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Propeller Shaft & Universal Joints

FACTORY

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RREC, RROC & Misc Articles

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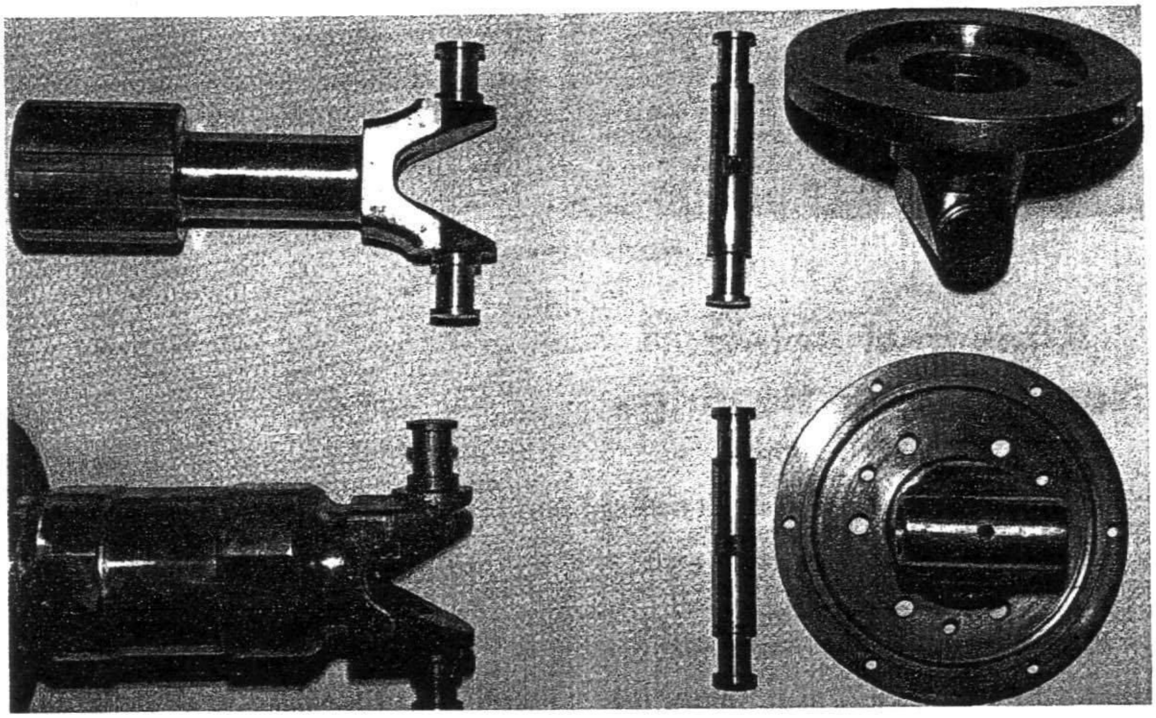
Universal Joints: Bentleys

(and maybe other models)

by David Mico



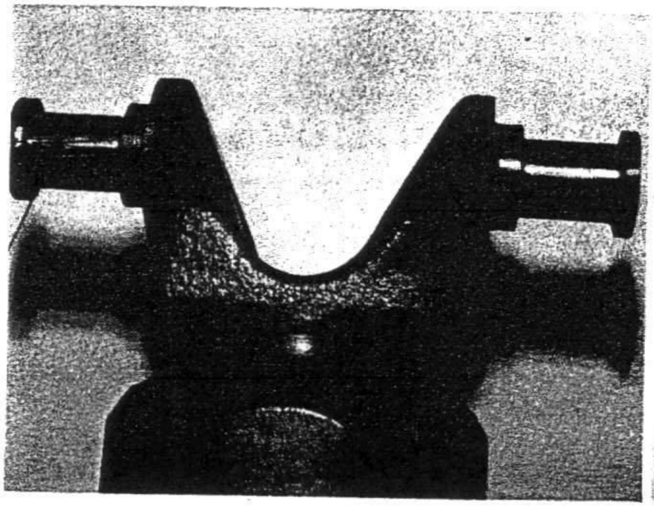
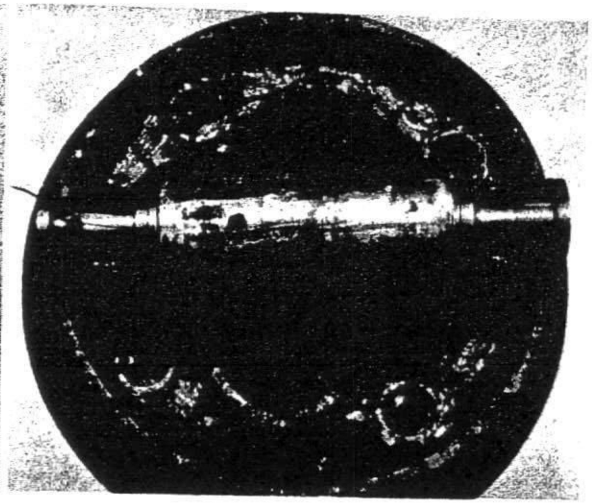
Having had a very messy time with universal joints, I send you these notes on the subject as it currently exists, as the Club will undoubtedly have cries for help from members in due course.

1. Appleyard's are sending out needle bearings of 98 thou. diameter. It is possible that they have been doing it for years!. 2. They are too big. A final empty place is left, about two thirds the width of a needle. 3. The whole set of needles then move about 20 degrees diagonally and *lock solid* in this position. 4. Thereafter, the constant hammering from the drive ruins the inner and outer bearing surfaces, which become corrugated.

To put the above right is, of course, irritating but straightforward. I order new outer bearings from Appleyard's at the not inconsiderable price of £140, plus new needle bearings. At this point, however, the problem becomes compounded, because the internal diameter of the bearings sent are 0.695. The size of the original bearings is 0.708. The needles sent are still 98 thou.

So, with the co-operation of Will Fiennes, the car will be non-standard in future in respect of the universals. As follows: Grind down the inner pins to 0.500 from 0.513. This has the twofold advantage of removing corrugations and also enabling the 98 thou. needles to be used. I am not awfully happy about reducing this dimension, with its reversing stress and the thought of fatigue. However, the needles fit well, if a little tightly, with 20 microns (or two fifths of a thou.) to spare. I often thnk that Henry pushed his luck to the limit on some dimensions, and I don't like reducing them. The pins will be low temperature heat treated.

Will Fiennes believes that the dimension of the new bearings is an RR error. But it is clear that as members get around to having a go at their universal joints they are going to be faced with all the above problems, which, in due course, will be presented to the Club.

Note diagonal corrugations.

Universal Joints: Needle Roller Bearings

by Donald Bastow

The article by David Mico on page 20 of Bulletin 138 brought back memories. Someone in the Rolls-Royce Design Ooffice spotted an article in the American Society of Automotive Engineers Journal for August 1933 and extracted from it the formulae and table reproduced with this article. I used these for all the needle roller designs I did from that time and I am sure they were used also in the Detail Drawing Office. Certainly all the Phantom III and Wraith IFS needle roller bearings were based on this information.

On a pin diameter of 0.513in there is excessive clearance. The needle rollers used were obtained from Hoffman and would have been 2.5mm in diameter (0.0984in). Nineteen of these could be fitted into the bearing. The minimum recommended clearance of 0.0001in per needle gives a pin diameter of 0.5in. Even with zero diametral clearance this implies a bearing bore of 0.6968in, so unless the needles are slightly undersize the bearing bore of 0.695in seems on the tight side.

The reduction in pin size would increase the bending stress in the pin by about 8 per cent. This is less important than the radius in the corner between the pin and the inner shoulder, which determines the stress concentration factor there. Care must be taken in grinding down not to reduce this radius. The other important thing in grinding down is not to rush it — light cuts only. Too rapid reduction by grinding can overheat the surface and cause minute cracks.

My copy of the SAE-based formulae has a not very decipherable pencil note to the effect that the indentation load is obtained by using a shaft speed of 5 to 7 rpm in the formula. Using the higher figure, which gives a lower load, this indentation load corresponds to about three times the possible propeller shaft torque in bottom gear. It is difficult to arrive at a bending stress without a knowledge of the corner radius referred to above, and the consequent stress concentration factor, but I should assume it to be still reasonable.

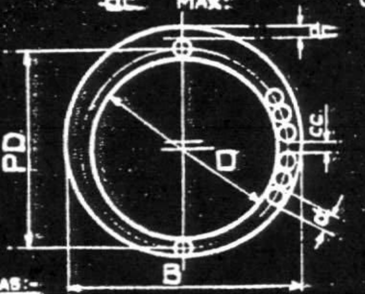
I am not sure what David Mico means by the pins having been low temperature heat treated. In my post-RR experience it is important not to lose hardness on needle roller bearings; from memory (which I hope is reliable) the desirable hardness is Rockwell 'C' 62 to 64. A case-hardening steel would normally be used. Some such steels can begin to lose surface hardness at as little as 150°C. Keeping water out, by adequate lubrication, is also important in avoiding bearing deterioration.

I hope these recollections and remarks may be useful, not only in this case but perhaps for any similar ones too.

FROM S.A.E. JOURNAL AUG: 33. — NEEDLE ROLLER BRGS. (1)

RECOMMENDED VALUES FOR DIAMETRAL CLEARANCE

SHAFT DIA: INS:	UP TO 2"	2"–4"	4"–6"
dc MIN:	0·0005	0·0010	0·0015
dc MAX:	0·0015	0·0020	0·0025



NOTATION:

- B = BORE OR INSIDE DIA: OF OUTER RACEWAY (INS:)
- CC = CIRCUM. CLEAR ON PITCH DIA: (INS:)
- D = DIA: OF INNER RACEWAY OR SHAFT (INS:)
- d = DIA: OF ROLLERS (INS:) (NOMINAL)
- dc = DIAMETRAL CLEARANCE (INS:)
- K = CONSTANT FROM TABLE II FOR DESIRED N
- L = EFFECTIVE LENGTH: ROLLER (LESS CHAMFER ETC.) INS
- N = Nº ROLLERS IN FULL BRG
- PD = ACTUAL DIA OF P/C INCLUDING CIRC: CLEARANCE
- R = RATED SAFE LOAD (LBS)
- S = SHAFT SPEED (R.P.M.)

FORMULAS:-

$N: \text{APPROX} = [(D: \text{APPROX} + d) \times \pi] / d$

$PD = [(K\, d\pi + C.C.) / \pi]$

$D = PD - d$

$B = PD + d + dc$

$R = (N \times L \times d \times 11\,250) / \sqrt[3]{S}$

$\quad = [(D + d) \times L \times 35340] / \sqrt[3]{S}$

NEEDLE ROLLER BRGS (2)

RECOMMENDED VALUES FOR CIRCUMFERENTIAL CLEARANCE ARE

CC MIN: = 0·0001 INS PER ROLLER

CC MAX: = 1/4 ROLLER DIA:

MIN: C.C. SHOULD BE AVOIDED WHEN LUBRICATION IS NOT GOOD

Nº	K	Nº	K	Nº	K	Nº	K	Nº	K	Nº	K	Nº	K	Nº	K
10	3·2360	22	7·0267	34	10·8380	46	14·6602	58	18·4706	70	22·2901	82	26·1062		
11	3·[illegible]	23	7·3441	35	11·4563	47	14·9727	59	[illegible]	71	22·6077	83	26·4236		
12	3·8637	24	7·6611	36	11·4747	48	15·2905	60	19·1058	72	22·9253	84	26·7482		
13	4·1786	25	7·9789	37	11·7912	49	15·6083	61	19·4275	73	23·2428	85	27·0657		
14	4·4940	26	8·2966	38	12·1093	50	15·9261	62	19·7448	74	23·5603	86	27·3833		
15	4·8100	27	8·6140	39	12·4273	51	16·2438	63	20·0625	75	23·8334	87	27·7008		
16	5·1258	28	8·9312	40	12·7453	52	16·5616	64	20·3801	76	24·2010	88	28·0183		
17	5·4423	29	9·2490	41	13·0644	53	16·8792	65	20·6977	77	24·5186	89	28·3358		
18	5·7587	30	9·5666	42	13·3812	54	17·1999	66	21·0152	78	24·8362	90	28·6533		
19	6·0756	31	9·8845	43	13·6991	55	17·5176	67	21·3364	79	25·1537				
20	6·3926	32	10·2025	44	14·0170	56	17·8353	68	21·6549	80	25·4712				
21	6·7095	33	10·5197	45	14·3349	57	18·1530	69	21·9726	81	25·7887				

Universal joints — needle roller bearings

by Will Fiennes

Recently David Mico (B138) and Donald Bastow (B139) have written on the subject of universal joints. In the course of rebuilding David Mico's universal joints I followed through the design calculations for a needle roller bearing in a universal joint, consisting of 19 rollers with a 2.5mm diameter. These gave a pin diameter of 0.4997in-0.0003in. Subsequently, I located an original Rolls-Royce drawing of the pin, which shows a diameter of 0.4997in-0.0001in. Clearly the design criteria available at the time were in line with current thinking, although it would seem that the Detail Drawing Office, seeking as near perfection as possible, specified a much tighter tolerance. Current thinking on universal joints in which the motion is reciprocating rather than rotating calls for a reduced minimum total circumferential clearance of 0.001in for the rollers, one third of that recommended for a standard needle roller race, and half that which Donald Bastow would have used.

The oversize pin diameter, as in David Mico's universal joint, would appear to be standard for the M-series cars, as is borne out by Norman Burt's letter in B139; indeed, S. Brunt of Silverdale, Staffordshire, has a stock of pins for this series. When I first encountered David's problem I thought that perhaps someone had made an error in machining a batch of the outer trunnions oversize in the bore and, to compensate, oversize pins had been made. Very un-Rolls-Royce! It is clear, however, that these dimensions were achieved by design. Unfortunately, in no way can the design criteria be modified to produce a suitable crowded needle roller bearing with a central race diameter of 0.513in using 2.5mm rollers. The extra circumferential clearance with this race diameter will very likely lead to the needles skewing and jamming at an angle to the axis of the pin as David Mico found. I have no idea why the design change was made.

Finally, new pins for David Mico's universal joints are to be made to the smaller diameter which can be used with trunnion GB911. The pins will be made in EN36 — a case-hardening steel with a final core strength of 55 tons per square inch. There will be a number of pins surplus to requirements, should anyone be in need.

STEERING CLEAR OF TROUBLE

MEMBERS sometimes have a problem with steering boxes of pre-war Rolls-Royces, in that while the steering displays no defects when the wheel is turned, it fails to return to centre after completing a long corner of constant radius. When, subsequently, the front axle is jacked up, the steering appears to be quite smooth and fully reversible.

As has been explained in the Bulletin, the cause is the squeezing out of lubricant from the contact faces of the worm and nut caused by static loading. Metal to metal contact results and causes irreversibility of the mechanism. A most unpleasant and potentially dangerous characteristic.

The solution I have seen recommended is the use of EP 140 lubricant in the steering box. This may cure the problem, but there is an even better lubricant, Shell Macoma R1000, though it is only obtainable from suppliers of Shell industrial lubricants and probably in quantities not less than one gallon. EP 140 is formulated to prevent metal to metal contact and local welding of rapidly sliding, highly loaded gear teeth. Such conditions are encountered in hypoid bevel final drives.

Macoma is formulated to prevent metal to metal contact in near static conditions of high loading, e.g. the bearings of steel rolling mills. These latter conditions are of a type more comparable to those in back axles. I experienced this irreversibility in my 20/25, GXO 12, and Macoma R1000 has completely cured it.

This advice was given to me by Cecil Mitchell, a senior engineer in Shell Technical Department, who is also chief scrutineer to the RAC and the owner of a type 43 Bugatti.

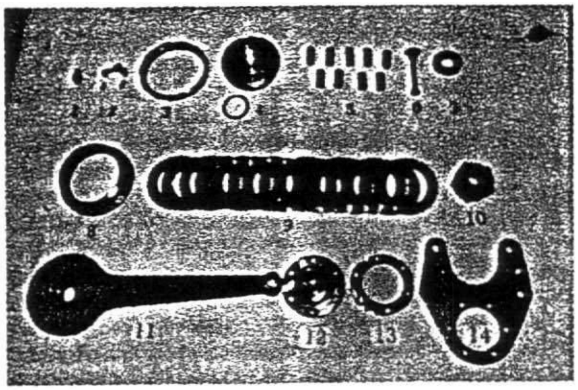
H. ORR-EWING

and screw home the two 6BA set screws. A small adjustment to the position of the locking ring may be necessary for this to be achieved.

Fill the unit with grease and check action. If the force exerted by a spring balance on the lever ball end is between 20lb and 40lb to move it relative to the bracket, then the friction damping will be about right. If the force is lower than 20lb to start the relative movement, then either change to stiffer springs if you can find a suitable set, or else use packing pieces beneath the springs. If the force is above the top limit, which is very unlikely, then slacken the locking screw back several castellations until it is within limits.

I was very fortunate when overhauling mine as I found no apparent wear in the components. One had been dismantled by a previous repairer and had not been tightened up properly – an easy repair. The other had a broken central splined hub or journal and to date I have failed to locate a replacement. If any member can put me in touch with one I shall be eternally grateful.

I hope to conclude with the overhaul of the front dampers in the next journal, together with a method which I successfully adopted to true up ball-pins in a lathe using a whetstone and patience.



The Technical Secretary comments:

I think Mervyn Woodward was lucky in that most problems arise from worn splines. A new shaft can be made up or the splines can be metal sprayed and recut. If they are made slightly oversize the discs can be dressed up with a stone or small file to fit. Efficient dampers keep the wheels in contact with the road at all times, a distinct advantage if you only have two wheel brakes!

MICHAEL SAPSFORD

Prewar Bentleys: the propshaft damper

By David Mico

At first sight it looks like one of those jobs where you undo dozens of nuts, mostly the wrong ones, until eventually something drops off.

Open the service instructions and consult Propshaft, Section H. This does not take long; it is completely empty. However, this is only the service instructions' little joke. Proceed to, of all unlikely places, CLUTCH, Section F. Here you will find 'Removal of Gearbox', which can be cut out and more suitably placed under Gearbox, Section G, but, before doing so, take out 'How to remove propshaft and damper', which can now be put in Propshaft, Section H.

We are in business!

No one ever mentions the propshaft damper. It appears that in its youth it did something very naughty, which is best not discussed. Suffice to say that it was fitted in 1935 (chassis 2DG), hastily withdrawn, then quietly popped back in again in 1937, chassis 53JY.

'Propshaft damper' describes what it does, not what it is. It is a flywheel, fitted with a safety device. The centre part is a truly beautiful machined piece of bronze. Put on the mantelpiece it looks like a small Henry Moore. Its circular bronze fin is clasped between two hardened steel rings. The intensity of the embrace depends upon twelve springs around the circumference. It weighs over 8lb, and nearly all of this weight is on the extreme periphery, where it orbits in a circle of eight inches diameter.

On the works drawing are indecipherable figures relating to slip. However, the Club has researched these figures, which are:

Slip at 5lb (+ or – 1lb) at 17.5in radius. With dry surfaces.

There are no methods of adjustment, unless you start leaving springs out, which is not the idea. However, you will judge from the pristine condition of the damper that it strongly objects to ever slipping at all. The stored energy within this flywheel exerts its authority over the propshaft, prancing up and down behind it. In the event of a sudden crash stop from perhaps a propshaft speed of 4,000 rpm to 0 rpm, this stored energy would impose intolerable strains on adjacent parts. The energy is then released by slipping.