The Foden Engine

During the year 1939, the management of Fodens Limited decided that the company should make itself independent of engine manufacturers. Led by Engineering Director Eddie Twemlow and known by everyone at Sandbach as ET, a team was formed to design an advanced diesel engine suitable for automotive use and marine applications.

A trio of Engineers, Jack Mills (Chief Engineer), Martin Britain (Engine Department Manager) and Rod East (Head if Vehicle Development) took the basic design to high levels of reliability, power and economy throughout the ‘50s, ‘60s and ‘70s. An account of the development has been written by Rod East and reproduced here.

A TALE OF A TWO-STROKE - THE FODEN FD ENGINE

By Mr R W East BSc. CEng. MIMechE

# Preface

Maybe I am not the right person to relate this story; I joined Fodens Ltd in January 1956 and did not get involved directly with engine development until December 1957. For the account of what happened before that date, and for a list of what happened thereafter, I have had to rely heavily on other peoples’ memories and records, and particularly on the two men most closely involved with the Foden engine; Jack Mills, Chief Engineer and later Technical Director of Fodens Limited, and Martin Britain; Experimental Department Manager and later Engine Department Manager. Without their generous and invaluable help I could never have started. I am grateful to them both for their help and patience, and to many others, like me, who were lucky enough to be involved in such an interesting engine by such a small but enterprising British company.

Although he had died before I had started to write this book, it would be quite unfair not to mention the contribution of Eddie Twemlow to the development of the engine. A member of the Foden family, and a self-taught engineer with a natural flair for design, he kept a watching eye on the engine design and development work and made major contributions to the team effort throughout the life of the engine.

For a variety of reasons, the original company, Fodens Limited, went into receivership and was bought by the American multi-national group, Paccar. At this time a very large part of the old company’s technical records were destroyed or dispersed. Consequently this book will not be a technical record from day one but a story of how a small engineering company could, in those days, tackle a major technical project with severe restrictions on resources, but none on enthusiasm and talent. Inevitably, relying on peoples’ memories of events taking place many years ago, mistakes are likely to creep in and I apologise for any that may have occurred.

Chapter one

Why and How?

In the 1930s Fodens made a rather belated change from steam-propelled vehicles to diesel lorries. The first diesel vehicle was fitted with a Gardner 5L2 engine and was sold to Samuel Jackson of Haslington (it was later bought back and preserved as no. 1 Diesel). However the range of engines that soon established themselves as head and shoulders above other diesel engines suitable for use in road vehicles, was the Gardner LW range, in 4, 5 and 6 cylinder versions.

Although Fodens tried other makes of engines, none were considered good enough for what was one of the best, if not the best, lorry available in the U.K. However Gardner engines were in great demand by other vehicle manufacturers as well as many other users, particularly marine engineers. Fodens output of trucks was in effect limited to eleven vehicles per week by the "ration" of Gardner engines they were able to get.

In a typical enterprising Foden decision William Foden, then head of the Company, decided that Fodens would make their own engine. In the first place the aim was to produce a four-stroke engine to compete with the Gardner LW engines. To this end a small development team was set up consisting of two engineers with two fitters and a single-cylinder Russell and Newbury engine. This engine was chosen as it had the same bore and stroke, 4.25" (108 mm) x 6" (152 mm), as the LW engines.

Initial tests of this single-cylinder engine resulted in fuel consumption some 25% worse than that given by a 6LW. The engine was converted to a Gardner LW block, piston and combustion system and then gave fuel consumptions 10 to 15% worse than a 6LW. This was in part due to the proportionally bigger parasitic losses, friction, power to auxiliaries etc, inherent in a single, which was accentuated by the very low parasitic losses of the Gardner LW engines. Another factor was one of the characteristics of these engines which were very sensitive to fuel injector performance and only injectors, or sprayers as they used to be called, made or reconditioned by Gardners would produce the excellent fuel consumptions for which the LW engines were famed. Try as one might injectors reconditioned elsewhere never produced the same standard of performance.

Realising that this problem was not going to be easy to solve, and that to match Gardner performance and reliability would be very difficult and likely to take a long time, consideration was given to other features, which could be improved over the standards set by the 6LW. The importance of payload and consequently of unladen weight suggested that a light engine would have an advantage but Gardner engines with their aluminium crankcases and separate bolt-on cast iron cylinder blocks were fairly light already. The one way to go which would make an even lighter but still durable engine was to go smaller - to a two-stroke with its potential for the same power from half the swept volume.

The Russell & Newbury engine was then converted to a uniflow two-stroke, (a uniflow two-stroke is one with piston controlled air intake ports in the lower part of the cylinder and exhaust valves in the cylinder head), using Foden designed and made cylinder, piston and cylinder head. The piston was made of cast iron in Fodens own foundry incorporating a hemi-spherical combustion chamber and was designed to use Gardner piston rings. The cylinder head had a single exhaust valve. Despite having been heat-treated (again at Fodens), after two hours running the piston "grew" 1/32" and hit the cylinder head! It was re-machined, re-fitted and the tests went on. Trouble was experienced with distortion of the single non-rotating poppet exhaust valve. A sodium-cooled version did not effect a cure and as the best brake mean effective pressure was half that of the four-stroke, no progress could be made in reducing the swept volume of the engine. A further attempt was made by converting the engine to a loop scavenge two-stroke, with inlet and exhaust ports in the cylinder liner but this gave even worse results.

This point was reached in the middle of 1939 and the outbreak of war diverted attention from engine development to war work. However, two years later pressure on the small design team had eased and some thought was again given to the Foden engine.

Armstrong Whitworth Securities, an engineering development organisation at Slough, had been doing a lot of work on two-stroke diesel engines using the Kadenacy patents. These patents centred on the use of pressure waves in the exhaust system to improve scavenging of the cylinder after combustion of the fuel. By the timing of the opening of the exhaust valve in the cylinder head when the pressure wave action creates a low pressure at the exhaust port, and the closing at the time of a high pressure wave at the port, good scavenging of the exhaust gases can be obtained without excessive loss of clean scavenge air and with minimum pumping losses. This is associated with a supply of air from a mechanical blower through piston-controlled air ports in the cylinder liner. Air movement through the cylinder is thus from bottom to top, hence the use of the name "Uniflow" as opposed to "Loop Scavenge" used in some two-stroke engines which have both inlet and exhaust ports in the cylinder walls. Loop scavenge engines usually sacrifice performance for simplicity of construction. This efficient scavenging of Kadenacy engines results in effective clearing of the exhaust gases and a denser charge of clean air for the next firing stroke. The power developed by an internal combustion engine depends very much on the weight of air or more accurately, oxygen, available for combustion.

Two senior Foden engineers visited Armstrong Whitworth at Slough and were most impressed to see Kadenacy two-stroke diesel engines pulling 100 pounds per square inch brake mean effective pressure, which was not a bad level of performance for a naturally-aspirated four-stroke in those days, and this with clean exhausts. In particular a Petter engine of 85mm bore and 120mm stroke gave 21bhp per cylinder. Such a performance in a five-cylinder engine would give 105bhp, which compared nicely with the 102bhp of the Gardner 6LW yardstick. It was not thought that the Petter engine itself would be suitable for a lightweight vehicle engine as it was designed for industrial use and was very sturdily made of cast iron, however it showed what could be done from a performance point of view. Fodens were very impressed by what had been seen at Slough and took out a licence from Armstrong Whitworth to design and build diesel engines in line with Kadenacy principles. This included a set of drawings of the 85mm x 120mm engine and these basic bore and stroke dimensions were adopted for the proposed Foden engine.

It was now necessary to design a suitable vehicle engine based on the combustion system of the Armstrong engine. As previously stated, low engine weight was very important and Fodens were familiar with the lightweight construction of the Gardner 4LK engine. Consequently they decided to adopt a one-piece aluminium cylinder block and crankcase with cast iron liners. The firing loads were taken by long through-bolts from cylinder head to crankcase main bearing caps so as to avoid high stresses in the DTD 424 aluminium casting.

In order to get the level of output wanted from the engine, a high degree of swirl was needed to ensure efficient mixing of the air and fuel to get good combustion. This was to be achieved by casting guide vanes in the crankcase air chest to line up with ports cast in the cylinder liners. This constituted a challenge for Fodens own Pattern Shop and Foundry, a challenge that was accepted and met. Similarly the target to produce swirl-inducing air ports in the cylinder liners, which would not require expensive finish machining was achieved - but more of this later. Suffice it to say that the Pattern Shop and Foundry made major contributions to the project.

There are several highly stressed components in a high performance two-stroke diesel engine. The piston and piston rings have to face severe thermal conditions. The pressures generated in the engine cylinder during combustion are very high making the design of such components as the gudgeon pin, connecting rod small end and cylinder head gasket critical. The working conditions of a poppet exhaust valve and fuel injector in the cylinder head are also very difficult and, because of the higher rotational speeds (engine speed instead of half engine speed as in a four-stroke), the valve gear and fuel injection pump need special consideration.

Chapter two.

Engine air movement and valve gear.

As previously stated, the power developed by an engine is very closely related to the weight of air that can be put through it. High rotational speed, easy breathing, efficient scavenging and pressure charging are all aspects to be considered.

The engine was designed to run at a governed speed of 2400rpm with an over-run safe speed up to 3000rpm and these parameters were used in the initial calculations. Also all early engine tests ran up to 2400rpm, but the speed was later limited to 2000rpm to improve overall fuel consumption. Fodens had realised that one of the factors involved in the good fuel economy of the Gardner LW range was the lower rated speed as compared with competing four-stroke engines from other manufacturers.

A full-power rotational speed of 2000 rpm was therefore chosen and also the use of a Roots-type scavenge blower. Initially the use of a single, large diameter 2.205", (56mm) exhaust valve was tried, as used in the Armstrong Whitworth engine. However this distorted and leaked, and it was later replaced by two 1.280" (32.5mm), diameter valves, an arrangement that stayed with the engine throughout its production life, despite occasional unsuccessful flirtations with three or four valve systems. There was a substantial improvement in engine performance and fuel consumption when the twin valves were introduced indicating much superior scavenging. The core of the cylinder being directly under the large valve head of the single valve, it was thought that this was not being cleared.

Valve lift was 0.382” (9.7mm) and, on the early and later marks of the engine, the valves opened 101deg after top dead centre and closed 42deg after bottom dead centre. The timing of "valve opening" and “valve closing" were taken at the moment when the valve emerged from a short 0.067" (1.7mm) counterbore or recess in which the valve head was a fairly close fit. This ensured that the valve was moving fairly rapidly at this point which accentuated the formation of pressure waves in the exhaust system. On Mk3 engines, on which the rated speed was raised to the maximum design rated speed of 2400rpm in order to develop more power, the exhaust valve was opened earlier to make the engine breathe more easily at the higher speed and to reduce the power required to drive the blower. On this version of the engine, the six-cylinder version of which was rated at 150bhp at 2400rpm, the exhaust valve opened at 98deg after top dead centre and closed 49deg after bottom dead centre.

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Air inlet ports of course, were cast in the cylinder liner and were uncovered by the piston at the outer end of the piston travel. Use of a fire ring in the top ring position and right at the top of the piston (see details of this in chapter four) gave precise and "sharp" control of the opening and closing of the air ports. On MkI and MkII versions of the engine, the ports opened 49deg before bottom dead centre, and of necessity, closed 49deg after BDC, on the MkIII engine the figure was 50deg either side of BDC, whilst the MkIV and MkIIV engines used ports opening and closing 44deg before and after BDC.

One of the results of the air chest guide vanes and the tangential cylinder air ports was, as desired, a high rate of air swirl in the cylinder. This was highly advantageous in that it made the use of a single hole fuel injector practicable, which led to good injector reliability and life. However at one stage it was thought that the high swirl rate might be leaving a core of unscavenged exhaust gas in the middle of the cylinder. Consequently some rig tests and engine tests were done on cylinder liners incorporating air ports, which were radial at the bottom and tangential at the top. It was hoped that this would improve cylinder scavenging but, whatever it did to the cylinder scavenging, it produced deterioration in engine performance.

As with virtually all highly rated two-stroke engines, a blower was necessary to give adequate positive scavenging of the cylinders. Fodens chose a Roots blower of their own design and manufacture (of which more later). This was mounted on the left-hand side of the aluminium combined crankcase and cylinder block, and shaft driven from the gear train at the rear of the engine. Blower speed was 2.04 times engine speed. The air was delivered into the air chest, which was a section of the cylinder block below the water jacket and above the crankcase. Guide vanes were cast into the air chest, which were aligned with air port bars of the cylinder liners so as to create the swirl of air movement in the cylinder necessary to get good mixing of the fuel and air prior to and during combustion. The accuracy and positioning of these guide vanes in what was a complex aluminium casting was one of the triumphs of Fodens own Pattern Shop and Aluminium Foundry. Another triumph was the accurate casting of the ports, with a smooth surface finish, in the centrifugally cast iron cylinder liners which eliminated the need for a costly finish-machining operation.

The blower drive was designed to take up to 26 bhp, a figure that could be reached on over-run at maximum speed. In considering blower horsepower the Kadenacy principle must be borne in mind. The blower horsepower on a Mk1 engine at full load at 2000rpm was 15 with a pressure in the air chest of 4psi. If fuel was cut off and the vehicle was allowed to over-run, the air chest pressure would rise to 6psi, corresponding to a blower horsepower of 20. This shows the benefit of the Kadenacy effect on blower power and consequently on fuel consumption.

The blower itself had an aluminium casing, which also acted as a mounting face for the fuel injection pump (on some verry early engines the injection pump was mounted on a bracket cantilevered off the timing case). Straight, two-lobed DTD 133B aluminium rotors were die cast direct onto stainless steel shafts to get a coefficient of thermal expansion close to that of aluminium; another technique which showed the high standards achieved in the Foden foundries.

It is vital to minimise rotor to casing and rotor-to-rotor clearances if a Roots blower is to have a reasonably good volumetric efficiency. Two factors enabled Fodens to keep these clearances low, the rotor to shaft fixing which eliminated any possibility of looseness of the rotor on the shaft, and high quality, precision, driving gears, which controlled the meshing of the rotors. Here expertise gained in the manufacture of high quality gearboxes stood Fodens in good stead. A dog clutch was incorporated in the drive between the blower and the driving gear in the timing case so as to minimise damage in the event of a blower seizure, a problem that, while not unknown, was infrequent.

The straight, two lobed blower rotors had austenitic steel shafts to get a coefficient of thermal expansion fairly close to that of aluminium. They were turned between centres and then flats were machined on them to locate the rotor lobes. A shaft and two sand cores (to give hollow lobes) were positioned in a gravity die, which was filled with DTD133B aluminium. The lobes were form milled in four cuts: one rough, two semi-finishing and one finishing cut. Finishing on one side was not done until the other side had been roughed, to maximise accuracy. The rotors were dynamically balanced, a special lug being provided in the lobes for balancing holes to be drilled.

As mentioned above, good blower volumetric efficiency demands tight rotor to rotor and rotor to case clearances. These clearances must be kept small both when the blower is cold and when it is at its working temperature, particularly the latter. On the other hand rotors must not touch each other, or the casing. Also any dirt or grit escaping the clutches of the air filter must pass through without causing damage. A lot of development work was necessary to optimise these clearances. Rotor to case radial clearance was set at 0.005" to 0.007" (0.127mm to 0.178mm), rotor to case end clearance at the drive end at 0.007" to 0.009" (0.178mm to 0.229mm), and rotor to case end clearance at the free end at. 0.012" to 0.018" [0.305mm to 0.457mm).

End clearances were adjusted by shimming bearing housings and end plates. Rotor to rotor clearances were set by adjusting the phasing of the rotors. The two rotors were driven by meshing helical gears, and by altering the relative axial positions of the rotor and the gear by shimming, correct phasing could be obtained. The backlash between the two gears was limited to 0.001" (0.025mm) and the circumferential rotor to rotor clearances were not to be less than 0.005" (0.127mm) in any position and not more than 0.010" (0.254mm), with slightly greater clearance between the leading edge of the driving rotor (the rotor driven directly by the timing gears) and the trailing edge of the driven rotor (the one driven through the meshing helical gears).

In the later stages of engine development various schemes were tried to reduce blower clearances even more in the interests of better volumetric efficiency. The risk of seizure precluded the tightening of the manufacturing tolerances and attention was given to conformable rotor or casing coatings. One method tried was to cover the rotors with flock. This was a coating of short nylon filaments, which reduced leakage through the blower internal clearances. Unfortunately it increased the frequency of blower seizures and had to be abandoned.

An alternative method, suggested by Shell, was to apply a coating of grease to the inside of the blower casing. Many different types of grease were tested but none were found that would stay in place. However, during normal use, blower rotors, and the casing internal surfaces gradually collected a thin coating of oil and fine particles which gradually reduced running clearances, improved volumetric efficiency and did not promote blower seizures.

One of the drawbacks of the Roots blower is its high noise level, caused by shock waves at the blower air intake. General Motors, on their two-stroke diesel engines, tried to reduce this by using helical rotors. The way Fodens tackled this problem was to modify the involute shape of the blower rotor very slightly by machining a small radius on each side of the tip of the lobe so that the opening and closing of the blower chambers was less rapid. This made a slight improvement in noise level and was adopted, but it did not disguise the typical two-stroke note.

The exhaust valves were operated by forked rockers from a gear-driven, high-mounted camshaft, using roller-ended cam followers and short push rods, which were mounted in the cylinder heads. Camshafts speeds of 2000 R.P.M. and over necessitated careful design and manufacture to achieve satisfactory life and reliability. The valves were of KE965 heat-resisting steel with stellite tips although later versions of the engine needed higher grades of material for the valves

Chapter three

### The combustion system

For a two-stroke engine to have a reasonable power-to-weight ratio it needs to run at a reasonably high rotational speed. Additionally, combustion needs to be completed early in the power stroke because of the early opening of the exhaust valves necessary to obtain satisfactory scavenging of the cylinders. Apart from the difficult combustion dynamics resulting from this, the fuel injection equipment has to operate at engine speed. At the time of the initial development work on the engine this presented major problems.

Initial tests of the development single-cylinder two-stroke were done with a centrally mounted single exhaust valve and an offset, single-hole fuel injector. It soon became evident that the large single valve had a very short life and the change to two smaller valves was made. The first two-valve head had a central injector (of the type fitted to the Gardner 4LK engine). This was a three-hole injector and, being mounted between the two exhaust valves, proved impossible to keep cool enough to avoid lacquering of the nozzle. In order to get adequate cooling of the injector it was moved away from the exhaust valves and to the side of the combustion chamber. This helped in two ways. In the first place it made it practicable to house the fuel injector in a copper tube in the cylinder head well exposed to the cooling water flow. Also, as the injector was now at the side of the combustion chamber where the air swirl was at its maximum, it was possible to revert to a single-hole nozzle spraying fuel parallel to the side of the combustion chamber in the piston crown. Thereafter nozzle lacquering and hole blockage were virtually unknown.

Digressing for a moment, Jack Mills had designed a single-cylinder engine to carry out initial two-stroke development in 1942, and this was made and assembled, and first ran in the early part of 1943. This engine took over all the development work from the old Russell & Newbury based engine until multi-cylinder engines joined it in 1944.

Early development pistons had had a hemi-spherical combustion chamber in the crown but tests were done using the Foden single-cylinder engine to optimise the combustion chamber shape. Out of eight shapes tested the one that showed most promise was of toroidal shape and this continued more or less unchanged throughout the production life of the engine.

C.A.V, "A" type fuel injection pumps were used at first with an Armstrong Whitworth governor but they had to be fitted with stronger camshafts to cope with the forces generated by the higher rotational speeds involved. They were working at or above their limit and were later replaced by the Bryce rocker pump until the C.A.V. "N" type pump became available, followed later still by the "NN". The Bryce pump was designed for an operating speed of 3000 R.P.M. (pump speed). It was used with a Bryce hydraulic governor. The "N" and "NN" pumps were fitted with the C.A.V. hydraulic governor. This was an expensive but very good unit though not without its teething troubles. One of the advantages of the hydraulic governor was the light accelerator pedal load that it gave. Mechanical governors at high (or low) rotational speeds, gave unacceptably high pedal loads, as anyone who has driven a vehicle fitted with a Gardner engine will know. The C.A.V. "N" type pump was sturdy enough to have a satisfactory life at the high rotational speed at which it ran. It was used for all production engines until the advent of the "NN" which was even better, particularly with its engine oil lubricated cam box. This pump was first used on the Mk III engine. Fuel pump plunger size was 7.0 mm diameter on the Mks 1, 2, 3 and 6 engines, and 7.5 mm diameter on the Mks IV and VII engines, which were turbo-charged.

Injector pipes were originally 6.0 mm outside diameter 1.5 mm bore with a conventional nipple formed directly from the pipe material (soft iron). Repeated breakages were cured by going to 7.0 mm outside diameter soft steel pipe, 1.5 mm bore, and the use of a carefully designed double-cone fixing.

All 85 mm bore engines had short stem nozzles but long stem nozzles came in with the 92 mm bore Mk6 and Mk7 versions. All production engines were fitted with single hole nozzles, so positioned that the fuel spray was directed down the side of the combustion chamber through a gash in the piston top surface. This put the fuel where the air density and velocity were greatest, leading to good mixing of fuel and air. The swirl ratio, the ratio of air rpm in the cylinder to the engine rpm, was about 20:1

Chapter four.

Mechanical components.

It is now worthwhile considering some of the mechanical aspects of the Foden engine in its early stages. In high performance two-stroke diesel engines, problems are created by high operating speeds of valve gear and fuel injection equipment, high cylinder pressures and high heat transfer rates to combustion chamber components. All is not black however and lower levels of vibration due to better engine balance, and absence of high inertia loads on connecting rods, as seen in the induction stroke of four stroke engines, make some aspects of the designers work easier.

One group of critical components is the piston and its ring pack. With 2000 or more firing strokes per minute, thermal loading is high. All diesel engines have high maximum cylinder pressures in part due to the high compression ratio required to ensure satisfactory cold starting. This is accentuated in a blower-scavenged two-stroke by the supercharging effect of the blower at higher engine speeds and combustion pressures of the order of 1000 pounds per square inch were considered likely when initial design calculations were being made. At this time no satisfactory engine indicator was available for engines of this sort of rotational speed, which would have enabled cylinder pressures to be checked. It was much later in the engine development work when reasonably reliable engine indicator test results showed that pressures were reaching 1600 pounds per square inch. Thank goodness for conservative design loadings and factors of safety! These results were obtained using a cathode-ray indicator developed by Standard Oil. Fodens purchased one of these in 1944.

Piston and ring pack development was done in conjunction with Hepworth & Grandage. Early pistons were of aluminium with taper-sided compression rings to minimise the risk of ring sticking. A "cocktail shaker" was fitted to the top of the connecting rod. This is a device to catch oil and throw it at the underside of the piston crown to improve the piston crown cooling. Piston rings had to be located circumferentially on the piston to ensure that the ends of the rings ran up and down an air port bar and did not catch in the air port itself. This was done by means of a radial snug on the inner face of the ring and a corresponding recess in the ring groove of the piston. The aluminium pistons were supplied by Hepworth & Grandage including a batch with solid crowns into which Fodens machined different combustion chamber shapes. The shape chosen gave a nominal compression ratio (taken from cut-off by the piston over the air inlet ports and with a top dead centre piston-to-head clearance of 0.030" (0.76mm)) of 14.12:1. The weight of piston with rings was 2 Ibs 15 ozs, 1.33 kg. These pistons used a conventional top ring, not a fire ring, and did not give a very positive cut-off of the air ports due to the clearance between the piston top land and the cylinder liner particularly when cold, so that starting was poor.

It soon proved necessary to go to cast iron pistons to withstand the high running temperatures of the piston crown. It was also necessary to adopt the so-called fire ring right at the top of the piston to give sharp control of the opening and closing of the air ports in the cylinder liner. The development and use of this ring, pioneered in the Junkers Jumo opposed-piston diesel aero-engine, was one of the key factors in the successful development of the engine. The fire ring was free to rotate though it obviously it had to pass over the air ports and snagging of the ends of the ring was avoided by the two-piece construction of the ring with the gaps of the two pieces at 180 degrees to each other. The lower part, the fire ring sealing ring, restrained the upper part, known as the fire ring chamber ring, and vice versa. Also the horns of the chamber ring were chamfered.

The ring pack also included three wedge section compression rings, one air seal ring and an oil control ring. The wedge section compression rings were mounted in the top third of the piston and the wedge section was necessary to minimise the risk of ring sticking. The oil scraper was at the bottom of the piston skirt with the air seal ring just above it. The purpose of the air seal ring was to prevent leakage of boost pressure air from the air chest into the crankcase when the piston was in the upper part of the cylinder and the air-ports were below the three compression rings. As stated earlier the three compression rings were pegged to the piston to prevent rotation. This was also necessary for the air seal ring and the oil scraper for, although they did not travel over the air ports, they did travel over the lower part of the cylinder liner which incorporated two slots to accommodate the angular movement of the connecting rod.

This ring pack was retained for all production engines up to Mk III when one of the compression rings was deleted. On the 92 mm bore Mk VI and Mk VII engines, further improvements in piston ring technology permitted one ring to combine the duties of the air seal ring and the scraper, making a ring pack of four rings including the fire ring.

The connecting rod design had to take into account the high gas loads on the piston crown. The fact that there was positive pressure on the piston crown whenever the piston was in the upper part of the cylinder helped to counter the inertia forces generated as the piston approached and receded from top dead centre. With long cast iron pistons these inertia forces would have been high but they would only exceed the gas loads at engine speeds over 3800 rpm. Their absence allowed the design of the small end of the connecting rod to be shaped so that there was a large bearing area to accommodate the high downward gas loads plus the downward piston inertia loads and a narrow strap round the top where the gas load was in the opposite direction to the inertia loads. The gudgeon pin bosses in the piston were correspondingly shaped to give large bearing areas in the upper part of the bosses. Gudgeon pins that were of substantial size 1.625" (41.0mm) diameter, were a light press fit in the piston and did not rotate. They were however retained axially by circlips.

The big end bearings on Mk I and Mk II engines were 2.625", (66.7mm) diameter. Initially they were of white metal running with a "soft" shaft but experience showed that an improvement in life was needed and the bearing material, of both big ends and main bearings, was changed to lead-bronze lead lined, the crankshaft was left unhardened. The big end bearings were pressure-fed with oil from the crankpin and oil was fed round the back of the rod half of the bearing to a 1/4" (6.35mm) diameter hole rifle-drilled up the rod to the small end. This oil supply lubricated and cooled the heavily loaded small end bearing and also fed the piston crown cooling jets in the top of the rod, which had been introduced in place of the cocktail shaker, with the cast iron piston. The continual downward load on the small end made the supply of oil to the lower part of the small end bearing difficult and it was necessary to give a lot of design and development effort to maintaining a satisfactory oil film in the critical area.

The early crankshafts had concentric lightening holes in the crankpins and only one cross-hole drilled to feed oil to the big end of the connecting rod. Small end bearing life was not satisfactory and an additional hole, diametrically opposite the existing hole, was drilled through the crankpin. This communicated with a groove in the big end bearing and gave a continuous feed through the rod to the small end, improving the life of the small end bearing. Also the phosphor-bronze small end bush had three oil holes in the pressure half, fed from a groove in the outside diameter of the bush, each hole feeding an axial groove extending almost the full length of the bush. These grooves were so located circumferentially that the angular movement of the connecting rod caused the grooves to overlap their arcs of travel on the gudgeon pin. On later marks of the engine, where piston loads had risen considerably, more grooves were added to improve the lubrication of the highly loaded part of the small end bearing. This raised a nice matter of judgment between the conflicting demands for better lubrication and cooling of the bush, and adequate areas of bearing metal to carry the load. Another investigation into the oil supply to the small end showed that this was cut off in certain crank positions. The big end bearing shell oil grooves were modified to rectify this. Although rig tests showed that this was beneficial, worries about the effect of the modification on the big end bearing durability stopped any further work and the change was not adopted. This became even more important when turbo-charging led to cylinder pressures in excess of 2000 psi. The big end split on the rod was diagonal so as to allow the piston and rod assembly to be taken out upwards through the cylinder bore.

The cylinder liner, as mentioned before, was made in Fodensown iron foundry. Having consulted specialist liner manufacturers, Fodens were told that what they proposed was impossible - so they made them themselves. In order to get adequately long guide vanes in the liner walls it was necessary to have a thick-walled liner. This suggested wet liners, which was also desirable to ensure good cooling of the upper part of the liner by direct contact with the coolant. The high thermal and pressure loadings demanded a good, strong, homogeneous casting and, in order to minimise manufacturing costs, the cast-in air port guide vanes needed to be positioned and shaped accurately, and to have a good, smooth finish. The best way to meet these needs was to use centrifugal casting and this is what was done. The production techniques demanded a big effort from the Foundry personnel. They were equal to it and produced a good, consistent and reliable component.

The liner material posed a problem as the strength required for mechanical reasons conflicted with the need to have a material that would give good ring and bore life. A special form of cast iron, designated F.20, was developed by Fodens foundry and laboratory. This had a low combined carbon of 2.90%, silicon 2.0%, manganese 1.25%, nickel 1.25% and chromium 0.40%. This had an ultimate tensile stress of 18 tons per square inch and a Brinell hardness of 270. It proved to be very satisfactory.

The ports, of which there were eight in each liner, were cast in and were finished by sand blasting, the accuracy of casting being such that the critical, upper cut-off edge was held within 0.015" (0.38mm), as cast. After dressing with a hand grinder the ports were used for location purposes in subsequent machining operations. The bore of the liner was rough bored, finish bored and then honed.

In later engine development, in particular when piston speeds went up by 20% in the Mk3 engine, trouble was experienced with scuffing of the cylinder liners and rings when engines were being run-in (all Foden engines were run-in on the test beds before being checked for power, fuel consumption, etc and then being passed off). Following advice and help from Hepworth & Grandage, finished liners were phosphate treated to improve lubrication of the liner/piston interface. In the first place the liners were Bonderised and later the treatment was changed to Parco-lubrising. This eliminated the running-in problem. I personally took the first experimental batch of 50 finish-machined liners from Sandbach to Hepworth & Grandage at Bradford in an Austin A70 estate car. I went by way of Holmfirth and the weight of the liners in the back made it quite exciting with the brakes working very hard and the front wheels making only light contact with the road surface!

Rubber O-rings were used to seal the cylinder liner/crankcase joints apart from the joint between the top flange of the liner and the recess in the top of the crankcase on Mk1 engines. This joint was made using a jointing compound but, on all later Marks of engine, a rubber 0-ring was used in this location also. As the piston had to be long so that the bottom ring would prevent air leaking from the air chest into the crankcase when the piston was at top dead centre, the liner had to be long to retain this ring at bottom dead centre. This in turn necessitated slots in the skirt of the cylinder liner to give clearance for the articulation of the connecting rod.

The cylinder heads were of high duty cast iron and were made in Fodens iron foundry. The four-cylinder engine had one head covering all four cylinders, as did the five cylinder. The six, apart from the very first, which had a single six-pot head, had two non-interchangeable three-pot heads. The exhaust ports from each pair were separate right to the face of the head on the right hand side, to which the exhaust manifold was bolted. The manifold, which was of cast aluminium, kept the exhaust gases from each cylinder separated for a further distance so as to minimise the interference of exhaust gas pressure waves between different cylinders.

The design of the heads with a cross water flow so that all the water had to pass through a duct between the exhaust ports after coming into the head from the crankcase via a hole under the copper injection sleeve was a Foden patent.

The heads were cast in another Foden alloy cast iron, F19 and subsequently F21 and F24. All the cores for the internal passages were produced using resin bonded sea sand giving great accuracy of location. The value of this was increased when the importance of keeping the thickness of the heads between the valve seats to a minimum was realised. The tolerance on this thickness in the finished machine condition (water side as cast) was 0.010” (0.25mm). Great efforts were also made to remove any sand or scale from this critical area. The heads were cast flame plate uppermost with an exothermic sleeve to keep the metal molten long enough for any impurities to float to the top. This top face was given an extra thick machining allowance to ensure that all impurities were machined off.

As the camshaft was carried high up in the left hand side of the cylinder block, roller type cam followers were mounted in the cylinder heads; valve operation was by short push rods and forked rockers. This arrangement kept the inertia of the valve gear down, essential in view of the high operating speed.

The camshaft ran in split bronze bearings, the outside diameters of which increased in 0.010” steps, smallest at the back and largest at the front. During engine assembly the camshaft, with bronze bearings fitted and held in position with circlips, was inserted from the front so by stepping the bearings’ outside diameters, fitting was eased. A ball bearing supported the camshaft behind the drive gear and also provided end location. Return springs were fitted to the push rods. The tappet guides were phosphor bronze while the roller tappets were EN39 hardened on the outside diameter, the spherical seat for the push rod, the roller pin bore and the side faces of the roller slots.

The twin exhaust valves were of high-grade heat resisting steel running in wet valve guides. As the engine was developed to higher and higher power levels, valve material was also upgraded, finishing up in Nimonic 80A and Nimonic 90. All valve material except Nimonic 90 required a hard tip welded on the top of the stem to avoid wear on the stem tip.

The single hole, side mounted injectors were outside the valve covers making them easily accessible and reducing the risk of fuel dilution of the lubricating oil. The injectors were on the left hand side of the engine, close to the fuel injection pump and avoiding excessive heat pickup from the exhaust passages in the head.

The cylinder head gasket was originally a conventional copper-asbestos gasket but it proved to have a very short life, it was replaced by a gasket made from two sheets of copper, the upper sheet being folded over the lower sheet round the cylinder bores. This made a substantial improvement in gasket life but the component turned out to be Achilles heel of the Mk I engine and led to the Mk II version, which overcame the high incidence of gasket failures. Great efforts were made before taking this step, to get satisfactory gasket performance on Mk I engines, but the failure rate on automotive engines was never brought down to an acceptable level. Failures of the cylinder head gasket were, however, not a problem on engines in marine use. It is likely that current gasket technology would have found an answer to the problem but that was not so in the 1950s.

The crankshaft was a “soft” 3% nickel chrome steel forging with 3” (76.2mm) main bearing journals and 2.625” (66.7mm) crankpins, with a stroke of 120mm. This gave a generous overlap of 0.45” (11.4mm) between journals and pins, resulting in a stiff trouble free component. Timing gears were mounted on the back of the shaft and were hardened steel with ground profile. Experience on Gardner engines where chain stretch caused timing variations, convinced Fodens that gears should be used in preference to a chain drive and the desirability of minimising torsional vibration inputs to the drives of the various auxiliary components determined that the gears should be driven from the ear end of the crankshaft. This location made them rather inaccessible, so the gears and their bearings were designed to last the life of the engine. The camshaft to fuel pump drive gears were straight tooth but the main gear train gears were helical. The gears were an exact copy of the tooth form and pitch of the FP gearbox, which had been designed immediately prior to the engine, so saving on gear cutters etc. they were made of 8DP, 20deg PA with a face width of 1.25” (31.75mm) and a right hand spiral angle of 200 22’ 8”.

The fuel pump was driven by an extension shaft from its spur gear, and its timing could be set by adjustment of the pump couplings or the split gears in the timing case. Once set, the pump could be removed and replaced without disturbing the timing. The 49-tooth crankshaft gear drove a 65-tooth idler gear which in turn drove the 49-tooth camshaft drive gear, 24-tooth blower drive gear and the 52-tooth dynamo drive gear. The fuel injection pump being driven off the camshaft as mentioned above. By use of an auxiliary drive housing at the front end of the crankshaft to drive water pump, oil pump, and auxiliaries such as a power steering pump or a sea water pump and a shaft drive from the front end of the camshaft to drive the fan, the engine was free of belt drives. This was a welcome feature at a time when belt drives did not have the reliability or durability that they were to achieve in later years. The drive from the crankshaft to the transverse auxiliary drive shaft was by skew gears in the initial stage but they proved to be inadequate to cope with increasing levels of power take off and they were superseded by worm gears. These were capable of dealing with increasing power loads and operating speeds and gave good life through all later marks of engine.

The crankcase itself was made in DTD424 aluminium in dry sand moulds with many oil sand cores, which had to be accurately located. After fettling the castings, the crankcase guide vanes were carefully finished by hand and then the vanes were used for location purposes during machining. The long crankcase through bolts, which in effect bolt the cylinder heads to the main bearing caps, take most of the stress out of the crankcase casting. However, to minimise the effect of consequent distortion of the crankcase, wherever advisable, crankcase machining was done with dummy heads bolted on. This ensured the accuracy of such features as the main bearing parent bores. A steel tube to take oil to the valve gear was cast into the aluminium crankcase. The completed crankcase moulds contained over 40 cast iron chills.

The cooling system was conventional with a simple centrifugal pump driven off the right hand end of the auxiliary drive shaft through a spring loaded dog clutch which was include to prevent damage to the impeller in the event of the water pump becoming frozen. The dog clutch was later deleted as it was found that it would jump out of engagement if the engine over speeded for example when a vehicle was running downhill using the engine as a brake. Initially stiffer springs were tried but they seemed to make matters worse. The problem was traced to spring coils surging due to the crankshaft torsional vibrations at engine over-speed. The more common use of anti-freeze as time went by reducing the need for the clutch, so it was deleted. The pump delivered coolant through a short pipe into the water jacket section of the cylinder block at the front right-hand side and through O-ring sealed ports in the top face of the block onto the heads. Water leaving the heads was controlled by a bellows type thermostat. This directed the flow either to the radiator or to the water pump inlet if the coolant was below the desired operating temperature. Later marks of engine had a triple element wax capsule thermostat to give better control of the cooling and to mitigate against any danger of thermostat failure.

The lubrication system was typical of a well-designed system of that time. A gear-type oil pump was driven off a gear on the nose of the crankshaft via the auxiliary drive shaft, which also drove the water pump and, through the oil pump, a drive was available on the left hand side for other auxiliaries. The pump drew oil from the sump well and delivered it to a full flow oil filter mounted on the right had side of the engine. The felt element in this filter was spring loaded so that on the event of the filter becoming choked, the element lifted off its seat and allowed unfiltered oil to pass through into the system. From the filter it went to the main gallery pipe which supplied the main bearings, drilled passages in the crankshaft carried oil from the main bearings to the crankpins from which it was fed to the big end bearings, and thence through the connecting rods to the small ends and the piston cooling jets.

From the rear of the main gallery an internal pipe took oil to the rear camshaft bearing and thence via the hollow camshaft to cross drillings to all the other camshaft bearings. The front camshaft bearings in turn gave an intermittent feed to the rocker shafts, which provided feeds to rockers and pushrods. Surplus oil draining down from the rocker boxes lubricated the cam followers before returning to the cam tunnel. Oil built up in the cam tunnel to a level determined by a tapping for an oil supply to the blower and a passage at the rear camshaft bearing which fed the timing gears at the rear. A supply was also taken from the front of the cam tunnel to feed the fan drive shaft. The oil reaching the front (free) end of the blower from the front part of the camshaft tunnel filled a reservoir in the blower forward end cover, from which an oil thrower picked up oil to lubricate the blower front bearings, the level in the reservoir being set by a longitudinal passage in the blower casing communicating with another reservoir at the rear (drive) end of the blower. The level in this second reservoir was maintained by a weir in the blower end drive cover so arranged that the blower rotor gears picked up oil to lubricate themselves and the blower drive end bearings, finally spilling into the timing gear case.

A spring loaded oil pressure relief valve in the oil filter cover spilled oil back to the sump when the oil pressure exceeded the valve opening pressure. This was to limit the oil pressure when the oil was cold or when the engine speed was high. Early automotive engines had an air-cooled oil cooler mounted in front of the radiator, but on later marks a water-cooled oil cooler mounted between the water pump outlet and the inlet to the cylinder block (replacing the water transfer pipe) was used. This gave a very neat and effective installation with the jacket water warming up the oil when the latter was cold so giving the engine a more rapid warm up.

Another factor which made a big contribution to the success of the engine was the availability of good detergent lubricating oils, such as Shell Rotella or Essolube HD, or any oil that met US specification MIL-O-2104. The improvements in lubricating oils which came in during the late 1940s and the 1950s led significant improvements in engine cleanliness, longer overhaul periods and reduced wear. The modern highly rated diesel engines would not have been possible without the continuing strides made by the oil companies in providing better lubricants.

Mention has been made of the shaft driven fan, driven off the front end of the camshaft. Another aim of the Foden designers had been to eliminate belt drives from the engine, as belt drives were by no means as reliable or as capable of transmitting high powers as they are nowadays. This target was achieved using the shaft drive for the fan, a dynamo driven by a gear in the timing case through a Metalastic coupling and the provision of the auxiliary drive housing at the front of the engine mentioned earlier. In later years when bigger, faster running fans became necessary as the engine developed more power, the camshaft drive had to be abandoned and replaced by a belt drive but, by that time, belt drives had improved greatly.

Chapter five

## The 85mm bore engines

Although the original intention had been to build the engine in a five-cylinder form to compete with the 102 bhp Gardner 6LW engine, only three FD5 engines were built. Initial road tests of the FD5 gave more favourable fuel consumption figures than had been expected and news of a potential upgrade of the 6LW to 112bhp changed Fodens plans.

The engine went into production in January 1950 in the six-cylinder 126bhp form, closely followed by the 84bhp FD4, the engine had of course generated lots of interest in the road haulage world and despite the innate conservatism of the industry, two-stroke powered Fodens were soon evident on the roads. An initial attraction was the low weight of the installed engine, which gave an immediate increase in payload capability of about 4cwt (200kg). As vehicles went into service their road performance became attractive to some operators. It was not just the power level, 126bhp, but the liveliness of the engine which, in the hands of an enthusiastic driver, shortened journey times particularly those on hilly routes. This gave increased earning capacity over and above that which came from the lower unladen weight of the two-stroke vehicle. Owner drivers, particularly those who were prepared to turn a blind eye to things like speed limits and drivers’ hours were able to offer their customers big improvements in service. The classic was the “Flying Scotsman” who regularly took his two-stroke eight-wheeler between Glasgow and London overnight in the days of the 20mph limit, no motorways and no tachographs.

Fuel consumption was a different matter, whilst a skilled and enthusiastic driver could turn in a fuel consumption comparable with contemporary four stroke vehicles, a driver who did not have these attributes would produce fuel consumption figures that were disappointing to the man who paid the bills. The availability of a multi-speed gearbox gave the skilled and keen driver a means of keeping the engine working at or near its best point and the Foden twelve speed gearbox, introduced in 1951/52 was ideal, but it did need a good driver.

Generally, especially for a new, highly rated engine, reliability was satisfactory apart from cylinder head gasket failures. Despite all efforts the problem could not be brought down to an acceptable level on automotive engines. The patience of operators was tested as thoroughly as alternative gasket materials and designs. It was eventually decided to introduce major modifications to cure the problem once and for all. That became the Mark II engine.

The change made was to widen the flange at the top of the cylinder liner so that the liner flange could be fitted with six 3/8” UNF studs, thus making it possible to clamp the liners direct to the cylinder heads. A ring gasket of copper, fitted over the studs made a seal which virtually eliminated gasket trouble

For the rest of the production life of the engines the multi bore heads were replaced by unit heads (one to each bore) which otherwise were similar in function and design to the Mk I heads. The cylinder head/liner assemblies were retained in the cylinder block by through bolts as in the Mk I but special balance washers were used to bridge the gaps between individual cylinder heads and between the end heads and special end blocks or spacers. Much of the valve gear remained unchanged but rocker boxes and covers, rocker shafts, oil supply components etc had to be changed to suit the unit head construction. Thus was the cylinder head gasket problem cured although it was not until 1956, six years after the engine was introduced, that the Mk II was put into production and marine engines for the admiralty at least remained as Mk Is for the time being. Over the following years alternative head gasket materials were tried from time to time (in Mk II form) including some in soft iron, but copper rings remained supremely reliable. Water and oil drain connections between the cylinder heads and the block were sealed by O-rings as in the Mark I. Any occasional gasket trouble could be traced to some quality aspect, but!!!

As I have said, the Mk II design got rid of the major Mk I service problem but it introduced another, more serious problem – cylinder head cracking. This will be dealt with in a later chapter but it proved to be quite intractable and remained with the automotive engine to the end of its days. Once more the marine engines, when the unit head was adopted, were more or less immune. Any cracked heads on marine engines could usually be traced to a cooling system failure such as seawater cocks left closed. However, on automotive engines it was never reduced to an acceptable level. The cracks ran across the narrowest part of the land between the two exhaust valve seats and were caused by thermal fatigue. As it was a major problem that defied our very considerable efforts, I will deal with it later, but once again, the long suffering operators put up with the trouble for a long time and the problems remained with us on Marks two, three, four, six and seven engines. You may be wondering what happened to Mk V. That will also be revealed later, back to MkII.

At the time that the Mk II was being introduced, a need could be seen for more power. After considering various options it was decided to get the increase in power by lifting the rated full power speed of the engine from 2000 rpm to 2400 rpm. The target was 150 bhp and to get this it was necessary to produce practically the same bmep at 2400 rpm as the Mk I and Mk II had been developing at 2000 rpm. It was obvious that engine breathing would require attention to minimise the power required to drive the Roots blower. There was also a speed increase for the valve gear and for the fuel injection pump.

To make the engine breathe more easily the air port timing was altered slightly from that of the earlier engines, opening and closing 50 degrees either side of BDC instead of 49 degrees. The changes to the exhaust valve timing were greater and necessitated a new camshaft with the same lift but greater dwell. Exhaust valve opening, as defined earlier, was 98 degrees after TDC and closure was 49 degrees after BDC. There were no other changes to the combustion dynamics of the engine and the power increase was obtained by the alterations in engine speed and timing, and by injecting more fuel. Fuel consumption was slightly worse than earlier engines but the weight advantage compared with the 150 bhp Gardner 6LX was 6 cwt (305 kg) and 11 cwt (559 kg) compared with the 200 bhp Cummins NH engine which was now available.

Because of the increase in rotational speed it was necessary to stiffen up the crankshaft to move the critical torsional vibration speed away from the new rated speed of the engine. Main journal size was increased from 3.0” (76.2 mm) to 3.208” (81.5 mm) and the crank pins from 2.625” (66.7 mm) to 2.832” (72.0 mm), going metric at the same time. By going from thick walled main and big end bearing shells to thin walled shells the crankcase and connecting rods required no major alteration. The crankshaft changes gave a journal/crankpin overlap of 0.66” (16.75 mm) and the new shaft could safely run at 2400 rpm. The main and big end bearing shells were lined with copper-lead, lead flashed. Later, both big end and main bearing shells were changed to aluminium-tin, an excellent bearing material supplied by Glacier Metal. It was kinder to the soft crankshaft than was copper-lead, and prompted the comment from one operator’s maintenance foreman that the Foden engine had the best bottom end in the business. Also the labyrinth type of seal at the rear end of the crankshaft on Mk I and Mk II engines was replaced by a piston ring type of seal which worked a lot better. At the same time the rear diaphragm of the crankcase in particular and the bottom of the crankcase generally was stiffened up.

The increased rotational speed prompted the introduction of the CAV “NN” injection pump to replace the earlier “N” type. On the later pump the cambox was lubricated with engine oil taken from a suitable tapping in the lubrication system with a drain back to the sump via the Roots blower free end casing. To avoid fuel dilution of the lubricating oil, (from leakage past the plungers), a scavenge pump incorporated in the hydraulic governor pumped fuel that had accumulated in the pump tappet chamber into the return fuel line back to the fuel tank.

One problem that arose during the development of the Mk III engine, and which delayed its introduction, was erosion of the water pump casing. This was caused by cavitation in the pump, which in turn was caused by the increase in rotational speed. The problem was compounded by a change in the gear ratio between the crankshaft and the auxiliary drive shaft which had been speeded up from 1:1 to 1.262:1 on the FD6 only so that the water pump and oil pump would have adequate capacity for the power uprate.

During dismantling of an engine after development testing, traces of erosion were seen in the aluminium water pump casing and water pump cover. Examination of the water pump on another MK III development engine showed that this was not an isolated case. Erosion had never been seen before on Foden engines and the cause, cavitation, was immediately recognised.

Finding a cure was not so quick. A period of intense water pump development work ensued. A rig was built on which a water pump could be driven by a variable speed electric motor and a close approximation to a vehicle cooling system was included in the rig together with all the relevant instrumentation. We found that we could reproduce the symptom in a relatively short time and we were quickly into designing, making, testing and discarding new water pump impellers, casings and covers in large numbers. This was one way in which the flexibility of a small company with its own design team, pattern shop, foundry. machine shop and experimental department paid dividends. Pumps were designed, made and tested within a week. No solution to the problem had been found and the threat of working right though Christmas was real. Fortunately the design team had the bright idea of buying a book on rotary pump design and the solution followed very rapidly after that. An impeller, better capable of developing more pressure combined with a casing and cover in which the water passage areas were more carefully blended gave us a pump with no cavitational problems and which needed no further alteration during subsequent years. The opportunity was taken to incorporate a better shaft sealing arrangement at the same time. The attachment of the bronze water pump impeller to the stainless steel shaft was not without its problems. Use of splines was not satisfactory as the impellers came loose on the splines. A cure was made by screwing the impeller onto the shaft on a left hand thread, and securing it with a nut on a concentric right hand thread.

As mentioned earlier, the Mk III also saw the introduction of a water cooled engine oil cooler which eliminated the need for a separate radiator to cool the oil in vehicle installations. The oil cooler fitted very neatly between the water pump outlet connection and the inlet elbow on the side of the crankcase. Oil piping from the oil pump to the cooler and from the cooler to the oil filter was much simpler than with an air-cooled cooler in front of the radiator. So this was a very good change giving a technically better installation and a much lower installed cost. Reliability was improved by the elimination of the oil radiator and its vulnerable pipe work and a more rapid warm up of the engine oil was obtained as well as better control of oil temperature.

Opportunity was taken to replace the bellows thermostat with a much more reliable wax element type as mentioned previously. The one used incorporated three wax elements, one of which opened slightly before the other, thus giving better control over engine temperature under cold low load conditions. The system was arranged to give a continuous flow down the by-pass pipe to the water pump inlet which again gave better control of engine water jacket temperature and more uniform operating conditions in the water pump itself, the increased flow of water round the engine caused pressure in the system and higher quality hoses and clipping were introduced.

At about that time, a fresh look was taken at turbocharging. I say a “fresh look” because some work had been done a few years earlier using a Napier turbocharger which, although tried on an FD6, was really of a size more suited to an FD12. The results were naturally poor as both compressor and turbine were working a long way from their high efficiency points However, in the late fifties new small turbochargers with radial as opposed to axial flow turbines were coming available from BSA and from Holset, the latter offering the American Switzer turbocharger built under licence at that time. Work started again on the six-cylinder engine on what was to become the Mk IV, a turbocharged Mk III.

For automotive use it was found that the BSA turbocharger (the development of which was later to be taken over by CAV Ltd) gave the best performance. Although in theory it would be possible to rely solely on the turbocharger to provide the engine with air, in practice it was necessary to retain the Roots blower to give an adequate air supply when starting the engine and during rapid changes of engine speed and load, vital in an automotive engine. The gas flow was from turbocharger compressor to intercooler, to Roots blower, to air chest, to cylinders, to turbocharger turbine, to exhaust system. As the Roots blower was being fed with compressed air from the turbocharger its swept volume could be reduced. After testing a few different sizes a capacity of 78% of the normal Mk III engine blower size was selected as most promising. The blower size was reduced by shortening the lengths of the rotors and fitting packing blocks in the blower casing to take up the gap. In production a shorter blower would be made. At higher powers it was found that this reduction of blower capacity was insufficient to keep the air chest pressure (and consequently the compression pressure and maximum cylinder pressure) within bounds. This was overcome by a modification to the blower casing so as to incorporate a simple spring-loaded piston valve, known as the recirculation valve, to pass air back from the delivery side of the blower to the inlet side. This worked very well once we had changed the piston material from aluminium to cast iron and eliminated the tendency to “pick up” in the bore in the aluminium blower casing. It was also found necessary, now that the blower was pressure fed by the turbocharger, to turn the oil seals round to face the other way so that the increased pressure in the blower casing did not lift the lips of the seals off the shafts, also as the exhaust temperature was now considerably higher than on non-turbocharged engines, it was necessary to change the exhaust manifold material from aluminium to cast iron.

The increase in air chest pressure from about 7 psi at full power on blower scavenged engines to something of the order of 15 psi at full power on turbocharged engines increased considerably the quantity of air leaking into the crankcase and raised the crankcase pressure. This made the oil reluctant to drain back down from the fuel pump cambox to the Roots blower and a lot of testing was necessary to develop a crankcase breather which would relieve the crankcase pressure without oil carry over. The breather was threatening to be the same size as the engine air filter till technology came to the rescue.

As the engine was now working with quite a high level of boost, particularly at that time, it was decided to adopt intercooling from the beginning. Vehicle applications used an air to air cooler of brazed aluminium construction mounted in front of the water radiator. The method of intercooler construction resulted in a very strong, trouble free unit, which was adopted by other vehicle manufacturers many years later. The connecting pipes delivering air to and from the intercooler were large and of complex shape and were carrying turbocharger delivery pressure air which, in the upstream side of the intercooler was quite hot. Aluminium would have been a good material but the quantities required could not justify the tooling cost. Instead Fodens turned to the expertise they had built up in making cabs from glass-reinforced plastic, to make the pipes from the same material. It was advisable to incorporate a steel-reinforcing ring in the ends of the pipes and to mould a lip on the end to minimise the risk of a hose connection blowing off. Whilst the pipes did not look very elegant, they worked quite well and overcame the cost problem but it was necessary to keep a close control of manufacturing quality. Marine engines were to use a seawater cooled intercooler, which made a very neat installation. Both these arrangements brought the charge air temperature down quite low, thus feeding the roots blower with dense air and, in this way, were considerably better than intercoolers using engine jacket water as the cooling medium.

The performance of the Mk IV engines obtained after a lot of hard work at Fodens, BSA and CAV was a revelation. As compared with the Mk III engines the automotive rating had gone up from 150 bhp at 2400 rpm to 210 bhp at 2200 rpm, whilst maximum torque had gone up from 365 lb.ft at 1500 rpm to 500 lb.ft at 1400 rpm. Specific fuel consumption had improved, a best point on the torque curve of about 0.420 lb/bhp/hr to 0.380 lb/bhp/hr. On the road, as tested in an 8x4 vehicle at a gross weight of 24 imperial tons (24392 kg), the improvement was even more impressive. The improved torque showed up as a big improvement in “luggability”, and good road fuel consumptions compared with the best four-strokes (now the Gardner 6LX), could be obtained by much less skilful drivers because the engine was no longer so sensitive to driver skill. The higher power available, coinciding with the opening of the M1 motorway with no speed limits or lane restrictions on heavy goods vehicles at that time, led to a test programme investigating potential earning capacity, fuel consumption and reduced journey times. The test vehicle rear axle ratio was geared up and up until motorway cruising speeds of over 80 mph (130 kph) were reached. It must be admitted that suspensions and brakes were not, at that time, suitable for such speeds, nevertheless the tests showed what benefits could be obtained for operators by very short journey times and satisfactory fuel consumptions.

However, this sort of performance could not have been obtained without operating problems. As mentioned earlier it was necessary to take steps to limit the air chest pressure (or boost pressure) at high speeds and loads as maximum cylinder pressures were of the order of 2400 psi. The introduction of the recirculation valve in the roots blower provided a simple means of doing this. Better results could be obtained by using both a recirculation valve and a turbine by-pass valve or waste gate, but it was decided that it would be too complicated. Also at that time, the technology to provide reliable non-sticking waste gate valves was in its infancy. Because of the high cylinder pressures the release pressure of the fuel injectors had to be increased from 175 atmospheres to 225 to prevent the nozzle needles being blown open by the gas pressure. This method of eliminating blowback was quite satisfactory and introduced no problems. Tests in which the nozzle needle lift was measured and timed showed that it was necessary to complete fuel injection before the pistons reached TDC, if a steady state dirty exhaust was to be avoided. This led to the use of 7.5 mm diameter plungers in the injection pump on turbocharged engines, as opposed to 7.0 mm plungers in blower scavenged engines.

Problems were experienced with the connecting rod small end bush. The bush used in the Mk III engine, which was solid bronze had a life of between 100 and 150 hours. In the Mk IV development engines the mode of failure was either hammering into the piston cooling jet oil feed groove or loosening in the connecting rod small end eye. It was replaced by a steel backed copper lead bush which required a rod with a larger small end eye, to take the larger outside diameter of the steel backed bush. This modification eliminated the problem.

No changes were needed to the piston design despite the increased engine rating. Many tests were done measuring piston temperatures. Two methods were used; fusible sentinels and templugs. Fusible sentinels are small pellets of metals of different melting points, in a way different sorts of solders, which are hammered into small holes drilled into the component whose temperature is to be determined. By putting groups of pellets of different melting points in a small area of the part and running the engine for a short period at a specified speed and load, usually full power, the temperature reached by the component can be estimated by seeing which of the sentinels have melted and which are still in place. Alternatively, if the component is small, or the critical area of the component is small, a series of tests can be done using a few identical sentinels in each test, but varying the melting points between tests. Templugs are small screws made of special steel, which are screwed into threaded holes made in the part concerned. After the test the component has to be carefully sectioned through the middle of the templug. The temperature reached can then be ascertained from measurements of the hardness of the templug.

From tests such as these, piston crown temperatures approaching 600 degrees C on turbocharged engines, which was about 100 degrees C over temperatures measured on blower scavenged engines. Those temperatures were those reached at full power and even with cast iron pistons were of some concern. However, full power tests of engines showed no piston burning and no change was made to the piston design.

One further modification necessary was the introduction of a variable position maximum fuel stop in the fuel pump governor, the position of the fuel stop being controlled by the air pressure available in the air chest This was necessary to avoid excessive smoke during engine acceleration from low load/low speed operation. This device was known as the boost torque control.

The MK IV engine never went into production in any quantity, only four being sold. One of which went to Loughborough University for research work and for student instruction. That was not because the company had doubts about the engine, far from it, but because other development work in progress at the same time was pointing towards an increase in swept volume; the Mk VI and Mk VII engines This work was being done on the FD6 Mk V engine, only one of which was built. The performance of the Mk V was sufficiently good to persuade the company to go to a bigger engine; 4.8 litres in a six cylinder as compared with 4.1 litres for all the earlier FD VI engines. These new engines would incorporate the improvements made in the Mk V.

The Mk V had its beginnings in a discussion between Martin Britain and myself, we both had a fair amount of four stroke diesel engine experience and, talking about how we could improve the Mk III we agreed that it offended our instincts to open the exhaust valve when the piston was only little more than half way down the cylinder. This made us consider a significant change in the air port and exhaust timing of a six-cylinder development engine. Liners incorporating air ports of a lower height were made and fitted in conjunction with a short dwell Mk II camshaft. This gave us a longer timing stroke and a substantial improvement in performance. Minor adjustments made to exhaust valve timing by widening tappet clearances brought no further significant changes in engine performance. Other minor changes to the engine to vary the timing included shimming the liners down by fitting two additional cylinder head gaskets and swinging the camshaft. Special heads with larger exhaust valves which gave an 8% increase in area for the same lift, and liners with 16 air ports instead of the standard 8 ports were also tried. While the larger exhaust valves gave an appreciable improvement in engine performance, the head cracking problem discouraged the company from making a change in head design which was likely to accentuate the failure rate. The 16 port liners gave a poor performance but it is possible that a change from 8 port to 10 port might have effected an improvement, similarly shimming the liners down was followed by a deterioration in performance. Timing changes gave us the choice of bigger improvements in mid speed performance or in high-speed performance.

The other major change in the Mk V was a change in the roots blower size. Development work showed that two alternatives were available to us – higher torque and power but at the expense of higher fuel consumption with a standard (100%) roots blower. The loss of fuel efficiency was due to the change in engine timing making engine breathing more difficult and to putting up the air chest pressure and consequently the power required to drive the blower. The other alternative was to reduce the amount of air going through the engine by fitting a blower of 10% less swept volume (90% blower). This gave us no increase in power over the Mk III albeit developed at 2200 rpm instead of 2400 rpm but a substantial improvement in fuel consumption and a very clean exhaust. The cleanliness of the exhaust was such that at full power the exhaust was virtually invisible. Both ratings were tempting and led to the idea of getting both at the same time (having our cake and eating it!) by incorporating the new timing with a 100% blower in an engine of increased swept volume; the 92 mm bore Mk VI engine.

No tests were done on a turbocharged Mk V because of the intended change in swept volume, but the fuel consumptions obtained in the one Mk V engine built were the best I ever saw on any blower scavenged Foden engine of 85 mm or 92 mm bore. However, it does not necessarily mean that production engines would have been as good, for as anyone who has been involved with engine manufacture knows; every now and again you get a really good engine and every now and again you get a bad one and maybe we were lucky with our one Mk V. No matter, it was a good engine and paved the way to the range of 92 mm bore engines, which were the best production engines that Fodens ever made.

Chapter six

# The 92 mm bore engines

As stated in the last chapter, the promise of the Mk V, 85 mm bore engine and the need for more power led to an interest in a “bigger” engine. The demand for more power was stimulated by such things as no increases in legal speeds, increases in permitted gross weights, the opening of motorways which, with high power high speed vehicles, gave promise of faster longer journeys and the ever increasing pressures on hauliers’ margins creating a need to get more out of each driver day and each vehicle day.

Experience over the years with 85mm bore engines had indicated that liner bore could be increased without seriously affecting the effectiveness of the air ports in creating swirl. Increasing the “size” of the engine without increasing the external dimensions was a very satisfactory way of getting the increase in power; engine installation would be virtually unchanged. Changing the bore meant only changing the pistons, rings and liners although in fact opportunity was taken to increase the gudgeon pin size to improve the life of the small end, and cylinder head gaskets had to change to suit the new liner. To increase the stroke would have created a need for many new parts probably including a crankcase. A change in bore was the obvious choice.

On the MkV it had been found that reducing the height of the air inlet ports and altering the timing of the exhaust valves, the engine could be run with a 10% reduction in roots blower swept volume and still produce the required power at a good fuel consumption and with a clean exhaust. Logically it was important to use the same roots blower as on the Marks I, II and III with the blower-scavenged engine of increased swept volume. It was therefore logical that an increase of at least 10% in swept volume could be matched with the existing blower and, for a small sacrifice in exhaust cleanliness and possibly fuel consumption; a bigger change in swept volume could be supported by the blower. The increase in bore from 85mm to 92mm gave an increase of 17% in swept volume – quite an optimistic step. However, this would give power output from the blower-scavenged engine of 175bhp at 2250 rpm if we could achieve a mean effective pressure similar to earlier engines. On a turbocharged engine, based on the rating of the Mk IV, 250bhp was within reach. In actual practice the Mk VII was rated at 225bhp, which was a power rating suitable for the market at that time. This represented an increase of 7% on the Mk IV and 28% on the blower scavenged Mk VI.

As the piston design was being changed, the ring pack was simplified, taking advantage of improvements in piston ring technology. The two-piece fire ring was retained, this being one of the key components of the engine. Below the fire ring were two single taper-sided, taper-faced compression rings and one parallel-sided, taper-faced air seal ring at the bottom of the piston skirt. The use of single taper-sided compression rings reduced cost and improved quality. By having the bottom face of the ring and groove parallel (i.e. at right angles to the piston axis) it was easier to achieve flatness of both machined surfaces. As, on two stroke engines, the ring is always sitting on the bottom face of the groove, good seating could be achieved while the top taper side ensured freedom from ring sticking. The revised design of the air seal ring enabled it to act as an effective oil scraper. The gudgeon pin diameter was increased to 1.875” (47.6mm) to give the small end bush and the piston bosses more bearing area to compensate for the increase in piston area.

The cylinder liner incorporated ten ports as opposed to eight on earlier marks of engine and the air port height was reduced to .550” (14mm) as compared with 0.680” (17.3mm) on the Mk I, 0.730” (18.5mm) on the Mk III and 0.560 (14.2mm) on the Mk V. The crankcase bores had to be increased in diameter to take the larger Mk VI liner and in so doing, the air chest guide vanes were too small to be effective so they were deleted from the casting.

The crankshaft was interchangeable with the Mk III. The con rods had a larger small end eye to suit the larger gudgeon pin and the big end caps were secured by four set screws into tapped holes in the con rod as opposed to two nuts and bolts used on earlier engines.

The cylinder heads and valve gear were virtually unchanged apart from modification to take long stem injectors. The Mk1 camshaft was used and not the long dwell Mk3 shaft. Timing gears were interchangeable with Mk3 but were soon changed from helical to straight spur, partly as a cost saving measure and partly to offer a high power, power take off from the rear end of the camshaft drive. On vehicle engines a high-speed dynamo drive was taken off the rear of the blower drive gear but on other applications the earlier separate dynamo drive below the blower was used.

As mentioned before, the blower was unchanged. Other modifications included a new crankcase breather, oil filter and a common shorter fuel pump driveshaft for 4 and 6 cylinder engines. An updated CAV “NN” injection pump was used, still pressure lubricated with engine oil. A new fan and drive using a four belt drive off a pulley on the front of the crankshaft and the auxiliary drive housings on both four and six cylinder engines were commonised with the 1.26:1 drive gear ratio. Unified threads were introduced.wherever practicable,

The engine timing was finalised as follows:

Exhaust valve opens 101deg after TDC

Inlet port opens 136deg after TDC

Exhaust valve closes 42deg after BDC

Inlet port closes 44deg after BDC

Fuel injection at 29deg to 31deg before TDC

The Mk6 engine was introduced at a rating of 175bhp at 2200rpm, though when the Gardner 6LXB engine came onto the market with a rating of 180bhp at 1700rpm, the engine was up rated to 180bhp at 2250rpm and called a Mk VIB, peak torque was 424 lb.ft at 1600rpm. Field trials of a Mk VI in a four wheeled tractor running at 24 tons GCW with a two axled semi trailer gave a fuel consumption of over 10mpg and a lubricating oil consumption of 5000mpg. The driver reported the performance as outstanding while the fuel consumption above was comparable with the best four strokes in the fleet.

The performance of the engine was certainly a significant improvement on what had gone before and giving obviously more power, but also a better torque curve and better fuel consumption. However, despite all the intensive development work on cylinder head cracking, of which more later, this problem still troubled us. That apart, the engine was very reliable.

The development of the Mk VII engine followed two separate paths; one for automotive engines and one for marine and industrial applications. The power requirements, or more accurately the torque requirements, for the duties were quite different. For automotive use high torque at low speed is important implying high boost from the turbocharger at lower engine speeds. Marine and some industrial engines only require power and torque at the rated speeds or thereabouts.

Looking at these requirements in more detail, the performance needs of the Mk VII could be listed as follows:

1 Required horsepower at rated speed without excessive boost pressure.

2 A good torque curve with maximum torque as low down the speed range as practicable

3 Good fuel consumption throughout the operating speed range

4 Low smoke levels throughout the operating range including during engine acceleration.

5 Low noise level

With marine and industrial engines including generator sets, the requirements would be:

1 Satisfactory power at continuous rated speeds

2 Possibly a combat rating at a higher speed or an overload capacity at the rated speed.

3 Low fuel consumption at rated speed and load

4 Reasonably clean exhaust at rated speed and load

In the case of the automotive engine, the requirement called for a small low inertia turbocharger, which would give good boost at low engine speeds and would accelerate rapidly when required to do so by demand for more power, so providing the increased air needed quickly. The need to limit boost pressure at full power required an additional control of airflow to the engine, the blower mounted recirculation valve used on the Mk VI engine was used. Also, to help reduce smoke during rapid engine acceleration (as occurring during downward gear changes) a “bigger” roots blower was required.

On marine engines, lack of a low speed torque requirement and the slightly lower emphasis on a clean exhaust during engine acceleration enabled us to use a larger turbocharger, which gave the required boost at the rated power and speed without the need for additional control. Also, a “smaller” roots blower could be used giving a reduction in blower output and a consequent improvement in fuel consumption.

The experience on the Mk6 was directly applicable to the Mk VII automotive engine and the use of a 78% roots blower and a CAV type 12 turbocharger proved to be quite a good combination. As before, an air-to-air intercooler was mounted in front of the vehicle radiator, a system not adopted by other manufacturers for many years. Fuel injection was by CAV “NN” pump with hydraulic governor. The pump was fitted with 7.5mm diameter plungers and injectors were still single 0.5mm hole long stem nozzles set at an opening pressure of 225atms, about 3375psi

The governor included the boost torque control as used on the Mk IV engine in which the position of the maximum fuel stop was varied according to the air chest pressure available. This limited the smoke during rapid engine acceleration and also at speeds below the torque peak where the boost pressure generated by the turbocharger and the roots blower were insufficient to provide enough air for clean combustion at full load.

The engine performance obtained was very satisfactory. The rated power of 225bhp at 2200rpm was achieved quite comfortably with a maximum torque of 600lb.ft at 1400rpm. This gave a very good over the road performance with a strong luggability, rather better than we had anticipated. Fuel consumptions were a vast improvement on previous engines, and for the first time we got below 0.36 lb/bhp.hr, which was quite a respectable fuel consumption at that time (although not as good as a Gardner). Furthermore the specific fuel consumption remained good over a wide range of loads at any fixed engine speed, these two factors produced excellent road fuel consumptions. A 24 ton 8 wheeler tested by the technical press on the M6 recorded a fuel consumption of 9mpg at an average speed of 57.4mph over a run of 87miles. The engine was as smooth as ever and, primarily because of the effect of the turbocharger on the roots blower intake noise and exhaust noise, was quieter.

The main part of the engine was virtually unchanged for the Mk VII version. The crankcase was the same as the Mk VI though it was round about this time that we started to anodise the crankcases to minimise the risks of corrosion from using unsatisfactory types of anti freeze. The crankshaft was unchanged as were the aluminium-tin bearings. Apart from the dimensional changes and the reduction in the number of piston rings, the pistons were very similar to those used in earlier marks of engine. However, exhaust valves were running a lot hotter and KE965 was no longer good enough to stop the occasional valve head coming off. In that event, the head usually came out through the turbocharger, and in so doing modified the shape of the inward flow turbine wheel to the detriment of the engine performance, normally it was not necessary to change a piston. Various alternative higher-grade valve materials were tried during the development of the Mk VII until we found a satisfactory solution with a Nimonic 80A valve. Some further development was needed by the valve manufacturer to get a satisfactory weld of the stem to the head but it was also sorted out in good time.

It was of course necessary to change to a cast iron manifold and, this being on the right side of the engine, kept the drivers’ legs warm when the engine was working hard. Unfortunately it did that in the summer too so it was necessary to develop a suitable heat shield to protect the drivers’ legs and the glass-fibre bonnet side.

The other minor component that gave us a lot of trouble was the crankcase breather. The high air chest pressure now being generated greatly increased the airflow through the crankcase from blower seal leaks, leakage past the air seal rings and from filling and emptying the hollow gudgeon pins with high pressure air as the pistons went up and down the cylinders. It proved quite difficult to find a breather that would pass this amount of air without passing oil as well and with satisfactorily low crankcase pressure.

As mentioned earlier, when we moved to 92mm bore, we took advantage of the need for a new piston design to increase he diameter of the small end gudgeon pin, so as to increase bearing area on that critical component. At full power on a Mk VII the maximum load on the small end was about 10 tons. Tests of wrapped bushes in the small end of the rod were unsuccessful and we had to use an expensive, steel backed, lead-bronze bush. This proved to be quite successful, however we were looking to increase the power to 250bhp or thereabouts so we thought that at that rating we may get back into trouble with the small end or piston crown temperature, or both. We started to look more closely at oil flow to the small end and rig tests of a motorised engine showed that the rate of oil flow from the piston cooling jet fell away as the speed rose above about 1500rpm. This showed that the acceleration of the rod on the upward stroke was causing reversal of the oil flow in the rod, the answer was to fit a non-return valve in the big end and this took the form of a spring-loaded ball. We tested these on an engine running on durability trials in the factory powerhouse. This engine was running at continual high load at 2250rpm and the piston life was significantly increased. The modification never went into production as the up rate was never made but one interesting aspect was that the only material that gave satisfactory ball life was glass!

Incidentally, a lot of durability testing was done in the factory powerhouse where we had up to six engines running. Most of these ran at 1500rpm so as to give 50 cycles per second AC power, but with the aid of a 12-speed box we were able to run one engine at 2250 rpm. The 1500rpm engines ran at 20% overload continuously night and day. It was a very effective way of doing durability testing as the value of the electrical power generated offset at least part of the cost of running the engines.

Development did not stop when the engine was put into production, we were continually looking for improvements in fuel consumption, torque especially at low speeds and higher power. One of the key components affecting these aspects is the turbocharger. Although we had gone into production with the CAV type 12 turbocharger, as standard fitment on the automotive engine, we did a lot of testing with Holset turbochargers. At first they did not match the requirements of automotive rating as well as the CAV type 12, but later models such as the 4LE gave engine performances closely matching that of the standard arrangement. We also did quite a bit of work with a Swedish turbocharger made by Flygmotor. Flygmotor was a company, which manufactured gas turbines such as the Rolls Royce Avon and the Pratt & Whitney JT8D-22 under licence for use in Swedish (SAAB) military aircraft. They decided to use the experience they had gained to produce a turbocharger, no doubt with the pioneers of turbocharged commercial vehicle engines, Volvo and Scania Vabis in mind. The Flygmotor turbo was unique in that the bearings were mounted outboard of the turbine and compressor wheels, when others had the bearings between the two wheels. An FD6 Mk VII was sent to Sweden so that Flygmotor could carry out tests in their own development department in parallel with work going on at Sandbach. However, some months later, Flygmotor abandoned the project.

As on the Mk VI, the automotive intercooler was an air-to-air matrix mounted on the front of the vehicle radiator and was connected to the engine by glass fibre reinforced plastic pipes. The brazed aluminium matrix was very strong and light. The glass fibre pipes were relatively cheap but were working near their limit of high temperature strength (in the case of the pipe joining the compressor delivery connection to the intercooler inlet), and it was necessary to take great care with quality of manufacture.

The engine had a mixed reception in the market place. Some operators treated it with a great deal of caution, others particularly those who valued the opportunities to complete journeys more rapidly, (possibly with no questions asked) welcomed the step up in performance. However, we met an unexpected problem when one or two operators decided that they knew better than we did what the engine output should be. They followed their age-old practice of opening up the maximum fuel stop on the governor until the onset of smoke told them to stop. While this worked reasonably well on non-turbocharged engines (after a mild uprate, the exhaust would become dirty), it did not do so on turbo charged engines as the turbo charger speeded up and delivered more air. So the fuelling went up and up and up until the engines started to run into serious thermal and mechanical problems, for which we got blamed!

The marine Mk VII, of which more later, used a Holset model 4 turbocharger and a seawater cooled intercooler. The roots blower had a swept volume of 66% of that of a non-turbocharged engine as compared with 78% for an automotive version. The seawater-cooled intercooler gave a very neat installation and worked very well. Obviously it had to be made of materials, which would not be adversely affected by seawater, but this was easily within the technology available at the time.

Chapter 7

Other Foden diesel engines

There were other Foden diesel engines or engine derivatives, which never got very far. On the 92mm bore engine there were two different experiments with four valve heads, i.e. four exhaust valves. The aim was twofold. By using four valves it was hoped to get a much more rapid increase in valve outlet area as the valves opened, so getting a bigger exhaust pulse and intensifying the Kadenacy effect Also, because of the bigger open valve port area, exhaust blow-down and scavenging would be improved and it should be possible to open the valves later and get a longer firing stroke. However, there was one possible disadvantage in that the more rapid reduction of the exhaust port area could have a deleterious consequence for the Kadenacy effect.

To some extent these experiments were not pursued with the effort and enthusiasm, which characterised so much of the other engine and vehicle projects undertaken by Foden. For example no special cam shape was produced to go with these heads to give either a longer firing stroke or a more gradual closing of the valves. A small gain in firing stroke was achieved by widening the tappet clearance. This obviously reduced valve lift and from that point of view was counter productive.

One of the designs of heads had a central multi hole injector, casting aside one of the well-proven good features of the other Foden engines - the single hole nozzle. This was done because of the difficulty in accommodating a side injector with four exhaust valves. However, engine performance was poor and nozzle life was unsatisfactory. The second type of valve head reverted to the single hole, side mounted injector and tackled the complications and congestion of this injector location with four valves. This again resulted in deterioration in engine performance and was not persevered with.

For the operation of four valves, the valve gear had to be revised. Cam loading could not be doubled so it was necessary to provide two cams per cylinder and new heads were required to carry the two cam followers. For each cylinder the forward cam operated the two valves forward of the cylinder centre line and the rear cam the two valves to the rear of the cylinder centre line.

One possible reason for the lack of enthusiasm for the 4 valve head was the problem of cylinder head cracking. This was still a major problem on the 2 valve head and it may have been felt that the problem would have been intensified if a 4 valve head were put into production. On the other hand it might have eliminated the problem – we didn’t know. Nowadays some clever computer could have done a few sums and told us whether or not the new arrangement would be free from cracking problems but this facility was not available to us then. Even if it had been, we may not have believed it.

Another major change attempted late in the life of the engine was a further increase in the bore from 92mm to 100mm. If the same specific performance (bhp per litre of swept volume) were achieved, this would have taken the blower scavenged 6 cylinder engine up to 210 – 215 bhp and a turbocharged version up to 265-275 bhp, with the possibility of 300 bhp from an uprate in line with the rating of the 85mm bore Mk VI. Tests on a 6 cylinder engine gave poor results so a development programme using the single cylinder research engine was begun. It was found that the single would perform as satisfactorily with a 100mm bore provided that the air chest volume was increased substantially. This suggested that interference between air flows to different cylinders was taking place in the air chest of the 6 but as the volume of the 6 cylinder engine air chest could not be increased significantly without the design and manufacture of a completely new cylinder block and crank case, the project had to be abandoned and the way towards higher powers sought by other means.

The last major engine development project started but never finished was an FD8. The FD8 was an 8 cylinder in line engine; the blower scavenged Mk VI version would have given 240 bhp and turbocharged Mk VII, 300 bhp with a possible uprate to 330bhp. Maximum torques would be about 590lb.ft on the Mk6 and 800lb.ft on the Mk VII. Although many of the special components were made, including crankshafts, crankcase castings, camshafts and injection pumps, a complete set of engine parts was never collected together and an engine was never assembled. Components such as heads and valve gear were common to the 6 cylinder engines but there were one or two interesting variations.

It was not thought that the roots blower could be lengthened to make a blower of the required throughput, as the rotors would become rather long and spindly with possibly a greater tendency to seizure because of increased flexibility, and to increase its diameter would entail a lot of new tooling and would require the blower drive moved outwards from the centre line of the engine and consequently a new timing gear train, the expected number of engines to be built could not justify this. The solution chosen for the Mk VI version was to use two FD4 size blowers. A shaft drive was carried forward from the timing gears at the back of the engine to the mid point of the crankcase, between cylinders 4 and 5 on the left hand side; this shaft drove a small transfer gearbox. One standard FD4 blower was mounted forward and above the line of the driveshaft and a special mirror image FD4 blower with the drive at the front end of the blower was mounted behind the transfer box and above the primary driveshaft. This was a similar blower to the one used on the A side of the FD12 engine but FD4 length. While this arrangement introduced a lot of new drive parts, a new blower casing and new end covers etc., the rotors were standard FD4 and minimal new tooling was required, there were also some advantages in having the transfer box in this location which we will come back to.

Thinking of the Mk VII version of the FD8, at first sight it might be thought that one standard FD6 blower could be used giving 75% as compared with the 78% blower used on the FD6 Mk VII. However this would only be feasible if a single large turbocharger was to be used. Other factors were pointing to the use of two small turbochargers; that arrangement would simplify the manifolding and two small turbochargers might well be lighter than one large one. It would be easier to match small turbochargers to the engine and their rate of acceleration would be better, easing the problem of accelerative smoke. However, the use of two small turbochargers would demand two completely independent air systems, as there is danger of compressor instability if two units feed into one air receiver. In other words each turbocharger would feed through its own intercooler into its own root blower and to its own air chest, two completely independent systems. Everything was beginning to point in the same direction; in this size of engine two small turbochargers would probably be cheaper than one large one and would probably operate in a more efficient area of their characteristics. Manifolding would be simpler but other piping to and from two separate intercoolers might be tricky. Two roots blowers would be required in a similar layout to the FD8 Mk VI and the air chest would have to be split.

There was already another good reason why the air chest should be split between numbers 4 and 5 cylinders. The firing order chosen, 1-8-2-6-4-5-3-7, meant that numbers 4 and 5 cylinders fired at an interval of 45 deg. This meant that both cylinders would be drawing air from the air chest together for a short time. To avoid interference of airflow between the two adjacent cylinders it would be necessary to divide the air chest between them. Also, because these two cylinders fired in quick succession the load on the main bearing between them was greater than on other main bearings and a longer bearing was required. Incidentally the crankshaft was located longitudinally at this number 5 bearing.

In effect, in terms of gas dynamics this would be like two separate 4 cylinder engines. Turbocharging of 4 cylinder engines is not as easy as 6s because it is difficult to get the turbine to operate efficiently. As single entry turbine casing is less efficient than two entry one because of the small arc of casing feeding gas to the turbine and, if a two entry casing is used, gas supply on a 4 cylinder engine is not continuous in each entry. At this time some work was in progress at Queen Mary College of the University of London on a turbocharged version of the 92mm bore FD4 engine by M.S Janotta and N Watson.

The inward flow turbine of the typical small modern turbocharger likes to be fed with gas at a constant pressure. The engine on the other hand, and particularly a 2 stroke engine, performs better if the exhaust gas flow pulses in such a way as to improve the scavenging of gas from the engine cylinders. Where a manifold branch feeding a turbine inlet is carrying the exhaust from only two cylinders the pulses are so far apart that the turbine performance suffers much more than it would if it were carrying the exhaust from three cylinders exhausting at 120deg. intervals as it would on one half of a 6 cylinder, 2 stroke engine. With two manifold branches each carrying the exhaust from two cylinders as on a 4 cylinder 2 stroke or half of an 8 cylinder 2 stroke, the turbine performance can be substantially improved by connecting the two branches of the manifold together before the turbine inlet, in a specially designed junction called a pulse converter. This devise enables the turbine to operate more efficiently without upsetting the pulses at the exhaust valves, which are so important to cylinder scavenger efficiency. The design and development of a pulse converter was the work that Janotta and Watson were doing, and the progress they were making was sufficient for us to decide that the FD8 Mk VII would have two small turbochargers rather that one large one and two completely independent air intake and delivery systems. As far as the size of root blower for the Mk VII was concerned, this could be adjusted by varying gear ratios in the mid mounted blower drive transfer box.

This transfer box also drove an output shaft axially in line with the input shaft from the timing case, but forward to drive the water pump mounted at the front of the left hand side of the engine. The water pump delivered water through an oil cooler horizontally mounted on the right hand side of the engine and then into the right hand side of the cylinder block between nos.4 and 5 cylinders. A water rail mounted on top of the cylinder heads as on the FD6 and the FD4 engines carried the outlet water to a thermostat housing which incorporated two, 3 part Weston Thompson thermostats (to minimise cooling system pressures) from which there was an outlet for the radiator connection and a bypass pipe down to the water pump.

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An alternator was to be mounted on the left hand side of the engine below the blower drive and driven by a short shaft from a gear in the timing case.

A three-belt drive off the front of the crankshaft drove the fan and the airbrake compressor, a torsional vibration damper being sandwiched between the flange on the crankshaft and the pulley.

The firing order chosen 1-8-2-6-4-5-3-7, at 45 degree intervals gave a smooth turning moment, which would make life easier for the transmission. It also gave perfect primary and secondary force balance with perfect secondary couple balance but left a small primary couple out of balance. This was rather less that one third of the primary couple present on the FD4 engine and, being in an engine of almost twice the length, it was unlikely to be a problem. However such a long crankshaft would need to be stiffened up so the journals and crankpins were increased in diameter to meet this need. This, of course, meant new main and big end bearings and new connecting rods. As mentioned above, it was also necessary to fit a large torsional vibration damper, but there was some small spin off in that the flange provided on the front end of the crankshaft to carry the damper also provided a full engine power take-off at the front of the engine. The revised drive for engine auxiliaries enabled us to dispense with the auxiliary drive housing mounted on the front end of the FD4 and FD6 engines thus minimising the overall length of the engine.

Various other minor alterations were necessary such as larger diameter oil passages to cater for an increase in oil flow up to 32 gallons per minute as compared with 24 gallons per minute on the FD6. A water pump and a new oil pump were required. The oil pump was mounted at the rear of the engine picking oil up from a sump well at the rear and delivering it through twin oil filters mounted at the rear of the right hand side of the engine to the oil cooler, also on the right hand side and thence with the crankcase gallery pipe via a tapping more or less in line with no. 4 cylinder.

The last skeleton in Foden’s engine cupboard was the VF, last to be mentioned here but not, the last chronologically as this was being worked on in the late 1950’s. This was another uniflow two-stroke 120mm bore by 150mm stroke. It was to be built in vee form with a cylinder bank angle of 87½ degrees. The engine was to have a cast iron crankcase/cylinder block partly because the Vee contribution precluded the use of long through bolts (from cylinder heads to main bearing caps) which took all the firing loads in the FD engines. It was to be built in 4, 6 and 8 cylinder forms, all 87½ degree vees with an output, blower scavenged, of 40 BHP per cylinder at 1800 rpm. This, with turbocharged versions would give a range of powers from 160 to 400 BHP for a turbocharged VF8.

Apart from the change from aluminium to cast iron crankcase, there was another major difference between the VF and FD engines; the VF had a central multi-hole injector. Also the combustion chamber in the piston crown was a deeper toroid with re-entrant sides. The engine had a four valve head. The piston was similar in general design to the FD piston.

A single cylinder development engine was built and initial tests were aimed at obtaining the desired power level and improving fuel consumption and exhaust smoke levels. Tests of the blower scavenged single cylinder engine were going quite well with promising fuel consumption figures when the project was stopped, it being decided that the company could not afford to tool up to make the engine even if it turned out well. This was a disappointment to the Experimental Department staff who felt that good progress had been made, but there is no doubt that it was a wise decision.

Chapter eight

# Cylinder head problems

The Mark I engine was generally quite reliable, apart from problems with the cylinder head gaskets. These problems were too numerous to ignore and it proved impossible to eliminate them with the materials and techniques available at the time. A complete redesign of the cylinder block/cylinder liner/cylinder head assembly was decided on.

The conventional multi-cylinder heads (two heads covered six bores on the FD 6 and one head covered four bores on the FD 4) were replaced by unit heads, one for each cylinder. The head was secured directly to the top flange of the cylinder liner by eight studs, carefully designed to provide a lot of stretch when torqued up so as to minimise loss of squeeze on the gasket. The gasket was a plain copper ring of very simple design. It was also very successful as cylinder head gasket problems disappeared. Unfortunately an even worse problem replaced it; cylinder head cracking.

The heads cracked between the two exhaust ports, a very hot part of the head. This problem did not affect marine engines but was far too prevalent on automotive engines to be acceptable. What was happening was that the metal between the two ports was getting very hot and trying to expand. It was prevented from doing so by the comparatively cool and rigid box shape of the unit cylinder head. Consequently when very hot, compressive creep of the metal took place between the exhaust ports, then when the metal cooled down the metal went into tension, a force that cast iron is not very fond of. Repeated cycles of compressive creep and resulting tension cracked the heads. To overcome the problem what was needed was a material that had the good features of cast iron and also;

1. A very high rate of thermal conductivity
2. A very low rate of thermal expansion

In order to find a suitable design and material we had to devise a test that would sort out the sheep from the goats quickly. We had to cycle the heads though a very hot period followed by a cool period, simulating a vehicle climbing a long hill then coasting down the other side. This was fairly easily done and we found we could crack the current production heads in about 50 hours running.

To start with, to improve the life of the heads we modified them to make the material thickness between the two exhaust ports as thin as we dared, bearing in mind production tolerances this produced only a slight improvement. The next step was to try different materials for the castings. We tried aluminium, aluminium bronze and tungum, a form of brass, what we gained in thermal conductivity we lost in other ways. We also tried coating the flame plate, firstly with copper plating to encourage the heat to move to other parts of the flame plate, no improvement! Then we tried spraying a ceramic coating on the face of the head to reduce the heat getting into the iron, the ceramic came off!

Time was passing, we were getting nowhere, customers were understandably getting fed up and it was time for more drastic measures. We resorted to protecting the cast iron by fitting a plate of heat resisting steel to the bottom face of the head. In the first modification the protective plate was quite thin and was brazed into a shallow recess machined into the bottom face of the head. It warped and came loose. We then moved into the realms of commercial fantasy, going to a thicker protective plate fitted into a deeper recess, which of necessity, broke through into the water compartment of the head and the insert had to be electron beam welded into the head. Getting 100% watertight welds between the Nimonic plate and the cast iron head, including sealing the exhaust passage, was just too difficult and the cost of the process was far too high.

This was the end of our attempts to solve the problem. If we had had a computer of the power now available to design engineers, it is possible that we would have found a solution, we hadn’t and we didn’t. It ended the life of the engine for automotive purposes but production of the engine for marine customers and particularly for the Royal Navy continued.

Chapter nine

# Marine engines

Admiralty engineers who looked at the FD6 Mk I engine when it was launched at the commercial motor show found it to be quite interesting. They were looking for a diesel engine to be adapted as a “standard engine” for various duties. Features, which appealed to them, were its compact size, light weight and, thinking of magnetic mines, its low iron and steel content, in other words its low magnetic weight.

Subsequently an engine was sent to the Admiralty Engineering Laboratory, then at West Drayton, for their type test and after a few minor problems the test was passed and the engine adopted as a standard engine for use by the Royal Navy. It was used for many applications, as a propulsion unit for small craft, as an auxiliary generator for larger craft and for use in cranes, dockside locomotives, fire pumps etc.

It soon built up a reputation for excellent reliability (in contrast to the automotive application) and long life, and consequently was adopted as a standard engine for other navies such as the New Zealand Navy, the Danish Navy, the Indian Navy and others.

An interesting result of its excellent reliability in naval use was that the Navy over-ordered spare generator sets, and some were sold to the Merchant Navy, where again they performed well. One ship used to start up the Foden generator set before it left the UK and didn’t stop it until they got to Australia six weeks later!

The Navy was delighted when the later marks of engine gave them more power from the same space, particularly the Mk VII turbocharged engine. At an earlier stage, again because of the reliability, they asked if we could supply an engine of twice the power. This request created the FD12 double bank engine. Fodens did not want to go to a vee configuration as this would have meant the aluminium crankcase having to take the force from the firing stroke, this being absorbed in the single bank engine by the through bolts from cylinder heads to main bearing caps. Consequently the FD12 came out as a double bank engine with two six-cylinder crankshafts geared together. This created a large complex crankcase casting, but again the foundry was amply good enough. There was also a minor advantage in that the power could be taken off either of the two crankshafts, which offered alternative directions of rotation. There was also the possibility, which never came to anything more than a passing thought, of having two FD12s back to back with a 350 kw generator in between, one engine driving off one bank with the other driving off the other bank. Taking this fantasy a bit further, by incorporating some sort of clutch between each engine and the generator, one engine could be declutched for maintenance work and the other engine could give up to half the power for the generator.

One of the last major developments of the marine engines was the non-magnetic FD12 for minesweepers of the glass fibre hulled Hunt class. Here, to the best of my knowledge, Foden produced an engine that no other manufacturer in the world managed to do. By changing the materials of the major steel and iron components, such as crankshafts, camshafts, connecting rods and pistons, the engine had a very small magnetic signature. One change was to go for non-magnetic stainless steels for some parts. This produced problems of lubrication for the crankshafts because stainless steel is not as easily lubricated as conventional steels. Here Foden were helped considerably by bearing material makers and oil companies.

The FD12 also caught the eye of makers of high speed pleasure craft, in particular racing boats that had to have reasonable passenger accommodation, as opposed to out and out racing boats. One such craft was Anglesey II fitted with two FD12 Mark VII engines set to a special sprint rating of 450 bhp each. Anglesey was entered for the third international offshore power boat race over the 170 mile course from Cowes to Torquay in 1963, but at scrutineering the passenger accommodation just failed by a few inches to meet the requirements, and she was not allowed to enter the race. Not to be put off by that, the crew waited until all the other boats had left the harbour then set off. They passed most of the other boats and they were the seventh boat of all the entrants and the first diesel boat into Torquay. Had she been an official entrant, starting as planned, she would have been third overall and first in the diesel class.

Chapter ten

Epitaph

The difficulty in curing the head cracking problem occurring on automotive engines, together with trading problems in the late ‘70s resulted in a decision to sell the FD engine manufacturing rights and replacement parts business to Rolls Royce. Sadly that did not result in further development or manufacture, but it did ensure a good supply of replacement parts for several years.