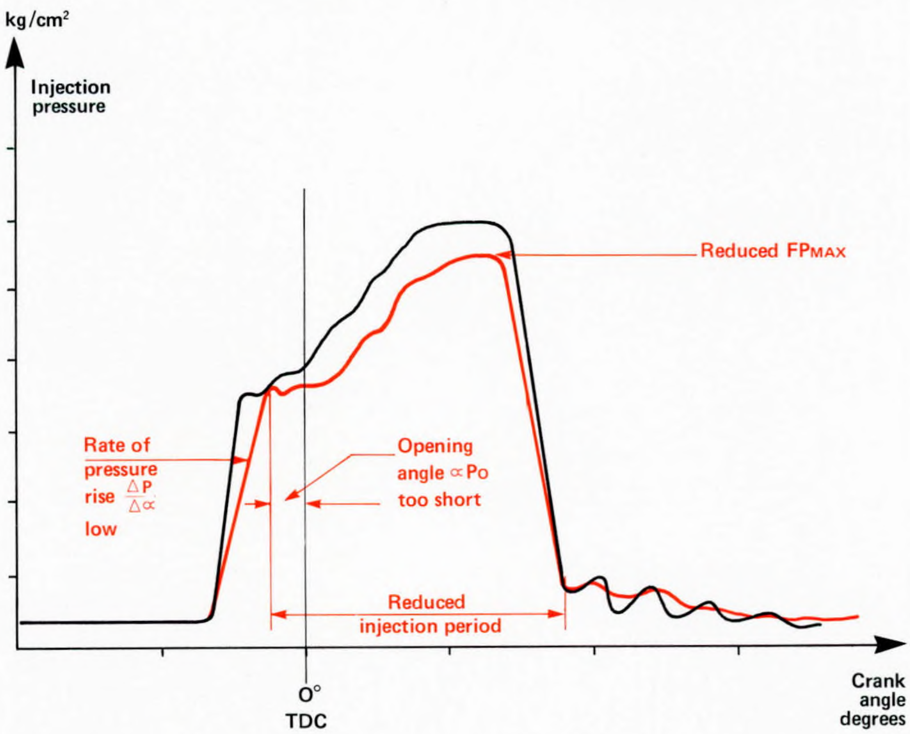
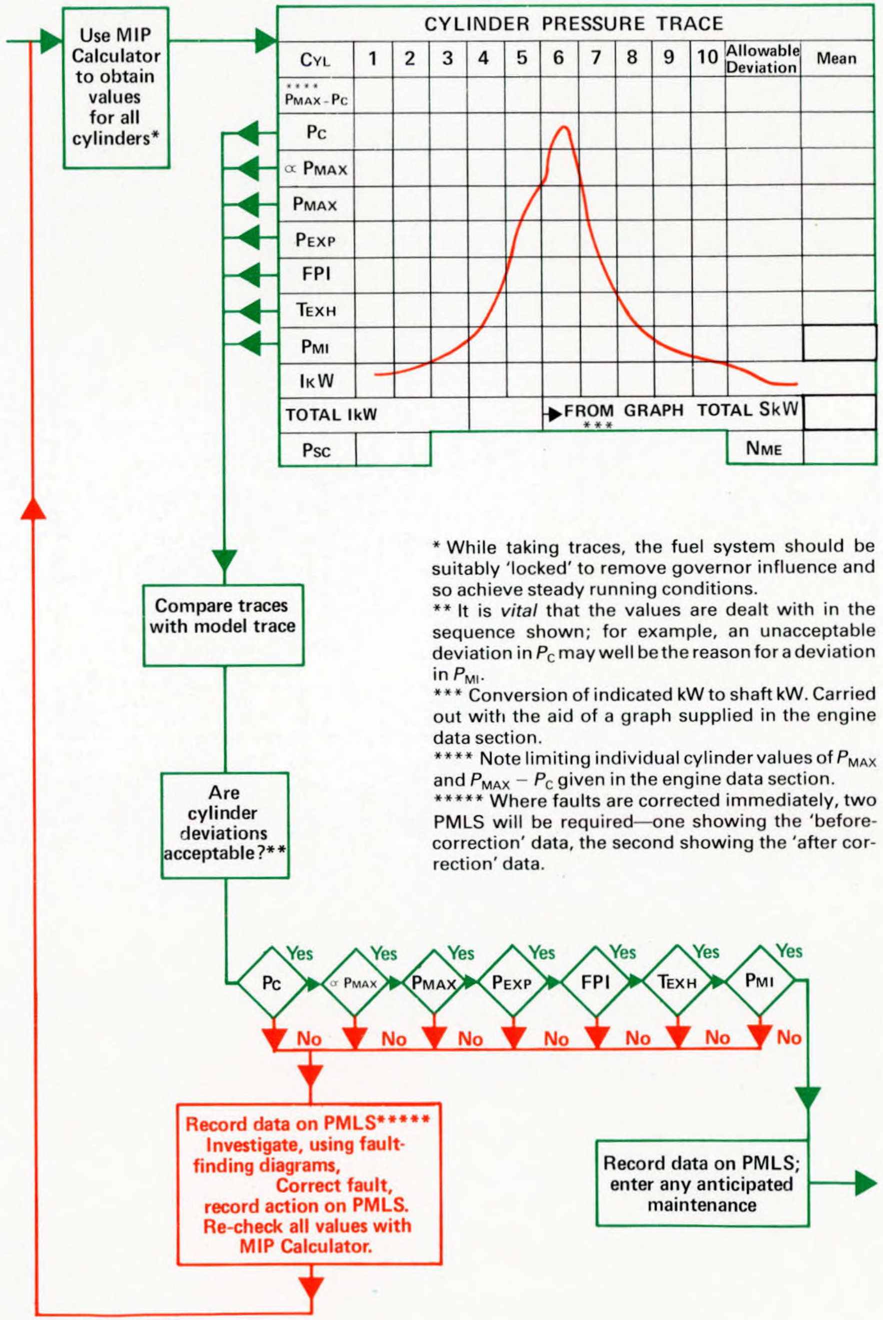


FIG. 7 Effects on fuel-pressure curve of: worn fuel-pump internals; leaking valve on fuel pump

INDICATIONS
- Reduced maximum fuel pressure.
- Low rate of pressure rise.
- Reduced injection period.
- Opening angle before TDC too short.
- Reflecting pressure wave damped.

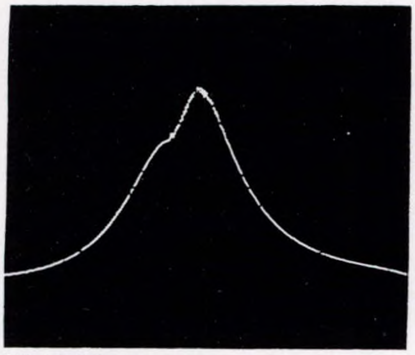
FAULTS	$\frac{\Delta P}{\Delta \alpha}$ LOW	FPo LOW	FPo HIGH	PMAX HIGH	PMAX LOW	∝Po EARLY	∝Po LATE	Injection period LONG	Injection period SHORT	Reflected pressure wave DAMPED
Worn fuel pump internals	■				■		■		■	■
Leaking valve on fuel pump	■				■		■		■	■
Fuel valve spring set too HIGH			■				■		■	
Fuel valve spring set too LOW or broken		■				■		■		
Fuel valve nozzle worn excessively or damaged					■					
Fuel pump index setting too HIGH				■				■		
Fuel pump index setting too LOW					■				■	
Choked fuel valve nozzle				■						
Fuel pump timing too ADVANCED (LEAD)						■		■		
Fuel pump timing too RETARDED (LEAD)							■		■	
Fuel viscosity too HIGH				■						
Fuel viscosity too LOW					■					

FIG. 8 Pre-performance check: fault-finding table

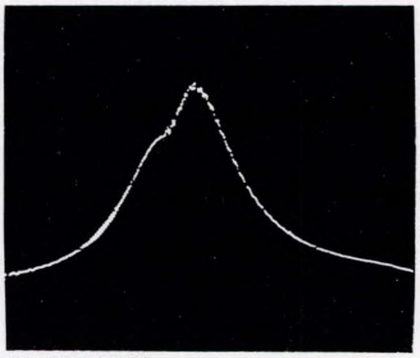


* While taking traces, the fuel system should be suitably 'locked' to remove governor influence and so achieve steady running conditions.
** It is *vital* that the values are dealt with in the sequence shown; for example, an unacceptable deviation in P_C may well be the reason for a deviation in P_{MI}.
*** Conversion of indicated kW to shaft kW. Carried out with the aid of a graph supplied in the engine data section.
**** Note limiting individual cylinder values of P_{MAX} and $P_{MAX} - P_C$ given in the engine data section.
***** Where faults are corrected immediately, two PMLS will be required—one showing the 'before-correction' data, the second showing the 'after correction' data.

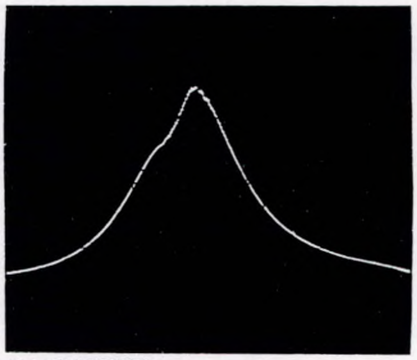
FIG. 9 Performance check ▶



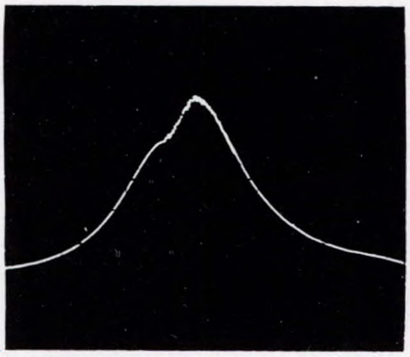
NO. 1 CYLINDER
P_{MI} 9.00 kg/cm^2
P_{MAX} 57.30 kg/cm^2
P_C 44.3 kg/cm^2
αP_{MAX} 15.6 deg
P_{EXP} 37.8 kg/cm^2
T_{EXH} 449°C
P_{SC} 0.65 kg/cm^2
N_{ME} 117 rev/min
Power 1184 kW
FPI 44



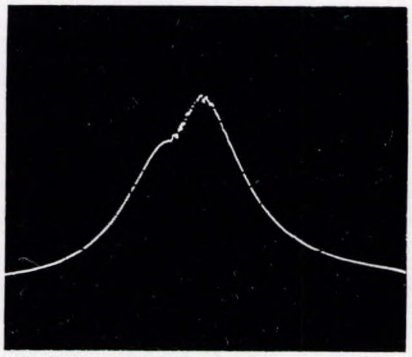
NO. 2 CYLINDER
P_{MI} 8.38 kg/cm^2
P_{MAX} 60.00 kg/cm^2
P_C 46.0 kg/cm^2
αP_{MAX} 16.5 deg
P_{EXP} 38.5 kg/cm^2
T_{EXH} 435°C
P_{SC} 0.65 kg/cm^2
N_{ME} 117 rev/min
Power 1103 kW
FPI 36



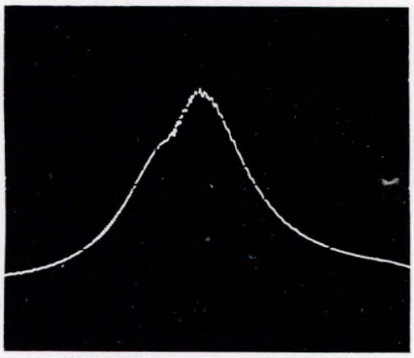
NO. 3 CYLINDER
P_{MI} 9.00 kg/cm^2
P_{MAX} 61.00 kg/cm^2
P_C 45.7 kg/cm^2
αP_{MAX} 15.8 deg
P_{EXP} 38.7 kg/cm^2
T_{EXH} 425°C
P_{SC} 0.65 kg/cm^2
N_{ME} 117 rev/min
Power 1184 kW
FPI 44



NO. 4 CYLINDER
P_{MI} 8.45 kg/cm^2
P_{MAX} 60.50 kg/cm^2
P_C 45.5 kg/cm^2
αP_{MAX} 15.3 deg
P_{EXP} 36.80 kg/cm^2
T_{EXH} 431°C
P_{SC} 0.65 kg/cm^2
N_{ME} 117 rev/min
Power 1112 kW
FPI 39



NO. 5 CYLINDER
P_{MI} 8.75 kg/cm^2
P_{MAX} 62.50 kg/cm^2
P_C 46.8 kg/cm^2
αP_{MAX} 15.8 deg
P_{EXP} 38.20 kg/cm^2
T_{EXH} 444°C
P_{SC} 0.65 kg/cm^2
N_{ME} 117 rev/min
Power 1151 kW
FPI 42



NO. 6 CYLINDER
P_{MI} 8.45 kg/cm^2
P_{MAX} 60.40 kg/cm^2
P_C 44.6 kg/cm^2
αP_{MAX} 14.4 deg
P_{EXP} 36.20 kg/cm^2
T_{EXH} 449°C
P_{SC} 0.65 kg/cm^2
N_{ME} 117 rev/min
Power 1112 kW
FPI 44

▲ **FIG. 10 Polaroid photographs of combustion traces**

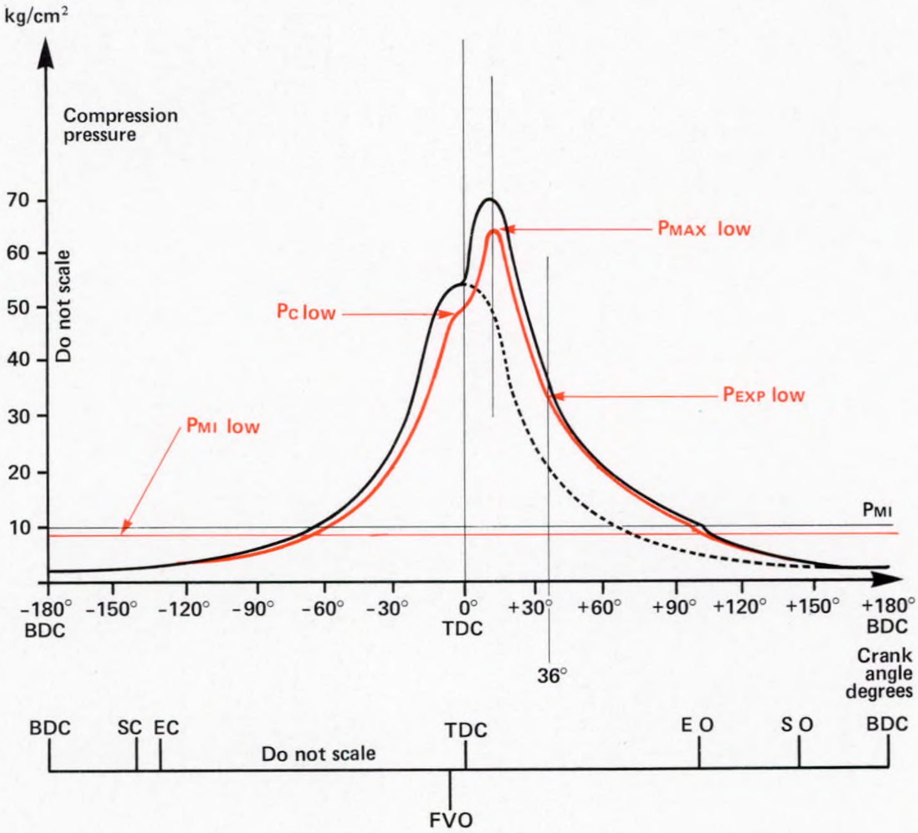


FIG. 11 Effects on pressure curve of: worn or defective piston rings; burnt piston crown; worn liner

INDICATIONS
Compression pressure (P_C) low.
Maximum pressure (P_{MAX}) low.
Expansion pressure (P_{EXP}) low.
Mean indicated pressure (P_{MI}) low.

FIG. 12 Performance monitoring log-sheet

Ship Name	Date	Log Sheet Number	Hours Run: Since Last Log Sheet	Hours Run: Since Last Dry Dock	Governor: Index	Governor: Speed Signal	Draught: FOR'D	Draught: AFT	Sea Temp.	Sea Condition		
M.V. EXAMPLE	19-11-80	12	205	5323	6.8	CONTROL GAGGED	4.2 M	7.8 M	22	1	2	2

Cylinder Pressure Trace

CYL No	1	2	3	4	5	6	7	8	9	10	Allowable Deviation	Mean
PMAX-PC	12.5	13.7	15.2	13.9	15.2	11.1					Max Value 17	13.6
PC	44.1	46.9	45.8	46.3	45.2	47.2					±1	45.92
α PMAX	15.8	16.5	15.8	15.6	16	14.1					±0.5	15.6
PMAX	56.6	60.6	61.0	60.2	60.4	58.3					±2	59.5
PEXP	37.3	37.4	37.9	36.8	36.9	36.3					±1	37.1
FPI	39.5	39	38.5	38	39.5	38.5					±1	38.83
TEXH	437	446	425	418	469	483					±20	446
PMI	8.5	8.17	8.84	7.93	8.78	7.9					±0.5	8.35
IkW	1113	1071	1158	1040	1150	1036						
TOTAL	IkW			6568	→ FROM GRAPH TOTAL SkW							5920
PSC	0.65		CYLINDER CONSTANT = 1.1253								NME	116.5

Fuel Pressure Trace

CYL	1	2	3	4	5	6	7	8	9	10	Norm	Limits
FPO	381	354	340	376	348	367					350	±10
FPMAX	630	634	663	646	560	683					650	±30
αPO	-1.6	-0.5	-2.1	-1.3	-1.0	-2.2					-4°	
ΔP/Δα	90	91	89	88	70	88						
Injection Period	19.2	18.7	18.5	18.5	19.5	19.0						

Engine Data

CYL	1	2	3	4	5	6	7	8	9	10
Fuel Pump Lead	19	17.5	21	18	19.5	20				
Fuel Cam Lead	21	18.5	23	20	21	21.5				
Exh Cam Lead	18.8°	18.7°	19.6°	18.5°	18.3°	18.3°				
FPI at Stop	5	-3.5	5	0	2.5	7				
FPI Read	44.5	35.5	43.5	38	42	45.5				

Fuel Consumption Test

	Fuel Meter Reading	Time
Finish	41550.6	13.00
Start	40000.0	12.00

Fuel System

Surcharge Pressure	2.5
Viscotherm cSt	12
Fuel Temp at Meter	55

Intermediate Values

PCA = PC + 1	46.92
TEXHC = TEXH + TCORR	438
N · FPI / SkW	0.764
TCORR = (45 - TSC) × 2	-8
TSC	49
PSC kp/cm²	0.65
PSC mmHg	478.3
PSCA = PSC + 760	1238.3
TAMB D	49
TAMB W	
ΔPWHU	55
PIWHU	
TTBI	49

		Turbo Blower No. 1	Turbo Blower No. 2	Turbo Blower No. 3
ΔPAF		70	50	
ΔPAC		190	175	
NTB		9100	9200	
PTB mmHg	$\frac{\Delta PAC}{13.6} + PSC$	492.3	491.2	

Trend Graphs

Ordinate	Model Value: Equation	Model Value: Value	Deviation % *	Graph No.
PCA	0.04 PSCA +3.1	52.63	-10.8	1
PMAX	3.147 PMI +38.6	64.9	-8.3	2
FPI	4.2 PMI -1.4	33.67	+15.3	3
TEXHC	0.0224 SkW +230	362.6	+20.8	4
N × FPI / SkW =	A constant between 70-100% power	0.655	+16.6	5
From SFC calculation sheet				6
NME	6.36 PMI +57.6	110.7	+5.24	7
PSC	125 PMI -630.8	412.9	+15.8	8
ΔPWHU	0.024 SkW -92	50.1	+9.8	9
ΔPAF	0.0667 PSC +16.67	48.6	+44.1	10.1
ΔPAC	0.24 PSC +24	139	+36.7	11.1
NTB	7.5 PTB + 4987	8679	+4.8	12.1
ΔPAF	0.0667 PSC +16.67	48.6	+2.9	10.2
ΔPAC	0.24 PSC +24	139	+26	11.2
NTB	7.5 PTB + 4987	8671	+6.1	12.2
ΔPAF	PSC			10.3
ΔPAC	PSC			11.3
NTB	PTB			12.3

Base Parameters from heavy lined boxes

Anticipated Maintenance

Fuel Valves:
NOS. 2 & 3 TO CHANGE
NO. 1 CHANGED SINCE LAST LOG
NOZZLE HAD COLLAPSED

Fuel Pumps:
ALTHOUGH NOT ADJUSTED
NO. 1 INDEX IS UP 2.5 SINCE
LAST READINGS

Pump Leads:
CHANGE NO. 5 FUEL PUMP

Cam Leads:

Waste Heat Unit:
WATER WASH LAST PORT SHOWS
DROP IN ΔPWHU

Turbo-Blowers:

Air Coolers:
REDUCE CLEANING INTERVAL
NO. 1

Air Filters:
CHANGE FWD AIR FILTERS

Other Remarks:
INVESTIGATE LOW PC NO. 1

Signature of Chief Engineer
S.S.B. Blow

All temperatures in degrees centigrade

Air/gas-circuit pressures in mmH$_2$O unless otherwise indicated

Cylinder pressures and fuel-system pressures in kp/cm^2 (kg/cm^2)

$$\text{* \%Deviation} = \frac{\text{Actual value} - \text{Model value}}{\text{Model value}} \times 100$$

adjusted accordingly. However, if engine balance is poor at this stage, care should be taken to ensure that individual cylinders are not overloaded.

Optimization of performance

Although perfect engine balance may have been achieved during the pre-performance and performance checks, it is unlikely that the engine will be operating at the optimum specific fuel consumption. For example, the maximum pressures in all cylinders may be low, due to a restriction of the air flow or a retardation of the camshaft. Optimization commences by calculating, for each performance characteristic, a percentage deviation from the relevant model curve.

These calculations are recorded in space provided on the PMLS (Fig. 12), which is also the central recording document for all numerical information obtained during the monitoring process. An example is shown below.

Mean

P_C = $\boxed{45.92}$ $\qquad$ P_{SC} (mmHg) = $\boxed{478.3}$

$\boxed{P_{CA} = P_C + 1 = 46.92}$ $\qquad$ $\boxed{P_{SCA} = P_{SC} + 760 = 1238.3}$

$\boxed{P_{CA} = 0.04\ P_{SC} + 3.1}$

$$= (0.04 \times 1238.3) + 3.1$$
$$= 52.63$$

$$\%\ \text{Deviation} = \frac{\text{Actual} - \text{Model}}{\text{Model}} \times 100$$

$$= \frac{46.92 - 52.63}{52.63} \times 100$$

$$\text{Deviation} = -10.8\%$$

All model-curve equations, with the exception of that for specific fuel consumption, are straight lines of the form:

$$y = mx + C$$

Calculation of the specific fuel consumption and any deviation from the model value are of major importance. Figure 13 shows the method, which is described below. Volumetric flow from the fuel meter is converted to mass flow by using the specific gravity of the fuel, corrected for temperature at the fuel meter. An initial value of SFC is obtained by dividing the mass flow rate by the engine SkW. This initial value is then corrected for three significant factors:

- The difference between the actual scavenge air temperature and the system standard of 45°C.
- The difference between the actual turbo-blower air inlet temperature and the system standard of 27°C.
- The net specific energy of the fuel.

This last item is based on the following empirical formula:

Net specific energy (MJ/kg)

$$= [46.392 - 8.792\rho^2 + 3.187\rho][1 - (x + y + s)] + 9.4205s - 2.449x$$

(Reference: BS.MA100:1982 'Specification for Petroleum Fuels for Marine Oil Engines and Boilers'.)

where

ρ = density at 15°C.
x = proportion by mass of water (% divided by 100).
y = proportion by mass of ash (% divided by 100).
s = proportion by mass of sulphur (% divided by 100).

Use of this formula is limited to where a complete analysis of fuel is obtainable at the time of use in the engine. Since this information is rarely available, a degree of simplification was essential. This was achieved by studying the analyses of approximately 1000 bunker samples and drawing the following conclusions:

- Ash and water content had relatively little effect on specific energy.
- A large proportion of the fuels had a sulphur content in the order of 3.3%.

The BS.MA100:1982 shows, in graphical form, for fuels of various net specific energies, the relationship between density and sulphur content. From this graph, it was possible to deduce, for a sulphur content of 3.3%, a relationship between specific energy and density. This had the following straight-line form:

$$\text{Specific energy} = 52.981 - (13.35\rho)$$

The fully corrected SFC value is then compared with the model value and the percentage deviation is calculated.

Another useful method of monitoring specific fuel consumption is by calculating the '*fuel consumption index*'. Since the quantity of fuel passing to the engine is proportional to the fuel-pump index (FPI) and the engine speed (N_{ME}), then their product, divided by shaft kilowatts (SkW), will bear a direct relationship to the specific fuel consumption, i.e.:

$$\text{Fuel consumption index} = (N_{ME} \times \text{FPI})/\text{SkW}$$

It should be noted that the index will tend to increase slightly with fuel-pump wear, but this is a long-term effect. A trend graph of the index is useful but, more important, calculation of the index immediately before and after adjustments to the main engine will clearly show whether the specific fuel consumption has changed.

The optimization process is completed by reference to a matrix (Fig. 14) showing, for high positive or negative deviations, the faults present in the engine's overall condition. Deviations are then transferred to the long-term trend graphs (Fig. 15), which are designed to cover approximately two years' operation of the engine.

Apart from their major function as the basis for optimizing engine performance, the concise nature of the PMLS (containing all numerical data, fuel-system settings and recommended maintenance work) and the trend graphs means that they have additional uses:

- As a permanent on-board record showing main-engine performance, maintenance history and fuel-system settings.
- As an aid to 'handing over' the vessel to relieving engineers.
- As a quick reference for visiting Superintendents to gain a rapid assessment of engine performance and trends.
- As a reference, either complete or in an abridged form, in Head Office in case of emergency.

Figure 16 shows a complete flow chart for the 'pre-performance check', 'performance check' and 'optimization of performance'.

IN-SERVICE RESULTS

Evaluation

The MIP calculator was evaluated in one ship over two years, during the latter half of which the performance monitoring system was developed and subsequently installed on the same vessel. Installation in a further eight ships has so far enabled the accumulation of a total 12 ship-years of operating experience with the calculator and the performance monitoring system.

The first readings from each engine indicated a clear pattern of common faults, which can be summarized as follows:

- Compression pressure was marginally low, due to normal wear and tear on the piston and liner parts.
- Injection timing was retarded by up to 3 deg, resulting from:
  - — incorrect cam adjustments (improper down-rating);
  - — incorrect fuel-pump adjustments;
  - — high fuel-valve opening pressures;
  - — low fuel-pump discharge pressures.
- Low rises in cylinder pressure (as a direct consequence of retarded injection timing) resulted in low maximum cylinder pressures.
- Fouling of air coolers, air filters and waste heat systems was higher than expected.
- Cylinder power output was, generally, imbalanced and well outside the normal tolerances.
- Specific fuel consumptions were in the order of 4 to 8% too high.

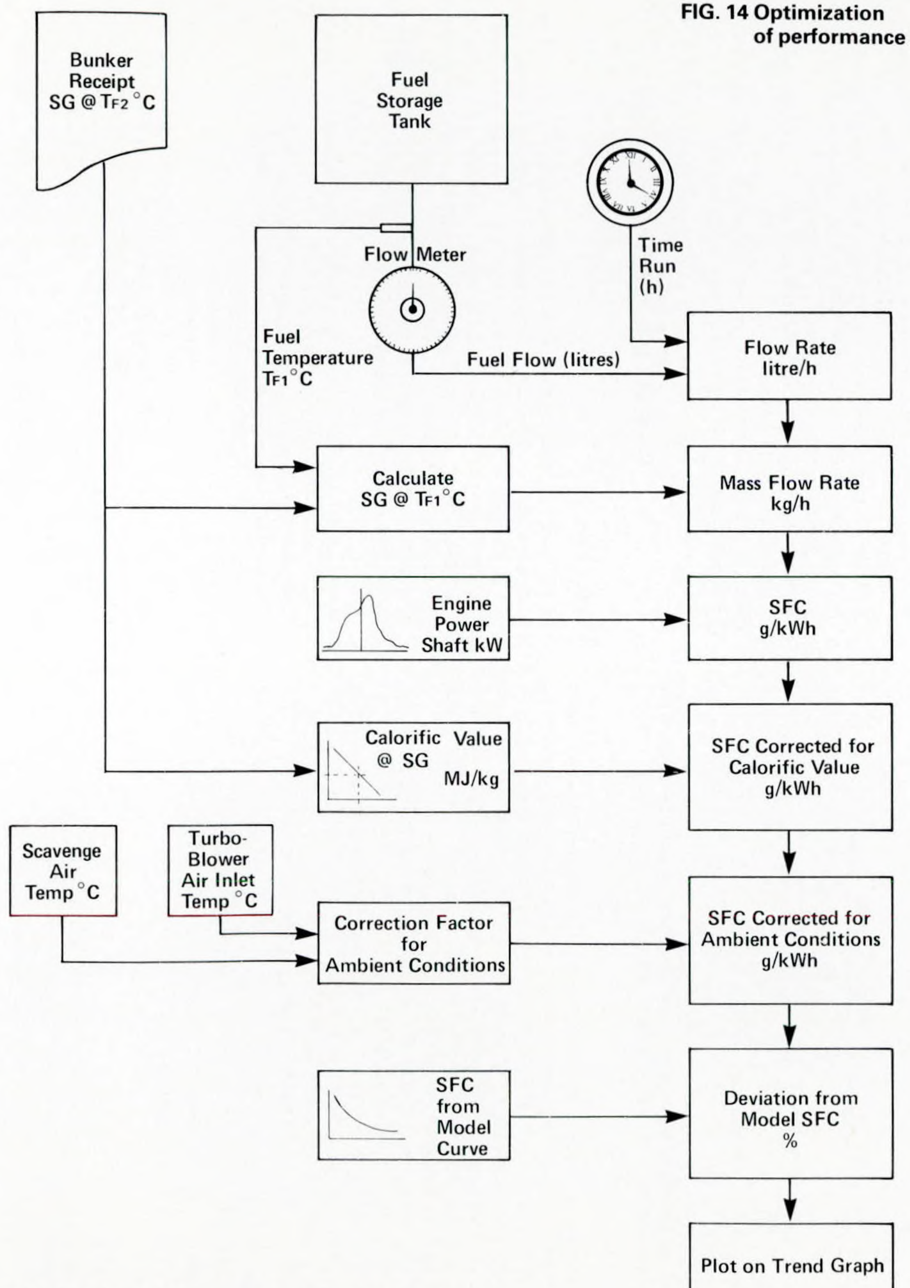


FIG. 13 Calculation of specific fuel consumption

FIG. 14 Optimization of performance ▶

CURVES	1	2	3	4	5	6	7	8	9	10	11	12	
BASE	P_{SCA}	P_{MI}	P_{MI}	SkW	SkW	SkW	P_{MI}	P_{MI}	SkW	P_{SC}	P_{SC}	P_{TB}	
ORDINATE	P_{CA}	P_{MAX}	FPI	T_{EXHC}	$\frac{N \times FPI}{SkW}$	SFC	N_{ME}	P_{SC}	ΔP_{WHU}	ΔP_{AF}	ΔP_{AC}	N_{TB}	
Defective: ExhV/Vs, Sulzer reed V/Vs, piston rings and crown. Worn liner. Choked scavenge ports.	−			+									
Decreased ship resistance due to reduced draught.		−					+						
Increased ship resistance due to fouled hull and/or damaged propeller.		+					−						
Dirty or damaged turbine/compressor.		−		+				−		−			
Coking in turbine nozzle ring.								+					
Fuel pump plunger worn. Leaky suction valve.			+										
Air cooler fouled on water side.				+									
Air cooler fouled on air side.		−		+							+	+	
Air filter fouled.		−		+						+	−	+	
Retarded fuel injection timing.		−											
Advanced fuel injection timing.		+											
Waste heat unit fouled.									+			+	
Retarded exhaust valve cam timing.								−					

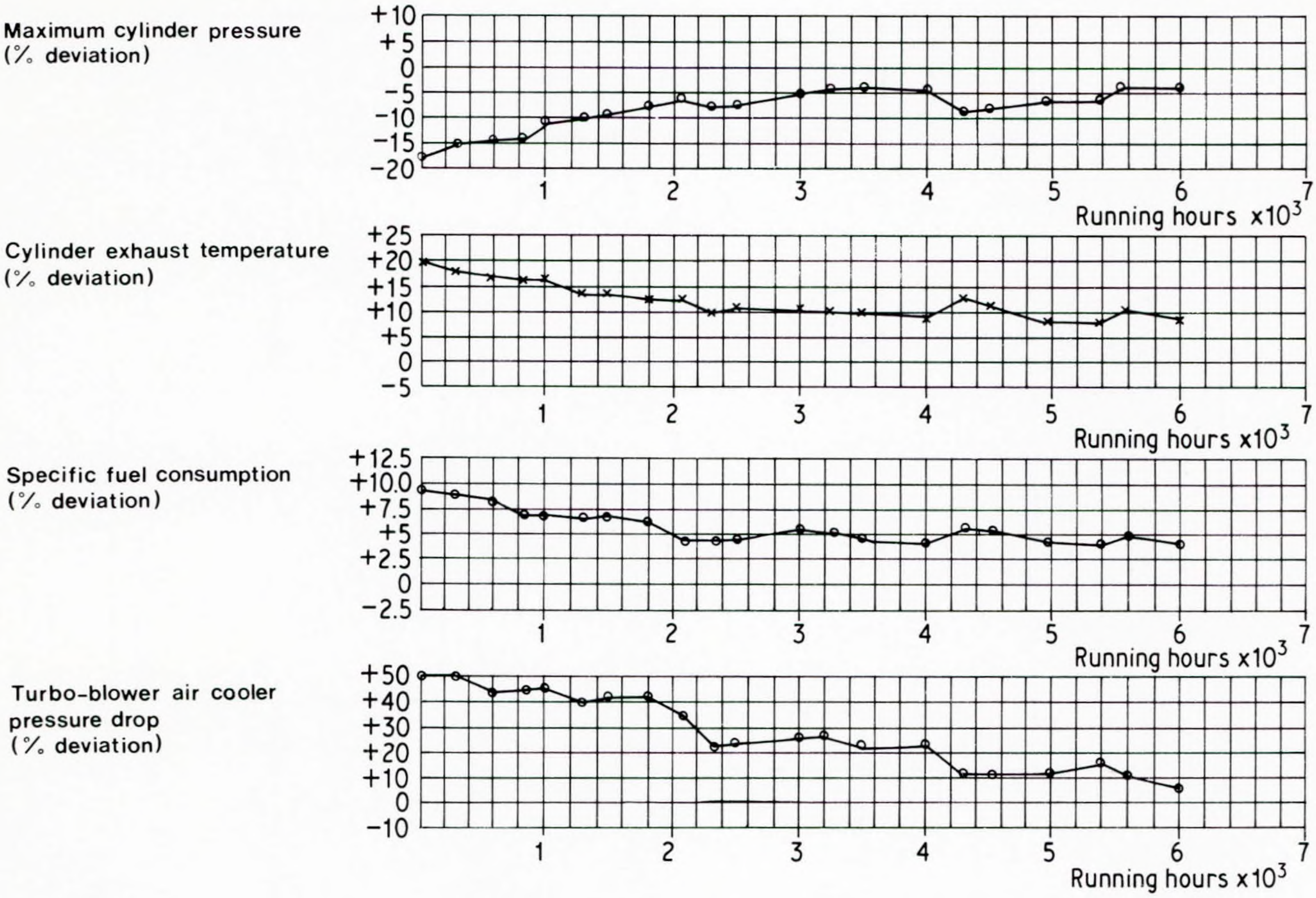


Fig. 15 Trend graphs

Typical results

Prior to the installation of the performance monitoring system, the engines, without exception, were considered to be operating well. Routine planned maintenance, including fuel-valve changes, piston-ring inspections, etc., were up to date.

Table I shows the percentage deviations from the model-curve operating points for various parameters measured on two vessels immediately after the performance monitoring system had been installed.

Improvements

Table II shows in-service results from a typical engine over a 2000-hour monitoring period and gives a clear indication of the improvements in performance.

Three monitoring times are shown in Table II—at 0 (installation of the system), 296 and 2096 hours respectively. In practice, between 10 and 12 log sheets would be completed during this time and, at each stage in the process, data would be plotted on the trend graphs to show the overall improvement in engine condition.

Table I: Percentage deviations from model-curve operating points immediately after installation of PMS

PARAMETER	PERCENTAGE DEVIATION	
	Vessel 1	*Vessel 2*
Compression pressure	−5.5	−9.0
Maximum cylinder pressure	−8.7	−10.3
Fuel pump index	+16.0	+15.1
Exhaust temperature	+11.3	+8.1
Fuel consumption index	+17.4	+16.9
Specific fuel consumption	+9.6	+9.4
Engine revolutions	+0.4	−2.9
Scavenge pressure	−4.4	−7.3
Pressure drop: waste heat unit	+98.0	+114.0
Forward air filter pressure drop	+65.0	+41.0
Aft air filter pressure drop	+38.0	+112.0
Forward air cooler pressure drop	+48.0	+17.7
Aft air cooler pressure drop	+85.0	+43.9
Forward turbo-blower revolutions	+3.2	+2.1
Aft turbo-blower revolutions	+2.8	+1.3

Table II: Percentage deviations from model-curve operating points up to 2096 hours after installation of PMS

PARAMETER	PERCENTAGE DEVIATION		
	0 hours	*296 hours*	*2096 hours*
Compression pressure	−12.0	−11.5	−9.1
Maximum cylinder pressure	−18.0	−17.2	−6.7
Fuel pump index	+8.0	+6.9	+5.0
Exhaust temperature	+21.0	+20.0	+13.7
Fuel consumption index	+11.0	+9.2	+5.4
Specific fuel consumption	+5.7	+4.8	+2.8
Engine revolutions	+1.0	+0.2	−1.3
Scavenge pressure	+2.0	+1.4	−6.7
Pressure drop: waste heat unit	+30.0	+29.0	+18.1
Forward air filter pressure drop	+135.0	+129.0	+60.0
Aft air filter pressure drop	+60.0	+40.0	−11.2
Forward air cooler pressure drop	+115.0	+105.0	+40.0
Aft air cooler pressure drop	+40.0	+36.0	+16.0
Forward turbo-blower revolutions	+5.5	+5.2	+1.9
Aft turbo-blower revolutions	+9.5	+9.3	+7.8

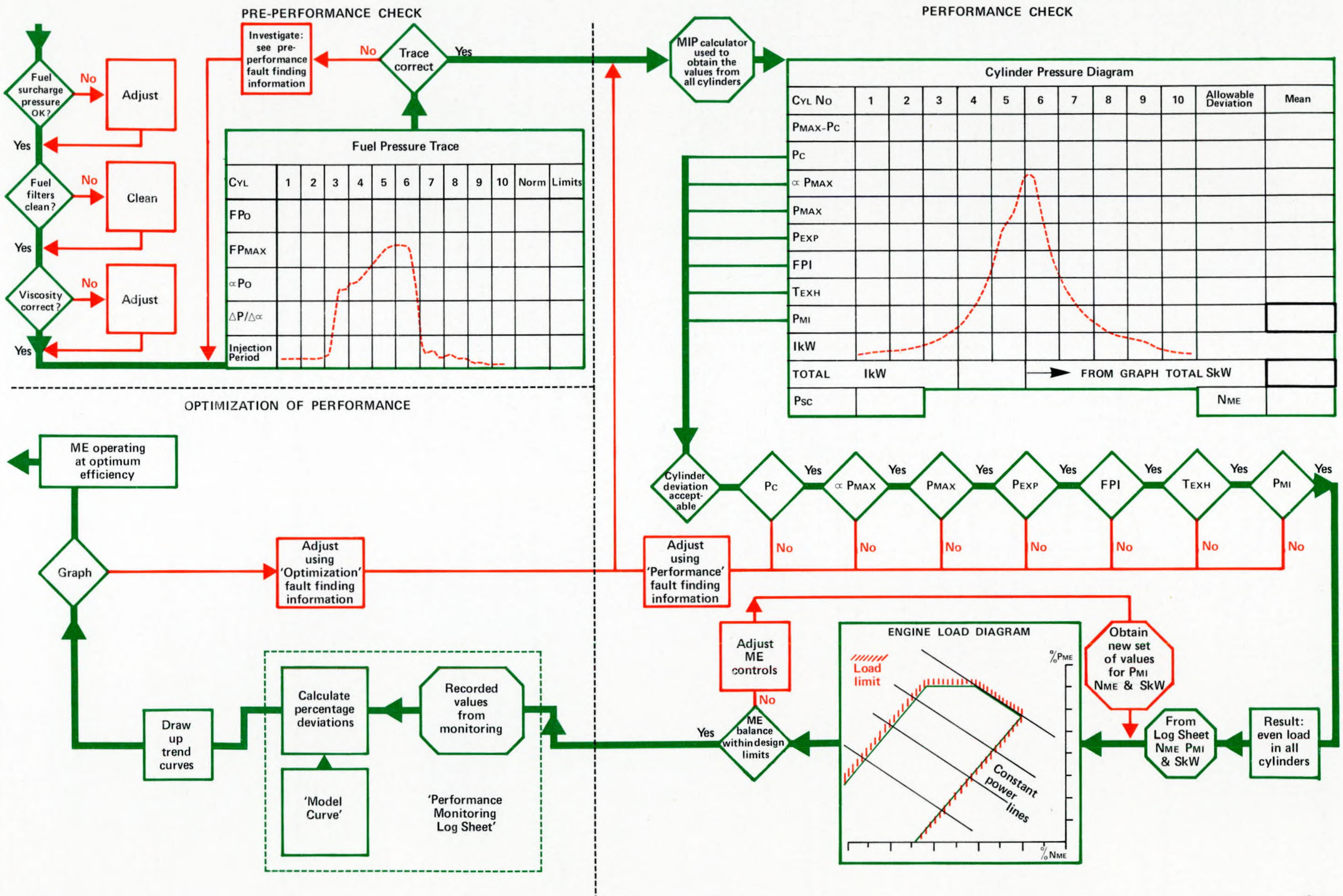


FIG. 16 Fuel system/engine monitoring diagram

Presentation of results

The final stage in an energy-conservation system is presentation of the results in a clear, concise manner, to both technical and commercial management.

For this purpose, graphs can be constructed for each of the model-curve relationships to show the comparative positions of:

- The performance monitoring system model curve;
- The condition of the engine on the trial trip;
- The condition of the engine at the time when the performance monitoring system was installed;
- The condition of the engine after the performance monitoring system had been in use for, say, 2000 hours.

Figure 17 shows these comparative positions for the relationship between maximum cylinder pressure and mean indicated pressure. The normal operating point is indicated by a vertical line, the distance between intersections being proportional to the percentage improvements of that particular ordinate (P_{MAX}).

Figure 18 shows a graph of specific fuel consumption to a base of shaft kilowatts. The approximate yearly savings to be made by using the PM system can be calculated as shown, provided the running hours and average power are known.

CONCLUSIONS

Use of the performance monitoring system has shown that the efficiency of operation of marine diesel engines can be improved, thereby making a worthwhile contribution to an energy-conservation programme.

Initial investigations carried out on a class of vessel fitted with identical engines showed quite clearly that, even immediately after delivery, the engines did not perform to a common standard. It is concluded that this was mainly due to the following factors:

- Insufficient emphasis was placed on proper engine down-rating at the time of building and trials (before the energy crisis), resulting in low thermal efficiency.
- Monitoring equipment available at that time was inadequate to give accurate, repeatable results.
- No proper adjustments were made to engine timing to suit operation on heavy oil after trials on diesel.
- No corrections were made on a consistent basis to allow for variations in fuel specific energy.

Production of a relatively small number of model curves has shown that engine performance can be monitored successfully by comparison with a known standard on a time base (trend graphs).

This graphic approach proved worthwhile, giving ships' staff a sense of achievement and a rapid system for handing over. Visiting Superintendents could also tell 'at a glance' whether their performance standards were being maintained.

The degree of accuracy of the system was considered satisfactory, taking into account:

- The individual accuracy of the system's components, including transducers, meters and electronics.
- The simplifications made by using approximations and the elimination of certain 'insignificant factors'.
- The restraints of a simple, clear systematic approach, with the minimum of technical obstacles for the operator to overcome.

This simplicity was achieved mainly by designing within the limits of Pareto's law; i.e. 20% of the total effort will realize approximately 80% of the possible fuel savings.

It has been concluded that results from the MIP calculator and oscilloscope output, when related to performance, reflect the condition of the engine's moving parts. Consequently, the results may be used in the future to regulate planned-maintenance schedules and, possibly, extend periods between overhaul.

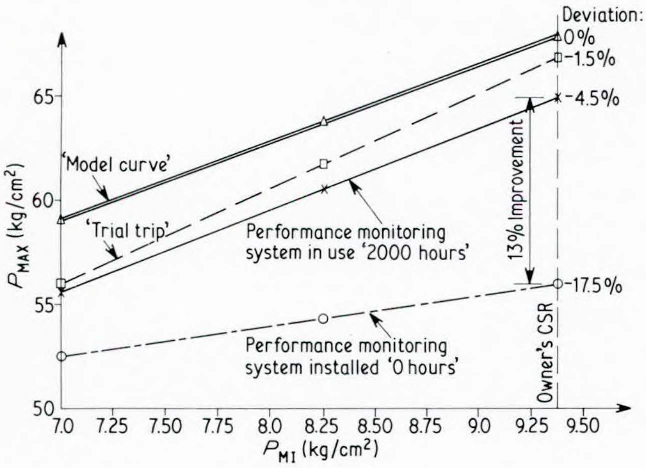


FIG. 17 Improvement in performance (P_{MAX})

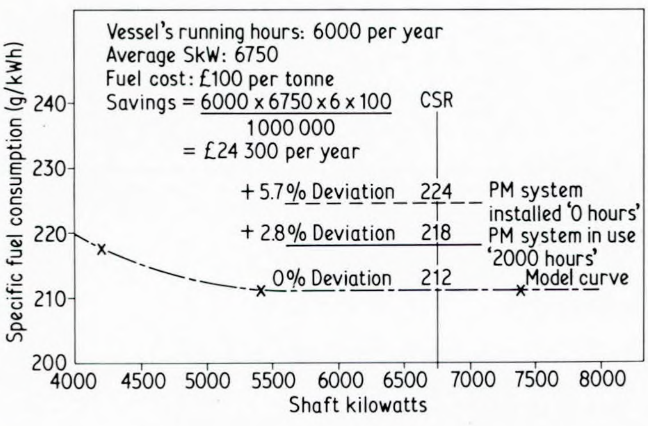


FIG. 18 Improvements in performance (SFC)

Shipboard computers

Current developments are concentrated upon the use of a shipboard computer to produce model curves from trial data and calculate deviations from the model curves. Eventually, storage of the complete performance monitoring log sheet in the computer is envisaged. A direct link between engine instrumentation and shipboard computer is considered to be impracticable, since some degree of 'filtering' is required. Ideally, the ships' engineering staff should vet data being input to the system, in order to eliminate results which are suspect due to adverse weather, unusual trim and intermittent engine faults such as choked indicator cocks, etc.

Finally, the author believes that it is possible to develop the applications of a shipboard computer to assist in the fault-finding process. Figure 19 shows a schematic diagram representing this hypothesis. The engine transducers are coupled to the MIP calculator in the normal way, the output being displayed on a VDU. This gives the operator the opportunity to accept data using the keyboard, through which he may input additional information. Upon acceptance, the data are analysed by the fault-analysis computer, which holds software describing the:

- Trend graph equations;
- Engine specification;
- Mathematical model of the engine, including the effects on performance of outside influences such as ambient temperature, ambient pressure, scavenge air temperature, fuel specific energy value, etc. (see Table III).

Output from the computer could be displayed (as with the existing system) in terms of percentage deviations, trend graphs and specific fuel consumption. However, using the software already described, this output could then be enhanced by a detailed analysis of the specific fuel consumption error. Figure 19 also shows this proposed output and gives a hypothetical breakdown of the excess fuel being used by the engine, in terms of the associated fault conditions. A manual analysis of these data, coupled with the following information:

the vessel's operating schedule;
the time required to overhaul various components;
the cost of spare parts required;
the current fuel costs;
the penalty incurred by delaying the vessel;

will enable the operator to take the most cost-effective corrective actions.

For example, the output may indicate that the savings to be gained by a simple adjustment in the fuel system could well be greater than the time-consuming and costly replacement of a large component.

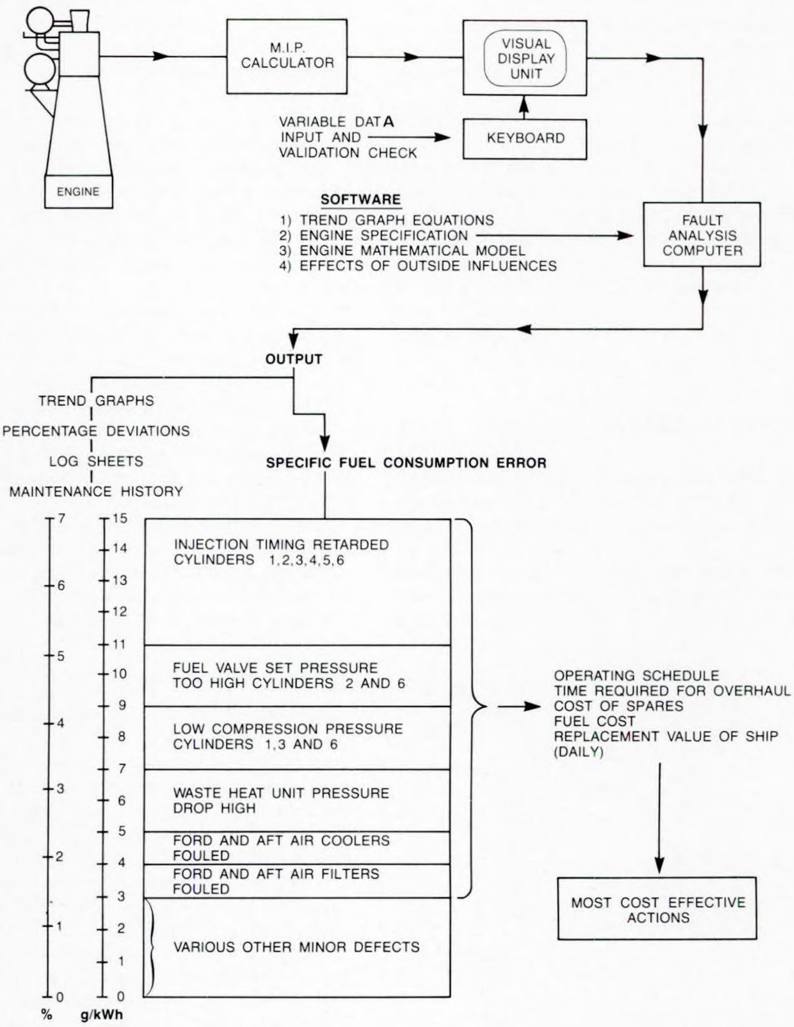


FIG. 19 Schematic diagram of possible future fault-finding system

Table III: Selection of outside influences and their approximate effects on engine performance

	CHANGE IN CONDITIONS FROM STANDARD[a]			
PERFORMANCE PARAMETER	*Ambient temperature +10°C*[b]	*Ambient pressure +10 mmHg*[b]	*Scavenge air temperature +10°C*[b]	*Net specific energy +1%*[b]
Specific fuel consumption	+0.31%	−0.06%	+0.91%	−1.0%
Air mass flow to engine	+0.31%	+0.18%	+0.36%	—
Scavenge pressure (absolute)	−2.03%	+0.5%	+1.45%	—
Exhaust temperature after turbine (absolute)	+1.91%	+0.04%	+1.35%	—
Cylinder maximum pressure (absolute)	−1.37%	+0.3%	+0.27%	—

[a] Standard conditions, used as reference values, are as follows:
Ambient temperature: 27°C.
Ambient pressure: 750 mmHg.
Scavenge air temperature: 45°C.
Net specific energy: 42.0 MJ/kg.

[b] Opposite signs are applicable in all cases. For example, if the ambient temperature is 17°C (i.e. 10°C below the reference value), the specific fuel consumption will be reduced by 0.31%.

ACKNOWLEDGEMENTS

The author gratefully acknowledges the assistance of his many colleagues in the BP Group who were involved in the development of the system, its evaluation at sea and the subsequent preparation of this document. Finally, he wishes to thank the Group for their agreement to the publication of this paper.

Discussion

G. S. MOLE (M.A.N.-GHH (Great Britain) Limited): The overall benefits gained from performance-monitoring of the diesel engine are savings in fuel costs, resulting from the running at the best efficiency for any operating condition. Further benefits may be gained from reducing the strain on a number of engine components by the balancing of the output from the individual cylinders, which should reduce the risk of major failures caused by maloperation and provide improvements in the average wear rates.

The fuel-cost saving given in the case of the vessel analysed in Fig. 18 is £24 300 per annum. Can the author relate this saving to the additional cost of maintenance and replacement components which had to be expended to achieve the optimum engine-performance level?

In addition, can Mr Warkman give an assessment of the age of a vessel to which this type of performance-monitoring system could be retrofitted? Obviously, the greatest benefit will be gained by fitting to new vessels, as older ships with worn components could involve the operator in considerable expenditure of time and cost for replacement components to restore the engine to an acceptable level of maintenance.

M. S. BRADLEY (Michael Bradley & Associates): Having been involved myself for some time in the field of voyage monitoring and ship performance measurement, rather than simply engine performance measurement as described by the author in his paper this evening, I should like to make the following comments and observations.

I cannot agree with the author's assumptions regarding the reasons for the observed spread in the engine shop and sea trial results. Even between sister ships, it is a commonly observed fact that considerable differences will arise in the results of comparative trials and acceptance tests. The reasons for these variations are many and varied, ranging from gas-flow variations along the scavenge and exhaust distribution manifolds of a particular engine to changes in hull resistance and weather loadings. The residual 'errors' remaining after correction of the data back to the standard ISO conditions cannot therefore be attributed, either solely or in the main, to the variations in particular engine-timing arrangements or fuel-system components as Mr Warkman proposes. More properly, the deviations arising in the measured values can be allocated, in the case of a motor ship, to the variations which will exist between the cylinders in the same engine and to variations arising between each particular engine installation and trial performed.

In practice, this scatter in the data measurements represents an unpredictable variation about the expected value; or, in control engineering terms, a noise source. The problem with which we are thus faced in any ship-monitoring programme is the analysis of noisy signals from a system whose transfer function represents the performance characteristics we are seeking to identify. If this transfer function is a constant then the system under investigation is termed 'linear' and if it varies, say with time, then it is termed 'non-linear'. A linear system can in general be analysed using the techniques obtained from classical control or transmission analysis. In the simplest cases, having constant coefficients and low associated noise levels, then these techniques may reduce to no more than averaging.

The ship and engine combination is obviously a non-linear case, since the performance coefficients are known to vary with time. Simple methods are neither accurate nor reliable when applied to the analysis of such systems and recourse must be made to the more sophisticated techniques of stochastic signal processing, using digital filters and system modelling. The difficulties are further augmented for the ship system where the variations in observed values over a period of time are very large. In some cases these variations can amount to more than 50% of the mean value, implying the existence of noise sources having very high energy values.

For a simple input/output system with added noise, the optimal estimate of the system response can be derived from the signal cross-correlation and auto-correlation functions. In the case of the ship engine system considered by the author, two signal noise sources exist. One of these sources may be termed 'plant noise' and is identified with variations arising between the cylinders of any particular engine; and the other may be termed 'system noise' and is associated with the variability arising because of the differing conditions in each installation. It is possible in principle, by analysing the data between and across the cylinders of a number of sister vessels, to identify the strength of each of these noise sources, and I wonder if the author has attempted to refine the data in this manner?

The advantage of this more complete and sophisticated aproach to the data analysis is that the distribution of variability within the system being examined is far more accurately known than before. As a result, more reliable estimates of the expected and achievable tolerance limits for each of the parameters being measured will be obtained. There is little point in trying to determine performance and efficiency to more accurate limits than can be perceived through the residual background noise levels, so that it becomes of crucial importance to achieve the lowest possible levels.

In particular, it must be pointed out that the 'model' or ideal performance characteristic curve for the system is the mean estimate of the system performance made by a filter after removing the noise signal and can thus be estimated at the onset of the performance-monitoring programme and at any other time subsequently. Its position is not fixed but will change with time as the conditions on board and around the ship vary, the certainty of being able to calculate its exact position at any time being associated with the background noise level.

The great advantage of the Kalman or adaptive type of filter is that it will always seek to determine the best value of performance coefficient at any time which satisfies the data observations made, taking into account the presence of noise. It will thus 'track' changes in performance and can therefore be used directly as a trend recorder and indicator. A further great advantage is that the 'cleaning up' of operational information through efficient modelling and filtering techniques considerably eases the problems of resource allocation at maintenance intervals and allows rational budgeting to take place by management. The method described, of positioning such a line above all the observations and having the same average slope, as shown in Fig. 1 of the paper, does not generate a line of optimum performance as is claimed and cannot be justified on the basis of mechanical wear or fouling of the manufacturer's test-bed engine.

It may be argued that the positioning of the model line is relatively unimportant, since all that it represents is a target, but this can lead to difficulties in practice since one is in the position then of presenting the ship operators with a goal which is forever denied to them no matter what their endeavours may be. This is obviously not a satisfactory recipe for maintaining interest and motivation in a project over long periods of time, which is one of the main requirements for a successful performance-monitoring system. The practice of allowing ships' staff to 'vet' data prior to the analysis is catastrophic under these (or any other) circumstances, since the natural bias will be to select only improving results at each exercise. This may in part explain the apparently good results so easily achieved.

Adjustments to the model curve in the light of service experience may go some way to correct this defect but what has now happened is that adaptive elements are introduced into the system in a non-optimal manner. Surely it must be the better and more realistic alternative to design the monitoring system to be adaptive in the first place as part of an overall integrated monitoring system?

During the course of a ship's operational lifetime it will be subjected to continuously varying weather conditions. Wind, wave and current forces affect the results of any engine-performance measurements since they alter the characteristics of the load dynamometer to which the engine is connected (the hull and propeller system). Recognizing this, therefore, it is possible to do one of two things. The first is to take measurements only during 'calm' weather conditions but this is less than satisfactory, since identical conditions can never be reproduced. The amount of data made available using such a system is severely limited, depending as it will on the combined vagaries of climate and opportunity. Since regions of good weather are in general associated at particular times of the year with specific regions, then there also arises the disadvantage that any such constraints on the data measurements will result in an emphasis being given to particular sections of the trading route and will thus work to the detriment of an overall picture of the vessel's performance.

As an alternative, measurements could be made over a much broader weather spectrum and the data corrected back to standard conditions prior to analysis. There are difficulties with this but we are still speaking about system behaviour, which is governed by the laws of physics. The adaptive filter can be used in these circumstances to track model hull/weather performance characteristics in much the same way as it can for the engine alone. In 1926, Professor E. V. Telfer proposed a system of monitoring based on plotting of the Admiralty Coefficient against broad sea-state codes, in an attempt to reduce the analysis problems raised by weather. Unfortunately, the Admiralty Coefficient is an unstable measure of ship performance for many reasons, so that little success was achieved. Has the author had any more success with his 'fuel consumption index' or any other simple Buckingham group?

In Fig. 14 of the paper I see that amongst the list of effects considered in the optimization of performance are hull and propeller fouling, the inference being that measurements on the engine will somehow uniquely quantify these effects. I presume that this interaction matrix is in fact a diagrammatic illustration only, since it contains non-unique solutions. For example, the diagram shows that the combination of retarded exhaust-valve timing and coking in the turbine nozzle ring will result in no measurable effect on the engine, the positive and negative results of the effects described cancelling out.

Figure D1 shows the loading diagram for a diesel engine, on which has been superimposed the propeller law absorption curves corresponding to various hull resistance or fouling conditions. Considering an increase in hull fouling will result in the effects shown on Fig. D2. In this figure, the effects of an increase in hull or propeller resistance are shown as projections on to planes of constant shaft-rate (rev/min), brake mean effective presssure (BMEP) and shaft power (shp) and are derived directly from the last figure. Thus, for example, to maintain constant shp whilst increasing resistance will require an increase in BMEP and a reduction in rev/min.

In practice, constant rev/min, BMEP or shp are rarely obtained, actual effects being a combination of such elements of behaviour. This is depicted in Fig. D3, where the engine operating point is observed to move from point (0) to point (3) via orthogonal shifts (1) and (2). In principle, the order in which the elemental changes are made is unimportant since the diagram is linear, but there is no constraint on the vector lengths or directions within each plane needed to assemble the connection between (0) and (3). These can only be accurately determined from a knowledge of the model characteristics of the system at that particular time. It is these elemental changes in the constant property planes which can then be used for rational diagnostic purposes.

Finally, I wish to comment on the use of Pareto's so-called law. Cautions are often given but seldom heeded about the use of this alleged relationship between effort and achievement. In engineering terms, Pareto's law is a restatement of Fick's law of diffusion and, when described in these terms, its deficiencies become immediately apparent: it is nothing more than a probabilistic model, where the probability of succeeding at an opportunity is assumed to be proportional to the number of opportunities for success remaining. It is a relevant description for simple low-technology occupations such as clearing stones from fields by using manual labour—initially production is very high but falls off as the distance between stones yet to be discovered and removed increases—but has nothing to do whatsoever with modern technology.

Today, significant investments or resources may be required in a project before any benefit at all is manifest, although the rate at which improvements are achieved later on may be many times greater. It is more meaningful to make use of management models based on the value of success as well as on the opportunity for success in these circumstances, and rational techniques have been established.

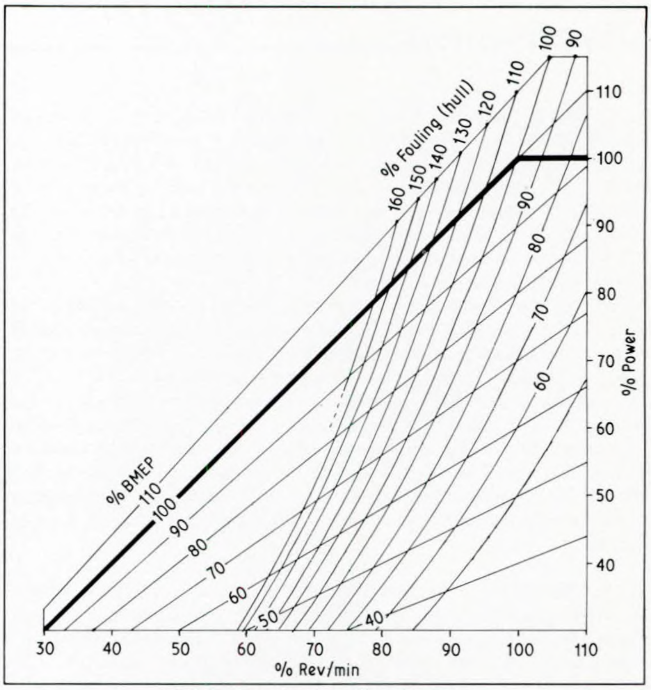


FIG. D1 Engine loading diagram

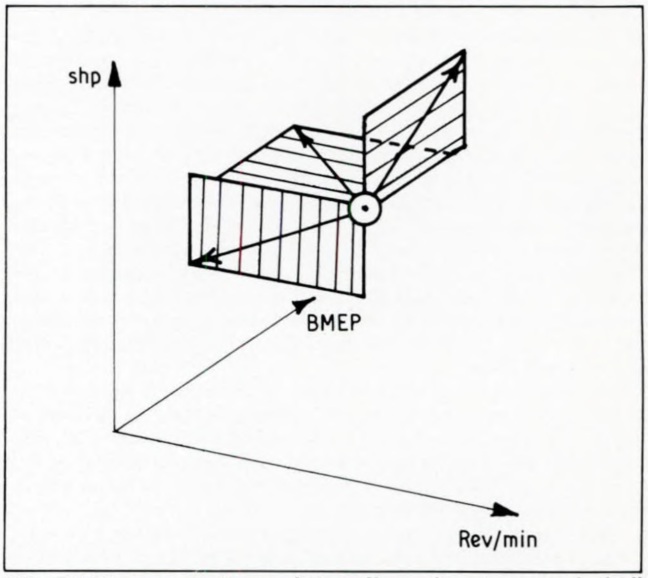


FIG. D2 Plane projections of the effect of an increase in hull resistance

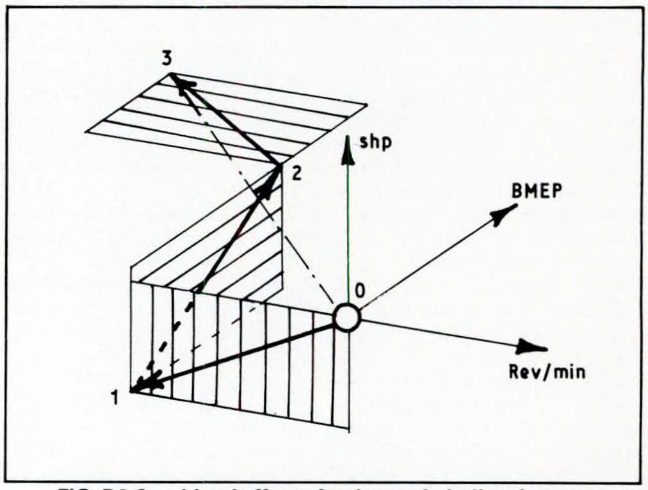


FIG. D3 Combined effect of a change in hull resistance

M. M. LYNG (Autronica Marine UK): In answer to a comment about other ways of achieving the same results with greater accuracy, I would comment that in all presentations there will always be someone with a better method of achieving the same result. In the marine field, however, it is more a question of other factors relating to the special problems existing for ships.

A supplier of marine equipment must always be aware of the fundamental problems onboard ships. Because of the crew, equipment must be easy to understand, use and install. Maintenance and repair work must be simple to perform and, preferably, be able to be done by the ships' crew. Reliable service facilities and depots of spares must be available in principal ports throughout the world. The PMS system together with the Autronica MIP calculator meets these requirements and presents the data in a form which is well understood by the ships' personnel.

There were also some comments about the absolute accuracy which could be achieved by the system. Even though we claim an overall accuracy of better than 2%, this is not of supreme importance. More important is that the reproducibility of the measured value is better than 0.5%. We are more interested in changes than absolute values.

R. C. HASLAM-JONES (Blue Star Line Limited): The paper has described the engineering application of a measuring system that has sometimes been considered as an 'extra' or 'add-on' to marine engines. Mr Warkman has demonstrated that it has become a standard item of equipment on the engine and a leading item in the appraisal of operating conditions.

There is a tendency to take this invasive technology to the point where the engine is almost inaccessible to the engineer for maintenance until the probes and sensors have been removed. Does this situation seem imminent to the author; and how does he see the type of diagnosis described in his paper being extended to other parts of the engine without invasive technology?

With regard to the accuracy of the system there are a number of points that have always been a feature of this and similar MIP calculators. First, the accuracy of the pressure sensors is important to the end result. Do the ship operators test these sensors at regular intervals or are the manufacturer's figures for keeping accuracy accepted?

Second, the position of piston TDC does not relate to crankshaft TDC as indicated on the flywheel when the engine is under load. The setting-up procedures for this equipment allow some compensation for this, but a change of engine load requires resetting of the calculator TDC position. Has this been appreciated or are these errors accepted as a part of the system, for the sake of simplicity?

Finally, the author outlines some assessments made by measuring the rate of pressure rise in the fuel system. Does the response of the sensor in this system keep within the actual pressure response or does the sensor response cause false readings to be made?

The results of this deductive process will, hopefully, filter out any changes caused by operator error but there are changes which have to be identified clearly for their cause. In particular I refer to the use of different grades of fuel and the occurrence of damage caused by a poor-quality fuel. Has the author any examples of results being submitted that gave warning of impending damage following a sudden change in the readings made? Have there been any circumstances where the results showed likely damage and the unit or engine was opened up, out of schedule, and damage was either found or not found?

The flow-chart analysing methods are a good way of engaging the interest of the operating staff. How does the author see this being carried over to a computer situation while keeping up the operator's interest?

The underlying reference to engine models is very interesting because it would appear to mark the next basis of condition monitoring. Would the author care to relate on any subsequent use of engine modelling and its particular application?

Finally, there has been considerable work in this field by manufacturers and operators alike, but I would refer the author to the work in Australia of Mr Nils Holgersson and the equipment being used on the BHP central power station. This is a computerized engine-management system which incorporates an on-line diesel engine tuning and analysing system of a similar type to that described in the author's paper.

Author's Reply

Mr Mole's question relates to cost of spare gear associated with improving performance and the age and condition of the particular vessel at the time of fitting the system.

As indicated in the paper, the Pareto principle was continually kept in mind during the development of the system's theoretical basis and naturally applies also to the actual operation of the system. Apart from material failure requiring immediate remedial action, the operator of the system should therefore concentrate on the faults in the engine systems giving the greatest return for the least outlay. My experience during evaluation of the MIP calculator can be used to illustrate this point quite well.

Fuel valve test equipment comprised the normal bench arrangement with high-pressure hand pump, pressure gauge and a fuel valve clamp. (Setting the valves using this arrangement can lead to errors in the static lift pressure, since the exact point of lift can be extremely difficult to judge by eye from the pressure gauge.) Readings taken from the engine (Table DI) proved this point, indicating that, on average, fuel valve settings were 57.5 kP/cm^2 high.

It was shown in practice that a change of ± 35 kP/cm^2 gave a corresponding change in engine timing of ± 0.5 deg.

From the fuel cam profile, a 1 deg advance was equivalent to a 1.5 mm increase in fuel cam lead or a 1.55 kP/cm^2 rise in the cylinder maximum pressure.

A 1 kP rise in maximum pressure reduces the SFC by 0.67 g/kWh. Therefore, by simply adjusting the fuel valve spring setting, a saving of $(57.5/35) \times 0.5 \times 1.55 \times 0.67 = 0.85$ g/kWh was made.

This would equate, over a year of 6000 running hours, average SkW of 6750 and fuel cost of £100/tonne, to:

$$(6000 \times 6750 \times 0.85 \times 100)/1\,000\,000 = £3442/\text{year}$$

In other words, for a few hours work nearly £3500/annum can be saved. Similar sums are easily recouped by a few hours work and very little spare gear outlay: for example, replacement of fuel pump suction valve plates, cleaning of scavenge spaces, air coolers and waste heat unit gas passages.

Moving to more costly replacement items, these generally tend to have a rapid payback in relation to their capital cost. A worn feed pump barrel/plunger can have a devastating effect on the injection efficiency of a particular cylinder. It must be borne in mind, however, that wear on engine parts can be related to fuel savings, as indicated in the system, and therefore an economic evaluation can be made before replacement is considered.

Therefore, as Mr Mole states in his question, the greatest benefit (over its life cycle) will be obtained by fitting the system to a new vessel. Naturally, considerable benefits are also available from a ship of any age where, in the case of badly maintained engines, the initial improvements will be considerable.

In conclusion, the system can and should be used to extend the periods between planned maintenance events by monitoring performance and only expending time and money when performance drops off. This in itself has the result of saving considerable man-hour effort by reducing maintenance man-hours.

I thank Mr Bradley for his very detailed and elaborate contribution to the paper. As Mr Bradley states, he has been involved in the field of voyage monitoring and ship performance monitoring for some time and it is obvious from his comments that he has applied fairly complex analysis methods.

Rather than enter into the many arguments put forward, I am of the opinion that two approaches exist for performance monitoring of marine diesel engines and both have points in their favour. The complex study has been tried many times with little success, mainly because of development costs and the difficulties in acceptance by ship's staff. The more simple approach, however, has been proved to work despite the many approximations, is accepted by seagoing staff and is producing the desired cost savings in terms of fuel and maintenance time.

I should like to thank Mr Lyng for his contribution and endorse his comments regarding the fundamental problems onboard ships.

The performance monitoring system meets the current needs in that it uses well-proven hardware, ideally suited to the marine environment, and the software is uncomplicated. More important, however, is that onboard management is possible, giving the Chief Engineer the relevant information to assess economically as well as practically his problem and make the most cost-effective decision. Equally, feedback to a shore-based establishment can be introduced when required in a case of unusual complexity or when continuous monitoring is required on individual vessels or across a class.

Repeatability of the equipment has proved to be well within the manufacturer's stated tolerances.

Finally, I should like to thank Mr Lyng and his associates in Autronica Marine for their help and co-operation during evaluation of the MIP calculator and the subsequent development of the performance monitoring system.

Table DI

CYLINDER NO.	FUEL VALVE LIFTING PRESSURES (kP/cm^2) *Actual value from engine*	*Maker's recommended value*	*Deviation*
1	395	350	+45
2	440	350	+90
3	420	350	+70
4	380	350	+30
5	425	350	+75
6	385	350	+35

In reply to Mr Haslam-Jones, with this particular application, invasion of the engine structure is limited to the measurement of four basic parameters, i.e.:

(i) Scavenge pressure: by transducer fitted to the scavenge manifold.
(ii) Flywheel position and engine speed: by position pins fitted to the flywheel and an adjacently mounted proximity probe.
(iii) Cylinder pressure: by an air-cooled piezo-electric pressure transducer fitted to the indicator cock.
(iv) Fuel pressure: by a piezo-electric pressure transducer fitted to a custom-designed high-pressure valve attached, for example, to the fuel distribution block.

Regarding a complete diagnostic system of liner wear, liner temperature, bearing temperatures and thermal load involving further invasion technology, I consider that the majority of such systems are acceptable in terms of access for normal maintenance. In each case, the value of such systems must be considered and weighed against their potential for savings. For example, a reliable cylinder scuffing temperature probe can be invaluable in controlling cylinder oil injection rates and striking a good balance between liner wear and lubricant cost.

Accuracy of pressure transducers is obviously of paramount importance; this is reflected in the system design (transducer cost approximately 20% of total system cost). Testing of the transducers on board is not possible with the normal deadweight tester above 1000 lbf/in^2 and therefore the manufacturer's figures for keeping accuracy have to be accepted to a certain extent. Naturally, if transducers are suspect, they are changed but, with the normal precautions of keeping them dry and clean, no major problems have been encountered.

Piston TDC reference is also critical in the overall accuracy of the system and, unfortunately, this varies with respect to the flywheel markings under load. Setting of the reference marks is carried out dynamically to compensate for shaft twist and inaccurate flywheel markings and can be checked periodically by the ship's staff. Under normal circumstances, engine monitoring is only carried out at, or very close to, full load and the minor errors introduced have therefore to be accepted for simplicity.

Regarding response to rate of pressure rise in the fuel system, it is my opinion that the time-constant of the system is sufficient to reflect accurately actual changes in pressure. Good response of this transducer is of vital importance to the system, as changes in the rate of pressure rise are a valuable source of information regarding fuel pump condition and the actual point of fuel injection into the cylinder.

Changes in fuel quality can have significant effects on the ignition quality of the fuel affecting, for example, the rate of pressure rise in the cylinder and ignition delay. Such deviations are identified immediately provided that good historical records of actual trace configurations and salient pressures are maintained on board the vessel. It is debatable, however, with the normal engine having manual adjustments for timing changes, whether fine tuning would be cost-effective when fuel quality could change every voyage. The exception to this would be, of course, an excessive cylinder maximum pressure affecting bearing loads.

More subtle changes, such as the intoduction into the fuel of abrasive particles (possibly catalytic fines), showed quite clearly in service that they could have a catastrophic affect. In one particular case, abrasive wear on the fuel pump plunger/barrel caused a reduction in maximum fuel delivery pressure of 50 kP/cm^2 in a matter of a few hundred hours running.

Generally speaking, analysis of the effects of fuel quality on cylinder performance is extremely difficult to carry out on a multi-cylinder engine in service, since discrepancies will always exist in the timing across the engine cylinders. I would suggest that a laboratory test engine under carefully controlled conditions is the only way to guarantee good results in this area.

Flow charts have been used in the actual shipboard paperwork system in preference to voluminous text which would never be read. Computerization of the system would be best carried out using colour presentation on the VDU, to assist with holding the operator's interest, together with a highly interactive approach to programming.

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