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Development of the Doxford engine from 1960

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SYNOPSIS

This paper concentrates on the technical-scientific aspects of the development of the Doxford engine. The mechanical design of the engine is considered as the technology in this area was well developed by the early 1970s. Thus, the engines in service experienced few mechanical problems. In thermodynamic design the technology was more patchy. Knowledge in turbocharging and combustion improved with the Seahorse engine but the geared engines are handicapped by specific fuel consumption compared with the direct drive engines. The effort spent on the Seahorse engine was partly wasted as it never entered service. The BS 42-100 project was terminated largely due to the high predicted fuel consumption. The direct drive Doxford engines were competitive until the single piston engines were developed for high cylinder pressure and turbocharger efficiency. However, the success of the latest Doxford 3 and 4 cylinder engines suggests that the opposed piston engine has great development potential. This paper finishes by describing recent improvements in the fuel system, the control system and the pistons. These improvements are supported by service results and substantiate the development potential of the engine.

INTRODUCTION

When British Shipbuilders decided to stop building Doxford engines in 1980 it marked the end of an era. The Doxford engine was the last British designed marine engine. It had its heyday before, during and immediately after the second world war and was considered by many to be the best engine in the world. However, in the early fifties considerable problems arose and Doxford Engines Limited was subjected to many changes. The author started work with the company in 1959 and an account of its history from 1960 until today therefore seemed appropriate for this paper. In this period many technical ideas were tried, changes took place and many personalities were involved.

It is still a much debated issue why the Doxford engine went out of production and there are a multitude of opinions on this subject. Crossland gave some reasons why Harland & Wolff preferred the poppet valve design to their opposed piston design about 15 years before the termination of the Doxford engine.[1] Many technical reasons have also been given for the failure of the Doxford engine.[2,3]

To throw some light on the subject this paper will describe the scientific developments which took place at Doxford and their influence on the confidence and thinking within the company. The company took many decisions which in retrospect appear to be wrong but these should be seen in the light of the experience and technical knowledge available at the time. Perhaps inadequate development of this knowledge was one of the main causes of failure.

Finn Ørbeck, who is Norwegian, received his BSc and PhD from Glasgow University. He was employed by Doxford in 1959 and from 1973 to 1980 was responsible for the development of the 58JS3 and 76JC4 engines. He has presented papers to many of the major institutions and started Orian Technology Ltd this year.

MECHANICAL DESIGN

Vibration and shafting design

In 1959, when the author started work with Wm Doxford and Sons (Engineers) Limited great changes were taking place in the company. The turbocharged LBD engines had performed satisfactorily without scavenge pumps and the company was about to launch the P-engines. All but one of these were turbocharged and were designated PT-engines. The one normally aspirated engine was designated a PN-engine.

However, the company was still badly affected by the crankshaft failures which had occurred in the 75LB6 engines and in 1960 Atkinson and Jackson,[4] presented a paper to set the matter straight. This paper gave a good indication of the state of the technology at the time. The paper dealt with crankshaft geometry, manufacture, alignment, torsional and axial vibrations and described the results from strain gauge measurements which were relatively sophisticated. Many remedies were suggested and a drawing is shown of the proposed crankshaft of the six cylinder P-engine which incorporated many of these. Considerable heart searching also took place as to why the company had got themselves into these problems in the first place and attention was focused on the accuracy of predicting shaft vibration and alignment.

The torsional vibration calculations had essentially remained the same since Dr W Ker Wilson had left the company and there was no information on how to design the Bibby detuner which was fitted to all the engines. Measurements of torsional vibrations were therefore carefully carried out for a considerable number of installations and analysed by detailed calculations. The detuner stiffness was measured using static torque tests and the actual value was found to be much lower than the value supplied by Bibby. Much effort went into the calculation of weights, inertias and stiffnesses which could be calculated accurately, and the results of measurements were then used to calculate the crankshaft stiffness. It had been Doxford's practice to use an adaption of Carter's formula,[5] to predict the stiffness of the crankshaft of new designs. Comparison with the results of measurement, however, now showed that the predicted values could be more than 10% too high. The 75LB6 engines were designed to have the seventh order III node torsional vibration resonance peak just above maximum rev/min. However, due to errors in the crankshaft and detuner stiffness this resonance peak occurred 3 – 4 rev/min below maximum rev/min and could reach a stress of 14 N/mm^2, ie high enough to make an important contribution to the crankshaft failures. Subsequent calculations of the crankshaft stiffness were therefore based on a formula due to Dr W Ker Wilson which gave more accurate results and underestimated the stiffness slightly. A method of designing tuned vibration dampers (detuners) was also developed as part of the torsional vibration calculations and the Bibby detuner was replaced by a radically redesigned unit. This was only fitted when required. In addition to much improved torsional vibrations this progress also resulted in considerable savings in the initial costs of the engines. The new detuner is shown in Fig 1. It was described in detail by the author,[6] who also dealt with the question of how to phase the propeller to the engine in the best possible way. Computer programs had been developed for the standard torsional vibration calculations and a program for forced-damped Holzer tables, introduced in 1963, was very advanced for its time. Thus, by this time the torsional vibrations were well under control.

Axial vibrations were also dealt with effectively although these were considerably less important than the torsional vibrations.

Another field of major importance to the reliability of the engines was shafting alignment. In the Doxford engine the combined stroke is divided between the side and the centre cranks. The centre crank radius, therefore, tends to be smaller in relation to the cylinder bore than for the single piston engine. As a result the bedplate tends to be shallow and of low stiffness in vertical bending. This effect is compensated for by the high flexibility of the crankshaft but the 75LB6 engines had bedplates of inadequate strength and stiffness. Atkinson and Jackson,[4] gave an example of the effect of ship loading on the shaft alignment and this could be substantial. Until 1960 there was no clear relationship between the web deflections and the stresses in the crankshaft and it was generally considered to be necessary to measure the alignment of the bearings. North Eastern Marine had developed the NEM line gear for this purpose whereas Doxford used taught wire equipment. In further studies after 1960 a considerable amount of work was carried out to gain a better understanding of shaft alignment. The bending moments on a shaft are due to two separate force actions, ie dead weight and misalignment. The merits of treating the effects of these two force actions separately became apparent. The effect of dead weight had to be obtained by calculation and many considerations were devoted to how to treat a complicated body like a crankshaft for this purpose. It was decided to represent the shaft by a continuous beam with a constant moment of inertia along its length but paying detailed attention to the positions of the individual loads. This gave the bending moment distribution across each of the centre cranks with sufficient accuracy. The stiffness of different centre crank designs was obtained by supporting the shaft in the crankshaft lathe so that Cylinder Section No 1 was overhanging. The bending moment distribution on this part of the shaft could be calculated accurately and the web deflection obtained by turning the shaft. A formula was developed to co-ordinate the two parameters and this could be used for similar shafts. Finally, a set of corresponding alignment and web deflection readings was available for a 75LB6 shaft. This information covered a variety of different alignment

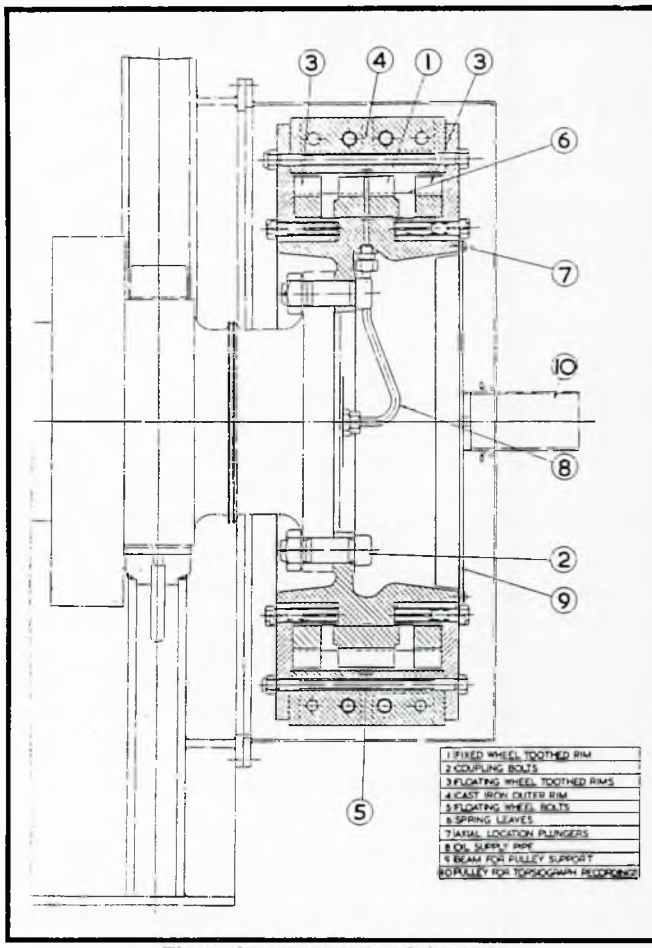


Fig 1: Arrangement of detuner

conditions. It was subjected to a multiple regression analysis to establish the influence coefficients and then the web deflections for the straight shaft could be calculated. These results were compared with the results of the continuous beam calculations to verify the latter and the results were satisfactory. Thus, the web deflections for the straight shaft, ie due to dead weight, could be obtained for any shaft.

The web deflections due to misalignment were obtained for a given alignment condition by subtracting the web deflections for the straight shaft from the actual readings. The corresponding bending moment (BM) distribution was then obtained as follows. At Main Bearing No 1 the BM is equal to zero. The BM at Bearing No 2 can then be obtained from the web deflection of Cylinder Section No 1 as the BM distribution is linear between the bearings. Using the web deflection for Cylinder Section No 2 the BM on the third bearing is obtained and so on. The method was used to establish instructions for web deflections for all Doxford engines in service, as shown in Fig 2 for the J-engines. Later the method was expanded to include the use of strain gauge readings on the intermediate and tail shaft as described by the author elsewhere.[7]

The technical advances in torsional and axial vibrations as well as alignment had a profound effect on the company. Crankshaft failures became a thing of the past and a new confidence in the crankshaft of the Doxford engine was established. Apparently, around 1960, C C Pounder of Harland & Wolff stated publicly that the Doxford engine could not be built with more than six cylinders on account of the torsional flexibility of the crankshaft. The personnel at Doxford believed that Pounder was wrong and a new design philosophy emerged. All

DOXFORD J C ENGINE

Engine No: *435* Engine Type: *67J6* MV: *Ship's name* Date: *26-2-89*

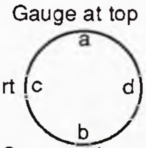
CRANKWEB DEFLECTION READINGS

CRANKSHAFT *FB*
Draft: ford.
Temperature °C:

DETUNER *YES*
aft:
Sea: Lub: Oil in Drain Tank: *LIGHT ENGINE HOT*

Note:- Crank No 1 at Forward end.
All measurements in hundreths of a millimetre

For defelections for straight shaft fs
See Page B.20/A/6



VERTICAL PLANE

Crank No	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*
b = reading, gauge at bottom	*24*	*9.5*	*–3*	*0*	*4*	*49*			
a = reading, gauge at top	*0*	*0*	*0*	*0*	*0*	*0*			
fm = b – a measured deflection	*24*	*9.5*	*–3*	*0*	*4*	*49*			
fs = deflection for straight shaft	*18*	*10*	*13*	*13*	*10*	*18*			
fv = fm – fs = misalignment deflection	*6*	*–0.5*	*–16*	*–13*	*–6*	*31*			

A = fv crank 1: *6* B = fv crank 2 – A: *–6.5* C = fv crank 3 – B: *–9.5*
D = fv crank 4 – C: *–3.5* E = fv crank 5 – D: *–2.5* F = fv crank 6 – E: *33.5*
G = fv crank 7 – F: H = fv crank 8 – G: J = fv crank 9 – H:

Realignment is necessary if any of A B C D E F G H or J fails to lie between
62 and –37
or if fm exceeds 76

HORIZONTAL PLANE

Crank No	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*
d = reading, gauge at starboard	*10*	*1*	*6*	*9*	*9*	*20*			
c = reading, gauge at port	*14*	*4*	*–5*	*–4*	*7*	*29*			
fh = d – c = misalignment deflection	*–4*	*–3*	*11*	*13*	*2*	*–9*			

A = fh crank 1: *–4* B = fh crank 2 – A: *+1* C = fh crank 3 – B: *10*
D = fh crank 4 – C : *3* E = fh crank 5 – D: *–1* F = fh crank 6 – E: *–8*
G = fh crank 7 – F: H = fh crank 8 – G: J = fh crank 9 – H:

Realignment is necessary if any of A B C D E F G H or J fails to lie between
50 and –50
or if fh exceeds 51

Inspected by

Fig 2: Crankweb deflection readings and calculations

the spherical bearing supports, which were characteristic of earlier Doxford engines, were removed in the P-range of engines and the bearings were made with a bigger diameter and a smaller length.[8] Thus, the crankshaft became stiffer and the ninth order III node resonance peak was placed above full speed. Earlier engines ran between the nineth and the seventh order III node resonance peaks.

The highest power of the 67PT6 engine was 10 000 bhp at 120 rev/min but the company was anxious to be able to offer engines of higher power. Jackson,[9] showed that an 85PT6 engine would require a crankshaft diameter of 770 mm to make sure that the nineth order III node resonance peak was sufficiently far above full speed and this was found to be unacceptable. A design was then evolved in which the main journal and two side webs were combined in one cylindrical part, and this became known as the J-engine. This resulted in further improvements in the crankshaft stiffness and the prototype J-engine was a nine cylinder engine of 760 mm bore x 2180 mm combined stroke developing 20 000 bhp at 115 rev/min. The confidence in the crankshaft design was now such that Jackson,[9] envisaged powers of 27 000 bhp or even above 30 000 bhp in 1963. This was confirmed by Abell in 1964.[10] Their confidence

Semi-built crankshafts (metric units 1/100 mm)

Crank	*fs* deflection for straight shaft 1	2	3	4	5	6	7	8	9	Allowable limits: Vertical plane *x*	Vertical plane *y*	horizontal plane *z*	horizontal plane *w*
67J4*	+15	+13	+10	+20									
67J4	+25	+8	+10	+18									
67J6*	+18	+10	+13	+13	+10	+20							
67J6	+28	+10	+13	+13	+10	+20							
76J4*	+18	+10	+10	+18									
76J4	+25	+8	+10	+18									
76J6*	+18	+8	+20	+10	+10	+20				+62	–37	+50	–50
76J6	+28	+3	+23	+8	+10	+20							
76J7*	+15	+10	+8	+20	+8	+10	+23						
76J7	+23	+8	+8	+20	+8	+10	+23						
76J8*	+15	+13	+8	+23	+8	+13	+10	+15					
76J8	+23	+10	+8	+23	+8	+13	+10	+15					
76J9*	+15	+10	+13	+8	+23	+8	+13	+10	+18				
76J9	+23	+8	+13	+8	+23	+8	+13	+10	+18				

Fully-built crankshafts (metric units 1/100 mm)

Crank	*fs* deflection for straight shaft 1	2	3	4	5	6	7	8	9	Allowable limits: Vertical plane *x*	Vertical plane *y*	horizontal plane *z*	horizontal plane *w*
67J4*	+20	+10	+10	+20									
67J4	+28	+8	+10	+20									
67J6*	+18	+10	+13	+13	+10	+18							
67J6	+28	+8	+13	+13	+10	+20							
76J4*	+20	+10	+10	+18									
76J4	+25	+8	+10	+18									
76J6*	+18	+5	+20	+8	+10	+15				+62	–37	+50	–50
76J6	+25	+3	+20	+8	+10	+15							
76J7*	+18	+10	+8	+20	+8	+10	+18						
76J7	+25	+8	+8	+20	+8	+10	+18						
76J8*	+18	+10	+8	+23	+8	+13	+10	+18					
76J8	+25	+10	+8	+23	+8	+13	+10	+18					

The gauging points are between the webs on the centre lines approximately 25 mm from the outer edges.
(*Detuner fitted)

Fig 2: Crankweb deflection readings and calculations (continued)

was completely justified as no major failures ever occurred to the crankshaft of the P and J-engines.

The work on shaft vibration and stresses continued and a computer program was developed for the calculation of the crankshaft stresses due to firing loads, dead weight, inertia forces etc.[11] Much work was devoted to obtaining stress concentration factors for the crankshafts and an improved theory of failure, which suited the step-by-step calculation used in the computer program, was proposed.[12] However, much of this later scientific work was ahead of its time. The calculation methods were not accepted by the Classification Societies and therefore did not result in any practical advantages.

Running gear and bearings

One of the criticisms levelled against the opposed piston engine was the greater number of running gear bearings which require attention. Because the crankshaft has to be turned to specific positions when work takes place on the running gear, only one cylinder section can be overhauled at a time and this criticism became of increasing importance with the J range of engines which were offered as up to and including 10 cylinder engines. The running gear for the earlier J-engines was essentially of the same design as for the P-engines. The designs of the side and centre top end bearing are shown in Figs 3a and 3b. In particular the centre top end bearing of this design gave problems with a maximum cylinder pressure of 64 bar used in the early J-engines. It can be seen that the segmental pad did not

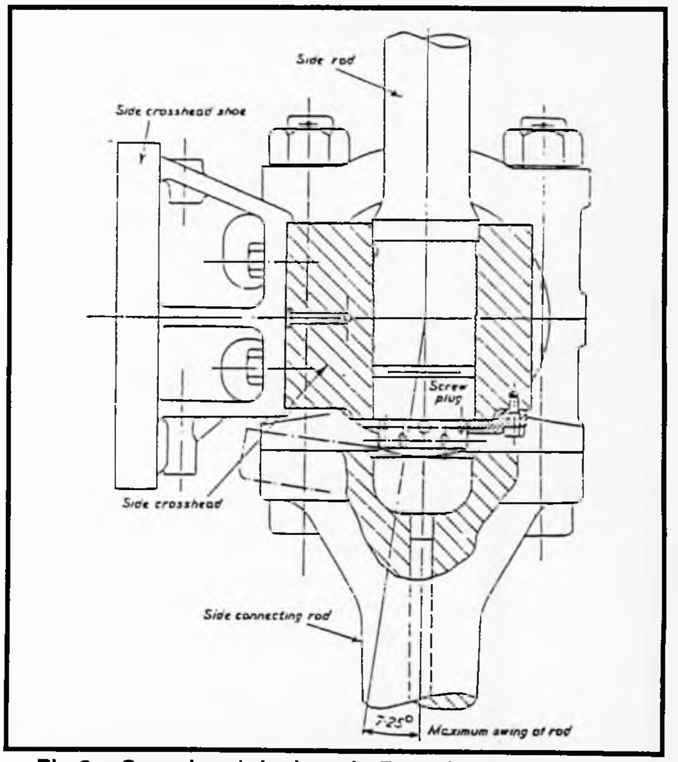


Fig 3a: Crosshead designs in P-engines and early J-engines; side rod and crosshead assembly

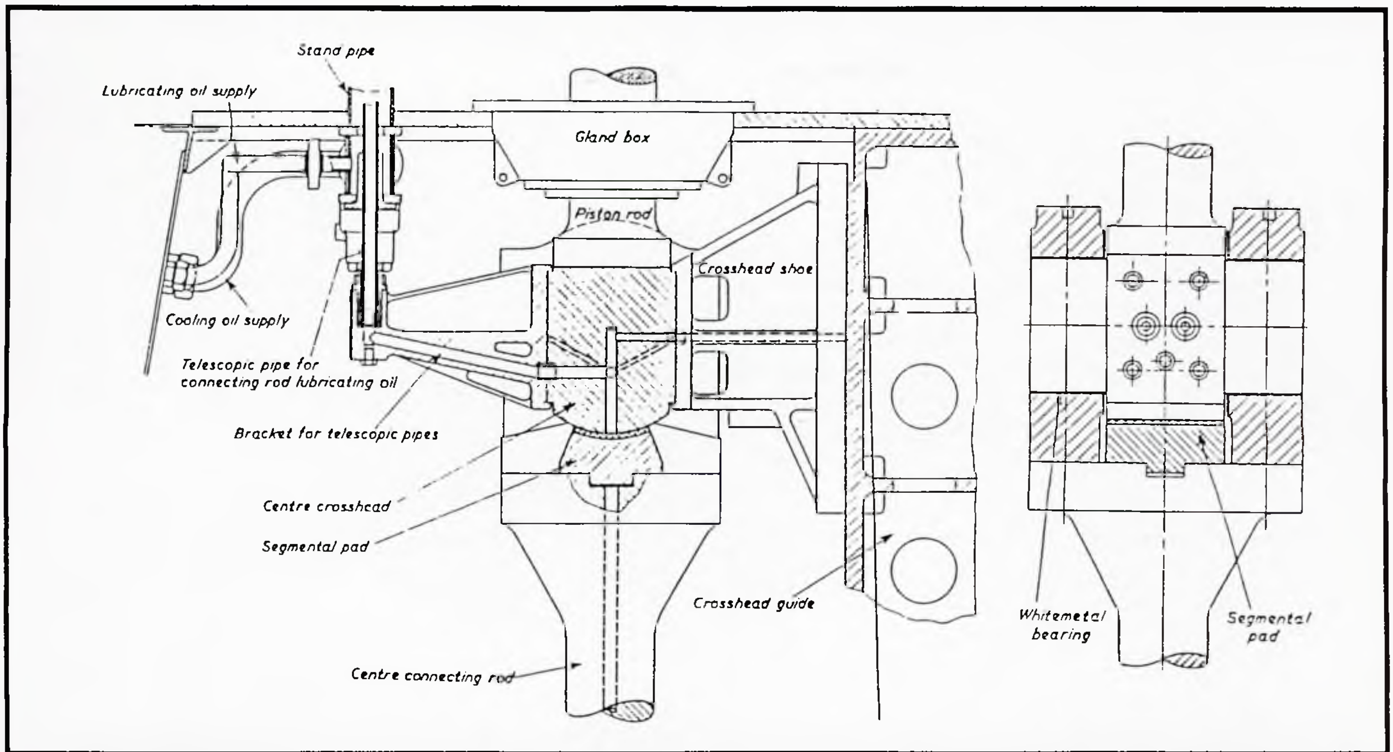


Fig 3b: Crosshead designs in P-engines and early J-engines; centre top end with pad, crosshead and shoe assembly

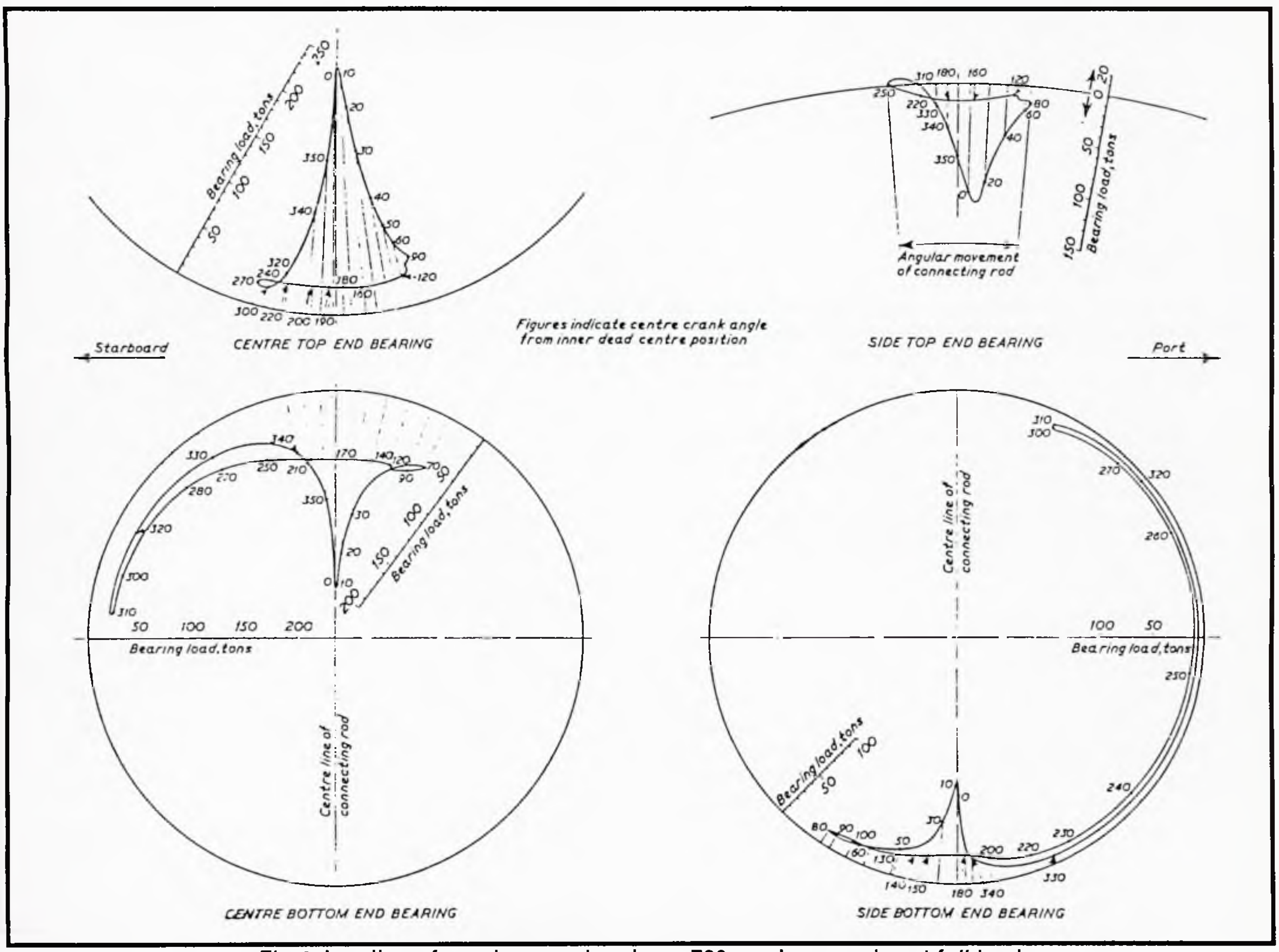


Fig 4: Loading of running gear bearings: 760 mm bore engine at full load

give a good bearing area in relation to the size of the crosshead. Bearing failures became an issue of considerable concern and the BICERA variable ratio piston was discussed for some time.[13]

Generally, the problem was solved by an improvement in bearing technology as described by Taylor.[14] Bearing load diagrams were calculated as part of the design process and a typical example is shown in Fig 4.

The manufacturing technique was improved and as a result thin shell bearings could be introduced in the design of the main journal, side and centre top end bearings.

The bottom end bearings were still built to the conventional design. A new centre top end bearing was evolved in two stages, as shown in Figs 5a and 5b, and considerable improvements were incorporated in the side top end bearings.

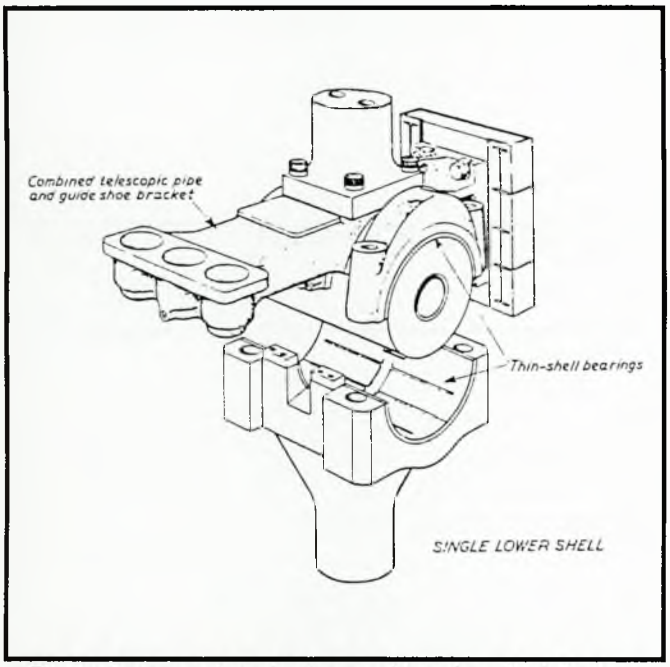


Fig 5a: Improved centre top end bearing; first design

After these modifications the running gear in the J-engines became very reliable and it was possible to increase the maximum pressure.

For the pulse turbocharged J-engines the design maximum pressure was increased to 70 bar and pressures of about 73 bar were used in a series of nine 67J4 engines tested in 1975 – 6.[15] A series of nine 76JC4 engines tested in 1978 – 9 had a design maximum pressure of 85 bar and finally the last Doxford engine to be built, which was tested in February 1981, used a design maximum pressure of 90 bar. The actual pressures used on the test bed for the 76JC4 engines were however 2 – 3 bar lower. The 76JC4 engines used the same running gear as the later 76J pulse turbocharged engines whereas the last engine, designated 76JCR4, used Glacier Sovereign 87 bearing material. It is interesting that it took nearly 10 years before the improvements in running gear design, which were described in 1969,[14] were properly utilised.

Finally, it should be mentioned that a new side rod design was introduced with the Seahorse engine,[16] and this was later used in the 58JS3 engines.

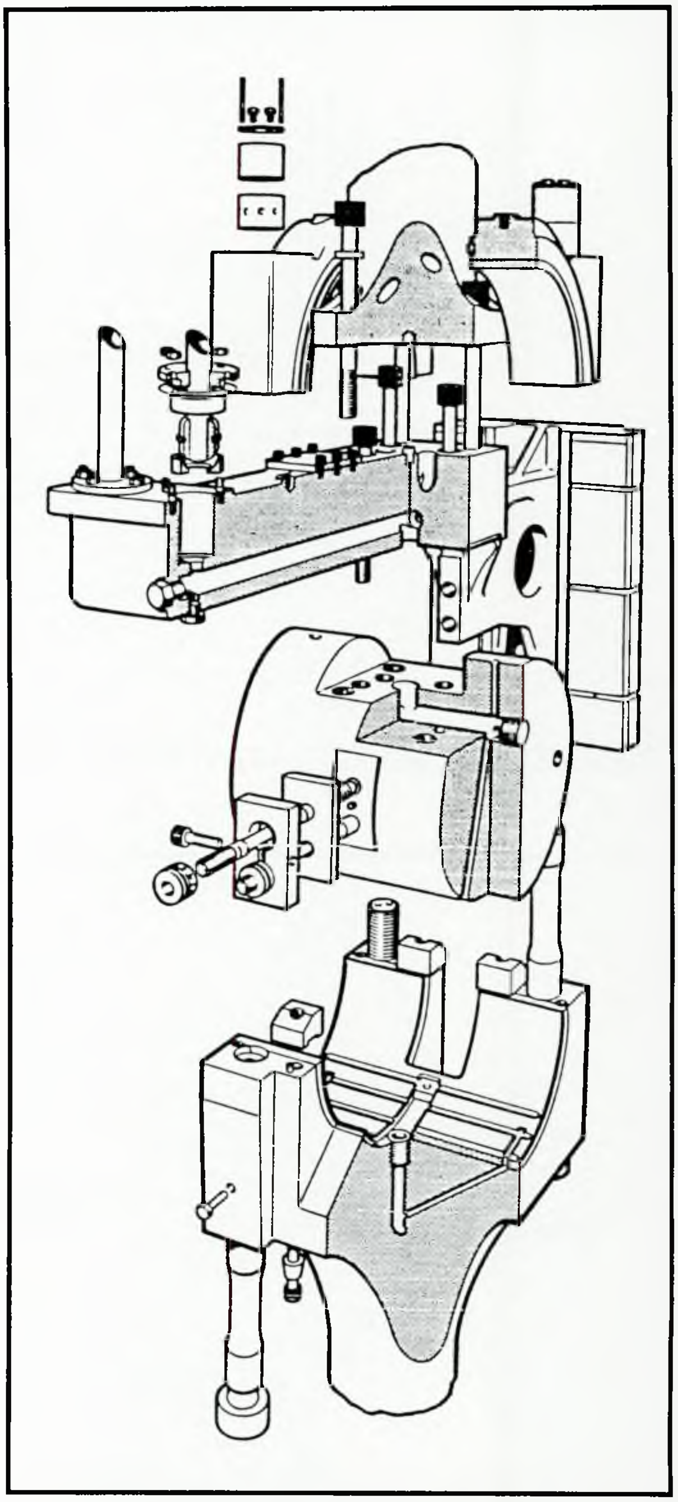


Fig 5b: Improved centre top end bearing; later design

THERMODYNAMIC DESIGN

Turbocharging

The main reason for choosing the opposed piston principle is the large flow areas available for the scavenge of the cylinders offered by this principle. As a result the opposed piston engine offers little resistance to flow through the cylinders and is ideally suited for turbocharging.

Considering the low scavenge pressures used at the time it was decided to use the pulse principle in the first turbocharged Doxford engines and this was so successful that the scavenge pumps were no longer required.

A small electrically driven fan, which was arranged in parallel with the turbocharger(s), gave satisfactory slow running performance.

Jackson,[17] described the above turbocharging system in 1963. He does not, however, mention the auxiliary fan but states that to give the turbochargers enough energy for satisfactory slow running the exhaust ports were opened earlier by giving the exhaust piston a 10 degree crank lead.

The piping arrangements proposed for the J-engines were also described and they are shown in Fig 6.

The six and nine cylinder engines had relatively simple arrangements.

For the other cylinder numbers the arrangements were more complex and less efficient although much thought went into finding firing orders which gave the best combination of balance and turbocharging.

Jackson also compared the Doxford pulse system with the constant pressure system used in the Götaverken engine.

The constant pressure system uses a large gas receiver before the turbine(s) and all the cylinders are connected to this receiver.

As far as turbocharging is concerned all the cylinder numbers and all the firing orders are equal, which is simple. However, it was necessary to keep the added complication of scavenge pumps in the Götaverken engine.

All round it seemed that Doxford had chosen the best system for their engines and the pulse system was retained for the J-engines until the mid seventies.

The calculation procedures associated with turbocharging within the company were limited to the calculation of port area diagrams.

Most of the turbochargers were supplied by Brown Boveri and apart from the port area diagrams, most of the performance prediction was left to them.

In 1963 Jackson mentioned that BSRA had during the past two years engaged Professor Horlock of Liverpool University, and later Professor Benson, to develop a computer program which would enable the exhaust pulses to be calculated and their energy to be determined. Their experiments and calculations had not led to any practical results and under the DSIR rules, which governed BSRA, no results of this research could be given to a foreign manufacturer. Doxford was therefore left in a cleft stick and had no alternative but to rely on Brown Boveri.

Considering that Doxford was the main possible beneficiary of the work at Liverpool University, it would have been much better to have let them control the research.

However, the calculation of port area diagrams had the beneficial effect of drawing attention to the importance of generous flow areas.

In 1967 Butler,[18] described the design of a new exhaust belt which, by virtue of its large flow areas, made it possible to raise the power per cylinder of the 76J engines from 2222 bhp to 2500 bhp. Thus, 20 000 bhp could now be developed by an eight cylinder engine instead of the earlier nine cylinder engine.

Butler also introduced the concepts of the blow-down integral and the instantaneous effective area and from the latter the resultant scavenge integral was obtained.[16] This completed the blow-down and scavenge diagrams as shown in Fig 7.

The instantaneous effective area is defined by the following equation:

$$A = \frac{1}{\left(\frac{1}{As^2} + \frac{1}{Ac^2}\right)^{\frac{1}{2}}} \qquad (1)$$

Where As and Ac are, respectively, the scavenge port area and the exhaust port area.

Butler also described in detail the air flow calculations used for the Seahorse engine and they resulted in the diagrams reproduced in Fig 8. These calculations were based on turbocharger technology and an assessment of the resistance to the airflow through the engine.

The 76J engines used a scavenge pressure of 1.7 bar absolute and for a pressure drop across the cylinders of 0.105 bar obtained a specific air flow of 7.3 kg/bhp h.

It was possible to design the Seahorse engine so that a pressure of 2.4 bar absolute would give a specific air flow of 8 kg/bhp h with a pressure drop of 0.205 bar across the cylinders.

The overall turbocharging efficiency shown is the effective efficiency from discharge from the exhaust ports to delivery to the scavenge ports. It is therefore lower than the product of the turbine and the compressor efficiency by approximately 10% in accordance with Butler's estimate.

Thus, it was estimated that the Seahorse engine required a turbocharger efficiency of 64% and, although it used constant pressure turbocharging, Butler concluded that such turbochargers were not available. The engine was, therefore, fitted with a mechanically driven turbo compressor.

Academically, the above calculations marked a considerable advance within the company but in practice things went wrong.

Doxford Seahorse engines

The Doxford Seahorse engine was based on the idea that the inherent balance of an opposed piston engine makes this design well suited for geared applications.

Many advantages would result from this approach,[16] and since the idea was supported both within Doxford Engines Limited and Hawthorn Leslie (Engineers) Limited, these two companies formed Doxford Hawthorn Research Services Limited to develop the engine.

A prototype engine was built and this started its test bed trials at the beginning of November 1971. The engine incorporated a number of novel features and these were well described in the literature.[16,19,20,21] The turbocharging system was, however, of particular significance and Fig 9 has therefore been reproduced from Ref 19.

Because the exhaust piston cranks in the crankshaft had an 8 degree crank lead, the shaft had a rotating out-of-balance moment. This was counteracted by balance weights mounted on a balancing shaft. On its own this was a costly and complicated solution.

The balancing shaft had to rotate at the same speed as the crankshaft ie 300 rev/min max. A step-up gear box was driven by the balancing shaft and increased the speed to 22 500 rev/min to drive a centrifugal compressor arranged in parallel with the turbochargers.

As the engine was reversing and the centrifugal compressor non reversing, the gear box had to be reversing in addition to providing the step-up ratio of 1:75.

Butler and Crowdy refer to the auxiliary blower drive as a novel feature,[16] and two gear box designs had been allowed for, ie one designed by company A (gear box A) and the other company B (gear box B). The turbochargers were Brown Boveri VTR 500.

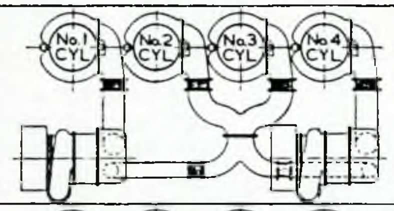
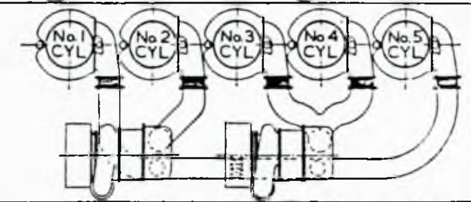
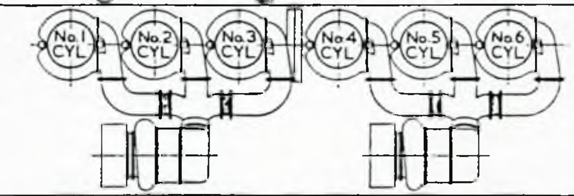
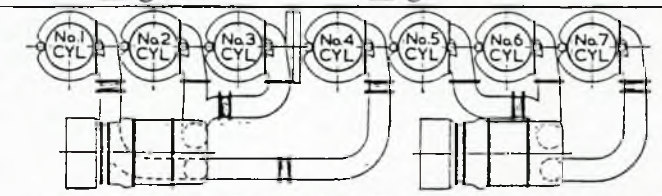
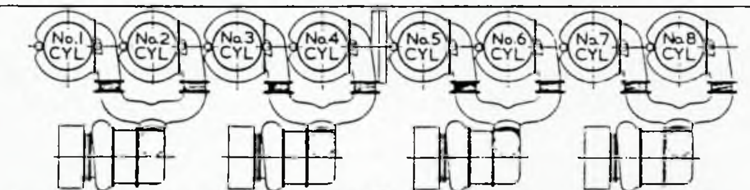
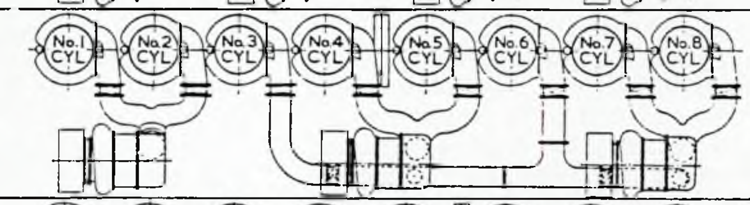
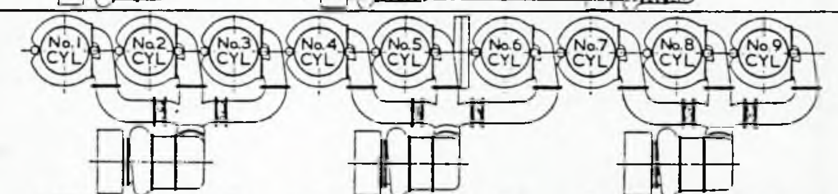
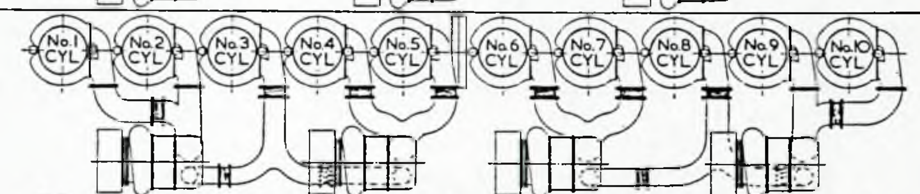
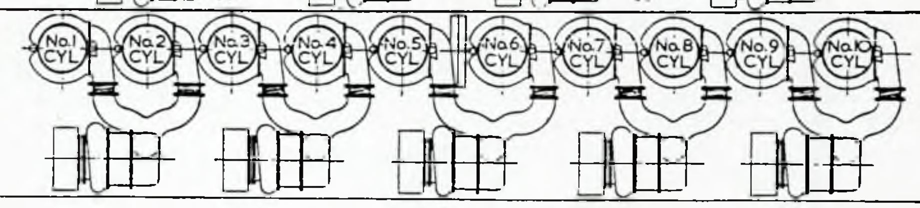
NUMBER OF CYLINDERS	FIRING ORDER	TURBO-CHARGERS	ENGINE TURBINE EXHAUST ARRANGEMENTS
4	1342	2 VTR 630 INLETS 1 AND ½	
5	13542	2 VTR 630 INLETS 1 AND ½	
6	153624	2 VTR 630 SINGLE INLETS	
7	1725436	2 VTR 750 TWO EQUAL INLETS	
8	14627538	4 VTR 630 SINGLE INLETS	
8	18347256	3 VTR 630 ONE SINGLE INLET TWO INLETS 1 AND ½	
9	167349258	3 VTR 630 SINGLE INLETS	
10	19463102758	4 VTR 630 INLETS 1 AND ½	
10	18549276310	5 VTR 630 SINGLE INLETS	

Fig 6: Arrangements of exhausts and turbo-blowers for J-engines having 4 – 10 cylinders

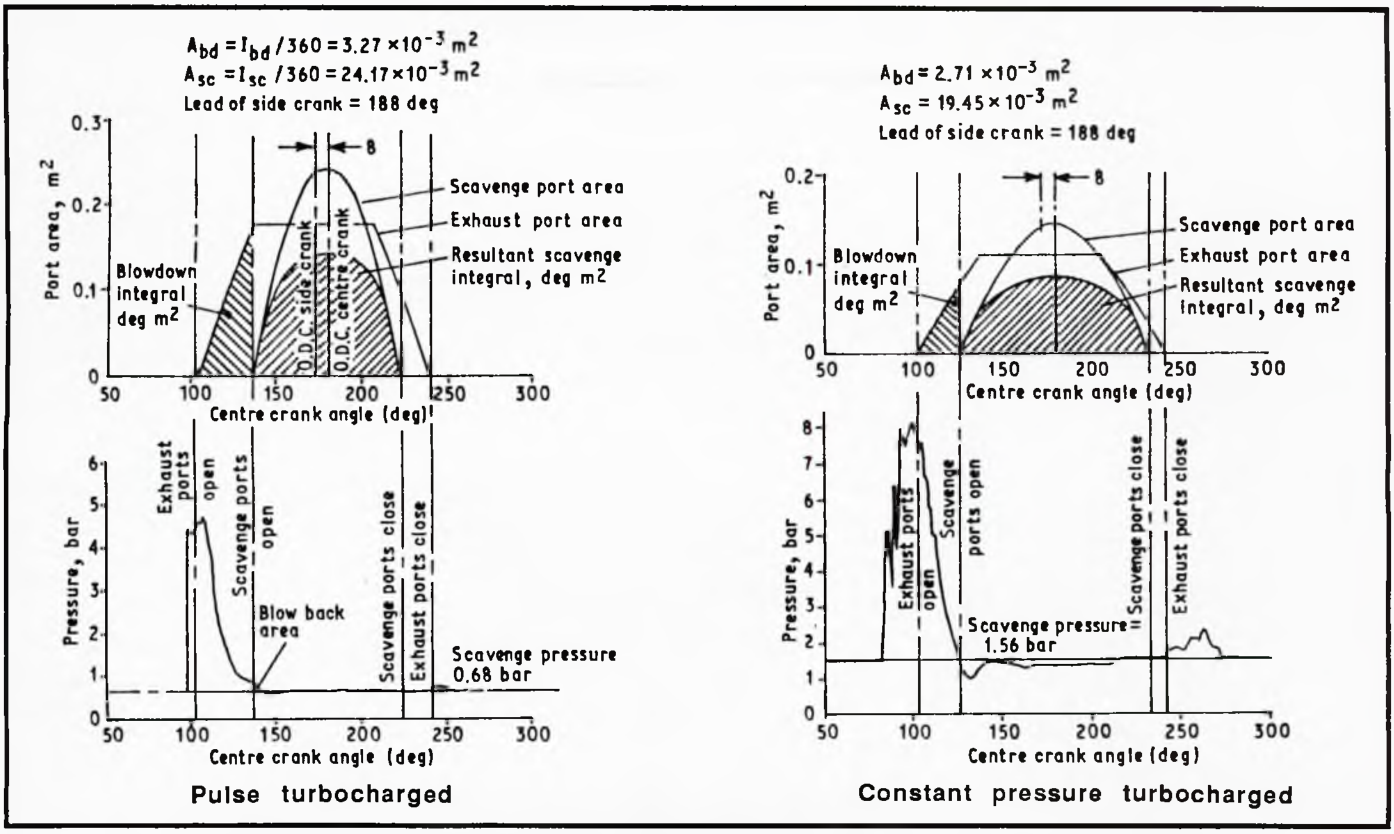
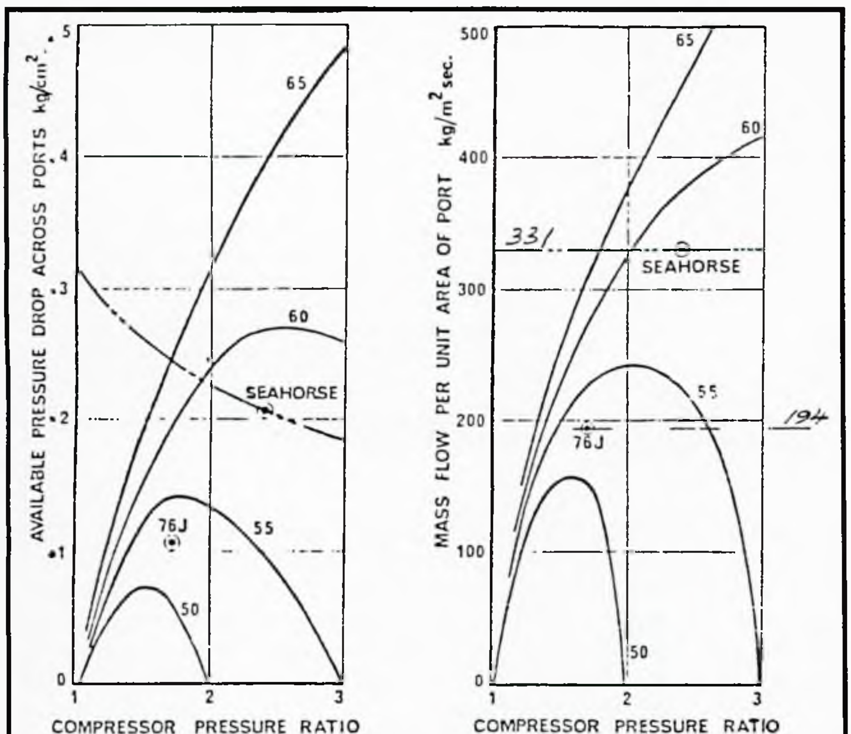


Fig 7: Comparison between the blow down of pulse and constant pressure turbocharging



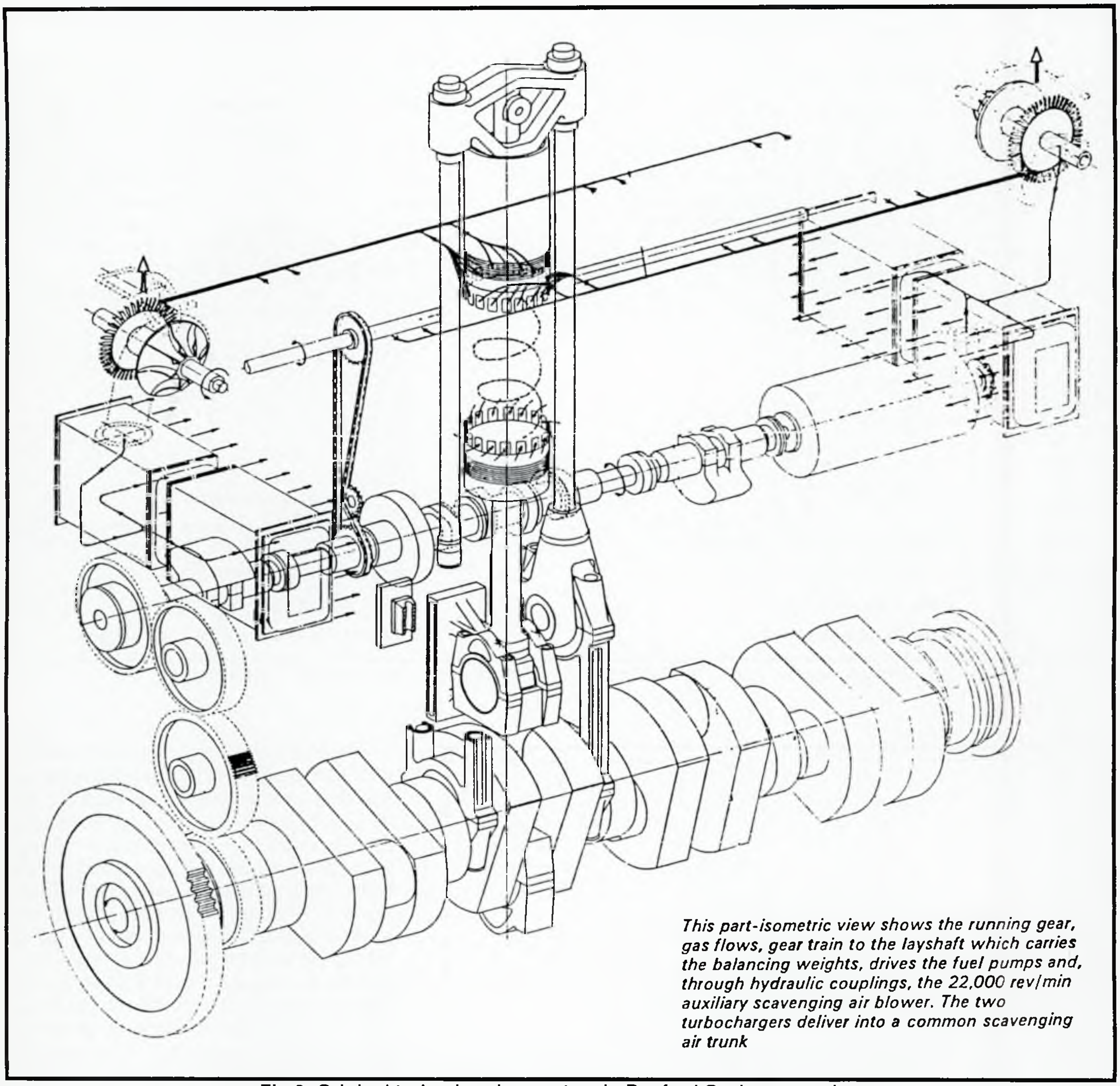
Engine	Equiv scavenge area	Mass flow/cylinder	Mass flow
76J	0.0262 m²	0.0262 x 194 = 5.08 kg/sec	7.32 kg/hp h
Seahorse	0.0168 m²	0.0168 x 331 = 5.56 kg/sec	8.01 kg/hp h

Both engines develop 2500 hp/cylinder.

Fig 8: Airflow calculations for Doxford Seahorse engine

The arrangement turned out to be a disaster. An analysis of the running hours of the two gear boxes was carried out in January 1973 by the Chief Test Engineer and this is shown in Fig 10a. After 14 months neither of the gear boxes had managed as much as 200 running hours. At the CIMAC Conference in Washington in April 1973 Butler and Crowdy presented a paper,[22] and Brown Boveri's technical representative made the following statement in the discussion in connection with the Seahorse engine: 'Doxford have decided to charge their newly developed Seahorse engine at constant pressure with an auxiliary blower working in parallel with the turbochargers, although this two stroke engine could be charged, if operated on the pulsation principle, with turbochargers alone, ie, without any auxiliary blower or scavenge pumps, except at very low speed.' He also advised changing to the VTR 501 turbochargers. The author, who was Technical Manager at Doxford Engines at the time was asked to look into the problem and agreed with this statement. The outcome was that two VTR 501 turbochargers were fitted in the autumn of 1973 but the constant pressure turbocharging system was retained, probably to incur minimum changes. The engine was tested, with the new turbochargers, in November 1973 and the results, which are shown in Fig 11, were very satisfactory although the specific airflow was lower than the originally stipulated 8 kg/bhp h.

At this stage one must ask the question as to



This part-isometric view shows the running gear, gas flows, gear train to the layshaft which carries the balancing weights, drives the fuel pumps and, through hydraulic couplings, the 22,000 rev/min auxiliary scavenging air blower. The two turbochargers deliver into a common scavenging air trunk

Fig 9: Original turbocharging system in Doxford Seahorse engine

why did Butler stipulate a specific air flow of 8 kg/bhp h for the Seahorse engine when the J-engines had used 7.3 kg/bhp h. The answer probably is that the J-engines had until then suffered a significant number of combustion belt failures,[23] and established a poor reputation in this respect. It was believed that a higher specific air flow would prevent similar failures in the Seahorse engine.

In September 1973 Butler presented a list of the top 10 remaining problems with the Seahorse engine as shown in Fig 10b. With the removal of the mechanical blower the gear box problem had been eliminated and piston ring cuffing was now the main remaining problem. When the engine was first started the piston ring scuffing was confused with piston skirt scuffing but the latter had been solved mainly by fitting balanced crossheads. However, piston ring scuffing remained a problem and although improvements were achieved in piston design and lubrication, the engine had to be nursed through its 500h trial at the end of the test bed period. The criteria for the 500h trial had been set by DTI.[24] The engine was designed with a mean lower piston speed of 8.8 m/s,[16] whereas the highest piston speed used for any of the J-engines was under 7 m/s. The piston ring scuffing problem in the Seahorse engine was clearly related to its high piston speed.

Open days were held at Doxford Engines Limited for the Seahorse engine in April 1975 but no engines were sold. The tests on the prototype were terminated later that year. The engine was put in store for some time and then scrapped. It is believed that the total cost of the project was about £6M and that was a heavy price to pay for the lessons learned.

Constant pressure turbocharged J-engines

In the period from April 1975 to July 1977 a series of 67J4

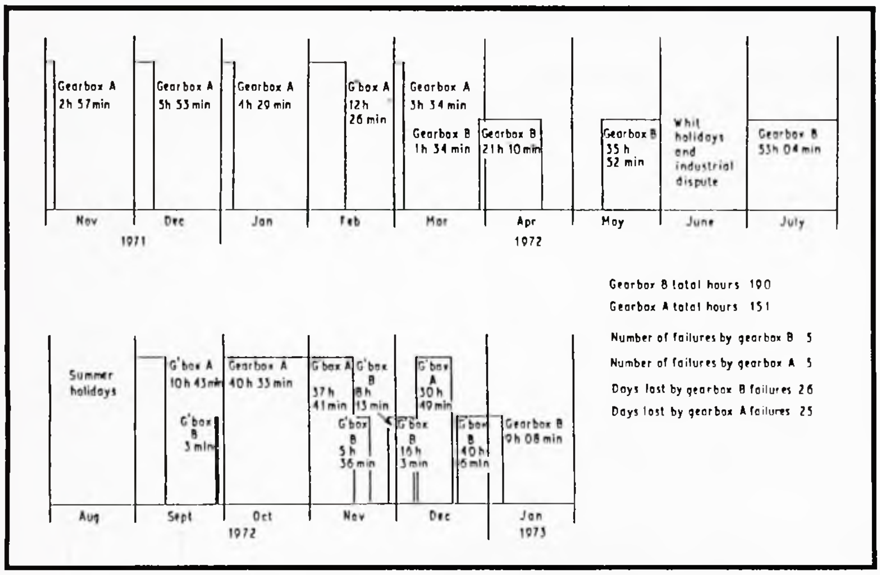


Fig 10a: Analysis of running hours of mechanical blower

	Solved	Remaining
1. Piston ring scuffing		x
2. Piston skirt scuffing	√	
3. Mechanical blower gearbox		x
4. Leakage from diaphragm gland		x
5. Excessive movement of side rods	√	
6. Wear of centre crosshead shoes	√	
7. Bearing failure		x
8. Slip of couplings on camshaft drive	√	
9. Oil leakage from crankshaft end seals	√	
10. Cracking of cylinder liners	√	

Fig 10b: Seahorse engine; top ten problems (25 September 1973)

engines for South American owners completed their test bed trials. Earlier four cylinder engines had been pulse turbocharged but the order for these nine engines had been obtained after quoting a lower specific fuel consumption and it was decided to adopt a pulse converter in the exhaust pipe system.[25] The engines also had a higher compression ratio, a longer expansion stroke and four fuel injectors per cylinder as for the Seahorse engine. Earlier J-engines used two fuel injectors per cylinder. As can be seen from Table I the results were good and an improvement in fuel consumption of 6 – 7 g/kw h had been obtained compared with earlier J-engines. One single entry Napier S610B turbocharger was fitted to this engine type and this represented a considerable cost reduction compared with earlier four cylinder engines. However, the slow speed performance was not as good as for the pulse turbocharged engines although it was acceptable.

One of the above 67J4 engines was run on the test bed during the open days for the Seahorse engine and caused a lot of interest. The author, who was responsible for the J-engines at that time, felt strongly that in the prevailing poor market Doxford should concentrate on three and four cylinder engines. For these engines the inherent balance of the opposed piston principle was of particular advantage. In particular, customer interest was shown in a small three cylinder engine and preliminary design was started in the autumn of 1976. Based on the success of the constant pressure turbocharging system in the Seahorse engine it was decided to adopt this system as described in a paper early the following year.[15] Great emphasis was placed on reducing the resistance to the air flow through the engine as much as possible. The exhaust ports were made as big as possible and a new design of exhaust belt with bigger flow areas was developed. This type of exhaust belt was also adopted for the constant pressure turbocharged 76JC4 engines.[26] Earlier, in the winter of 1977, Doxford Engines Limited received orders for five of the three cylinder engines which were designated 58JS3 engines and this was soon followed by an order for a further two. Not long after this orders were also received for eight 76JC4 engines. Two of these latter engines were fitted with one Napier 650 turbocharger each. The other six were fitted with two MAN turbochargers each. The design department was therefore in an acute overload situation as the 58JS3 engine was a new design and the 76JC4 engine incorporated constant pressure turbocharging. Nevertheless, the first 76JC4 engine completed its test bed trials on 7 July 1978 and the first 58JS3 engine on 10 July 1978. This was followed by sea trials on 13 October 1978 and 16 November 1978 for the two engines respectively.

A detailed technical description of the first 58JS3 engine was given by Henshall and Ørbeck,[27] a discussion of the practical application of the engine was provided in the Supplement to 'The Motor Ship', February 1979,[28] and a users view was presented in February 1980.[29] With the 58JS3, which was a two stroke crosshead engine, Doxford had penetrated a market hitherto dominated by trunk piston four stroke engines and in this respect the engine was a forerunner of the small bore engines later launched by Sulzer and B&W. Thus, in 1978 Briner,[30] described the Sulzer RLA56 engine as an extension of the RND-M engine range at the small output end. The smallest RLA56 engine was the four cylinder engine of 3720 – 3940 kW at 155 – 170 rev/min. This compared with 3680 kW, later increased to 4048 kW at 220 rev/min for the 58JS3 engine.

The 76JC4 engine had an MCR rating of 8832 kW at 123 rev/min and took the place of the earlier 67J6 engines of 8832 kW at 124 rev/min. This resulted in substantial reductions in engine length and cost as well as a reduction in specific fuel consumption from 206 g/kW h to 200 g/kW h. Thus the 76JC4 engine must have been quite competitive. One further 76JC4 engine was sold and this engine completed its test bed trials in May 1979. It was followed by a 76JC4R engine which was tested in August 1980. This latter engine was similar to the 76JC4 engine but with an MCR of 6624 kW at 96 rev/min. After some guarantee problems all the above engines have given very satisfactory performance and all the expenses of guarantee and development in service were about £6/hp. However, the British Shipbuilders leadership believed in the single piston engine and in 1980 Leo Curran, board member for engineering, persuaded the board to stop building Doxford Engines. A 58JS3 test engine was kept at Doxford Engines Limited.

In 1981 BS(ETS) (British Shipbuilders (Engineering Technical Services)) was formed and Doxford Engines Limited became Doxford Engines Limited (Spares Components and General Engineering). During his term as Chairman of British Shipbuilders R Atkinson tried to resurrect the opposed piston engine. A project named BS42-100 was started in 1982 to design an engine with 420 mm bore and 1000 mm combined stroke. The work was conducted by IPE (International Power Engineering) in Copenhagen in conjunction with BS(ETS). The project, which had many similarities with the Seahorse

Table I: Summary of specific fuel consumption of Doxford engines at mcr (based on diesel oil of 10 200 kcal/kg calorific value)

Engine type	Test bed dates	Turbo charging	BMEP bar	Max press bar	P-max/ BMEP	Ex crank lead	Exp stroke deg	Comp ratio	Inj per cylinder	SP F C g/kW h	Ref
67J6	April 1966 to January 1978	Pulse	9.42	68.6	7.28	8°	105	10.6	2	206	15
76J8	July 1967 to December 1977	"	9.32	67.1	7.20	8°	105.4	10.32	2	207	15
67J4	April 1975 to July 1977	Pulse convertor	9.42	72.2	7.66	8°	107.6	11.43	4	200	15
4 cylinder Seahorse	November 1973	Constant pressure	10.71	103.9	9.70	8°	102.5	12.86	4	198	16 21
76JC4	July 1978	"	1087	84	7.72	0	112.3	11.46	2	200	26
Original 58JS3	July 1978	"	11.42	84	7.36	0	109.8	11.37	4	200	26
Final 58JS3	June 1983	"	11.42	94	8.23	0	109.8	11.37	4	193	–
76JC4R	August 1980	"	10.45	90	8.6	0	119.2	12.16	2	193	–
BS 42X100	Design project 1983	"	14.7	150	10.2	0	105	–	–	185	–
B&W MAN L70MCE	1985	"	13	126	9.69	–	–	14	–	166	33
Doxford 49X122	Suggested	"	13	126	9.69	0	–	14	–	–	–

Table II: Comparison of engines

Author	Engine builder	Scavenge system	Cylinder charging	Turbo charging	Engine type	MIP kg/cm^2	Maximum pressure kg/cm^2	Scavenge pressure Kg/cm^2	Ex temp °C	Specific air quantity	Specific fuel consump-tion g/bhp h
Gugliel-motti	Fiat	Cross	A	Constant pressure and SC pumps	B.750S	9	75	1.05	–	–	–
Taylor	Doxford	Opp piston uniflow	B	Pulse	76J8	9	62	0.58	335	7.4	150
Scobel	MAN	Loop	B	Pulse for 6,9 and 12 cylinder constant pressure others	–	–	–	–	–	–	–
Wolf and Stoffel	Sulzer	Loop-cross	B	Constant pressure and UP Ch*	8RND 105	9	76	0.80	295	7.4	146
Andersen	B&W uniflow	Ex valve	–	Pulse	K98FF	9	68	0.60	295	8.7	150
Thulin and Dahlbring	Gotaverken	"	–	Constant pressure and SC pumps	–	9	65	0.93	–	7.7	–

A Scavenge ports close after exhaust ports
B Scavenge ports close before exhaust ports
* underpiston charging

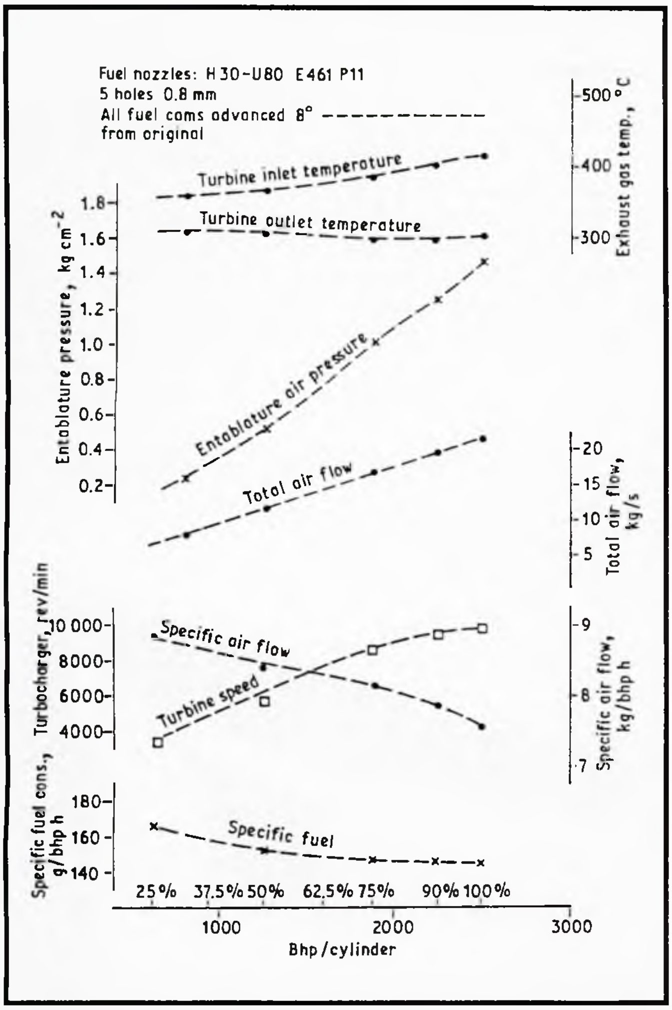


Fig 11: Doxford Seahorse engine; test bed performance curves

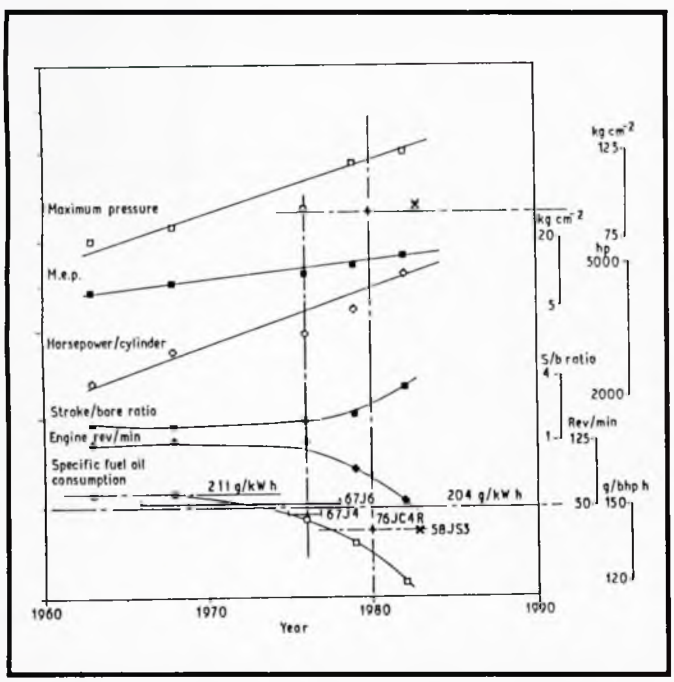


Fig 12: Development of 2-stroke engines over 20 years (from A F Harrold[2]) compared with the results from Doxford engines

engine project, never went beyond the design stage and faded out in 1985. Doxford Engines Limited (Spares, Components and General Engineering) was closed by W Scott who became board member for engineering after G Day had taken over as Chairman of British Shipbuilders.

Fuel consumption

From a thermodynamic point of view, efforts made to reduce the fuel consumption of diesel engines are well understood. The modifications are increased maximum cylinder pressure, increased compression pressure, extended relative expansion stroke, which can be obtained with higher efficiency turbochargers, improved scavenging and combustion and finally reduced heat losses. The effects of the first three modifications are relatively easy to assess, using engine simulation programs, and a special program for Doxford engines was brought into use in 1978. The last three of the modifications are, however, less defined and improvements are mainly achieved by clever design and testing.

As a contribution to the discussion after a paper by Wolf and Stoffel,[13] in 1969 the author prepared the table shown as Table II. The table is not comprehensive but of considerable interest since today, after about 20 years and a period of intense competition on specific fuel consumption, only the uniflow-scavenged engines with exhaust valves have survived. The Sulzer engine obtained the best specific consumption but with a high maximum pressure. The B&W and Doxford engines obtained the same fuel consumption but the Doxford engine used the lowest maximum pressure.

A summary of the specific fuel consumptions of Doxford engines at MCR is given in Table I. This table also provides information about the main parameters which affect the specific fuel consumption. For comparison purposes Fig 12 shows some results compiled by Harrold,[2] for single piston two stroke engines. The 67J6 and 76J8 engines were chosen as representatives of the pulse turbocharged Doxford engines. The specific fuel consumptions of these engines, 206 – 207 g/kW h, compared favourably with the results for single piston engines before 1970, ie 211 g/kW h. The 67J4 pulse converter turbocharged engine gave very satisfactory results considering its modest maximum pressure. The specific fuel consumption was 2g/kW higher than for the Seahorse engine but this was achieved with pmax/BMEP of 7.66 compared with 9.7 for the Seahorse engine. The increased expansion stroke of the 67J4 engine was probably a contributory factor in this respect.

The first 76JC4 and 58JS3 engines were comparable in performance with the 67J4 engine but considering that a bigger engine should return a better specific fuel consumption than a smaller engine the 58JS3 engine gave better results than the 76JC4 engine. This was probably due to the four fuel injectors per cylinder. The 58JS3 engine was originally fitted with a Napier NA550 turbocharger with a 510 turbine outlet casing and the auxiliary blower in series. A Brown Boveri VTR 501 turbocharger with ball and roller bearings was then fitted and the better slow speed performance of this turbocharger made it possible to fit the auxiliary fan in parallel with the turbocharger.[26] This resulted in an improvement of 2 g/kW h in the specific fuel consumption. Finally in 1983, the engine was tested with a new design of piston head as well as increased fuel pressure. This gave a further 5 g/kW h reduction in the specific fuel consumption but the maximum cylinder pressure increased to 94 bar. The 76JC4R engine also obtained 193 g/kW h but again this result was not as good as for the 58JS3 engine suggesting that four fuel injectors per cylinder gave better combustion than two. Referring to Fig 12 again it can be seen that by 1980 – 83 the competition had overtaken Doxford. However, both the 58JS3 engine and the 76JC4R engine compared favourably with the single piston engine with a 90 bar maximum pressure.

The 76JC4R engine was fitted with a Napier NA650 turbocharger and a controllable pitch propeller and the engine required to accelerate from idling to approximately 85% power in under 2 min. To achieve this the turbocharger was assisted by a hydraulic turbocharger accelerator at low loads as described by Fowler and Ørbeck.[32] This device was fitted just before the sea trials.

It is likely that fitting four fuel injectors per cylinder and improved piston heads, which will be described later, will improve the specific fuel consumption of the 76JC4R engine by 2%, ie to 189 g/kW h. Further an increase in maximum pressure to 104 bar will probably result in a further reduction to 185 g/kW h.

The Seahorse engine required a gear box with expected losses of 3 g/kW h. Thus, the Seahorse engine would return 201 g/kW h compared with the expected 185 g/kW h for the comparable direct drive engine, ie 8.6% higher.

The particulars of BS 42-100, which was mentioned earlier, are also entered in Table I. The predicted specific fuel consumption of this engine was 185 g/kW h. Adding 3 g/kW h for the gearbox gives 188 g/kW h on the propeller shaft which, compared with the best long stroke engines of about 166 g/kW h, is 13.3% higher. Although approximate, the above considerations illustrate the disadvantage of the geared engines with regard to specific fuel consumption.

Improved piston head design

During the development of the Seahorse engine a lot of attention was focused on piston ring scuffing. Two factors are important in this connection, ie the distortion of the firing ring groove and the temperature of the firing ring, and to improve these two factors it appeared to be desirable to position the firing ring further away from the piston crown.

The piston designs for the pulse turbocharged J-engines, the Seahorse engine and the 58JS3 engine are shown in Figs 13a, 13b, and 13c respectively and it can be seen how the ratio of S/D was increased up to and including the value in the original design of the 58JS3 engine. The 76JC4 engine used a similar

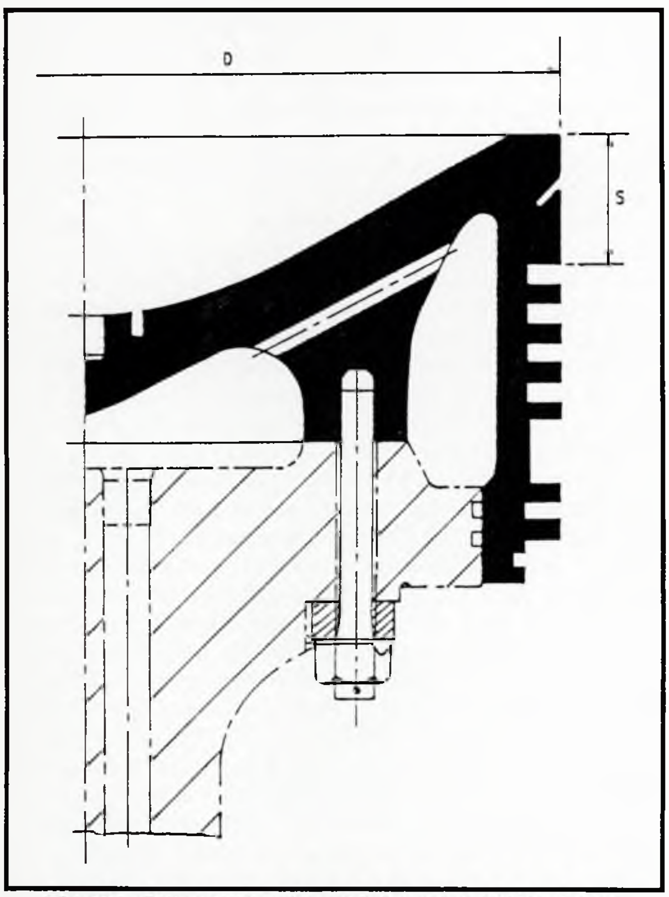


Fig 13a: Pulse turbocharged J-engine piston head; $S/D = 0.139$

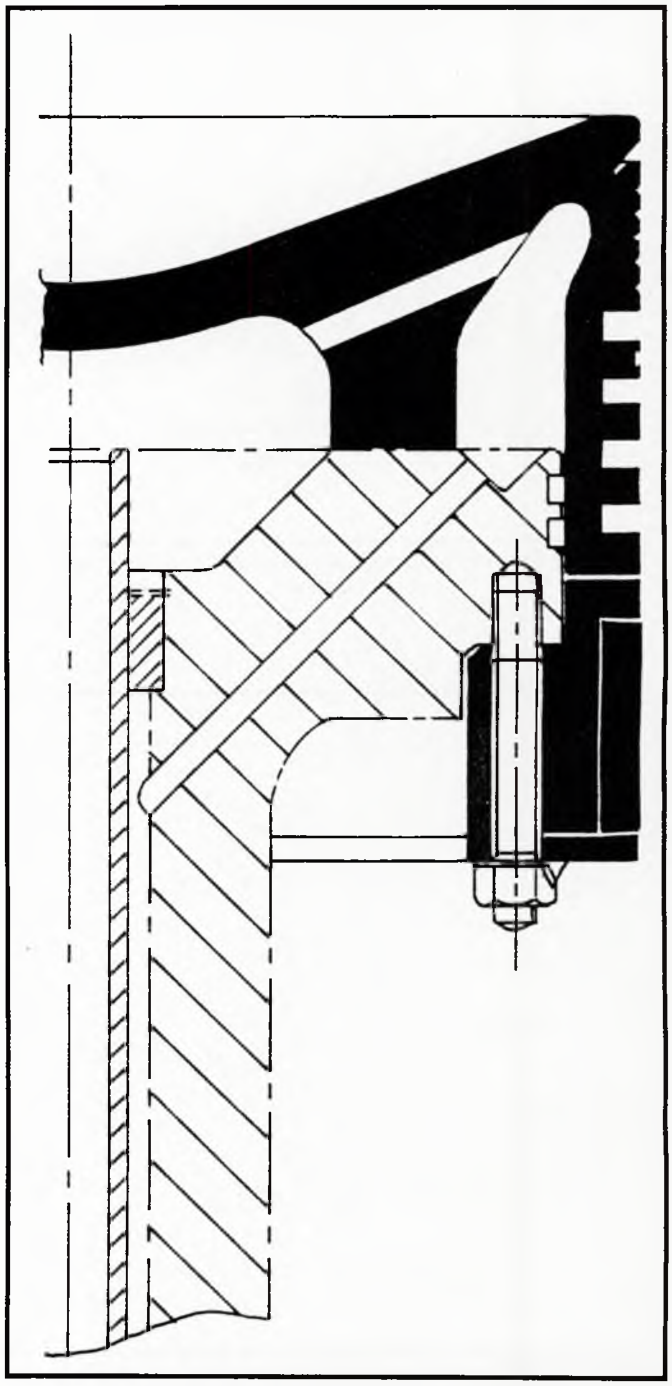


Fig 13b: Piston design for Seahorse engine; $S/D = 0.172$

piston head design to the 58JS3 engine. It was reported from the test bed trials of the above two engines that they had unusually oily entablatures. This was found to be cylinder oil which had been blown back through the scavenge ports.

Later it was found that in order to calculate the compression line of the cylinder pressure diagram from the 58JS3 engine it was necessary to use a compression index of 1.32. This question was discussed with IPE (Copenhagen) who told us that B&W used 1.38. On further consideration it was concluded that this difference had to be due to leakage past the piston crowns and out through the ports when the piston rings did not cover the ports as shown in Fig 14. This effect would be particularly prominent for the exhaust ports. After a detailed investigation into the flow of the coolants in the pulse turbocharged J-engine piston heads it was concluded that a coolant deflector, as shown in Fig 13d, would improve the cooling so that the piston ring

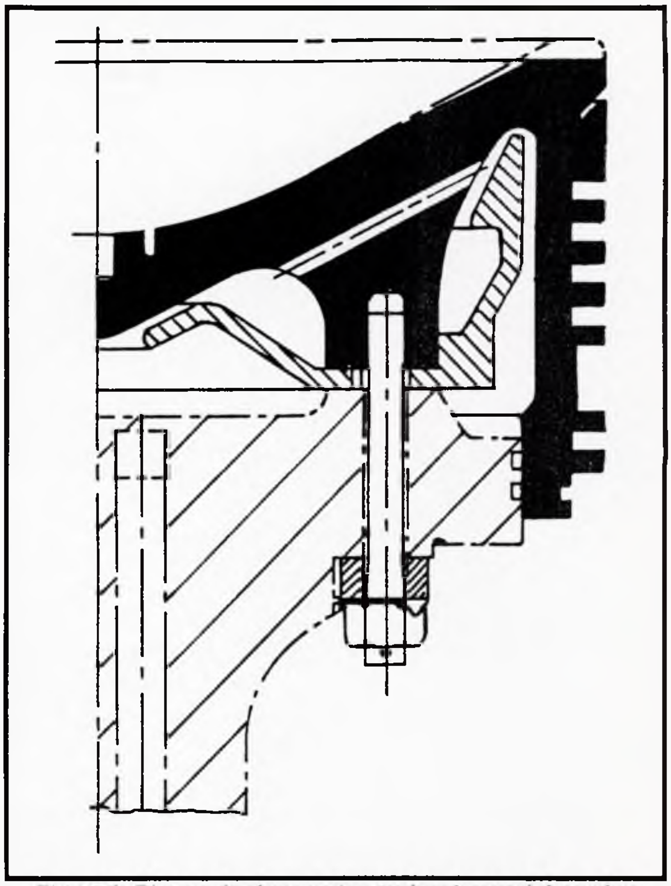


Fig 13d: Piston design; pulse turbocharged J-engine upgrading package; $^S/_D = 0.118$

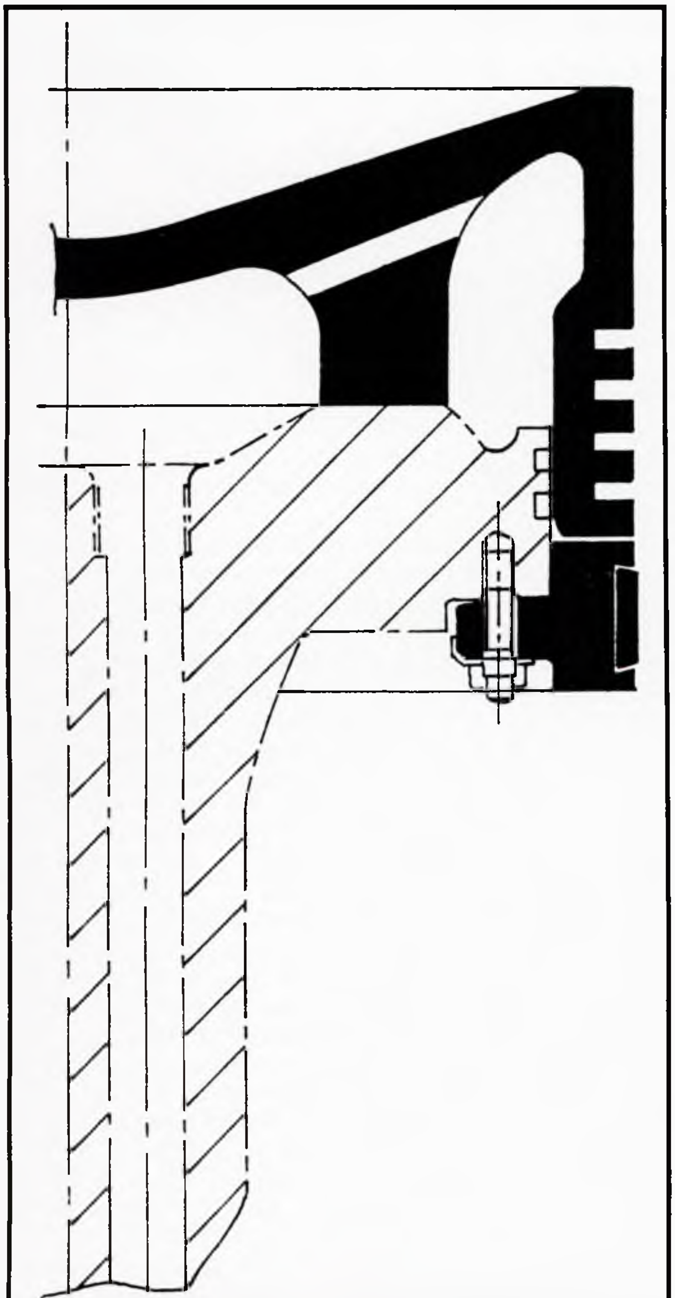


Fig 13c: Piston design for original 58JS engine; $^S/_D = 0.213$

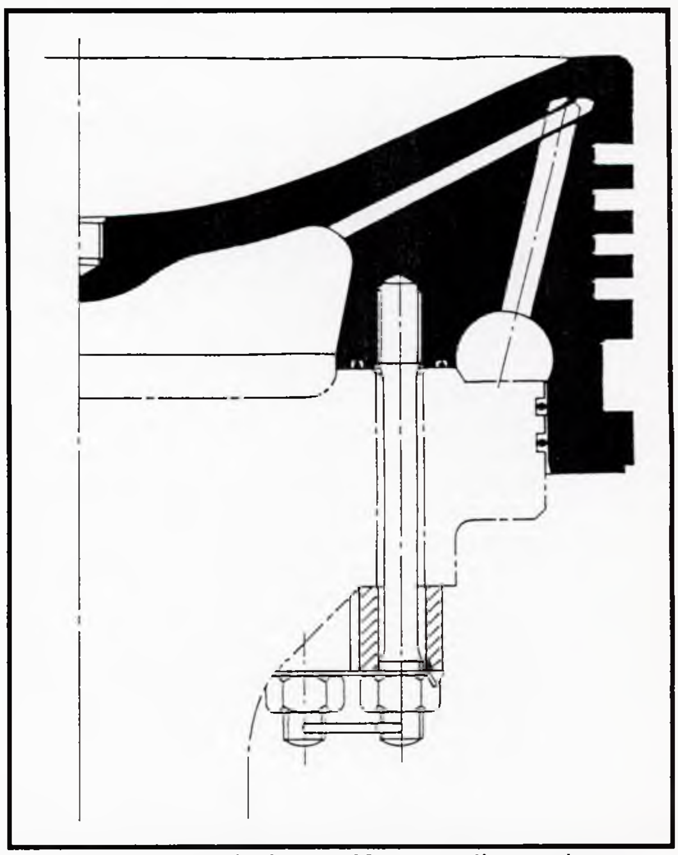


Fig 13e: Piston design; 76JC4 upgrading package; $^S/_D = 0.092$

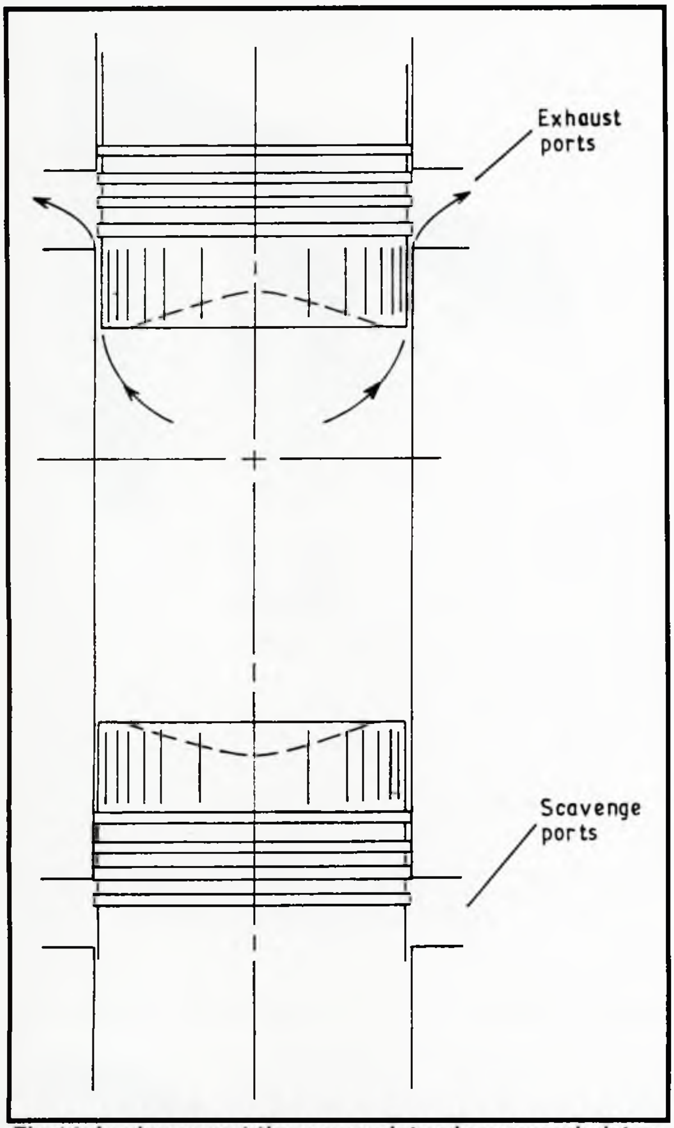


Fig 14: Leakage past the upper piston in opposed piston engine

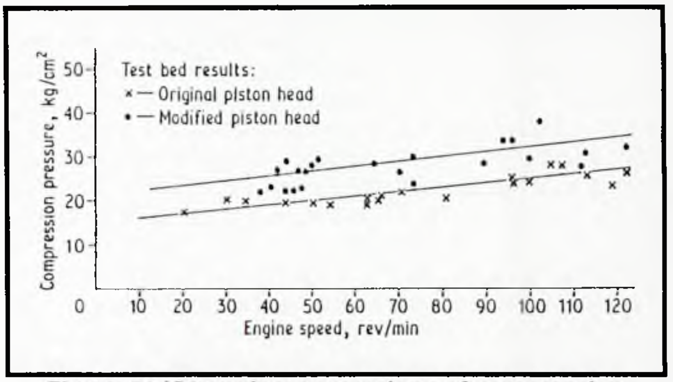


Fig 15: 58JS3 engine; comparison of compression pressure measured with original and modified pistons

Table III: Comparison of daily fuel consumption before and after fitting piston modification

	Average speed of ships (knots)	*Average consumption HVF (tonnes/day)*
Original engines	14.63	22.84
Modified engines	14.61	20.82

pack could be moved 15 mm towards the combustion space. The positions of the cylinder lubrication quills in the cylinder liners could also accept this change and an upgrading package was fitted to five 76J6 engines owned by Andrew Weir & Co Ltd. The MCR rating of these engines was 15 000 bhp but the vessels were operated at a reduced power of about 6000 bhp. The owners carried out a detailed analysis of the performance of the ships before and after fitting the piston modification. For three of the ships one voyage to the Far East and back just prior to and one voyage just after the modification were compared. In the case of the last two ships one of them provided the information for a voyage with the original pistons and the other ship the information for the modified pistons. The result of this analysis, which is shown in Table III, indicated a surprising improvement in fuel consumption for the same average speed. Primarily this must have been due to the piston head modification although it is possible that the general state of the original pistons was poorer than for the modified pistons and that this contributed to some of the difference.

It was mentioned above that the 58JS3, 76JC4 and 76JC4R engines in service incurred expenses of £6/hp for guarantee and development in service. Most of these expenses were associated with the 58JS3 engines which was the most advanced of the engine designs. After a relatively short time in service it became evident that the engines were too hot and it was agreed with Napier Turbochargers Limited to replace the NA510 turbine outlet casing by the NA550 design so that the turbochargers became completely NA550. This latter design had a bigger exhaust outlet and the velocity losses were substantially reduced. The exhaust trunking had to be changed to suit. The starting positioner was also redesigned to make it a more robust unit.

The above turbocharger modifications improved the exhaust temperatures substantially. However, those ships which used fuel oil homogenisers rather than purifiers, suffered excessive cylinder liner wear and when the liner wear reached 3 mm on diameter, the engines suffered starting difficulties. These difficulties were traced to lack of compression. In the mean time the above piston head development had taken place and it was decided to design a new piston head for the 58JS3 engines. Great attention was paid to the design of the cooling passages and the firing ring was positioned 60 mm from the edge of the crown. This compared with 124 mm for the previous piston head used in this engine. A very similar piston head was later designed for the 76JC4 engine and this is shown in Fig 13c. These pistons were fitted with four plain compression rings and since there was now enough room, the bearing ring was mounted on the piston head instead of separately as on the Seahorse and earlier 58JS3 pistons. The piston head was very rigid around the ring grooves and this minimised groove distortion.

As mentioned above this new design of piston head, in conjunction with a higher fuel oil pressure, reduced the fuel consumption at MCR by 5 g/kW h. The corresponding increased maximum cylinder pressure was explained by an increased compression pressure as shown in Fig 15. At 120 rev/min the increase was 7.5 bar and at 220 rev/min this would be about 10 bar.

To fit the new pistons it had been necessary to move the upper lubricator quills below the exhaust belt. New cylinder liners and piston heads were supplied to all the 58JS3 engines in service and all the ships now use purifiers for the fuel

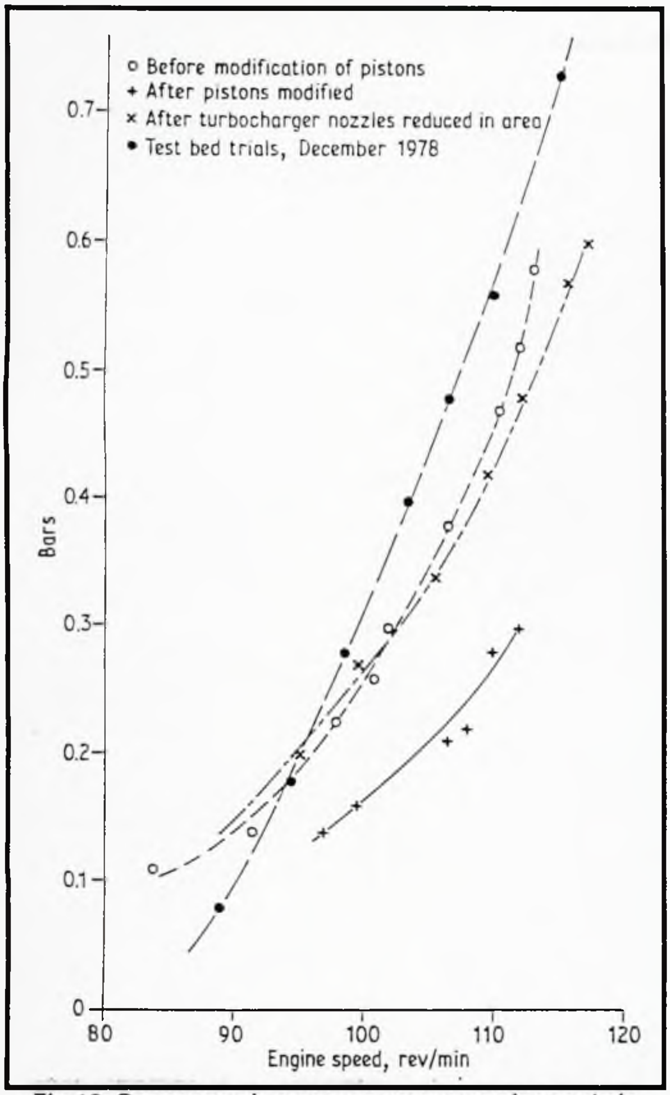


Fig 16: Scavenge air pressure versus engine rev/min

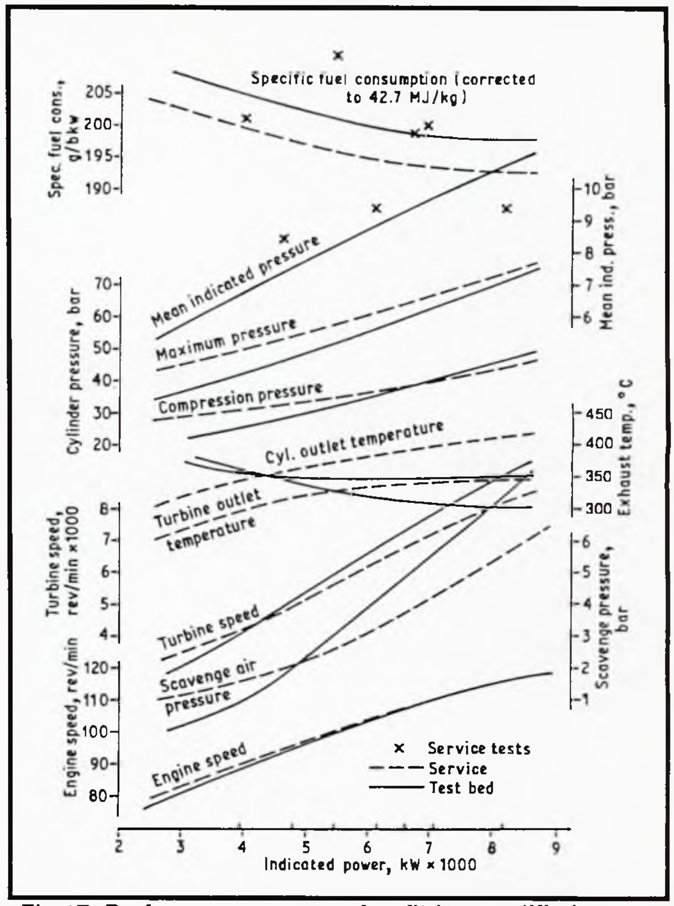


Fig 17: Performance curves after fitting modified crown pistons

treatment. After these modifications the engines started very satisfactorily, the cylinder lubricating oil consumption was reduced by 30% and the cylinder liner wear was very satisfactory. The new design of piston head has so far been fitted to one of the 76JC4 engines. A special jig was designed so that the new positions of the upper piston lubricator quills below the exhaust belt could be drilled in-situ. This was cumbersome but saved removing the cylinder liners from the engine. The order was received in August 1984 and covered upgraded piston heads, timing valves and fuel injectors. Due to the organisational changes in British Shipbuilders described above, with transfer of manufacture from Doxford to North Eastern Marine and later to Kincaid, the fuel injectors were first delivered in September 1985, the piston heads in April 1986 and the timing valves were never delivered. After fitting the new piston heads it was necessary to reduce the turbine nozzle area in the two MAN turbochargers. The scavenge pressure had dropped substantially as shown in Fig 16. The turbine nozzle area was changed from 512 cm^2 to 413 cm^2, ie a reduction of 19%. The forward turbocharger was modified first and the engine was operated in this state for some time before the aft turbocharger was modified. After both turbochargers had been modified a series of tests were carried out in December 1986 with the engine running between 90 and 120 rev/min. The scavenge pressures obtained in these tests are also shown in Fig 16. For example, at 100 rev/min the scavenge pressure before the modification was 0.26 bar. This dropped to 0.16 bar after the piston modification and was brought up to 0.265 bar after the turbocharger modification. Since there was little change in scavenge pressure between the original and the final build, the airflow must have been reduced in about the same proportion as the turbine areas, ie by 19%.

A comparison between the results after the final modification and the test bed results is shown in Fig 17. At 100 rev/min the turbine outlet temperatures were about the same as for the test bed condition and one would expect this to be the case also for the condition in service before the modification. Nineteen per cent less heat would therefore have been taken away by the exhaust gases after the modification and this must correspond to a substantial saving in fuel. Both the scavenge pressure and the turbine speed are higher at low engine power, than the test bed results and lower at high engine power which is advantageous. At 100 rev/min the test bed trials gave a scavenge pressure of 0.31 bar and a compression pressure of 32 bar. After the modification the results were; a scavenge pressure of 0.26 bar and a compression pressure of 35 bar. Adjusted to the test bed scavenge pressure the compression pressure would be 35 x 1.31/1.26 = 36.4 bar, ie the trapped air lost in the test bed condition was (1-32/36.4) x 100 = 12%.

Calculated from cylinder pressure diagrams the temperature of the gas reaches a maximum of about 2000K during combustion. Increasing the amount of trapped air by 12% would reduce this temperature by 240K and have a considerable effect on the heat transfer through the cylinder walls. It is, therefore, now believed that the Doxford combustion belt failures were primarily caused by insufficient trapped air and that Butler's

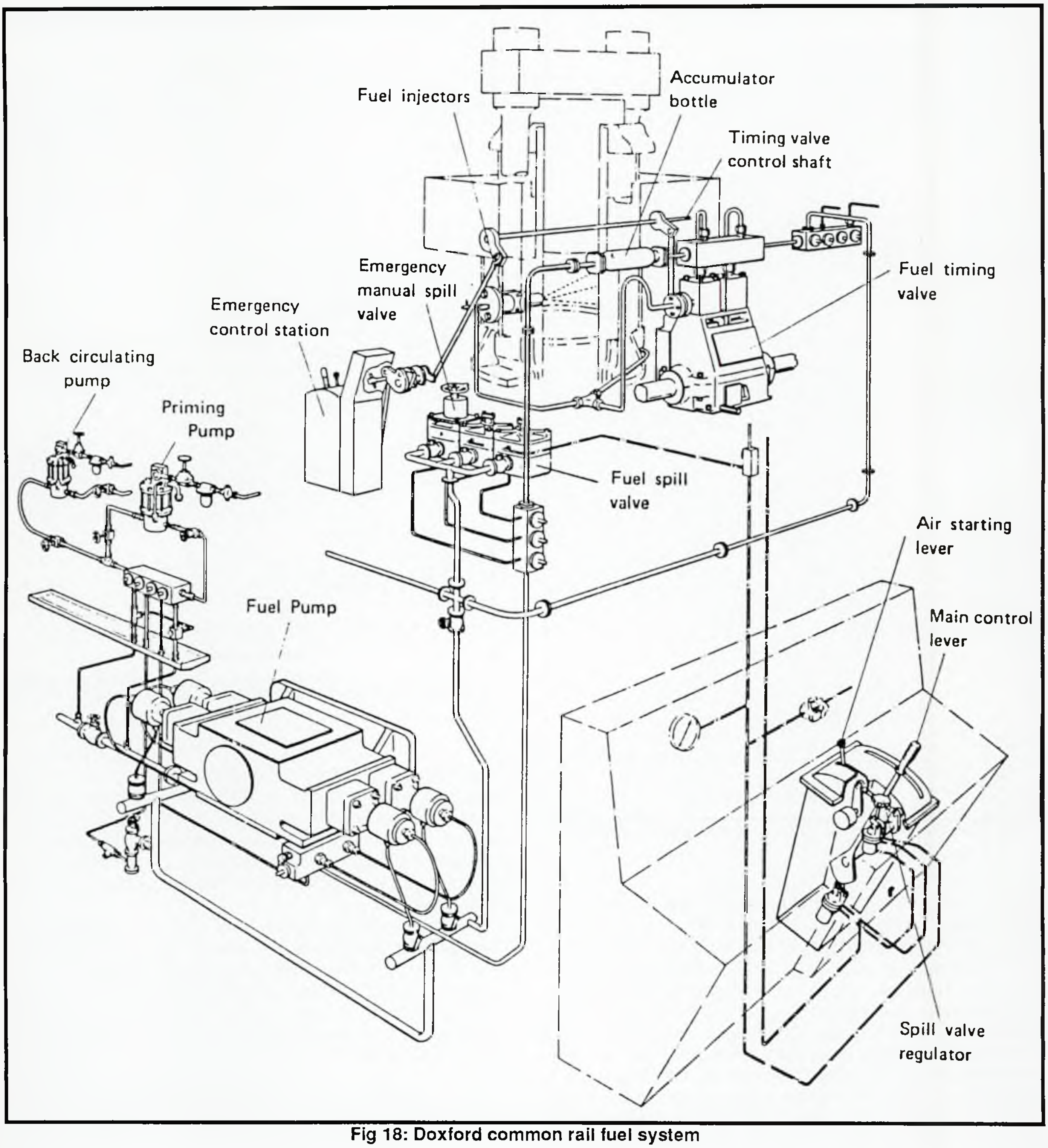


Fig 18: Doxford common rail fuel system

requirements of a high air flow through the Seahorse engine were largely irrelevant to this problem.

The specific fuel consumption results after the modification showed considerable scatter as would be expected. On average they were 2.4% lower than the test bed results, as indicated by the curve. The airflow considerations indicated a bigger reduction in specific fuel consumption but broadly speaking the saving was similar to the one obtained for the 58JS3 engine (see Table I). The earlier suggestion, that the specific fuel consumption of the 76JC4R engine could be reduced to 185 g/kW h, also seems reasonable in view of these results.

Fuel and control system

The Doxford common rail fuel system is well described in the literature but it is shown in Fig 18 for reference purposes. The system has many advantages over the jerk-pump system and from the engine performance point of view the most important feature is that the fuel injection pressure can be controlled over the operating speeds of the engine. Thus, an adequate pressure for proper atomisation can be obtained at slow speed without having to use an excessively high pressure at full speed. However, the Doxford system also has some disadvantages. Since the timing valve control shaft requires a

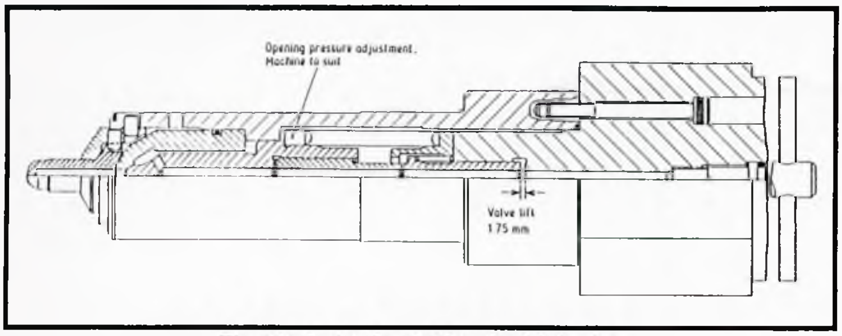


Fig 19: Autocirculating fuel injector

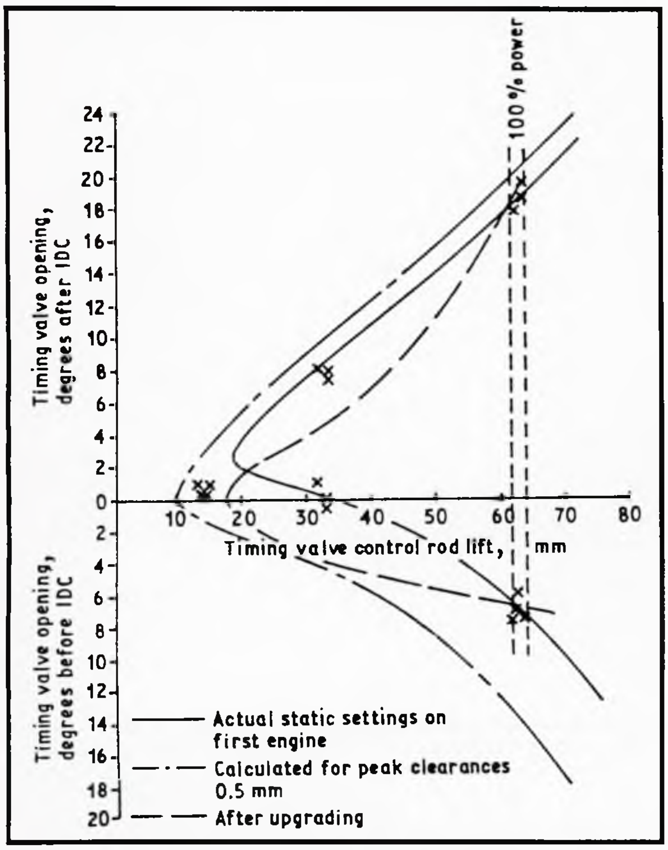


Fig 20: Timing valve characteristics of 76JC4 engine

relatively big torque to operate it and presents shock loads on the operating mechanism, it was relatively difficult to find a suitable governor for the Doxford engine. Because of its high mechanical efficiency this engine also required a governor with a good response. To meet these demands Doxford was the first engine builder to use an electronic governor. This was introduced in 1972 and, although there were some earlier problems, this action was generally successful. Later the governor was on one occasion integrated in a very sophisticated control system as described by Fowler and Ørbeck.[32]

With a relatively extensive high pressure pipe arrangement in the common rail fuel system, careful attention must be paid to pipe coverage and the problems of pipe coverage were increased by the necessity of steam tracing and the use of four injectors per cylinder. Water cooling of the injectors added to these complications and it was a disadvantage that circulation of hot oil through the system had to be done manually before starting the engine. To improve this situation a fuel injector, as shown in Fig 19, was introduced in 1982 and has been further developed since then. This injector does not require water cooling. It is fitted with a Stellite nozzle which gives much longer service life between overhauling than the earlier injectors. The injector also incorporates a valve which allows automatic circulation of fuel oil while the engine is stopped. Spring loaded tappets in the timing valves are also required to achieve this purpose. Modified fuel systems have been in service on several engines since 1983 and work satisfactorily. There has been little feed back of information but what we have received indicates that the non-cooled injector gives a slight improvement in fuel consumption probably because the fuel oil is closer to its correct temperature when injected into the cylinders.

Finally, the improved timing valve design must be mentioned. Referring to Figure 17 again it can be seen that the curve of cylinder maximum pressure versus power shows a slope which is increasing with power. A much better part load fuel consumption would be obtained if the maximum pressure could be raised for these powers and held constant over the top end of the pressure range. The present pressure variation is due to the timing valve characteristics for the Doxford engine and typical characteristics for the 76JC4 engine are shown in Fig 20. The design curve and the actual curve for the first engine were similar, but the actual curve was obtained for a bigger peak clearance and retarded approximately 3° relative to the design curve. The curves show how the start of injection is advanced when the power of the engine is increased and as a result the cylinder maximum pressure will increase rapidly with power. A timing valve modification designed to give the desired maximum pressure variation is now available and the characteristics of this modification are shown as a dotted line.

In conclusion it can be said that the fuel and control system of the Doxford engine has been improved substantially since the last engines entered service and this can be reflected in the engine performance.

FUTURE DEVELOPMENT

Today the longitudinal scavenge exhaust valve engine reigns supreme among the two stroke crosshead engines. It is therefore natural to compare the future potential of the Doxford opposed piston engine with this engine. A detailed comparison will not be carried out as this could only be given justice in a complete paper. The discussion will be confined to specific fuel consumption as it was in this field that the exhaust valve engine established its superiority. The engine chosen for comparison purposes is the MAN, B&W L70MCE,[33] and the relevant particulars of this engine are given in Table I. The specific fuel consumption of 166 g/kW h is impressive and this is obtained by using a very high compression ratio, and a high maximum pressure in relation to the BMEP. The compression ratio was calculated from the compression pressure by using a compression index of 1.38.

With the new piston head and timing valve design there is no reason why the above cylinder pressure characteristics cannot be obtained in an opposed piston engine. Figure 21 shows the cross section of the 58JS3 engine. If the cylinder bore was reduced to 490 mm, and the maximum cylinder pressure increased to 126 bar, the forces on the running gear would remain substantially unaltered. The smaller bore liner is indicated and clearly the accessibility round the liner will be slightly improved. With a BMEP of 13 bar the three cylinder engine would

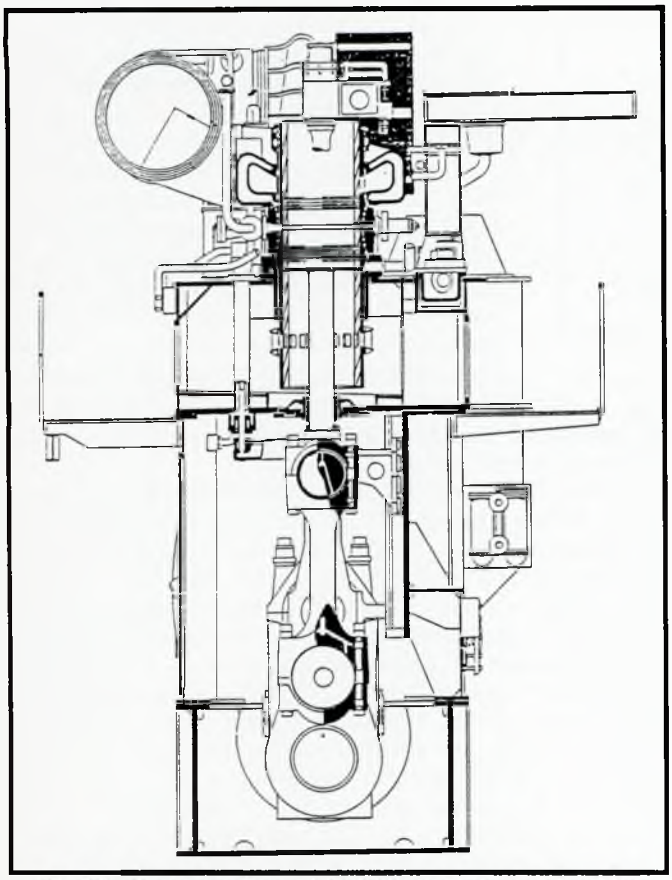


Fig 21: Cross section of 58JS type engine indicating reduced cylinder bore

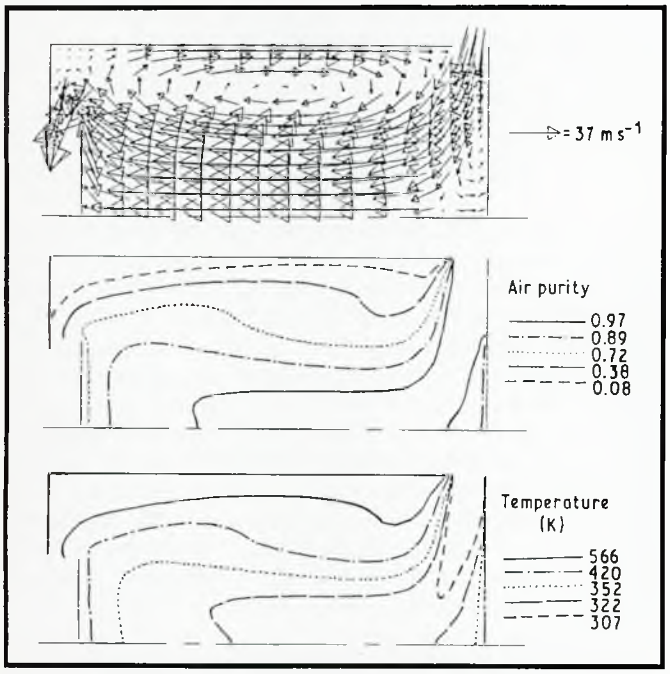


Fig 22a: Flow structure in two-stroke exhaust valve engine at 20° BBDC (results from Cartellieri and Johns[34])

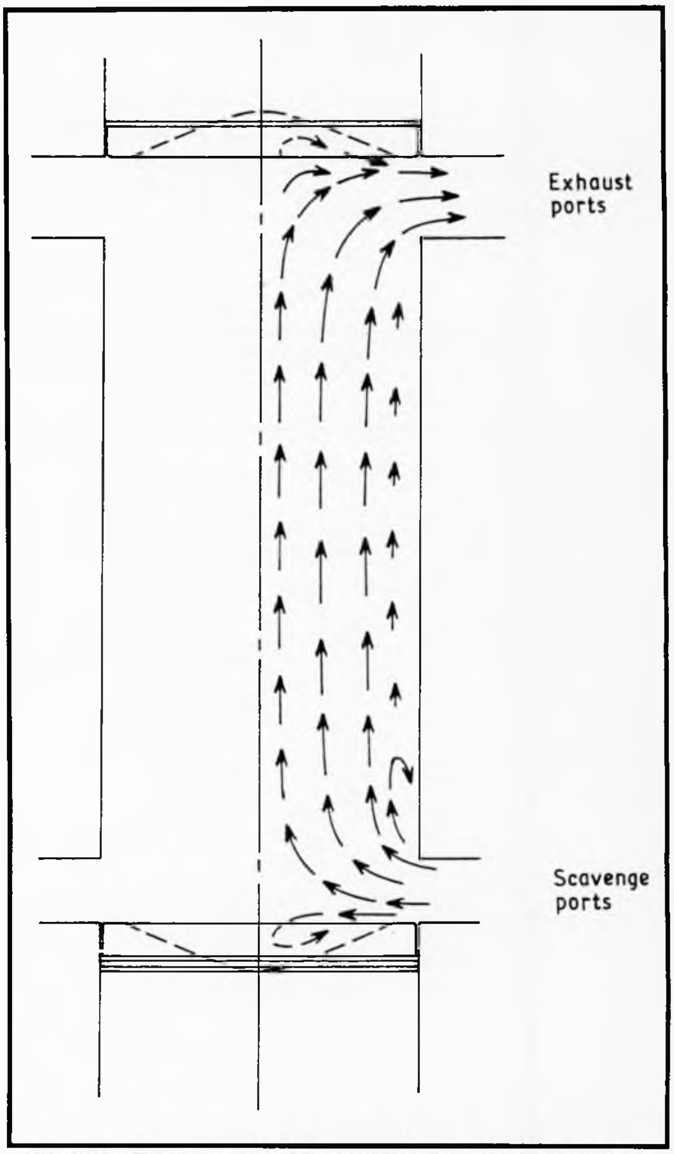


Fig 22b: Flow structure in two-stroke opposed piston engine

develop 3290 kW at 220 rev/min. The only difference between the opposed piston engine and the exhaust valve engine, as far as the cylinder pressure diagram is concerned, is that since the exhaust ports are much bigger relative to the engine size than the exhaust valve areas, the expansion stroke can be made longer in the opposed piston engine than in the exhaust valve engine. This effect suggests that the former engine type should reach a slightly better specific fuel consumption than the latter.

The flow of the gases in the cylinder during the scavenge period should also be considered. This was studied by Cartellieri and Johns for a smaller exhaust valve engine and some of their results are reproduced in Fig 22a. It can clearly be seen how the scavenge air moves up through the centre of the cylinder and, as some of the escaping exhaust gas hits the inner surface of the valve, it is deflected down the cylinder wall while the major part escapes through the valve opening. A similar diagram has been constructed for the opposed piston engine and is shown in Fig 22b. With the exhaust ports there will be almost no backflow along the cylinder wall and a more complete scavenge. This effect also suggests that the opposed piston engine should obtain a slightly better fuel consumption than the exhaust valve engine.

Without having gone into great detail the above two effects indicate that the opposed piston engine possesses a greater development potential than the exhaust valve engine.

CONCLUSIONS

In the first part of this paper it was demonstrated that the shaft design of the opposed piston engine will permit direct drive engines of very high power. The mechanical design techniques were well established and the risk involved in this development would be small.

Maximum power per cylinder was raised from 2500 hp to 3000 hp with relative ease for the latest J-engines. Since the main cause of the combustion belt failures in earlier J-engines was later identified as leakage of trapped air past the piston crown and this was substantially improved by an improved piston design, 5000 hp per cylinder should be well within reach.

With continuous emphasis on fuel efficiency, direct drive opposed piston engines are preferable to geared engines. The inherent primary balance of the opposed piston engine makes 3 and 4 cylinder engines of this design particularly advantageous.

Recent improved understanding of the fuel injection system and the engine characteristics in connection with earlier fuel consumption results suggests that the opposed piston engine can be designed for a very competitive specific fuel consumption.

ACKNOWLEDGEMENTS

The author wishes to thank his colleagues for their help and discussions in relation to this paper. He would also like to express his appreciation for the co-operation given by Mr R M MacLeod of Andrew Weir and Company Ltd and Mr R Dick and Mr R Drew of Ellerman Lines Ltd (later Cunard Ellerman Ltd) in connection with the latest Doxford engines. Most of the

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Discussion

G Victory (Past President, IMarE) Dr Ørbeck says that the Doxford engine 'was considered by many to be the best (marine) engine in the world.' He may be pleased to know that I am one of the many. However, that was not the engine which Dr Ørbeck speaks of in his paper, though, given time, it is possible that it might have regained its previous good name.

So, although the paper deals with developments from 1960, we have to go back beyond that date to find out why the good name of the Doxford engine was so eroded that it could not be resurrected except by a complete redesign and the jettisoning of many of the features which had made the Doxford so popular with shipowners and marine engineers.

I think that I am entitled to comment on these problems as I served part of my apprenticeship at Doxford, I first went to sea on a Doxford engines ship built in 1926, and I was DOT surveyor in Sunderland and covered Doxford during the crankshaft problem period of the early 1950s.

Mr Keller, an engineer of genius to rank with the greatest, designed a completely new engine in almost all respects, one which depended for its success on a number of novel features which broke with almost all existing concepts of the day, and which combined to make the Doxford a favourite in the marine engine world for over 30 years. Yet all this has now been jettisoned. The features referred to were:

1. The opposed piston concept – borrowed from Junkers.
2. The solid injection of fuel – the first engine, I believe, to adopt this method.
3. A brilliant design of fuel valve, using a pressure balancing spindle to reduce the operating effort, leading to a smaller camshaft and quieter fuel pump. With its ability to vary the lift period, the point of opening and the duration of injection it was an engineer's delight, allowing an engine to be tuned like a fine piano.
4. A cast iron bed-plate (until about 1933), which limited the flexing of the crankshaft. The much more flexibly fabricated bed-plate coupled with the more flexible hulls of later ships could well have been a major factor in crankshaft breakages.
5. 'Spherical' bottom end bearings which could accommodate a reasonable amount of crankpin deflection.
6. 'Bottle' guides which provide better location for the crosshead than do flat guides.
7. Water cooled pistons, which are less likely to get clogged up and cause contamination of the lub oil, and piston skirts which shielded ports and avoided leakage after the piston had passed the ports.
8. A swing arm supply and return of the piston cooling water – no glands, reduced pulsating pressures and, if properly looked after, no leakages. (The crankcase gland and elbow wear could have been solved by using an oil-proof hose and a quadrant as in later upper piston assemblies.)
9. Even the centre scavenge pump, providing large quantities of air at relatively low pressure, might have been an advantage as it effectively separated Nos 1 and 2 cylinders from Nos 3 and 4 cylinders, and with its lower stresses could better accommodate crankshaft deflections – always greater on the scavenge crank webs.

Perhaps Dr Ørbeck could tell us why it was necessary to abandon all these except the opposed piston and solid injection, which were successful features in the 1950s and 1960s.

So why did Doxford lose the good name and good reputation which they had in the wartime years and why, in the late 1950s, was it necessary to consider redesigning the entire engine? Perhaps it was because of the changing pattern of operation as Engineer Officers changed ships more often – the Doxford relied on personnel who had good experience of the engines, and preferably of a particular ship and engine, in order that maintenance would be carried out in a tried and repetitive manner; perhaps it was because of the move towards higher power and mep. Many consider that it was because of the onset of problems with broken crankshafts in the late 1940s and early 1950s, but in my opinion a good deal of the damage had already been done by the complacent and easy going attitude of the management. Times were good, orders came easily and they expected this to go on for ever, with the result that they failed to realise that when breakdowns did occur the easy answer that the Doxford engine could not be to blame, and, therefore, any breakdown must have been due to mismanagement on the ship, was just not good enough.

As an example of this attitude I know of one twin-screw Doxford, built about 1943, which had six pistons fracture around the inner corner of the upper piston ring groove, after about three years service. This happened on several ships built at about the same time. Doxford did not appear to have much interest and did not examine the pistons. When new pistons were obtained and fitted throughout it was found that a change of design had taken place.

When older pistons had developed cracks on the crowns it had been decided, in the early 1940s, that this could be cured by thinning the upper part of the piston wall above the ring groove. This appears to have been overdone and the wall thickness in way of the upper piston ring groove, a fairly obvious stress raiser, had been reduced to 7/16 in – and that is where the fractures had propagated. However, the replacement pistons had been modified so that the thinning was not taken so close to the ring groove. Yet when in the 1950s I was able to examine the Doxford records for the ship in question I found a terse comment: 'Operator fault; engine run without cooling water.' And that was without examining the evidence which would have indicated fatigue failure. I knew that the verdict was not true for I had been Chief Engineer on that particular ship at the time and I would have known that the engine had not run without water, nor had the low pressure alarm operated at any time.

The same apathy persisted when cases of crankshaft breakages began to come in – not in 5 cylinder engines as Atkinsons would have us believe but in the No 4 forward dog-leg of 4 cylinder engines. The first two were not taken too seriously, the blame was put on the engineers for allowing the engine to operate with excessive crank web deflections. Only when the 5 cylinder engine breakages began to come in was the problem taken seriously. By that time much of the confidence in Doxford engines had been lost – and it was never regained.

It is my opinion that the real cause of the breakages on a type of engine which had been in service satisfactorily for nearly 40 years has never been established, though the effort to get higher powers and a shorter engine did not help. One possible cause, the adoption of Michell pads for the shaft bearings of the thrust shaft, was never investigated though Bill Cantess and I thought it should have been.

The change meant that, although the engine main bearings and the tunnel bearings could wear down, the Michell thrust could not, and it is possible that the shaft had to 'run over the hump' so increasing the deflections and the stresses in the No 4 dog-leg.

Another possible cause of the breakages might arise from the machining of a semi-circular recess in the dog-leg faces in order to reduce the length of the engine slightly. Atkinsons said that this increased the strength of the dog-leg, but this is difficult to comprehend as the fractures all started at this fillet. Admittedly the engine needed a stiffer bed-plate and a stiffer hull in way of the engine bed-plate, but does Dr Ørbeck think these other alterations to design could have contributed?

It was because of this loss of confidence and the fashion, at the time, to go into higher powers by increasing the mep and maximum pressures that caused Doxford to go off in a 'lemming like' rush to get out another engine in an endeavour to get back into the market, and this, in effect, is the background to Dr Ørbeck's paper. Practically everything was changed at one time, and the cardinal virtue of changing one thing at a time and looking to the effect before changing the next was lost. So problems and partial solutions came thick and fast – as did the losses. And to compound the problem of cash flow it was decided to embark on the Seahorse engine which, because of its exhaust port angle of advance, was condemned to be unidirectional, with the consequential penalty of higher fuel consumption from reversible gearing or electrical drive. It is worthy of note that in 1933 I saw Mr Keller running the 3 cylinder 1929 North East Coast Exhibition engine at 300 rev/min. Why could that not have been taken as a basis for development without any exhaust port advance? I understand that Mr Keller was aiming to market it as a land generator but so far as I know it was killed by the depression. The Seahorse in turn was killed by the over- complication of the scavenge compressor drive.

With such a background and without a fathomless well of finance it seems that Dr Ørbeck and others were batting on a sticky wicket. It was ironical that during this time the 'fashion' among diesel engine builders and users swung back to the slow speed, long stroke concept for which the Doxford design is most suitable. It says much for the patience and professionalism of Dr Ørbeck and others that they reached the position where Dr Ørbeck is able to say that the unique advantages of the direct drive 3 or 4 cylinder opposed piston engines, to their latest design, would have a very competitive specific fuel consumption, and therefore be a marketable commodity. It was, indeed, unfortunate that at this juncture Leo Curran and the British Shipbuilding Board decided to stop building Doxford engines. However, although they killed off 'a' Doxford engine, they did not kill off 'the' Doxford engine – the best marine engine in the world. That was killed in the 1940s and 1950s by the apathy and sloth of the management and workforce at that time. Doubtless Mr Keller turned in his grave for in the 1940s there was no one of his calibre to carry on or to maintain the discipline in the works for which he was noted.

F Ørbeck (Orian Technology Ltd) First of all may I thank all the contributors to the discussion for their many and varied contributions. This shows that the Doxford engine still is a subject which is viewed with interest and fascination.

G Victory has experience with Doxford engines going back many years prior to my appointment by the company in 1959. I am grateful for his reference to earlier Doxford engines as this gives a greater depth to the subject and I will discuss the nine features he mentioned in the same order as in his contribution.

1. The opposed piston concept was rightly retained by the company till the end and it should be clear from the paper that I still believe this to be the best two–stroke principle.
2. We believe that Doxford was the first engine builder to adopt solid fuel injection. This method has now been adopted by all engine builders but few, if any, have made use of the unique fuel system pioneered by the Doxford engine.
3. The mechanically operated fuel injectors were abandoned because they were expensive and required camshafts both on the front and the back of the engine. However, in some respects they had better injection characteristics than the present fuel system and something was, therefore, lost in the changeover.
4. The early flexible fabricated bed-plates were indubitably a contributory factor to the crankshaft failures.[4] However, later improvements in the design made these bed-plates very satisfactory. Fabricated bed-plates are cheap when only a few units of a particular design are required, and they are easy to repair. This method of construction is, therefore, preferable for new engine types. A properly designed cast iron bed-plate can, however, be preferable if the casting is not too big and this alternative was considered as a future development of the 58JS3 engine.
5. The development towards higher power and careful attention to the torsional and axial vibration characteristics resulted in much stiffer crankshafts for the P and J-engines compared with earlier engines. This made the 'special' bearings superfluous and resulted in considerable cost reductions.
6. The bottle guides for the upper pistons were removed with the introduction of the P-engine as it was considered that the bearing ring on the piston, in conjunction with the crosshead guides, provided adequate location. This was generally successful.
7. Water is a better cooling medium than oil and the upper piston, which is the hotter of the two pistons in the cylinder, was water cooled in all the later Doxford engines as in the earlier engines. The lower piston can be adequately cooled with oil and this avoids the danger of water contamination of the crankcase oil. The later Doxford engines, therefore, changed to oil cooling of the lower piston and this was generally successful.
8. Properly designed telescopic pipes work very well and the same can be said for swinging links. Swinging links were introduced again for the upper piston cooling of the Seahorse engine and they worked satisfactorily during the extended test-bed trials. However, telescopic pipes are more compact and neater, and introducing these was generally considered a step forward.
9. The centre scavenge pump was replaced by lever driven scavenge pumps to reduce the engine length and improve the torsional stiffness of the crankshaft. The dilemma often is that for alignment purposes one wants a flexible shaft whereas for vibration reasons a stiff shaft is preferable. Throughout the life of the Doxford engine the development has been towards stiffer shafts and to satisfy the alignment requirements it has, therefore, been necessary to increase the stiffness of the bed-plates and the double bottoms.

Mr Victory talks about the complacency of the management at Doxford in the late 1940s and 1950s. I started with the company in the autumn of 1959 and the technical management of the company had by then changed substantially. It may be easy to blame the management of the past but I believe that many opportunities had been lost in the post-war period and that it was difficult to recover the situation. On the positive side I must say that both the P-engine and the J-engine represented substantial technical advances and much credit must go to Percy Jackson's technical leadership.

Mr Victory then turns to the crankshaft failures in the 4-cylinder engines. To the best of my knowledge these were confined to aft end installations which used very stiff intermediate shafts in order to place the 4th order I-node torsional vibration adequately above the running range. These thick intermediate shafts were unable to accommodate the hull deflections and the thermal rise of the engine, and the results were problems with the crankshaft after end. The adoption of Michell pads for the thrust shaft bearings could have aggravated this situation. When I arrived at Doxford, Mr G Oliver had made considerable progress with these problems having established factual information about hull deflection and thermal lift. Such information was included in the alignment instructions shown in Fig 2. The limits in the vertical plane are bigger on the positive side than on the negative side, which is a result of the effects of hull deflection and thermal lift of the engine.

The undercut fillets used in the side webs came from some research work carried out on the continent. In my opinion this was a poor design.

I do not agree with Mr Victory's statement that a 'lemming like' rush took place to get out another engine. Steady development in engineering has many virtues but the pace of development must be adequate to keep up with the competition. Both the P-engine and the J-engine represented substantial progress. They both suffered initial development problems but by the end of the 1960s Doxford was in a much better position competitively than at the end of the 1950s.

I am grateful to Mr Victory for his comments about the patience and professionalism of myself and the few others who have served the Doxford engine till today. May I again thank him for his contribution which contained a number of valuable points.

Cdr E Tyrrell (Retired) I should first like to congratulate Dr Ørbeck on the technological excellence of his paper. I had been with DSIR and DTI over the period covered by the paper, and would like to refer to the work carried out by Professors Horlock and Benson on turbochargers of which the results had been offered to Napiers. At that time Napiers were plagued with labour problems and low productivity at the Liverpool works resulting in the manufacture of turbochargers being transferred to the Ruston Gas Turbine works at Lincoln. This move caused considerable disruption and Napiers concluded that they did not have the resources to undertake further research work on turbochargers at that time.

DTI provided some financial assistance for the trials of the Seahorse engine and the criteria for the 500h trial was agreed between DTI and Doxford Hawthorne Research Services Ltd.

DTI investigated a number of studies on both the technological and commercial aspects of the Seahorse development. It seemed that the thermal loadings on the Seahorse were higher than any being experienced in a medium sized 2 stroke engine then operating satisfactorily. This contention was to some extent confirmed by the piston ring scuffing experienced during the trial and referred to in Fig 10b of the paper.

A visit to Fairbanks-Morse in the United States, who were then engaged in the development of a very similar engine to the Seahorse, provided some interesting information.

Fairbanks-Morse were satisfied that the technological problems with their engine had been solved, but at somewhat lower thermal ratings than those proposed for the Seahorse. However, detailed production costs showed that the engine could not compete in price against the established competition, and that the more favourable propeller characteristics with a gear box did not fully compensate for the gear box losses. Hence the fuel consumption in a ship was likely to be greater than with a direct drive engine. Fairbanks-Morse, despite an expenditure of some $35M dropped the project and took a Pielstick licence.

Production costings at Doxford showed that productivity, measured in terms of value added per man hour, was under half that in an engine works on the continent. Doxford labour rates, measured at the then exchange rate, were about half those on the continent. However, the continental works were fully operated for two shifts and partially for three. Doxford were fully operational for one shift and partially for a second. Hence the overhead costs on the continent were less and the continental production costs lower than at Doxford.

Further investigations showed that it would require an output of about 200 cylinders a year to pay for the research and development costs over the projected life of the engine. This was quite outside the capacity of the Doxford works.

DTI therefore decided that the engine was probably not technically viable at its design ratings, nor commercially viable.

As a result of the technical studies DTI and their technical advisers suggested that Doxford might be better engaged in the development of a large 2 stroke engine with a cylinder head exhaust. This suggestion was rejected. Time has shown that the valve in the head 2 stroke has been the subject of remarkable development in recent years.

F Ørbeck (Orian Technology Ltd) Cdr Tyrrell's contribution throws some valuable light on the attitude of DTI to the Seahorse engine and their recommendations for further action. First, however, he provides some further information on the research work carried out by Professors Horlock and Benson referred to in the paper and I would like to answer as follows. This work did to some extent apply to turbochargers but was mainly concerned with predicting the pressure pulsations in exhaust pipes, ie a problem in engine design. The work was too academic to be of much use to Doxford and this was unfortunate because Doxford needed support in this area.

The Fairbanks-Morse engine was then mentioned in connection with the Seahorse engine development. The Fairbanks-Morse engine used a smaller bore for the upper piston than for the lower piston in contrast to the Doxford engines. This design resulted in relatively small exhaust ports and high thermal loading of the engine. The Seahorse engine on the other hand had satisfactory thermal loading with higher air flow and trapped air than the conventional Doxford engines. The piston ring scuffin which plagued the Seahorse engine was, in my opinion, basically due to the high piston speed of the lower piston, 8.8 m/s, which was well above 2 stroke engine practice.

The following, quoted from Cdr Tyrrell's contribution and concerning the Seahorse engine, is interesting: 'DTI therefore decided that the engine was probably not technically viable at its design ratings, nor commercially viable'. The same conclusion had been reached by many of the personnel at Doxford but it was contrary to company policy. DTI reached this conclusion partly on the basis of the production costs at Doxford which were high on account of low productivity. Nevertheless, when in 1977 the company received orders for seven 58JS3 engines and eight 76JC4 engines it became profitable for a while, ie these latter engines must have been relatively cheap to produce. After some initial problems the 58JS3 and 76JC4 engines have given excellent service performance.

DTI apparently suggested the development of a large valve in the head 2 stroke engine, but even if this had been done with the same technical skill as by B&W, Doxford would have lost out on account of its production costs. Therefore, the only route to success for Doxford was to develop their own direct drive engines. This course of action was missed as a result of 'red herrings' and lack of foresight, but I believe that it is still open.

J P P Pillai (Marine Consultant) My first and last experience with Doxford engines was as a first year 'look do not touch' apprentice in the early 1960s. I have not had the pleasure of working with a Doxford since then. Hence I value the content of your excellent paper as an education to myself.

Could you give me an indication of the number of Doxford powered ships still afloat? Which organisations look after the servicing of Doxford engines at present? As a matter of interest have any Doxford engines been used for land installations, like power stations?

F Ørbeck (Orian Technology Ltd) In answer to Mr Pillai, in June 1986 when we last investigated the matter, there were 164 Doxford powered ships in service. This has probably by now been reduced to about 100.

The following organisations offer service to Doxford engines:

Britparts: supply of spare parts and service;
Doxford Design Engineering Ltd: design and drawing services;
Orian Technology Ltd: technical consultancy and design service.

No Doxford engines have, to the best of my knowledge, been used for land installations.

A J Wickens (Retired, formerly Chief Engineer, Hawthorn Leslie (Engineers) Ltd) It is my good fortune to have been associated with Dr Ørbeck for very many years. This is a paper which had to be written and no one is better qualified to recount the events.

The author shows how a fundamental approach to transmission system vibration and alignment calculation paid handsome dividends in eliminating crankshaft failures. At the time Doxford were probably ahead of the field in this area and I would like to ask whether they adopted the practice, subsequently used by other engine builders, of defining to installing engineers acceptable bending moments and shear forces at the crankshaft coupling.

To introduce some personal recollections, I was closely involved with the development of the Seahorse engine in the 1974–1977 period. I would not disagree with the author in his condemnation of the mechanical auxiliary scavenge blower arrangement adopted for the initial design. However, it should be remembered that the prototype unit was not an engine on a test bed but a test bed in itself and as such possessed a number of redundant features which were included solely for evaluation. As well as powering the auxiliary blower (which would have subtracted some 500 hp from the useful crankshaft output), the auxiliary drive operated counter rotating balance weights to deal with the primary couple referred to and cams for a possible jerk pump fuel system. All these can be seen in Fig 9. The production engine, whose design was well advanced at the time of abandonment and for which crankshaft forgings had been delivered, would have had zero lead exhaust pistons, turbochargers assisted only by a small electric blower and the traditional Doxford common rail fuel system. These would have rendered the auxiliary drive arrangement redundant and, in fact, the engine would have had a marked resemblance to a 58JS4.

A retrospective account of the Seahorse engine could form a paper in itself. However, Dr Ørbeck has struck just about the right balance in his recounting of it within the total Doxford story.

F Ørbeck (Orian Technology Ltd) Mr Wickens asks if the practice by other engine builders of defining to installing engineers acceptable bending moments and shear forces at the crankshaft coupling was ever adopted by Doxford. I have high regard for this practice but the answer is no. Doxford was near the end of their engine production when this practice came into use and since Doxford prepared the complete alignment instruction for their last installations, there was no call for the use of this practice.

Mr Wicken's contribution makes the story of the Seahorse engine more complete and I agree that this could fill a paper on its own. It is certainly true that the complexity of the first Seahorse engine was largely due to this engine being the test engine. Unfortunately, I think we fell for the temptation of introducing too many new features, and when some of these failed it tended to give the engine a bad name. On the positive side the engine was the test facility for a number of new successful designs. For example, four injectors per cylinder were introduced with this engine and the success of this arrangement was valuable information during the development of the 58JS3 engine. The Seahorse engine was also the first constant pressure turbocharged Doxford engine. However, in the long term the engine would never have been competitive on fuel consumption and I strongly believe that we should rather have pursued the direct drive engines.

J R Bambridge (Doxford Design Engineering Ltd) As one who worked with the author over the period covered in the paper, I would like to compliment him on the technical content and the interesting comments raised.

I would like to add some comments of my own under the running gear and bearings section covered in the paper. In Fig 10b it is shown that after two years of prototype testing there still remained the problem of bearing failure. The Seahorse engine was a radical step in Doxford engine bearing design in that it introduced thin wall shell bearings to the bottom ends of the side and centre connecting rods. The bearing pressures and speeds were much higher than had been experienced previously and, together with new lubrication methods and bearing materials being tested, it was understandable that problems could be experienced. However, at the end of the development period, I think that the answer was the need for greater manufacturing accuracy in the crankshaft and the bearing housing/shell interface, and also in the pre-tensioning of the running gear bolts and studs.

The experience gained with the Seahorse engine was invaluable to the design of the 58JS3 engine in 1977. In view of the very short design period, and the necessity to keep the weight and engine dimensions to within specified limits, it was imperative that bearing sizes were optimised at an early stage. The successful design of the crankshaft, running gear and bearings was achieved in no small part due to the experience gained between 1972 and the end of 1975.

F Ørbeck (Orian Technology Ltd) Mr Bambridge described how the Seahorse engine was used to advance the design of running gear and bearings. This is another area in which this engine served as a valuable test facility and I wholeheartedly agree with Mr Bambridge's contribution.

H D Makinson I read with interest that the calculation procedures associated with turbocharging within the company were limited to the calculation of port area diagrams.

One is left wondering if Professor Benson's earlier work on flow through exhaust ports and on port timing, published in the *Journal of the Royal Aeronautical Society* in 1955 and 1957, and in the *Engineer* in 1957 and by IMechE in 1959, was utilised by the company prior to 1961.

F Ørbeck (Orian Technology Ltd) Mr Makinson has touched an interesting point. I am certain that Professor Benson's earlier work on flow through exhaust ports and on port timing was not utilised by the company. Probably there was no one available within the company with sufficient time and capability to be able to utilise this work in practice. Perhaps the money would have been better spent on employing such a person than on taking part in centralised projects.

Dr A Fowler (The University of Newcastle-upon-Tyne) Dr Ørbeck's paper presents fascinating reading for those such as myself whose initial introduction to marine propulsion plant was via the courtesy of Doxford of Sunderland. The historical content of the paper lays down for posterity an important chapter in the evolution of marine diesels, written with an insight which must be virtually unique.

The paper also addresses several issues which extend beyond historical interest, involving questions concerning combustion and the gas exchange process in uniflow engines.

1. First a relatively minor question of clarification. When discussing the specific air flow for the Doxford engines, how are the stated values defined; for example, does a figure of 7.3 kg/bhp h with a pressure drop of 0.105 bar correspond to the gross turbocharger output (giving an estimated full load air-fuel ratio of approximately 49:1)? If this interpretation is correct, what percentage of this flow would be typically 'trapped' in the engine cylinders?
2. The information given in Table III shows a significant improvement of in-service fuel consumption, following the fitting of modified piston heads. It is suggested that a partial contribution to the improvement was due to the relatively poor condition of the original piston heads. Is it possible to separate out these two effects, using knowledge gained from past experience, to predict precisely to what degree the improvement obtained could be attributed to the geometric modifications to the heads? Also can it be confirmed that 'like for like' comparisons were made, and that other variables were not simultaneously changed by, for example, cleaning of the hull or the turbochargers?
3. It is stated that after fitting the modified piston heads to the 76JC4 engine the scavenge pressure dropped (by typically 0.2 bar), whereby smaller turbine nozzles were required to restore the scavenge pressure. However, the corresponding exhaust pressures (or alternatively the scavenge pressure gradient) is not stated in the paper. The deduction that the airflow reduction of 19% corresponds directly to the area reduction arguably rests on the assumption of equivalent exhaust conditions also. Could the author clarify this point?
4. The proposed modifications to the timing valve to provide improved part-load performance appears to be a promising development of the common rail fuel system. Can the author provide a typical cylinder peak pressure against engine speed characteristic, corresponding to the proposed fuel control system, together with an estimate of the equivalent limiting peak pressure characteristic, defined by engine loading constraints? Are these likely to be significantly different to the values applicable in more conventional engine designs?

Stimulated by the contents of this paper a computer simulation study has been implemented at Newcastle University to assess the implication of some of the points raised. Detailed results are as yet unavailable, but preliminary investigations appear to support Dr Ørbeck's predictions of improved sfc and other opposed piston engine operating characteristics. It is anticipated that the outcome of this study will be disseminated in due course, and it will be interesting to assess to what degree this computational analysis explains and confirms the ideas presented to us in Dr Ørbeck's paper.

F Ørbeck (Orian Technology Ltd) Dr Fowler raises a number of points of academic interest and it is appreciated that accurate answers to these would be of importance for further development of the engine. Unfortunately the figures available should be considered with some reservation.

1. The figures of 7.3 kg/bhp h airflow and 0.105 bar pressure drop across the cylinders refer to 76J pulse turbocharged engines. Heavy fuel oil has a stoichiometric ratio of 14 and we estimated that 60% excess air was trapped in the cylinder. Thus, the trapped air ratio would be typically 22.4:1.
2. Referring to Table III it can be confirmed that the comparisons were 'like for like' to the extent possible. Unfortunately we are not able to separate the effect of worn pistons from the effect of the upgrading package.
3. We have no further information on this case but thought that it was worthwhile drawing attention to the information provided.
4. An order was received for the timing valve modification which gave the characteristics shown by the dotted line in Fig 20. Unfortunately the order was cancelled as a result of much delayed delivery from the manufacturer. No service results were therefore received, but our own approximate predictions suggested that the engine characteristics would improve substantially.

It should be apparent from the above that much has yet to be learned from the results of the various upgrading packages and a detailed computer simulation study should be most valuable in this connection. I therefore look forward with interest to any further results from Newcastle University.

In conclusion I would like to thank all the contributors to the discussion. We may differ on a number of points but I am pleased that the discussion has enhanced the subject substantially and contributed to a more objective view of the history of the Doxford engine, as well as future possibilities.