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TRANSACTIONS(TM)

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Improvement of Load Acceptance in a Slow-speed Diesel/CPP Propulsion System

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SYNOPSIS

This paper describes a case study resulting in modifications to improve the load-acceptance performance of a propulsion system comprising a Doxford 76JCR engine with controllable-pitch propeller and single-lever combinator-type control system. An investigation into the causes of poor acceleration and excessive smoking when attempting rapid manoeuvring resulted in changes to the turbocharging and propulsion control systems. A significant improvement in performance was thereby achieved with some particularly interesting features of the overall system interaction being identified during the course of the study.

INTRODUCTION AND BACKGROUND

In 1979 Doxford Engines Ltd accepted an order for a four-cylinder direct-drive engine to be fitted in a bulk carrier of 36 400 m^3 cargo capacity, designed for dual-role operation in the St Lawrence Seaway and conventional deep-sea trading. The engine was to be coupled to a KaMeWa controllable-pitch propeller (CPP) with an unusually low maximum speed of 96 rev/min. The propeller pitch control was arranged to impose constant bmep (brake mean effective pressure) down to 85 rev/min.

The installation therefore represented one of the earlier slow-speed diesel/CPP applications and the contract included stringent requirements for fuel consumption. The engine was a slow running version of the Doxford 76JC4 and designated 76JC4R, where R signifies reduced power. Along with the 58JS3 range of engines these were the first constant-pressure turbocharged Doxford engines to enter service.

With the large port areas and relatively low resistance to the scavenge air flow through the cylinder, the Doxford engine is relatively easy to turbocharge. Thus, the earlier pulse turbocharged J engines required only a small electrically driven auxiliary blower arranged in parallel with the turbocharger(s) to provide satisfactory slow-speed performance. This parallel arrangement of the auxiliary blower was simpler and much cheaper than the alternative series arrangement, and was therefore adopted initially for the constant-pressure turbocharged engines also.

However, constant-pressure turbocharging offers little energy to the turbocharger at low speed and it subsequently became necessary to change the 58JS3 engines to a series arrangement.[1] This was also the case for six of the 76JC4 engines, which left four of this engine type, including the 76JC4R engine, with the parallel turbocharging arrangement.

For three of these last four engines, adequate acceleration and slow-speed performance was obtained simply by changing the turbochargers from plain bearings to ball and roller bearings to reduce friction. However, during the test-bed trials on the 76JC4R engine it became clear that such a modification would be insufficient to meet the special requirements of this particular contract. Thus a technical investigation and development programme was initiated, leading to the improvements of the turbocharging and control systems recorded in this paper.

Dr Fowler completed an Engineer Officer Cadetship and gained further seagoing experience and DTI qualifications with Furness Withy & Co Ltd. Subsequently, he took a BSc degree in Engineering Science from Edinburgh University, followed by MSc and PhD degrees from Newcastle University. Following periods of employment spent at Sunderland Shipbuilders and South Shields Marine and Technical College, he is now employed as lecturer in Marine Engineering at Newcastle University. His main research interests are autonomous underwater power sources and propulsion control system design modelling and simulation.

Dr Ørbeck took a BSc in Mechanical Engineering at Glasgow University in 1953 and a PhD in 1956. As Technical Manager at Doxford Engines Limited he developed the Doxford 76JC4 and 58JS3 engines. He is now Advanced Engineering Manager at Marine Design Consultants Limited.

Objectives

The primary objective of this paper is to trace the investigation through its various stages leading to the implementation of changes associated with the turbocharger and the propulsion control systems. A secondary objective is to illustrate and reinforce the philosophy of the 'systems approach' as a structured method of investigating engineering problems involving dynamic interactions between sophisticated components of machinery.

The structure of the paper essentially follows chronologically the progress of the investigation, in which the following phases may be identified:

1. The design and installation of a hydraulically powered turbocharger accelerator to provide scavenging assistance at low engine powers.
2. A review of the propulsion system components, their interactions, performance, and definition of an appropriate method of study.
3. A detailed account of specific operational problems and an analysis of ship and test-bed data available from previous engine trials.
4. A detailed consideration of the behaviour and interactions of control system components leading to specifications for modification.

THE MACHINERY INSTALLATION

The Doxford 76JC4R engine with a maximum continuous rating (MCR) of 6300 kW at 96 rev/min is basically a derated version of the 76JC4 engines, which were originally rated at 8950 kW and 123 rev/min. These engines are of the four-

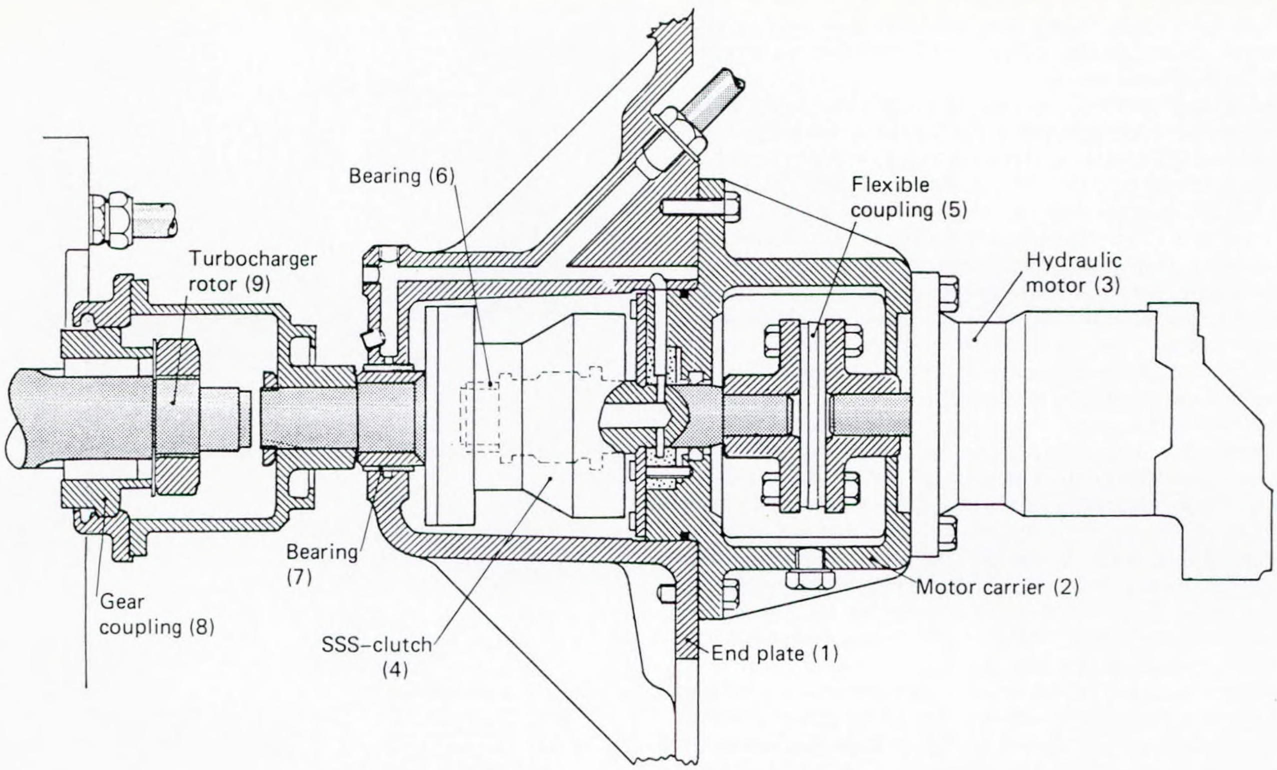

FIG. 1: Hydraulic motor drive to turbocharger

cylinder, single-acting, crosshead opposed piston type with a bore of 760 mm and a combined stroke of 1870 mm.

The 76JC4R engine was fitted with a single Napier NA650 turbocharger located at the after end of the engine. The turbocharger was fitted with ball and roller bearings, as previously described. Constant-pressure turbocharging was adopted with an additional 45 kW auxiliary fan arranged in parallel. The engine was direct coupled to drive a KaMeWa CP propeller with a single-lever combinator being used simultaneously to set the shaft speed (56–96 rev/min) and the propeller pitch angle. Continuous unidirectional engine operation was facilitated with astern power achieved by suitable pitch reversal.

Test-bed performance

With the equipment available on the test bed it was difficult to simulate the service load to be experienced by the engine during acceleration. The engine was contracted to accelerate from 56 rev/min (idling) to 85 rev/min in under 1 min, but in practice could only achieve this speed in approximately 3 min. Initially, it was found to be particularly difficult to accelerate beyond the load at which the auxiliary fan had to be shut off to avoid surging. The exhaust from the engine under this condition was observed to be black with smoke. At this load, the turbocharger speed was 3000 rev/min and barely self-sustaining. However, by the time the turbocharger speed had reached 4000 rev/min the exhaust was clean and the engine accelerated well.

Improvements were therefore required to meet the contractual obligations, but since the projected electrical load in the engine room during manoeuvring was high, a solution which further increased the demand for electrical power was to be avoided if possible. After alternative proposals had been investigated it was decided to design a novel hydraulic turbocharger accelerator, as described below. This system was considered to offer significant advantages in terms of low inertia, flexible speed/torque characteristics and the ability to utilise the high-efficiency blower and air ducting incorporated into the existing turbocharger. The existing electric blower was retained as an emergency backup.

Hydraulic turbocharger accelerator

Although the parallel auxiliary fan was capable of providing a maximum pressure of 0.11 bar at zero flow, surging of the fan was encountered when the turbocharger speed had reached about 3000 rev/min, corresponding to a scavenge pressure of about 0.07 bar. This is quite common and occurs because of pressure fluctuations in the scavenge belt. Under this condition the corresponding engine power and air flow were 1650 kW and 10.2 kg/kW h, respectively, and the pumping power imparted to the air was 27 kW.

It was anticipated, by extrapolating from the salutary effects achieved by fitting low-friction turbocharger bearings, that a substantial improvement in output would be achieved by fitting the hydraulic assister. It had previously been established that when the NA650 turbocharger was fitted with plain bearings, the power losses in the bearings at 8000–9000 rev/min would be about 10 kW. Friction losses in plain bearings are proportional to the square of the speed and the loss at 3000 rev/min would therefore be about 2 kW.

It was estimated that the turbocharger efficiency at 3000 rev/min would thus be improved from 57.5% to 60% simply as a consequence of fitting ball and roller bearings instead of plain bearings. This would obviously produce beneficial results with respect to cylinder scavenging. Allowing for the improvement achieved by changing from plain to roller bearings in the turbocharger it was estimated that providing an additional 20 kW of hydraulic power at 3000 rev/min should prove more than adequate.

At the engine idling condition the assistance to the turbocharger would require to be of the same order as the power provided by the auxiliary fan. However, it was considered that the turbocharger compressor and its ducting would be more efficient than the auxiliary fan and associated ducting, with the result that a smaller power could therefore be accepted from the hydraulic motor.

The system which was proposed to meet this specification comprised an hydraulic motor driving the turbocharger rotor through an SSS clutch, as shown in Fig. 1. A Volvo Bent Axia hydraulic motor type F11-39 was chosen, capable of developing 52 kW continuously at 4200 rev/min and 22 kW at 1000 rev/min. The synchro-self-shifting (SSS) clutch was designed to disengage when turbocharger speed exceeded 4200 rev/min, whereafter the motor could then be stopped.

During slowing down of the engine the hydraulic motor is started before the turbocharger speed has fallen to 4200 rev/min. Clutch engagement occurs automatically at this speed and the hydraulic motor then drives the rotor for all speeds below this point.

The design characteristics of the unit, including an estimate of power absorbed by the turbocharger compressor are shown in Fig. 2. The hydraulic system was designed for a maximum pressure of 245 bar, which provides hydraulic motor power proportional to speed up to 2800 rev/min. Above this speed the power remains constant, which implies falling pressure as shown. This is achieved automatically by the unit's own hydraulic control device. The hydraulic motor is supplied with oil from a swash plate pump driven by an electric motor.

In the lower part of the speed range the power produced by the hydraulic motor is well in excess of the power absorbed by the compressor, as shown in Fig. 2. Even neglecting the turbocharger turbine power contribution it is readily seen that there should, consequently, be adequate scope for acceleration.

Referring to Fig. 1, the whole assembly is mounted on an end plate (1) which replaces the original plate on the turbine end of the NA650 turbocharger. A motor carrier (2), which is bolted to the end plate, supports the hydraulic motor (3). The motor output shaft is connected to the input shaft of the SSS clutch (4) by a flexible coupling (5). This coupling gives radial support and angular flexibility. The clutch output shaft is located to the input shaft both radially and axially by the bearing (6) and the output shaft is supported by a bearing (7). A gear coupling (8) connects the clutch output shaft to the turbocharger rotor (9), the driven part of this coupling being secured with keys to the rotor locking plate. Thus, thermal expansion and any axial movement of the rotor can be allowed for.

The principle of the SSS clutch is well documented and has been described for example by Clements.[2] Apart from its primary function as an overspeed clutch, it can readily accept the large speed difference between the input and the output shaft which occurs when the engine is running at maximum power while the hydraulic motor is stopped. The clutch and its bearings use the same oil supply as the turbocharger.

Installed performance

The hydraulic accelerator was fitted prior to sea trials which took place in the autumn of 1981, and after some minor difficulties, mainly caused by dirt in the oil supply to the motor, the system was successfully commissioned. With the engine stopped the turbocharger speed was about 1000 rev/min, rising to about 2000 rev/min with the engine idling. During sea trials the engine accelerated from idling to 85 rev/min in under 1 min with clean exhaust, thereby completely satisfying the contractual requirement for acceleration.

Unfortunately, two problems associated with the accelerator system came to light early in the ship's service. The first of these happened after an emergency stop of the engine which was caused by loss of the lubricating oil. The failure of the accelerator unit was discovered after noting increased exhaust temperatures when the engine was brought up to full power after the stop. On investigation it was found that the bearings in the SSS clutch had seized, this failure subsequently being

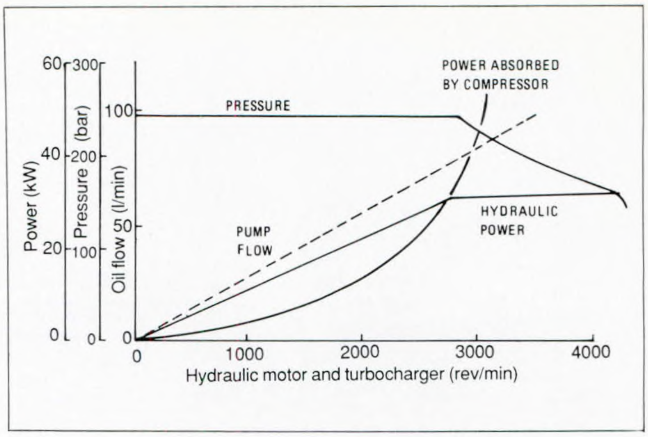

FIG. 2: Turbocharger accelerator characteristics

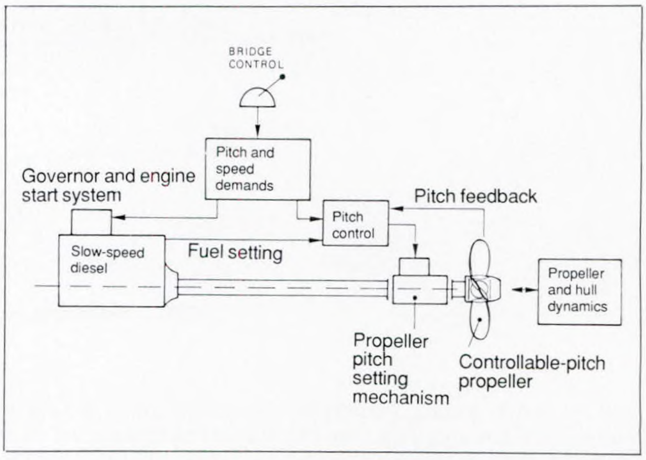

FIG. 3: Schematic diagram of propulsion system

explained by loss of lubricating oil to the clutch. During emergency stops, it is normal for the turbocharger to run for a few minutes after the engine has stopped and it is therefore fitted with a header tank which automatically takes over the supply of lubrication if the engine supply fails. This system had not been connected to the clutch, an omission which was rectified after the bearing failure. The second problem with the system was wear at the teeth in the gear coupling because of incorrect engagement. This situation was easily rectified by adjusting the axial location of the assembly.

During the period while the hydraulic system was out of action the engine was operated using only the auxiliary fan with the result that the performance was less than satisfactory as had been predicted by test-bed experience. Under these conditions, certain problems were traced to the control system, in particular the interaction between the propeller load control and the engine fuel control and torque limit. The control system in this installation was more complicated than in an installation with a fixed-pitch propeller and BS(ETS) therefore engaged Dr Fowler of the Marine Engineering Department at the University of Newcastle upon Tyne to carry out a full review of the system. Both KaMeWa, the propeller makers, and Bond Instruments & Control Ltd, who had supplied the engine governor, offered invaluable assistance during the subsequent investigation.

The study of the control system identified some weaknesses in the overall control strategy and it was decided that these features should be modified irrespective of whether the engine was to be operated with the hydraulic acceleration system or with the auxiliary fan.

THE CONTROL SYSTEMS

For the purpose of control system and dynamic performance evaluation, the main elements in this particular marine diesel propulsive system are identified in Fig. 3. These may be grouped as follows with the objective of formulating a 'systems' approach to the study:

1. Propulsion engine with associated fuel and turbocharging systems.
2. Controllable-pitch propeller, actuator and controls.
3. Hull resistance and inertia effects.
4. Overall control system.

Each of these 'sub-systems' may obviously be further sub-divided depending on the detail of analysis required.

The method of study

Since the sub-systems all interact with and impose demands on each other it may often prove extremely difficult to distinguish between cause and effect, especially during dynamic operating conditions, eg when manoeuvring. Steady-state analysis is usually easier, although in practice dynamic history often determines the steady-state solution, which may not be unique, especially where non-linear limiting devices are active in several areas, as is the case in a complex propulsion system.

It must be emphasised that the ideal 'systems approach' to such a complex non-linear and interactive configuration as outlined above would involve the utilisation of computer simulation techniques to define fully and quantify the physical processes involved, particularly during transient operating conditions. This technique is well developed and is referenced in numerous papers.[3-6] Unfortunately, under pressure of time and cost, such an option was considered to be impractical in the present context. The investigation therefore developed as a 'desk top' analysis of the characteristics and inter-relationships between the most significant sub-system components.

The techniques used were similar in philosophy to those in computer simulation, both in terms of system representation and interpretation of results. However, in this case physical performance data were extremely sparse compared with those which would have been available following a detailed simulation exercise. Nonetheless, important conclusions were reached, as will be shown.

The propulsion control system is shown schematically in Fig. 4, the main sub-systems for the purpose of the present study being briefly described as follows.

KaMeWa bridge control system

A single-lever combinator system simultaneously transmits pneumatic signals to the governor set point receiver and to the CPP pitch control system. The pitch and speed schedules for this particular vessel are shown as functions of combinator lever position in Fig. 5.

The schedules nominally determine the 'steady-state' values of pitch and engine speed for a given demand lever position. However, the pitch schedule can be overridden by the propeller load control system (within limitations) to prevent engine overloading.

Servodyne speed control system

The pneumatic combinator speed demand signal is converted to an electrical signal at the electronic governor input, then processed and compared with the actual engine speed. The resulting error signal is amplified by proportional and integral terms (derivative action is not used). The resulting corrective signal is then passed to the positioner servo card, which operates the fuel control shaft through a high-response pneumatic servo, incorporating electronic feedback.

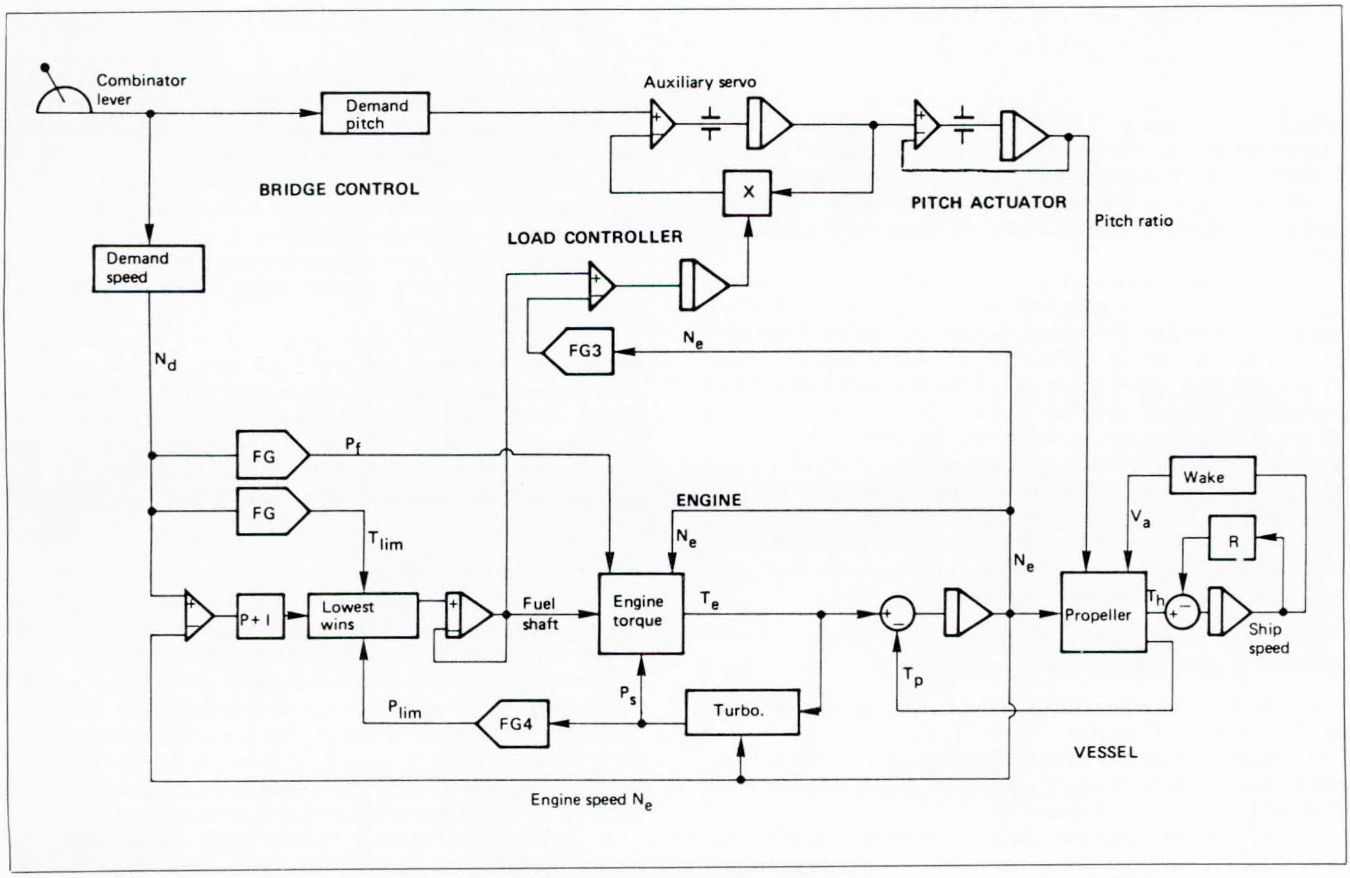

FIG. 4: Schematic diagram of control system

In addition to the normal speed control function described above, three important limits are applied to the fuel control signal in the governor:

1. A fixed maximum limit on fuel shaft position.

2. A function generator which relates maximum allowable fuel shaft position to demanded engine speed, and limits the fuel shaft output accordingly. This feature is intended to ensure that the engine can not be operated outside its design operational envelope.

3. A scavenge pressure limit which operates to ensure that fuel shaft setting can not increase unless a minimum boost pressure exists at any given demand speed.

Doxford fuel control system

The Doxford common rail fuel system is well documented and has been described in detail by Jackson.[7] Briefly, the system incorporates ram displacement, engine-driven, helix-controlled pumps delivering into a high-pressure 'reservoir'. The 'fuel racks' are nominally fixed on this particular engine, and fuel pressure is regulated by a pneumatically loaded 'spill valve' which bypasses excess fuel back to the pump suction. The 'rail' pressure is set by a mechanical open-loop control system linked to the fuel control lever and therefore depends on demanded speed.

The governor controls the position of the fuel control shaft with respect to demanded speed and imposed load. The fuel shaft operates the timing valves through suitable linkages, and hence determines the duration of the fuel admission period.

A highly significant feature of this type of fuel system is that the rate of fuel delivery per engine stroke is a function not only of the fuel shaft position (corresponding effectively to fuel rack position in the more conventional jerk pump system) but also of fuel rail pressure and speed of camshaft rotation.

To provide a quantitative assessment of the engine torque capability as a function of these variables, the following expression was developed, as detailed in Appendix I:

$$T = \frac{84(\theta - 22)\sqrt{p_f}}{N_e}$$

where T is the torque, θ is the fuel shaft angle in degrees, N_e is the engine speed in rev/min, and p_f is the fuel rail pressure in bar. This expression was subsequently used at various stages in the investigation, particularly when estimating the magnitudes of potential overloads under off-design operating conditions.

The implications are particularly important when considered in the context of governor fuel rate limiting implementation. Governors used in marine applications are often fitted with 'torque limits'. In a mechanical governor this is a cam-controlled limit set as a function of the demand speed. When applied to a jerk pump fuel system this device will limit the stroke of the pump and therefore the fuel delivered per revolution. Thus in effect it becomes a torque limit. When applied to the 'common rail' system the governor limiter will limit the timing valve opening period in degrees. The length of time the valves are open is, however, inversely proportional to speed and the limit therefore effectively becomes a power limit. This can be seen from the above expression.

KaMeWa controllable-pitch propeller

Three distinct but interacting sub-systems may be identified in the overall CPP system:

1. Propeller characteristics giving torque and thrust as functions of advance coefficient J (where $J = V_a / N_p D$), pitch ratio p/d, and propeller shaft speed N_p.

2. Pitch control and actuator system which converts the pneumatic combinator signal to actual pitch ratio through hydraulic servo mechanisms.

3. Electro-hydraulic load control system which is activated from the engine fuel shaft position and is capable of over-riding the combinator pitch control system to trim load according to engine capability.

Propeller data were available as power against speed for pitch ratios between 0.5 and unity, steady-state vessel speeds being superimposed on the same diagram. However, it must be remembered that these are steady-state data and do not represent the transient situation where, for instance, high torque can be imposed at relatively low engine speed if, for instance, speed of advance V_a is low.

Operation of the pitch control actuator and load control systems is illustrated in the overall control system diagram (see Fig. 4). An auxiliary hydraulic servo is pneumatically positioned from the bridge control system and actuates the main hydraulic servo piston in the propeller boss. The valve rod provides a mechanical pitch position feedback to the control system. The load control system (when active) effectively modifies the feedback ratio by means of a small hydraulic ram, with the result that a pitch correction signal is proportionally superimposed on the nominal combinator pitch demand signal.

The essential feature of the load control system is that a comparison is made between the actual engine fuel control shaft position and a reference fuel shaft setting, the latter being specified by the engine builder as a function of engine speed. Any error between these signals is effectively integrated (within limits) and used to modify pitch, so that the engine is nominally loaded according to the reference signal during those periods when the load control system is 'active'. In this way, it is intended that acceptable torque loading profiles will be ensured during vessel manoeuvring or other 'off-design' operating conditions.

A variable potentiometer is also provided on the load controller and provides a facility for adjusting the load characteristic from the nominal configuration.

REVIEW OF DATA FROM SHIP TRIALS

At the time of conducting the investigation, the vessel was laid up for the winter and consequently ship trials were not feasible. Data used in the investigation were therefore restricted in accordance with the following sources:

1. Steady-state engine test-bed data provided by the Builder (see Fig. 6) showing fuel shaft angle, fuel pressure, torque and engines speed as functions of control lever position.

2. System details provided by the main sub-contractors.

3. Steady-state and transient results recorded during ship trials at an earlier date.

The available ship trial data had not been recorded specifically to assess control system performance and consequently instrumentation had not been installed comprehensively enough to define completely the complex interactions between the respective systems. However, careful sifting and extrapolation of the available data did yield invaluable insights into the system's performance, as discussed below.

Steady-state trial data

Even from consideration of steady-state recordings, it was apparent that control system performance was less than adequate. For example, it was noted that in one case with a demand speed of 90 rev/min, the engine was deficient in both torque and speed under an apparent steady-state condition. This suggests that one of the governor limiting functions must have been activated, since a P+I device would not normally tolerate steady-state error but would have attempted to push the fuel shaft towards the maximum position.

A detailed explanation which would account for the existence of this situation is given in Appendix 2. Summarising this explanation it may be postulated that there existed, in

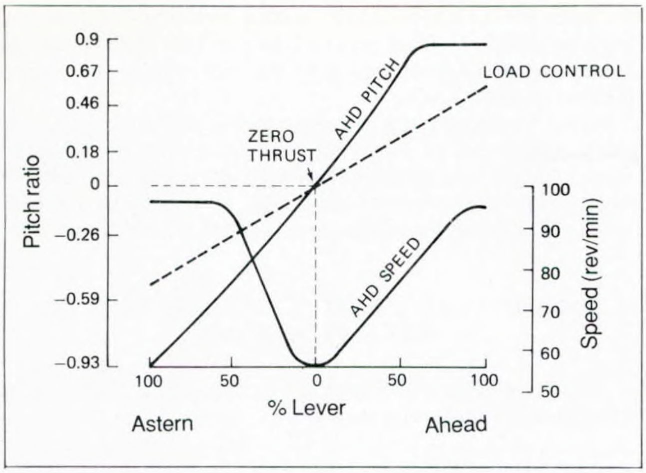

FIG. 5: Steady-state speed and pitch ratio schedules

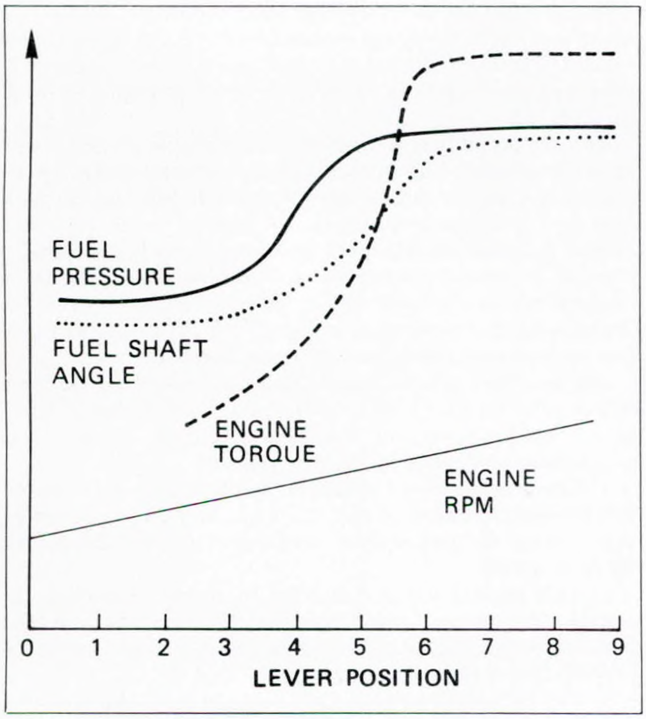

FIG. 6: Engine test-bed design data

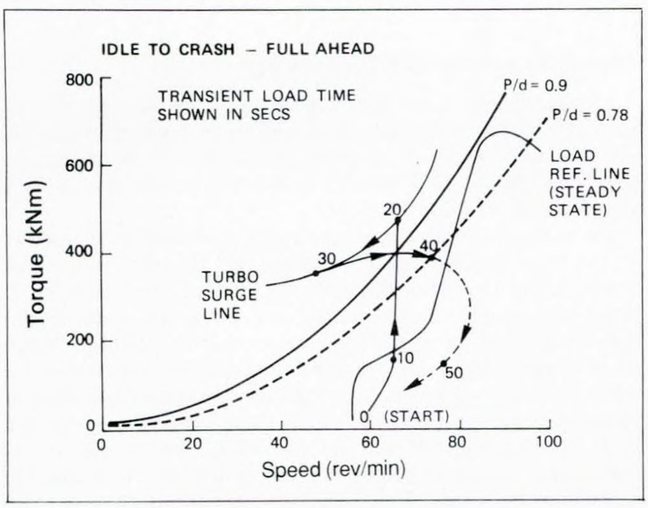

FIG. 7: Sea trial manoeuvring response

fact, a situation where because of low fuel pressure, the governor torque limit was activated by the fuel shaft attempting to compensate by increasing beyond its design setting. In the process, the load controller would have been activated since the actual shaft angle would have increased above the load controller reference line. The result would be a 'pseudo-steady state' situation, as recorded.

Transient trial data

The most significant insights into the system's behaviour were actually illustrated by considering a number of the strip chart recordings showing torque, cylinder peak pressure and thermocouple temperatures on a continuous time base while manoeuvring with the engine during the Lake trials. These results were plotted for convenience in the torque against speed plane, an example being shown in Fig. 7. This recording corresponds to an attempt to accelerate the engine in response to a sudden movement of the demand lever from position 0 to 8.

A detailed step by step description of events during this transient is presented in Appendix 2, together with an explanation of how such a situation could arise.

In summary, it may be readily observed that the load control system does not appear to operate effectively and the engine speed is unable to increase, with the result that as load increases the turbocharger surge limit is eventually violated and the engine has eventually to be returned to the idle condition.

Subsequent consideration of the available data as detailed in Appendix 2 led to the conclusion that such a situation could arise in the event that low scavenge pressure was causing limiting of the fuel shaft angle in the governor, while the propeller load control system, being unaware of any overload violation (since its limit had not been transgressed), remained inoperative.

PERFORMANCE OF CONTROL SYSTEMS

Consideration of the evidence suggested that the fundamental reason that malfunctions were possible was not faulty operation of any one component but was related to the unique and unfamiliar dynamic interactions which are possible between the KaMeWa load control system, the Doxford common rail fuel system and the Servodyne governor. In particular, the load controller plays a significantly important role in this particular propulsion system at speeds above 80 rev/min, since above this speed it is required continuously to back-off pitch from the maximum p/d ratio of 0.9.

The significance of this feature is that in the event of a load control system failure, the combinator profile will prevail and the pitch ratio will be maintained at 0.9. Although this pitch is acceptable at 80 rev/min, it would produce gross propeller overloads at, for instance, the full rated speed of 96 rev/min.

Governor torque limit

For a vessel with CPP, the propeller load control system should prevent overloads from being presented to the engine. However, if under some circumstance such an overload is presented, the governor torque limit feature is intended to prevent acceptance by the engine. A partial stall situation will then result, with engine speed decreasing until the propeller torque matches a load the engine can safely sustain.

The philosophy of the torque limit feature may therefore be defined as restraining the fueling rate, and hence preventing the engine from generating torque above a certain level which for one reason or another is considered to be excessive (eg bearing loads, turbocharger surge or exhaust smoking considerations). It follows that the limit should be a variable function of actual speed and not demand speed as was the case originally.

The importance of this point is revealed when it is considered what happens if the engine partially stalls and settles at some speed which is below that demanded. There may then exist a 'pseudo-steady-state' condition, in which the value registered by the torque limiter will not correspond to the real engine speed but will correspond to some other (probably higher) demand speed. Hence the fuel shaft position will be allowed to advance further than is intended before limiting eventually occurs.

Also the engine torque cannot be assumed to be a simple function of fuel shaft angle and speed on this particular type of engine because fuel pressure can also vary. This variable is not present in more conventional fuel systems.

The torque limiting problem is potentially compounded by the fact that the fuel pressure control cam setting is also a function of demand speed rather than actual speed. Thus fuel pressure could also be higher than intended, potentially resulting in increased overload capability.

It thereby became apparent that even if the original torque limiter was capable of providing adequate steady-state protection, performance would not be guaranteed under certain conditions arising when attempting rapid engine accelerations.

Scavenge pressure limit

The scavenge pressure limiter is a device based on a filtered pressure transducer signal which is passed through a simple linear function generator consisting of variable slope and intercept. Its primary function is to prevent engine smoking during transients by limiting fuel shaft angle.

However, its influence on system performance could be very significant since it also has the capability to limit engine torque generating capacity without exerting any direct influence on the load imposing device, ie the propeller. It was therefore apparent that close attention to this item of equipment was also required, in the context of turbocharger performance and implications for the load control system.

Load controller performance

It is worthwhile reconsidering, at this stage, the philosophy behind the profiling of the load line which in turn determines the setting up of the load control function generator. Traditionally, the line is constructed somewhere near to the fixed-pitch-propeller load characteristic.

The reason is that the turbocharger, being a rotodynamic machine, possesses pressure ratio/flow capacity characteristics which also follow a propeller-type law, with the result that if correct matching of engine and blower is ensured at full load then part-load matching throughout the range should also be approximately correct. In practice, turbocharged slow-speed engines with fixed-pitch propellers can be accelerated fairly easily, as long as the rate of acceleration is not too great, so that engine load does not deviate significantly from the 'propeller law' characteristic.

Perhaps the most critical area for torque generation in this particular installation occurs in the vicinity of the relatively flat part of the engine operating characteristic towards the upper end of the power diagram. It was required that the torque limit should be typically only 3% above the normal operational load line in this vicinity.

However, implementation of such a tight specification introduces some reservations in terms of practicalities associated with accuracy of analogue electronic function generation. This is because it is essential to ensure that at any operating condition the limiting value of control shaft angle associated with the governor torque limit does not fall below the control shaft reference setting associated with the load controller torque limit. In other words, the load control system must always be activated before the fuel shaft angle is limited in the governor.

If the reverse sequence occurred, the engine output torque would indeed be limited, but the load control system would be 'unaware' of an overload existing and would not reduce pitch accordingly. This would result in the pitch and load increasing unhindered towards the new steady-state combinator demand setting.

It will be apparent, upon considering the working of this particular system as originally configured, that it was by no means certain that load limitation would always precede the engine torque generation constraint. It was therefore concluded that further attention would be required in this area.

MODIFICATIONS TO CONTROL SYSTEM SPECIFICATIONS

The proposed modifications will be discussed below and are illustrated schematically in Fig. 8.

Combined torque generation and load control limits

It was envisaged that, in practice, it would prove very difficult to set up two separate function generators, one in the governor and the other in the propeller load controller, within the constraint of a combined accuracy of 3%. In fact, it was estimated that to achieve the correct margin a resolution of 1° of control shaft angle or 0.15 V at the actuator amplifier would be required.

This suggested that it would be more sensible to use a single function generator for estimating both the load control reference characteristic and the governor torque limiting function. The new characteristic would, of course, be defined with respect to actual engine speed and would also include effects from fuel pressure variations, a feature which was missing from both functions in the original system. The governor torque limit would then always be set slightly above the required load line by a margin dependent on actual speed N_e.

The modified governor-generated load controller function would then be used by the existing KaMeWa feedback controller system. However, the following additional specifications would now be defined in the new system:

1. The propeller load controller would always be activated before any limitation of fuel shaft position was imposed by the engine protection system, irrespective of the source of the limiting signal.
2. This implies that in addition to simply restricting the engine fuel rate, as was originally the case, the scavenge pressure limiter would now also be capable of activating the propeller load control system to reduce pitch.
3. The limitation of fuel shaft position from the governor would be implemented using an open-loop technique to avoid the difficulties associated with switching, stability and integral wind-up, which could arise if a second closed-loop system was introduced to limit engine torque generation capability.

Predicting settings for the scavenge limiter

Ultimately the scavenge limiter must be set up experimentally during ship trials. However, to provide guidance an attempt was made to produce an analytical approach to scavenge limiter definition using the admittedly sparse data which were currently available.

An outline of the principles used is presented in Appendix 1. The basic approach is to postulate that the trapped air:fuel ratio (AFR) must not fall below a certain limit (say 20:1) when the engine is accelerated from any point on its design operating characteristic. This permits an estimate to be made of the maximum tolerable fuel shaft setting which can be sustained at any given engine speed, assuming that the turbocharger discharge pressure and fuel rail pressure are at the level corresponding to the normal steady-state condition for that particular engine speed. This maximum fuel shaft position must then be imposed electronically as a limiting voltage in the governor.

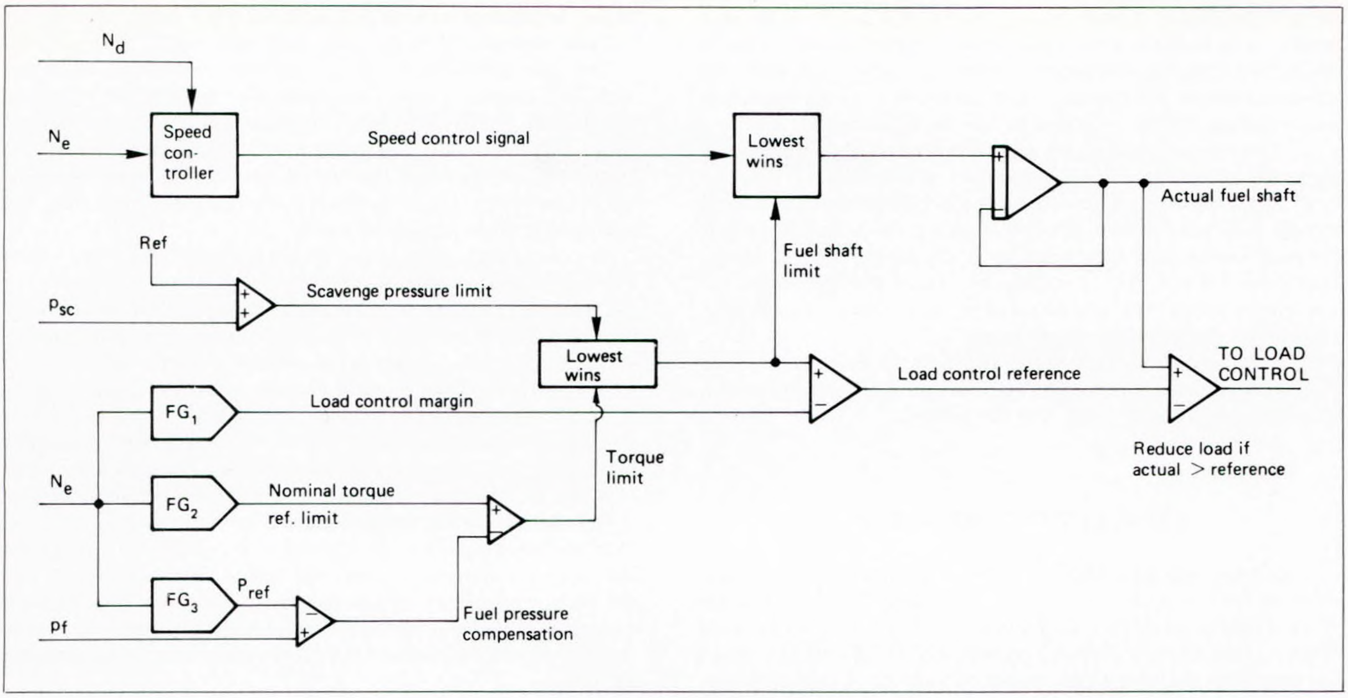

FIG. 8: Modifications to control system

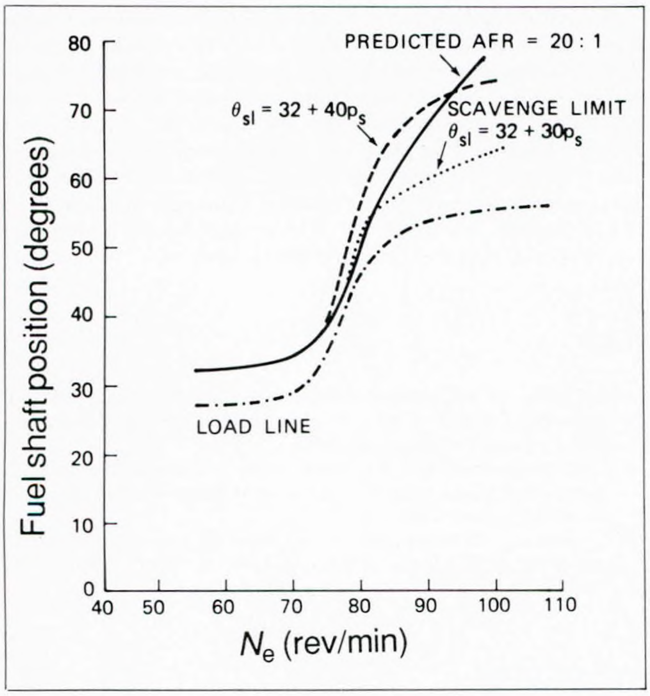

FIG. 9: Scavenge limiter function generation

In this way, the limit characteristic effectively determines an acceptable margin of advancement for control shaft position above the normal steady-state operating point, assuming the correct boost pressure exists. If the boost pressure then falls for any reason the fuel limit will reduce, and conversely if boost pressure rises above its normal steady-state condition the limit will be automatically raised.

Figure 9 illustrates the scavenge limiter characteristic predicted, assuming the AFR = 20:1 criterion is adopted. It will be noted that the margin between the AFR limit and load characteristics is extremely small in the vicinity corresponding to an operating speed of 80 rev/min. This confirms the observed tendency for the engine to be potentially short of air in this vicinity.

In practice, a close approximation to the AFR = 20:1 limit is predicted using a scavenge pressure function of the form:

$$\theta_{s1} = 32 + 30P_s$$

where P_s is scavenge pressure in bar g and θ_{s1} is fuel shaft angle in degrees. The predicted characteristic arising from such a function generator is shown superimposed on Fig. 9, and in the critical region of interest is seen to show excellent correlation with the anticipated AFR = 20:1 operating line.

The margin between the limiting and loading characteristics may be increased if required, and as an example an alternative characteristic function is also shown in Fig. 9. It is estimated, however, that under this condition AFR could fall to 17:1 during the transients.

The setting of the scavenge limiter potentiometers is therefore seen to be a compromise between conflicting requirements, which in the absence of simulation can only be fully reconciled by ship trial adjustment.

Summary of control system modifications

A summary of the key features associated with the proposed modifications, as shown schematically in Fig. 8, is as follows:

1. The nominal engine torque limit signal is now generated as a function of actual engine speed by FG_3 according to the characteristic specified by the engine manufacturer.
2. Any error between actual and nominal fuel pressure at defined engine speed is used to modify the torque limit using a linear approximation technique, thereby producing the actual engine torque/speed limit.
3. The scavenge pressure limiting signal is generated by FG_1 as a straight-line function of boost pressure.
4. The actual fuel shaft limit is selected by a 'lowest wins' comparison of the torque and pressure limiter signals.
5. The speed-controller-demanded control shaft position must pass through the second 'lowest wins' circuit where it is compared with the fuel shaft limit. The selected output determines the actual control shaft angle.
6. The reference line for the KaMeWa load control system is generated by subtracting a variable margin from the fuel

shaft limit signal. The margin between the torque limit and load line is expressed as a function of engine speed. Thus it is ensured that the load line reference is always less than the governor torque reference, so that governor limiting can never occur before the load controller has been activated.

7. The load controller can now be activated by low scavenge pressure in addition to excessive fuel shaft angle.

8. The error signal upon which the propeller load control system operates is now generated using the same fuel shaft position sensor and speed sensor as are used by the governor fuel shaft limiters. This precaution ensures that problems are not encountered because of relative errors between different sensors in two separate subsystems.

9. Inclusion of variable gain and bias on the output error signal transmitted to the existing load control system provides flexible compatibility with that equipment.

CONCLUDING REMARKS

This paper has described an unusual hydraulic turbocharger acceleration system which was designed, built and commissioned successfully with the main engine during acceptance trials. The results achieved completely validated the main assumptions and decisions taken during the initial stages of consideration.

The paper has also provided details of a complete propulsion system review leading ultimately to suggestions for control system improvements. Following agreement between the various sub-system manufacturers on the implementation of the suggested modifications, replacement circuit cards were designed, manufactured and fitted into the existing electronic governor. The system was commissioned and set up according to the defined specifications and sea trials were subsequently conducted on 21 August 1985. It was not possible at this time to record the transient response of the significant operational parameters, but it was established by general observations that the system worked effectively. In particular it was ensured that the correct sequence of events occurred during the periods when the system automatically transferred from combinator to load control following large load increases.

The manoeuvrability of the vessel was entirely adequate with full engine speed attainable well within the specified interval of 1 min from idle. The exhaust remained clear on these occasions and the system performed generally to the complete satisfaction of the operating staff. The vessel has now been in service for a further period of one year and the performance has remained good.

In concluding, it is stressed that although this case study details specific problems and their solution, the underlying philosophy relates to a technique of analysis derived from the concepts of dynamic modelling and simulation. Whilst commonly applied in designing propulsion controls for warships, where particularly responsive control is essential, such studies are rarely applied to merchant vessels where cost and profit margins are particularly critical and response rates generally less significant. However, the study plainly illustrates the value of such an exercise particularly where relatively unconventional machinery configurations are to be integrated.

It is suggested that as the cost of computer simulation hardware continues to diminish while famiiiarity with the software techniques simultaneously expands, the modern marine propulsion engineer will increasingly come to accept these methods as standard for all design and trouble-shooting activities.

ACKNOWLEDGEMENTS

We would like to express our gratitude to Marine Design Consultants Ltd of Sunderland and to Upper Lakes Shipping Ltd, Canada, for permission and assistance in producing this paper. S.S.S. Gears Ltd and Volvo provided valuable design assistance in connection with the hydraulic turbocharger accelerator which we gratefully acknowledge. Appreciation is also expressed to Bond Instruments and KaMeWa for their co-operation in providing technical information during the study. Finally, we thank Mr H. Henshall for his advice and assistance during the course of this study.

REFERENCES

1. S. H. Henshall and F. Ørbeck, 'The Doxford three-cylinder Engine'. *Trans. North East Coast Institution of Engineers and Shipbuilders*, vol. 95, pp. 61–92 (1978–79).
2. H. A. Clements, 'Operational Experience of the S.S.S. (synchro-self shifting) clutch'. *Trans. ASME* (March 1972).
3. A. M. Dorrian, 'An introduction to the use of model simulation in system design'. *Trans. I.Mar.E. (TM)*, vol. 97, paper 17 (1985).
4. J. B. D. Rush and D. R. Broome, 'Optimum control of marine propulsion units'. *Trans. I.Mar.E. (TM)*, vol. 97, paper 18 (1985).
5. A. Fowler, 'Hybrid computer simulation and validation of a closed-cycle diesel engine'. UKSC Conference on Computer Simulation, Bath University (September 1984).
6. M. G. Parsons and J. Y. Wu, 'Limit cycles in diesel/CPP propulsion systems using load control'. *International Shipbuilding Progress*, vol 32, no. 375 (November 1985).
7. P. Jackson, 'Developments in fuel injection and the combustion process in diesel engines'. Paper presented to I.Mar.E., 20 October 1966.

APPENDIX 1

Since it was apparent that torque overloading could occur, it was necessary to establish a method of estimating the engine torque for any given set of conditions, and to assess the viability of the existing protective devices, as listed below:

1. Propeller load control system.
2. Governor torque limiter (function of demand speed).
3. Governor absolute limit.
4. Mechanical fuel control shaft stop.
5. Maximum capacity of the fuel pumps.
6. Absolute mechanical limit on ahead pitch ratio.

This was achieved by developing a semi-empirical torque generation equation based on the postulation that torque developed in the engine cylinders is proportional to fuel per stroke, according to a relationship of the form:

$$Q = k_1 \int_0^{t_i} \sqrt{p_f}\,\mathrm{d}t$$

where Q is the fuel per revolution, p_f is the fuel rail pressure, t_i is the injection time interval. Also, $T \propto Q$, where T_i is the indicated torque.

Integrating assuming constant p_s during the injection interval, ie

$$T_i = k_2\sqrt{p_f}\, t_i$$

but

$$t_i \propto \frac{\theta}{N_e}$$

so

$$T_i = \frac{k_3\sqrt{p_f}\theta}{N_e}$$

where θ is the fuel shaft angle in degrees and N_e is the engine speed in rev/min.

The no-load fuel setting ($\theta_0 = 22°$) corresponds to the deficit caused by friction and other losses. Hence the actual brake torque, T_e, to a first approximation can be expressed in the form:

$$T_e = \frac{k_3(\theta - \theta_0)\sqrt{p_f}}{N_e}.$$

The constant k_3 was estimated for the various power levels from engine test-bed data and plotted against engine speed. It appeared that an average value of $k_3 = 84$ gave excellent correlation between test-bed data and the estimated torque, and the above equation was therefore used in subsequent estimates throughout the study.

Derivation of a scavenge limiter analytical function

In an attempt to establish suitable settings for a scavenge limiter, a simple model of the engine was developed. It must be stressed that data availability was poor and a number of assumptions had to be made, which obviously affect the results obtained. However, the model provided a useful indication of trends and allowed an approximate estimate of the required settings for the intercept and slope of the existing limiter.

The key assumptions implicit in this approach are as follows:

1. The air trapped in the cylinder is estimated assuming that the trapping volume is 85% of the total cylinder volume (4 m^3).

2. The % purity of the trapped air varies linearly from 60% when the scavenge pressure $p_s = 0$ bar (gauge) to 100% when $p_s = 1$ bar.

3. The cylinder pressure at the end of trapping is the same as the turboblower discharge pressure (available from test-bed data), and the temperature is assumed to be 47 °C.

Hence mass flowrate trapped in the cylinder is given by

$$\dot{M}_a = 0.85\rho(0.6 + 0.4p_s)(p_s + 1) \times 4 \times N_e \times 60 \text{ kg/h}.$$

Implementation of this equation over the steady-state operating range of the engine defined from test-bed data suggested that trapped air did not fall below 22:1, which appears quite acceptable.

An approximate expression was also developed to calculate fuel delivered as a function of fuel pressure, control shaft angle and engine speed. This derivation was similar in form to that used in the torque equation described above:

$$\dot{M}_f = 1.717(\theta - 19.3)\sqrt{p_f} \text{ kg/h}.$$

Checks made with the available data showed very reasonable correlation between prediction and observation.

The combined fuel flow and air flow equations were subsequently used to estimate the maximum value of θ for any specified air:fuel ratios.

APPENDIX 2

Explanation of recorded anomalies in steady-state performance

One of the recorded situations arising during lake trials had shown the engine to have settled with both speed and torque below the demanded values. Attempts to increase speed and load by adjusting demanded values has proved futile. The following explanation was postulated (with the benefit of hindsight).

In effect, the torque produced at a particular steady-state speed for an engine having a common rail fuel system is determined as a function of fuel control shaft position and fuel pressure. In the event that control of pitch changes from combinator demand to load control, the fuel shaft setting will always tend to be returned, under the influence of the load controller integral action, to the reference value associated with the function generator FG3 in Fig. 4. However, in the event that fuel pressure is low, the torque generated by the engine (which at steady state is equal to the propeller load) will, in practice, be lower than the anticipated design value.

Now, if at some stage during the acceleration transient the governor senses that its torque limiter setting is exceeded, it will hold the actual control shaft setting and prevent further acceleration of the engine. It should also be noted that a similar action could be initiated from the scavenge pressure fuel limiting device. Under such governor limiting circumstances the actual fuel shaft position will be substantially higher than the load control reference, since the governor reference limit is set higher than the load control reference. The load control system will therefore act to reduce pitch to the limit of its capability. However, the indicated governor overload may still not necessarily clear because of the excessive shaft position required to compensate for low fuel pressure.

Under this condition, increasing the load control reference by a small amount in an attempt to increase load (as was tried during the trials) will not produce an increase in engine torque, but if anything will only reduce speed to restore propeller torque at the increased pitch ratio equal to available engine torque.

It was subsequently discovered during the ship trial that fuel pressure had in fact been low on this occasion, and upon rectifying the fault it again became possible slowly to accelerate the engine up to its rated load and speed.

In addition to postulating an explanation of the recorded events, this discussion emphasises the complex interactivity of the various sub-systems and the dependence on accurate control of fuel pressure.

Detailed description of transient trial data

Upon considering first the transient response to a step increase in lever position from notch 0 to notch 8 (see Fig. 7), the following observations can be made. The engine speed and load increase reasonably linearly from 60 rev/min and 55 kNm to 65 rev/min and 165 kNm over the first 10 s of the transient.

The load then increases at a nominally uniform rate with no change in engine speed, until the turboblower surge line is intersected at 65 rev/min and 465 kNm, 20 s into the transient. Thereafter speed and torque drop rapidly to 50 rev/min and 400 kNm. After a further 18 s of surging at an average torque of 400 kNm the engine speed has only increased to 64 rev/min. Power is then reduced and the surging terminated, whereby the transient response tails off as shown.

These data yield some extremely interesting interpretations of the actual system behaviour. Most significantly, it appears from this result that under certain operational conditions the load control system fails in its capacity as a load limiting device. At notch 8 it is anticipated that maximum torque of 600 kNm should be produced at 90 rev/min (when the ship has finally accelerated up to speed). The engine, in fact, does not remotely approach this speed, because pitch is apparently brought on at such a rate as to absorb all available engine torque, so limiting engine speed to 65 rev/min until surging eventually intercedes.

During the transient, torque is seen to climb well in excess of the reference load limit of 190 kNm at 65 rev/min and continues to increase for a period of some 8 s before surge occurs. However, throughout this period pitch does not appear to be reduced by the load control system to restore normal engine acceleration. Neither does it appear that the pitch control system intervenes subsequently, and only the advent of ship acceleration with corresponding decrease in propeller torque (at assumed fixed pitch) appears to allow a margin for engine acceleration during the surging period. This result suggests that the combinator schedule was in fact retaining control of pitch during the entire transient, in which case the maximum ahead pitch ratio could presumably apply, especially during the later phases of the transient.

Trials were also conducted in which the engine was accelerated by gradually moving the control lever through its range from position 0 to 8. From the recorded results it was observed that even under this condition the actual load line was still above the limit at which the pitch control system should be operating to reduce load. The actual load line was, however, below the turbocharger surge line, so that in this case the engine did eventually come up to rated power.

However, it was also stated in the trials report that acceptable engine acceleration could be achieved in response to large stepwise increments of control lever, if the load control setting was reduced to 90%. This did suggest that the load control system was not in fact completely inoperative, although from the recorded results a problem obviously existed.

Possible explanation of transient recorded behaviour

Careful consideration of the interactions between the various system components and analysis of potential sequences of events eventually led to a plausible explanation of the events described above.

As the engine demand speed was increased and acceleration commenced, the scavenge air limit may have activated (perhaps because of unsatisfactory turbocharger performance) and this would have prevented any further increase in control shaft angle. However, if the control shaft had not yet moved above the reference line associated with the load control system, the load control system would not have known that the engine was incapable of developing more torque. Hence pitch would have continued to increase, heading towards its demanded value. There would, therefore, be a tendency for the engine to slow down as the applied load exceeded developed torque. However, such a speed decrease would tend to allow more fuel per stroke because of the nature of the common rail fuel system. This, in turn, would tend to develop additional torque, and a fall in engine speed would be prevented or at least restricted.

Meanwhile, the turbocharger would be slowly accelerating, providing more boost pressure, and gradually releasing the scavenge limit upwards, allowing the fuel control shaft angle to increase gradually, thus supporting a load increase at constant engine speed. Since the fuel pressure would be high under these conditions, corresponding to the demanded value at a higher engine speed, the fuel rack may still not have advanced beyond the load control reference line, and the load controller would have continued to impose a higher load than was intended for that particular engine speed. Hence the load would always be maintained just equal to the developed torque, increasing gradually upwards into overload, but never allowing sufficient acceleration margin to increase the engine speed.

Although it is not certain that this explanation is valid, it does fit the observed facts and clearly illustrates the potential danger which can arise in the event that fuel control shaft limitation occurs without also invoking appropriate action at the load control system.

Discussion

Dr P. S. KATSOULAKOS (Lloyd's Register of Shipping): First, I would like to congratulate the authors for presenting a very clear description of the systematic approach they adopted to solve a well defined and interesting engineering problem. The main tools used in the investigation include traditional mechanical design concepts, diesel engine performance aspects, control scheduling and simulation principles, illustrating the necessity to combine expertise from different engineering disciplines in order to arrive at a near optimum solution.

The investigation reported in the paper was instigated by a contractural requirement to improve the acceleration performance of the 76JC4R Doxford engine coupled to a CP propeller. The solution outlined in the paper, which involved a novel turbocharger accelerator, has obviously produced the necessary improvements. However, I suspect that the initial problem would have been reduced significantly if the new generation of high-efficiency turbochargers used today in both slow- and medium-speed diesel engines were available. As the peak efficiency of these turbochargers can reach 72%, the turbocompounding arrangement, in which the excess energy at high loads is used in a power turbine, could provide improved acceleration performance in slow-speed diesel CPP propulsion systems. Could the authors comment on the alternatives, and are there any current applications for the hydraulic turbocharger accelerator?

The systems approach adopted in the investigation, including the behaviour and interaction of control components, is very interesting. The resulting modified control system described in Fig. 8 could provide the main principles for the design of control systems in similar applications. To accommodate the current difficulties of engines operating on variable quality fuels it may become useful to incorporate an ignition limiter in parallel to the scavenge limiter. This could be based on a criterion for the maximum ignition delay period at steady-state conditions, and its dependence on the charge air temperature. Do the authors believe that ignition difficulties could affect the performance of control systems during manoeuvring? Would an additional scavenge limiter increase the complexity of the system or produce any practical problems?

With regard to some of the technical details of the control system could the authors explain why the fuel pressure function was not taken directly from N_e rather than from N_i, hence avoiding the need for a pressure transducer. Also did the authors identify any of the control stability problems during their investigation?

Finally, apart from the value of simulation techniques in the study of the unconventional machinery configurations, which I fully support, another application of simulation using detailed thermodynamic models could be in condition monitoring applications. Could the authors comment on whether they think that the accuracy and flexibility of the control systems such as those described in the paper could improve if they were linked to the appropriate monitoring systems and simulation models?

J. P. P. PILLAI: I would like to commend the authors on a very interesting paper—another intelligent design approach, making the competence level of seagoing engineers more redundant!

I find the use of the hydraulic turbocharger accelerator quite interesting. This is particularly relevant, especially as today a good many ships are sailing at speeds well below their design value at what is considered as 'economical speed'.

My questions are:

1. Do the authors consider it practical to retrofit existing slow-speed diesel engines with hydraulic turbocharger accelerators.
2. What would it cost for a typical retrofit?
3. Have any existing engines in service been retrofitted with such accelerators, in addition to the one fitted on the engine described in the paper?

P. BAK (MAN–B&W Diesel A/S): With reference to Fig. 7, it makes one wonder how it would be possible to start the engine with a fixed-pitch propeller, when the engine cannot perform with a CP propeller.

A CP propeller requires at neutral pitch about 30% of the torque a fixed-pitch propeller would require. At idling speed the propeller torque for the CP propeller should be expected to exceed 100 kN, and it seems possible that a simple delay of the pitch increase would have allowed the engine to pick up speed.

The requirement for the engine was to accelerate from idling speed to 85 rev/min in less than one minute; there should then be no reason for the pitch to increase from neutral to full pitch in less than one minute, one and a half minutes would probably be adequate, and for manoeuvring it should be possible to reduce the pitch to neutral in about 10 s.

Anyway, the authors have presented an interesting paper, which shows how to improve the airflow to the engine, reveal the control system and make it work.

M. F. CRAIG (Marine Design Consultants): In their concluding remarks the authors comment that the analysis techniques described will increasingly come to be used as standard for all design and trouble-shooting activities.

Would they like to expand on this statement and give examples where dynamic modelling and simulation techniques can be used, mentioning the advantages to be gained?

Dr K. H. AITKEN (YARD Ltd): The authors are to be congratulated on their paper which clearly addresses the combined problems of engine/turbocharger and control system performance. It is refreshing to hear of solutions to these familiar problems which have been applied to a 'real' installation.

As the authors are no doubt aware [from their Ref. (3)] YARD has considerable experience of simulating ship propulsion systems and we have encountered similar problems on a number of diesel/CPP installations with demand speed torque limits and/or pressure-based fuel limiters in the engine governors. The problems stem, in our opinion, from a lack of coordination between the engine builder (who generally specifies the governor/fuel limiter) and the controls contractor or CPP supplier (who generally specifies the load control function) at an early stage in the system design.

The authors were fortunate in the case study that an electronic governor had been fitted which could be (retrospectively) integrated with the load control system to ensure that load limiting would precede fuel limiting. We have on a number of occasions, where mechanical governors had been specified, had to resort to less elegant solutions incorporating an engine speed error based load control function in addition to the fuel rack based function. As an alternative, a double ball-head governor, with fuel limiting (and load limiting) as a function of achieved engine speed instead of demand speed or pressure will ensure satisfactory load control, but this does not necessarily give smokeless operation and is resisted by some engine builders.

In our experience, load control, particularly using a pressure signal, can give rise to pitch hunting. Would the authors care to comment on this given that no transients from the case study are available for inspection?

Can the authors provide any further information on ship and engine acceleration performance before and after both the turbocharger and control system modifications? In retrospect, do the authors consider that the modifications to the load control system would have been sufficient to give adequate ship performance without the hydraulic turbocharger accelerator.

In addition could the authors comment on:

1. The costs of the turbocharger modifications.
2. The effect on fuel consumption.
3. The applicability of the turbocharger modifications to medium- or high-speed engines.
4. The accuracy and reliability of pressure measurements for load control.

G. A. WEBB (Vosper Thornycroft (UK) Ltd): I would like to congratulate the authors on an interesting paper. They have managed to present an example of a sometimes complex and involved subject in a comprehensible form.

I was intrigued to see the use of a hydraulic turbocharger accelerator. Did Dr Ørbeck consider using an electric drive for this and do the authors know of many other applications of accelerators to turbochargers? How much of an impact do the authors feel the accelerator will have in relation to cost and reliability, and in changing the transient engine performance?

I would concur with Dr Fowler's comments in regard to the importance of using computer simulations to highlight these sort of interactive sub-system responses. As Dr Fowler points out, the use of such simulations in testing and developing control systems is now standard for warships. However, to do this with sufficient accuracy a considerable amount of transient data on the propulsions system is required for the database to reduce the amount of assumptions that may have to be made on unknown data. Acquiring this data even for warship applications is often a problem, so for merchant ships this would be compounded.

Would the authors agree that manufacturers of all units of the propulsions system would be greatly assisting the customer, and the propulsion control designer, in knowing the unit's transient performance and transfer function, ie its equation of dynamic performance. With this information the simulation can be confidently used to iron out any complex interactive control system problems that might otherwise have expensive consequences if allowed to filter through to the 'onboard' control system.

N. J. SMAIL: While the speed and pitch demand schedules for bridge control of the propulsion system are intended only to represent the steady-state condition, their programming can nevertheless be used to ensure compliance, at least under that condition, with the requirement that pitch must be shed at speeds above 80 rev/min so that constant bmep is maintained. The programme given at the authors' Fig. 5, however, does not appear to be designed to do so.

There may be some very good reason for this that I have failed to see, but a simple cam modification could be made so that beyond about 65% lever position ahead, as engine speed continues to rise to its maximum 96 rev/min, the propeller pitch is reduced progressively through the remaining level travel. The engine torque load would thus be limited and hence the constant bmep requirement met. The programme in Fig. 5, where $p/d = 0.9$ is reached and then held throughout the upper part of the demand speed range, with the propeller then behaving like a fixed-pitch one, is a recipe for black smoke (when, that is, cylinder pressure is critical).

A modified programme for bridge control demand would eliminate steady-state overload, leaving the load controller and the other 'long-stop' devices to take care of the transients.

Figure D1 shows the constant bmep diagram and Fig. D2 is the revised bridge control programme which could be used to achieve it.

Prof. J. R. HEWIT (Loughborough University of Technology): This is a very interesting and readable account of the application of control technology to the retrofitting of a power-

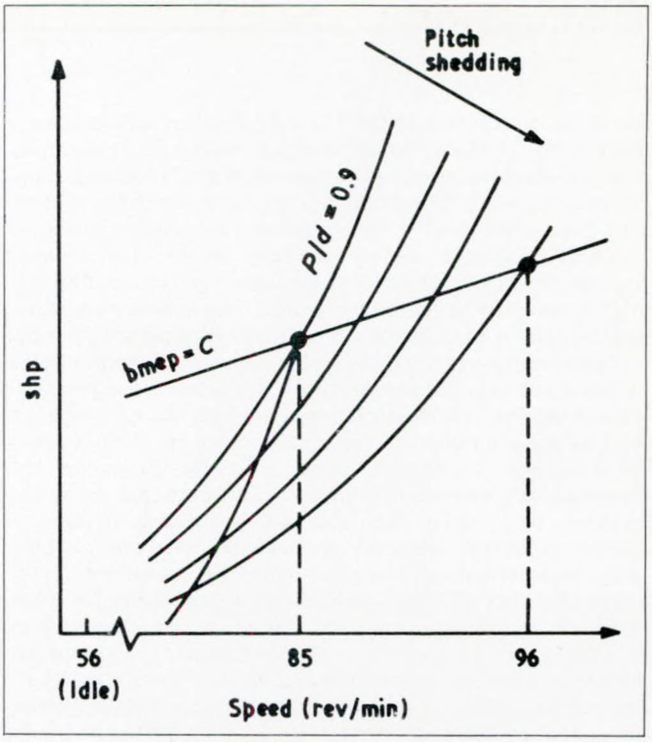

FIG. D1: Constant bmep diagram

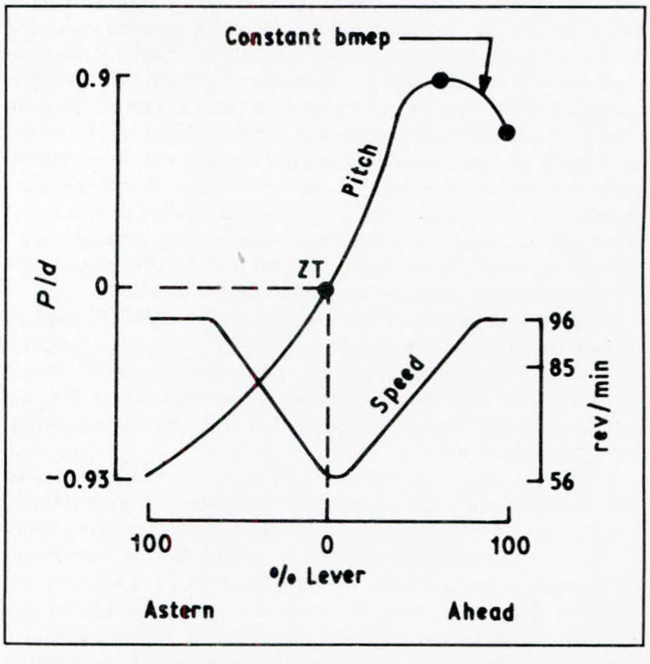

FIG. D2: Revised bridge control programme to achieve constant bmep

boosting system to an otherwise underpowered propulsion system. It seems to me a great pity that the sophisticated modelling and control theoretic principles mustered in this paper were not utilised during the initial ship design process to ensure that an adequate propulsion system was fitted a priori.

The authors say that a solution which further increased the demand for electrical power was to be avoided if possible and that it was decided to design a novel hydraulic turbocharger accelerator. They then go on to say that the hydraulic motor is supplied with oil from a swash plate pump driven by an electrical motor. Could they please explain this.

D. GLENNIE (University of Newcastle upon Tyne): The modifications to the propulsion control system, as proposed by the authors, highlight the importance of preventing overload conditions by ensuring that the propeller load control system is activated before the governor (torque and scavenge pressure sensing) fuel limiting devices. Having re-designed the system to implement this objective, do the limiting features of the governor system become redundant?

Authors' reply

The authors would first like to acknowledge their appreciation of the contributions made and the interesting points thereby raised for discussion.

Dr Katsoulakos suggests that the latest generation of high-efficiency turbochargers could prove efficacious in alleviating the acceleration performance of slow-speed two-stroke engines. While it is true that the efficiency is such that there is sufficient energy available to drive a power turbine, this situation is usually associated with the upper range of the power spectrum at steady-state operating conditions. It is by no means certain that the improved efficiency under these conditions implies sufficient margin at the 'off design' condition to provide the necessary boost required to meet the stringent acceleration specification which was encountered in this case.

The use of turbocompounding in the upper part of the power range offers the added advantage of a smaller turbine area at low power when the passage of the gas to the additional turbine is closed. It is therefore true that the light-load performance is improved considerably. However, it is unlikely that two-stroke engines with constant-pressure turbocharging could operate without an auxiliary fan and the hydraulic accelerator is best considered as a superior alternative to the auxiliary fan.

There are no other applications of the hydraulic accelerator at present apart from the one described in the paper. The reason for this is probably that fairly major design changes would be required for engines fitted with an auxiliary fan to draw the full advantage of the accelerator.

Turning to the control system, the point is raised that the electronic governor provides an extremely flexible device when implementating modifications or additions to the system. It would therefore be relatively easy to incorporate an additional feature such as the proposed limiter based on prediction of ignition delay effects, providing a suitable signal can be derived, which is representative of the condition to be controlled. Charge air temperature is mentioned, and this could be implemented in a fairly simple and practical way. However, it must be remembered that the control system is only capable of (a) reducing load if necessary and (b) restricting fuel flow to the engine. It does not directly influence the timing and may not necessarily produce particularly salutary effects with respect to ignition delay effects.

It is not believed that ignition problems in themselves would constitute problems for the control systems, as described in the paper.

The question is raised concerning the fact that fuel rail pressure was measured and used to modify a nominal torque limit, rather than following an apparently simpler option of referencing the fuel rail pressure function generator from achieved speed rather than demand speed. The option was considered, but in practice the fuel rail pressure is set by a mechanical control system associated with the engine control linkage. This involves a cam and mechanical spill valve operated directly from the demand lever. There is therefore no convenient mechanical output corresponding to achieved speed, which can be used to replace the original system.

A secondary advantage also accrues to the transducer-based system. Pressure control by the mechanical spill valve is not necessarily particularly accurate, especially if filters are partially blocked for example. The pressure transducer will detect any such anomolies and adjust the limits accordingly.

The issue of control system stability is always an important and potentially difficult one. The question from Dr Aitken elsewhere in this discussion offers an illuminating example in this respect.

Since on this occasion a simulation facility was not available for evaluating modifications before installation, it was decided to deviate as narrowly as possible from the original design. In addition to simplifying the task at hand and minimising cost, this approach was considered less likely to invoke stability problems, since the original system was known initially to be stable. By incorporating the governor limits without generating error summations and true closed feedback loops, it was possible to produce a system which was not too dissimilar from the original stable system.

Finally the point is raised concerning the potential for more general applications of simulation. The case study presented offers one typical example, and Dr Katsoulakos draws attention to another important area involving detailed simulation of engine thermodynamic performance. Such studies are well advanced in several industrial and academic institutions. For example, the University of Newcastle upon Tyne, in conjunction with Lloyd's Register of Shipping, is currently investigating the application of such techniques to the problems of condition and performance modelling of diesel engines.

Essentially the modelling philosophy entails the synthesis of subsystem blocks into a meaningful representation of the total system. The technology is already well developed and hopefully the activities described in this paper may contribute another small step in that direction.

Mr Pillai asked three questions relating to feasibility and cost of retrofitting, which should be seen in relation to the common practice today of operating ships well below the rated power.

It is quite practical to retrofit existing slow-speed diesel engines with the hydraulic turbocharger accelerator. In fact, in the case which was described in the paper, the unit was designed after the test-bed trials and fitted just prior to the sea trials.

The cost of the retrofit described in the paper, which was to an engine with one large turbocharger, was £10 000 to £15 000. No other engines have been fitted with this system, as explained in our reply to Dr Katsoulakos.

Mr Bak has asked a very relevant question enquiring why it is apparently so difficult to accelerate a CP propeller system when common experience with FP propellers appears to be relatively straightforward. It is probably true to say that if the response of the system is deliberately slugged using rate limiters in the demand signals, then the problems identified in this study would be alleviated. In practice this is the way in which the ship was being operated, prior to making the modifications, since during accelerations the operating staff had resorted to a practice of moving the bridge command lever slowly, step by step, at a rate which they had determined to be acceptable by experience.

The incorporation of rate limiters, which in effect convert step inputs from the operator to ramp inputs at the controllers, is common practice with fixed-pitch propeller and also some controllable-pitch propeller installations. While being entirely adequate in many applications, this approach is not necessarily optimal in terms of achieving the maximum rate of ship acceleration. To achieve this objective it is generally necessary to drive the engine as hard as possible throughout the transient, subject to limitations imposed by the engine operational envelope. However, this is done at the expense of complicating the control arrangement, as was the case here. Nonetheless, such systems have been successfully designed and are commonplace amongst warships.

The behaviour apparently experienced during service, as illustrated by Fig. 7, is in fact not uncommon when compared with the results obtained during simulation design studies, and potentially occurs even with four-stroke engines if propeller pitch is increased in an uncontrolled manner. The problem is that the engine is initially running at idle speed with relatively low load and boost pressure, when suddenly pitch begins to increase with the ship virtually 'still in the water'. Hence load is increasing without necessarily implying an increase in shaft speed. This situation is not of course possible with a fixed-pitch propeller, where load only increases as a result of a shaft speed increase. Also, because of the shape of the propeller characteristics, relatively high load torque can result at low advance speed, even with modest shaft speed, and there is little or no torque margin left to accelerate the rotating parts, since the propeller load increases as quickly or even faster than the engine developed torque.

The above explanation represents an attempt to describe these dynamic events intuitively, but it will be apparent that the only effective way to present a quantitative appraisal is by resorting to comprehensive dynamic modelling of the system.

It is worth noting that the problems identified above are potentially aggravated on this particular installation since the propeller can actually assume a pitch ratio of 0.9 during the transients. This is high compared to the normal full power (equivalent fixed pitch) propeller which would have a pitch ratio of only 0.78.

Mr Craig has referred to the wider implications of simulation philosophy. The paper has illustrated, by way of example, a typical consequence of system synthesis, which is based predominantly on consideration of essentially steady-state design characteristics of key components. This traditional approach to system design is appropriate in many situations, especially if previous experience exists and/or in cases where dynamic performance is of minor consequence.

It is probably true to say that most designers will intuitively follow the 'steady-state' approach, often relying on satisfactory 'blending' of component characteristics at the commissioning stage, using essentially empirical methods to establish controller settings etc. However, as has been demonstrated, the results of the intuitive steady-state approach do not always lead to the optimal solution, and unforeseen problems can arise, especially when one or more components in the system possess relatively unusual characteristics.

The philosophy of using simulation techniques, as referred to in the paper, has generally evolved from the aerospace and process industries where extremely complex, interactive and expensive subsystems must be optimally synthesised. Response times can typically vary from milliseconds to hours or even days, often leaving little flexibility in terms of empirical tuning. Additionally, there may be disastrous consequences, including danger to personnel, in the event of failure or poor performance in a dynamic environment.

For this reason simulation has evolved as a means of avoiding unforeseen problems, which might otherwise not come to light until, during or after commissioning. For example, the problems associated with the propulsion plant of *Canadian Pioneer* would probably have been identified had a comprehensive simulation been implemented at the design stage.

It should, however, be stressed that the results of any simulation exercise are only as good as the data input to the model and, in practice, it is often the case that refinement and modification of the model will often arise during commissioning, as more empirical and validatory data become available.

A secondary but crucially significant advantage accruing to the simulationist is that the designer usually acquires a deeper insight into the characteristics and interactions which occur between various items of equipment. This develops naturally when the designer actually sits down and starts defining these relationships mathematically. Such an insight is obviously most advantageous, particularly when troubleshooting in a dynamic situation. In such cases engineers inevitably find themselves attempting to unravel cause from effect in a situation where, without some form of structured approach, it is often difficult to know where to start.

With regard to typical applications of simulation techniques, it is reasonable to assert that these requirements potentially arise whenever a complex, interactive or non-linear system is being designed or synthesised. If novel concepts or unfamiliar components are involved, the value is further enhanced.

In addition to the propulsion control system problems, as identified in the paper, typical marine applications arise in dynamic positioning, underwater operations, development of auxiliary systems and prediction of environmental influences and responses. Numerous applications arise in the offshore industry, examples being submersible deployment, behaviour of umbilicals, pipeline studies, trenching, mooring and crane dynamics. More generally a range of applications is found in mechanical, electrical power and process engineering and even biotechnology and modelling of economic systems.

Recent advances in software and the availability of relatively low-cost, powerful and flexible digital computers has brought the technique within the grasp of a substantial professional clientele. It is therefore suggested that the rewards of investing the necessary intellectual effort will be profitably returned for engineers engaged on applications related to any of the above areas.

Dr Aitken has added some interesting and informative comments derived from his organisation's experience in the field. The problem of pitch hunting was not to our knowledge experienced in this particular propulsion system; certainly there have not been any reports from the ship operators suggesting that this was a problem. It will be appreciated that the pressure-operated device is normally activated only during transients, and would not be expected to prevail during steady-state operation. It is therefore possible that low-level, short-duration periods of hunting may have occurred without being regarded as significant.

Certainly this is an interesting area for investigation, since when the pressure control limit is activated, an alternative closed-loop feedback system is potentially created involving the dynamics of the turbocharger, the governor, the engine and the pitch control system. The usual control system constraints associated with loop gain, stability, hysteresis etc. will obviously apply here, but the only realistic approach to defining the system precisely is either through simulation or/and copiously instrumented ship trials.

Dr Aitken also refers to the accuracy and reliability of the pressure-measurement system. Again this only affects the transients and minor errors should not prove too problematic. It is worth noting that the signal from the transducer was filtered, and obviously choice of transducer would centre upon robustness and reliability, even if this incurred some sacrifice in terms of sensitivity. If used in a feedback system the dynamic characteristics of the transducer and signal processor must, of course, be correctly incorporated.

Before leaving this point, it is worth stressing that the governor limiters were actually implemented by lowest-select circuits without generating true closed-loop feedback systems incorporating error summation. It will be appreciated that this philosophy applied to both pressure and fuel shaft limiting functions in the governor, and was deliberately adopted because of uncertainties regarding possible stability problems in a true feedback system. However, it will be apparent that the actual load control system does constitute a true feedback system and the normal constraints will therefore apply. Without resorting to simulation these effects could not be fully quantified, but fortunately, on this occasion, performance appears to have been quite acceptable.

Unfortunately, recorded data in the form of strip charts showing pitch, rev/min etc. as functions of time were in general not available from the vessel (a feature which considerably complicated the study). Even the critical data which were used to reconstruct the events shown in Fig. 7 had in practice only been recorded, in passing, as part of a combustion system study. It is therefore not possible to provide quantitative data on the transient performance before and after the retrofits, as the results were accepted purely by observation during trials.

With regard to the efficacy of the modified load controller as a 'stand alone' modification without additionally incorporating the turbocharger assister, this appears most unlikely judging from initial tests of the engine on the test bed, when even with manual loading of the engine, suitable acceleration was not attainable without the hydraulic assister.

The load controller by definition should protect the engine from adverse effects, but to obtain the specified rate of acceleration with this particular two-stroke constant-pressure-charged engine, it is considered that adequate scavenging must be ensured by an external device.

With respect to cost (a point raised also by Mr Pillai), the retrofit described in the paper was between £10 000 and £15 000. The cost depends on a number of factors. The hydraulic pump(s) and motor(s) are standard units and therefore relatively cheap. The clutch, couplings and support plate may, however, have to be designed for the application and this results in design costs as well as added production costs. Lastly, the make and type of turbocharger are important. The turbocharger described in the paper could be easily modified as it was lubricated from the engine system and therefore did not require a rotor-driven lubricating oil pump. This made access to the rotor locking arrangement easy. Retrofit cases must, in conclusion, be treated individually but the cost is probably commercially acceptable.

Concerning the implications for efficiency, at full power the system is declutched and it therefore has no effect on fuel consumption. At light load the system provides the engine with more air. Since constant-pressure turbocharged engines in particular tend to be short of air at low powers, an improvement in combustion and therefore in fuel consumption can be expected. Additionally, the hydraulic drive to the turbocharger gives a constant-pressure turbocharged engine better acceleration than the corresponding pulse turbocharged engine. Considering the superior full-load performance of the constant-pressure system, an all round improvement in fuel consumption is therefore possible.

Turning to the question of high-speed applications, Bentaxia hydraulic motors are available for speeds over 10 000 rev/min. This means that turbochargers of 20 0000 rev/min maximum speed can be assisted up to half speed. The corresponding engines comprise all marine medium-speed engines and also some engines rated above 1000 rev/min.

Finally the question of pressure measurement has already been addressed earlier in the context of scavenge pressure measurement. Robust bonded strain gauge transducers were also used for the fuel pressure measurements, and these appear to have provided both high accuracy and acceptable reliability.

Mr Webb asks if an electric motor was considered instead of the hydraulic motor. The turbocharger manufacturer had experience with a much earlier system which used an electric motor. A two-pole induction motor would be the best proposition and with a 60 Hz supply this can reach a speed of slightly less than 3600 rev/min. This compares with the speed of 4000 rev/min for the hydraulic motor. A step-up gear box could be used, but this is a further complication. The electric motor would have had to be supported separately on account of its size and weight, whereas the hydraulic motor could be supported on the turbocharger.

Finally, the electric motor would have influenced the dynamic behaviour of the turbocharger rotor whereas the inertia and mass of the hydraulic motor is insignificant. A hydraulic motor was therefore chosen. It should be pointed out that the hydraulic motor could be supplied by an engine-driven pump, thus making the engine self-contained. However, for a retrofit it is easier to use an electrically driven pump.

It is our opinion that the hydraulic accelerator could have a considerable impact on diesel engine design. Both two- and four-stroke engines could be universally designed for constant-pressure turbocharging and yet achieve excellent acceleration. Thus improved fuel consumption as well as simplicity of exhaust pipe and turbocharger design could be achieved with no sacrifice in performance.

Mr Webb also refers most appropriately to a problem which ultimately faces all simulationists: how to retrieve adequate data to construct and validate the model and so improve fidelity. This requires carefully planned and well instrumented ship or shop trials, with data collection certainly extending well beyond the familiar requirements associated with traditional merchant ship practice. With regard to the characterisation of subsystems, it is rather unusual to encounter situations in which complete dynamic representation is available from manufacturers in the form of suitable mathematical models of their equipment. Ideally such models must incorporate at least those dynamic features which are directly significant within the frequency range of interest. The models must also include non-linear and discontinuity effects associated with the physical equipment.

The authors agree absolutely with Mr Webb that the interests of the system designer, the user and ultimately the component manufacturer would be well served by paying closer attention to this important area of specification.

Mr Smail correctly draws attention to the fact that the pitch schedule combinator cam is not arranged to reduce pitch over the upper part of the load characteristic, with the result that the load control system is left to reduce pitch in order to maintain constant bmep in this region, even under steady-state conditions. This feature was contemplated during the investigation when it was disclosed that the system is designed and installed essentially as a standard package, which has been seen by experience to operate successfully in numerous propulsion systems.

The operation of modifying the cam was considered during the initial phase of the study, although it would not of course have removed the requirement to implement the other modifications described in the paper. This is because in practice the problems associated with this study were not of steady-state origin but were uniquely related to transients in which the load controller is active.

It was concluded that since the system was nominally designed to operate permanently on load control at the upper end of the power spectrum, the additional effort of dismantling the control cams, specifying appropriate reprofiling, machining etc. was not justified within this particular context.

It is agreed that the suggested cam reprofiling would nominally satisfy steady-state requirements, although it should be appreciated that even this would only be true at a particular set of operating conditions with respect to draught, hull fouling etc. One advantage of using load control in this vicinity is that engine loading becomes independent of these factors, which is presumably why the manufacturers have opted for this particular design philosophy.

Prof. Hewitt has supported the arguments raised elsewhere on the potential merits of applying mathematical modelling and simulation techniques to identify and resolve problems at the design stage.

His point regarding the choice of power source for the turbocharger accelerator is also an interesting one, since the power to drive the swash-plate pump obviously imposes additional electrical load, ultimately on the ship's system. However, the alternative to using an hydraulic assister would have required either a separate large capacity electric blower or a motor drive connected directly to the main turbocharger shaft.

In practice, auxiliary blowers and their associated ducting are of lower efficiency than the near optimised turbocharger, and the equivalent electrical load would then be substantially higher. The disadvantages of a directly coupled motor on the other hand are associated with speed and power control (an area where the hydraulic unit is ideally suited) and also bulk in an otherwise confined space.

In practice, using the arrangement as described, air delivery was improved without drawing more electrical power than was required by the original auxiliary fan which the system replaced.

Mr Glennie has raised an important issue which has implications both for the control system designer and for engine designers. Certainly from a steady-state consideration it would appear that if the load control system is adequate in all respects, then the function of the engine-based protection devices is rendered redundant. Indeed the argument may be extended further, since the interaction between the two systems can actually produce a detrimental effect overall, as has been seen in this case study.

The redesigned system has alleviated the possibility of unwanted interactions while retaining a back-up system to protect the engine in the event of, for instance, a failure in one of the hydraulic components associated with the load controller. This is probably the most significant argument for retaining an independent engine limiter, certainly from the perspective of the engine designer, who does not always wish to rely exclusively upon the fidelity of equipment provided by a separate and independent sub-contractor.

However, there is a secondary point to consider. While it is apparent that under steady-state conditions an active load controller will (or should) preclude the activation of the engine limits, this cannot necessarily be guaranteed under all dynamic conditions. It is possible that during the period while the load controller is attempting to reduce load, the engine could dynamically find itself temporarily overloaded. In this case, both the engine limiter and load controller could be activated simultaneously. Intuitively it might be difficult to appreciate how this could come about, but the only safe way to ensure that such a requirement is or is not needed is to either model the system accurately or conduct exhaustive sea trials with a completed system, experimenting with every conceivable operating condition.

At the present time it therefore seems most likely that engine builders will take the apparently safer option and continue to specify on-engine protective features, but further fundamental investigation of these considerations certainly appears justified.