The Oz Vincent Review

Edition #93 November 2021

Disclaimer: The editor does not necessarily agree with or endorse any of the opinions expressed in, nor the accuracy of content, in published articles or endorse products or services no matter how or where mentioned; likewise, hints, tips or modifications **must** be confirmed with a competent party before implementation.

Welcome to the final edition of OVR.

A big thank you to all those readers who have contributed articles and letters over the almost 10 years of publication. My passion for all things Vincent has waned and having recently passed my entire collection of Vincent related items, as well as my Vincent, on to a much younger and passionate enthusiast, it is time for me to say goodbye.

In signing off I must make mention of OVR subscriber, contributor and personal friend, Franco Trento who has provided OVR with a treasure trove of original documentation from the Vincent works at Stevenage. Thanks Franco for your unswerving support especially in recent troubled times.

For now the OVR archive will remain available for you to access, read and if you wish, download. To access the complete OVR archive from any device, just go to the OVR web site https://ovr270.wixsite.com/ozvincentreview

Martyn

Melbourne, Australia.
Email : Ozvinreview@gmail.com

Letters to the Editor

Hi Martyn,

I found the items on puncture repairs amusing. The author asserts that " whereby those who can and do mend punctures are forever segregated from those who cannot and do not" but that misses a far more important point. There is the third category of those **who will not** repair motorcycle tyres, a category which includes myself.

I learned a long time ago that repairing inner tubes on British motorcycles is a lost cause. The patches on inner tubes always seemed to come away within a few thousand miles. For the last forty years I have always insisted on a new inner tube and a new rim tape whenever I replaced a tyre, and I will not enter into a discussion on how "new" the used items appear to be. In that time I have never had a flat tyre on any of my motorcycles.

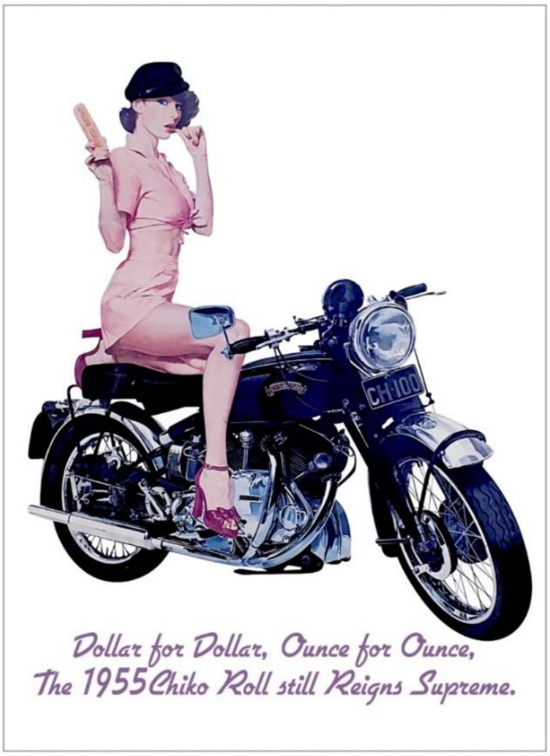
Regards, Holger, Australia

Hi Martyn

I scanned my old Chiko Roll poster from Melbourne in the early 1970s. I can't say I was ever tempted to try one. I remember someone explaining it in The Review as "something processed slowly through a dog".

Keep up the great work and I hope that you can be riding again soon. I've just done a ride around the South Island on my Prince but I think that COVID Delta is slowly winning the war so I'm not sure how much longer we'll have only limited restrictions here in the South Island.

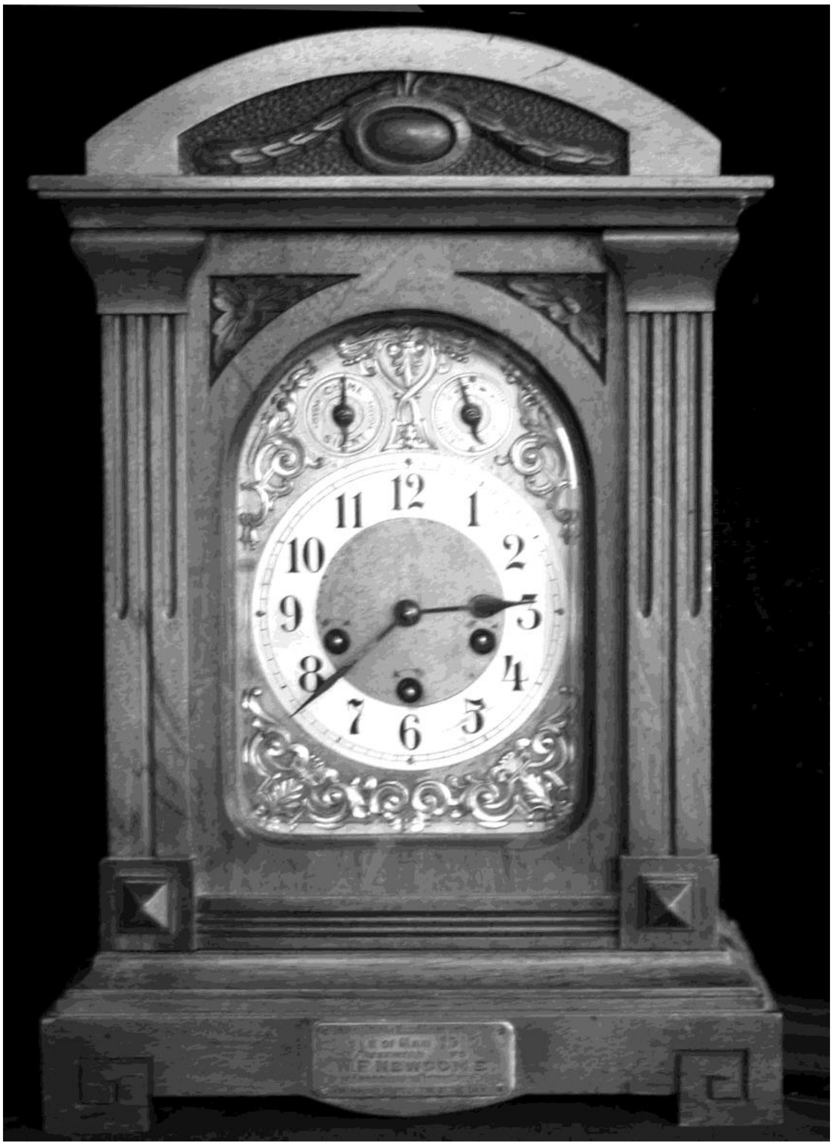
Cheers Bill, New Zealand



Tick Tock . . . . . .

Another superb contribution from David Wright, IOM

Most of the mechanical devices associated with the Isle of Man Tourist Trophy meeting of 1910 have long gone for scrap, but some 110 years on, there is one that is still running as sweetly as the day it was made. However, this lone survivor from 1910 is not a motorcycle, but a clock.

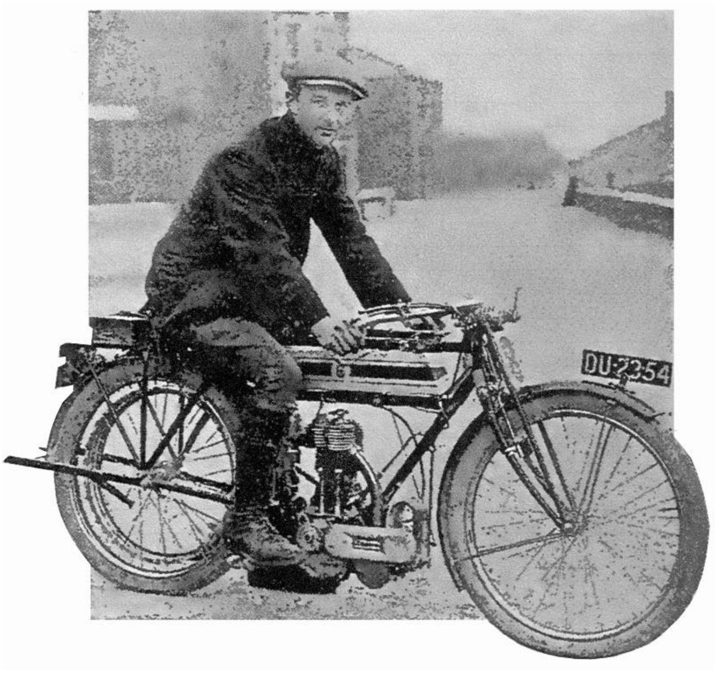


It was a fairly standard mantel clock of the day, nicely cased, with an attractive chime and with gearwheels to govern the ratio of movement between its components, an arrangement that had been developed and refined in clocks over many years. By contrast, at that 1910 TT race the provision of transmission gears for motorcycles was in its infancy, most machines having just a single-speed with direct belt drive, although Scotts and Indians had two speeds and one or two others were fitted with a variable-sized drive pulley to allow a small alteration in drive ratio.

The very first TT race of 1907 was created by the Auto Cycle Club (later the Auto Cycle Union - ACU), "for the development of the ideal touring motorcycle" and one feature that really did need developing for touring motorcycles, was the ability to climb the hills which riders would meet on their everyday travels. Over poorly surfaced roads, steep hills either slowed gearless machines to a labouring, belt-slipping, overheating crawl, or defeated them; leaving riders a sweaty push to the summit.

Motorcycle manufacturers were slow to adopt variable gears, so in an effort to stimulate development, the ACU organised challenging hill-climbs in association with the TT races of 1909 and 1910. In those days the TT ran over the undulating St John's Course in the west of the Island, but the hill-climbs involved a far more demanding 6 mile ascent of Snaefell, running from just outside Ramsey to a point some 300 yards before the Bungalow. Winner in 1910 was W. F. Newsome mounted on a 3½ hp (500cc) Triumph and this is where the clock comes into the story, because his hill-climb victory was held in such high regard by those at the Triumph factory, that the clock was presented to him in recognition of his efforts.

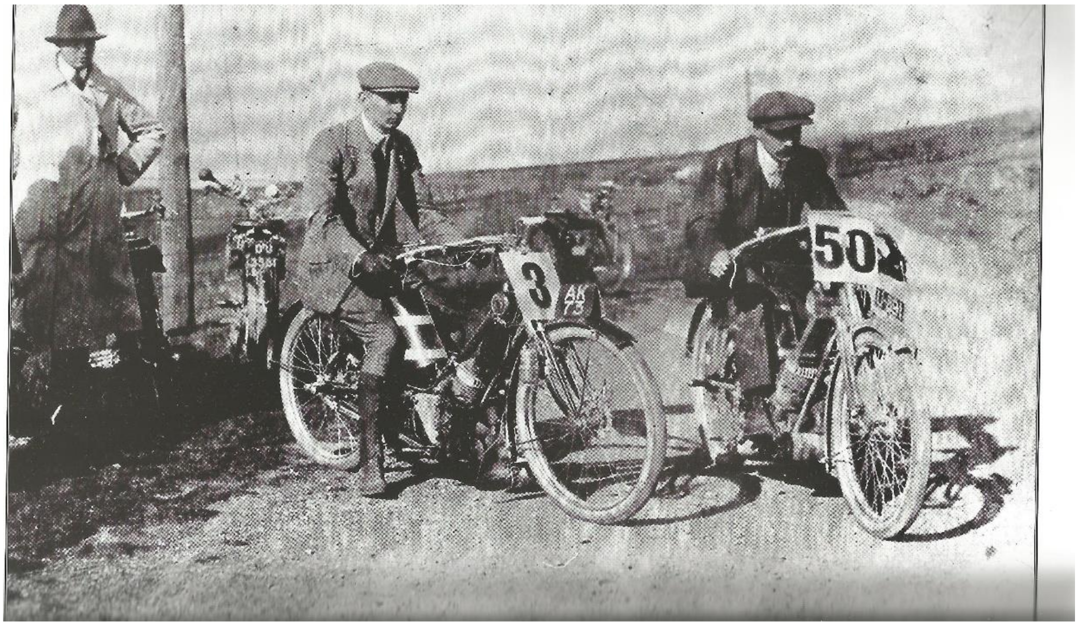
Newsome's win was achieved over a climb having a basic unbound macadam surface which 'The Motor Cycle' described as "very loose and gritty, but good on the whole". It told how the two most demanding bends were at Ramsey Hairpin and the Gooseneck, and of the latter it wrote "the engines are already heated by some stiff-collar work, and unless the riders are very adroit the engines begin to knock on the steep corner and stop immediately above it . . . . the great majority of the engines clanked heavily when the machines got their noses straight and work recommenced". A rider could lessen the handicap of a single gear by skilful juggling of levers controlling air, spark and throttle, but he could do little to counter the major drop off in performance brought on by over-heating, which was a characteristic of the iron engines of the day.



The ACU had probably expected victory in that 1910 hill-climb to go to one of the 670cc twin-cylinder Indians, with their two-speed gearboxes and chain drive. Instead, on a course where variable gears might have been expected to offer an advantage, it was Newsome on his single-speed Triumph who showed everyone the way with a time of 8 minutes and 43 seconds. The next fastest, W.O. Bentley, took 5 seconds longer on his 5 hp Indian.

A busy scene as competitors gather on the outskirts of Ramsey, prior to the start of the 1910 Hill Climb.

While the result of the 1910 hill-climb might not have achieved the ACU's aim of showing the advantage of multiple gears, as event organisers, they decreed that the 1911 TT races would move from the St John's Course to the longer and more demanding Mountain Course where, if they wanted a chance of victory, manufacturers really would have to fit their machines with variable gears. The Indian marque went on to prove the point by taking the first three places in the 1911 Senior TT with their two-speeders.



These two competitors on Scotts show the nature of the Mountain Course in 1910.

Thanks go to Alan and Mike Kelly of Mannin Collections, current owners of Newsome's clock, and to Vic Bates for photography.

Treasure from the Stevenage Works

contributor, Franco Trento

THE VINCENT

H·R·D

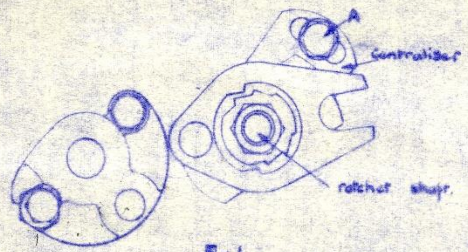


Fig.1

Stage 1: Engage ratchet shaft in 2nd gear. (Fig 1).

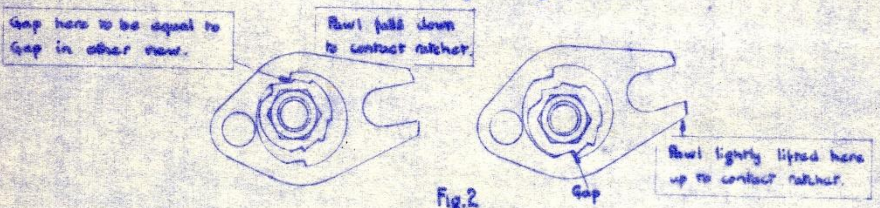


Fig.2

Stage 2: Adjust centraliser so that pawl engages ratchet equally in up and down positions (Fig 2) and tighten screw A when this condition is obtained.

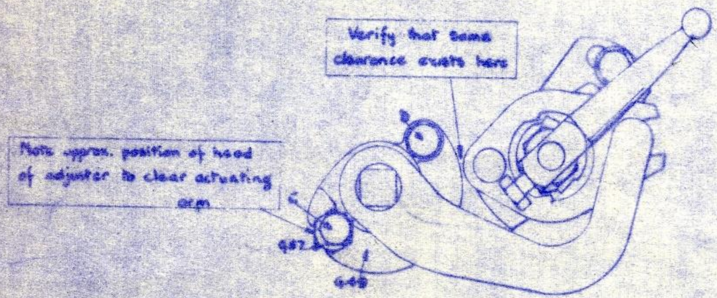


Fig.3

Stage 3: Adjust position of lever stop G48 by turning eccentric adjuster G57 so that when actuating arm is moved by its squared boss upwards to full extent bottom gear is correctly engaged and when downwards, third gear. Tighten screws C & D.

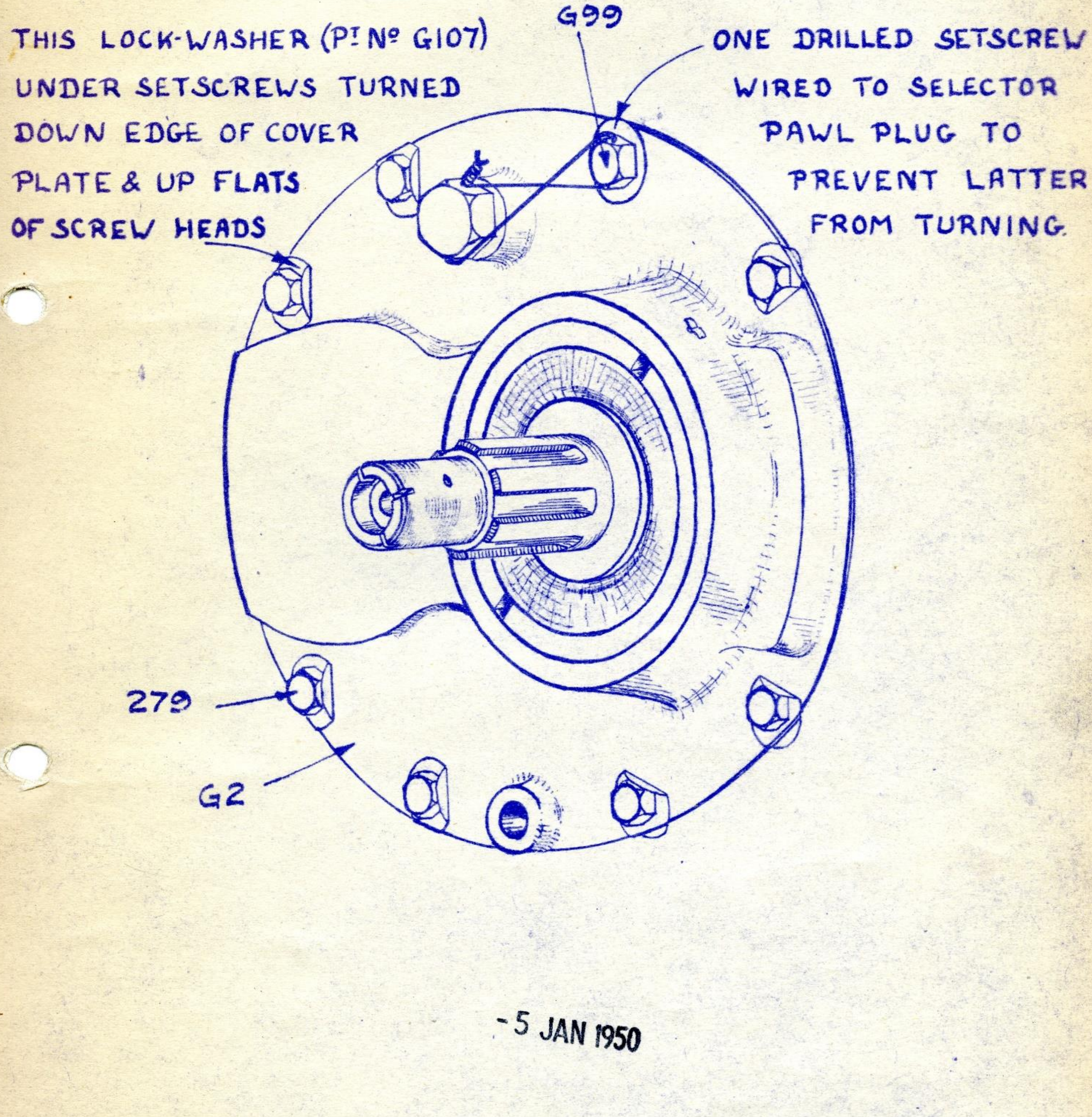
Stage 4: Check for gear-change. In all ratios ratchet shaft must be firmly notched in position after actuating arm has been slowly moved to its full extent. The indicator lever is used to check this condition (Fig 3 – which shows third gear engaged). It may be necessary to reposition G48 slightly to obtain correct engagement in all gears.

-5 JAN 1950

CORRECT METHOD
OF ADJUSTING GEARCHANGE
MECHANISM

M008.
RWJ 22.8.47

H·R·D



-5 JAN 1950

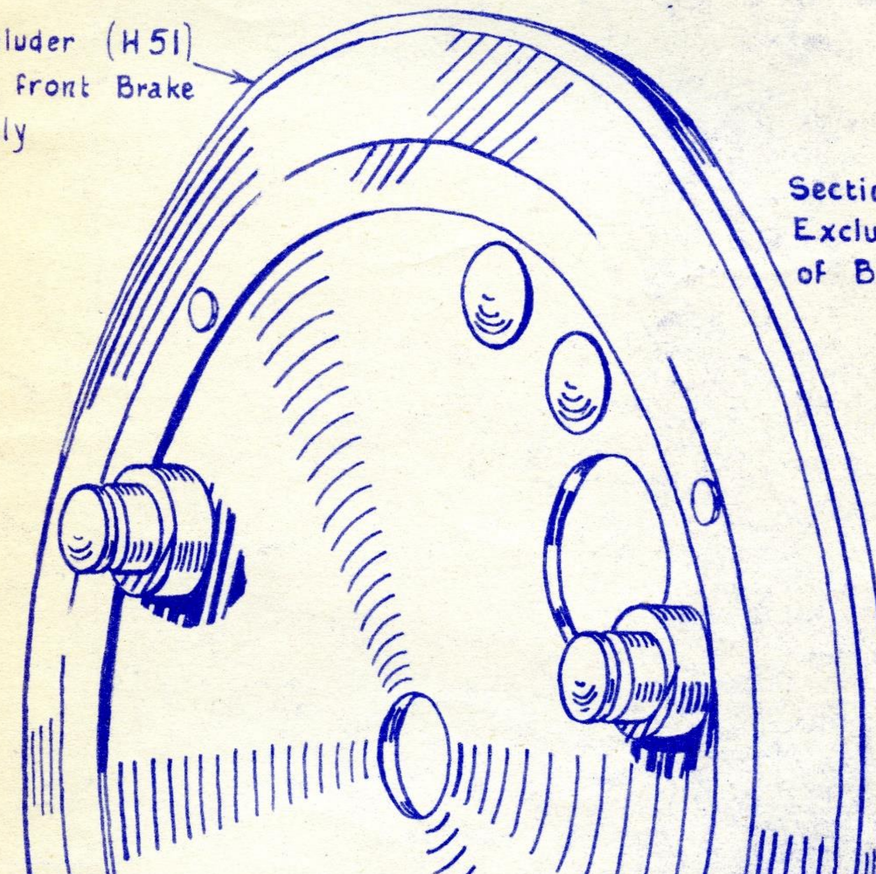
LOCKING of GEARBOX COVER SCREWS & SELECTOR PAWL PLUG

M023.

ASR 6.10.47

H.R.D

Water Excluder (H 51) fitted to front Brake Plates only



Section through Excluder and Rim of Brake Plate

Excluder locates automatically on Brake Cam Boss

Drill four holes $\frac{1}{8}$" dia. in Brake Plate to match Excluder. Rivet up with $\frac{1}{8}$" dia Aluminium Rivets. Flat heads on outside

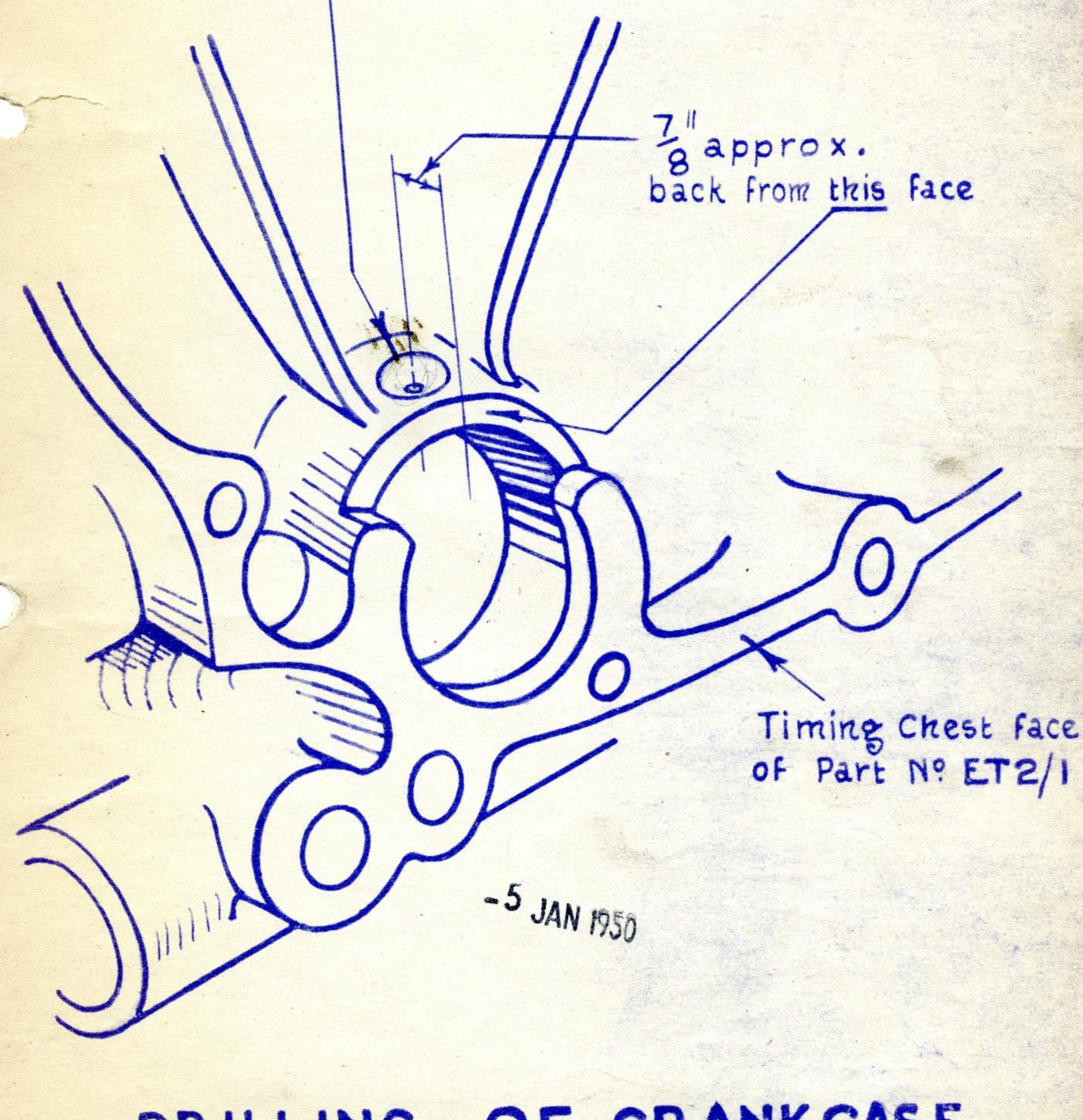
BRAKE PLATE WATER EXCLUDER

M024

R.A.K. 7.10.47

1/8"

Drill $\frac{1}{16}$" dia. through to bore at convenient angle as near to vertical as possible Counter-sink not less than $\frac{1}{2}$" dia. to provide catchment for oil.



DRILLING OF CRANKCASE

To provide additional oil-way for the lubrication of the Oil-Pump Worm

R.A.K. 12·11·47

M027

An Interview with Phil Irving, from 1990

GUEST of honour at the 1990 Australian VOC rally was the one-time chief designer of HRD-Vincent/Vincent, Phil Irving MBE, FIMechE, MSAE (Aust), then 87. He was president of the Vincent HRD Owners Club and honorary president of the club's spare parts company.

Phil was involved in the rally events going on the runs, speaking at the dinner, presenting the awards and contributing to a round-table seminar about the finer points of the Stevenage thoroughbreds. Between engagements, he found time to answer a few questions.

What was the extent of your contribution to Vincents? “Well, I arrived in England in 1930, after a trip through New Zealand and Canada with Jack Gill on his HRD sidecar outfit. HRD paid our fares from Canada, but I worked initially for Velocette. After a short while, Phil Vincent offered me a job and I worked at HRD until 1936.

Initially, we were using proprietary engines (JAP and Rudge Python) in Philip Vincents sprung chassis, but I wasn't happy with them, and then supplies dried up. My main contribution there was design of the A-series: first the 500cc Comet single and then the Rapide 1000cc V-twin, in 1936.

Then I moved to Velocettes (working on the Model 0, LE and chassis work for other machines), and went on to help with the Porcupine racer at AJS. I was involved in war-time munitions as well as the bikes.”

[Phil was out of action for three months thanks to a German bomb coming through a factory roof].

“I returned to Vincent after the war, where I designed the B-series, which included the Black Shadow, and followed that with the C-series. I left before the D-series and returned to Australia in 1949. From then, I really moved away from bikes. “

[Phil Vincent continued with the bikes (and the D-series) until 1953 when production tailed off.]

The thing that strikes me about Vincents is the attention to detail: everything seems so well though-out. “Yes, they were very practical. That was one of the keystones of our design: we considered the sort of things we thought you ought to get on bikes, but didn't. Of course, that gave us a problem with price, which wasn't helped when our alloy and castings firm went to the wall.

Generally, I handled design and detailing of the bikes, while Phil Vincent looked after policy and administration. This meant I was free to incorporate most of my ideas.”

Which of the Vincents was your favourite? “The C-series Black Shadow. It was the most advanced of all the bikes I worked on.”

What about the Black Lightning? “That was purely a race bike. The Black Shadow was more useful.”

You left Vincent in 1949. What did you do then? “I came back to Australia. At first, I was working on Chamberlin tractors, and then went to Repco research (in 1954). The first task there was to convert 5-litre Holden engines for (saloon car) racing. After a period spent writing, including "Tuning for Speed," I again worked for Repco on the three-litre V-eight which Brabham and Denny won the Formula One world championship with in 1966 and 1967. That was the first Australian-designed engine ever to win a world championship. It'll be the only one, too.

After that, I worked on the engines for Formula 5000, a race class which doesn't exist any more.”

You've been based in Australia since 1949? “Yes. I've been overseas many times but I live here. I've still got my house near Melbourne.”

I'm very impressed with the Vincents here. You must be pleased? “You know, originally there were only about 100 Vincents in the UK, we had to export most production; now there are many more. Worldwide, we've got over 1,000 members VOC members, mainly in the UK, USA, Australia, Germany and so on. We've got all the national VOC events in each country every year, plus the International Rally, which is in California next year, although I don't think I can manage an 11-hour flight across the Pacific, so I'll have to miss that.”

Do you follow the modern motorcycle scene? What do you think of the current machines? “I don't really follow it any more. As for the machines, generally I don't think much to them — they're money-oriented, market-oriented. I'm only really interested in the classic stuff . . . and the people — you get really interesting, knowledgeable people in the classic world.

We've seen a bit of a re-awakening with manufacturers outside Japan recently. The Italians, Harley-Davidson, Norton, Matchless, Triumph. Do you see Japanese predominance ever being challenged? “No, Japan is here to stay! Take the Norton — they've been working on the Wankel for over 20 years now, and I still don't know if they've got around all the problems. The Japanese can introduce a new model in a fraction of that time, The world has changed: when I started engineering in Melbourne in the early twenties, there were six independent manufacturers of engines in Melbourne; now there are none in Australia.”

More Treasures from the Stevenage Works

contributor Franco Trento, Australia

CONFIDENTIAL

The VINCENT HRD COMPANY Ltd.

STEVENAGE, HERTS, ENGLAND

Service Letter No.: 5. 7th November, 1947.

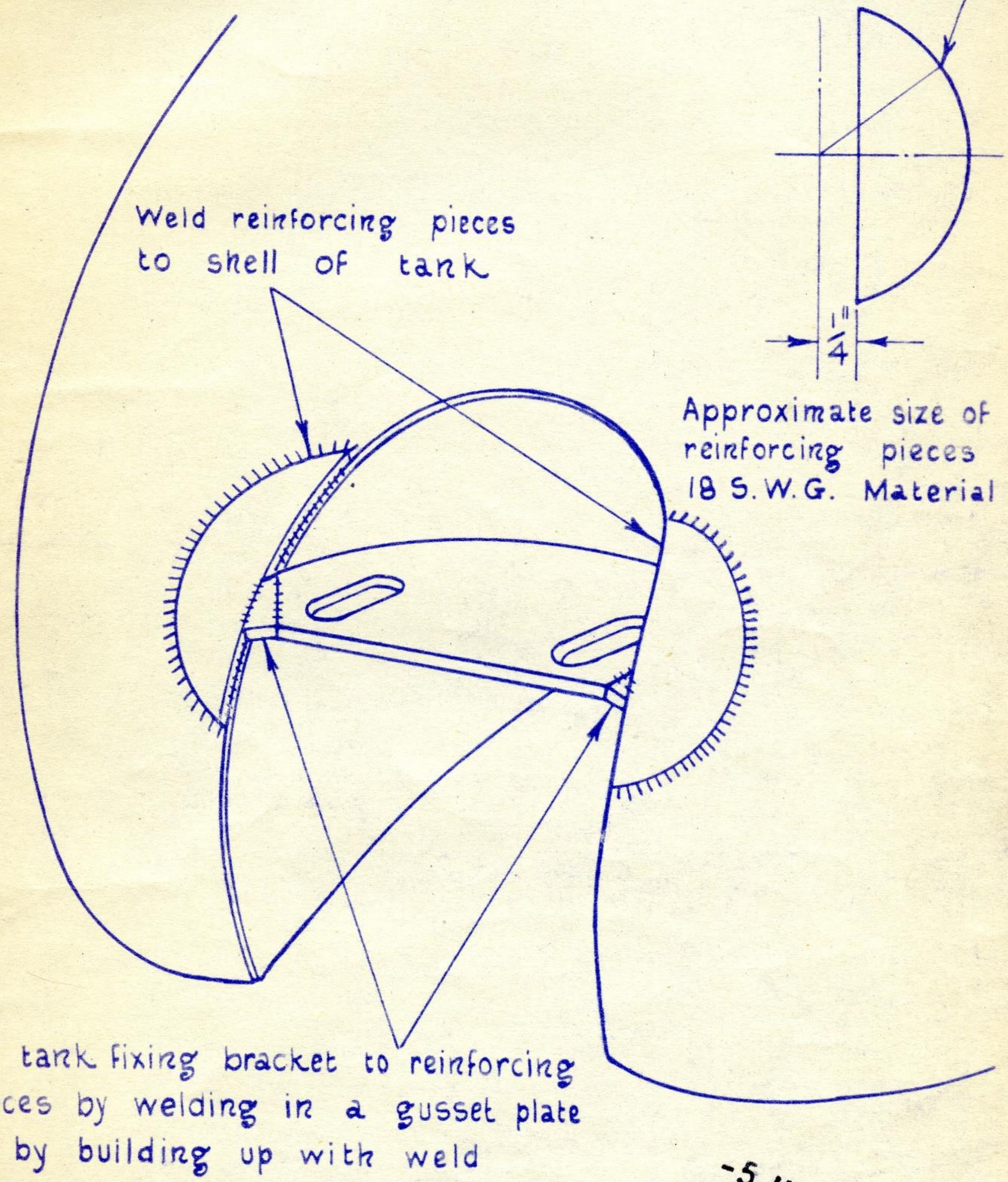
Model RAPIDE "B" Eng. Type No. F10AB/1/3 onwards Frame Type No.

Subject TANK REPAIR AND REINFORCEMENT.

Tanks which develop cracks adjacent to the rear fixing bracket can be repaired in a satisfactory manner by welding the cracks and adding reinforcing pieces as shown in the drawing MO26. Tanks repaired in this manner are stronger than before, and if the weld is dressed smooth the reinforcing plates are virtually invisible after enamelling.

When remounting the tank, use a tank rubber FT173 below the bracket, another rubber FT173/1 above the bracket, and two tank bolts Part No. FT172/1 which are longer than the original bolts FT172 to accommodate the additional rubber thickness. All tanks are now mounted with rubbers as described, but some early machines were not fitted with a rubber above the bracket.

A.R.D



-5 JAN 1950

Weld cracks and pressure test before adding reinforcements

STRENGTHENING OF PETROL TANK

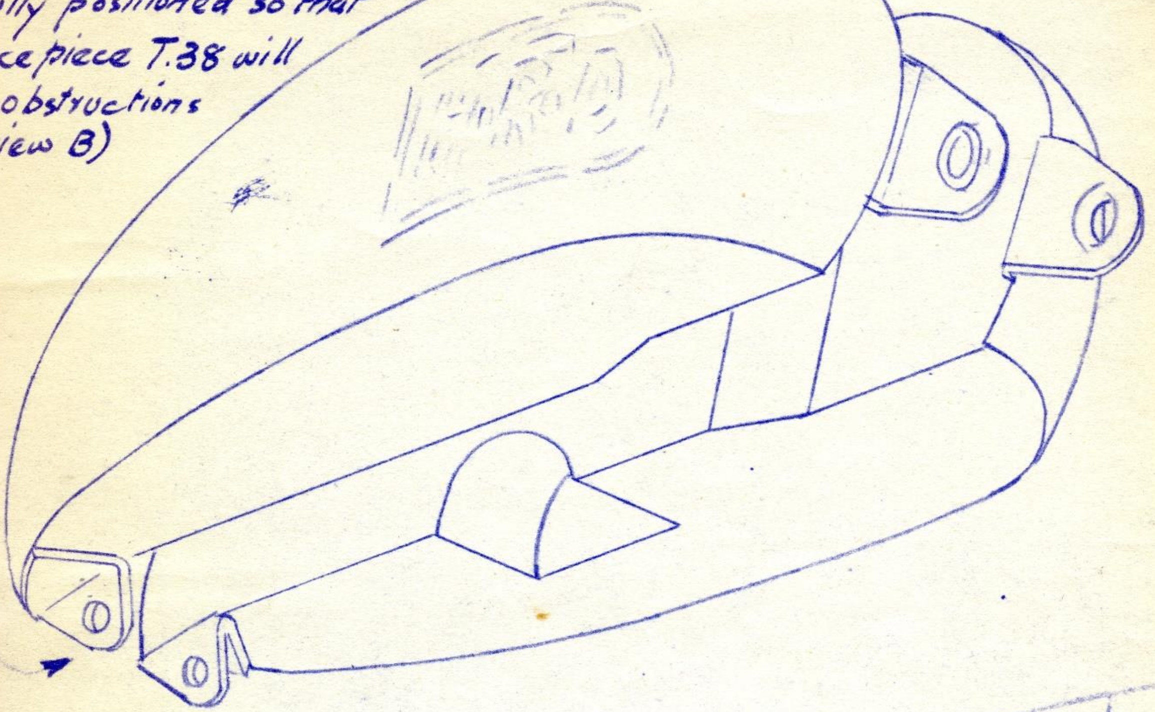
To be performed on tanks which have developed leaks adjacent to rear fixing after repairing such leaks.

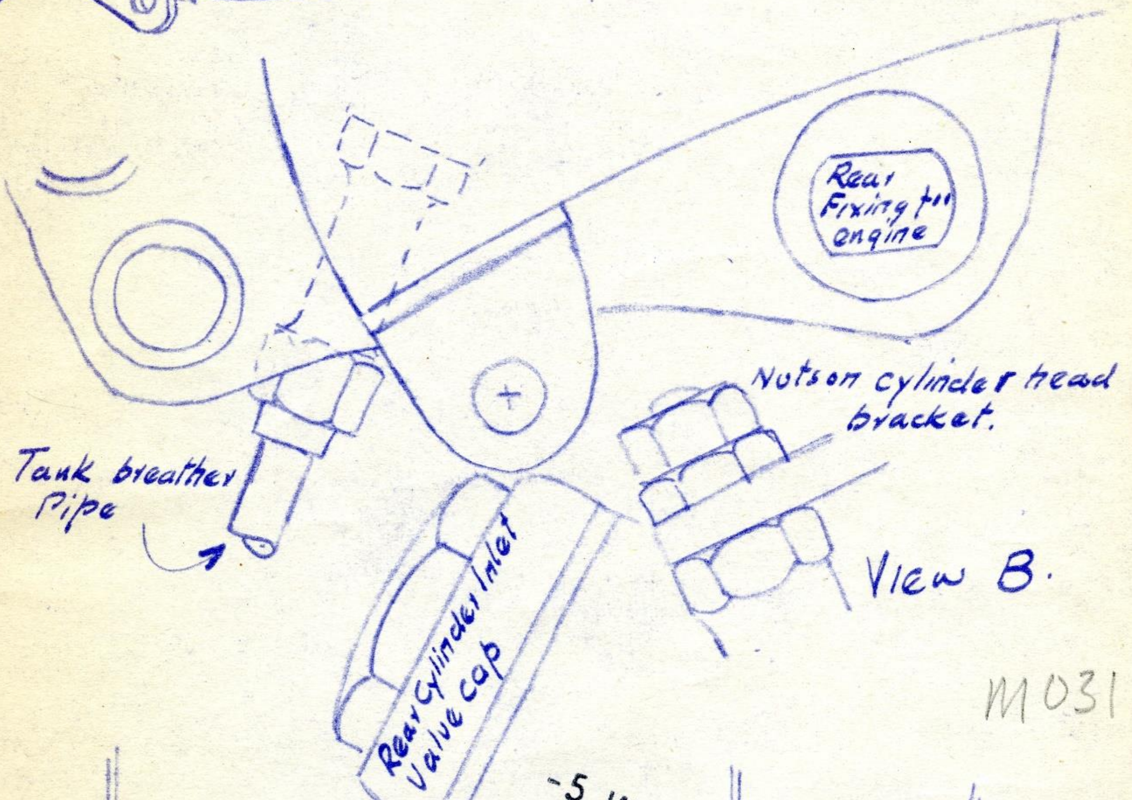
R.A.K 6.11.47

MO26

THE VINCENT HRD

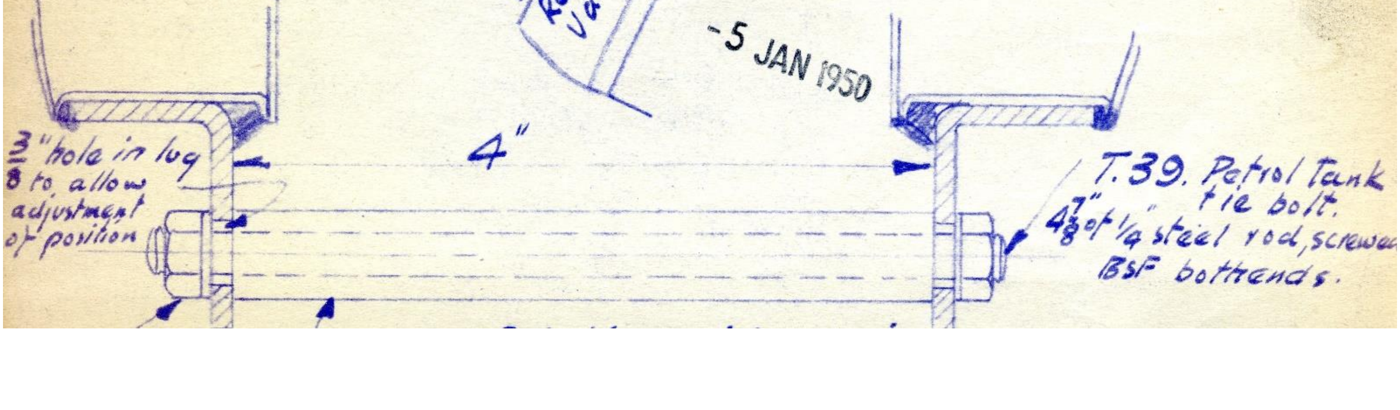
Two lugs in 10 or 12swg steel welded to shell and tunnel. Carefully positioned so that distance piece T.38 will avoid obstructions (see view B)

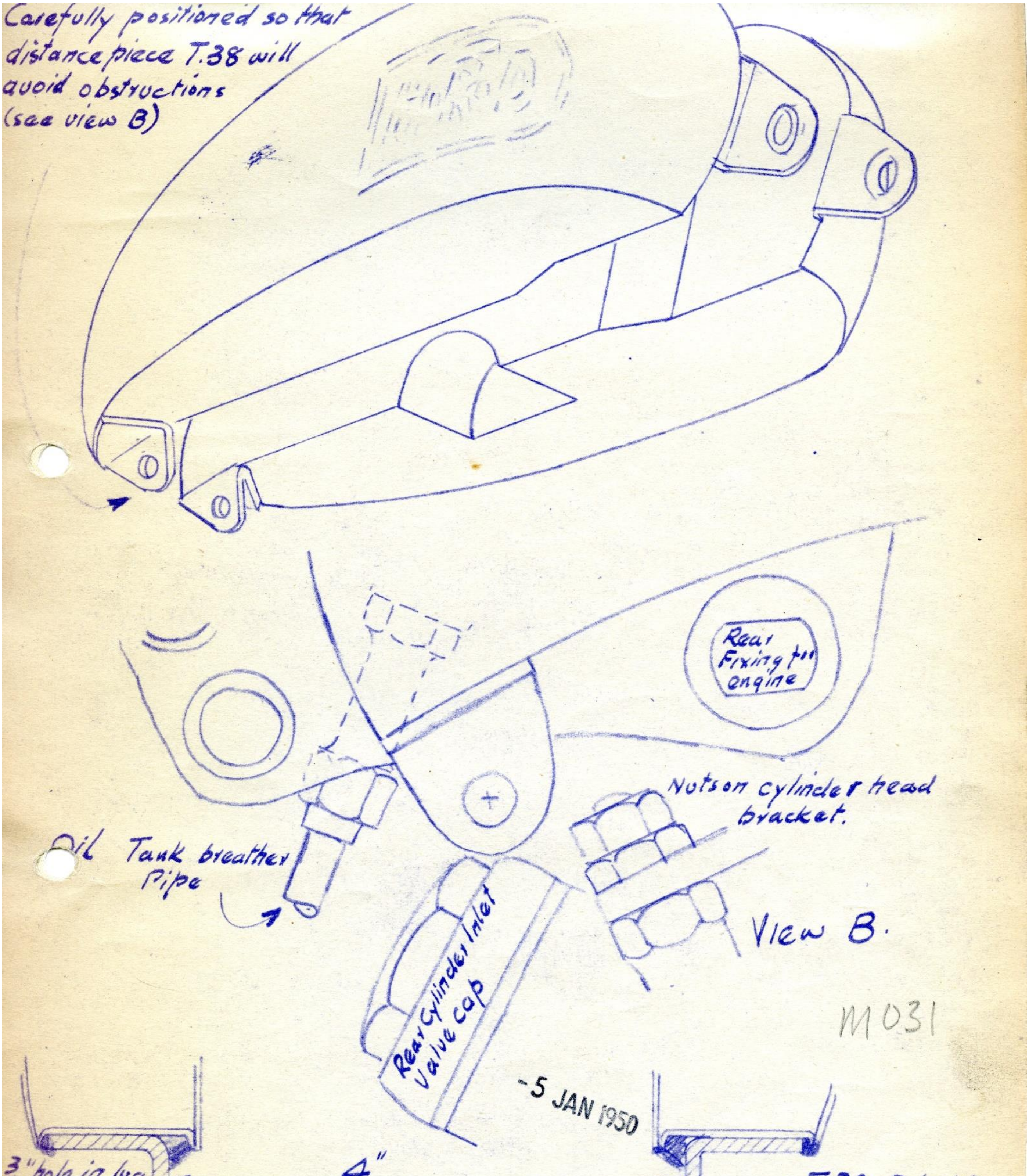




M031

-5 JAN 1950





Rear Tie for Petrol tank.

M.031.

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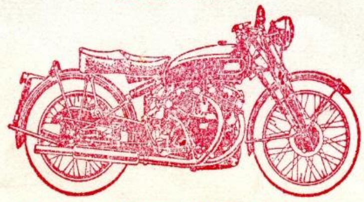
The VINCENT H·R·D COMPANY Ltd.

MAKERS OF



The World's Fastest Standard Motor Cycle

STEVENAGE

HERTS, ENGLAND

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SECRETARY: E. F. CHELSOM

Our Ref...
Your Ref...

12th December, 1951.

The Kallin Motor Co. Ltd.,
20 Hanson Street,
Adelaide,
S. Australia.

SERVICE LETTER NO. 15.

SUBJECT:- WHEEL BEARINGS - ALL SERIES "C" MODELS.

1. Part No. H22/3 has now been allotted to the Metric taper roller bearing, the subject of our Service Letter No.11 dated May 1951 and this part number which is identical for front and rear wheels should be added to the list of parts.

2. Owing to the acute shortage of bearings it has been found necessary to introduce a new combination of wheel bearings, namely the Pollard LS8 in conjunction with the ET94 roller race. Where these bearings are fitted they will be in both front and rear wheels and the following new parts are incorporated:-

REAR WHEEL

H38/10	Rear Hub.
H62	Retaining circlip.
H59/2	Brake Plate Spacer.
H59/3	Brake Plate Spacer.
H22/2	Pollard Ball Race.
ET94	Roller Race.

FRONT WHEEL

H38/9	Front Hub.
H62	Retaining Circlip.
H59/2	Brake Plate Spacer.
H59/3	Brake Plate Spacer.
H22/2	Pollard Ball Race.
ET94	Roller Race.

Where these bearings are fitted the frame number will have the following suffix:-

R.C/1/......./D. RC........../D. RC..........B/D.

3. A slightly modified system of retaining the Pollard ball race has been introduced in place of the H62 circlip referred to at point 2. As before the bearings will be fitted in both wheels and the undermentioned new parts are incorporated:-

REAR WHEEL

H15/6	Hollow axle.
H38/8	Rear Hub.
H63	Lockring (ball-race side).
H59/3	Brake Plate Spacer.
H59/2	Brake Plate Spacer.
69	1/16"x ½" Split Pin.
H22/2	Pollard Ball Race.
ET94	Roller Race.

FRONT WHEEL

H15/5	Hollow axle.
H38/7	Front Hub.
H63	Lockring (ball-race side).
H59/3	Brake Plate Spacer.
H59/2	Brake Plate Spacer.
69	1/16" x ½" Split Pin.
H22/2	Pollard Ball Race.
ET94	Roller Race.

Where this combination is used the frame number will have the following suffix:-

RC/1/.........../E.
RC............../E.
RC.............B/E.

4. The ball race side lockring H63 has been further modified to take a felt washer and the new parts incoporated are the following:-

Lockring (ball-race side)	H63/1.
Hub felt	H47/3.

Yours faithfully,
For VINCENT "H.R.D." COMPANY LIMITED.

Paul Richardson

Manager,
Technical Information.

STANDARD WIRING LOOM VINCENT H.R.D. SERIES ‘B' AND 'C' MODELS

FITTING NOTES

1) From the headlamp shell, the loom should be routed along the left hand side of the U.F.M.. Ensure that it passes over the top of the front petrol tank mounting rubber, between the petrol tank and the U.F.M. and then over the top of the oil return pipe. The loom should form a gentle loop from the headlamp, avoiding any form of restraint. It may be clipped to the oil return pipe, close to the point where this passes beneath the U.F.M. to the banjo bolt, as there is adequate length for cable flexure to accommodate fork movement from this point. In this case however, care must be taken to ensure that the loom is not put under strain as the forks move from lock to lock.

2) Pass the loom rearward, between the front carburettor and the U.F.M. then over the top of the rear cylinder head bracket, (or the rear frame tie on singles), through the space beneath the U.F.M. This provides support for-the loom at this point. Route the connections to the frame earth screw from the forward end of this U.F.M. anchorage point.

3) Pass the loom down behind the rear cylinder and beneath the battery mounting on the twin, or behind the rear frame tie on the singles. If preferred, the loom may be clipped to the main oil feed pipe for additional security.

4) On single cylinder machines, feed the horn and dynamo cables through the space between the top gearbox mounting and the rear frame tie.

5) Cables feeding the rear and stop lights should be clipped to the undersides of the respective lower R.F.M. tubes. The connecting cables from the loom provide a neater finish when these are routed beneath the R.F.M. pivot casting. Provide sufficient slack here to connect up the bullet connectors; pack. the connectors with waterproof grease to prevent corrosion.

Cut and connect the stoplight cable to suit the switch location. Under normal circumstances, the stop light switch is mounted on the right hand side of the machine. On a standard switch, connecting the live feed to the lower of the two terminals effectively prevents short circuits through the brake operating rod when this is disconnected to allow removal of the rear wheel.

Care is needed, when routing the cables in the vicinity of the rear axle, to ensure that they will not become snagged during removal of the wheel.

6) Thread each cable through the hole in the bottom of its respective side of the lifting handle and out through the hole adjacent to the mudguard hinge. This can be eased by baring a short

length of the cable and guiding this out through the top hole, using a piece of wire with a hook formed on the end.

7) Cables to the horn and dip switches should be clipped to the underside of the handlebars, between the switches and the handlebar clamps and then routed in gentle, unrestrained, loops to the headlamp.

8) Where a 5" speedometer is fitted, bare the end of the cable plus a short section close to the end of the sleeve and connect up to the two bulb holders. The loop between the bulb holders should roughly follow the shape of the mounting bracket.

Where a 3" speedometer is fitted, cut the speedometer lead to length as required.

9) An in-line fuse holder and fuse can be incorporated in either of the battery leads to protect the system against short circuits. However this will only protect the system against battery discharges; it will not protect the direct lighting circuit between the generator and the lights against defects.

10) The headlamp earth lead may be connected direct to the shell, using a small nut and bolt with a connection to the lamp unit, or to the lamp unit direct

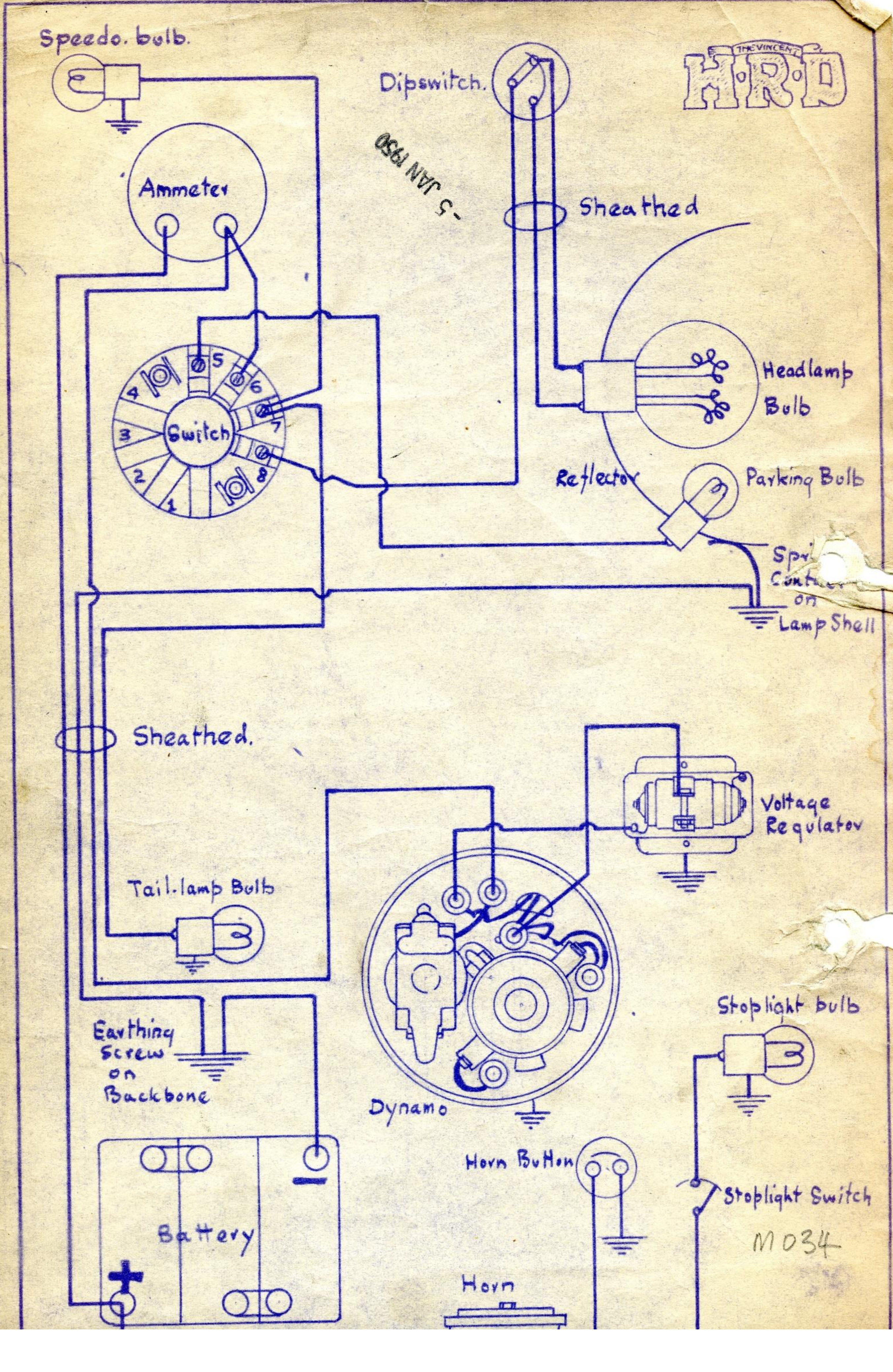
11) Should the ammeter show a charge when the lights are switched on with the engine not running, then the terminal connections to the ammeter should be changed over.

TERMINAL IDENTIFIERS for machines wired for NEGATIVE Earth (Ground)

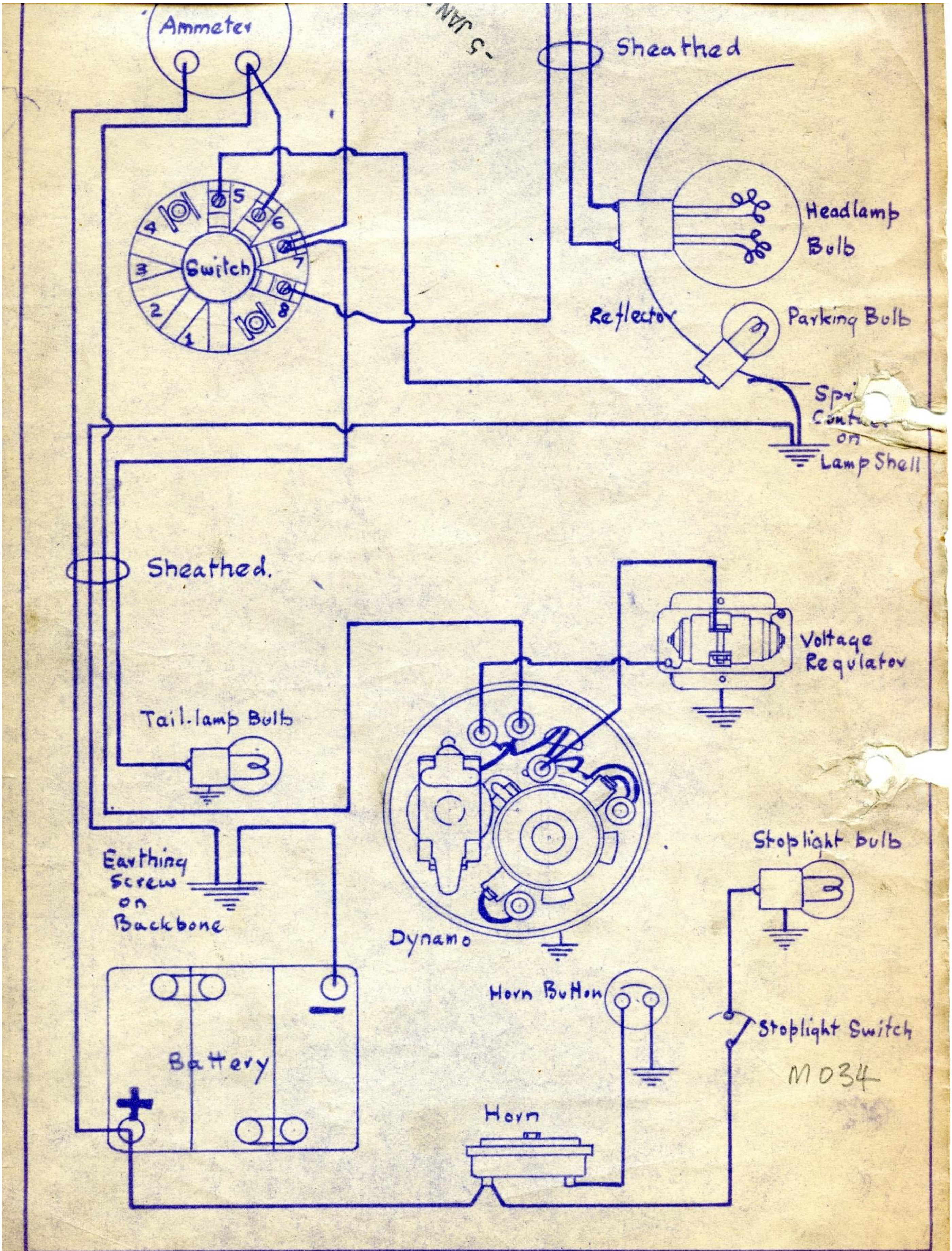
No. & Colour	**Cable Route**
1 Brown	Switch Terminal 7 to Speedometer
1 Brown	Switch Terminal 7 to Rear Lamp
2 Red	Ammeter to Battery Live (Unearthed Terminal)
2 Red	Battery Live to Horn
2 Red	Horn to Brake Light
3 Orange	Horn to Horn Button
4 Yellow	Switch Terminal 5 to Pilot Light
5 Green	Switch Terminal 8 to Dipswitch
6 Black	Headlamp Earth to Frame Earth
6 Black	Frame Earth to Battery Earth
7 Purple	Dipswitch to Main Beam
8 Grey	Dipswitch to Dipped Beam
9 White	Switch to Ammeter
9 White	Ammeter to Dynamo / Control Unit

Positive Earth (Ground) Where the machine is to be wired for POSITIVE Earth (Ground), then the RED battery lead of the wiring loom must be connected to the BATTERY NEGATIVE terminal, and the BLACK battery lead of the wiring loom connected to the BATTERY POSITIVE terminal.

It is most important to ensure that the polarity of the charging system is correct. This is especially so where a modern electronic control unit has been fitted, as these are liable to be destroyed if subjected to reversed polarity.



Speedo. bulb.
Dipswitch.
THE VINCENT H·R·D
-5 JAN 1950
Ammeter
Sheathed
Switch
1
2
3
4
5
6
7
8
Headlamp
Bulb
Reflector
Parking Bulb
Spring Contact on Lamp Shell
Sheathed.
Voltage Regulator
Tail.lamp Bulb
Earthing Screw on Backbone
Dynamo
Stoplight bulb
Horn Button
Stoplight Switch
M034
Battery
Horn



WIRING DIAGRAM M.034.

M 034 contributed by Franco Trento

Even More Stevenage Treasure From Franco

CONFIDENTIAL

The VINCENT H.R.D COMPANY Ltd.

STEVENAGE, HERTS, ENGLAND

Service Letter No.: 6 23rd December, 1947.

Model RAPIDE 'B' Eng. Type No. Frame Type No. R2340 onwards.

Subject REAR MUDGUARDS.

Cracking of the rear mudguard can be prevented by the following modifications:-

(1) Duplicate the reinforcement strip FT152 in the blade, and the strip FT151 in the flap.

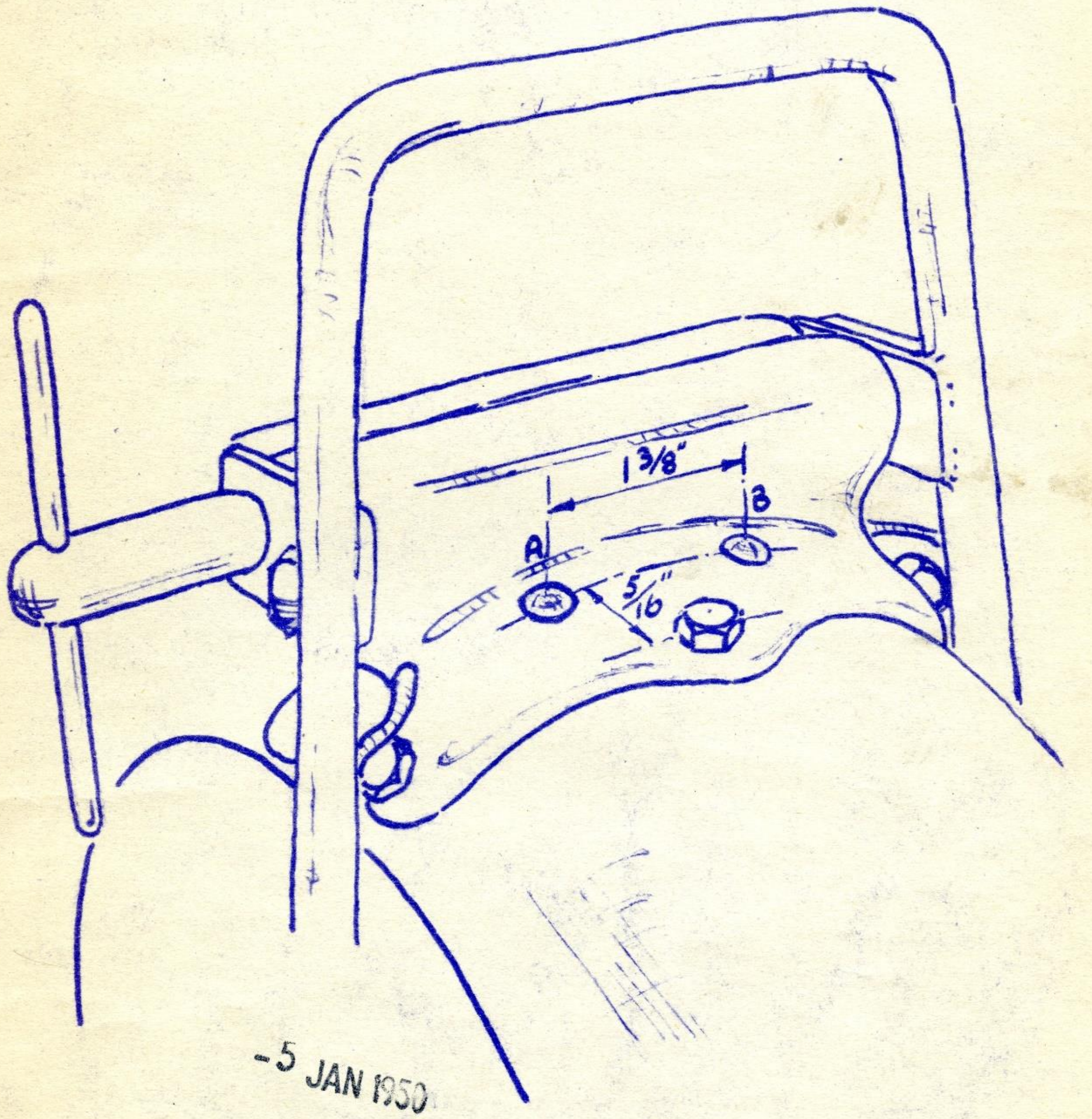
(2) Add 2 3/16" rivets or bolts through the hinge bracket 1½" apart, on either side of the single central bolt.

(3) Add 2 additional stays, Part No. FT22/2 running upwards from the fork-ends to a point about 1" behind the single bolt under the Dual-seat.

To carry out modification 3, the necessary stays and bolts will be supplied on request for the nominal sum of 7/10d, subject to normal discount.

All the above modifications are standard on all machines subsequent to Frame No. R2340. The other two were introduced earlier, and many existing machines have been converted.

H·R·D



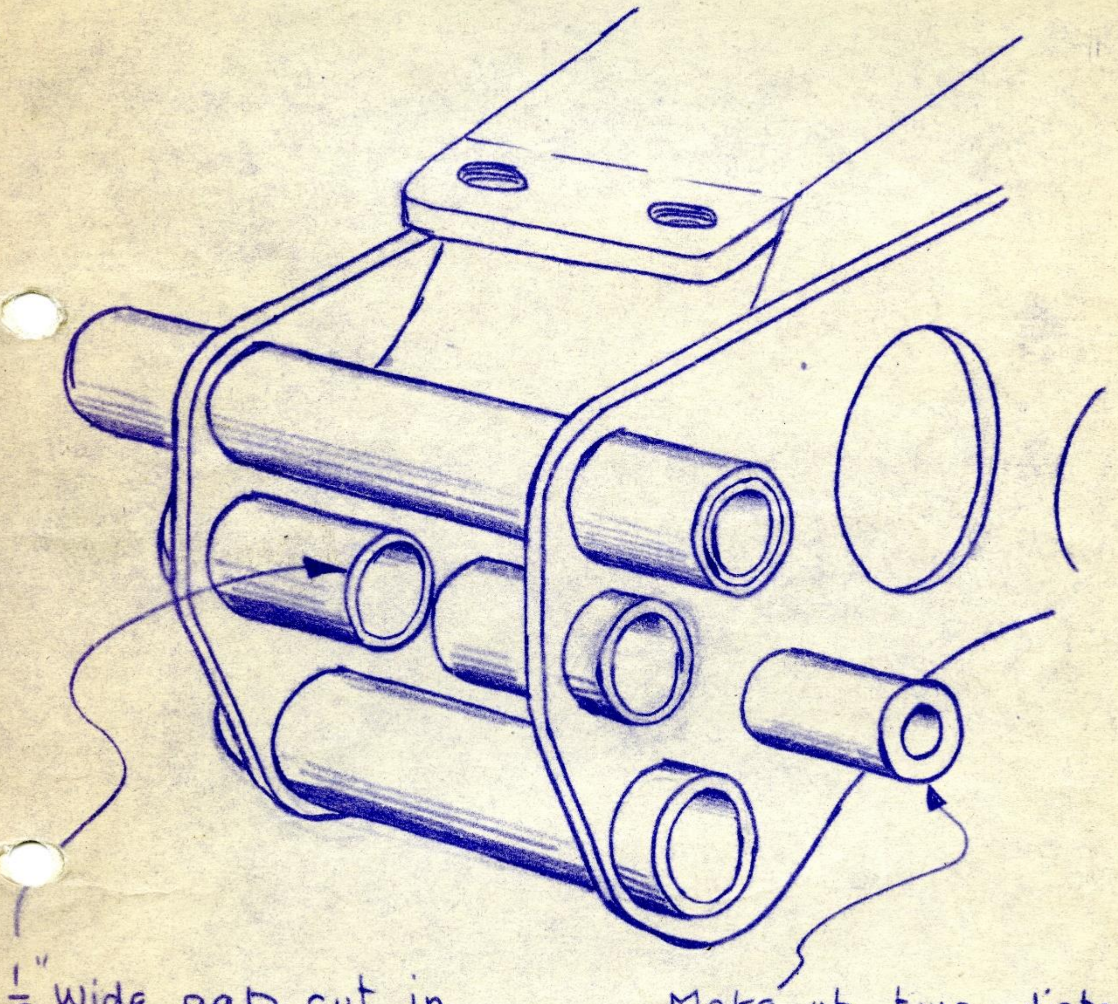
Drill thro' hinge & guard 2 holes, No. 22 drill (.157" dia.). Rivet up with 5/32" dia x 3/8" long snap-head aluminium rivets with washers (5/32" I/D x 3/8" O/D x 1/32" thick) inside guard. As at A & B.

EXTRA RIVETING OF MUDGUARD BLADE HINGE.

RWJ 21·8·47.

M020

M059

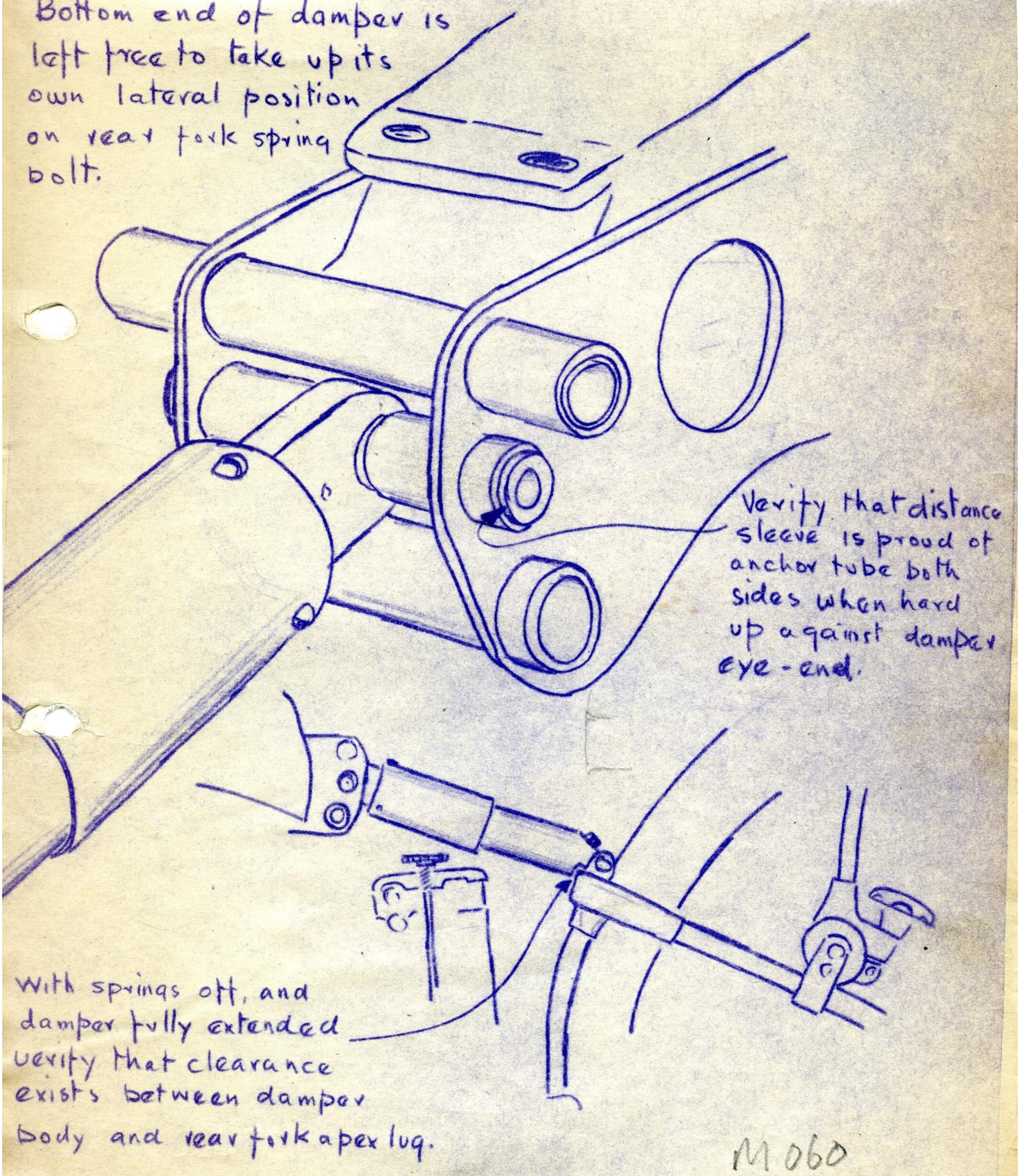


$\frac{1}{2}$" wide gap cut in spring anchor tube in upper frame member. This gap must be central with tube.

Make up two distance sleeves $1\frac{27}{32}$" long from existing sleeve. It is important that the cut ends are square to the bore.
Alternatively two new sleeves Part No F57/3 may be used.

Preparation of Upper Frame Member to receive Hydraulic Damper.

Drg No M 059



M060

Fitting of Hydraulic Damper.

Drg. No. M060

CONFIDENTIAL

The VINCENT HRD COMPANY Ltd.

STEVENAGE, HERTS, ENGLAND

Service Letter No.: 7.

Model RAPIDE "B" Eng. Type No. F10AB/1/70-370 Frame Type No.

Subject OIL PUMP WORM. PART NO. OP 31.

Cases of failure of this component after a short mileage have occurred, and the cause has now been definitely traced to the use of incorrect material by the suppliers. Fortunately, only a small number were made of this material, but there is no means of stating definitely to what engines they were fitted, except to say that the engine numbers were between F10AB/1/70 and F10AB/1/370. It is very advisable to examine any of the engines within this bracket at the earliest available opportunity and replace the worm should heavy wear be apparent. Subsequently please notify our Service Department of any change, so that we will know when all the faulty worms have been replaced.

To extract the worm, remove the timing cover, mainshaft nuts, pinion and key, and the oil pump plug which lies outside and to the rear of the timing chest behind the oil pipes. Screw a bolt $\frac{1}{4}$" B.S.F. and 2" long approximately into the tapped hole in the end of the pump plunger which is exposed when the pump plug is removed. When the bolt is tight, continue to apply pressure in a clockwise direction and commence to rotate the engine slowly forward. This should have the effect of forcing the worm outwards, moving the inner race of the roller bearing ahead of it. When the roller race has moved outwards a short distance it can be taken off the shaft and the worm finally extracted with the aid of 2 B.A. screws fitted into the two extractor holes provided. (If no screws of this size are available, use two of the clutch spring retaining screws.)

When replacing the worm, first make absolutely sure that the No.6 Woodruff driving key is in its place on the shaft. Then refit the worm, making sure that it slides easily over the key, and replace all the other parts in the reverse order, with the dotted tooth on the pinion meshing with the tooth - space marked with one line, and the pinion key located in the dotted keyway. Tighten the pinion nut fully and finally punch-lock it into one of the keyways.

CONFIDENTIAL

The VINCENT HRD COMPANY Ltd.

STEVENAGE, HERTS, ENGLAND

Service Letter No. 8

26th April, 1948.

Model RAPIDE 'B' Eng. Type No. .... Frame Type No. ....

Subject CLUTCH SEALING WASHERS.

Trouble has occasionally been experienced with oil leaking up the splines of the clutch shoe carrier, past the copper washer, C.18, and hence to the clutch shoes.

This trouble can be cured by assembling the clutch shoe carrier with a liberal coating of non-setting gasket cement on the splines. We have, however, now developed a washer comprising a steel outer ring to which is bonded an inner ring of rubber. This part, C.18/1, is being fitted as standard to all models subsequent to No. F10AB/1/656 and can be supplied for fitting to machines previous to this number.

Owing to the section of the steel outer ring, C.18/1 can only be used if the diameter of the face of the clutch retaining nut, C.20, is greater than 1.218" (31 m.m.) measured at the inner edge of the chamfer. In some cases it may be necessary to obtain a new nut or, alternatively, the face of the nut can be machined back to give the required diameter, and a similar amount machined off the collar adjacent to the hexagon of the nut. Such machining must be done square to the head.

Great care must be taken when assembling to ensure that the washer, C.18/1, is fitted correctly into its recess in the clutch shoe carrier, otherwise it may fall out of position and be trapped and damaged by the nut.

Telephone & Telegrams:
Stevenage 375 (2 Lines) Head Office, Spares & Service
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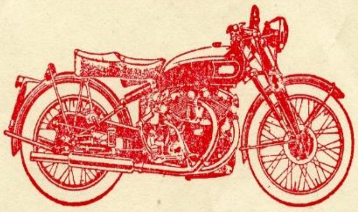
ON WAR OFFICE & AIR MINISTRY LISTS

Code: A.B.C. 7th Edition

The VINCENT H.R.D COMPANY Ltd.

E. C. BAILLIE, C.A. RECEIVER

MAKERS OF



The World's Fastest Standard Motor Cycle

STEVENAGE

HERTS, ENGLAND

Our Ref.

Your Ref.

28th June, 1951.

SERVICE LETTER NO.12

Models: Series 'C' Black Shadow.

Subject: Gear Ratios.

With effect from June 1951 all "Black Shadow" Engines will be fitted with "Standard Rapide" gear ratios, unless otherwise ordered.

Engine No. F10AB/1B/7076 is the first "Black Shadow" unit featuring the 9.1 bottom gear and we would add that this change has been effected as a result of many suggestions in this direction from "Black Shadow" owners throughout the world.

For a limited period we shall be in a position to supply "Black Shadows" with the original 7.2:1 bottom gear if specially ordered.

PR/MCJ.

The End Is Nigh

From Aaron League

According to the UK Department for Transport's decarbonisation plan, a 220-page document released recently, all new vehicles must be 'fully zero emissions at the tailpipe' in just over a decades' time.

The news comes less than nine months on from the UK government's announcement that the sale of all new petrol and diesel vans will end come 2030, whether we're ready for it or not.

And while many pundits, action groups and organisations were quick to claim there was no mention of bikes, it'll be just five years later that the same fate hits two-wheelers.

Not only will racers need to rethink their transporters from 2030 with the van ban on its way, but by 2040 the axe is set to come down on HGVs as well, as part of the final stage of phasing out ALL non-zero emissions vehicles that same year.

By mid-century the planned zero retail demand for petroleum fuels will see the disappearance of many petrol (gas) stations across Europe and the UK, forcing vehicles reliant on them into retirement.

"While cars and vans outnumber motorcycles on UK roads, motorcycles are an important and sizeable vehicle population, with 1.4 million licensed in 2020 and we do not want to see them remaining fossil fuelled as the rest of the vehicle fleet cleans up," the plan said.

"The opportunities for zero emission light powered vehicles (ZELPV) are enormous. We will build on our existing support in this segment, such as with the plug-in motorcycle grant, to benefit urban logistics and wider mobility and look to grow a new UK industrial supply chain.

"We will use Zemo's strategic partnership [formerly known as the Low Carbon Vehicle Partnership] with the Motorcycle Industry Association (MCIA) to stimulate and coordinate activity in this area and publish options to develop this at national and local level this year.

"Zero emission motorcycles and other powered two wheelers are an efficient and clean form of mobility that can reduce congestion, improve urban air quality and reduce noise – we will take forward measures to remove these emissions…ensuring we support the development of new industrial opportunities for the UK."

In 2019, motorcycles and mopeds made up just 0.4 per cent of all domestic UK greenhouse gas emissions. Cars and taxis were responsible for 55.4 per cent while heavy goods and light duty vehicles made up 31.6 per cent.

The plan is among a number of new regulations expected to be set out in the run up to 2021 November's United Nations Framework Convention on Climate Change (UNFCCC) Conference of the Parties (COP) in Glasgow.

“At this meeting, potentially one of the most important events in recent history, almost every country in the world will be represented”, DoT officials added.

“They will decide whether to deliver, and whether humanity takes what many believe to be its last best chance to get runaway climate change under control. As the president and host of the conference, the UK’s own intentions and commitments will significantly affect the chances of an ambitious global deal.”

Strangely there is no mention of restrictions on petrol powered and clearly polluting lawnmowers and similar equipment

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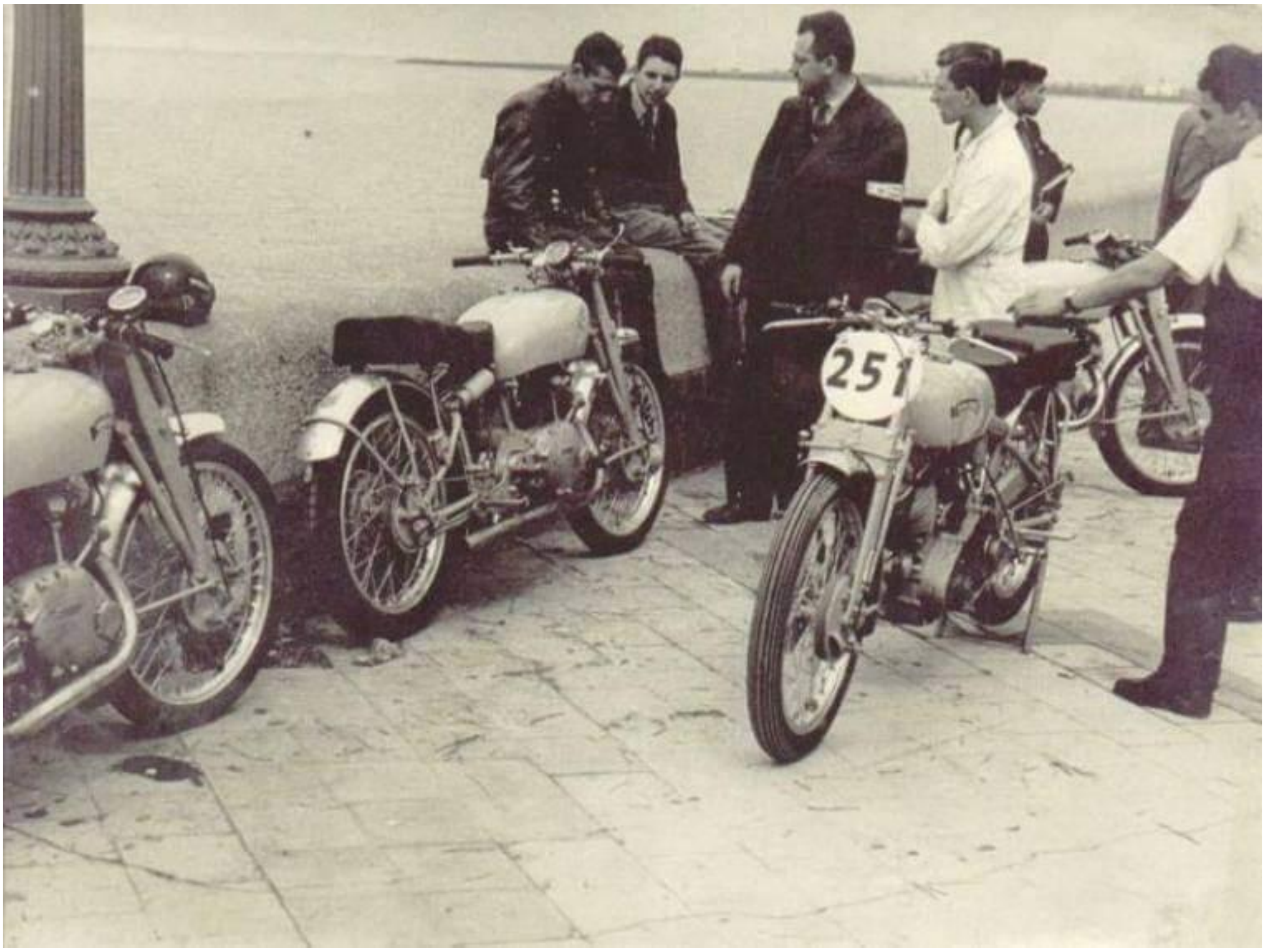

Four Grey Flash’s in Brazil, mid 1950’s

Want a Cheap Black Shadow? Here's How.

Stevenage 600 (4 lines) Accounts and Purchases
Stevenage 1100 (2 lines) Sales, Spares and Service

ON ADMIRALTY, WAR OFFICE
AND AIR MINISTRY LISTS

Code
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VINCENT ENGINEERS (STEVENAGE) LTD.

MAKERS OF

The World's Fastest Standard Motor Cycle

STEVENAGE
HERTS, ENGLAND

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Our Ref. JB/BW
Your Ref.

CORRESPONDENCE SHOULD BE ADDRESSED TO THE COMPANY AND NOT TO INDIVIDUALS

29th January, 1957.

J. W. Tenhoff, Esq.,
Mountain Lake,
Minnesota,
U.S.A.

Dear Sir,

We thank you for your letter and note that you have purchased an open Series "D" Rapide.

To convert the engine to the same specification as a Series "D" Black Shadow or Black Prince the only modifications required are the polishing of the con rods, rockers, cam followers and valves. It would also be necessary to open out the inlet ports and carburettor stubs to 1⅛" bore and to fit the type 389 Monobloc carburettors, part number PR5/6.

Assuring you of our best attention at all times, we are,

Yours faithfully,
for VINCENT ENGINEERS (STEVENAGE) LIMITED.

J. BLAND.
SPARES & SERVICE MANAGER.

Customers' Motorcycles are driven by our own Staff at Customers' own responsibility. No Liability is accepted by this Firm for any Motor Vehicle, Motorcycle or Stock entrusted to them

Contributed to OVR by Lou Collodetti, Australia

MOTOR CYCLING with SCOOTER WEEKLY PERFORMANCE DATA CHART **PART 1**

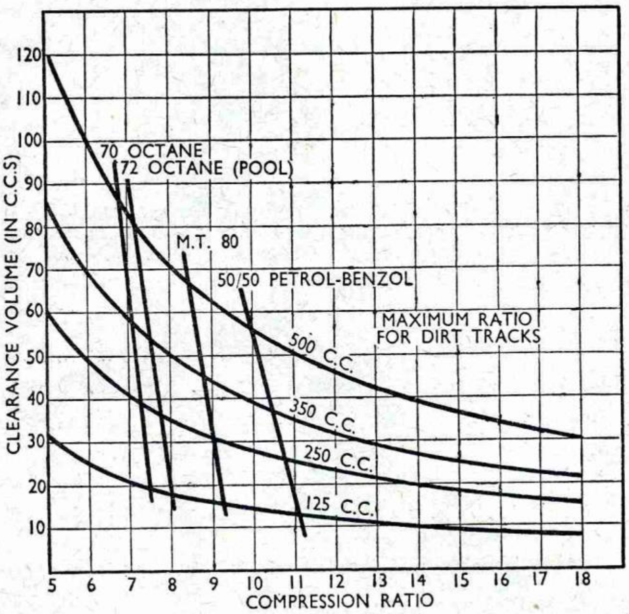
TUNING for SPEED

charts by Phil Irving

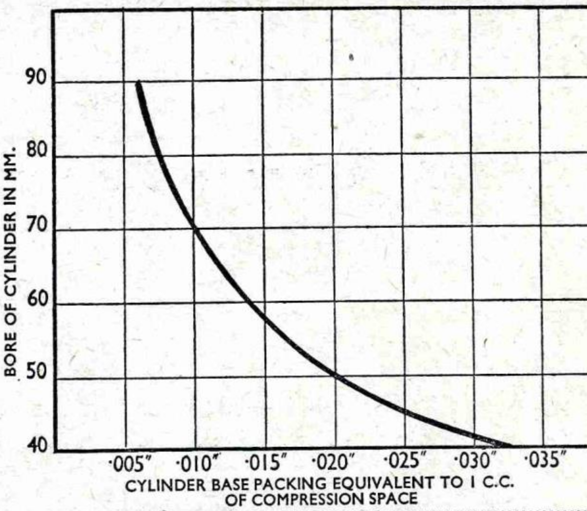
Valuable time may be wasted in searching for technical formulae and information. In these charts much that amateur tuners may require is set out in easy-to-find form.

Finding Compression Ratios

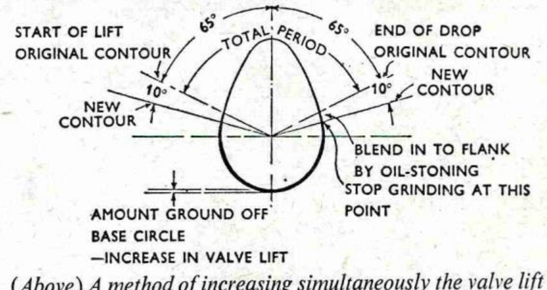
The graph on this page has been drawn to provide an easy method of finding out the C.R. given by any particular combustion space volume (or vice versa) in conjunction with cylinders of various capacities; these curves represent the



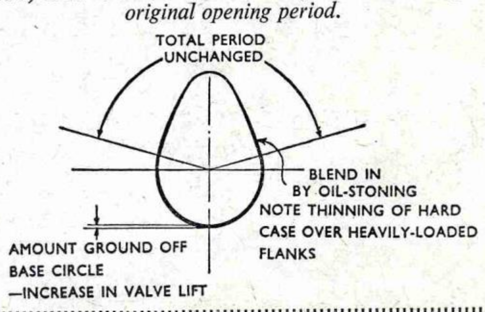
most commonly used cylinder (*not* total engine) capacities; the 125 c.c. curve, for example, applies also to 250 c.c. twins or 500 c.c. four-cylinder engines. To find the ratio given by any known combustion chamber volume, read across from the volume figure to the curve of the cylinder capacity concerned and down to the ratio line. Conversely, to find what compression space is required for, say, $7\frac{1}{2}$ to 1 in a 250 c.c. cylinder, read up from 7.5 on the ratio line to the 250 curve, then across to the volume line (38 c.c.). If it is desired to raise the ratio to 8, for which the volume should be $35\frac{1}{2}$ c.c., the second graph shows packing or material removal required.



Improved Cylinder Filling



(Above) A method of increasing simultaneously the valve lift and period of opening by grinding the cam base circle. (Below) How to obtain increase in lift whilst retaining the original opening period.



Power Requirements

The power required to propel a vehicle depends upon two main factors—rolling resistance and air resistance. The former depends upon the gross vehicle weight, the state of the road surface and the size and inflation pressure of the tyres, but it remains substantially constant over a wide range of speed provided the factors stated do not vary. Air resistance depends upon the frontal area and the aerodynamic shape, and increases as the square of the speed. Therefore, the power absorbed in overcoming it increases as the *cube* of the speed, whereas that required to overcome rolling resistance increases only directly as the speed. In consequence, on a good surface, the power absorbed by wind pressure at speeds in the region of 100 m.p.h. is nearly 10 times that absorbed by rolling resistance, hence the necessity for adopting the most compact riding position possible. The air resistance can be found from the formula:

$$\text{Resistance (pounds)} = A.V^2.C$$

where A = the frontal area in square feet.

V = velocity in feet per second.

C = a constant called the " drag coefficient."

A, the frontal area, is difficult to determine accurately for such an irregular shape as a motorcycle and rider, but it can be taken as being about 5 sq. ft. for a rider of the average physique, lying down on a Senior T.T. machine; it would obviously be less, perhaps $4\frac{1}{2}$ sq. ft., for a small man on a 125 c.c. model. For the factor C, a figure of .0008 has been found by experiment to give results which tally reasonably well with known performance figures, though it may be slightly less for a clean design of machine with the rider exceptionally well tucked in.

Using these figures, the air resistance of a Senior T.T. model at 124 m.p.h. (equal to 182 ft. per sec.) is

$$5 \times 182^2 \times .0008 = 132 \text{ lb.}$$

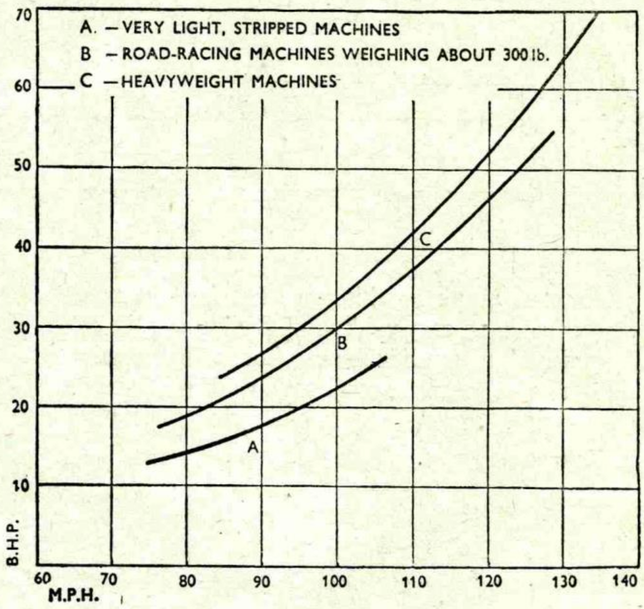
Taking the total all-up weight as 520 lb. and rolling resistance as 2 per cent., which is a good average figure to employ, then the rolling resistance is approximately 10 lb., a very small figure compared to the air resistance. The total resistance is therefore 132 plus 10=142 lb.

The horse-power required to overcome a known resistance as a known speed is found from the expression

$$\text{H.P.} = \frac{\text{resistance} \times \text{feet per second}}{550}$$

therefore, in this instance, $\text{H.P.} = \frac{142 \times 182}{550} = 47.$

This is the power required at the rear tyre; assuming 96 per cent. transmission efficiency, 49 b.h.p. would have to be developed by the engine to achieve the quoted speed of 124 m.p.h. This result agrees very closely with the power development and maximum speed of known racing machines, but there is always bound to be a certain amount of guesswork in estimating the factors involved. The drag coefficient C for instance can be brought down to .0003 by full streamlining and though this usually results in a simultaneous increase in frontal area, the net gain is very considerable. The accompanying graph has been compiled from a number of known performance figures and gives a fairly accurate guide to power requirements; the lower curve is for light, small-capacity machines in track condition, the middle curve being for heavier machines in road-racing trim, and the upper curve is for large, heavy mounts in the big-twin class. With the aid of this graph and the performance



In graph form this approximate guide to the tie-up between b.h.p. and m.p.h. gives the tuner an idea of the results his efforts should produce—always bearing in mind wind and other resistance factors.

figures of the engine an estimate of the speed attainable and of the gear-ratio to employ can be obtained which will be sufficiently close to serve as a " jumping-off place " pending final settlement on the course.

Power Calculations

$$\text{B.M.E.P.} = \frac{\text{B.H.P.} \times \text{C}}{\text{R.P.M.}}$$

C depends upon size and type of engine, though not upon the number of cylinders, and is as follows:

For 250 c.c. 4-strokes	52460
350 c.c. 4-strokes	37480
500 c.c. 4-strokes 250 c.c. 2-strokes	26230
350 c.c. 2-strokes	18740
1,000 c.c. 4-strokes 500 c.c. 2-strokes	13125

For other capacities the value of C can be obtained by direct proportion.

$$\text{B.H.P.} = \frac{\text{PLAN}}{33{,}000}$$

where P = Brake Mean Effective Pressure (B.M.E.P.).

L = Stroke in feet.

A = Area of one piston in square inches.

N = Number of explosions per minute.

$$\text{Torque} = \frac{\text{B.H.P.} \times 5{,}250}{\text{R.P.M.}}$$

Therefore at 5,250 r.p.m. the torque in lb. per ft. is numerically equal to the b.h.p.

Froude Brake Calculations

$$\text{B.H.P.} = \frac{\text{P} \times \text{R.P.M.}}{\text{C}}$$

where P = Pull in pounds shown on dial.

C = A constant, usually either 4,500 or 5,500, depending on type of brake.

To find torque or B.M.E.P. directly from the "pounds-pull" reading, multiply by the following factors, irrespective of the number of cylinders in the engine:

	ENGINE SIZE	BRAKE CONSTANT 4,500	BRAKE CONSTANT 5,500
TORQUE	ALL CAPACITIES	1.167	.965
B.M.E.P.	250 c.c. 4-str.	11.56	9.46
	350 c.c. 4-str.	8.26	6.76
	500 c.c. 4-str. / 250 c.c. 2-str.	5.78	4.73
	350 c.c. 2-str.	4.13	3.38
	1,000 c.c. 4-str. / 500 c.c. 2-str.	2.89	2.36

Example: If a 500 c.c. four-stroke engine is pulling 30 lb. at 5,000 r.p.m. on a 4,500 constant brake, it is developing

$$\frac{30 \times 5{,}000}{4{,}500} = 33.3 \text{ B.H.P.}$$

Its torque is 30 × 1.167 = 35 pounds-feet.

Its B.M.E.P. is 30 × 5.78 = 174 pounds per sq. in.

Atmospheric conditions exert quite a large effect on power, which increases when the barometer reading is high, or when the air temperature is low, and vice versa. For that reason, it is always advisable, particularly in a changeable climate, to correct all power readings to N.T.P.—i.e. normal temperature and pressure, which are respectively internationally fixed at 15° C. and 29.92 in. mercury. Correction is made by the following formula:

$$\text{Corrected B.H.P.} = \frac{\text{Observed B.H.P.} \times 29.92}{\text{barometer reading}} \times \frac{400 + \text{air temp. (degrees C.)}}{415}$$

From this it will be seen that a drop of 1 in. in the barometer reading lowers the horse-power by approximately 3%, and although the effect of temperature change is less important over the ranges normally met with in an enclosed room with the engine running, it is certainly advisable to correct the readings fully to avoid subsequent confusion.

Other figures, such as the torque of the engine and its B.M.E.P. can also be obtained from the brake test.

Engine Calculations

$$\text{Mean Piston Speed (ft. per min.)} = \frac{\text{R.P.M.} \times \text{stroke in inches}}{6}$$

$$\text{or } \frac{\text{R.P.M.} \times \text{stroke in mm.}}{152.4}$$

$$\text{Mean Gas Velocity Through Port (ft. per sec.)} = \frac{\text{Piston speed}}{60} \times \frac{D^2}{d^2}$$

$$\text{Mean Gas Velocity Through Valve Seat (ft. per sec.)} = \frac{\text{Piston speed}}{60} \times \frac{D^2 \times 22}{d_v \times L \times 7}$$

where D = Diameter of piston. d_v = Diameter at throat of valve.

d = Diameter of port. L = Lift of valve.

(Either in. or mm. can be used on the right-hand side of these equations, provided the same scale is used for each of the dimensions used.)

Inertia Load of Reciprocating Parts

$$\text{Load at T.D.C. (in pounds)} = .0000142WN^2S\left(1 + \frac{S}{2L}\right)$$

$$\text{Load at B.D.C.} = .0000142WN^2S\left(1 - \frac{S}{2L}\right)$$

where W = Weight of components in pounds.

N = R.P.M.

S = Stroke in inches.

L = Length of connecting-rod in inches.

Centrifugal Load on Crankpin

$$\text{Load} = .0000142WN^2S$$

where W = Weight of big-end and rollers.

N = R.P.M.

S = Stroke in inches.

Road Speed Calculations

To calculate speed in m.p.h. when time and distance factors are known:

$$\frac{\text{Distance (in miles)}}{\text{Time (in seconds)}} \times 3{,}600$$

For speed over quarter mile, divide 900 by the time in seconds.

Speed & R.P.M. for 3.25-in. and 3.50-in. x 19-in. Racing Tyres

R.P.M.

m.p.h. / Gear ratio	30	40	50	60	70	80	90	100	110	120	130	140
3.5	1,382	1,844	2,304	2,765	3,226	3,688	4,146	4,608	5,068	5,350	5,990	6,452
3.6	1,422	1,895	2,370	2,843	3,316	3,790	4,264	4,740	5,210	5,686	6,160	6,632
3.7	1,462	1,948	2,436	2,930	3,410	3,896	4,386	4,872	5,360	5,846	6,334	6,820
3.8	1,500	2,000	2,500	3,002	3,502	4,002	4,503	5,003	5,503	6,004	6,504	7,004
3.9	1,540	2,054	2,566	3,080	3,593	4,108	4,620	5,134	5,646	6,160	6,673	7,186
4.0	1,580	2,106	2,634	3,160	3,687	4,212	4,740	5,266	5,794	6,320	6,847	7,374
4.1	1,620	2,160	2,700	3,240	3,780	4,320	4,860	5,400	5,940	6,480	7,020	7,560
4.2	1,660	2,212	2,764	3,318	3,870	4,424	4,978	5,530	6,082	6,638	7,188	—
4.3	1,698	2,264	2,830	3,397	3,963	4,528	5,096	5,660	6,277	6,794	7,360	—
4.4	1,738	2,318	2,896	3,476	4,055	4,636	5,214	5,792	6,372	6,952	7,530	—
4.5	1,777	2,370	2,964	3,555	4,148	4,740	5,332	5,925	6,520	7,110	—	—
4.6	1,817	2,422	3,030	3,634	4,240	4,846	5,450	6,057	6,664	7,268	—	—
4.7	1,856	2,476	3,094	3,713	4,332	4,952	5,570	6,188	6,807	7,426	—	—
4.8	1,896	2,528	3,160	3,792	4,424	5,056	5,688	6,320	6,952	7,584	—	—
4.9	1,935	2,580	3,226	3,871	4,515	5,161	5,805	6,452	7,096	7,742	—	—
5.0	1,975	2,634	3,292	3,950	4,608	5,267	5,925	6,584	7,242	—	—	—
5.1	2,015	2,686	3,360	4,030	4,702	5,372	6,045	6,716	7,388	—	—	—
5.2	2,054	2,738	3,424	4,108	4,793	5,476	6,162	6,847	7,532	—	—	—
5.3	2,093	2,792	3,490	4,187	4,885	5,584	6,280	6,979	7,677	—	—	—
5.4	2,133	2,844	3,556	4,266	4,977	5,688	6,400	7,110	7,822	—	—	—
5.5	2,172	2,896	3,620	4,345	5,070	5,792	6,517	7,241	—	—	—	—
5.6	2,212	2,950	3,686	4,424	5,160	5,898	6,636	7,372	—	—	—	—
5.7	2,251	3,002	3,752	4,503	5,253	6,004	6,754	7,504	—	—	—	—
5.8	2,291	3,054	3,818	4,582	5,345	6,109	6,873	7,636	—	—	—	—
5.9	2,330	3,108	3,884	4,661	5,438	6,216	6,990	7,768	—	—	—	—
6.0	2,370	3,160	3,950	4,740	5,530	6,320	7,110	7,900	—	—	—	—

Difference Table

Gear ratio	1	2	3	4	5	6	7	8	9
3.5	46	92	138	184	230	277	323	367	415
3.6	47	95	142	190	237	284	332	379	426
3.7	49	97	146	195	244	293	341	390	439
3.8	50	100	150	200	250	300	350	400	450
3.9	51	103	154	205	257	308	359	411	463
4.0	53	105	158	211	263	316	367	421	474
4.1	54	108	162	216	270	324	378	432	486
4.2	55	111	166	221	276	332	387	442	498
4.3	57	113	170	226	283	340	396	453	510
4.4	58	116	174	232	290	348	405	464	521
4.5	59	118	178	237	296	355	415	474	533
4.6	61	121	182	242	303	363	424	485	545
4.7	62	124	186	248	309	371	433	495	557
4.8	63	126	190	253	316	379	442	506	569
4.9	65	129	193	258	323	387	452	516	580
5.0	66	132	197	263	329	395	461	527	592
5.1	67	134	201	269	336	403	470	537	604
5.2	68	137	205	274	342	411	479	548	616
5.3	70	140	209	279	349	419	488	558	628
5.4	71	142	213	284	356	427	498	567	640
5.5	72	145	217	290	362	434	507	579	652
5.6	74	148	221	295	369	442	516	590	664
5.7	75	150	225	300	375	450	525	600	675
5.8	76	153	229	305	382	458	534	611	687
5.9	78	155	233	311	388	466	544	622	699
6.0	79	158	237	316	395	474	553	632	711

Note.—For 3.25-in. and 3.50-in. × 20-in. tyres *subtract* 4% from r.p.m. figures. For speeds intermediate between even tens, *add* r.p.m. from "Difference Table." For ratios lower than 6:1 select any sub-multiple and multiply r.p.m. accordingly, i.e., for 12:1, select speed for 6:1 and multiply by 2.

Average Speed Table (1 Kilom. s.s.)

Time secs.	Speed m.p.h.	Time secs.	Speed m.p.h.	Time secs.	Speed m.p.h.
21.00	106.52	25.00	89.48	29.00	77.14
.2	105.52	.2	88.77	.2	76.61
.4	104.53	.4	88.07	.4	76.09
.6	103.56	.6	87.38	.6	75.57
.8	102.61	.8	86.70	.8	75.07
22.00	101.68	26.00	86.04	30.00	74.56
.2	100.76	.2	85.38	.2	74.07
.36	100.04	.4	84.73	.4	73.58
.4	99.86	.6	84.10	.6	73.10
.6	98.98	.8	83.47	.8	72.63
.8	98.11	27.00	82.85	31.00	72.16
23.00	97.26	.2	82.24	.2	71.70
.2	96.42	.4	81.64	.4	71.24
.4	95.60	.6	81.05	.6	70.79
.6	94.79	.8	80.47	.8	70.34
.8	93.97	28.00	79.89	32.00	69.90
24.00	93.21	.2	79.32	.2	69.47
.2	92.44	.4	78.77	.4	69.04
.4	91.68	.6	78.21	.6	68.62
.6	90.93	.8	77.67	.8	68.20
.8	90.20				

Conversion Factors

To convert—	Multiply by—
Miles to kilometres	1.609 (or 8/5 approx.)
M.p.h. to k.p.h.	1.609 (or 8/5 approx.)
Kilometres to miles	.621 (or 5/8 approx.)
K.p.h. to m.p.h.	.621 (or 5/8 approx.)
Gallons to litres	4.536 (or 4½ approx.)
Litres to gallons	.2205 (or 2/9 approx.)
Pounds to kilograms	.4536 (or 9/20 approx.)
Kilograms to pounds	2.205 (or 11/5 approx.)
Cubic centimetres to cubic inches.. ..	.061 (or 1/16 approx.)
Cubic inches to cubic centimetres.. ..	16.39 (or 33/2 approx.)
M.p.h. to feet per second	88/60 (or 1½ approx.)

NEXT WEEK'S SUPPLEMENT

The second part of our Performance Data Charts will deal with carburetter types and settings, fuel analysis tables, valve settings and complementary data.

Make sure of your copy by ordering *Motor Cycling with Scooter Weekly* **from your newsagent today.**

The information given in these data charts is extracted from the 294-page book "Tuning for Speed" by P. E. Irving, M.I.Mech.E., M.S.A.E., published in association with *Motor Cycling with Scooter Weekly* by Temple Press Limited, Bowling Green Lane, London, E.C.1, from whom copies may be obtained price 9/4d. including postage; or from any bookseller, price 8/6d. net.

"DO-IT-YOURSELF SERIES" No. 9 - - - by BERNAL OSBORNE

The 192 c.c. "Flat-twin" Model LE

VELOCETTE

Practical Work on a Popular Water-cooled Lightweight of Unique Design

APPEARANCES are deceptive in that, to the complete layman, or to the owner of a conventional motorcycle, the Velocette LE model seems to be an inaccessible machine from a maintenance point of view. This impression, however, is readily disproved by study of LE component assemblies which comprise a horizontally opposed, water-cooled side-valve twin-cylinder engine with gearbox and shaft-drive transmission units built into a forward sub-frame carrying the radiator. Superimposed is a pressed-steel "body," accommodating a deep-slung fuel tank and battery carrier. With standard tool-kit equipment only, the radiator and the upper part of the 1956-type frame structure can be detached, leaving the complete "works" exposed and ready for workshop operations.

Actually, this seemingly extensive task can be carried out in less than 20 minutes; further, the removal of five nuts suffices to split the engine and gearbox units, revealing the clutch assembly—a job which can be carried out easily in a further 10 minutes. Therefore, the LE, so far from having inaccessible engine and gearbox components is, in fact, more conveniently and quickly dismantled than are many more simple-looking machines of conventional character. Normal running adjustments and decarbonization can, of course, be carried out with the "body" in situ.

Special Tools

The manufacturers list over two dozen appliances of this kind. Probably the private owner, carrying out a general overhaul, could get by with three or four and, in some cases, could fabricate a device similar to the appropriate factory tool, or fashion a substitute. In the following paragraphs I have listed the tools officially recommended but have kept the total to a minimum. Incidentally, I understand that members of the LE Owners' Club have certain facilities in this respect. The club, apparently, possesses a stock of special tools available to members on a loan-against-deposit basis. Undoubtedly most Velocette dealers would offer similar help.

Dismantling Procedure

Separating the "body" and front fork assembly from the engine gearbox and rear transmission units necessitates draining the radiator and disconnecting the water pipes, the fuel line, all cable controls, the battery and the pulling out of the plug-type electrical connections. Take off the gear-change lever knob and unscrew the lever; also unscrew and remove the two long-headed hexagon bolts which anchor the tops of the rear suspension units. Elevate the machine slightly by means of a support beneath the sump. Slacken the four bolts—two each side—securing the "body" at a point adjacent to the rear swinging-fork hinge-pin. Two crescent-shaped nut-plates will be released by this operation. Now remove both lower legshield fixing bolts, which serve also to lock the mitred joint in each of the lower frame lugs, and spring the shields clear of the protruding ends of the lower cross tube on each side. It should now be possible—after some 20 minutes' work—to lift from the rear and move the "body" forward on the front wheel, leaving the engine/gearbox and transmission components intact.

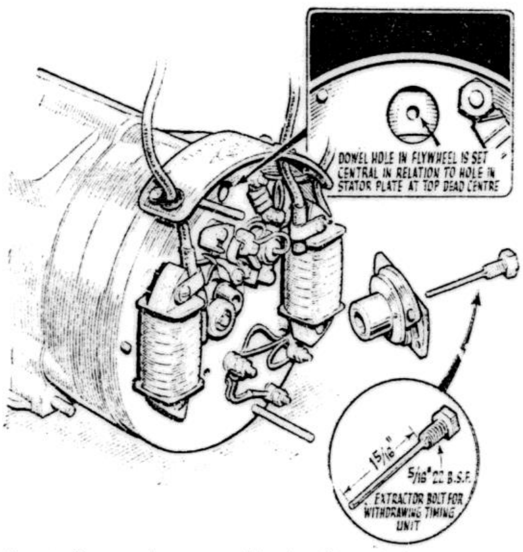
One now has free access to the cylinder heads, cylinders and carburetter, and stripping-down can commence, although it is wise to leave the barrels and pistons in position so that the connecting rods have a measure of support, pending removal at a later stage.

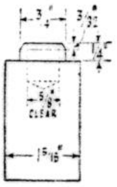
Cursory inspection shows that removal of the five nuts (two on each side of the clutch housing and one on top) suffices to split the engine and gearbox assemblies. The clutch is now exposed and, if it is to be dismantled, the complete clutch-housing assembly should be parted from the crankcase. To do this, take off the two nuts at the base—then be careful to keep your hands clear, because as the parts separate the starter spring is released and suddenly protrudes. To dismantle the clutch, the opposite end of the shaft must be supported on a flat surface and the springs depressed while the eight nuts and lock-washers are removed.

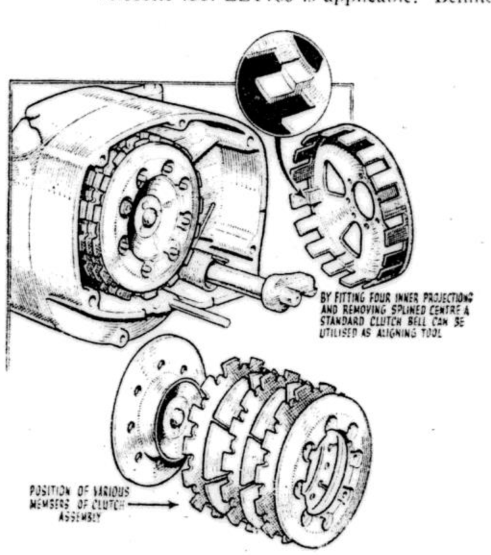
The eight nuts secure the clutch assembly to eight driving pillars on the clutch back-plate and the pillars transfer torque to three intervening friction discs which are serrated to engage with the driven bell-member—the part still left assembled on the withdrawn gearbox.

Reduction Gear

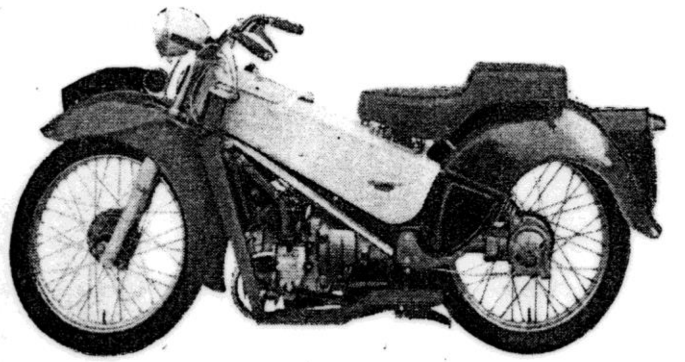
When the clutch plates have been removed it will be found that the back-plate is secured by a ring nut on the splined shaft of the reduction gear—which is actually a form of primary drive, gearing down clutch speed from the rate of engine r.p.m. This reduction gear, which is helical-cut, is located ahead of the clutch housing and meshes with the mainshaft driving pinion. That pinion, too, is secured by a ring nut to which Velocette tool LET780 is applicable. Behind







(Above) Stator plate assembly details and the LE tool 948, used to extract the automatic advance component. (Right) Accurate clutch assembly is aided by a bell member modified to act as an alignment tool. (Inset, centre) A fitting tool for the gearbox end-cover oil seal.



With its near-scooter degree of enclosure, "silent" water-cooled power unit and car-like driving characteristics, the LE Velocette is undoubtedly the most original of British post-war motorcycles. The 1956 machine shown is in the new two-colour finish.

the timing-gear pinion is an outrigger plate, supporting the front end of the clutch shaft and the timing-pinion collar.

With dismantling work almost completed at the rear of the engine, it is feasible to remove the timing unit (by means of tool LET948, an extractor bolt) and to take off the stator plate and Miller flywheel. A $\frac{1}{2}$-in. Whitworth nut locates the flywheel and a tubular spanner of suitable size does this work efficiently. Withdrawing the flywheel, however, calls for the application of a puller tool, with a muff threaded internally $1\frac{1}{2}$ in., 26 t.p.i., to fit the thread of the flywheel boss. This tool is listed as LET949, with a subsidiary part LET647/1, this being a cap designed to fit over the end of the shaft to avoid the risk of damage as the centre-bolt tightens.

Surrounding the flywheel is the flywheel housing which, when removed, leaves an aperture for the withdrawal of the big-end assembly and con.-rods. That final phase, of course, must be preceded by dismantling of cylinder barrels, pistons and rings.

Incidentally, when working on the forward mainshaft you will note that there is a keyway. This is a legacy of the old LE series in which a B.T.H. generator was used; the Miller generator flywheel is simply a taper-fit on the shaft. Before the big-end assembly can be withdrawn it will be necessary to extract the oil pump (operated by a worm gear on the mainshaft), and for this purpose, automatically, the sump must be taken off.

Complicated? Not really; in fact, recapping briefly, and for clarity omitting detail which becomes obvious as you proceed, there is a simple dismantling sequence:—

(1) Gearbox.
(2) Clutch housing, complete with the clutch and reduction-gear wheel.
(3) Reduction gear pinion at rear end of mainshaft and cam gear.
(4) Timing unit.
(5) Stator plate and flywheel.
(6) Cylinders and pistons.
(7) Sump and oil pump.
(8) Withdraw crank/con.-rods assembly.

A complicated arrangement of oil pipes is revealed by the removal of the sump base component. Study the connections and, if in doubt, avoid mistakes later by labelling each of the pipes. The joints are silver soldered, and annealing to make the pipes pliable during assembly can be carried out without risk of damaging them.

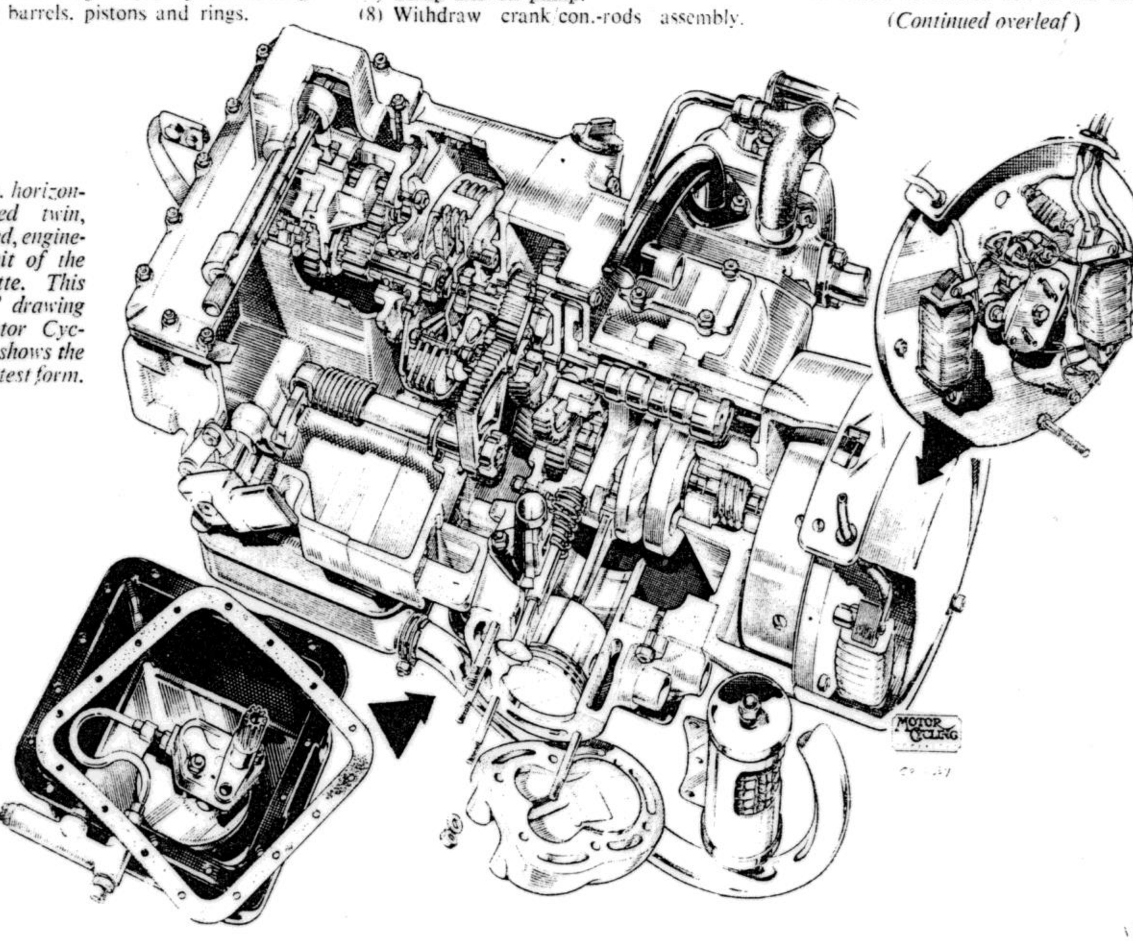
Normally, in this series, advice to go ahead with rebushing and similar work follows dismantling details, but in the case of the LE there are technical considerations which create difficulties. The small-end bushes are probably the only parts of their kind which can be renewed without special equipment. Pay particular attention to the Veloce requirement that one end face of the bush must stand out a stipulated distance beyond the end face of the big-end eye. For engines up to number 200/22677 the dimension is .236 in. Subsequent models have wider big-ends and, in this case, the end-face overhang is only .205., the con.-rod thickness being $\frac{1}{16}$ in. greater. The nominal $\frac{1}{2}$-in. bush bore must be reamed to $\pm$ .0025 in.

Support for the crankshaft is by four bushes. Two are in the flywheel housing, one in the crankcase and one in the reduction-gear plate. All of them are line-reamed after fitting, and here special equipment is required. Another difficulty arises from the fact that the big-end bushes are split and cannot be fitted satisfactorily without application of Velocette bushing fixture LEF 1267. And then, of course, there is the further complication of accurate reaming and the need for yet another Veloce fixture, LEF 1268.

Work can be carried out on the crankpin

(Continued overleaf)

The 192 c.c. horizontally-opposed twin, water-cooled, engine-gearbox unit of the LE Velocette. This "cut-away" drawing by a "Motor Cycling" artist shows the unit in its latest form.



"DO-IT-YOURSELF" (LE VELOCETTE) (Continued from previous page)

sleeves but, once more, factory service is probably a better bet in that replacement crankpin sleeves are graded for selective assembly from "AA" (smallest) to "A," "B" and "C" (largest).

Cylinder, piston and valve components—including guides, which are of cast-iron material and are pressed in—can be dealt with by the owner without undue technical difficulty.

Assembly

A reversal of the eight-point dismantling sequence provides a good guide.

In-line markings on the faces of the cam gears aid correct valve setting; note that the camshaft should have from .003/.007 in. float, this being controlled by the thrust button and packing shims fitted to the reduction gear plate.

To establish a basis for spot-on ignition timing is less simple. Veloce, Ltd., have a rig (LET 953) which locks the near-side piston at a point half-way along its travel and, at the same time, a dowelled setting plate (LET 952) determines a flywheel position giving maximum magnetic flux at the firing point. Emergency starting will probably not be satisfactory unless this juxtaposition is preserved. With the stator plate assembled, the flywheel dowel hole registers centrally in the hole at the top of the stator plate, the timing device being retarded and the contact-breaker points about to break at that stage. With the timing nut on the taper, check and set the points before retiming.

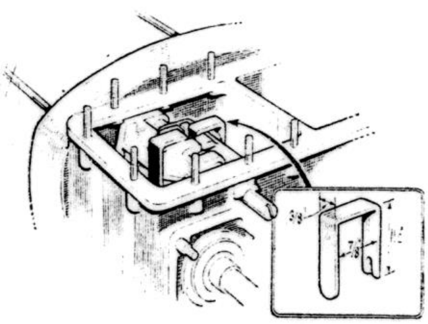
Transmission

Access to the gearbox is by way of the bolted-on top cover. Set the gears in neutral and, with the clutch side of the box towards you, push back both selector forks. This will lock the gears while the clutch bell member, retained by a ring nut and tab washer, is removed. Now take off the selector fork rod nut on the gearbox front cover and extract the rod, catching the spring-loaded selector-fork spring plungers as they are freed. Five nuts retain the end cover and if the universal joint is disconnected the "internals" can be withdrawn together with the front cover. Take care not to damage the oil seals. A cross-member frame, or carrier, bolts to the back of the gearbox and acts as a mounting for the die-cast, light-alloy swinging-fork member, in the near side of which is housed the universal joint and the propeller shaft.

Again an array of special extractors, key-type box spanners, mandrels and jigs are called for when taking down, or replacing, these final transmission parts, which include the bevel gears, and the Works emphasize the advantage of using their facilities rather than causing damage by improvisation.

Suspension

The front forks are of simple design, each leg comprising a main stanchion, with one bush at the bottom and one in the bore of a slider. The stanchions are drilled with a single hole to allow oil put into the slider on assembly to lubricate the bushes.

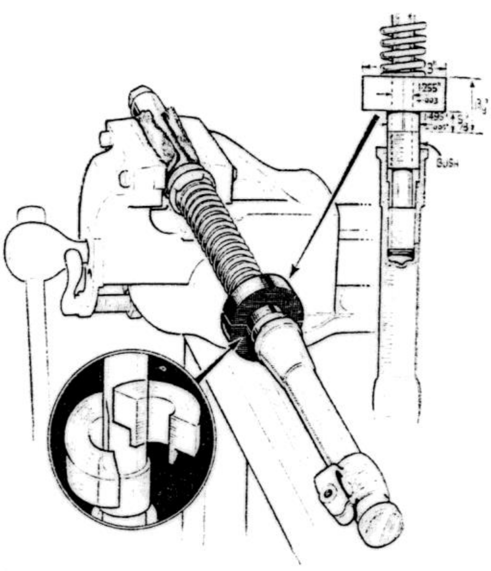


Replacing the plunger and spring in the selector sliders is made easy by the use of a special retainer tool, which may be made up to the dimensions shown. When in position, it holds back the plunger and spring.

Positioning the Spring

A clamp-type appliance, LET796 (see detailed sketch), is used to position the spring and to act as a "base" for the slider as it is tapped into position, the stanchion being secured in a vice as illustrated. At both ends, the helices engage with a scroll and the spring works both in compression and in tension.

At the rear, each "leg" carries a single coil spring also operating in compression and tension, with the top-end engaging in a slot and being movable to give hard or soft reaction to impact.



Dimensions of the LE tool 796 and its application in fitting the slider to the main stanchion.

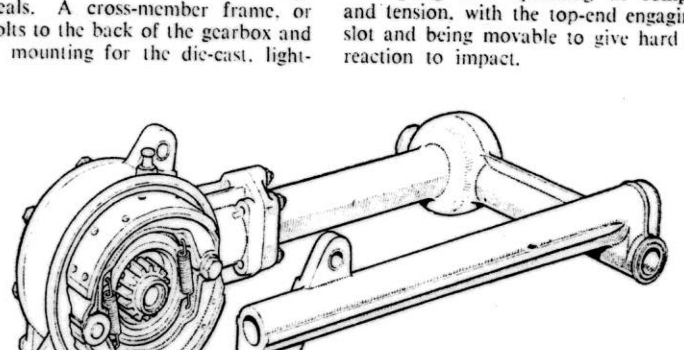
Lubrication

Wet-sump in principle, the system provides for 1¼ pints of oil carried in a reservoir at the base of the crankcase, whence it is picked up by a gear-type pump and delivered via the release valve (mounted externally) to the filter and then to the copper-piped distribution assembly in the crankcase cavity. From that point, direct feeds go to the big-end and main bearings, the camshaft bushes and gears, and the reduction gears. There is a feed also to the timing pinion bearing collar and clutch-shaft bearing, but splash from the rotating crankshaft is relied upon for cylinder-wall lubrication. Note that the filter element should be renewed—not, in any circumstances, cleaned—at 10,000-mile intervals: and that when this work is carried out the filter chamber, which is part of the pressure system, must be primed, otherwise no lubricant will flow when the engine is started.

Silencers

The right-hand side cylinder tends to get slightly more oil than that on the left, and after prolonged use the silencer or exhaust on the right only may become more heavily carbonned than its "opposite number." This has been known to cause misfiring on one cylinder, and has proved rather baffling until the reason was found. In cases where such misfiring occurs on machines that have covered big mileages, it would be advisable to suspect the exhaust system rather than the ignition side.

This article essentially condenses the wealth of detail contained in the 86-page LE service manual—a publication which anybody contemplating a serious LE overhaul would obviously study—but, at the same time, it is hoped that this instalment in the series will be a guide to the new, and perhaps non-technically minded, owner whose imagination may boggle at the thought of LE maintenance—for the reasons stated in the opening paragraph.



The die-cast, light-alloy swinging-fork rear member of the 1955-6 LE. The universal joint and propeller shaft are housed in the near side.

REFERENCE DATA

192 c.c. Flat-twin Water-cooled LE Velocette

CYLINDER—PISTON GROUP

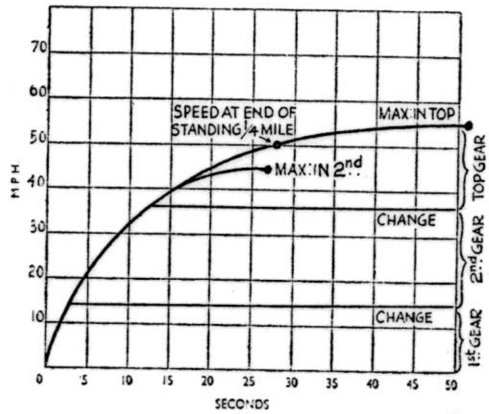
Bore: 50 mm.
Stroke: 59 mm.
Swept volume: 192 c.c.
Compression ratio: 7 : 1.
Rebore to +.020 in. or +.040 in.
Piston diameters:
At top land, 1.961 in.—.003 in.
At bottom land, 1.9655 in. ±.0005 in.
At skirt, 1.967 in. ±.0005 in.
Piston ring gap: .007/.012 in.
Piston ring groove depth: .0735 in.
Permissible vertical play: .0025/.001 in.
Gudgeon pin diameter: .499 in. —.0002 in.
Small-end bush diameter: .5 in. ±.00025 in.

VALVES AND VALVE GEAR

Valve stem diameter: .216 in. —.0005 in.
Bore of valve guides: .218 in. ±.0002 in.
Seat angle: 45°.
Free valve spring length: $1\frac{9}{16}$ in.
Camshaft journal diameter, front end: 1.106 in./1.1065 in. ±.0005 in.
Camshaft journal diameter, gear end: 1.264/1.2645 in. ±.0005 in.
Valve timing (with tappets set at .006 in.):
Inlet opens before T.D.C. 20°.
Inlet closes after B.D.C. 55°.
Exhaust opens before B.D.C. 55°.
Exhaust closes after T.D.C. 20°.
Normal tappet clearances: Inlet, .004 in.; exhaust. .006 in.

CRANKSHAFT GROUP

Crankpin track diameter: 1.377 in. —.0005 in.
Con-rod big-end eye diameter: 1.3775 in. +.0005 in.
Permissible side play: .003/.008 in.
Mainshaft end play: .003/.008 in.
Left-hand thread on engine components: None.
Location of contact breaker: On stator plate in Miller generator.



Road test performance chart for the LE Velocette, first published in "Motor Cycling" of February 3, 1955. Fuel consumption figures registered at the same time were 118 m.p.g. at 30 m.p.h., 100 m.p.g. at 40 m.p.h. and 82 m.p.g. at 50 m.p.h.

GEARBOX

Gearbox bearings: Ball type Hoffmann S7 (3 off). Hoffmann LS7 (1 off) at rear end of secondary gear shaft.
Internal reduction: .74, 1.1, 2.07 to 1.
Left-hand thread on gearbox: None.

TRANSMISSION

By universal joint, propeller shaft, crown wheel and pinion with a 3 : 1 reduction ratio.
Overall gear ratios: 7.3, 10.9 and 20.4 : 1.

WHEELS

Front: WM1–19.
Brake diameter: 5 in.
Spokes, brake side: 12 SWG by 7.875 in. (20 off).
Spokes, plain side: 12 SWG by 8.375 in. (20 off).
Hub bearings: Ball journal type Hoffmann LS8 9, $\frac{3}{4}$-in. bore by $1\frac{7}{8}$ in. O/D by $\frac{9}{16}$ in.

Rear: WM1—19.
Brake diameter: 5 in.
Spokes, brake side: 12/10 SWG by 7.687 in. (20 off).
Spokes, plain side: 12/10 SWG by 8.375 in. (20 off).
Hub bearings: As front wheel.

FRONT SUSPENSION

By telescopic forks, each with a single-coil spring, hydraulically damped, carried on cup-and-cone-type head bearings, with $\frac{1}{4}$-in. balls top and bottom (19 off). Head bearing race pitch circle 1.6 in., top and bottom.
Fork angle: 27°.
Trail: $2\frac{11}{16}$ in.
Damper fluid content: $\frac{1}{8}$ pint (2.5 fl. oz.) S.A.E. 30 oil.
Slider bush dimensions: Upper, bore 1.244 in. (—.001 in.) by 1.5025 in. O/D (—.001 in.). Lower, bore 1.187 in. (±.0005 in.) by 1.371 in. O/D (+.001 in.).

REAR SUSPENSION

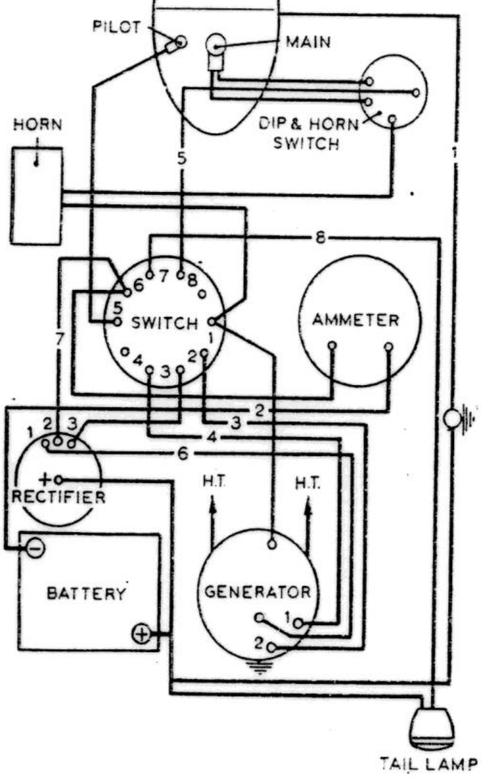
By swinging-fork with swivel adjustment for load. Pivot bush details: bore, .879 in. (—.001 in.) (flanged) by 1.066 in. O/D (+.001 in.).

CARBURATION

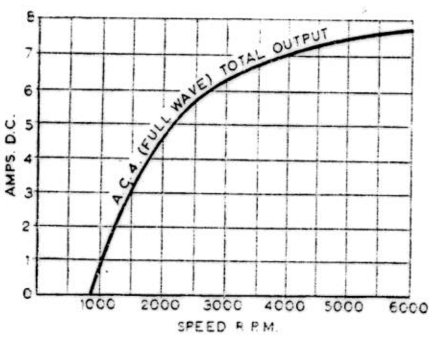
Amal fixed jet-type equipped as follows: Pilot jet, 25; pilot spray tube 20; main fuel jet 25; compensating fuel jet 20; starting jet 15. Late 1956 models are fitted with the Amal "Monobloc" light-weight type 363/1 carburetter, equipped as follows: Main jet, 65; pilot jet, 25 c.c.; choke, .475 in.; needle-jet, .045. in.; valve no. 2; needle position, no. 3.

ELECTRICAL EQUIPMENT

Miller generator, type 63E with rectifier charging Varley battery, type MC 7/12, through "Sentercel" rectifier, with alternative A.C. circuit for emergency starting. Standard bulbs fitted are: Main 6 v. 24/24 W. double filament S.B.C. cap; pilot and tail 6 v. 3 W. S.C.C. cap.



Wiring diagram of the Miller generator/rectifier set for the LE Velocette. Key to colour code:
1 = *Brown*
2 = *Grey*
3 = *Green*
4 = *Yellow*
5 = *Red*
6 = *Red and Black*
7 = *Blue*
8 = *White*



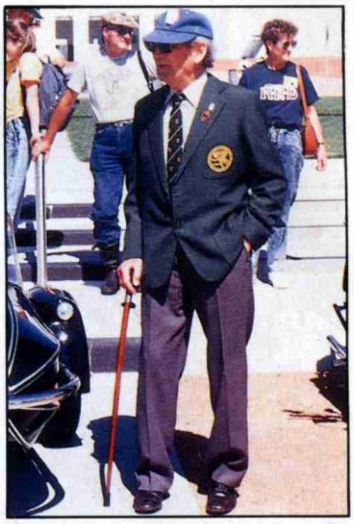
Output curve of the AC generator. The charge rate varies according to the voltage of the battery; with voltage low, the charge rate is increased and vice versa.

Why the "LE"? . . . . . . It was designed by Phil Irving !

30 Years ago …….. Cranky Cranks

NOT SO CRANKY CRANKS

Well-nigh 30 years on, a neglected Phil Irving brainwave shows what it could have done for the big parallel twin



Above: Phil Irving photographed in Australia this year

Opposite page: Nine-fifty leads eleven-fifty. On their way to second place at Brands Hatch in March 1990, Steve McFarlane and passenger Gary Hearl lead the oversize Vincent twin of Dave Mellows and Nick Archer

ARGUABLY the soundest and most imaginative engineer the erstwhile British motorcycle industry ever had was Australian Phil Irving, who in 1977 — long after the industry had sunk with scarcely a trace — was awarded the MBE for his remarkably comprehensive and versatile contribution to automotive engineering as a whole.

It borders on tragedy that Irving was also one of the industry's most underrated engineers, for which there were two basic reasons. The first was his refreshing modesty and objectivity, for he preferred to let his ideas and work speak for themselves. The second was the inability of the opinion-formers of the 1930s (such as Arthur Bourne, then editor of *The Motor Cycle*) to recognise raw engineering talent — being mesmerised instead by the bombast and cocksureness of such 'pillars' of the industry as Triumph's Edward Turner.

Surprisingly, one of the clearest proofs of this anomaly has come to light only comparatively recently with the metamorphosis in the character of Steve McFarlane's 992 cc (80.5 x 93.5mm) BSA parallel-twin classic racing sidecar outfit. It is nearly 30 years since Irving first propounded — both in the press and to the industry — a simple crankshaft modification calculated to transform the otherwise popular parallel-twin four-stroke from a mobile vibro-massage device to a civilised power unit. No manufacturer showed the slightest interest and the shakes got worse as engine capacity (hence reciprocating masses) increased. Anyone who has raced, or even tested on full throttle, both a 500cc Triumph Grand Prix and a silk-smooth BMW Rennsport flat twin of the same capacity will need no reminding of the chalk-and-cheese difference between the two.

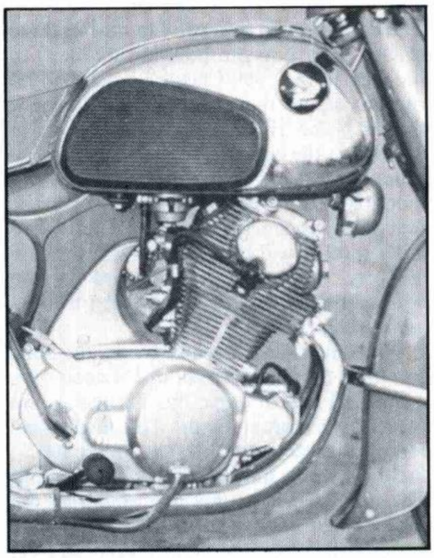
By virtue of its large capacity, McFarlane's bored and stroked A65 engine was a prime candidate for the Irving treatment. And it was Ron Valentine — the former Weslake designer and himself an Irving disciple — who suggested as much to McFarlane. Already, in 1988, McFarlane had stretched the capacity of an A65 engine to 770 cc, using a conventional 360-degree A10 crankshaft, thus enabling him and his passenger Paul Dewhurst to win that year's CRMC top novice awards. But engine vibration was so severe as to expose the weakness of the standard crankcase, which after all was never intended for such arduous use. Since he had the even larger engine in mind, to challenge the top-dog four-cylinder Imp-powered outfits (of 1,000-1,200 cc and possibly developing up to 125 bhp), it was clear that the crankcase would be decidedly vulnerable. So Valentine's suggestion was welcomed.

Peculiar though it seemed, Irving's brainwave was simply to substitute a crankshaft with two separate crankpins, spaced at 76 degrees to one another, for the conventional shaft (with pins in line). With sketches from Valentine, McFarlane had a shaft made and statically balanced by Dave Nourish, in Oakham. But when it was sent to Bassett Downe, in Swindon, for dynamic balancing, it almost jumped out of the machine. Calculating that the shaft should be balanced as two independent single-cylinder cranks rather than as a whole, Valentine gave McFarlane instructions for the necessary modifications. (Nourish himself has since acquired facilities for dynamic balancing.)

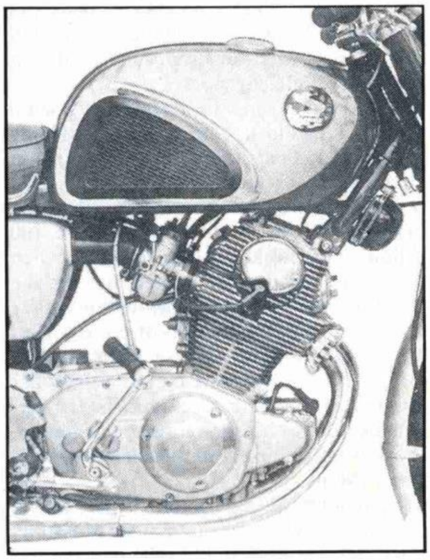
It was June last year by the time the engine was ready for action, at Lydden. Uncertain as to how it would behave, the crew asked to be allowed to start from the rear of the grid. But behave it did, for they

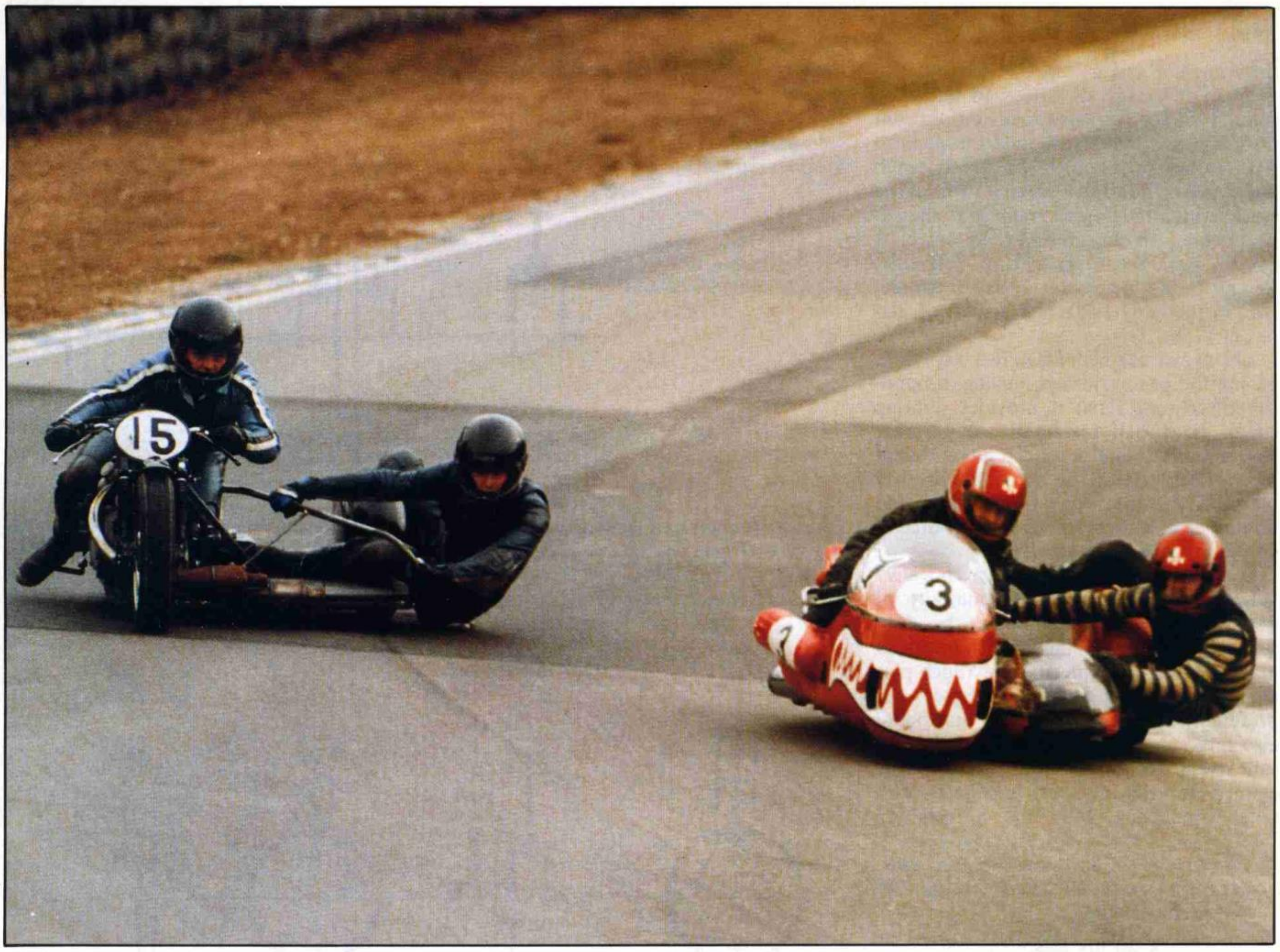
"When I briefly borrowed a Honda Dream Super Sport in 1961 I found it difficult to detect the uneveness of the firing . . . unmistakable, however, was the engine's much smoother running compared with that of the 'cooking' C72, which had a 360-degree shaft . . . 180-degree crankshaft's rocking couple could be distinctly felt in a Honda CB450 parallel twin . . . but was effectively reduced by Doug Hele's counter balanced crankshaft for the 350cc Triumph Bandit/BSA Fury"

250 HONDA C72



250 HONDA DREAM SUPER SPORT





sliced through the field to finish second. They were lying second in their next race, too, when a gremlin closed the fuel tap.

Following some drastic strengthening of the transmission to cope with the engine's huge torque, the outfit was taken to Snetterton for the Race of the Year meeting in the September. There it duly humbled the Imps to win its heat, though it was a non-starter in the final, courtesy of a burned-out ignition coil.

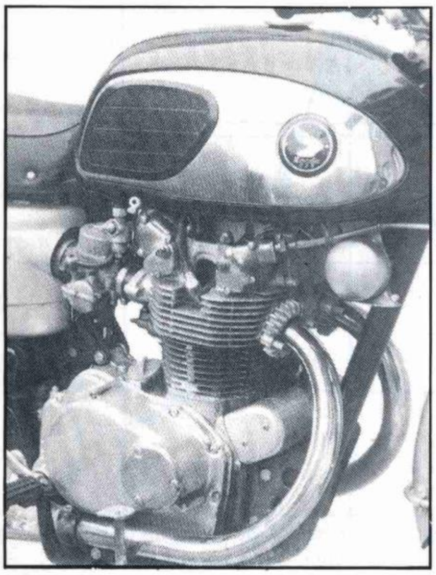
A shade undergeared in the heat, the engine spun up to 7,000 rpm on the Revett Straight, which translates to 127 mph. Yet McFarlane said it felt 'slow and lazy' — an impression that was dispelled whenever they streaked past other outfits! There was none of the frantic vibration and buzzing urgency of the 770 cc engine. Instead there was a throbbing sensation accompanied by an off-beat lilt reminiscent of a big Ducati V-twin. McFarlane was already a convert to cranky cranks for parallel twins.

Of course, the engine's commendable power and torque owed most to its greatly increased capacity, with carburettor and inlet-valve sizes to suit. But severe vibration is mechanically destructive, physically tiring and may even absorb a modicum of power. Hence its near elimination reduces both human and metal fatigue. But why 76 degrees and where did Irving's idea originate?

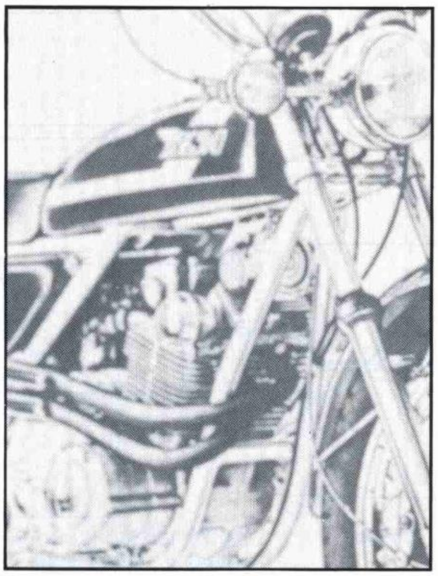
Re-reading his first article on the subject (in *Motor Cycling*, 4 July 1962) reminds me that the seeds were sown during a discussion we had in the paddock cocoa tent during practice for that year's TT, where Phil was preparing his annual technical review for *Motor Cycling* and I was doing a similar job for *Motor Cycle*. Our conversation turned to crankshaft layouts for four-stroke parallel twins, because Honda had recently adopted 180-degree shafts for their Dream Super Sport and 125 cc works racer.

Right from Alfred Angas Scott's first

450 HONDA



350 BSA BANDIT



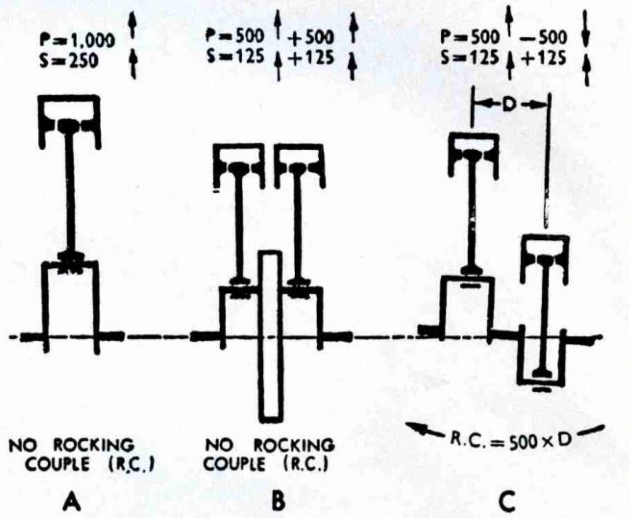
NOT SO CRANKY CRANKS

experimental engine at the turn of the century, that type of shaft had been the obvious choice for two-strokes but diligent historical research was required to unearth any four-stroke examples. (In 1914 Triumph had a 600 cc side-valve parallel twin on the stocks with a 180-degree crankshaft, though production was prevented by the outbreak of war. And, according to Ron Valentine, Peugeot's 500 cc double-knocker parallel twin in the early 1920s had a shaft of that layout.)

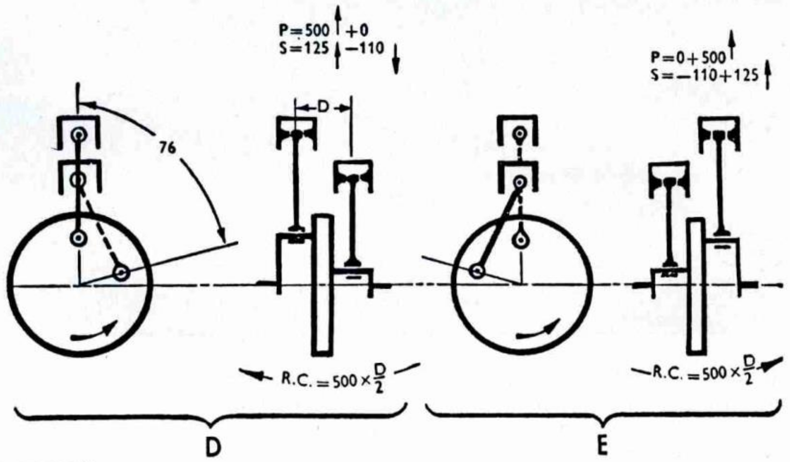
The two-stroke benefits in two ways from having the pistons always moving in opposite directions. First, it provides a firing stroke every half revolution, so ensuring very smooth torque; second, it has excellent mechanical balance because the primary inertia forces from the pistons at top and bottom dead centres (the greatest source of vibration) cancel one another.

So why was Honda's move such a novelty? Why had the overwhelming majority of four-stroke designers shunned the 180-degree layout for parallel twins in favour of arranging the crankpins in line? (Such pins are sometimes said to be at 0 degrees but preferably at 360 degrees since that figure also indicates the firing intervals.) Simply, said Irving, because of an irrational horror of uneven firing intervals. In V-twins with both connecting rods on a common

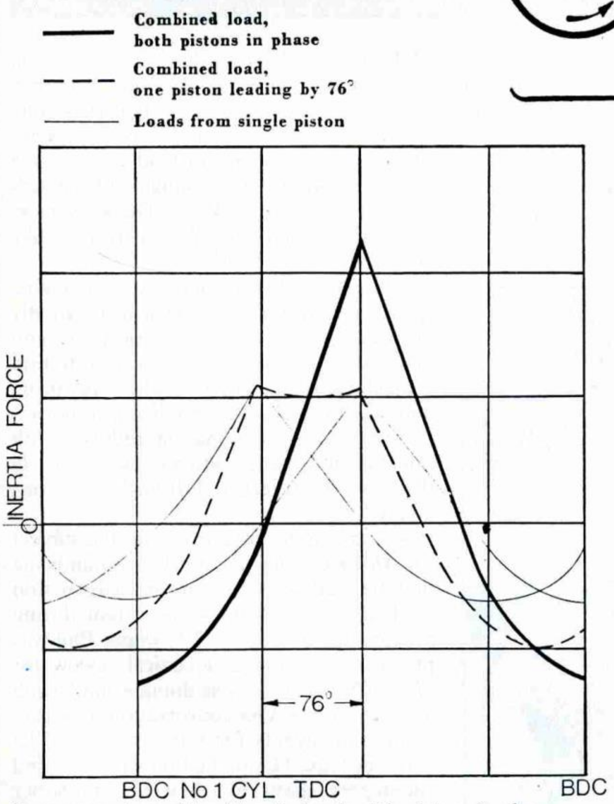
Phil Irving's 1962 diagrams (not to scale) showing four crankshaft layouts and their primary and secondary forces and rocking couples ▶



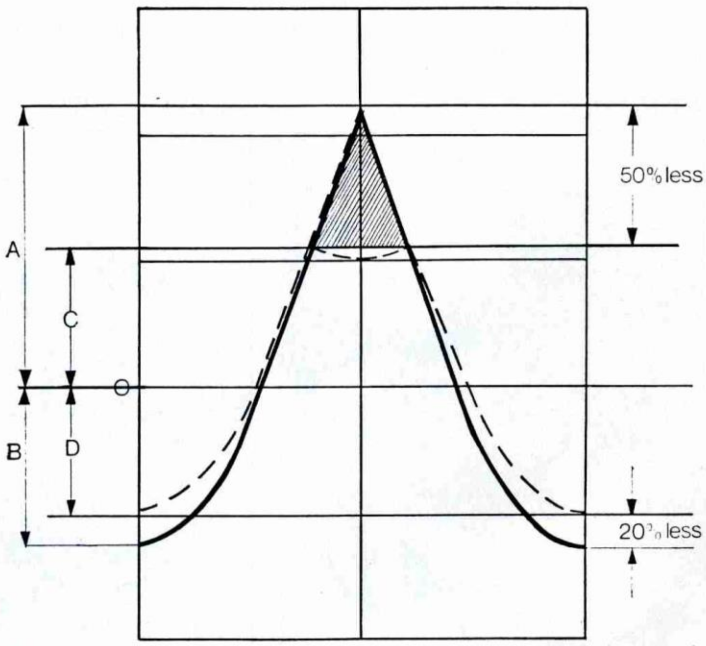
Effects of primary and secondary inertia forces in (A) the single, (B) the orthodox parallel twin, (C) the 180° twin and (D and E below) the 76° twin in two positions (P = primary force, S = secondary force; arrows show directions). If balance weights equal to 50% of piston weight were added, the primary forces would be halved, but the secondaries are unchanged



▼ *Phil Irving's 1972 graphs comparing vertical inertia forces for 360-degree and 76-degree crankshafts*

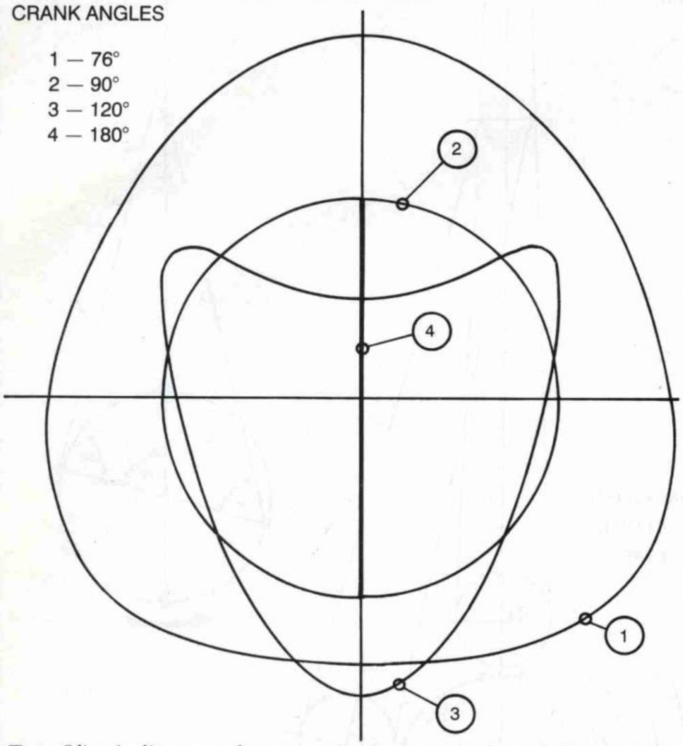


Separate and combined inertia loads with pistons in phase and at 76°



Superimposing curves from figure at left shows the reduction in forces with pistons 76° out of phase. (A is nearly double B, C is nearly equal to D)

PARALLEL TWIN
INERTIA POLAR DIAGRAMS FOR DIFFERING CRANK ANGLES



Tom Oliver's diagrams show comparative magnitude and direction of inertia forces for four different parallel-twin crankshaft layouts throughout one revolution crankpin, intervals ranging from 315/405 to 270/450 degrees had long been accepted but it was feared that the 180/540 degrees resulting from the pistons moving in opposite directions (ie, one interval three times as long as the other) would be intolerable.

Yet when I had briefly borrowed a Honda Dream Super Sport late in 1961 I had found it difficult to detect the unevenness of the firing, so well muffled was the exhaust note. What was unmistakable, however, was the engine's much smoother running compared with that of the 'cooking' C72, which had a 360-degree shaft. A more valid objection, where production cost is crucial, is the need to fit two carburettors, otherwise the wide overlap of the two inlet phases produces a strong mixture bias. (Incidentally, the longer of the two intervals, 540 degrees, necessitates a heavier flywheel than does a 360-degree twin.)

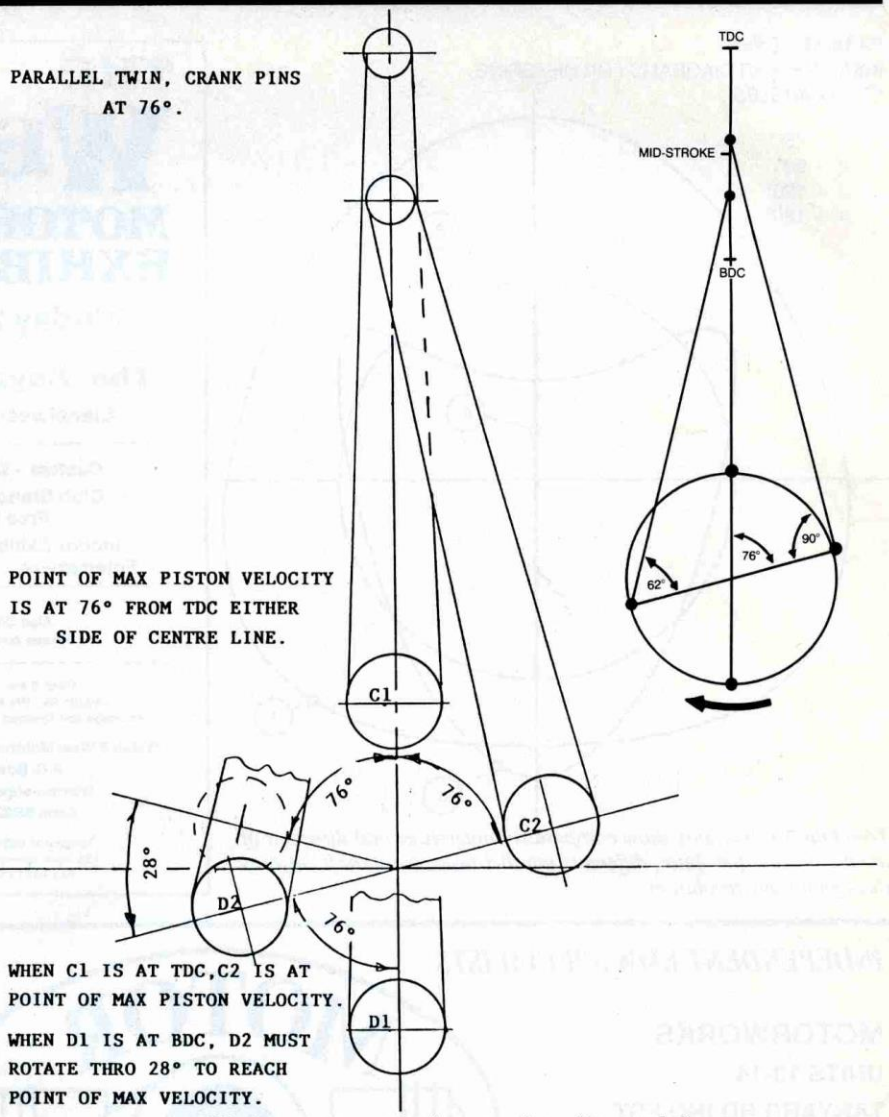
It is true that a 180-degree crankshaft produces a rocking couple, which is absent when the crankpins are in line. However, its magnitude depends on the spacing of the cylinder axes and it can be reduced to negligible proportions by clever design. For example, it is of no consequence in a BMW 'boxer' engine, where the con-rod big-end eyes are recessed into opposite sides of the middle crankweb to minimise cylinder offset; but it could be distinctly felt in a Honda CB450 parallel twin, where the two pairs of flywheels were separated by two roller-type main bearings and the camshaft drive. A shrewd design feature that effectively reduced the rocking couple was Doug Hele's counterbalanced crankshaft for the 350 cc Triumph Bandit/BSA Fury; as in Ron Valentine's 76-degree shaft, each half was treated as an independent single-cylinder crank.

In any case, although a parallel twin with the pistons moving in step is theoretically no different from a single of the same capacity so far as primary balance is concerned, it is usually worse in practice because of the wide spacing of the main bearings and the consequent crankshaft whip and less rigid crankcase.

Common disadvantage

So — given two carburettors and intelligent bottom-end design — a 180-degree crankshaft is a better bet for a four-stroke parallel twin than is a 360-degree shaft. Yet they share the disadvantage that they bring both pistons to rest simultaneously, and the flywheel has to give up kinetic energy to start them moving again. Thus there is a cyclic fluctuation in the energy content of the flywheel.

However, in a single-crankpin V-twin with a cylinder angle of, say, 50-90 degrees the situation is much better. For when either piston is at tdc or bdc the other is moving at approximately full speed. Thus the total kinetic energy content of both pistons is approximately constant and they contribute alternately to the flywheel effect, so that the actual flywheel can be lighter.



76-degree crankpin spacing (to scale)

When either piston is at tdc, the other is 76 degrees before or after tdc, hence moving at maximum speed by virtue of the 90-degree con-rod/crank angle. But when the stationary piston is at bdc, the other is moving more slowly because its con-rod/crank angle is only 62 degrees

As a long-time proponent of the V-twin (for reasons of installation as well as dynamics) Phil Irving had a better appreciation than most of the value of this feature and — when reflecting after the TT on all the points we had discussed — sought to exploit it as a possible aid in taming the parallel twin's shakes. Thus it was that his fertile mind spawned the idea of setting the two crankpins anywhere *except* in the same plane — ie, neither in line nor at 180 degrees.

Spacing the pins at 76 degrees was then a logical choice because, with connecting-rod length almost always four times the crank radius, when one piston is at tdc, the 'big end' angle (con-rod to crank) on the other side is precisely a right-angle so that the moving piston is travelling at top speed, where it contributes maximum flywheel effect but generates no unbalanced inertia force.

Firing intervals are an acceptable 284/436 degrees and — as Irving's graphs (reproduced from his original *Motor Cycling* article and another in *Motorcycle Sport* for May 1972) show more clearly than words — the balance compromise is first class. The maximum upward inertia force is halved and the downward force reduced by 20 per cent. The small, double-frequency secondary forces are almost completely balanced and the rocking couple is only half what it is with 180-degree pin spacing.

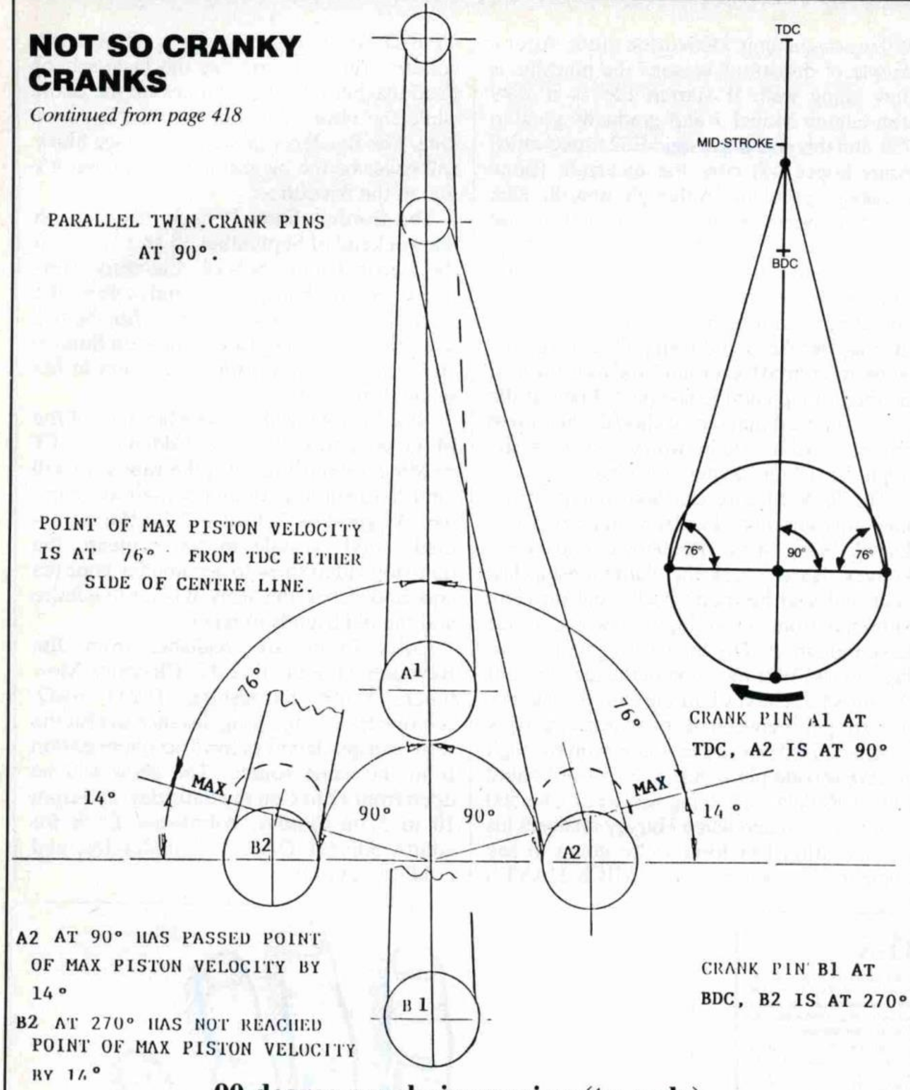
Computer

Once Steve McFarlane's brief experience proved that Irving's theory was borne out in practice, Ron Valentine decided to use his computer to explore the potential of other crankpin spacings. Produced by his mathematician colleague Tom Oliver, the four superimposed polar diagrams reproduced on the previous page, all assuming a 50 per cent balance factor, show that a 90-degree spacing (No. 2, a true circle) gives an even better balance compromise than 76 degrees (No. 1).

Continued on page 430

NOT SO CRANKY CRANKS

Continued from page 418



90-degree crankpin spacing (to scale)

At all four dead-centre positions (ie, top and bottom in both cylinders) the speed of the moving piston is the same because the con-rod/crank angle is always 76 degrees

Though No. 4, a vertical diameter of No. 2, seems to suggest that a 180-degree crank is best of all, this is misleading because polar diagrams take no account of rocking couples. (Incidentally, the diagrams are only comparative; to get the actual force values a mathematical formula is required.)

My own crank-angle diagrams for 76 and 90 degrees (elaborated by Ron Valentine) show an interesting difference in favour of the latter. With the 76-degree spacing, although when either piston is at tdc the other is moving at maximum speed (90-degree 'big-end' angle) that is not the case when the stationary piston is at bdc. Then, the 'big-end' angle for the moving piston is only about 62 degrees and the linear speed much less, so causing a high-frequency fluctuation in its flywheel effect. With the 90-degree spacing, however, whether the stationary piston is at top or bottom dead centre the 'big-end' angle for the moving piston is always about 76 degrees and the flywheel effect therefore constant, albeit the instantaneous piston speed is slightly below maximum.

Other differences are that the secondary forces would be completely, rather than substantially, balanced; the two peaks in the upward inertia curve would be slightly farther apart; and the firing intervals would be slightly more uneven (270/450 degrees) though identical with Ducati's, hence no problem. There would seem to be no appreciable increase in the rocking couple.

As with the 76-degree layout, the verdict will depend on running tests. Suffice it to say that McFarlane has three 90-degree shafts on order — one for his outfit, the others for solo use by colleagues.

For a large manufacturer some 30 years ago, trials of this sort would have been easy and relatively inexpensive to conduct had engineering been accorded its proper priority. 'Cranky cranks' might not have prevented the British industry's demise, for its fate was probably inevitable by then. But they could have vastly enhanced the riding pleasure of thousands of the manufacturers' customers and so salvaged some of the industry's lost respect.

VIC WILLOUGHBY

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Service Providers

The Service Providers listed have been used with a degree of satisfaction by OVR readers in the past. Just because they are listed does not imply an endorsment of them by OVR.

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V3 Products, Australia: (aka Neal Videan) has an extensive range of top quality Vincent Spares including multiplate clutches for twins, oil leak eliminator kits, socket head tappet adjusters, paper element oil filters and lots lots more. Ships worldwide. Email for a price list to nvidean@outlook.com

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Paul Goff, UK: A massive range of electrical spares and replacements including 6 and 12V quartz Halogen bulbs, LED lamps, solid state voltage regulators and lots lots more. Ships Worldwide. PayPal accepted. See Paul's website for more information www.norbsa02.freeuk.com

Union Jack Motorcycles, Australia: Full range of Triumph, Lucas, Amal and Venhill control cables. Ships worldwide. More info at the website www.unionjack.com.au or phone +61 3 9499 6428

François Grosset, France: Electric starter for Vincent Twin. Electronic ignitions for Vincent Single and Twin supplied complete with drive gear. Email pontricoul@gmail.com for more info.

Cometic Gaskets: Modern, reusable gasket sets for Vincent twins and singles. If you actually USE your Vincent you are mad not to use these. Contact Paul Holdsworth of the VOC Chicago section c/o pl_holdsworth@yahoo.com Located in Chicago IL USA.

Nuts n Bolts:

Classic Fastners, Australia: Their aim is to supply obsolete and hard to obtain fasteners for your restoration project be it a professional or private venture. The print catalogue, available for download, lists the current complete range. Ships Worldwide. http://www.classicfasteners.com.au/

Precision Shims Australia: All types of shims made to your requirements, ships worldwide. More info at their web site www.precisionshims.com.au

V3 Products (see entry under Spares above) also stocks a large range of Vincent specific nuts n bolts.

Restoration Services:

Ken Phelps, Australia – Qualified aircraft engineer and builder and daily rider of Norvins for over 30 years, who has the skill and experience to carry out overhauls, rebuilds, general repairs and maintenance to Vincent HRD motorcycles. Full machine shop facilities enabling complete engine and chassis rebuilds, Painting, wiring, polishing, aluminium welding and wheel building. Ken Phelps Phone: (61+) 0351760809 E-mail: ogrilp400@hotmail.com . Located in Traralgon, Victoria, Australia

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I hope you have enjoyed my clumsy efforts in producing OVR since my first edition way back in 2014.

Well this is the final regular edition. The OVR archive will continue and has been massively expanded. It now includes all issues of OVR, a wealth of original documentation from the Vincent works plus stacks of material on other marques.

To all who supported OVR and contributed content over the years – I thank you.


THANK
YOU
AND
GOODBYE