MODERN ENGINES

The 348 c.c. Overhead-camshaft 7R A.J.S.

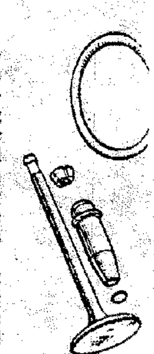
Analysis of a Successful Production Racing Power Unit Revealed by Discussion Between P. A. WALKER, A.M.I.Mech.E., Associated Motor Cycles Designer, and the Technical Editor, ALAN BAKER

THERE can be few machines in the history of motor-cycle racing which established themselves in the esteem of riders so quickly as did the 7R A.J.S. More than this: the "Boy's Racer," as the 7R is affectionately called, has more than maintained its early popularity, as a glance through the entry list of almost any three-fifty race will confirm. A strong reason is the quality of the 7R engine.

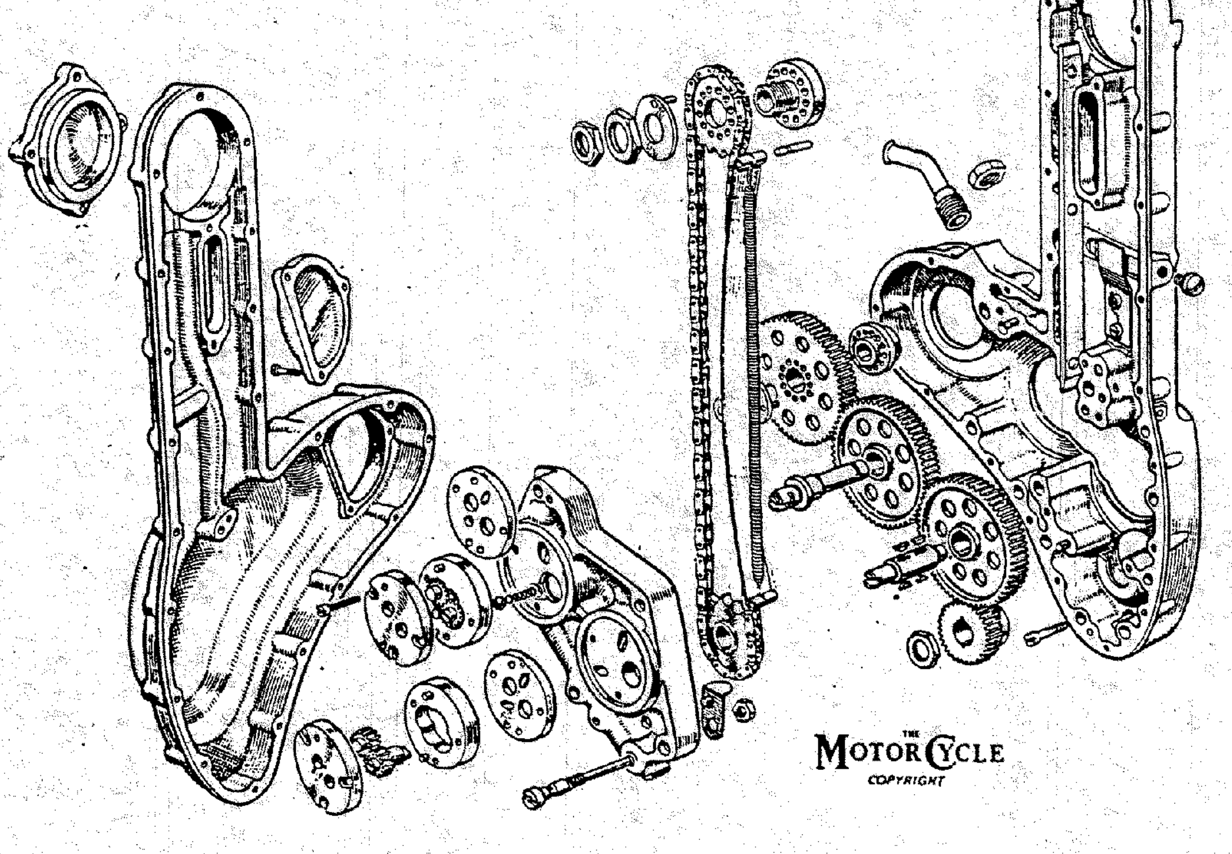
Compromise is inevitable in engine design, and the necessity for compromise is at least as prominent where racing power units are involved as it is with touring engines; the difference is one of kind rather than of degree. Happy is the designer who, in Shakespeare's words, can make a virtue of necessity, even the necessity for compromise. That the problems were successfully solved in the case of the 7R is emphasized by the facts that the engine performed remarkably well almost "off the drawing board," and that only minor modifications have been required in the subsequent six years, during which development in the racing sphere has been most intensive.

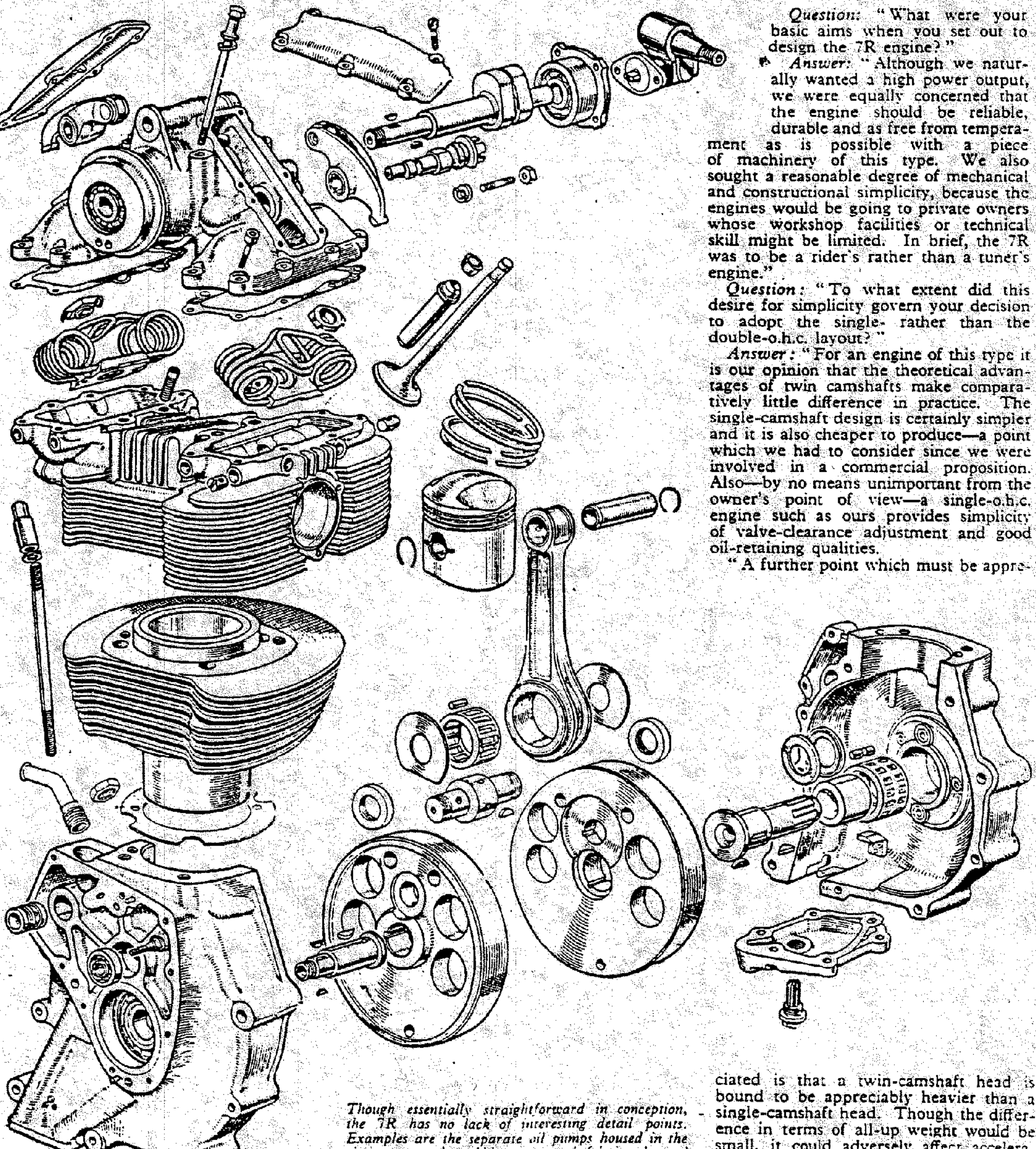
Not only in Britain has the 7R acquired its fine reputation, and not only in the hands of British riders. On the Continent, in the Americas and throughout the Commonwealth—in fact, wherever racing flourishes—is found this straightforward yet potent three-fifty, of which over 300 have been built since 1948. Among its victories are numbered first place in the Junior Manx Grands Prix of 1950, 1951 and 1952 (in the 1952 event 7Rs annexed the first three positions), and Bob McIntyre's almost fantastic second place in the 1952 Senior M.G.P. This list does not include the many stout efforts of the factory teams, nor the host of lesser successes of private owners, but it does indicate a good combination of engine power and stamina.

Philip Walker, who has many years of service with the Plumstead concern to his credit, was responsible for the design of the 7R engine, and subsequent development work has been in the hands of H. Hatch. The main modifications introduced have been stiffening-up of the crankshaft and main-bearing assemblies, alteration to the cylinder-head geometry, and the substitution of roller cam followers for the slipper type. Each point will be dealt with later in this article.



Before a designer can set to work on an engine, he must have a definite purpose in view and well-defined ideas on how certain requirements are to be achieved. This thought prompted my first question to Mr. Walker.





Question: "What were your basic aims when you set out to design the 7R engine?"

Answer: "Although we naturally wanted a high power output, we were equally concerned that the engine should be reliable, durable and as free from temperament as is possible with a piece of machinery of this type. We also sought a reasonable degree of mechanical and constructional simplicity, because the engines would be going to private owners whose workshop facilities or technical skill might be limited. In brief, the 7R was to be a rider's rather than a tuner's engine."

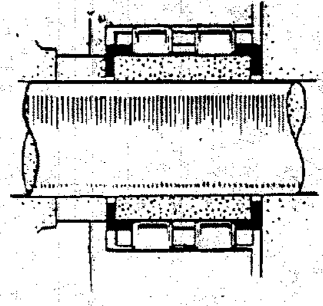
Question: "To what extent did this desire for simplicity govern your decision to adopt the single- rather than the double-o.h.c. layout?"

Answer: "For an engine of this type it is our opinion that the theoretical advantages of twin camshafts make comparatively little difference in practice. The single-camshaft design is certainly simpler and it is also cheaper to produce—a point which we had to consider since we were involved in a commercial proposition. Also—by no means unimportant from the owner's point of view—a single-o.h.c. engine such as ours provides simplicity of valve-clearance adjustment and good oil-retaining qualities.

"A further point which must be appreciated is that a twin-camshaft head is bound to be appreciably heavier than a single-camshaft head. Though the difference in terms of all-up weight would be small, it could adversely affect acceleration unless balanced by a proportionate increase in torque."

Though essentially straightforward in conception, the 7R has no lack of interesting detail points. Examples are the separate oil pumps housed in the timing case, the rubbing strip and friction-damped Weller tensioner on the chain drive to the camshaft, the roller cam followers and the unusually deep spigot of barrel into head. The crankpin and drive-side mainshaft are of composite construction; the crankpin has been turned through a right-angle in the drawing, to show the two oil-outlet holes. A single row of long rollers is employed for the big-end bearing; the outer race of the drive-side main bearing is positively located in the crankcase

In a racing engine, robustness of the crankshaft assembly is of the utmost importance because of the very high centrifugal and inertia loadings involved in full-throttle running. High power resulting from clever design of the combustion chamber, cams or ports, is of no consequence if the crankshaft cannot transmit



Thrust washers of the drive-side main bearing are flanged to avoid wear of the roller cage

it—and do so for long periods. For that reason the 7R flywheels are forged from KE.805, a high-grade carbon steel, and the mainshafts are of nickel-chrome-molybdenum case-hardening steel.

Question: "The shafts on current engines are flanged at the inboard end and are an interference fit in the flywheels, with key location; on earlier engines (before 1953) the drive-side shaft was inserted from the outside and was secured by a castellated nut. What was the reason for the alteration in construction?"

Answer: "There were three reasons. First, it was difficult to pull the shaft up really tight by means of the castellated nut because the C-spanner tended to yield under heavy pressure. Secondly, for adequate strength, the thickness of the nut had to be greater than that of a flange. Hence, by going over to a flanged shaft we were able to reduce the depth of the counterbore in the flywheel and so have a greater length of shaft supported in the wheel, with a consequent increase in rigidity. Thirdly, whereas the old one-piece shaft had two changes of diameter—always a potential source of weakness—we now have a shaft with only one change of diameter and a sleeve pressed on to form the inner race of the main bearing. The arrangement is more robust and permits sufficient case depth on the sleeve to resist wear."

Reduced Bearing Width

Question: "Another 'bottom-end' modification concerns the drive-side main bearing. Up to 1953, the outer race was a simple sleeve pressed into the crankcase, and there were three rows of rollers—two together in one Duralumin cage and the third in a separate cage. The present engines employ two rows only of $\frac{5}{16} \times \frac{5}{16}$in rollers in a common cage, and the outer race is positively secured in a flanged sleeve which is pressed into the case and held by screws. What lay behind the reduced bearing width and why do you not use a proprietary bearing unit?"

Answer: "To deal with the second point first, we could not obtain a proprietary bearing of the required dimensions and load capacity. As to bearing width, with the stiffened-up crankshaft assembly we decided that we could dispense with the outer, single row of rollers and could shorten the shaft accordingly. By shortening the shaft we could bring the primary-chain line closer to the drive-side flywheel, and so reduce the tendency of the mainshaft to deflect under chain pull. The resultant improvement in chain life has been marked.

"A point of interest on the bearing assembly is that the outer race, which is held into its sleeve by a coverplate and screws, is finish-ground after being fitted, to ensure accurate alignment."

The only engine modification for 1954 was to the cage and thrust washers of the drive-side main bearing. It was found that there was a tendency for the cage to wear against the plain thrust washers fitted last year. The latest washers are shouldered and the cage is recessed to take the shoulders, so that any axial movement results in the roller ends, and not the end of the cage, abutting on the washer concerned.

Forming the inner race of the big-end bearing is a sleeve which is pressed on to the crankpin; as on the drive-side mainshaft, this composite construction gives the desired combination of wear-resistance and strength. The pin is a parallel interference fit in the flywheels and is keyed into the drive-side wheel.

Question: "Since they are devoid of hexagons or castellations, how are the crankpin-securing nuts tightened, and why is the key in the drive end rather than the timing end of the pin?"

Answer: "Prior to 1953 we employed on the crankpin castellated nuts similar to that on the old drive-side mainshaft, and the same difficulty was met over tightening them adequately. We now fit nuts which have a hexagon on an extended shank; after the nut has been tightened, the hexagon portion is sawn off. This modification has given complete security of the crankpin, and the only disadvantage is that removal necessitates mutilation of the nuts.

"We key the drive end of the pin because the other end contains the big-end oil hole and we did not wish to weaken it any further."

Question: "The big-end bearing comprises a single row of $\frac{1}{4} \times \frac{1}{4}$in rollers in a Duralumin cage; prior to 1953, $\frac{1}{4} \times \frac{3}{8}$in rollers were employed. A single row of long rollers has the advantage of greater effective length over two or three rows of shorter rollers (of the same overall length), since there are only two end chamfers instead of four or six. Is it not a fact, however, that longer rollers have a greater tendency to skew, so that a higher degree of accuracy is necessary as regards parallelism of the crankpin? Also, why did you reduce the roller length?"

Shorter Rollers

Answer: "You are quite right regarding the advantages of the single-row bearing and the manufacturing accuracy. We set a tolerance of 0.0001in on the parallelism of our crankpin to ensure that the rollers have no tendency to run off the straight.

"A further advantage of the single-row bearing is that, since the oil outlet is midway along the pin, oil is fed to the middle of the rollers. In a two-row bearing, oil emerges between the rows, unless the feed is duplicated. On touring engines, the oil-feed aspect is not critical, but it can be so on a racing power unit where the inertia loadings at high r.p.m. are extremely heavy.

"We reduced the roller and crankpin length in order to give a shorter and stiffer crankshaft assembly. This modification was introduced concurrently with the redesigned mainshaft and main bearing. In spite of the higher unit loading resulting from the use of shorter rollers in the big end, the life of the bearing remains entirely satisfactory."

The connecting rod is forged from KE.805 steel and is subsequently heat-treated for maximum strength; a sleeve of nickel-chrome, case-hardening steel is pressed in to form the outer race of the big-end bearing. The big and small ends are each stiffened by two peripheral ribs which provide greater resistance to bell-mouthing of the eyes than would single ribs. Hence the rod is lighter than would be a single-rib design of equivalent rigidity at the eyes.

In the top of the small end is a hole through which oil mist from the crank-case passes to lubricate the gudgeon pin. Several radial holes evenly spaced round the middle of the small-end bush serve as oil reservoirs to lubricate the pin immediately after the engine is started.

Since it combines light weight with reasonable strength, Elektron (a magnesium alloy) is employed for the crank-case halves and also for the combined timing-case and camshaft-drive housing.

Separate delivery and scavenge oil pumps are carried in a cast-Elektron housing bolted to the inner half of the timing case. The scavenge pump is driven

TECHNICAL DATA

CAPACITY: 348 c.c.
BORE: 74 mm.
STROKE: 81 mm.
COMPRESSION RATIO: 10 to 1 approximately.
PISTON CLEARANCES: Top land, 0.021in; bottom of skirt, 0.008in.
PISTON RING END GAPS: Compression, 0.013 to 0.008in; scraper, 0.025 to 0.020in.
PISTON RING SIDE CLEARANCE: Compression, 0.003in; scraper, 0.0015in.
VALVE CLEARANCES: Engine cold—inlet, 0.008in; exhaust, 0.012in.
VALVE TIMING: With 0.008in valve clearance—inlet valve opens 50 degrees before top dead centre and closes 74 degrees after bottom dead centre; exhaust valve opens 70 degrees before bottom dead centre and closes 49 degrees after top dead centre.
IGNITION TIMING: Contact-breaker points begin to open 40 degrees before top dead centre on full advance.
ENGINE DIMENSIONS: Drive-side double-row roller bearing (24 rollers, $\frac{5}{16}$in x $\frac{5}{16}$in), 2.0629/2.0626in bore x 2.8168/2.8160in outside diameter x $1\frac{1}{8}$in wide; timing-side roller bearing, 1in bore x $2\frac{1}{4}$in outside diameter x $\frac{3}{4}$in wide; crankpin sleeve, 1.5156/1.5154in outside diameter x 0.696/0.694in wide; big-end bearing, single row of 14 rollers, $\frac{1}{4}$in x $\frac{1}{4}$in; big-end eye of connecting rod, 2.01600/2.01575in diameter x 0.685/0.683in wide; small-end bush, 0.875 (+0.00050−0.00025)in bore x 0.934/0.933in wide; con-rod length, big-end to small-end centres, 6.375 (±0.0025)in. Inlet-valve diameters: head, $1\frac{7}{8}$in; throat, 1.640in; stem, 0.3120/0.3115in. Exhaust-valve diameters: head, $1\frac{11}{16}$in; throat, 1.420in; stem, 0.4342/0.4332in. Valve-seat angle, 45 degrees; inlet-valve lift, 0.449in; exhaust-valve lift, 0.408in.
CARBURETTOR: Amal, type 10 G.P., $1\frac{3}{32}$in choke diameter, 12 degrees down-draught. No. 5 throttle slide; needle No. GP6; 0.109 needle jet; needle position No.2; 230 main jet.