CRANKSHAFT DAMPERS, SPRING DRIVES AND CAMSHAFTS – PART 2

THE HISTORY FROM 1953 – 1958

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PURSUING THE LATER HISTORY OF THE DAMPER AND SPRING DRIVE

Probably more has been written on the subject of crankshaft dampers than any other subject on Rolls-Royce engines and hours of time must have been expended on the setting up of dampers. The whole subject is one to set alight any editor's e-mail inbox or postbag.

With some notable exceptions, most of the previously written technical considerations are directed towards the choice of different damper friction materials or the arbitrary poundage setting of the damper. Most acknowledge the ability of the dampers to break, seize or stick, but suggest no long term cures, and even less has been said to indicate the circumstances during which a crankshaft will suffer damage, other than sheer engine over speeding. Even less has been said on the subject of the last of the 4.5 litre engine dampers, that had Ferodo friction washers and slotted dampers and which I covered briefly in previous articles.

One is forced to ask the questions, what happens if the damper is sludged or otherwise not working correctly and what can go wrong? What can be done to improve the situation? Possibly, more importantly, what did the company do to address the situation and what did they know? Whilst conducting research on the Ferodo dampers it became apparent that very little, if anything was known and inquiries to various specialists drew a blank. Fortunately the archive contains the answers to most of these questions and provides an interesting history, and the reasons why the original dampers with cotton duck washers were also modified.

Although this article is not intended to cover crankshafts in detail it is impossible to discuss the dampers without outlining some history of the post war six cylinder crankshafts. The results of a number of company reports, documents and memo's on crankshafts and dampers have been paraphrased, hopefully without losing the true translation.

Of all the engineering staff who were involved in crankshaft dampers at company level, one stands out in particular. He was Stanley Bull. In my own opinion he was so important to this later history of the damper. His approach seemed down to earth and practical, to say the least. Further consideration of the damper would, in fact, be an injustice without giving some background to this very practical engineer.

Stanley Bull joined Rolls-Royce in 1919 and knew Royce very well. He worked for Harvey Bailey Snr during the last war. Stanley, known internally in R-R parlance under the initials SB, became technical manager for cars, initially in Derby and then in 1945 at Hythe Road service depot in London. In 1959 he took up the post of World Service manager at Crewe and then became Service Director. After completing 50 years' service with the company he retired in 1970. Sadly he passed away on 5th August 1976 aged 72 years. An engineer, who appears to have kept a level, practical, engineering head. We shall see shortly how the initials SB featured in this story.

I make no apology for reminding enthusiasts who wish to learn of the early history of the R-R crankshaft damper that they should read Tom Clarke's book, which is published by the Rolls-Royce Heritage Trust, entitled 'Royce and the Vibration Damper'. This book, besides covering the history of the damper superbly, also contains a very extensive and interesting appendix by Ken Lea, formerly Director and Chief Engineer (Power Train) at Rolls- Royce Motors.

I would also remind readers to compare comments and particularly a later section of this article, called Actual Crankshaft Breakages, with a comment by Ken Lea on page 116 of this booklet. In no uncertain terms he points out that the damper, and no doubt the engine is doomed should the 3rd order frequency approach the natural frequency of the crankshaft. It is indeed educational (if you are interested in cars) to see just what parts break, and I hope that this article will leave no doubt, whilst covering a very interesting part of history.

Part 1 of this series of articles set out to explain some of the more practical aspects of the post war crankshaft dampers, whilst this article part 2 sets out to tell some of the later history. Inevitably some points are repeated to complete the story.

REPORTED CRANKSHAFT DAMAGE

I have known of two cracked 4.5 litre crankshafts, one visual and one found by crack detection, other specialists to whom I have spoken have admitted to never witnessing a damaged post war shaft. This is not to suggest that shafts cannot be damaged, but probably the occurrences are not as common as generally believed and when they do occur are the result of sheer vandalism on the part of the driver. However there is no doubt that many owners have experienced crankshaft torsional vibration, which has spoilt the experience of driving one of the company's products to say the least. It is not surprising that some specialists have never experienced a broken post war crankshaft. The company themselves did not face such an experience until November 1953, and it came with a vengeance!

At this particular time the company was in the middle of quite intensive experiments to improve the crankshaft liveliness and balance of the Siam engine, later to enter production as the 4.8 litre Bentley S1 / R-R Silver Cloud 1 engine. During the course of the following tests on dampers and crankshafts and their endeavours to improve on existing crankshaft vibrations, the company had a few surprises. They found that the swash or wobble on the flywheels reached 0.020 inch, that the outer rim of a damper also would move axially 0.020 inch and that the crankshafts acted in a concertina fashion by lengthening and shortening from front to rear. To add to the troubles the concertina action was not consistent with the front and rear ends acting together, but each end could have a mind of its own.

The first engine crankshafts were not fully balanced until late in 1953. The company, at this stage, were pursuing both out of balance and torsional vibration problems in parallel and this story crosses both paths. It is normal for the crankshaft vibration to be felt around 2500 rpm but company engineers had identified particular unbalances during other RPM ranges and even engines with trembles at idle speed.

A number of noteworthy points arose from the tests and in July 1953 the Engineering Department Report EER 627 was raised. This was conducted on an inter Siam crankshaft part number UE 209. It highlighted, for the first time, three specific points when it says amongst the general conclusions:-

"The tests show the present method of balancing rotating assemblies to be unsatisfactory"

"The most satisfactory solution would be to balance the rotating masses (and possibly the reciprocating masses) in the crankcase. It is suggested that the possibility of obtaining a dynamic balancing machine as outlined in this report, should be investigated"

"In the meantime it is recommended that all clutches be balanced when assembled on the flywheel, the necessary corrections being made by drilling the pressure plate through the springs"

Although the company did possess a dynamic crankshaft balancer, this was fully engaged on balancing military crankshafts. The EER 627 report highlighted some of the problems experienced and goes on to outline crankshaft balancing methods thus:-

“The present method with hand change gear boxes is to balance the crankshaft and flywheel together on knife edges, any bias being removed by drilling on the flywheel periphery. The clutch is then added, and tried in three possible positions, the best being selected. The clutch position is then marked, no further corrections being made. The clutch driven plate is also added without further balancing”

The last comment is very important because it was found that the largest out of balance force on the crankshaft assembly was the clutch driven plate itself. Indeed on this particular test it proved hard to obtain one in the whole works, which was near enough in balance to be useful. The report recommended, amongst other points, that all clutch driven plates be henceforth balanced separately on a mandrel.

Besides highlighting that crankshafts were not dynamically balanced, this report also discusses the use of a special rig, which was later modified to accept a disused alloy crankcase and later mounted on an Avery balancer. The alloy crankcase was one of a small number, one of which was especially used for testing pistons running directly in parent alloy bores. The special rig consisted of an inverted crankcase, which rested on a rubber sheet, in which the crankshaft assembly was rotated whilst being supported in no’s 1, 4 and 7 main bearings. The balance was checked by the movement of the crankcase and was sufficient to detect differences due to an out of balance clutch drive plate. Up to a point the simplest ideas work!

Only in December 1953 was the finalised dynamic balancing rig instructed for use, initially with all engines fitted with manual gearboxes although the balancing of automatic gearbox dressed engines was also becoming a problem. The glitch here was ensuring that the torus cover was filled with oil and the air was completely expelled.

Balance problems had been a thorn in the side of engineering staff for some time. There are regular references to out of balance crankshafts due to the bowing of the shafts after nitride hardening. Very strangely, the company allows a maximum in service bow of the crankshaft of 0.010 inch, far more than they experienced on production, and yet their small production tolerance was promoting balance troubles! There is no doubt that, nitriding apart, these shafts have a tendency to bow in service, and even a third of the allowance allowed by the company will make it almost impossible to centre and regrind the shaft, unless it is straightened beforehand. This is not an impossible task, but very time consuming, and it is imperative that these shafts should be checked for truth and straightened before regrinding.

ACTUAL CRANKSHAFT BREAKAGES

Here Stanley Bull (SB) enters our picture and his report of 18th November 1953 was not one the engineering staff at Crewe would have been happy to receive, in fact, at the very least it would have spoilt their day! The first sentence read, “We have to report two broken crankshafts, which have occurred on large bore Bentleys”. (Large bore in this case means the 4.5 Litre engine.)

Both these crankshafts had failed in an identical manner, namely across no 6 crankpin adjacent to the face of the forward web on that crankpin. In simple terms the failures were at the node point, or position on the crankshaft at which the torsional forces are calculated to act in the greatest manner. One shaft had broken completely, whilst the other was in an advance state of cracking.

The chassis numbers were B226 MD and B168 SR, both fitted with 4.5 engines and having completed 24,000 and 19,000 miles respectively. The crankshafts on these 4.5 litre engines were RE 11551 differing significantly from the earlier 4.25 litre crankshaft, in having a thicker rear end web. Also significant was that both these cars had manual gearboxes where down gearshifts were possible at high speed with no gas pressure above the pistons. Within months B 226 MD, in the same ownership, was to break yet another engine crankshaft, now making three shafts broken in service! This latter chassis was undoubtedly being driven with little sympathy for its engine survival!

This internal memo certainly caused the consternation one would expect. By early December 1953 the stresses in the crankshaft node area had been reworked, 3rd order crankshaft frequencies rechecked and tests arranged to find out the engine speeds and conditions that would break a crankshaft. Of particular interest to this current article is that it was feared that sludged dampers might even bring the engine crankshaft critical speed down into a normal drivers operating range, and this was confirmed.

Assuming a clean working damper, archive calculation showed that the 3rd order natural crankshaft frequency was 15410 cycles per minute, equivalent to 5136 engine rpm. On the other hand a solid sludged damper reduced the 3rd order frequency to 11460 cycles or 3820 rpm. These calculations were later proven on test rigs with both new and sludged dampers. It was further confirmed that these calculations placed the node or reaction point on the crankshaft between the centre of no 6 cylinder and the flywheel.

Over the years warnings regarding sudden throttling back from medium to high speeds, which have astounding consequences in immediately reversing the stresses, have been few and far between. Suddenly releasing the throttle and hence gas loads allows a wound up crankshaft to immediately and uncontrollably unwind. This condition is exaggerated by the overrun driving inertia of the car, and not helped if the engine is suffering maladies such as a misfire, when the gas loads will be erratic. It is possible that during the following related company tests, that throttling back between test cycles or misplaced plug leads may have been more significant than was believed at the time.

FIRST CRANKSHAFT BREAKAGE TESTS

On 26th January 1954 the first test to deliberately break an engine crankshaft was completed, this report was number EER 783. The rig engine chosen was BM134 built up as a 4.8 litre six inlet port engine and this dress was probably chosen as it was close to the type which was to be in production within the year. The crankcase was a standard 4.5 litre unit bored out to 3.75 inch bore and fitted with a six port head and twin S.U carburettors. The exhaust was passed through a water cooled test bed manifold to a 3.0 inch single pipe and thence to two silencers. It was fitted to a synchromesh gearbox run in neutral.

At full throttle the maximum rpm was 4800 and after a minute a plug lead detached. After shutting down and replacing the lead the engine was run again at 4800 rpm for approximately one minute. After this period the exhaust pipe and silencers were removed and the engine ran at 5000 rpm maximum speed for another minute.

FIG 1 DAMPER TRAPPED COTTON DUCK WASHER

The engine was then stripped and the spring drive extracted with some difficulty and, although the slipping load of the damper was 14 – 16 lbs., one friction cotton duck washer was trapped between the friction drum and damper wheel. This trapped washer was fairly significant, as we shall see later during the study of the actual dampers. A typically trapped friction washer is shown in Fig 1 but it should be pointed out that this photograph is from test number EER 1143 dated 19th October 1955. The front taper on the crankshaft had picked up; the flywheel had fidgeted on the crankshaft rear abutment and no's 5, 6 and 7 main bearing caps had fretted. Trailing all this damage the crankshaft had cracked right around the forward end of no 6 crankpin.

All this damage occurred in the space of three minutes and, as we shall see later, this and much more can occur in only one minute or less. This is the point to remind readers of my earlier warning and that of Ken Lea's, that even when in good condition, no damper can save an engine when it's the 3rd order frequency approaches natural frequency. This situation is clearly demonstrated in this series of tests.

Further high speed testing on this same engine rig was carried out and contained in the EER 926 report of 16th August 1954, when a second crankshaft failure occurred. This second test was time restricted to see if the same amount of damage occurred in a short run period, or if the frequencies needed time to build up. Four short periods of 15 seconds each, were run at 4900 rpm.

After the first two periods, totalling 30 seconds, a damage check was instigated. Inspection revealed that No 5 main bearing cap was broken and fretting had occurred between the three rear main bearing caps and the crankcase. Fretting had also occurred between the flywheel and crankshaft flange.

After an engine rebuild, the test was resumed with a further two 15 second periods of running. A strip examination then showed that fretting had again occurred at the previous locations and this time No 7 main bearing cap had broken. In addition, on this occasion, the crankshaft had suffered cracks on the radii of No 4, 5 and 6 crankpins.

Following these previous tests the company had experienced a crankshaft failure at lower speeds on a test bed engine being run on piston duration testing. Problems occurring with vibrations at approximately 2500 rpm after the initial engine running in procedure had been carried out, with runs above this speed but not over 4500 rpm. In this instance it is very probable that sudden throttling back during the test had some significance.

Quickly following these tests was an in service failure of the crankshaft on Bentley Continental BC12C and much later on 18th December 1956 Silver Cloud LSXA 105, with the 4.8 litre engine which failed the crankshaft across No 6 crankpin, whilst in a customer's hands.

With some considerable evidence now at hand it was decided to test the durability of the 4.25 litre crankshaft. Previous failures, at this time, had been with the 4.5 litre crankshaft. This 4.25 litre crankshaft test was contained in the EER 1027 report of 4th April 1955. The results of this test were discussed in the Part 1 article in these sequences.

In addition to the tests, which have been mentioned, the company were pursuing, in parallel, general crankshaft induced vibrations problems and a number of tests and experiments were conducted. These are too numerous to mention in this article but should any enthusiast wish to pursue that history, the following list may be helpful.

EER 624 on 2nd July 1953, Road tests on forged to size crankshaft UE 290.

EER 692 on 6th October 1953, Testing stiffened short throw crankshaft RE 20319.

EER 731 on 24th November 1953, Loosening of flywheel screws on slave engine no 8, chassis LWEME.

EER 897 on 14th June 1954, Evaluation of various crankshaft dampers on the boom period of Siam cars

EER 912 on 8th July 1954, Evaluation of engine roughness of crankshafts with different balance weights

ERR 1051 on 26th May 1955, Crankshaft balance with thin flywheel.

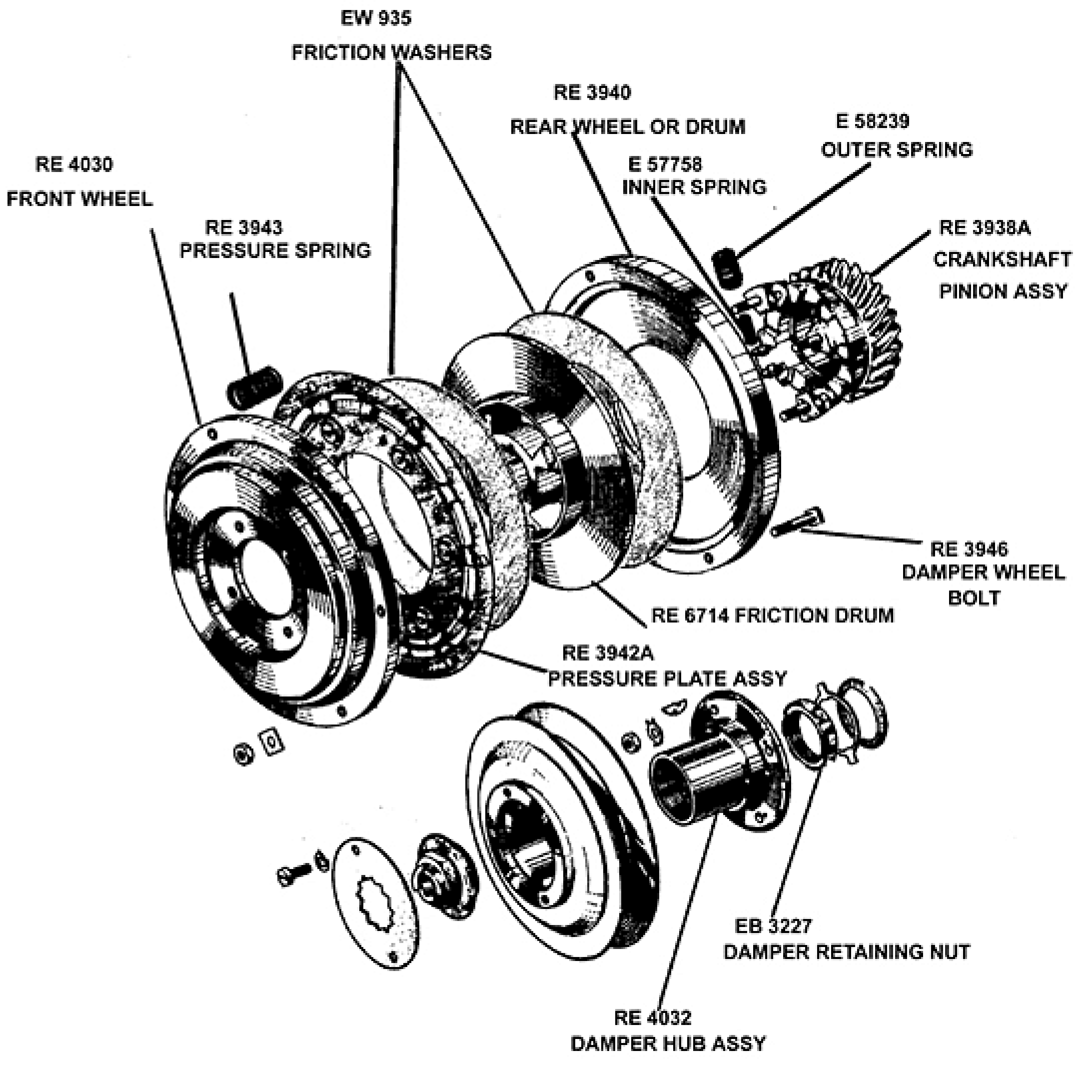
This crankshaft history is by no means exhaustive but sets the scene, at least a little. After completing the tests mentioned it was obvious the company knew that no crankshaft damper could save a crankshaft when it went "critical". When the crankshaft did reach this point the damage was severe to say the least. Their tests on sludge dampers had indicated that this type of damage could be expected at least 1000 rpm below the point when damage might otherwise be likely.

An extract from a letter dated 17th May 1957, sent by S. H. Grylls, chief engineer R-R Motor Car Division to Mr Ker Wilson, himself a brilliant engineer, of De Haviland Engine Co. Ltd, is fitting and informative to end this section.

"During the last few years we have found that engine mountings, body mountings and the behaviour of the flywheel are the three things which most determine the smoothness of an engine. The system of balance weights is chosen mostly for reasons of economics and today we use the four weight system pioneered by Jaguar and greatly favoured by the forgers.

We have made two other big contributions to smoothness recently. We now dynamically balance the entire crankshaft and flywheel assembly in a crankcase complete with bearings, in order to overcome the unbalance, which would result from straightening a crankshaft balanced in its bowed condition. Secondly, we determine by experiment the thickness of the flywheel back plate so that the inertia is least disturbed by the behaviour of the crankshaft flange. It looks as if we are on the eve of some further discoveries along this path"

FIG 2 DAMPER COMPONENTS FROM PARTS BOOK



HISTORY DURING 1953 TO 1958

This specific period I find is particularly interesting, as it represents the ultimate in the development of the six cylinder damper, answers a few questions possibly hitherto unknown to most enthusiasts, but also poses the question… "What if, damper development had continued?" For reference a typical parts manual view of the damper is repeated in Fig 2 from an earlier article sequence.

Coincidental with the first reports of crankshaft breakages in service, company engineers had been working on experiments using different materials for the crankshaft damper washers. The original cotton duck washers, part number EW 935, were supplied by British Belting and Asbestos Limited of Cleckheaton in Yorkshire, better known under the trade name of Mintex. The cotton duck washers had certain shortcomings, most notably the ability to stick to the faces of the damper drums. Sludging was also a known problem and it is most probable that Stanley Bull from the service side of the company was one of the internal forces that propelled the organisation to try a different approach to the problems of sludging and the shortcomings of the cotton duck washers.

SLOTTED DAMPERS AND FERODO FRICTION WASHERS

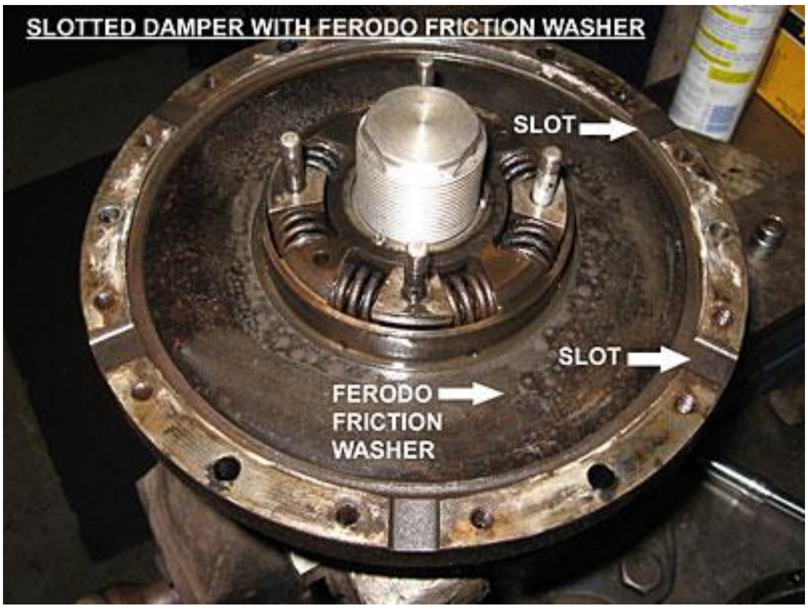
In April 1954, a Ferodo material known as VM41 was under test along with other Ferodo products, and in fact Ferodo themselves had been supplied with equipment by R-R to check the material characteristics in dampers. A month later Ferodo themselves were in the course of rigging up an oscillating device to obtain surface bedding in. It was originally intended that the VM41 material, which was made into rigid disk washers, should 'always' be used with slotted dampers, a point worth remembering. By early May 1954 a similar test rig was in the course of production for British Belting, as a company memo of 12th May confirms. By the 24th May this rig, complete with a damper and a damper assembly drawing RE 6715, had been dispatched from Crewe.

FIG 3 DAMPER SLOTTED TYPE FRONT DRUM

FIG 4 DAMPER FERODO FRICTION WASHER AND DRUM SLOTS

The term slotted was used to describe a damper shell or casing that had slots formed around its periphery so that sludge and oil could be expelled and centrifuged out of the damper. The slotted damper and the Ferodo VM41 material was later to be used in 1955, on the very last of the 4.5 litre engines and was also intended for the later 4.8 Litre engines. Reference to Fig 1 will show what looks like sludge drain holes drilled in the periphery of the rear damper drum. It is possible that this method was tried to test the theory. Fig 3 shows slots machined in a front drum, this one showing fretting evidence after a crankshaft breakage test.

Production dampers had the slots cast in during the manufacturing of the drums, which no doubt was a less costly exercise than machining. The slotted damper drums and Ferodo washers are listed in the modification bulletin index as the last modification made to engines prior to the launch of the Bentley S1 and Silver Cloud engines. The respective slotted damper drums becoming RE 21657 for the 4.5 Litre engine and RE 21656 for the later 4.8 Litre. Each rear drum contained six equally spaced slots 0.500 inch wide and 0.040 inch deep. The Ferodo VM41 friction disk was UE 2549 of some 6.690 inch O/D and



4.040 inch I/D. UE 2559 was a slotted rear drum to suit the early Wraith type damper. All measurements of course were subject to production tolerances.

A slotted rear drum taken from a late Z series R type part no RE 21657 is shown in Fig 4 after considerable service and the damper parts were all still intact and working.

FERODO INTERNATIONAL TECHNICAL NEWS F14

Friction materials

VM41

General description
Ferodo VM41 is a rigid moulded, non-metallic friction material with a random fibre asbestos base; it is fawn in colour. VM41 is available in disc and disc segment form and as flat sheets from which strips or special shapes can be cut; it is suitable for use under dry operating conditions only (i.e. it is not suitable for use in oil).
Although used in some cases as an automotive clutch facing, VM41 is principally used in the industrial field for clutch and brake linings, friction discs, friction drives, etc., in a very wide range of shapes and sizes.
This material is suitable for use at light or medium duty levels and at moderate loadings; it has a medium friction level with good resistance to wear and smooth take-up characteristics.
The size range information given overleaf is necessarily only a very brief summary of the principal limitations on the availability of this material at the time of publication. Particular requirements should always be referred to Ferodo Ltd for confirmation that the desired dimensions are currently acceptable.

Application
Industrial clutches
Industrial plate type brakes
Friction washers
Friction drives
Clamping devices

Technical data

μ for design purposes (see curves overleaf)	0.34	

Physical properties

Specific gravity	1.55	
Ultimate tensile strength	13.1 MN/m²	(1900 lb/in²)
Ultimate compressive strength	110.3 MN/m²	(16000 lb/in²)
Ultimate shear strength	5.5 MN/m²	(800 lb/in²)
Rivet holding capacity	79.4 MN/m²	(11500 lb/in²)
Thermal conductivity	0.001 J/m sec °C	
Brinell hardness	6.5	

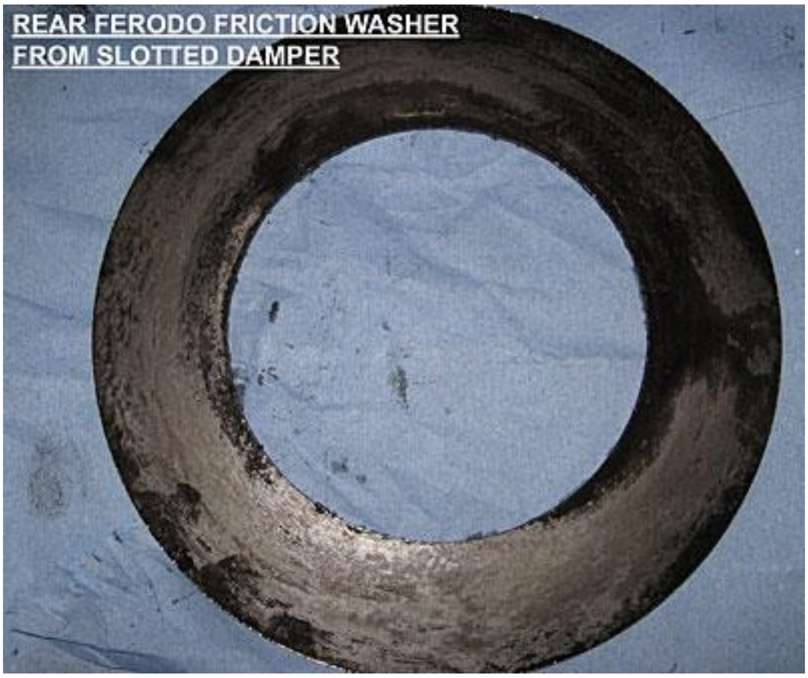
Recommended operating range

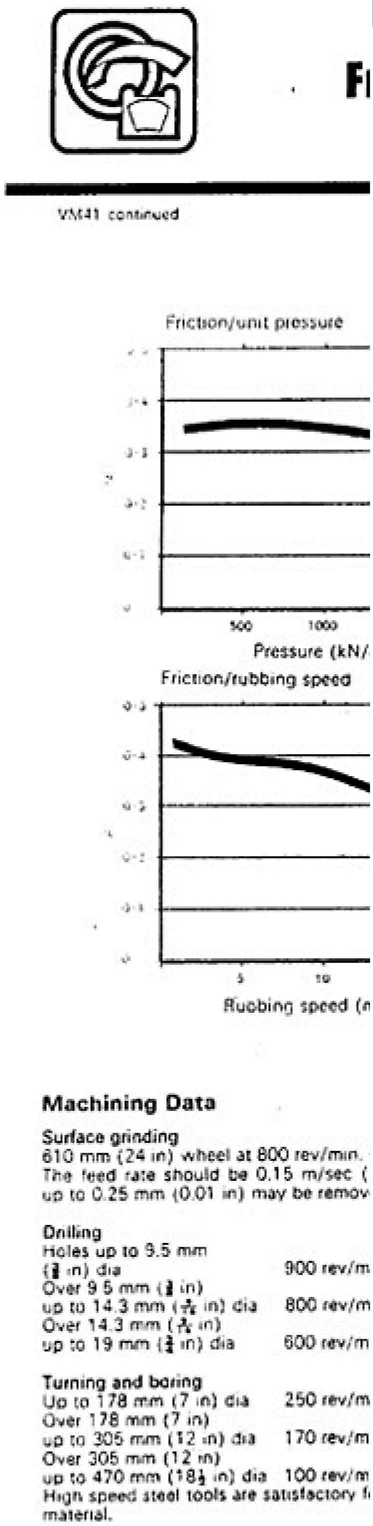
Unit pressure	70-690 kN/m²	(10-100 lb/in²)
Maximum rubbing speed	18 m/sec	(60 ft/sec)
Maximum temperature	290°C	
Maximum continuous temperature	100°C	

Note. The continuous temperature quoted is for constant slip conditions. For intermittent applications, bulk temperatures of 160°C are acceptable for long periods.

Bonding
VM41 may be bonded with any of the established bonding adhesives; for the best results, however, thermosetting adhesives should be used. The material is supplied ground on both faces and may therefore be bonded on either side.

Recommended mating surface
Good quality fine grained pearlitic cast iron. Cast steel is not suitable as a mating surface but forged or cold rolled steel with a Brinell hardness of 150 or more may be used.




VM41 continued

Machining Data

Surface grinding
610 mm (24 in) wheel at 800 rev/min.
The feed rate should be 0.15 m/sec (
up to 0.25 mm (0.01 in) may be remov

Drilling

Holes up to 9.5 mm (⅜ in) dia	900 rev/m
Over 9.5 mm (⅜ in) up to 14.3 mm (9/16 in) dia	800 rev/m
Over 14.3 mm (9/16 in) up to 19 mm (¾ in) dia	600 rev/m

Turning and boring

Up to 178 mm (7 in) dia	250 rev/m
Over 178 mm (7 in) up to 305 mm (12 in) dia	170 rev/m
Over 305 mm (12 in) up to 470 mm (18½ in) dia	100 rev/m

High speed steel tools are satisfactory f
material.

FIG 5 DAMPER FERODO VM41 MATERIAL SPEC

FIG 6 DAMPER USED FERODO WASHER OF VM 41 MATERIAL

Of all the data contained in this article the most difficult to track down was the actual specification of the Ferodo VM41 material; Fig 5 shows the original Ferodo specification sheet for this material. I am indebted for this information to Boyd Holmes of Industrial Brakes Ltd, Bilston, Wolverhampton, he went to considerable lengths to search for, and extract the information from his archives. Readers will note that the material is listed as not suitable for use in oil, an environment it was certainly to witness in the Ferodo and Rolls-Royce tests. Time has proven that the material is durable, Fig 6 shows a friction washer that has been removed from an engine apparently after 50 years' service and it is still within the original UE 2549 specification.

It would appear that, in spite of the slotted damper drums being intended for use with the rigid Ferodo friction washers, an attempt at introducing the slotted damper and cotton duck washers was made as early as May 1954. An extract from a memo dated 10th May 1954 reads,

"We are having an epidemic of trapped front cotton duck washers in Test Dept. car engines and of objectionable vibration at the half crank period. The only recent change to the damper of which we know is the introduction of oil release slots to prevent the accumulation of sludge in the damper. It is possible that a damper not full of oil vibrates more endwise and until we have evidence to the contrary we wish to omit the slots on production"

A further memo of 13th May 1954 seems to back up the production intention, in reply to the previous memo, the brackets are mine, and it reads,

"...on checking the position we find that of the slotted type, there are 38 only in F.P.S (Final Production Stores) plus a similar quantity on the assembly line. The material on the shop floor, a quantity of 300, is in the 'early stages' of manufacture and we have therefore, in order to omit the slots as quickly as possible, changed this batch of material over to make RE 4030 the original front damper wheel" (Note this mentions front damper wheels or drums not rear damper wheels, which were eventually slotted). It is worth noting that the washer trapping problem, at this time, was occurring with the front cotton duck washers only.

On the same date, 13th May 1954, EER 875 was released titled, 'Probable cause of outer duck washer becoming trapped on production engines'. The introduction of this report reads,

"There would appear to be only two possible methods whereby the washer can become trapped.

The presser plate vibrating relative to the inertia mass. With the presser plate springs 'choc-a-bloc' there is a clearance of 0.020 inch. This dimension is aggravated by the chamfer 0.020 inch by 45°, which should be removed from the C.W.P drawing.

On assembly we have tested such a damper which gave the standard slipping load so that it would be possible for this to occur without being detected"

The main body of this report also contains the following remarks:

"We know that it is possible for the outer mass of the damper to vibrate axially with amplitudes of up to 0.020 inch. The movement is limited by the clearance between the friction drum boss and the front damper wheel"....... (This comment is very important because if the damper pressure plate lifts off the friction surfaces it destroys at a stroke most of the past presumptions concerning the practical operation of these dampers.....NWG)

"...it would appear therefore that the only possible way in which the washer could move outwards between the presser plate and the rear friction drum, would be due to the presser plate, by some means, moving relative to the inertia mass. With the springs fully compressed the clearance through which the washer could move is 0.020 inch, this value being assisted by 0.020 inch by 45° chamfer. It should be pointed out that this chamfer only exists on the Works Process Drawing"

If the presser plate was to vibrate in such a manner, it would mean that the damping load was being removed completely. No coil clash markings were visible on the springs so that this theory would appear unlikely. The only alternative is that the washer was trapped on assembly".... As time went by it was realised that the damper load was indeed being removed completely.

On 14th May 1954 the decision was made to delay the introduction of the slotted damper and continue production with the normal damper fitted with cotton duck washers. Meanwhile, as testing still continued with the rigid friction washers, the company was experiencing troubles with the existing EW 935 cotton washers. They had a tendency to fray at the joint and bind on their inside diameter when assembled. The various damper tests had highlighted a number of potential trouble spots and alterations were made to the drawings around this time to alter the tolerances of the friction washers and the RE 3940 rear drum. In addition new jointing cement was used on the friction washer scarf joint.

Stanley Bull was obviously aware of the events to date and, convinced of the advantages of slotting or drilling the dampers, borne out by his experience through many years of drilling dampers at Hythe Road. In fact Stanley Bull had instructed the raising of drawing number SB401 at Hythe Road on 31/12/1953 and 1/1/1954.... they worked New Year's day! This drawing titled 'Scheme to overcome slipper drive sludging' shows six slots machined in the RE 7147 front drum of the earlier Wraith damper and the RE 4030 drum of the later damper that was carried through to the Silver Cloud. Note that these drawings show slots in the front drums.

STANLEY BULL WROTE FROM HYTHE ROAD ON 19TH MAY TO HARRY GRYLLS AT CREWE.

"It has occurred to me that the trouble which has developed since you introduced the slotted flywheels is that you have not at the same time increased the supply of oil to the cotton duck washers.

In our earlier experiments a few years ago, when we added radial oil escape holes, we thought it advisable to increase the oil supply by adding flats on the crankshaft to provide a more positive oil feed. The flats, of course, were coincident with the oil holes and were about 0.002 to 0.003 (inch) deep. We did quite a few dampers some three or four years ago and have had no trouble with them since"

FERODO WASHERS

By mid-June 1954 the company had achieved satisfactory rig test results with rigid friction washers made of VM41 and MP Ferodo materials. Arrangements were then put in hand to fit them to a number of experimental cars. A few days prior to this decision a test had been concluded and a report raised, EER 897, which had compared different dampers, including a 10 inch Girling viscous damper on the effects of a boom period on the new Siam car engine. The viscous damper was only found to make things worse, although the test was not particularly thorough.

KDA132

ROLLS-ROYCE CRANK SHAFT DAMPER TESTS

Object: - To determine stick/slip characteristics of various Ferodo materials in respect of the above application.

Procedure:- 1. Initial thickness measured.

2. Tested in oil (Static & Kinetic)

3. Soaked 48 in oil.

4. Checked for swelling

5. Tested as (2) above.

6. Clutch assembly fitted to standard test machine and given a 48 hour bedding in period

7. Tested as (2) above.

48 Hours = 320,000/396,000 full oscillations (X2 for total)

None of the materials tested showed any measurable increase in thickness after 48 hours soak in oil.

Method: - Spring balance (up to 28 lbs) suspended from torque arm at 17½" Radius. Container attached to balance and water poured gently into container until first movement was recorded and "pull" noted. Further water similarly added to full slip - Pull noted.

FIG 7 DAMPER FERODO TEST PROCEDURE

ROLLS-ROYCE CRANKSHAFT DAMPER TESTS 28.7.54

At 17½" Radius

MATERIAL	BEFORE SOAK	AFTER 48 Hrs. Soak	Before 48Hrs Bedding	After 48 Hrs.Bedding
1.L-Set 1	Static 11¾	11¾ Lbs	11¾ Lbs	11½Lbs
	Kinetic 12¼	12¼ Lbs	12¼ Lbs	12 Lbs
L-Set 2	Static 11½	11½ Lbs	11½ Lbs	11½Lbs
	Kinetic 12	12 Lbs	12 Lbs	12 Lbs
.LH.1 Set 1	Static 12	12 Lbs	12 Lbs	11¾Lbs
	Kinetic 12½	12½Lbs	12½Lbs	12¼Lbs
LH.1 Set 2.	Static 12¼	12¼Lbs	12¼Lbs	12 Lbs
	Kinetic 12½	12¼Lbs	12¼Lbs	12 Lbs
3.WF.3 Set 1	Static 11¼	11¼ Lbs	11¼Lbs	10¼Lbs
	Kinetic 11¼	11¼ Lbs	11¼Lbs	10¼Lbs
WF.3 Set 2	Static 10¾	10¾ Lbs	10¾Lbs	10½Lbs
	Kinetic 10¾	10¾ Lbs	10¾Lbs	10½Lbs
4.MP Set 1	Static 12¾	12¾ Lbs	12¾Lbs	10½Lbs
	Kinetic 13	13 Lbs	13 Lbs	10¾Lbs
MP Set 2	Static 13¼	13¼Lbs	13¼Lbs	10¼Lbs
	Kinetic 13¼	13¼Lbs	13¼Lbs	10½Lbs
5.VM.41 Set 1	Static 9¼	9¼Lbs	9¼Lbs	10 Lbs
	Kinetic10	10 Lbs	10 Lbs	10¼Lbs
VM.41 Set 2	Static 9½	9½Lbs	9½Lbs	9½Lbs
	Kinetic 10	10 Lbs	10 Lbs	10 Lbs
6.MF Set 1	Static 13½	13½Lbs	13½Lbs	14½Lbs
	Kinetic 14	14 Lbs	14 Lbs	14½Lbs
MF Set 2	Static 19½	19½Lbs	19½Lbs	18 Lbs
	Kinetic 20	20 Lbs	20 Lbs	18 Lbs
7.J1	Static -	11 Lbs	11 Lbs	11¼Lbs
	Kinetic -	11 Lbs	11 Lbs	11½Lbs

FERODO LIMITED
CHAPEL-EN-LE-FRITH
AUGUST, 1954

FIG 8 DAMPER FERODO TEST RESULTS

The pursuit of an alternative to the cotton duck friction washers continued, and the initial test results of rigid washer materials from Ferodo were received at Crewe in August 1954.

The test procedures and results are shown in FIG 7 & 8. The company had conducted their own tests and were to eventually decide on the VM 41 material for production, which is shown as item 5 in the results.

Methods of friction measurement and preparation of friction washers

The test conducted by Ferodo and some remarks made by Stanley Bull bought to light some rather strange and fundamental points. To discuss these points we must deviate temporarily from the chronological date order.

Anyone who has overhauled the crankshaft damper on one of these post war six cylinder engines will not have failed to notice the Workshop manual requirement to use a spring balance to determine the slipping poundage. In various articles written on the subject, this method is stressed to say the least, and yet strangely was not introduced as a post war production technique until September 1955.

Ferodo engineers had been well versed in the procedure for testing the slip loads by using a spring balance. When they forwarded their report on 25th August 1954, the covering letter contained the following paragraph.

“You will also note that the method of checking the pull was carried out by attaching a container to the spring balance and loading this accordingly with water, and it was considered that this was more accurate than trying to determine the pull on the spring balance by hand”

In a memo of 11th July 1955, commenting on the minutes of a Modifications Committee Meeting held on 15th June 1955, Stanley Bull had the following to say.

“One other point on which Hythe Road differs from Crewe is that we use a spring balance to test the poundage and not a dead weight. The latter may not detect a slipper drive, which is ‘juddery’, but a spring balance will, and this does enable us to throw out any dampers, which have a variation of more than 2lbs, which we know will give trouble” (This also applies when a torque wrench is used, thankfully almost unheard of in the U.K. NWG)

“We have always believed that the rotary ironing process used for many years was better than just compressing the lining, but, of course, this is laborious and it can be overdone.

We have tried to reduce the time required to obtain a reliable friction setting, and we believe this is now possible by a very simple expedient. This consists of soaking the washers in a lubricant consisting of 75 % (Castrol) Hi-press and 25% SAE20. The washers are soaked for no less than 24 hours, and are compressed for a further 24 hours”

It would appear that Stanley Bull’s opinion was respected as on 2nd September 1955 the following instruction was raised, the full text of which read:-

“Pre-build Treatment of Spring Drive Cotton Duck Washer EW 935

On production the treatment at present given to these washers is to soak them in SAE 20 oil followed by a period in a press. The soaking and pressing periods are supposed to be of 24 hours duration each but the procedure has apparently never been given the full status of a planned operation and the times are not adhered to. We therefore wish to regularise the treatment by making it a proper operation and also bring it into line with that which Hythe Road have found to be advantageous.

The treatment is to be as follows:

- The washers to be soaked for a minimum of 24 hours in a lubricant consisting of 75% Hi-press and 25% SAE 20 oils.
- The washers to be transferred directly from soaking into a press where they are to be retained under a pressure of 75 lbs./sq.in. For a minimum of 24 hours.

In addition to this, the method of measuring the spring drive slipping load during assembly is to be carried out by means of a hand operated spring balance instead of by weights as at present. The purpose is to ensure the detection of undesirable stick/slip characteristics”

It would appear from the above evidence that no post war slipper drive was set on production with a spring balance until the Silver Cloud / Bentley S1 engine production had already commenced. Latter tests did in fact show that the dual mix of oils used for initial setting of the friction slipping loads did not give any advantage over the use of straight SAE 20 oil.

Still pursuing a slotted damper

In spite of delaying the introduction of the slotted damper in May 1954, the pursuit continued. Re-joining our slotted damper story brings us back to 21st September 1954, when EER 949 was issued. The title of which was… "Spring Drive Unit, Shortcomings and Recommended Modifications". The conclusions of which are shown in full:-

"The principle faults and modifications required to overcome them are as follows:-

1. Increase in torque setting due to sludging can be avoided by provision of oil release slots
2. Fatigue failure of the spring plate spokes is primarily due to sludging, and this is relieved by the same modification.
1. Unsuitability of the cotton duck friction washers in the following respects:-
2. Inconsistent damping performance due to the presence of a joint.
3. Tendency to disintegrate or rot apparently as a result of faulty inhibiting treatment --- a not uncommon occurrence.
4. The ease with which it can become trapped on its outer periphery on account of its soft pliable nature.

Friction disks in Ferodo VM 41 material have proved on test to be at least comparable on general performance. Being joint less and fairly rigid they avoid faults (a) and (c) and there is no indication that they are susceptible to corrosion.

We therefore recommend the introduction of slots and Ferodo washers"

The main body of the report mentions that sludging could be avoided by the use of oil release slots, and this had been proven. That axial vibrations of the pressure plate and friction drum, to surprising amplitudes, appeared be the true cause of friction washer trapping and that this was most likely to occur when the engine was overdriven, not under its own power, to around 4500 rpm. Fatigue failure of the spring pressure plate spokes had been traced to a combination of high torque loading on sludged dampers and bending through axial vibration. It was reported that, except in one instance, no failures had occurred (of spring plates) on dampers with oil release slots. The exception was a suspect case since the pressure plate had previously been used in an un-slotted damper. Just after this report was published it was also decided to discontinue the separate balancing of the spring drive sub assembly before adding it to the crankshaft.

Ferodo was still conducting testing in January 1955 in respect of static and kinetic friction properties at various friction face temperatures and pressures. Initial production slotted dampers were being built in December 1954 with the Ferodo VM41 friction washers part number UE 2549. These washers having been reduced in outside diameter and the outer edges chamfered to prevent any fouling between the inside and corners of the damper drum. The slotted dampers were eventually used in the last 94 engines fitted in cars prior to the introduction of the Silver Cloud and Bentley S type.

At least by 1953, company engineers at the sharp end of the business were only too well aware of the problems arising from sludged dampers but this practical experience does not appear to have been shared by all the top division of engineering management. Stanley Bull was still a big believer in eliminating the sludge in dampers, he wrote to Grylls, chief engineer R-R Motor Car Division on 2nd February 1955, on the subject of slipper drives.

“Firstly, I think it is unquestionably a fact that we require oil circulation through the slipper flywheels to prevent sludging, which does appear to be extremely undesirable and probably is a major factor in causing breakage of the spring plate.

So far as I know, the decision to abandon slots has been a result of cases of trapping of the cotton duck washer. The theory presumably being that with better lubrication the cotton duck washer is better able to slide or move radially, and the only place it can move is the annular space between the flywheel and friction plate.

It seems to me that the flexible and deformable material like the cotton duck washer ought not to be spigoted or restricted on its outer diameter; also if it must move radially, there should be space for it to be accommodated.

On our pre-war designs there have been several different ways of avoiding this trouble. In the earlier designs the cotton duck washers were spigoted on the inner diameter and had ample space to spread radially. We had trouble with trapping on the spigot diameter until the dimensions were put right. The cotton duck actually was too small for the spigot.

On the pre-war Bentley no attempt was made to control the cotton duck washer either internally or externally. We had no trouble with trapping, and I do not think there was anything wrong with this design except that today we would probably criticise it because the cotton duck washer could assume an eccentric position and be out of balance.

I would like to add that on all pre-war Bentleys we had oil escape holes in the flywheels so there does not seem any good reason why we should not today have suitable escape holes or slots without detriment to the location of the cotton duck washers.

Summing up, it seems to me that we require oil escape holes or slots and either an internal register for cotton ducks or greater radial clearance to avoid trapping”

Grylls replied the next day, and although his day to day experience in the results of factory testing far outweighed that of Stanley Bull, the converse appeared true in regard to practical up to date experience at the sharp end. He wrote to Stanley Bull.

“…On the other hand, I think that it is true to say that our customers have had little trouble with spring drives since the war. We have objected to the appearance of sludge and we have objected to broken window plates, (he was referring to pressure plate broken spokes) but the customer has been blissfully ignorant of what is going on. There have been just a few cases where dissembling a spring drive has been said to cure a rough engine, but it is obviously difficult to substantiate this assertion when so much of the motor car has also been disturbed.

Ninety four engines have left the factory with Ferodo washers and it will not surprise me if they all give trouble.

A combination, which proved quite unsuccessful, was the use of cotton ducks and large release slots in the periphery of the damper. FJH thinks a likely explanation is that endwise relative motion of the window plate occurs because we have lost viscous damping. The tiny release holes used before the war may not have had this effect.

It is even possible that the pre-war release holes only worked for a short time and I seem to remember that this was the case.

Stanley Bull’s practical experience was obviously not going to let him rest the case at that point, on 7th February 1955 he replied.

"I am afraid the Service Department has never shared your views that the owner or driver of the car is unaware of the existence of a slipper drive which has become solid or has excessively high poundage.

We have many complaints of engine periods or vibration, which are cured by attention to the slipper drive, and we must have dealt with hundreds of post war slipper drives. Therefore, we need any improvement possible which will maintain a reasonably constant poundage..."

The only other document of interest in respect of the present discussion is report EER 1162 dated 30th November 1955. It is also of interest because, although raised by the same authors, it contradicts the finding of the EER 949 report raised only just over one year before, and is an example of the contradiction of findings throughout the whole subject thread.

The use of axial vibration stops mentioned in EER 1162 had previously been used to prove that the effective damper pressure plate load does in fact substantially reduce during certain periods of damper excitement. A point, which is extremely important to the practical understanding of what happens in practice compared to what should happen according to the masses of theoretical calculation. When axial stops had been originally fitted to test axial movement, the pressure plate had contacted the stops sufficient to make very heavy contact depressions on the ends of the stop studs. In this particular instance, any effective loading on the friction washers has all but disappeared when part of the pressure plate spring load is taken off the friction washers.

EER 1162 is significant in that it appears to close off any further development of dampers. It is titled "Results of various modifications applied to the spring drive on experimental car engines". The conclusion lists the following:-

"The damping performance of spring drives with oil escape slots has been generally most unreliable despite various modifications.

The existing un-slotted unit with its known major drawbacks of sludge choking is therefore still preferable.

We believe that an efficient slotted type can be developed, but we cannot at present afford the time in view of work urgently required on new designs. Further development will be carried out as and when time allows"

The reason for raising the report says:

"When the combination of a slotted damper and Ferodo washers was found wanting as regards damping performance it appeared that the fault lay in the friction characteristics of the Ferodo material. We therefore tried to retain the slotted unit together with cotton duck washers and the addition of axial vibration stops to overcome washer trapping which was the original objection to combining slots with this type of washer. Experiments with increased pressure spring loads and increased inertia dampers have also been conducted."

Modifications Tested:

1. Axial vibration stops: These took the form of 3 setscrews in the front damper wheel, which was drilled and tapped accordingly. The screw heads were on the inside of the wheel with packing washers under the heads so as to leave 0.010 to 0.015 inch clearance between the heads and the front surface of the spring plate with the damper finally assembled. The effect is to prevent axial movement of the spring plate relative to the damper wheels to the extent necessary for the front cotton washer to become trapped.
2. Increased pressure spring load: this was obtained at first by fitting an additional spring inside the standard pressure spring and, later when available, a special pressure spring. The theoretical increase in torque load is approximately 70% and 100% respectively but in practice the increase was

found to be some 20% to 40% less than this. The original object was to maintain adequate damping with Ferodo washers.
3. Spring plate RE 20256 on which the spoke width is increased from the standard 0.375 to 0.700 inch with the object of avoiding spoke fatigue failure.
4. Increased inertia damper CEX 8598.

Attached to this report are the tables of results on experimental car dampers. Most of the cars were still in service at the time and the dampers had largely not been stripped, in fact, of the eleven cars listed only four dampers had been physically stripped. The majority of this report was therefore based on the driveability of the car, and the same person was not conducting the car to car comparison! The tables are not included in this article. Having said that, the discussion section in this particular report is very illuminating. In view of this I have reproduced the report almost in full.

The section dealing with the reporting of the results makes initial reference to the tables and goes on to say:-

Results:

1. Where the only departure from standard is oil release slots or slots plus axial vibration stops, the damper is unsatisfactory due to early development of a pronounced half torsional period. When this obtains the damper torque loads are high and the action very jerky as a result of a large difference between the static and kinetic slipping loads developing.
2. Axial vibration stops do not influence damping performance but do restrict spring plate vibration at least in a slotted unit. No sign of washer trapping has been observed where stops are fitted.
3. With the departure from standard of increased load pressure springs in addition to oil release slots there is indication that the damping performance is less liable to deteriorate than with slots only.
4. Pre- build treatment of cotton duck washers by soaking in 75% Hi- press / 25% SAE 20 oil mixture as against SAE 20 only, although a help in obtaining a smooth slipping action on initial build, does not appear to have any subsequent value in a slotted unit.
5. The increased inertia damper with additional mods of oil slots, vibration stops, higher load pressure springs and wider spoke spring plate has run 12,000 miles in Continental test car 25B. The half torsional period is not severe but is nevertheless readily noticeable, and this damper is certainly no better than a normal damper with oil slots and high load springs. The strengthened spring plate was fitted in this damper when failure of the spokes on the standard plate was observed on the last strip. How it has fared is not known, as the damper has not since been available for inspection.

Discussion:

From previous experience we know the slotted damper with Ferodo washers to be unsatisfactory. We know that this is also true when cotton washers are used. The reason for the failure of Ferodo discs was thought to be loss of effective load due to kinetic friction decreasing. On the other hand the load with cotton washers increases but the result as regards loss of damping is the same.

Now if a damper is built with normal smooth action, but a low friction (10 lb. or less) the damping is inadequate, but with smooth action and a high friction load (up to about 30 lbs.) there is no noticeable adverse effect. It appears then that there is a further important factor, and this, we believe may be the very jerky action due to the static / kinetic load difference that develops in both cases.

We do not know for certain why the friction characteristic changes; rig tests conducted by Ferodo Ltd failed to reproduce it on their VM 41 material. The rig tests, however, did not centrifuge sludge bearing oil between the friction surfaces, whereas in every engine instance we find a sludge deposit on the inner bore of the washers, due to edge filtering action at this point, a film of sludge on the friction surfaces and cotton washers are invariably impregnated with sludge. It is therefore regarded as quite feasible that the change in friction characteristic is caused by sludge deposit formed in the bore of the washers penetrating to the sliding surfaces when axial vibration temporarily separates the surfaces.

If this were the case the reason for apparent longer useful life when stronger pressure springs are used in slotted dampers might be quite reasonably explained by the higher loading preventing axial vibration except under severe conditions. With un-slotted dampers a very rough action is not normally noticeable, but in this case it is the heavy sludge deposit rather than frictional load, which controls the damper motion and the same change in friction characteristics, may still take place. Here again, however, axial vibration will be less severe due to the damping effect of the oil trapped inside the damper. On the other hand there is the other major difference, namely the much 'drier' running condition on slotted dampers.

Un-slotted dampers will always suffer performance deterioration due to sludge deposits, but are still less prone to run us into serious trouble than any variation we have tried with the slotted type. It should nevertheless be possible to develop a reliable damper with a through oil flow, but at present urgent work required on new designs prevents us from devoting further time for intensive development on this line. From our present knowledge of spring drive behaviour and the arguments advanced in this report the arrangements which we will investigate when time permits, are as follows:-

a) A slotted damper with the present annular type of friction washer replaced by a number of friction pads of the Ferodo type of material bonded onto either side of the friction drum surfaces. This would allow free oil circulation and avoid the edge filtering action.
b) Stronger pressure springs in conjunction with annular type washers in a slotted damper with oil channels in the metal friction surfaces.
c) Axial vibration stops and the wider spoked spring plate in all slotted dampers.
d) Alternative friction materials, in particular a sintered metal which is actually being rig tested by Ferodo"

LAST OF THE ARCHIVE

From this point onwards the archive files do not appear to provide any further information on major damper development, although the dimensions of the existing cotton duck washers were altered as late as March 1956. The last significant memo was written on 22 April 1958 during discussions regarding the crankshaft and damper arrangement on the forthcoming F60 engine, eventually supplied to the then B.M.C operation at Cowley. The first paragraph reads:

"Experience with the present UE 281 spring drive on 'S' series engines shows that with a flexible crankshaft, torsional amplitudes of about + 0.2° produce audible gear rattle on some engines (but not on all) at about 2500 rpm, the 6 / rev. vibration speed, and at speeds above 4000 rpm where the amplitude of the 3 / rev. vibration is rapidly increasing, gear noise again becomes noticeable. Crankshaft torsional failure can be reproduced by over speeding up to (say) 5000 rpm on the over-run. (I have mentioned this fact in articles on many occasions….NWG) It does not seem to have been satisfactorily proven that the present spring drive does in fact work very effectively as a vibration damper"

DISCUSSION, HINDSIGHT, DEVIATION, AND PRACTICE

I have underlined certain sections in the paragraphs below that I believe the interested owner should take on board to ensure durability of the car engine.

The finite reason that the slotted dampers were discontinued and not carried over to the next generation is unclear, as the various reports are often contradictory. If any single reason could be given for the failure of dampers to perform repeatedly to a desirable path, it must be sludge that was the main cause. In no instance was sludged oil used during the initial friction poundage settings, and yet this was the environment in which the friction surfaces would eventually be operating. Sludge undoubtedly affects any friction medium used in these dampers and unless steps are taken to reduce sludging and to some degree eliminate edge filtering of sludge on the inner diameters of the friction washers, no friction material will perform for long.

There are however many distinctive conversational underlines in the files, which gives one the impression that although sludging was a problem, other factors were present, especially the axial movement of the pressure plate. It is clear that some dampers did not perform in practice entirely as the design envisaged, probably the result of sludge, and experiences with cotton duck were not much better than the Ferodo material, save that with slotted dampers sludge did not build up to the same degree.

Very smooth engines were found sometimes with damper poundage's set at over 50 lbs., quite contrary to the workshop manual setting. It seems equally obvious, with some hindsight, but also hinted at in the files, that the differences between the actual static and kinetic friction breakaway poundage were more important than the actual bench setting. There was also tremendous pressure on engineering staff to shake down any potential problems likely to be raised with the introduction of the new car model. In addition an understandable fear that any problems with the introduction of a new slotted damper would jeopardise the new car reputation. Other files in the archives also lead one to believe it was expected that the V8 engines would replace the six cylinder much sooner than it did in practice. Taking all things into consideration it is perhaps not surprising that very quickly all R & D work on dampers, whether it be cotton duck or Ferodo washers, was closed down.

This then is the later potted history of the damper, which appears in the archive records. Meantime it is at least necessary to discuss a few points in regard to the actual spring drive and also relate some practical experiences in connection with the dampers.

MATERIALS

After the demise of cotton duck friction washers, Tufnol type washers were offered as the alternative. Whilst the original cotton duck provided an initial smooth operation, eventually the material adheres to, or rusts, to the faces of the friction drum or rear damper wheel surfaces, a condition that we will call seized. This is a condition that cannot be avoided with any absorbent friction materials working against a ferrous surface, neither can it be avoided by regularly turning the engine over with the starting handle, as two friction faces of the four will tend to seize. With two working friction faces the damper will give the appearance of freedom, when tried with the starting handle, but the true slippage picture is very different.

In general, although the later Tufnol type material does not seize because its surface is impervious to liquids, particularly water, its ability to provide a smooth action between the friction surfaces is, to say the least, suspect. In short the static friction is very different from its kinetic friction.

Over succeeding years and especially at present, a large number of alternative materials have been suggested and placed into service. The one, almost overriding reason given is that the modern materials do not suffer the drawbacks of the original cotton duck or tufnol materials. Whilst this may be true, they are operating in a damper, which is not performing as either the supplier or enthusiast have been lead to believe, and sludge will eventually take its toll. In this situation the enthusiast is trapped between a rock and a hard place. Sludging of the friction surfaces takes place quite rapidly and the only option in my view, whatever material is used, is to drill the damper to centrifuge the oil, sludge and any water.

The choice of the friction materials is purely down to the owner, but careful thought needs to be given to the true actions of the damper and spring drive in service. Whatever friction material is chosen, consideration needs to be given to the number of actual working friction surfaces, the poundage settings that need to be higher than normally suggested, the disposal of residue sludge, the static and kinetic friction slippage and the setting and effect on the radial spring drive.

It would seem that it is sensible to provide for some form of radial gaps or depressions across the friction material faces or those of the mating metal faces, to provide an oil flow path. It is difficult otherwise to visualise how sludge build up on the inner lip and face edges may be flushed away. If sludge is allowed to accumulate on the inner lips centrifugal action will eventually force sludge between the friction surfaces in large quantities. Undoubtedly some minor sludging will occur but with radial slots in the friction material faces, this will be kept to a minimum. The physical properties of the chosen material will dictate how these radial slots can be arranged.

PERCEIVED OPERATIONAL TROUBLES AND OVERRUN DANGERS

The normal owner is unlikely to experience the extreme critical crankshaft period in the high engine revolution band during sensible highway driving, but the engine will experience vibrations in other RPM bands and in particular the next critical band around 2500 rpm. Much will depend upon the damper condition and power output demand from the driver, as the worst will only be experienced when the car is driven through the bands at full throttle. This latter point is important as the only method directly under the control of the driver is to drive through the 2500 rpm engine range at full throttle, to ensure a good damper keeps working. Unfortunately the average enthusiast might perceive this as ill treatment of the beloved car, but it should be placed on the list of necessities when out for a drive. Even so the damper friction surfaces will try to slip across two faces only, and this was discussed in an earlier article. On a frequent basis, dampers are seen in a seized condition, largely because of sludging and long periods of engine idleness, but also because the car has never been driven at full throttle through the mid-range critical band.

A largely unknown danger is over running of the engine from medium or higher speeds, it is destructive to many engine components. Most owners may have attained a higher engine rpm in an intermediate gear than they have in top gear, sometimes unwittingly, when descending a steep incline. If a downward gear change is undertaken and, particularly if heavy braking immediately follows this, the stresses caused by torsional reversals can be tremendous. The engine can be required to go from full or nearly full gas load above the pistons at a high power demand to a no gas load situation and full over run. Whenever fairly high powers or speeds are demanded it is a wise driver who avoids excessive and sudden engine overrun.

It is the suddenness of this action, the instantaneous engine speed change, combined with reversals that are so destructive. In any event, radial spring drive springs in a seized damper will be subject to hammering and will not survive this treatment for long, and neither will the fingers of the damper spring pressure plate, which have to transmit damper torque.

Analysis of the company crankshaft breakage tests and the few service breakages indicate that the most damage was done in the quickest time, in engine overrun conditions.

AXIAL MOVEMENT AND VIBRATION

Company experiments proved the existence of axial movement and vibration of the spring pressure plate and consequential breakage of the spring plate spokes or window panes. The experiments even went further and indicated the presence of axial movement at the rim of the complete damper assembly. If one accepts that the pressure plate loads will be relieved from the surface of the friction washers at some stage in the operation range of the engine, then this could result in the friction washers rotating in relation to the damper drums. Whether or not this relative rotation is normal, even when the pressure plate load is constantly applied, I would not like to say. I would however contend that this relative rotational movement can be seen to have occurred very easily, at least when the damper is fitted with any rigid friction washer.

As an example, I conducted a test with Tufnol friction washers. The Tufnol surfaces were drilled with fifty small holes, to provide small oil reservoirs in the friction faces, and the damper rebuilt as per the workshop manual. During the rebuild the friction washer positions, relative to the damper wheels, were marked. The engine was then operated for some 3000 miles over a one year period. When the damper was stripped, the friction washers had rotated. On another occasion, when the friction disk was drilled with just one marker hole and subsequently operated, the same rotation could be seen to have taken place. This rotation effect can be seen from the relationship of the witness marks, through the marker holes, caused by humidity on the metal mating surfaces. In the two tests I conducted the witness holes were some 0.500 inch apart. Whether the disks had moved 0.500 inch or actually rotated one or more turns is open to conjecture. One has to presume that in the case of flexible friction washers like cotton duck, the same rotation will try to take place. Strangely, no company records show any tests being completed to establish whether the friction washers rotate relative to the metal surfaces.

Although conjecture I believe this is what actually happened to the cotton duck washers that became trapped on testing, and that the washers rolled up on themselves when they were forced to rotate as the clamping pressure was destroyed when the pressure plates lifted off.

It was also known that the outer mass of the damper could vibrate axially with amplitudes of up to 0.020 inch. Just around the time when the UE 2549 Ferodo friction washers were considered, it was decided to relieve the outer edges of the washers by 0.030 inch at 45°. This was to prevent fouling in action around the outer periphery. When I originally conducted my own first test with tufnol washers I found that I could detect where the rear washer had contacted for about 0.062 inch on the outer rear edges with the face of the rear RE 3940 damper drum. Although the drum face was square and had been reground, the outer face was just highlighted. It occurs to me that if the outer metal mass of the damper moves axially it is possible that the outer edges of very rigid friction washers are also forced to move axially and perhaps the pressure is relieved slightly in the centre of the friction washer faces. Rather the rear drum had appeared to take on the cross section of a dinner plate, gripping the flat friction washers at the edges. I was eventually to use entirely different materials but this point was not lost on me and I ensured both outer and inner edges of the friction material cleared any metal face by 0.100 inch.

Today we are in a different environment than the engineers of 50, 60 or 70 years ago. We have better oils and in the main tend to change our oils and filters more often. Suitably modifying a damper for drainage and spring pressures and using better materials can overcome the past damper troubles, at least on post war engines. The same principles apply to pre-war engines and I have little doubt that similar modification would produce very desirable results.