From Humble Shunter to Transatlantic Blue Riband Winner in Sixty Years. Some recollections of the development of the famous English Electric RK diesel engine

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This paper describes some of the stages in the technical development of a classic British medium speed diesel engine. This resulted in a ten-fold increase in specific power over a period of sixty years.

Conceived in 1930, by the English Electric Co., in Rugby, the first production engine was delivered in 1934. This was the prime mover for a diesel-shunting locomotive. This six-cylinder engine was released at a rating of 50 bhp (37 kW) per cylinder at a speed of 600 rpm. Towards the end of the twentieth century the power per cylinder had been increased to over 500 bhp (373 kW) with twenty-cylinder engines being released for naval duties delivering 10,265 bhp (7,658 kW) at a speed of 1,032 rpm. Since inception, market needs required the engine to provide more power at lower cost with greater reliability within the same package size. Improved specific fuel and lubricating oil consumption and reduced weight for mobile applications were also required. Reductions in exhaust emissions other than those contributing to visible smoke were not of prime concern until the latter part of the engines life.

The paper will describe how some of these apparently conflicting requirements were achieved. Reference will be made to the fundamental design principles of the compression ignition engine with particular respect to the functionality of the major components. Several of these components will be discussed in greater detail. This will provide more insight into the diesel engine which has become, probably, the most fuel efficient prime mover known.

KEYWORDS: diesel, power, modification, development

Introduction

This is the story of a classic diesel engine which started life in the 1930s and is still in production today. The engine is a design classic; a term which is the subject of many unconcluded debates but when it is applied to a mechanical device it generally means that within its field it is highly regarded and is better than the majority of its competitors. Of course, there are many other engines no less deserving of this description and from many builders.

Conceived in 1930 by the English Electric Co., the engine was initially produced in six-cylinder form to power marshalling yard shunters and was installed at a rating of 300 bhp at 600 rpm. Sixty years later towards the end of the twentieth century twenty-cylinder versions of the engine were available producing 10,265 bhp at a speed of 1,032 rpm for naval duties. This is equivalent to a power per cylinder increase from 50 bhp to over 500 bhp, more than ten fold.

Background

The engine was known for much of its early life as the English Electric K engine. Following the many revisions to the design it was referred to as the "revised" K and then more simply as the RK. The English Electric Co. acquired Ruston and Hornsby of Lincoln in 1966 and thereafter the engine range was known as the Ruston RK, eventually becoming the Ruston RK270. Initially available in 4, 6 and 8 cylinder inline forms it was later offered in 8, 12, 16 and eventually 20 cylinder vee configuration. The cylinder bore, which for fifty years had been 10 inches (254mm), was increased to 270mm (10.63 inches) in 1980. The stroke remained unchanged at 12 inches (305mm). The K and the RK 270 engines are shown in Fgures 1 and 2.

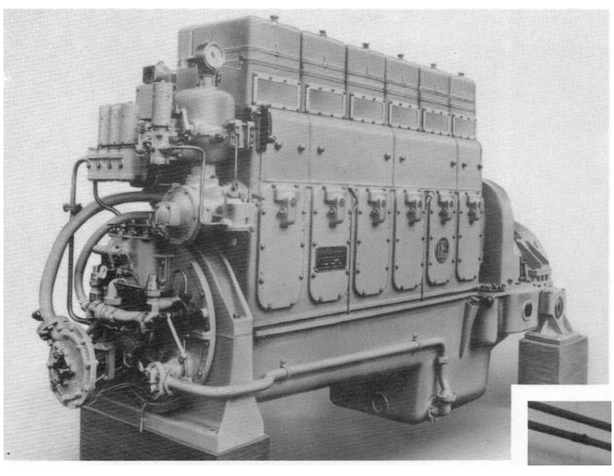


Figure 1. 1937 6K engine.

The design philosophy was to use the same components for each cylinder adding more cylinders to obtain more power and arranging them in either in line or vee formation to suit the application and to achieve best cost effectiveness. This modular approach meant that economies of scale could be achieved in production and that operators would benefit from reduced spares inventory.

Initially designed as a conventional direct injection compression ignition diesel engine running on standard distillate fuel it was later offered to run on natural or sewage gas, the charge being ignited by either spark or pilot diesel fuel. Engines were also available to use “heavy” fuels up to 1500 Redwood seconds viscosity, for marine and industrial use.

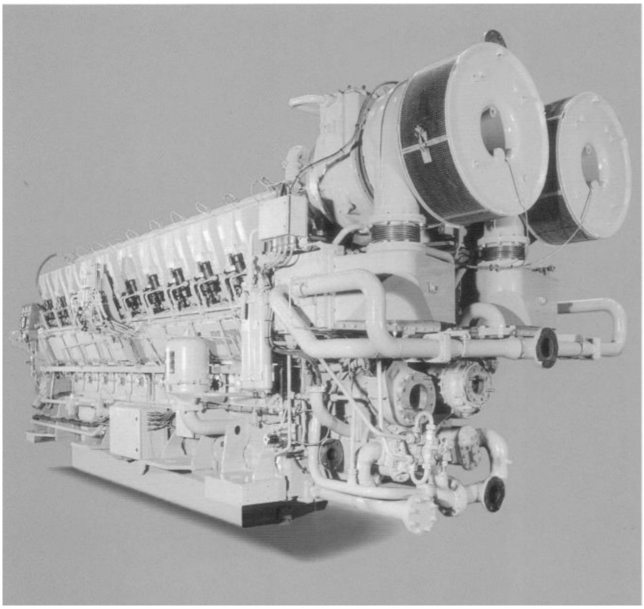


Figure 2. 1996 20RK270 Engine.

The applications were many, including rail traction, industrial and marine power generation, marine propulsion and direct drive gas and fluid pumping duties. This diversity of application was a result of the engine being in the medium speed class. It provided useful power between 600 and 1100 rpm and was small enough for use in mobile applications but large enough to provide better specific fuel consumption and longer periods between maintenance than the lighter higher speed engines. The engines were never mass-produced in the automotive sense and were able to incorporate customer specific requirements relatively simply. Over its main production life of seventy years more than 20,000 were produced. The engine was marketed as robust and heavy duty which means that it would provide at least 90% of its full load for 90% of the time within reasonable maintenance periods and with a major component life of 120,000 hours.

Since inception the engine was required by market forces to provide more power at lower cost with greater reliability within the same package size. Improved specific fuel and lubricating oil consumption and reduced weight for mobile applications were also required. Reductions in exhaust emissions other than those contributing to visible smoke were not of prime concern until the latter part of its life. So, how were these apparently conflicting requirements addressed?

Uprating Potential

The four stroke cycle gives designers the ability to increase the power of such an internal combustion engine by changing some or all of the operating parameters. The pressure generated in the cylinder by the combustion gases is proportional to the amount of fuel burnt and the rate of burn. If more fuel can be burnt in each cycle more energy is produced. This requires more fuel to be delivered and more air to allow the fuel to burn properly. The air is introduced by the downward motion of the piston on the induction stroke reducing the pressure inside the cylinder to below that outside. In the case of a naturally aspirated engine this is atmospheric. If the pressure of the external air can be increased more can be introduced into the cylinder. This can be achieved by either a mechanically driven compressor or in the case of the RK engine an exhaust gas driven compressor known as a turbocharger. The process of compressing the air causes an increase in temperature, which reduces the mass of air available for combustion. In order to mitigate this effect the air is passed through a cooling device enroute to the cylinder. This is known as a charge air cooler. Further power increase can be obtained by increasing the volume of each cylinder or the number of cylinders in operation. By increasing the speed of the engine the number of operating cycles per minute is increased resulting in yet further power increase.

Therefore from a given starting point an engine can be uprated by increasing the number of cylinders, fitting larger fuel pumps, a turbocharger and charge cooler and increasing the rotational speed. This was the route followed by our engine in many stages as a process of relatively safe evolution.

It would be misleading for me to claim that the power increases were achieved by using all the original components. This would have been impossible but the major defining features of the engine have remained unchanged. These include the use of a separate bedplate and crankcase, individual cylinder heads and fuel injection pumps, side-by-side connecting rods on the vee engines, side mounted camshafts with push rod operated inlet and exhaust valves and individual water cooled cylinder liners. The bore and stroke of the engine was as originally conceived 10 inches by 12 inches (254mm x 305mm). The bore was increased to 270 mm in 1980. This gave a swept volume increase of 13%, the stroke remaining unchanged at 305 mm. Some may say that this made it into a new engine and indeed it was a major change, but it was achieved without increasing the all important pitch between adjacent cylinder centres by reducing cylinder liner wall thickness and detail changes to the cylinder head to liner sealing joint design. One of the major contributors to the overall length, weight and cost of any engine is this cylinder centre distance and by retaining this the overall length was unaffected and the engine identity retained.

Having obtained all this extra power most of the components of the engine will be subjected to greater forces and the complete structure must be strengthened

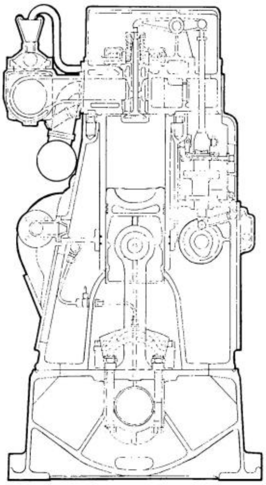
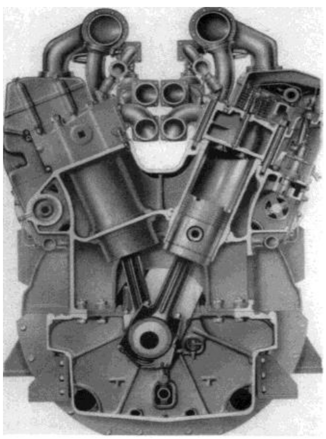


Figure 3. 1934 K engine cross section.



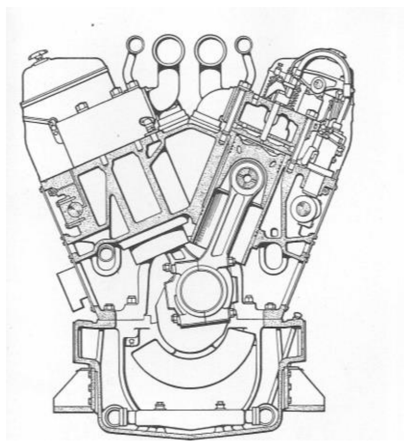


Figure 4. Early vee engine. Figure 5. 1986 RK 270 engine.

to hold it all together. To put these forces into perspective the 20 cylinder RK 270 engine delivering 10,000 bhp (7,460kW) will undergo 10,000 firing strokes each minute, each stroke delivering a force of 100 tons to each piston 500 times per minute. It is designed to do this for over 6,000 h per year without breaking for up to 20 years, and with reasonable maintenance requirements. This is no mean feat and it is amazing that the modern diesel engine can do all this and burn its fuel more completely, more cleanly and more quietly than the early pioneering engineers could have ever envisaged.

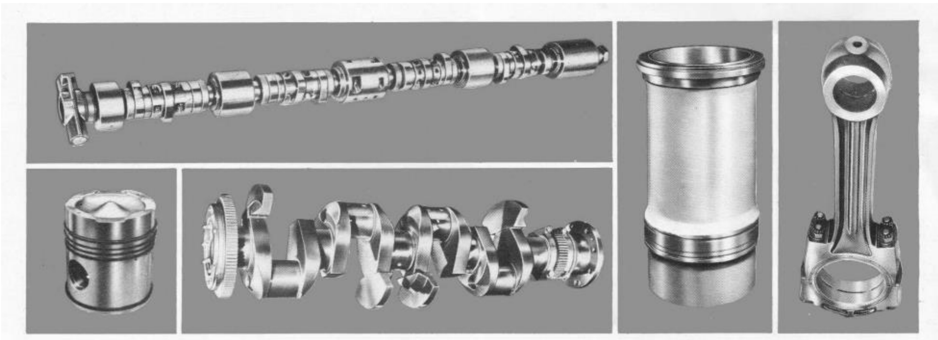


Figure 6. Some 1960s engine components clockwise from top left: sectional camshaft, cylinder liner, connecting rod, crankshaft, five ring piston.

Component Developments

The Bedplate Engine

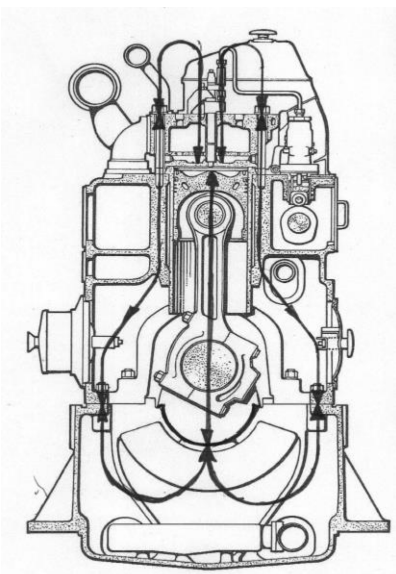


Figure 7. Force reaction loop. The red lines show the flow of forces from the firing loads through the cylinder head studs and into the bottom of the main bearing. This puts the connecting rod into compression and the crankcase into tension which tries to separate from bedplate and to open the joint face. The forces are reacted through the joint face flange by the connecting bolts and the clamping load must be sufficient to prevent separation.

The RK engine is known as a "bedplate" engine, a defining feature used by engineers to differentiate it from the other common design known as the "underslung crank" engine. Many large internal combustion engines designed at the beginning of the twentieth century followed earlier vertical steam engine practise wherein the crankshaft rotated in bearings mounted in a solid foundation structure also known as a bedplate. When correctly designed this provides the theoretically superior support this provides the theoretically superior support to the crankshaft main bearings but usually results in a heavier and more expensive engine than the underslung crank concept.

The technical advantage of better bearing support and reduced load on the main bearing caps and bolts is offset by the difficulty of maintaining a sound interface betweeen the crankcase and bedplate. This interface which takes the form of a flanged bolted joint is required to transmit the tensile loads from the combustion forces acting on the cylinder head to the main bearings via the crankcase and bedplate structures in order to complete the force reaction loop. (See figure 7) It is subjected to separation loads by twisting and bending and must resist fretting and remain oil tight. The RK engine has featured the bedplate throughout its life and during the uprating process this joint face flange has been the subject of much development and improvement.

The Bedplate

The bedplate is a cast iron structure which as well as forming the foundation of the engine and supporting the crankshaft also provides the attachment points for the engine mountings which can be either continuous beams or separate feet. This stiff structure enables flexible mountings to be employed to isolate the engine from, for example, the hull of a vessel or the chassis frame of a locomotive. This will prevent external deflections from these structures interfering with the crucial main bearing alignment. In the case of a close-coupled rail traction application, the bedplate also enables a continuous circular mounting flange to be used to link the engine and the electrical driven machine.

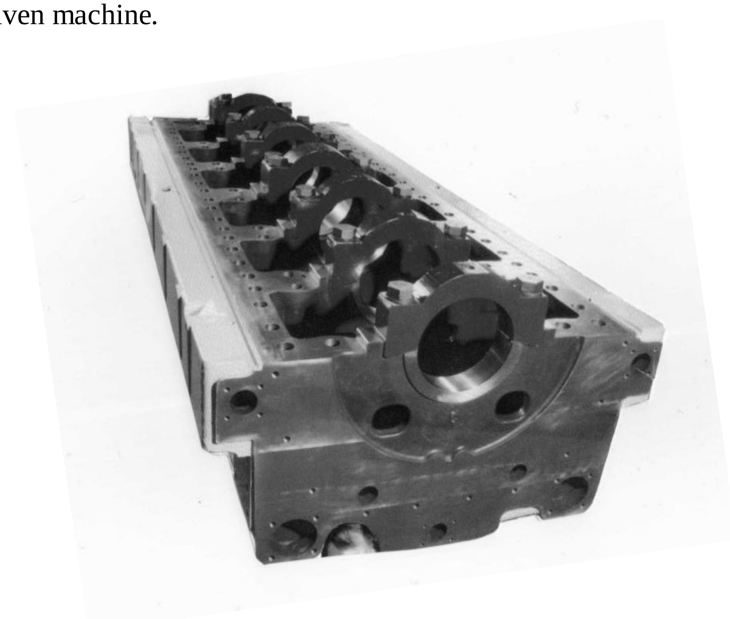


Figure 8. RK Vee engine bedplate.

The major changes required to this component to contain the forces imposed by the uprating process were material changes from grey cast iron to spheroidal graphite iron and increases in the material thickness to provide greater strength and stiffness. The main bearing housings were increased in section thickness and the all important joint flange thickness was increased to allow greater bolt loadings to be accommodated. The use of spheroidal graphite iron also provided the shock loading resistance required when the engines were used in military applications. Figure 8 shows a typical bedplate.

The Crankcase

This is also an iron casting which contains the cylinder liners with their cooling water and has always featured side mounted camshafts driven from the flywheel end. The auxiliary drive system for the lub oil and water pumps is taken from the other end. The top deck of the crankcase features tappings for the individual six bolt cylinder heads and the mounting points for the individual fuel injection pumps. As with the bedplate, the uprating required significant increases to the material thicknesses particularly with respect to the lower edges of the diaphragms, which run across the crankcase between the pairs of cylinders on either side of the vee engine banks. These diaphragms have thickened edges to increase the section modulus and to resist the tensile loads imposed by the firing forces. The top half of the joint face flange was also thickened and the end walls of the crankcase were increased to accommodate the extra loads from the heavier turbochargers and charge air coolers, which are mounted on these walls.

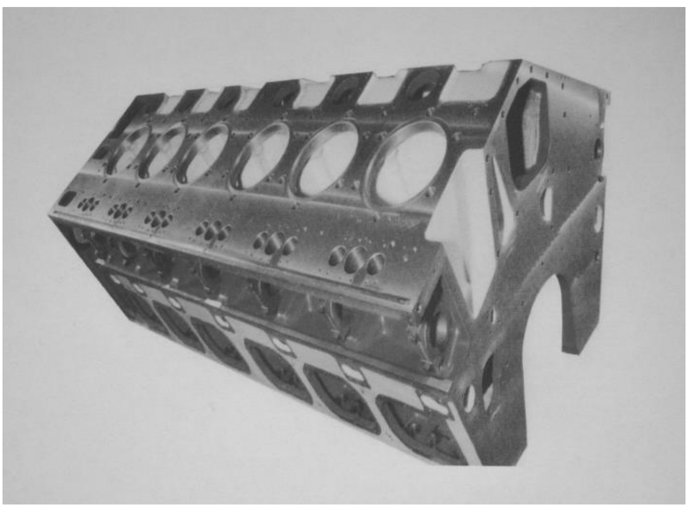


Figure 9. a late RK270 crankcase.

The camshafts were originally mounted in bearing tunnels and were inserted from one end, which was a tricky operation requiring special tooling. The larger fuel pumps required to produce the extra power are driven from the camshafts and in order to keep the fuel cam stresses within acceptable limits larger cams were specified. The tunnels in the crankcase were then changed to open channels with separate bearing housings enabling the shafts to be fitted from the side. The vee of the crankcase between the two cylinder-banks was originally open but was later closed to form an integral air chest for the induction air. This increased the torsional strength of the crankcase by closing the top and enabled the previously used external sectionalised air distribution system to be abandoned. This integral air chest was also used on the in-line engines again increasing stiffness. All these changes were aimed to improve strength and stiffness and the inevitable weight gains had to be less than or equal to the specific power increase. Figure 9 shows a late RK270 crankcase.

The Crankshaft

Considered by many to be the heart of the engine, this is the final link in the chain of components, which convert the linear motion of the piston into more usable rotary motion. The material has always been a continuous grain flow chrome molybdenum alloy steel forging. Improvements to the strength of the material were made over the years culminating in a final ultimate tensile strength of 60 tons/in^2. The shaft is subjected to alternating torsional, bending and shear forces and is highly polished in the critical areas to avoid stress concentrations that would lead to fatigue failures. The shaft is supported in the main bearings housed in the bedplate and the large ends of the connecting rods embrace the crankpins. The crank webs also provide attachment points for balance weights which counteract the out of balance forces caused by the reciprocating and rotating masses, and help to maintain suitable oil film thicknesses in the bearing system. The six-, twelve-, sixteen- and twenty-cylinder engines are perfectly balanced for primary and secondary forces and couples. The vee-8 engine uses three contra-rotating balancer shafts mounted in the bedplate to give satisfactory external balance. The RK vee engine has always used side by side connecting rods which means that each pair of cylinders across the two banks share a common crankpin, the rods being adjacent and of the same design. The cylinder centres are staggered bank to bank to allow this. This method allows commonality of components between vee and in-line engines.

The modifications to allow the power increase included changes to the material and the forging methods and detail improvements to the critical junctions between the webs, pins and journals to reduce the tensile stresses in those regions. The shaft has never been surface hardened or chrome plated on the bearing surfaces as this can lead to difficulties in repair in the event of a bearing seizure. The main bearing diameter was increased to give more strength to the shaft by increasing the critical overlap between pin and journal and to give reduced bearing pressures avoiding the need for local surface treatments. This was achieved within the same bedplate dimensions by the use of tri-metal thin walled bearing shells in place of the earlier thick walled shells but having the same external diameter and using the same housings. The early bearings were white metal lined onto thick steel shells which gave good seizure resistance but lacked the fatigue strength for uprating. As the engine ratings increased the steel backings were reduced in thickness and the bearing material went through a variety of copper-lead alloys overlay plated with a lead-tin alloy. These changes lead to the bearing load capacity to be increased by a factor of three.

When the number of cylinders was increased to twenty from sixteen the coupling flange at the end of the shaft was increased in diameter to allow a reliable connection to the driven machine. This connection was made by bolt clamping forces rather than by the use of fitted bolts in shear. A connection at the other end of the shaft was also available for a reduced output power take off system.

The thrust bearing required to react the end thrust on the shaft, caused by the driven machine or by installation at an angle to the horizontal, was moved from a position in the centre of the engine to one nearer to the drive end to reduce the axial "panting" effect of the increased firing loads on the crank pins. This is known as crank web "breathing" and if unchecked can result in unacceptable axial vibrations. Torsional vibrations are also present within the crankshaft system and the uprating required changes to the flywheel and vibration damper system to minimise them.

The Connecting Rod

The small-end is flexibly connected to the piston by a gudgeon pin and the large-end embraces the crankpin and is split to allow for assembly and disconnection. By definition, the piston assembly of a reciprocating engine is required to be accelerated from rest, brought to a maximum speed and then decelerated to rest twice for each revolution of the crankshaft. The function of the connecting rod is to transmit the force caused by the cylinder pressure acting on the piston into the crankshaft and whilst doing so the shank of the rod is under a compressive load. However, on the exhaust stroke the rod is under tension as it restrains the upward inertia of the piston assembly. Consequently the connecting rod experiences alternating compressive and tensile loads during each cycle. This causes the shank to tend to buckle and stretch and the small and large end eyes to deform as they ovalise under these forces. The firing forces are proportional to the engine load whereas the inertia forces are speed dependant. Although simple in concept the detail design required to make a connecting rod operate satisfactorily is considerable, as it has to be light and strong in tension, compression and bending, and has to be assembled and dismantled several times during its life. Generally for this class of engine the small end is a complete ring and the gudgeon pin is assembled to the piston outside the engine by simple insertion through holes in both components, but the large-end is assembled around the crankpin which requires it to be split. This is a designer's challenge as it is also desirable for the part of the rod, which is attached to the piston to be withdrawn through the cylinder liner. If the cylinder bore is too small the large-end is bolted to the shank. This is known as a three-piece rod, whereas if the bore is large enough a two-piece rod may be employed.

The RK engine started life with a three-piece rod which needs two bolted interfaces and the consequent fretting problems as the interfaces move microscopically are doubled. This fretting can result in micro metal transfer which then forms stress raisers and in extreme cases leads to catastrophic fatigue failure. In order to reduce this problem the interface should be reduced to one. The second generation of engine saw the introduction of a two-piece horizontal-split rod with the clamping bolts slightly angled to improve access and to provide a horizontal component of force to abut the top half of the rod snugly into the cap half.

As the large end ovalised mainly due to the speed dependant inertia forces, fretting and metal removal still took place which resulted in closure of the joint and a reduction in the diameter of the bearing housing. This could eventually lead to loss of bearing clearance and seizure. As the engine design speed was increased for more power the large-end split was changed from being horizontal to angled and the plain faces were given serrations to provide location and extra shear force reaction surface. The detail of this angled joint face was the subject of much consideration, the two halves of the angled face being parallel but not coincident with the centre of the bore as in a simple split. The clamping bolts were also set at an angle to bias the load to one side of the serrations. The consequence of the ovalisation of a ring when under tension is that the radius of the sides of the ring tends to increase and that of the ends tend to decrease. If the split is placed at the null points, the bending and shear forces across the joint faces are minimised. The null position is simple to calculate when the ring is of uniform stiffness but not so when it features transverse holes for the bolts and is blended into an H section shank. This design was the latest to feature in the RK engine and was a major factor in raising the operating speed (see Figure. 10).

The Piston

The piston forms a movable end to close the cylinder and is subjected to the force of the expanding combustion gases, which in the case of the RK engine can be as high as 100 tons at a firing pressure of 172 bar. At a rated speed of 1,000 rpm and for an expected life of 60,000h this equates to 1.8 $x10^9$ cycles. It transmits this force to the connecting rod and provides a location bearing for the top end. It has to be as light as possible to reduce inertia forces and able to withstand a temperature of 350ºC at one end whilst being at 80ºC at the other end. It also has to be gas and oil tight providing a seal against a pressure of 2500 psi whilst moving through the cylinder at a mean speed of 2000 ft per minute. The early engines in the 1930s used cast iron pistons which although having the required strength and bearing properties were too heavy to allow a speed increase.

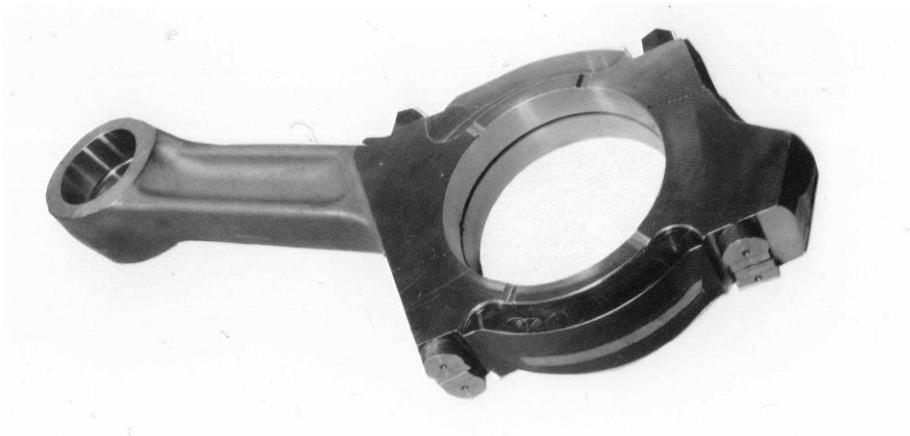


Figure 10. RK270 connecting rod.

As the engine was uprated, low expansion aluminium alloys were used which were initially cooled only by oil splashed from the large-end bearing outflow. With improvements in casting technology, an internal cooling gallery could be incorporated into the crown that was fed by oil from the connecting rod. This was needed to prevent the crown becoming weakened by excessive temperatures, which leads to thermal cracking and eventual failure. Hard anodising of the aluminium crown was used to improve fatigue strength, but as firing pressures and gas loads became higher the single-piece aluminium piston was changed for a composite design using a steel crown bolted to an aluminium body. This allowed the use of a 3 bar charge-air pressure with 172 bar peak firing pressure, and combined the hot strength properties of the steel with the lower weight and good bearing properties of the aluminium. The compression ratio remained at 12.8:1 for all the turbocharged engines to allow for reliable cold starting. To aid initial running-in and to provide longer-term lubrication, the bodies were also treated with a graphite based coating. In order to be the correct shape to fit the cylinder when it is hot and running under load the piston body when cold is made tapered, barrelled and oval. This requires some sophisticated machinery and much development to obtain the correct shape. The piston also has to run satisfactorily at partial loads and at different temperatures.

The gudgeon pin is usually hollow to reduce the weight but it is subjected to bending and ovalisation as it is loaded at each end and reacted in the centre. This causes edge loading of the bearings in the connecting-rod and the bosses of the piston with increased local stresses. The way to avoid this was to increase the diameter of the pin, but this also increased the weight and required a larger small-end to the connecting-rod to accommodate it, again increasing the weight.

The piston, Figure 11, is required to seal the combustion gases from the oil mist laden crankcase with minimal leakage and to this end it is fitted with a number of sealing rings each of which has different duty. At the top there are compression rings to seal the hot gases. Lower down there are oil-control rings to prevent the oil from entering the combustion chamber whilst allowing the surface of the cylinder liner to retain just enough oil for lubrication but not enough to burn excessively. The early engines had four compression-rings and two oil-control rings but as ring technology improved the later engines featured only three compression rings and one oil-control ring. The rings were always made from cast iron with a variety of additions to the alloys and some were plated with special iron oxides, ceramics and chromium. Different cross sections and wall thicknesses were used to optimise the sealing performance, wear rates and costs.

Figure 11. An early RK270 piston with a four piece ring set and hard anodised crown to improve thermal fatigue resistance

The Cylinder Liner

This has always been a thin-walled cast iron tube suspended from a built-in flange at the top which is clamped to the crankcase by the cylinder head and located in a housing bore lower down within which it is free to expand. It is surrounded by cooling water on the outside which is sealed at the top by a thin steel gasket and at the lower end by two synthetic rubber “o” rings. The interface between the underside of the cylinder head and the top of the clamping flange is sealed from combustion gas leakage by a joint that has changed during the uprating process from wide thick copper to narrower thinner copper and eventually to thin soft steel. The running surface of the bore has never been plated, as in some designs, but has undergone much development to obtain the correct surface finish to reduce the initial running-in period and to provide the optimum conditions for oil retention to minimise wear. This surface was originally obtained by boring the liner with a round nosed single point tool but was subsequently achieved by boring with a ceramic tool and honing with diamond sticks to obtain a plateau finish.

The liner is suspended from one end, and as the piston moves sideways at the top and bottom of its stroke due to the obliquity of the connecting-rod, it is subjected to a transverse vibration in much the same way as a tubular bell. This vibration is damped to a degree by the lower location but it can still give rise to cavitation erosion in the water space which, in extreme cases, will eat through the liner wall. This was successfully addressed by raising the water pressure and using chemical anti-corrosion treatment in the RK engine thus avoiding the need to surface treat the outside of the liner. Figure 12 shows an RK 270 cylinder liner with a modified top joint face.

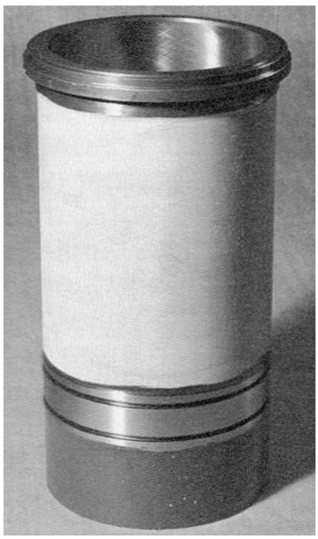


Figure 12. RK270 cylinder liner.

The Cylinder Head

This is a complex iron casting whose main function is to form a fixed end to the cylinder, the piston forming the moveable end. Individual heads are fitted to each cylinder and carry the inlet and exhaust valves and a centrally mounted fuel injector. Passages are cast into the cylinder head to transfer the charge air into the combustion chamber and carry the exhaust gases away from it through the camshaft-controlled valves. The cylinder head is clamped to the crankcase by a ring of studs and in doing so traps the top flange of the cylinder liner between the two components. The clamping system originally used only four studs but these were later changed to six and detail changes were made to the bottom end to reduce the stresses in the crankcase tappings. The casting also carries passages for the cooling water which is transferred from the crankcase and out through the head by transfer ports. It is a complex casting containing a myriad of holes, tappings, passages and internal supports to transfer the considerable firing load from the lower deck to the fixing studs and thence to the crankcase. This lower deck is in contact with the flame of the combustion gases and is known as the flame face. There is also an intermediate deck and a top deck to close the cooling space and to provide a platform for the valve springs and operating equipment.

The RK engine has always employed individual cylinder heads. These were initially of two deck construction and originally featured only one inlet and one exhaust valve per head. One of the first stages of uprating was to increase the air throughput. This was done by using two inlet and two exhaust valves per head (see figure 13) and turbocharging the induction air. The circular flame face, which is in the order of ¾ inch (19mm) thick and carries the firing load, is supported around its circumference and behaves like a diaphragm, deflecting upwards with each cycle. As the pressures were increased the intermediate deck was added to strengthen the structure by sharing the loads between the two decks via the port walls and other internal struts. The intermediate deck also enabled the cooling-water velocity over the flame face to be increased. The material was changed from grey cast iron to spheroidal graphite. The early cylinder heads carried a fifth valve whereby compressed air was introduced to push down the piston to start the engine. This was later removed to simplify the casting and reduce stress concentrations and the engines were then started by using electric or pneumatic motors, which drive the flywheel though a ring gear system.

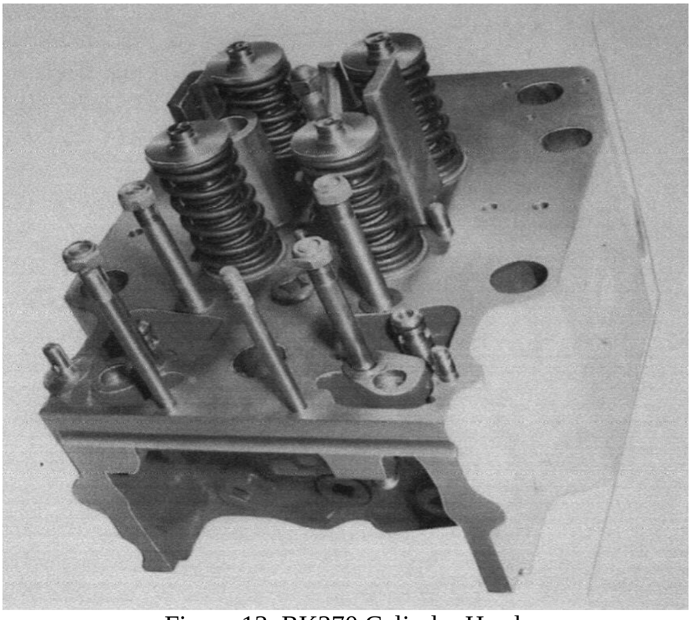


Figure 13. RK270 Cylinder Head.

The Valves

The inlet and exhaust valves were always conventional poppet type with angled sealing faces and were made from En 51 stainless steel with hard faced stellite 12 seats and chromium plated stems. Initially both inlet and exhaust valves featured 45 degree seat angles. This was later changed first to 30 degrees, then to 22½ degrees on the inlet valve in order to combat the seat wear caused by the increases in firing pressure. The exhaust valve angle remained although the material was changed to a nimonic alloy to suit the higher temperatures. The valves sealed against replaceable hard seat-inserts in the flame deck and the exhaust seats were eventually intensively cooled by the introduction of water passages within the seat. The valves were closed by coil springs and operated by camshaft driven push rods and rocker levers and bridge pieces.

The Camshafts

A single camshaft was used on the in-line engine and one for each bank on the vee engine. Each shaft carries three cams per cylinder, two of which operate the inlet and exhaust valves and the third in the centre operates the high pressure fuel injection pump. Uprated engines required larger fuel pumps which demanded a greater driving torque. The contact stress between the cam and the follower roller also increased. The camshafts are driven from the crankshaft at the flywheel end of the engine, originally by roller chain and eventually by a train of spur gears.

The camshafts have always been sectional. Each of these sections initially served two cylinders and was a simple steel bar onto which the valve cam was keyed. The fuel cam was split, clamped and dowelled for location. The shafts were connected by split, clamped couplings that also formed the bearing journals running in the tunnel housings in the crankcase. This sectional approach was originally taken to allow commonality of parts for different cylinder configurations, firing order and direction of rotation. However, the complexity of the clamping system and the presence of the many stress raising keyways and dowel holes in a highly loaded shaft was unsuitable for the greater loads required to deliver more fuel. Finally the camshaft was changed to a stiffer and stronger design whereby a single forging serving each cylinder carried the three cams and connecting flanges at each end. These sections were bolted together using the solid cylindrical bearing journals as couplings, sufficient holes being provided to enable different firing orders and cylinder configurations to be built without the damaging stress concentrations previously present. The gear drive system remained but the tooth width was increased as was the fuel cam face width and base circle diameter.

The Fuel Injection system

The high-pressure pumps were always individual plunger type whereby a simple cam lifts a plunger in a cylinder to pressurise the fuel to up to 1200 bar. This is then delivered through a thick walled pipe to a fuel injector fitted with a spring loaded needle valve controlled atomiser nozzle which is mounted in the centre of the flame deck of each cylinder head. The critical timing of the injection of this fuel in relation to the position of the piston is determined by the angular relationship of the fuel cam to the crankshaft, which is maintained by the gear drive train. This is fixed and therefore the timing is a compromise which cannot be optimum for all loads and speeds. The amount of fuel introduced into the cylinder is determined by the duration of the injection event that is controlled by a rack and pinion device within the pump which spills some of the fuel from the constant displacement pump chamber. The maximum amount of fuel delivered depends on the diameter of the plunger and the lift of the operating cam.

Although the concept of the injection pump and the mounting method did not change throughout the uprating, the pumping chamber was made larger and the fuel pressures were increased, requiring the changes to the camshafts previously mentioned. The rack control was actuated via a mechanical linkage from an engine driven hydraulic governor that matched the engine speed and load to suit external demands from the driven equipment. The final version of the fuel injection system uses a similar cam operated pump but instead of the mechanically operated rack control system, internal solenoid-operated spill valves are electronically controlled by a microprocessor. For the first time in the history of the engine the injection timing could be varied independently to suit different speeds and loads. With this came the promise of better specific fuel consumption and reduced exhaust gas emissions across the complete operating envelope. The system is now in use and under further development.

The Turbocharger

The original K engine was naturally aspirated, the amount of air available for combustion being limited by the volumetric flow efficiency of the inlet and exhaust gas porting and valve systems. In the 1940s, turbochargers, and later, charge air coolers were introduced. Initially up to four small turbochargers were used on the larger engines, one at each corner to keep overall height down. Later fewer larger turbochargers were fitted until the final version of the RK 270 engine featured only two units. These could be fitted at either end to suit the application. Over the last fifteen years the aerodynamic efficiencies of turbochargers has increased dramatically and the size and weight has fallen correspondingly for a given air mass flow. This has been a major factor in the ability to increase the power available from the engine.

The turbochargers have always been mounted on the engine and for the majority of the time they were single shaft machines whereby a centrifugal compressor is driven by an axial flow turbine. Radial and mixed flow machines were also used occasionally. The multiple exhaust pulses inherent in multi cylinder engine operation have been combined in different ways by the exhaust piping from the cylinders in order to smooth the flow into the turbine and improve the removal of the spent gases from the cylinder. The latest version of the engine uses a simple manifold whereby each cylinder exhausts into a tee shaped pipe that contains a convergent/divergent nozzle. The individual identical pipes are connected by flanged expansion bellows to form two straight pipes that run along and above the vee into the turbocharger. In some applications bank-to-bank balance pipes and waste-gate pressure relief valves are also fitted. This design makes for simple maintenance, which, in keeping with the tradition of the engine concept, uses the minimum number of different parts for its function.

Summary of Component Developments

Reference to the foregoing descriptions of some of the major component changes demonstrates that the considerable increase in power was achieved by the steady evolution of individual component parts rather than by dramatic concept changes to the complete machine. Although the designers were often highly innovative the Company would have been described as cautious, with a loyal customer base whose own businesses in turn demanded consistency, reliability and predictable through-life maintenance costs. Such customers would include the public utilities, the railway and fishing industries and the armed forces both at home and overseas.

This broad customer and application base provided the opportunity to develop specific features in a controlled manner using development partnerships with appropriate users. This enabled engines to be finally developed in the field under real working conditions whilst still earning revenue for the customer. The first stage of development always took place in the factory, but this was expensive and rarely replicated actual service experience. The benefits yielded by this two-stage process could then be passed on to others to become part of the engine pedigree.

Table 1. Significant stages in uprating of the RK engine

Year	**Type**	**Power per Cyl**	**Speed**	**bmep**	**Pmax**	**s.f.c**	**Piston speed**
		bhp	rpm	bar	bar	g/kWhr	m/s
1934	6K	50	600	4.8	?	220	6.1
1940	6K	82	685	7	?	228	7
1947	SV	100	750	7.7	?	228	7.6
1951	SV Mk2	125	750	9.6	76	225	7.6
1958	CSV Mk2	150	750	12	97	225	7.6
1969	RK Mk3	220	900	14	127	220	9.1
1977	RKC	300	1000	17.4	127	210	10.2
1980	RK 270	406	1000	21	152	210	10.2
1986	RK 270	437	1000	22.4	172	205	10.2
1996	RK 270	500	1000	25.6	172	205	10.2

Notwithstanding the above some of the applications were indeed revolutionary. A most notable example of this was the introduction of the sixteen-cylinder engine into the fast ferry industry in 1989. This eventually led to the creation of the twenty-cylinder RK270 engine. This in time honoured fashion was an entirely logical development of all that had gone before using as many of the same familiar and well proven components as possible. The six-cylinder shunting engine was about to mature into its ultimate form. Some of the main changes in the engine ratings and performance data are tabulated below. It is believed that the 1940 6K was mechanically supercharged, but thereafter they were all turbocharged, denoted by the prefix S. The prefix C denoted that the engine was also charge cooled. The prefixes were dropped when all the engines became the RK range.

The Fast Ferry era

In the late 1980s these vessels, designed and built in Australia, represented the latest in marine passenger-carrying technology. The vessels are fast and of low weight being built by welding relatively thin gauge aluminium alloys.

The vessels are unusual in that they are twin hulled, wave-piercing, catamarans with a third central hull which is normally clear of the water except in extreme sea conditions. The main hulls are sharply pointed at the bow in order to pierce the oncoming waves to ride through rather than over them. Each hull contains two engines driving individual water jets. The hulls are connected by a covered deck used to carry vehicles, above which is mounted a vibration free acoustically isolated passenger cabin with the bridge at the front. The bridge, as well as accommodating all the usual navigation equipment is also the engine control centre with all functions being remotely monitored and controlled from a simple desk. The engine rooms are usually unmanned. As well as being fast and of low weight, the vessels are much less expensive than a conventional ferry in both capital and operating costs, requiring a much smaller crew of both catering staff and professional mariners. The shallow draft of the low-drag, long slender hulls means that they can operate out of ports that would be denied to larger ferries. One of the major design criteria was to minimise the weight even to the extent of reducing material thickness by one or two millimetres and to leave much of the vessel unpainted where possible. Each ton of weight saved meant either extra speed could be enjoyed or another car could be carried for the same power A typical payload of the early vessels was 85 cars and 450 passengers and the ferries were particularly suited commercially to short journies with quick turn round times. Typical of these would be the English cross channel service between Dover and Calais where the 35 minute run each way would be made up to nine times per day.

The RK engine's first involvement with these vessels was in 1989 when engines were required to power the then latest and largest 74 metre version. The concept was to drive the water jets directly from the engine flywheel through shafting with neither clutch nor gearbox in order to minimise weight. Steering, reversing and stopping would be achieved by engaging thrust deflectors into the jet stream. The required engines had to be long and thin to suit the hull shape being mounted in line, astern, and slightly offset. They had to be light and fuel efficient to enable low fuel loads to be used. These were exactly the same requirements needed for main line locomotives and the 16 RK 270 engine running at 750 rpm fitted the bill perfectly. The engines were installed on six flexible mounting feet into the relatively flexible hulls which was where the rigid engine bedplate was to prove its worth. A flexible coupling was fitted between the engine and the solidly mounted driveshafts.

Large diesel engines are best suited to running at high loads for long periods, the starting and shut down being accompanied by gentle warm up and cooling down periods. For example in a power station an engine would run at 80% full load without stopping for up to six weeks at time.

The fast ferry operating regime meant that two of the four engines would be started in port and used to drive the vessel slowly out to the harbour mouth. Then the other two would be started and all four would immediately go to full load and speed until the destination was approached, at which point two of the engines would be shut down from full load and the other two used for berthing. These would then be shut down to avoid undermining the jetty or straining the mooring ropes which could occur if they were left running. Half an hour later the cycle would be repeated many times a day for the whole operating season. These frequent starts and hot shut downs put unusual demands on some of the engine components, the most affected of which are mentioned below.

Large end and main bearings

When a large diesel engine is started from rest the first few injections of fuel are rarely burned completely. This is often manifested by the presence of dark exhaust smoke for a few seconds after start up. Some of this unburnt fuel forms carbon which migrates down the cylinder walls past the piston rings and into the lubricating oil sump. This contaminates the oil and can form abrasive particles, which then cause premature wear of heavily loaded bearings. Although the chemical formulation of the oil and the normal filtration systems remove most of the particles the frequent stopping and starting continuously creates them and some remains. To deal with this, centrifugal oil filters to clean the oil, modifications to the piston crown lands above the top ring to reduce the creation of carbon, and bearings of different materials and construction to absorb the remainder, were developed.

The earlier bearings were made of either copper-lead or aluminium-tin alloys overlay plated with a softer lead-tin material. In the case of the aluminium the overlay was bonded to the substrate via a layer of nickel. The overlay was to absorb contaminants and would wear away exposing the nickel layer, which is a poor bearing material, in a sheet that might cause shaft seizure. This possibility was addressed by the use of a bearing type in which the nickel and overlay were deposited in grooves such that when wear occurred the nickel was exposed as narrow lines rather than the wider sheet.

Load governors

To minimise the formation of smoke upon start up and subsequent load application, special fuel control governing systems were put in place to provide the minimum amount of fuel to give a reliable start and to limit the amount of fuel introduced during load application and thus to match the air available from the turbochargers.

Exhaust Systems

The individual cast iron exhaust branches exiting from the cylinder heads are connected by bolted flanged joints to expansion bellows between each section. The flanges were integral with the castings and the frequent thermal cycling resulting from the many stops and starts caused fatigue cracks to form in the flanges with consequent exhaust gas leakage. This was addressed by changing the casting to stainless steel and making the integral flanges loose on the pipes to allow unrestrained thermal expansion and contraction to take place.

The Twenty-Cylinder Engine

Eventually the vessels grew larger and more power was required than could be provided by sixteeen cylinders. The direct drive to the water jets was replaced by a clutch and gearbox which enabled the engine speed to be increased. The other well-proven features of the engines and vessels were retained.

The increased power required was in the order of 25% and one answer to this was to increase the swept volume of the engine and to increase the maximum operating speed. To increase the stroke or cylinder bore further than the existing 270mm would have resulted in a completely different engine with many new components. The extra width resulting from such an engine would have made fitting into the hulls difficult, but as the larger ships featured longer hulls a longer engine could be more easily accommodated. The logical answer was to add another four cylinders of exactly the same type as those used by the proven sixteen-cylinder engine.

This was, surprisingly, a relatively easy route to follow as each individual cylinder firing causes the major stresses in the new crankcase, crankshaft and bedplate. Although adding more cylinders causes this to happen more often, the firing loads were limited to be no larger than those previously experienced on the sixteen-cylinder engines. The choice of firing order affects the shape of the crankshaft, torsional vibrations, main bearing oil films and the all important exhaust pulses. It is always a compromise but by using the appropriate firing order the peak firing loads were arranged so as not to overlap. This meant that the majority of load bearing components did not experience any greater loads. However, the crankshaft drive flange had to cope with a greater mean torque and was increased in diameter and the crankshaft and the camshafts experienced different torsional vibrations. These torsional vibrations were addressed by the use of commercially available, tuned-absorber type dampers rather than the simpler viscous fluid type used on the other engine configurations.

All the new features were verified by extensive use of finite element modelling of both specific parts and of global systems followed by 500 h of in house testing at various loads including 10% overload. The crankshaft integrity was assessed and approved by the various marine classification societies and in 1996 the twenty-cylinder engine was released for production.

The Blue Riband

This is a feature of the Hales trophy, a prestigious award offered in 1935 by Harold Keates Hales for the fastest crossing of the Atlantic Ocean and was to “serve as a stimulus to the craft of speed and mechanical perfection.”

Prior to the fast ferry attempt, the last winner was the American passenger liner United States which achieved a time of 3 days, 10 hours and 40 minutes in July, 1952. The magnificent Trophy, which is presented to and held by the owner of the winning vessel, is almost four feet high and made from 600 ounces of gilt silver. The Blue Riband itself is one of the many decorations embodied on the trophy and encircles the lower part of a globe held aloft by winged figures of victory. Originally competed for by ocean liners, it seemed unlikely that a car ferry would win this award on its delivery voyage, but one did just that.

On 23 June 1990 the fast ferry “Hoverspeed Great Britain” crossed the Atlantic in 3 days, 7 hours and 54 minutes, this being an average speed of 36 knots and took the award. Four sixteen-cylinder RK270 engines powered the vessel. As if such an achievement were not sufficient, in 1998 another larger fast ferry the “Catlink V”, powered by four twenty-cylinder RK 270 engines, beat this and took the award at an average speed of 41 knots.

The humble shunting engine had now become of age and won the Blue Riband award……twice!

Conclusion

It is hoped that the foregoing has demonstrated that the process of gradually changing components in a controlled and considered way can and indeed did lead to what, by any standards, is a remarkable achievement.

It took a long time but was built on sound principles by evolution and never losing sight of the pedigree. The original designers, armed with little more than their pencils, slide rules, log tables and determination would be astonished that their early conceptions could yield such an increase in power. If any of you are still around and able to read this, congratulations!

Notes on Contributor

The author is a professional mechanical engineer having obtained his degree in applied science from St. Andrews University in 1968. He joined English Electric Diesels as a graduate apprentice and has held various senior positions including those of Chief Designer of Ruston Diesels, and Chief Engineer of the Ruston Division of Alstom Engines, eventually retiring from the post of Chief Product Manager of MAN B&W Diesel Ltd., in 2006.
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