A REVIEW OF "IN-LINE" FUEL INJECTION PUMP DEVELOPMENT

By FRASER M. EVANS, M.B.E., B.Sc.

A Paper read before the C.A.V. Engineering Society on March 21st, 1961

FUEL INJECTION & ELECTRICAL EQUIPMENT
ACTON · LONDON · W.3

A REVIEW OF "IN-LINE" FUEL INJECTION PUMP DEVELOPMENT

By FRASER M. EVANS, M.B.E., B.Sc.

The term "in-line pump" is one which appears to have emerged in recent years without any formal definition, as a convenient means of referring to injection pumps which are not of the distributor type. For the purpose of this survey the term is taken to mean "multi-cylinder injection pumps in which a number of jerk-pump elements are arranged in-line along the axis of a common camshaft".

On this definition, the classic design was originated by the Robert Bosch Company in Stuttgart. As early as 1922, they, as electrical specialists, had the foresight to see that use of the high-speed diesel engine for road transport and other purposes was imminent and could imperil their magneto business. By the eventual development of a highly successful fuel injection system they not only secured a firm footing in the new field of the high-speed diesel, but greatly accelerated the advent of such engines.

The success of the Bosch injection system was largely dependent on three basic design features, none of which was originated by Bosch themselves.

The first of these features was the method of pump delivery control by means of a helically-grooved slide valve formed on the head of a rotatable pump plunger to co-operate with inlet and spill ports in the barrel wall. Designs of pump then existing were invariably complicated by the use of a separate mechanically-operated spill valve for control purposes.

A typical example is shown in Fig. 1 of a Bosch pump designed in 1923. This had a large steel block head having two lapped bores for each pumping line, the one containing a plain plunger and the other the spill and suction valve. The cam operated two parallel tappets through a rocker follower. The spill valve tappet was rotatable by a central control rack and had a face cam to vary the timing of the spill valve opening. Calibration was effected by an adjustable screw in the end of the spill valve tappet.

The now familiar helix-control method confined all the high pressure oil within two simple cylindrical parts which could be housed in a light alloy casing, and not only gave enormous simplification but also improved performance by minimising the volume of oil at high pressure. It has been claimed that the helix method was first used by Junkers during World War I for a low pressure petrol injection pump for a Zeppelin engine, but the Bosch development stems from a patent in 1926 by Fritz Lang of the Acro Company. About this time Bosch took over the Acro patents which also included the pintle nozzle and a design of pre-combustion chamber.

The second important feature of the Bosch injection system was the use of a closed injection nozzle with a differential needle valve, but this goes back at least as far as a patent by Ruston in 1909.

The third feature was the so-called Atlas type of

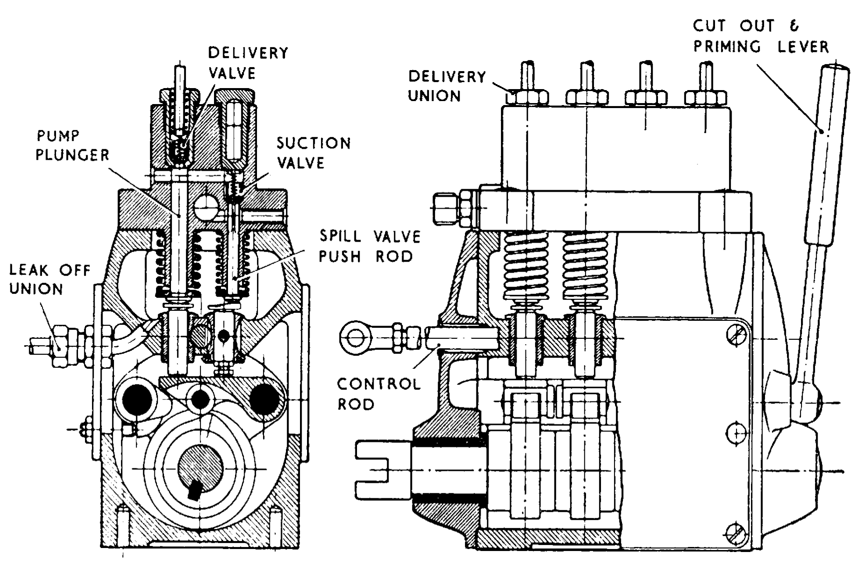


Fig. 1.—An early Bosch design.

unloading delivery valve which drops the pipe-line pressure at the end of injection to give rapid closure of the nozzle valve. This was invented in Sweden in 1924 and the patent rights were acquired by Bosch.

Bosch's own major contributions were threefold: in choosing the right elements to achieve a successful system; in meticulous attention to detail design and simplification; and perhaps most important of all, in the parallel development of production techniques to enable the equipment to be produced in quantity and economically.

By about 1929 Bosch were able to put a well-developed pump on the market together with injectors, governor and other ancillary items. This was the real beginning for the high-speed diesel engine, and engine developments followed rapidly. During the 1930's the Bosch equipment became virtually a world-wide standard, being manufactured not only in Germany but also under licensing agreements in Britain, France and the United States.

It was in 1931 that C.A.V. entered into an agreement, with Bosch, to manufacture fuel injection equipment in Britain. Then followed a period of intense activity during the build-up of production facilities and the training of personnel in the new techniques. The basis of a world-wide service organisation was also laid. On the engineering side the major function was Technical Sales work. The Drawing Office was largely engaged on translating German drawings into English. Until the British product and market became established there was no call for design development, but in 1936 a small development department was set up. I came as a new recruit to this department in 1937.

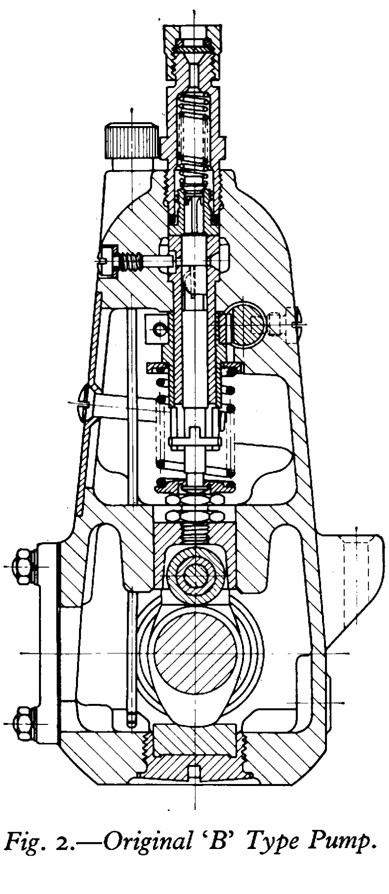


Fig. 2.—Original 'B' Type Pump.

The staple product at that time was the model 'B' in-line pump shown in Fig. 2. This is the model which made possible the successful development of diesel engines for public transport and haulage vehicles. A smaller model (the 'A') and two larger models ('Z' and 'C') were also manufactured, though in much smaller quantities.

The 'B' pump was of 10 mm. stroke and had a range of plunger diameters up to 10 mm. In its country of origin, engines were mainly of the ante-chamber type requiring moderate injection pressures of perhaps 250 atmospheres, whereas in Britain the more efficient direct injection engine was favoured because of the lack of indigenous fuels, and this meant higher pressures of perhaps 350 atmospheres together with higher injection rates and minimised high pressure volumes.

Designs for faster cams and reduced volume delivery valves were supplied by Stuttgart. In the interest of world-wide standardisation they kept a tight rein on design and Stuttgart approval was necessary for any change however small. Consequently for a time not much development was done by the new Diesel Engineering Department and the main work was of a trouble-shooting character. It was in fact housed at that time in an annexe off the main Service Department Workshop.

These were the years of the "Gathering Storm" and in 1938 it became known that the new German aircraft engines (by Junkers, Daimler-Benz, B.M.W. & Bramo) were fitted with petrol injection systems instead of carburettors. It was also known that Junkers had developed an aircraft diesel engine.

Interest in aero-engine injection systems was aroused in this country and we had a number of enquiries. For security reasons Bosch would give us no information at all, so at last there was one field in which we were free to get on with our own development.

Ricardo were experimenting with a high-speed two-stroke, aero-diesel cylinder having a sleeve-valve to control uniflow scavenging and high pressure supercharge. We supplied 'B' pumps with special cam forms and lightened tappets. The tappet roller proved to be a weak point under high-speed conditions and after trying all kinds of bush materials we found the best solution to be needle rollers.

After a year or so Ricardo converted the unit to petrol injection, in search of higher specific output, and produced a most remarkable engine. It was still a sleeve-valve, two-stroke with direct injection through the cylinder head but timed to occur when the piston was around bottom dead centre. The injected spray (from a pintle nozzle) traversed the full length of the cylinder to strike the piston crown and bounce back into the path of the swirling incoming air. This formed a rich-mixture skin which persisted during compression into a bulbous combustion chamber where it was ignited by twin spark plugs. This "stratified-charge" principle meant that the engine could be operated on an extremely wide load range (up to about 150 two-stroke b.m.e.p.) by changing only the fuel charge in similar fashion to a diesel. This eliminated any requirement for automatic mixture control and resulted in exceptional part-load economy.

In July 1939 two of us went to the Brussels Aero

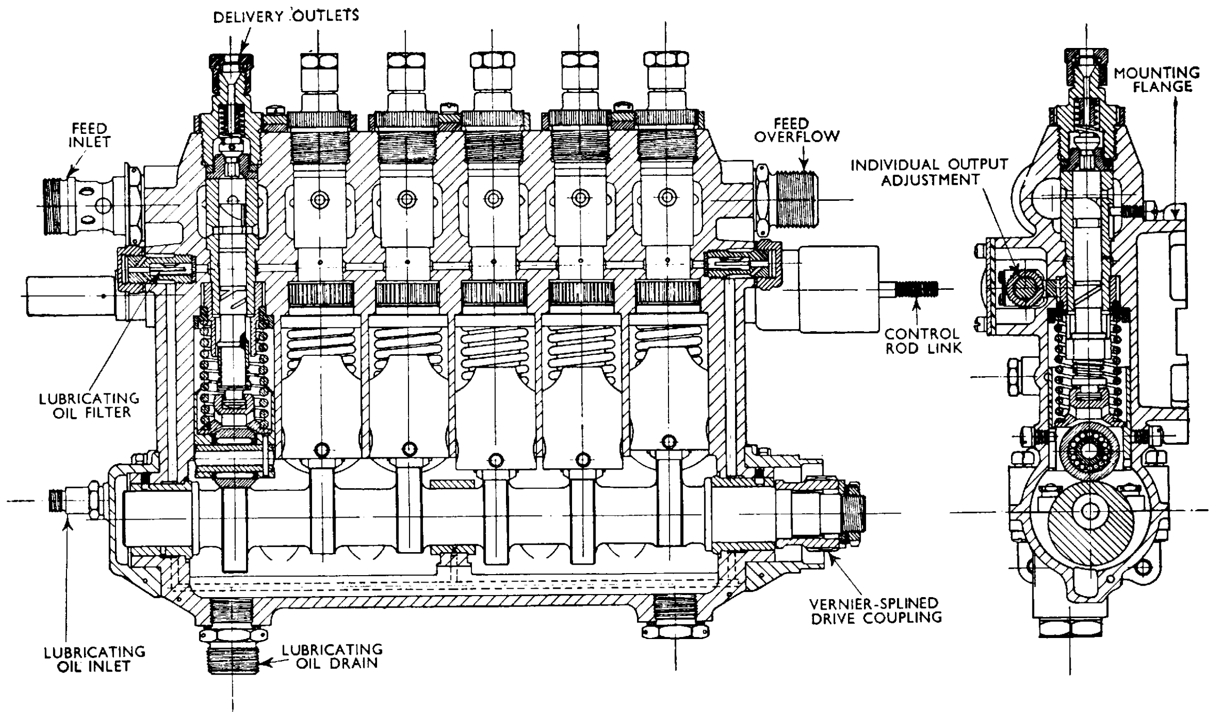


Fig. 3.—Petrol Injection Pump for Rolls-Royce "Crecy" Engine.

Show where for the very first time the new German aero-engines were being exhibited. No information was forthcoming but by inspection we were able to work out the main details of the petrol injection system used. A few weeks later the war started and the link with Bosch was broken.

Aero-engine development was by this time greatly accelerated and both Napier and Rolls-Royce took up further development of the Ricardo two-stroke. The Napier version was of short stroke and we supplied single cylinder pumps running up to over 3500 r.p.m., but their main effort was soon diverted to the "Sabre" 24-cylinder four-stroke engine.

Rolls-Royce persevered with a 12-cylinder Vee engine which became known as the "Crecy". This used two 6-cylinder pumps, one for each cylinder bank.

Fig. 3 shows the pump which we specially developed for the "Crecy" engine. Of the same 10 mm. stroke as the 'B' type, it could take larger diameter plungers (up to 12 mm.) yet weighed only 18 lb. as against 33 lb. for the standard 'B'. On the "Crecy" engine it was delivering 450 mm.^{3}/stroke at 3000 r.p.m. and with 350 atm. injection pressure. Noteworthy features were:—

Plain bearing camshaft hollowed out for lightness.

Forced feed lubrication to the camshaft bearings and to seal the pump elements against petrol leakage.

Spring-loaded dog clutch on each plunger to enable remaining plungers to be controlled if one seized.

Micrometer type calibration adjustment.

When the first sample was ready for test we were horrified to find that no test machine was powerful enough to drive it up to full speed and output. As an emergency measure we had to lash up a rig to drive the pump from a 250 h.p. electro-dynamometer newly installed in the Engine Test. This at least gave us the chance to measure the pump drive power, something we had never done before and rarely since.

A number of other petrol pumps were developed during the war, but none of them went into production because the piston-type aero-engine was rapidly overhauled by the jet type. Fortunately we had a foot in both camps, as the early development of fuel systems for the Whittle jet was done at C.A.V. and only transferred to Birmingham for security reasons when the fly-bombs arrived in 1944.

By the beginning of 1945 the end of the war was in sight, and it was decided that we would design a new diesel pump to replace the 'B' type. With our experience obtained on the aircraft pumps we believed we could develop a smaller but sturdier pump to do a similar job and also introduce a number of detail improvements. It was also decided to develop a completely new hydraulic governor to go with the new pump.

Before and during the development we consulted our principal customers for suggestions and criticisms. The smaller size was welcomed particularly by customers developing engines in the intermediate range around 1 litre per cylinder. At one time it was hoped that the new pump could replace the 'A' range as well as the 'B' but this just was not economically possible, particularly when during the post-war years small engines for agricultural tractors began to be produced in large quantities.

During the next three years, four main stages of design and development were passed through before we arrived at the model which was finally marketed as the type 'N' pump.

The first two stages of development are shown on

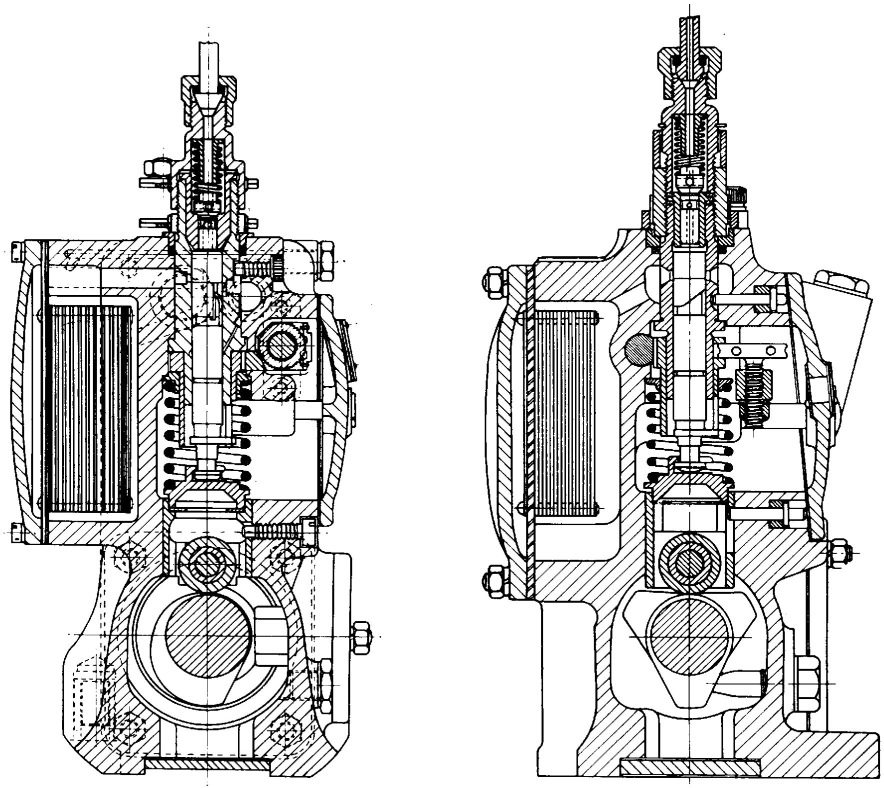


Fig. 4.—*'N' Pump development—Marks I and II.*

Fig. 4. The Mark I pump on the left used barrels in tension following a design we were currently developing for a petrol pump for the Bristol Centaurus engine. In the 'B' pump the barrel is inserted from the top and is clamped down against a shoulder by the delivery valve holder which screws into the aluminium housing. This puts the housing into tension and the barrel into compression under a load which has to be adequate to maintain the high pressure joints, making due allowance for differential expansion between housing and barrel over the working temperature range.

It seemed to be a good idea to reverse the procedure and insert the barrel from underneath, with a collar to contact the housing, and clamp it in position by a nut screwed on to the top end of the barrel. This put the barrel into tension and the housing into compression. It relieved the housing of the heavily stressed screw thread and enabled the housing scantlings to be considerably reduced and the element centres to be closed in from 40 mm. to 32 mm. Also the high pressure joints were completely divorced from the housing, the delivery valve holder screwing over the end of the barrel.

This arrangement as it turned out was splendid as regards stress, but failed lamentably as regards strain. The barrel, under tension, closed in diametrically and this was accentuated by the reaction at the collar and the slightest "out of squareness" of the nut face and threads. As the clearance between plunger and barrel is only 1 or 2 microns, a minute inwards distortion of the barrel wall was enough to give a tight control, and if run under this condition then seizure followed.

To get any reasonable running out of the pumps we had finally to lap the elements with the barrel clamped in position in a block simulating the housing, and control the clamping torque to a ridiculously low value which we could never hope to maintain in service.

The Mark I had micrometer type calibration adjustment similar to that of the "Crecy" pump. We were being pressed by certain customers to improve our calibration limits in the interests of engine performance and idling regularity. The micrometer adjustment could do nothing for variations of individual delivery curves over the speed range, which is a function of the plunger and barrel accuracy, but at any chosen condition it could allow the individual lines to be balanced within about 1 mm.^{3} quickly and without moving the control rod. It was very expensive however.

This pump also had divided suction and spill in an attempt to obtain smoother delivery curves free from odd bumps and kinks which, on the 'B' type, had been attributed to the spill wave from one element affecting the inlet of the next. Although over a dozen different arrangements of feed system were tried we could get no satisfactory result. There was a tendency for throttling at high speeds and at some conditions one or two elements would cut out altogether.

As a contribution to improving the service life of pump elements and delivery valves we embodied a built-in final filter contained in a cavity at the back of the pump housing. The element consisted of a pack of paper sheets interleaved with slotted cardboard spacers. In practice we found this construction unsatisfactory because the paper lifted in way of the spacer slots and caused by-passing. We know now that if by-passing had not occurred we would have had other trouble. The main filters in those days were of felt or cloth, comparatively inefficient, and an efficient final paper filter would have been the first to choke. Located inaccessibly in the back of the pump this would have been most inconvenient in service.

Pump stroke had been reduced from 10 to 9 mm. which was adequate for all applications then envisaged. This, together with a shorter plunger, allowed a reduction of camshaft diameter (because of the closer element centres), and the adoption of shim adjusted phasing gave an overall reduction of 36 mm. in pump height. Pump length had been reduced by 44 mm. in the 6-cylinder version.

The Mark II pump, Fig. 4 right, was following along so quickly that it still contained some of the snags found in Mark I, for example the paper filter and the tension element. This latter had been re-designed to save expensive barrel material but proved to be no better for distortion.

At the suggestion of A.E.C. Ltd., the old radial mounting base had been discarded in favour of a flat base, to give better rigidity, and the rear fixing bolt bosses extended right up past the filter to give good accessibility on the engine.

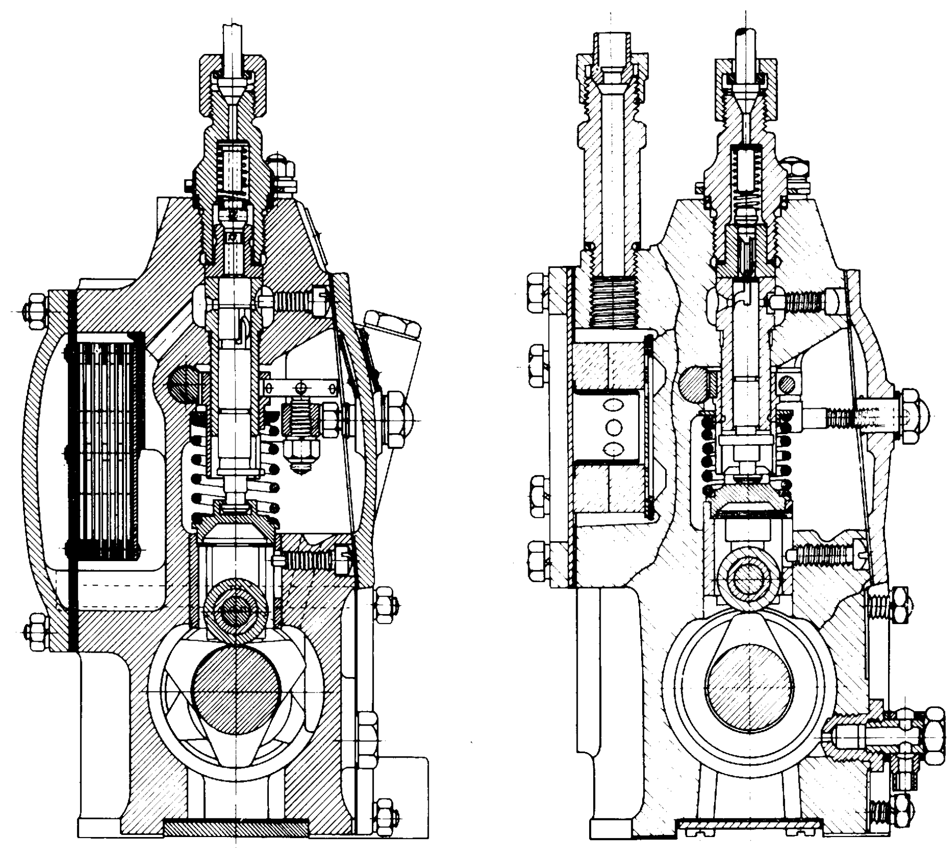
Mark II also embodied a cheaper design of micrometer adjustment. The control rod was moved back to the rear of the pump and was cut with helical teeth. A bar in the front of the pump carried a vertical screw with a large disc head which engaged a slot in the control sleeve so that the sleeve could be adjusted axially, the helical teeth converting this into a rotary motion.

Divided suction and spill had been discarded. Experiments on the 'B' type pump had by this time shown that irregularities in the speed delivery curve could be reduced to acceptable limits by a modified feed system using a self-regulating diaphragm type feed pump at about $4\frac{1}{2}$ lb/in^2 instead of the previous plunger feed pump with relief valve pressure control.

The inspection cover plate had been extended to cover the barrel and tappet locating pins as a second line of defence against leakage. Later, screws replaced these pins and removed the need for the special retaining plate necessary during pump testing.

The major problem of development was still distortion of the barrel in tension. All records and drawings of the Mark III pump appear to have been lost. I have the impression that it was a scheme only, reverting to wider element centres for the orthodox 'B' type compression elements and no samples were made.

Meanwhile we had done some static loading tests on a 'B' type housing specially machined with element centres of 32 mm. instead of the standard 40 mm. To our surprise and joy we found that although the alu-



MARK IV

N PUMP

Fig. 5.—'N' Pump development—Mark IV and final production design.

minium section between adjacent threads had come down from 20 to 12 mm. there had been no significant weakening. When tested to destruction by over-tightening the delivery valve holders, failure occurred at the same loading (about 900 lb. in.) and in the same manner, by cratering out from the bottom active thread, instead of by radial splitting which we had expected.

The way was therefore clear to employ orthodox barrels in compression without increasing the pump length. This was done in the Mark IV pump shown on the left of Fig. 5. A thicker but shorter barrel than 'B' was used, also an improved delivery valve joint arrangement. A new diaphragm feed pump was designed and, at the suggestion of A.E.C., provision was made for the drilling of a feed hole straight through the pump from feed pump face to filter chamber to simplify the external plumbing. One or two customers (but not A.E.C.) did eventually adopt this internal feed system, but it was never fully successful because it required the main filters to be on the suction side of the feed pump and it became difficult to track down air leaks.

Samples of the Mark IV gave much more satisfactory results and enabled us to get ahead with field proving trials and customer acceptance tests. With comparatively little change it went into production in 1948 as the 'N' type pump. It is shown on the right of Fig. 5 in its up-to-date form. The paper filter was finally discarded in favour of a felt pack type, and after a long struggle the micrometer adjustment was changed to the original

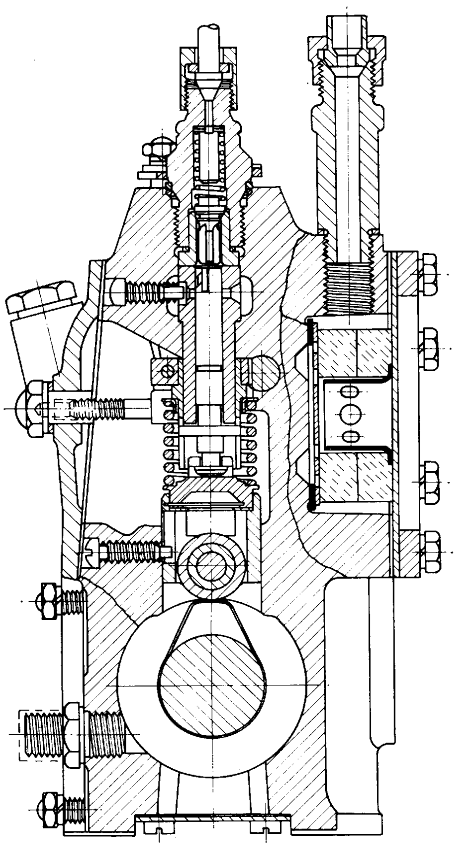


Fig. 6.—Model 'NN' uprated pump.

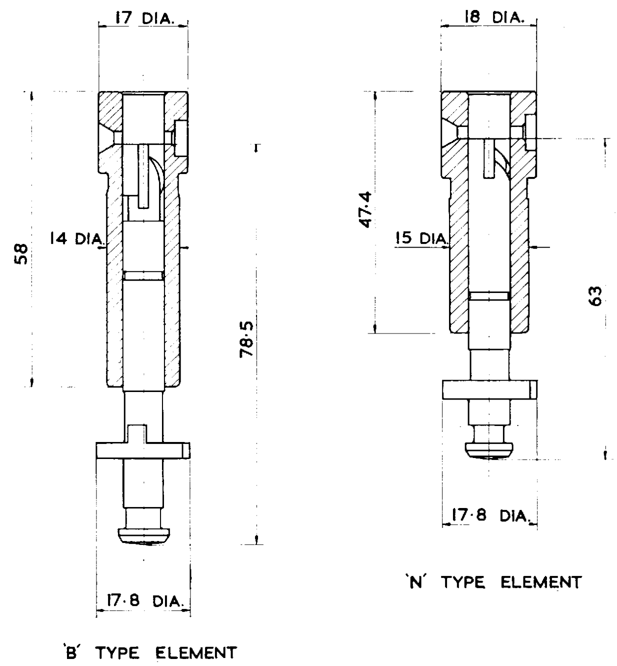


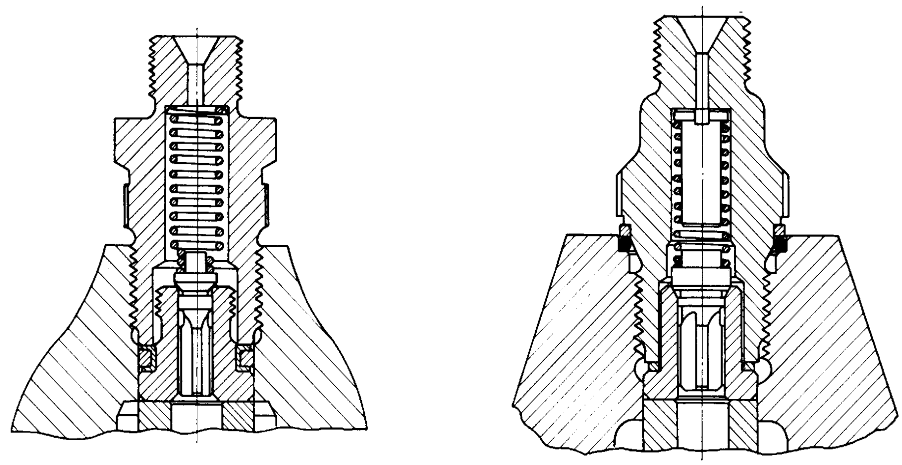
Fig. 7.—Comparison of Pumping Elements.

split quadrant type. The reasons for this change were threefold. The helical teeth were more prone to give control rod stiction unless the rod was very accurately guided against twist. The value of a more accurate means of adjustment was greatly reduced when it was shown by an exhaustive series of tests that nozzles alone could account for 5% variation of calibration when new, and with used nozzles this could rise to 11%. Also the micrometer type adjustment was more expensive.

It is one of the hazards of our sort of business that when a new model is introduced it often takes several years before the old one is completely superseded. Engine builders can be most vociferous in asking for improvements, but when a new model is presented some of them prefer that it should win its spurs on engines other than their own. The result is that we have to produce the two models for a considerable time, both in reduced quantities, which is uneconomic. The 'N' pump was no exception to this rule.

During the 1950's engine developments proceeded apace with specific outputs steadily increasing. Speeds crept up and supercharging came more to the fore. Every increase put greater demands on the 'N' pump in terms of speed, fuel output and maximum pressure. By about 1953 we were faced with conditions requiring 200 mm.^{3} delivery from 10 mm. diameter elements at pressures of around 550 atm. Under these conditions we ran into serious mechanical troubles with failures of camshaft bearings, cams and rollers.

To meet such conditions we designed an uprated version of the pump which became known as the 'NN'. As shown in Fig. 6 it employs the same carcase of the 'N', but has a stiffer camshaft of increased base circle, and lift restored to 10 mm. instead of 9 mm. This was accommodated by reducing the length of the tappet. Much bigger camshaft races are fitted in an improved mounting. Also we have gone back to the use of lubri-



ORIGINAL 'B' TYPE　　　'N' TYPE

Fig. 8.—*Comparison of Delivery Valve Assemblies.*

cating oil in the cam chamber instead of relying on fuel lubrication. For the highest ratings this may be supplied on a re-circulatory system from the engine.

Having traced somewhat sketchily the historical development of the 'NN' pump from its predecessors it is now intended to look a little more closely at detail development of the more vital components.

First the pumping element itself, consisting of barrel and plunger. We compare in Fig. 7 the original 'B' type element and the 'N' (and 'NN') type. Change is largely confined to proportions, the 'N' type being shorter but with thicker barrel wall. It requires 15% less bar material. The increase in thickness is only 0.5 mm. all round but for the same stress allows 9% higher injection pressure. The main source of weakness is the presence of ports in the barrel wall which, in a notch-sensitive material, causes a stress-raiser of around 3 to 1. Increase of barrel thickness gives diminishing returns. If we could make it infinitely thick it would only allow us to double the working pressure.

With our present En31 type material the limiting port-factored working stress is around 25 tons/in^2 which means that we can expect to start cracking 10 mm. 'NN' type barrels at injection pressures of 650 atm. The best hope for reaching high pressures is the use of a less notch-sensitive material such as Nitralloy En40C and on some of the larger pumps this is being tried.

The shorter barrel and plunger did not result in greater leakage down the shank because there was less bore distortion during hardening, and production was therefore able to achieve a more consistent clearance.

Apart from length, the main change to the plunger has been a simplified helix arrangement consisting of two slots (one axial, one helical) ground straight into the hardened part. Soft-stage milling is eliminated and a better control of clearance is achieved in way of the helix, particularly at the idling position. There is also a valuable reduction of high pressure volume.

In the 'B' type delivery valve assembly (Fig. 8) the joint ring has a double duty to perform. It makes a high pressure seal on its faces and also has to expand radially under compression, making a low pressure seal against the housing bore and preventing leakage from the feed gallery up the holder threads. The original joint ring shown was made of fibre with a copper sheath. This joint gave reasonable results for many years but in the post-war period when pump duty was on the increase we ran into a spate of troubles with joint leakage. This was particularly serious on one or two engines where the exhaust manifold was uncomfortably near to the pump. The copper-fibre joint did not have enough recovery to consistently withstand the load-cycling induced by thermal expansion of the housing. A large scale investigation was undertaken in which many joint materials and modified designs were tested.

Finally, for the 'B' pump a compromise solution was adopted, the joint ring being made in laminated fabric-filled bakelite without the copper sheath. This successfully saw out the remaining life of the 'B' pump and was also adopted for the 'A' pump and its uprated version the 'AA'. As a further improvement and cheapening measure a moulded nylon washer is about to be introduced for the 'AA'.

On the 'N' pump, from Mark IV onward we adopted the design shown in Fig. 8 in which the duties of H.P. and L.P. sealing are performed by separate joints. The H.P. joint started as a plain copper washer but was prone to work hardening, inadequate resilience and sometimes bursting. A change to brass afforded some improvement but finally we went to fully annealed mild steel which has proved most reliable. The L.P. joint is made by a rubber 'O' ring at the top end.

Regarding the functional parts, the tendency has been to minimise the high pressure volume as far as possible (by elimination of extractor thread and filling the interior of the spring with a peg) in order to reduce the hydraulic springiness of the system and so obtain better control of the actual injection at the nozzle. Valve diameter has tended to increase, particularly for higher duties and speeds. We now commonly use 7 mm. instead of the original 5 and 6 mm. dias. This results in smaller lift for a given flow area and unloading volume and requires a shorter spring.

Delivery valve design provides a very useful means of

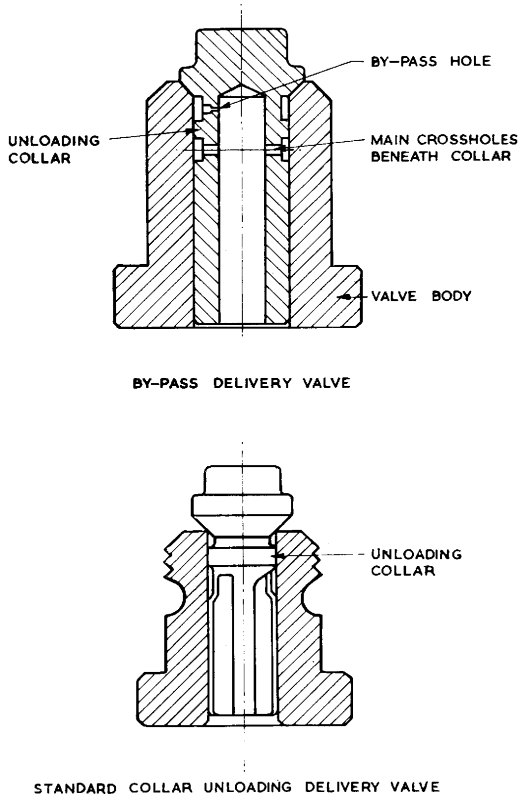


Fig. 9.—Comparison of Standard and By-Pass Delivery Valves.

tailoring the pump speed-delivery curve to suit engine requirements. The delivery valve unloading volume subtracts from the net output of the pump so that, if we can arrange that the effective unloading increases with speed, the pump output will drop with speed, which is often desirable.

Bosch originated this line of development with their so-called A.S.D. valves in which the flutes ran out to a fine point which defined the minimum unloading and by adjusting the spring pressure it was possible to trim the delivery curve. Results were never very consistent. An improved C.A.V. type is the By-Pass Valve (Fig. 9). This is hollow instead of fluted and the central bore feeds the main flow-holes beneath the unloading collar and also a small, accurately-drilled, by-pass hole above the collar.

At very low speeds the by-pass hole is able to pass the injected flow, so valve lift is small and the collar does not lift out the seat. Unloading is virtually nil. At high speed the by-pass hole is ineffective and the valve lifts its full normal amount so that the main flow-holes become operative and full unloading occurs. At intermediate speeds there is a progressive variation.

This subject of Control of Engine Torque Curves is described in great detail in an article by D. G. Burton which appeared in C.A.V. Engineering Review, Vol. 1, No. 6, May, 1958.

Fig. 10 compares the original 'B' type tappet assembly with the modern 'NN' type. The screw and locknut type of phasing adjustment resulted in highly loaded threads, particularly in the lock nut, and was liable to slacken or be maladjusted in service. The shim type adjustment gives a positive setting, readily made in the

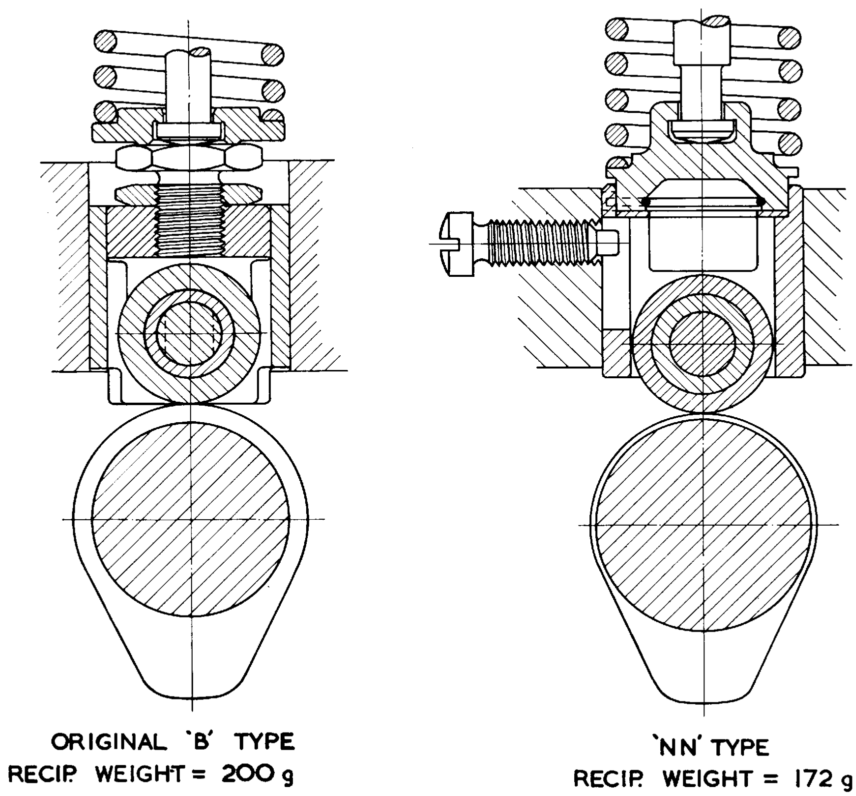


Fig. 10.—Comparison of Tappet Assemblies.

factory but not likely to be tampered with afterwards. It also results in a useful reduction of pump height.

The 'B' type tappet location was by flats on the ends of the roller pin which engaged slots cut in the housing bore. The part most liable to wear was the aluminium housing, and this involved a very expensive service replacement cost. Furthermore the width across the housing slots constituted a limit to the minimum possible centre distance between pumping lines.

The 'NN' type tappet is located by a dowel pin screwing into the front of the housing and engaging a longitudinal slot in the tappet body. The expendable part in service is usually the pin which is very cheap. This construction was one of the factors allowing the element centres to be reduced from 40 to 32 mm.

Total reciprocating weight has been reduced by 14% which gives a useful improvement in speed limit.

The roller, bush and pin have a very arduous duty to perform. In 1940 we discovered that for high-speed, two-stroke conditions (speeds of 2000 r.p.m. upwards) we could not get away with a plain bush. Many materials, fits and finishes were tried but success was only achieved by using needle rollers. On the other hand for four-stroke conditions, at pump speeds up to perhaps 1300 r.p.m., we have not been able to improve on the original hardened steel floating bush, although to meet modern loadings we have had to tighten up on the geometrical accuracy of the parts and limit the surface finish to about 8 micro-inch. We have also had to return to lubricating oil in the cam chamber, maintained at a higher level than before. Even so, lubrication is a chancy business being dependent on oil thrown off by the cam getting on to the pin and being drawn into the clearance. It is highly probable that any further uprating will require forced feed lubrication.

We have designed many different cams at one time and another. Until about ten years ago they were all constructed from circular arcs and tangents, probably for no better reason than the fact that a draughtsman's tools are a straight-edge and compasses. It confers no particular benefit to production because cams are produced by copy-grinding from a master cam, and the master is usually produced by milling from a table of lift figures at small angular increments, the minute remaining cusps being blended away.

On a cursory consideration it is often thought that the cam pushes the plunger up and the spring pushes it down again, but under running conditions this is not the full story. At bottom of stroke the plunger is at rest, and again at top of stroke. It follows that in order to get from bottom to top the plunger must be first accelerated and then decelerated. During the accelerating phase the cam is dominant, and during the decelerating phase the spring is dominant. The converse happens on the down-stroke. In other words the spring is only really required over the latter part of the up-stroke and the early part of the down-stroke. It would be quite possible to operate a pump in which the spring was only effective over the cam nose period, but this would involve impact and be very noisy.

From the pumping aspect, the plunger stroke is divided into three phases. First a period during which filling is completed, the ports are being closed and the plunger accelerated up to a velocity suitable to give a sharp first pressure wave. This is followed by the delivery period which is preferably performed with rising velocity, i.e. positive acceleration. Finally there is the spill period when pressure collapses and the plunger is brought to rest.

We thus require the first two phases of pumping to

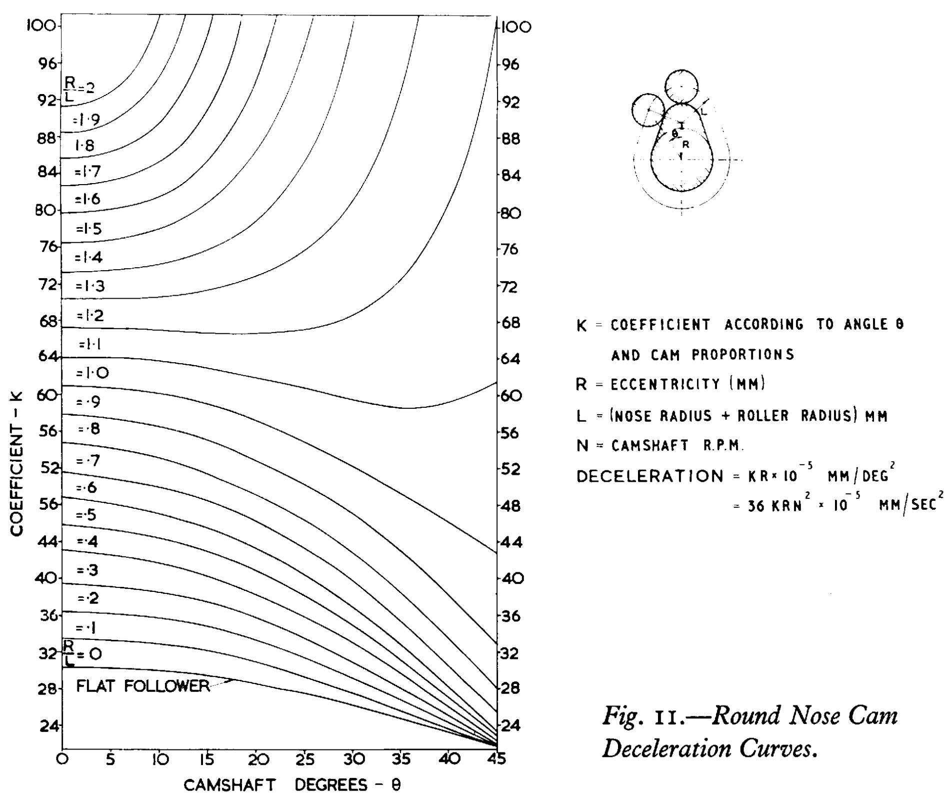


Fig. 11.—Round Nose Cam Deceleration Curves.

coincide with the accelerating phase of the cam lift, and only the spill phase to encroach on to the decelerating phase of the cam.

Re-entrant profiles are in general taboo from the production aspect because they limit the size of grinding wheel which can be used, and a large wheel is preferred to obtain rapid production, long wheel life and good surface finish.

This means that the highest rate acceleration phase which we can use is a tangent straight off the base circle of the cam and most cams for four-stroke applications are now made this way.

In order to increase the usable output of the pump the accelerating phase needs to be extended as far as possible which for a given lift means a reduction of the decelerating or nose phase. With a circular arc nose this brings about a double difficulty in regard to bouncing speeds.

Fig. 11 illustrates what happens to the deceleration curve shape as the round nose radius is progressively reduced. On a base of camshaft degrees before top of stroke we have plotted a deceleration coefficient K which when multiplied by a constant, the r.p.m.^{2} and the eccentricity of the nose radius gives the deceleration in mm./sec.^{2}

Over the nose phase, the motion is precisely analogous to that of a con-rod/crank mechanism of which the crank radius R is equal to the eccentricity of the nose radius from the cam centre, and the con-rod length L is equal to the sum of nose radius and the roller radius.

The ratio R/L determines the shape of the deceleration curve. In engines the crank/con-rod ratio is commonly between .2 and .3 giving a curve which peaks at top of stroke. This sort of shape continues up to $R/L=1$ which is about the ratio used in some of our two-stroke cams. Beyond $R/L=1$, the curves start to turn up progressively until at ratios such as we use in four-stroke cams of between about 1.4 and 2 the initial deceleration is higher than that at top of stroke. Unfortunately the spring force which has to provide this deceleration is highest at top of stroke. We thus pay a double penalty in regard to bouncing speed limitation.

We therefore decided to investigate the optimum form of nose profile to give highest bouncing speed. A typical case in point was the cam for the 'NN' pump, the original form being shown at Fig. 12 and consisting of a tangent flank running into a 4 mm. nose radius. This gave a transition point occurring at nearly 6 mm. of stroke. Considering the lift curve we wondered what would happen if at a given speed, having reached the transition point, the cam suddenly disappeared. Obviously the moving mass having reached a certain velocity and being opposed only by the spring would describe a trajectory which formed part of a simple harmonic motion having an origin at the free length of the spring and an amplitude dependent on the velocity at the point of release. Thus at three different speeds (1200, 1350 and 1500 r.p.m.) the trajectories would be as shown. At 1350 r.p.m. the trajectory just reaches the required 10 mm. stroke and this is the highest speed which can be achieved with the given spring and mass conditions. To achieve it we have to design the cam profile to just fit the 1350 r.p.m. trajectory, so that at this speed the roller floats round the nose. The form of curve required is a sine curve plotted in polar co-ordinates of form as shown. The immediate result is that the bouncing speed has been increased by 13% from 1195 to 1350 r.p.m.

The sine nose also gives an important bonus in regard to cam contact stresses. Geometrically, the contact between two cylinders is a line. But a line has no area and if loaded the stress would be infinite. In actuality, when a load is applied the cylinders become flattened, minutely but calculably, to give area contact and a finite stress is carried which, other things being equal, varies as the square root of the sum of the curvatures. So far as possible we avoid allowing full injection pressure conditions to encroach on to the nose because of the high stress conditions which arise, but in the search for more and more output this is not always possible. For instance, measurements have shown that in some cases the pressure on the plunger does not start to fall until nearly 1 mm. after the spill port has started to open.

We know from experience that the contact stress at which breakdown of the surface occurs is around

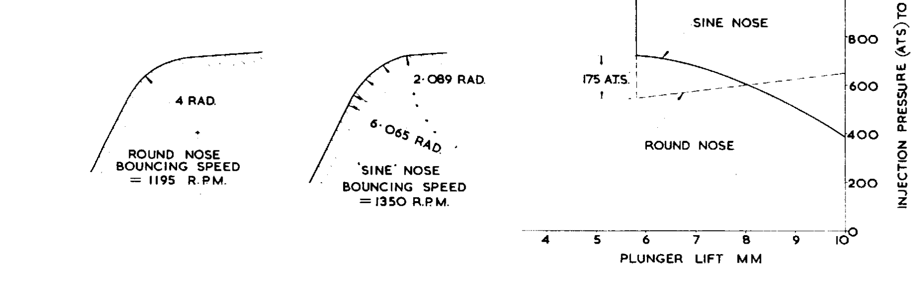


Fig. 12.—The Sine Nose Cam.

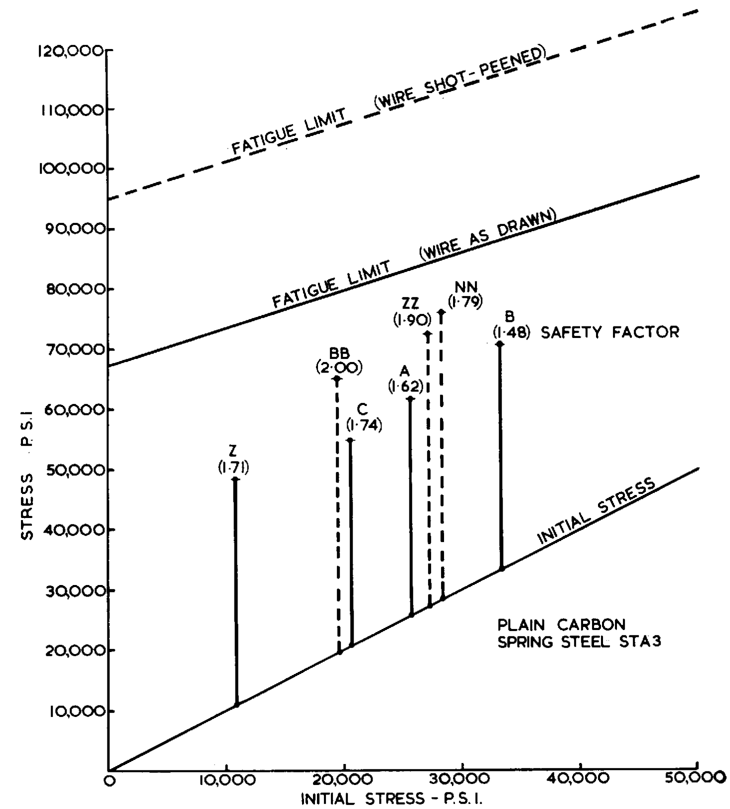


Fig. 13.—*Plunger Spring Stresses.*

150 tons/in^2, in fact for safety we try to keep to a maximum of 135 tons/in^2. The sine nose has less curvature at the transition point and more curvature at end of stroke than the equivalent round nose, as indicated by the instantaneous radii shown on the diagram.

On the right of Fig. 12 we have plotted for the case of a 10 mm. plunger the injection pressure to give 150 tons/in^2 contact stress. At the transition point the sine nose allows 175 atm. more pressure than the round nose. If injection was continued beyond about 8 mm. of stroke we would start to lose out, but we never go that far.

Plunger spring failures have given us quite a bit of trouble from time to time. They are highly stressed components and being subjected to a continually varying stress they fail by fatigue.

For fatigue conditions we have to consider not only the maximum stress imposed but also the range of operating stress.

This is best done by reference to a Goodman diagram as shown at Fig. 13 for our normal STA3 material. On a base of initial stress there is plotted the initial stress line and the fatigue limit line. Then the working stress range of any given spring can be represented by a vertical line and the safety factor can be obtained, being the ratio of the fatigue range to the actual range, at a given initial stress value.

The sloping full line is the fatigue limit of the material as drawn, and the full vertical lines represent the original springs in the B, A, C and Z type pumps, having safety factors of 1.48, 1.62, 1.74 and 1.71 respectively. At a time when we were experiencing an epidemic of 'B' spring failures a statistical investigation was made of the service failure figures and various endurance test results and it was recommended that a minimum safety factor of 1.75 should be worked to.

To achieve this we adopted the practice of shot-peening all plunger springs and this proved very successful. Zimmerli and others had shown the benefits of shot-peening in lifting the fatigue limit to somewhere about the upper dotted line. The peening is effective by ironing out minor surface imperfections, work hardening the skin and leaving locked-up compressive stresses which oppose the principal working stresses. On later pumps, as shown by the vertical dotted lines, we have designed springs for shot-peening and obtained wider stress ranges but still with safety factors greater than 1.75.

Plunger springs are not immune to the phenomenon of surge which besets engine valve springs and on which reams have been written without a positive cure-all being found. Every spring has its own natural frequency dependent on its mass and resilience. This frequency is invariably many times greater than the frequency of reciprocation in the pump. However as Fourier showed, any periodic relationship (such as the cam lift curve) can be represented as the summation of an infinite series of sine and cosine terms having frequencies which are integer multiples of the fundamental. For symmetrical curves the sine terms disappear leaving only the cosines. Spring surge can be stimulated at any speed at which one of these so-called harmonics comes into resonance with the natural frequency of the spring.

Over 20 years ago we had a number of failures on 'Z' pump springs on a particular application, and amongst other things we investigated surge by the crude method of soldering a gramophone needle to the centre coil of a

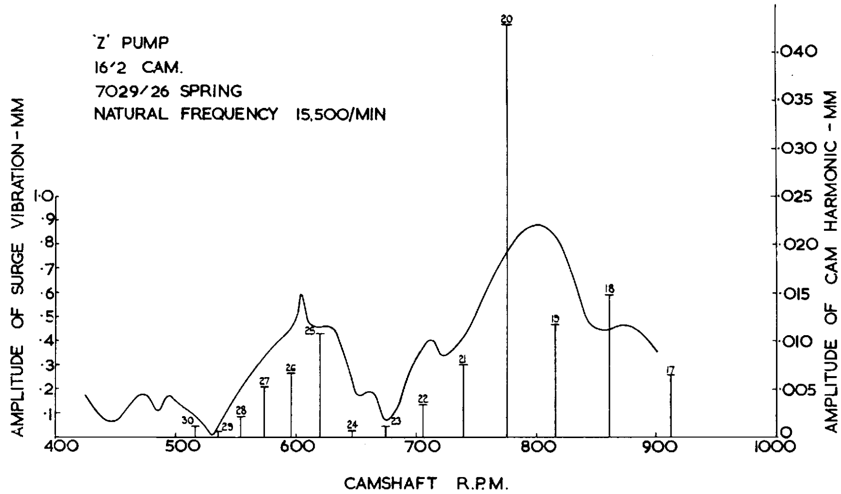


Fig. 14.—Surge in 'Z' Pump Plunger Springs.

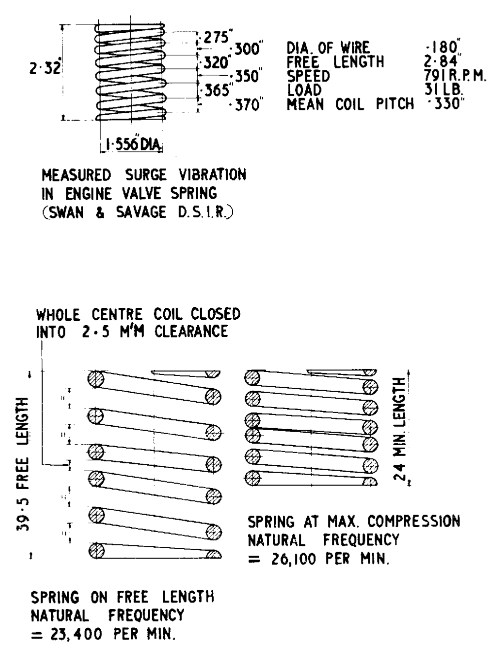
spring and traversing a guided smoked-glass plate across it whilst the pump was running at various speeds. From the oscillograms so obtained we were able to measure the amplitude of centre coil vibration and this was plotted against speed as shown by the wavy line of Fig. 14. We also did a laborious 144-ordinate Fourier analysis of the cam lift curve and plotted the amplitudes of the various harmonics against the speeds at which they would synchronise with the natural frequency of the spring. As will be seen, there is a rough correspondence which appears to be more than coincidental. However, we didn't know what on earth to do about it and were very relieved when it turned out that most of the failures were on marine applications and an eventual supply of stainless steel springs appeared to cure the trouble.

The valve gear experts usually recommend that the natural frequency of the spring should be sufficiently high that cam harmonics lower than the 11th do not become resonant within the working speed range. On the 'NN' pump we have pushed this up to the 18th harmonic but there is still some evidence of measurable surge occurring.

We have recently been looking into the question of surge stresses a bit more closely. First of all there is a 'Velocity stress' which is applied to the spring, over and above the normal deflection stresses, whenever a velocity is applied to one end. It can be shown that this stress is completely independent of the dimensions of the spring. This is well known, for instance, in gun springs and is the limiting factor in recoil velocities. For round wire steel springs the relationship is:

$$\text{Velocity Stress (kg/cm}^2\text{)} = 360\ \text{V (metres/sec.)}$$

In injection pumps the plunger velocity will rarely ezceed about 3 metres/sec. which gives a maximum Velocity Stress of about 15,000 lb/in^2. As long as this is damped out rapidly it does no harm because the velocity at both ends of stroke is zero so that although



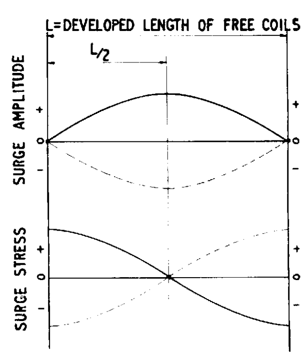


Fig. 15.—Surge of Plunger Springs.

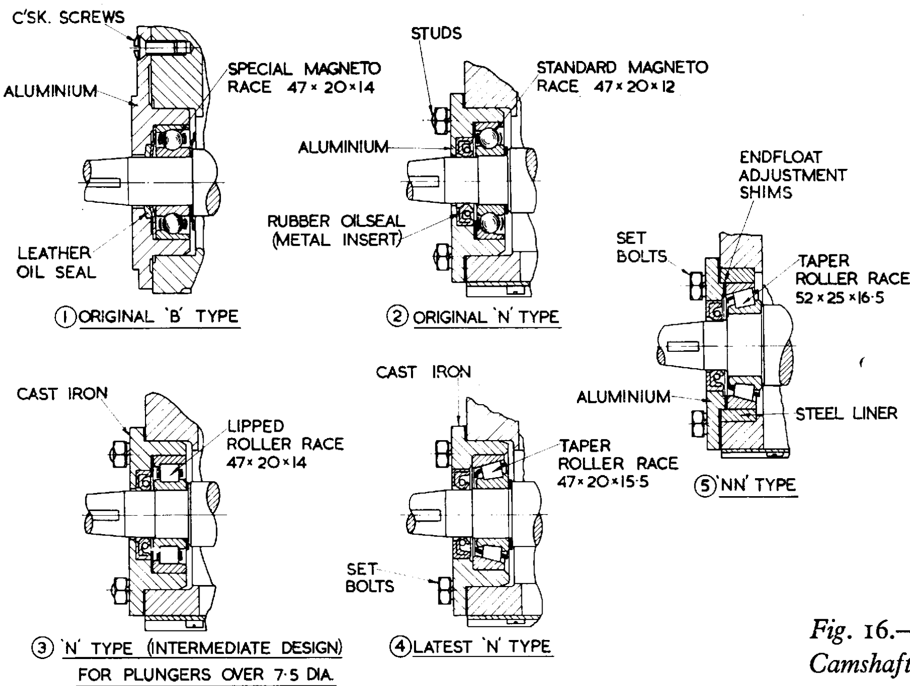


Fig. 16.—Development of Camshaft End Bearings.

the stress-time curve is distorted the stress range is not increased.

However, when harmonic resonance occurs the velocity stress develops by forced vibration and, by reflections from both ends, builds up into a standing wave.

If, as shown on the right of Fig. 15, we plot the amplitude of surge vibration of all points on a spring against the developed length of the spring wire we obtain a sine curve, and its negative reflection half a period later, shown dotted. Amplitude is zero at the ends and greatest in the middle.

The corresponding surge stress is proportional to the slope of the amplitude curve and is plotted below. This shows that stress is greatest at the ends (which accounts for springs usually failing there) but is zero in the middle. This interesting result is confirmed by figures given by Swan & Savage of measured coil pitch changes under surge conditions. The mean pitch without surge was 0.330 in. When surging the mean pitch of the two centre coils was 0.335 in., i.e. within experimental error the surge stress was zero at the centre.

It has frequently been proposed to wind the spring with one or two of the end coils closed into a smaller pitch so that they close down solid at part lift and so change the natural frequency to discourage resonance. As the clashing of coils can cause surface damage and cancel out any benefits obtained by shot-peening it would appear reasonable to arrange the closed-in coil at the centre of the spring, where surge stresses are zero, rather than at the ends. We have had some springs made like this to the design shown on the left of Fig. 15.

Fig. 16 illustrates the progressive increase of camshaft bearing capacity and mounting improvements. The original 'B' pump had an aluminium endplate, retained by countersunk screws. It contained a leather oil seal and carried a non-standard "magneto type" ball-race.

On the first 'N' type design we still had an aluminium endplate, but of reduced flange size and retained by studs and nuts to overcome locking and thread stripping troubles experienced with the countersunk screws. A metal-insert rubber oil seal was used and to improve seal life the shaft diameter in way of the oil seal was ground 0.15 mm. less than the bearing fitting diameter so as to prevent scoring when pressing on the inner race. A standard magneto ball-race was used having the same load capacity as the 'B' type although 2 mm. shorter.

By 1952 it was clear that for the higher duty applications (taken as 8, 9 and 10 mm. plungers) bearing life was inadequate. Trouble had also been experienced by the outer race turning in the aluminium endplate and then causing rapid wear. We therefore changed the endplate material to cast iron and introduced roller bearings having single-lipped outer races.

This change was only a palliative. The roller race, although of greater load capacity than the ball-race, was incapable of resisting the thrust loads imposed by drive or governor. Also the cast iron endplate could only be used at both ends with a pneumatic governor. With mechanical or hydraulic governors the bearing was carried at one end in the governor housing which was still of aluminium.

The next step was a further change of bearing to the taper-roller type which has much better endthrust capacity. This required tighter limits of control of camshaft endfloat during assembly.

For the 'NN' pump an even larger taper-roller race was used with the stiffened camshaft, in a new mounting which arose out of a salvage suggestion received from Canada. The outer race is mounted direct in a steel sleeve which is pressed permanently into the housing and finish-bored in position. This allows a return to an aluminium endplate (and governor housing) which has little more to do than carry the endthrust and support the oil seal. Endfloat adjustment is effected by shims between outer race and endplate instead of on the camshaft.

The use of plain bearings for the camshaft ends

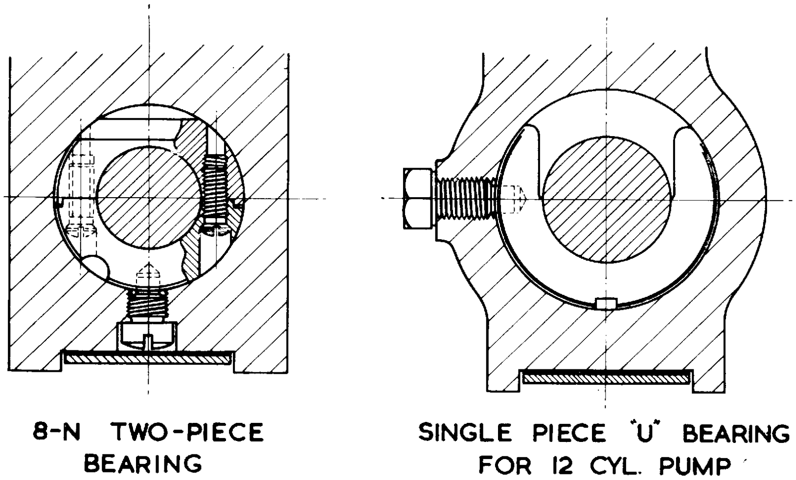


Fig. 17.—*Camshaft Intermediate Bearings.*

deserves more attention than it has received, particularly as we have now reverted to lubricating oil in the cam chamber. At a time when ball bearings were in short supply we endurance tested an 'A' pump with plain bearings for 7000 hours quite successfully. It could allow larger attachment diameters for coupling and governor regulator which in turn might allow the use of cast camshafts to be reconsidered.

On all pumps for more than six cylinders it is the practice to fit one or more intermediate camshaft bearings. Thus on 8-cylinder pumps we fit one intermediate bearing at the centre, and on a 12-cylinder pump recently designed we fit two. These bearings are required to prevent excessive bending of the camshaft under the injection loads. Apart from possibly overstressing the shaft, excessive deflections cause failure of the end bearings and the cam rollers by mis-alignment.

Plain bearings are used at the intermediate positions and as fitted to 8-cylinder B, N and Z pumps (shown on left of Fig. 17) consisted of a thick bush made in two halves, accurately dowelled and screwed together around a journal on the shaft. After assembly into the pump, the bearing is located by a dowel screw. This was an expensive construction and we have recently developed a cheaper one-piece "U-bearing" as shown on the right of Fig. 17. This provides bearing surface only in the direction of loading and unobstructed ingress of oil to the low pressure side.

In parallel with the main line of pump development we have from time to time had to divert on to special features to suit particular requirements. Three of these deserve brief mention.

Starting in 1949, and largely at the request of L.T.E., we produced a number of pumps embodying a system of Pilot Injection with the object of reducing engine noise in passenger service vehicles. The system involved considerable complications both of design and production techniques with consequent increased cost. Long term service results eventually showed that there was a further penalty in terms of increased fuel consumption such that the undoubted quietening could not be economically justified.

On railway applications extraordinary measures are taken in regard to safety. In particular a second line of defence is required in case the normal speed governor should fail or the controls become seized. This led to the development of overspeed trip mechanisms which can completely cut off the fuel supply to the engine within one pump revolution by means of spragging pins. These engage holes or slots in the tappets when they reach top of stroke and thus prevent them returning. This may be done mechanically or (as shown at R.H. view of Fig. 22) hydraulically on receipt of a signal from a speed-sensing device on the engine.

It has been demonstrated that with comparatively minor changes (chiefly to compression ratio) a diesel engine can perform adequately on a wide range of fuels varying between petrols, paraffins, gas oils and normal

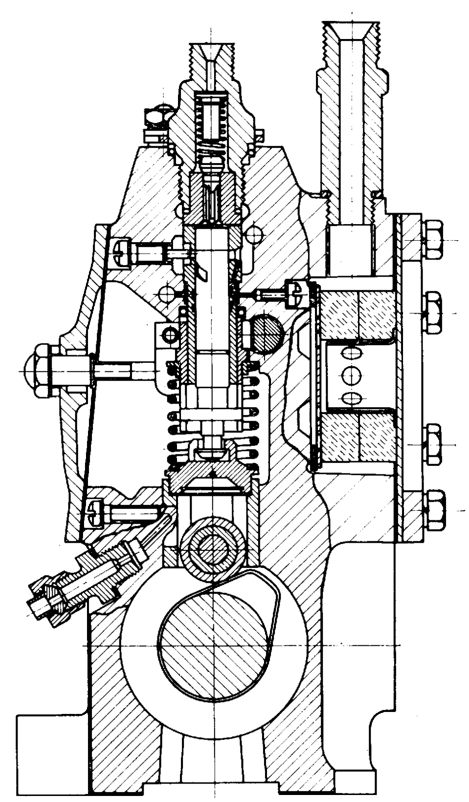
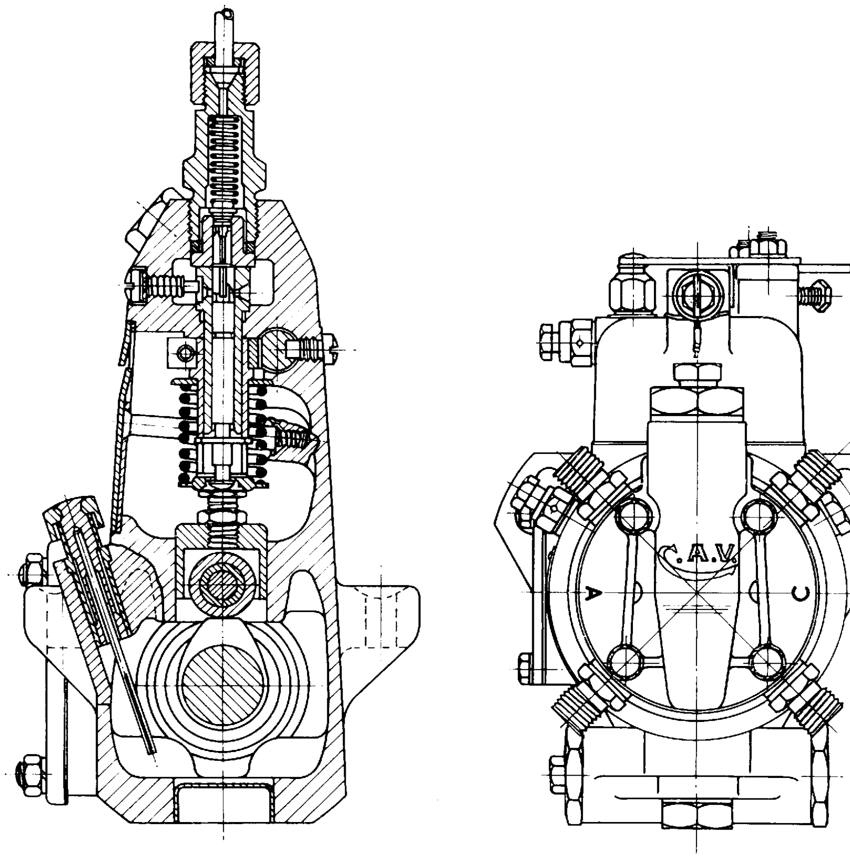
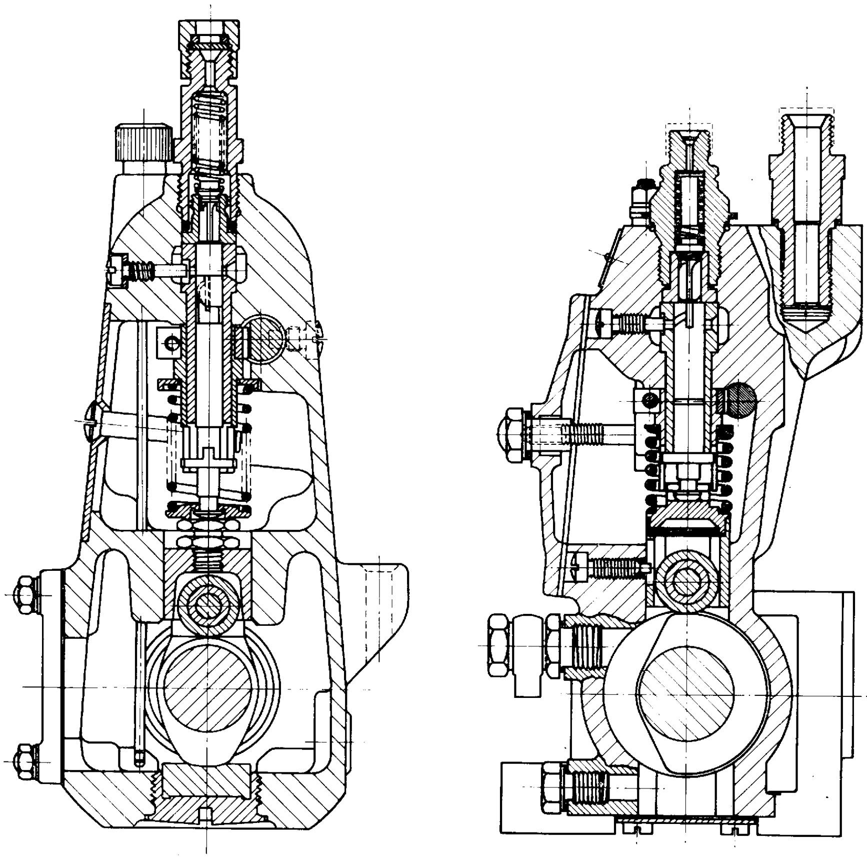


Fig. 18.—*Multi-fuel Version of 'NN' Pump.*



A PUMP DPA

Fig. 19.—*Comparison of 'A' and DPA Distributor Pumps.*



ORIGINAL B PUMP COMMONISED N.N.

Fig. 20.—*Comparison of 'B' and Commonised 'NN' Pumps.*

diesel fuels. This is obviously a valuable attribute in time of war when particular fuels may be in short supply in particular areas and the NATO military authorities have decided that all future engines for service vehicles must be of this multi-fuel type.

To operate injection pumps reliably on the thinner fuel such as petrol we require the plunger sealing arrangement first used on the "Crecy" pump. This arrangement is shown in Fig. 18 which is a multi-fuel version of the 'NN' pump. The circumferential grooves are machined in the barrel wall. The upper one is connected by a drilling to the feed gallery and serves to break down the leakage pressure to feed pressure. The lower groove is connected to a pressurised supply of lubricating oil from the engine to prevent the leakage of fuel down into the cam chamber. A metered supply of oil to the cam chamber is provided by drilling the connection into one of the tappet bores in the housing, the working clearance between tappet and bore thus forming a non-choking orifice.

Having now outlined the main course of development from the earliest days, it is fitting to conclude by indicating something of our future intentions. In particular it will be shown how the original models A, B, Z and C will all soon be superseded. In each of the remaining illustrations both pumps are to the same scale.

Referring to Fig. 19, the 'A' pump and its uprated derivative the 'AA', are both well on the way to being completely superseded by the DPA distributor type pump. The DPA is also replacing a number of 'N' pumps and eventually should do so completely.

The 'B' pump is already replaced by the 'N' and 'NN' models, the latter being the more expensive. With 'N' pump production reduced by the encroachment of the DPA it would be a great convenience to the factory to be making one model instead of two. To achieve this we have to get the cost of the 'NN' down to that of the 'N'. One of the changes which would help is the omission of the built-in filter as shown in Fig. 20. With the improved standard of main filtration already established by our paper element filters it has become difficult to justify the additional cost of the built-in final filter, particularly when our competitors do not use it. As a further cheapening measure we are currently developing a pressure die-cast housing needing a minimum of machining.

To replace the existing 'Z' pump we are about to go into production with a new pump called the 'BB'. The two designs are compared on Fig. 21. The 'BB' has the same 12 mm. stroke as the 'Z' and a range of plunger sizes up to 13 mm, but the pump base height and overall length is the same as the 'B' type. The main features follow the 'NN' pattern of design.

Finally we expect to supersede the 'C' pump, which is made in very small quantities, by a new design, the 'ZZ', which is in course of development. This has 15 mm. stroke and plunger diameters up to 16 mm. The particular version illustrated in Fig. 22 also has the "U-type" centre bearing and the over-speed trip

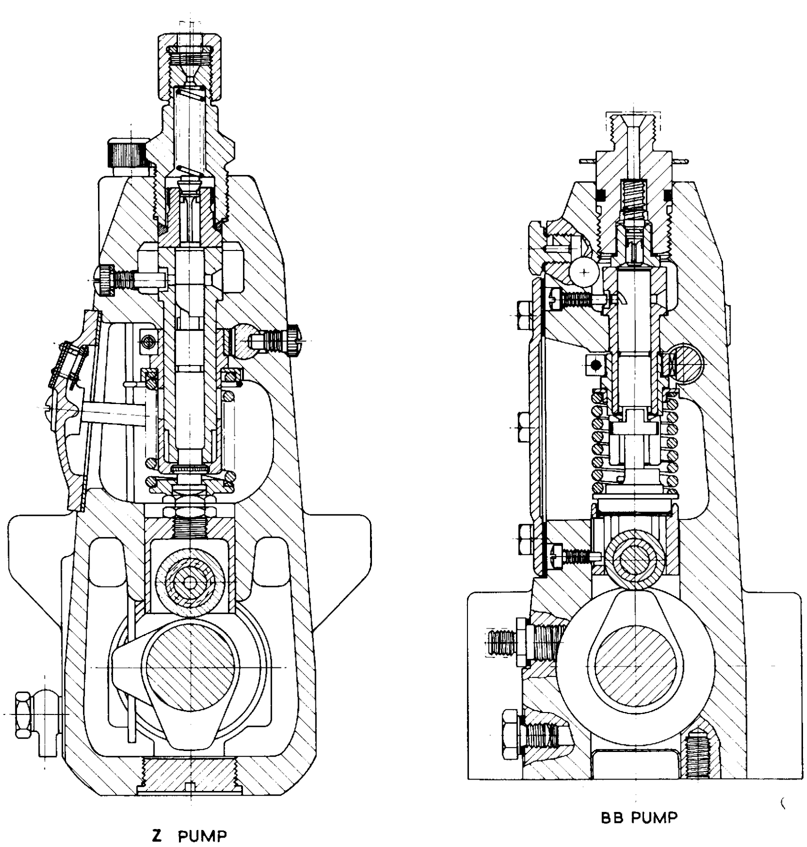


Fig. 21.—Comparison of 'Z' and 'BB' Pumps.

mechanism necessary for railway applications.

In a broad survey such as this many items of development have had to be omitted and others, which may have occupied many hundreds of hours of development time, have had to be dismissed in a few words. It is hoped that it has been useful as a background summary for the many new members of our Company and Engineering Society and also of interest to the older hands.

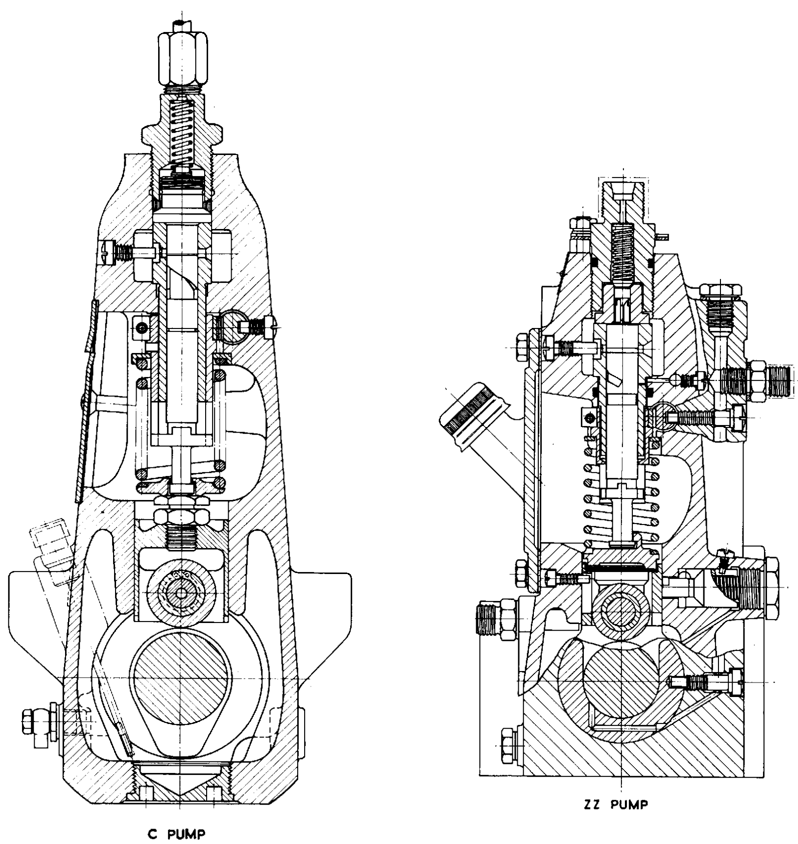


Fig. 22.—*Comparison of 'C' and 'ZZ' Pumps.*

C.A.V.

ACTON LONDON W.3

Printed in England

Publication No. 588