Phantom I Damper Front Side

Rear Side 37LC
Michael Forrest [UK], RREC Bulletin

ROLLS-ROYCE & TORSIONAL VIBRATIONS
A MONOGRAPH BY JOHN G. CROFTS
COLUMBUS, INDIANA

Rolls-Royce & Torsional Vibrations

BY JOHN G. CROFTS

1. Background

For the first 3 decades of this century, one of the significant technical problems in the development of automobile engines was crankshaft torsional vibration in 6-cylinder in-line engines. 2-, 3-, and 4- cylinder engines have short crankshafts, too stiff in torsion (twisting) to be seriously affected, but when 4-cylinder engine designs were expanded to 6-cylinders in line, a serious problem of torsional vibration was encountered.

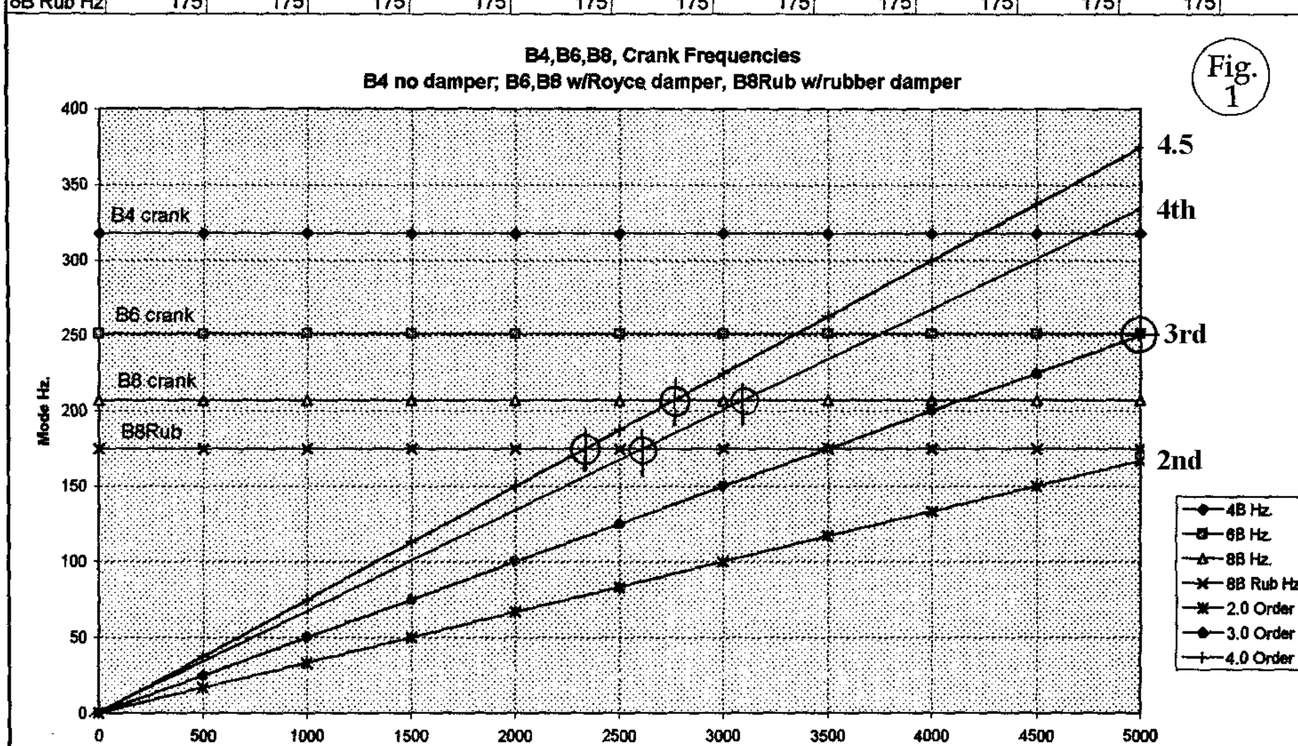
This form of vibration has the whole length of the crankshaft twisting against the heavy flywheel, so that the free end of the crankshaft alternately accelerates and decelerates instead of turning smoothly, whenever the natural frequency of the crank is excited. The complex force pattern of a single cylinder 4-stroke cycle contains many harmonics within it, which can excite the crankshaft at different speeds. Of these forces however, the major excitation comes from the firing frequency of the cylinders, which combine to twist the crank along its whole length <u>when a massive flywheel is attached to the shaft.</u>

By convention, vibrations within rotating machinery are identified by the number of cycles per revolution of the shafting, thus one complete cycle of vibration in one complete revolution is called a 1st order vibration, two cycles per rev. is called 2nd order, and so on. So a 4-cyl., 4-cycle engine has a 2nd order firing frequency, a 6-cyl. has a 3rd order firing frequency, and an 8-cyl. has a 4th order frequency, based on the number of cylinders which fire per revolution.

A convenient way of portraying the relations between orders of vibration, natural frequencies of shafts, and engine rpm is by way of a "Campbell" or Interference Diagram, shown in Fig 1 for three types of crankshaft, 4-, 6-, and 8-throws per shaft. When the frequency of the firing impulses of the engine approaches the natural frequency of the crankshaft in torsion, the phenomenon of resonance may increase the degree of twist to dangerous levels if some form of damping is not used to control the amplitude of vibration. This speed is known as a major critical speed.

Referring to Fig 2 (Transmissibility Curve), it

Eng. RPM	0	500	1000	1500	2000	2500	3000	3500	4000	4500	5000
4B Hz.	318	318	318	318	318	318	318	318	318	318	318
6B Hz.	251	251	251	251	251	251	251	251	251	251	251
8B Hz.	207	207	207	207	207	207	207	207	207	207	207
8B Rub Hz	175	175	175	175	175	175	175	175	175	175	175

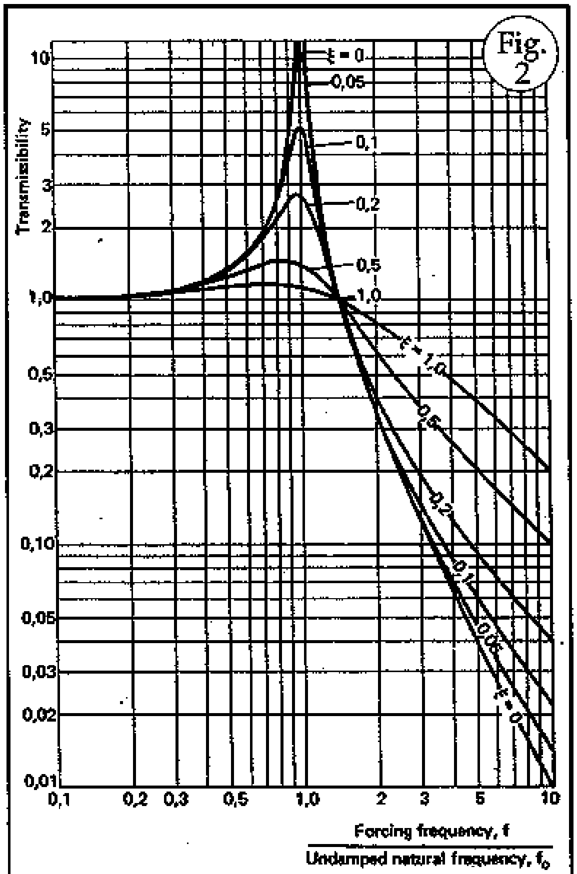


will be seen that when the forcing and natural frequencies are equal, and the system is in resonance, (ratio of 1.0 on the X axis), the motion of a system will be increased infinitely if there is zero damping (zeta = 0, transmissibility is infinite). In practice this can't happen, as all materials have some structural damping, but the more brittle the material, the less damping, and the greater amplification it will have.

This explains the breaking of a glass goblet by a powerful singer with a very pure tone of voice, sounding a note which coincides with the natural tone of the goblet when rung like a bell. The glass is driven into resonance, and the stress rises until it breaks.

Similarly, a steel crankshaft, having a fairly low damping coefficient, if driven to resonance will experience very high stress at the nodal point, and eventually break, although not instantly like glass, but in a short time.

The only way to ensure that this does not happen is to increase the damping in the system by some means, and/or ensure that the system can't be run at its resonant frequency during normal operation.



To produce isolation, select vibration isolators with f_0 such that the lowest machine forcing frequency is greater than 1,4 f_0. High damping ratio ξ protects best against vibration transmission as a machine passes through isolator resonance to attain operating speed. Low damping ratio gives best protection at high forcing frequencies.

If the natural frequency is much lower than the forcing frequency, (values higher than 1.0 on the X axis), then the phenomenon of isolation occurs, and the transmissibility of the vibration will be reduced at frequency ratios above 1.5 to 1.

The higher the ratio and the lower the damping coefficient, the greater the isolation will be.

Pure torsional vibration of itself is not usually felt as a vibration, since it is confined only to the rotating assemblies of the engine and drive train, and does not cause any unbalance, which is the usual source of perceived "roughness".

However it is frequently a source of noise if the timing gear train is driven from the free end of the crankshaft, when the angular acceleration causes noise and wear of the gear teeth as they "rattle" and "ring" instead of meshing smoothly.

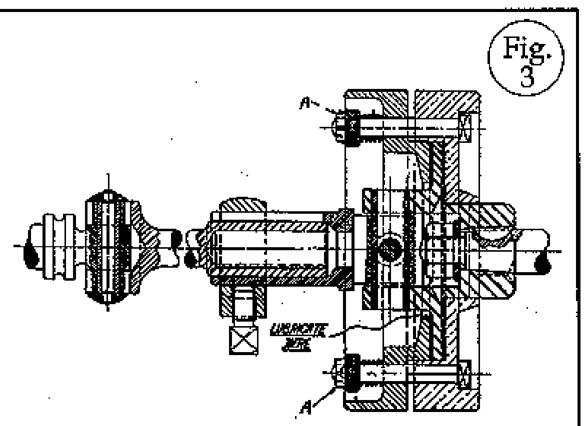
When Royce encountered the phenomenon in his first 6-cylinder engine, he spent much time and effort in trying to understand the causes, and developed pragmatic methods to reduce this vibration.

This paper hopes to outline the course of that endeavour (already covered by several writers), but with an analytical approach to design details and damper performance.

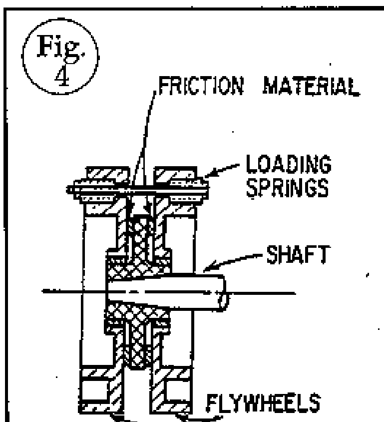
2. Origins

Henry Royce first discovered the principle of the friction torsional damper while trying to cure timing gear rattles in his first 6-cyl. engine, the 30 HP. This engine went into production in 1905 with a crankshaft designed as three twin-cylinder cranks each with 180 deg. throws, joined at 120 deg. intervals.

This design suffered from two defects, unequal firing intervals, which caused perceptible rough-



MAGNETO FLY-WHEEL



Schematic cross section through Lanchester damper.

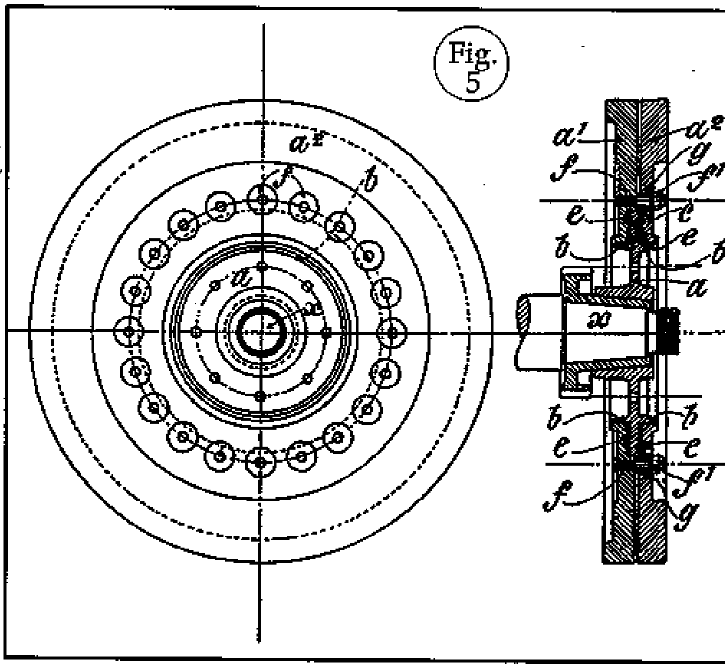


Fig. 5

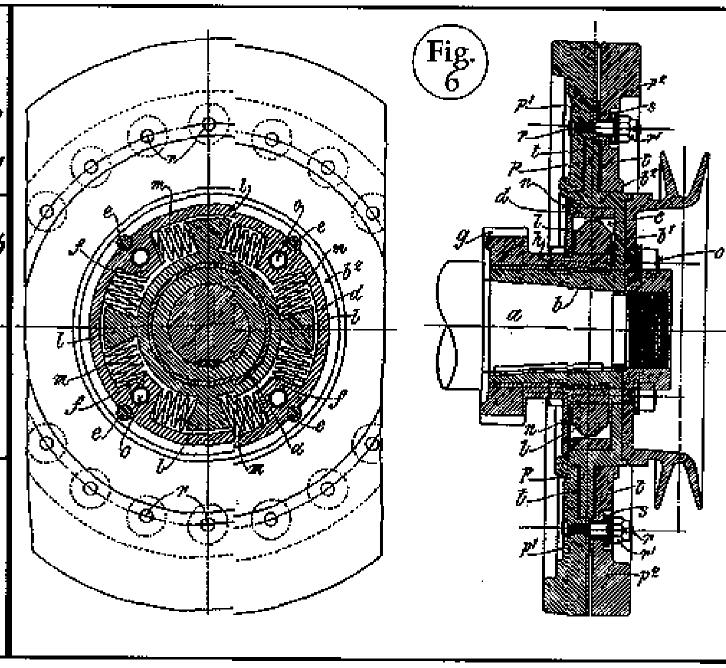


Fig. 6

ness in the engine, and a low natural frequency, as it was very slender, having the same proportions as the shorter crankshafts.

Royce bolted small fixed flywheels of various sizes to the nose of the crank in an effort to reduce the gear noise of this engine, and to begin with, he used a temporary hardwood adapter for the purpose, and later found that the noise was less with the wooden adapter than with a more permanent metal adapter.

Examination of the wooden adapter showed tell-tale scorch marks, showing that the flywheel had been slipping, and Royce deduced the damping effect of a slipping inertia from this evidence, and so called the device a "slipper flywheel."

His new insights into the nature of the problem then led him to design a very stiff crankshaft for the 40/50 HP engine, which initially was free of serious vibration, as it had a high natural frequency, a relatively low maximum speed, a short stroke and modest cylinder pressures. All of these factors combined to reduce the worst crankshaft torsionals, but another form of torsional vibration appeared, induced by the sharp 3rd order torque impulses of the magneto.

3. Magneto slipper flywheel

This component was driven by a fairly slender cardan shaft from the front gear train, and as a slight amount of wear developed in the pin joints, the natural frequency of this assembly was lowered enough to be forced into resonance, causing the gear train to rattle. The initial design had a form of friction brake, so called, to eliminate the problem, but a better solution was achieved in 1908 by mounting a slipper damper to the end of the magneto shaft, Fig 3.

4. The Lanchester patent

This is a somewhat controversial subject, and has been described in detail by Tom Clarke in his article "Royce and the vibration damper."[1] Recently more evidence has come to light, and their will no doubt be a further paper on the implications of this evidence in the future.

Dr. F.W. Lanchester, acting as a consultant to the British Daimler company, designed a multi-plate friction damper in 1909 as a retrofit device to solve crankshaft torsionals on their first batch of 6-cylinder cars. He patented it in 1910, and it became widely known as the "Lanchester" damper. Fig 4.

Dr. Lanchester, observing that Rolls-Royce was using slipper dampers in 1913, claimed that Royce had infringed his patent. Correspondence from Rolls-Royce, with copies of their drawings pre-dating the Lanchester patent, evidently settled the issue of infringement, but there has been some controversy since then over the technicalities of each invention.

5. Royce's first damper patents

In 1909, the 40/50 HP was stroked from 4.5" to 4.75", which lowered the natural frequency of the crankshaft to the point where Royce evidently realized the desirability of developing a new crank-

[1]The Flying Lady, p. 5179-5184, May/June '96.

shaft torsional damper, to prepare for the inevitable speed and power increases which the market would ultimately demand of this engine.

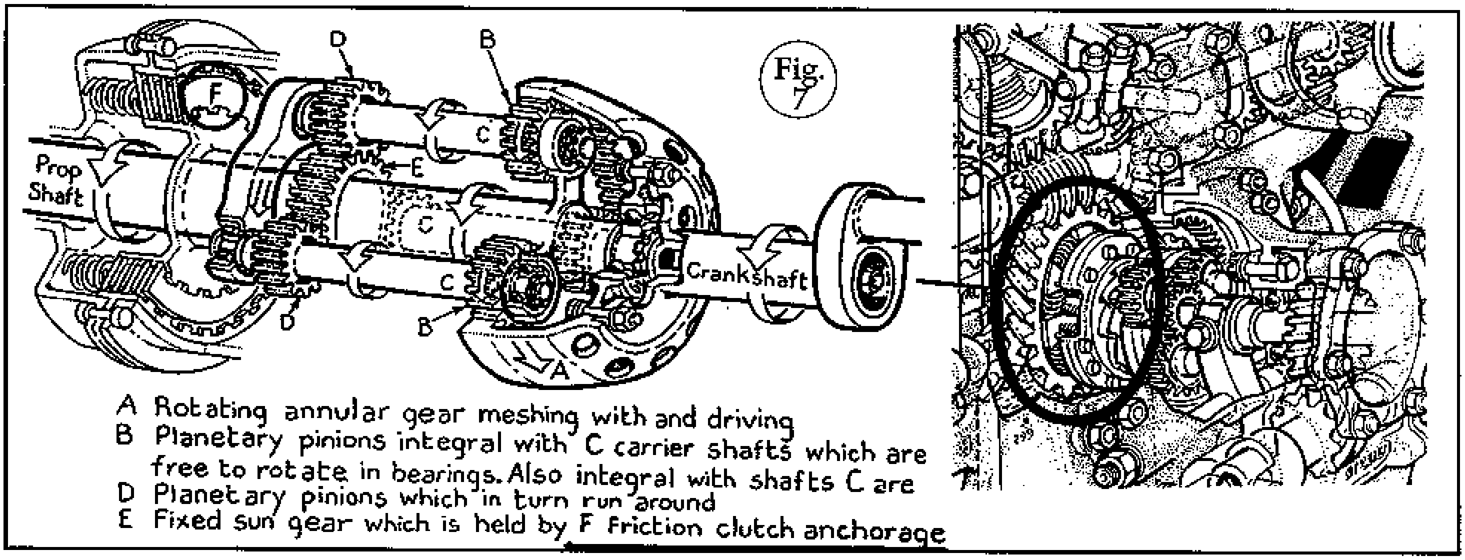
He secured two patents in 1911, the first involving an improved design of friction damper, and the second a novel approach to further reducing gear train noise by means of a spring-driven crankshaft pinion. Figs 5 & 6.

Note that the spring-driven pinion has a quite separate function from the damper, although cleverly incorporated into the basic mechanism to save space. In the light of modern knowledge, the only criticism one could make of this friction damper falls off considerably.

This was not easy to find in 1912, as there was a limited choice of suitable material, mostly natural organic substances, such as leather, cloth, felt, etc.

The material which finally found favor was a specially woven cotton disc, with the main threads running like spokes in a radial direction, sewn together and ironed and pressed at the joint to achieve a relatively uniform thickness of the disc. Oil lamp wick is probably the nearest modern equivalent.

With this material, and with the damper working in oil, the precise nature of the damping



design is that the size of the slipping inertia seems out of proportion to the friction surface, as the design challenge is the dissipation, without damage and wear, of the heat generated by the action of the damper, and the friction areas are small compared to the inertia flywheels.

The first application of a friction damper to the 40/50 crankshaft was an externally-mounted version of the 1911 patent damper on a 1910 Ghost chassis for testing purposes, dated 1912. The production version was enclosed in the gear case, so that engine oil could both lubricate and cool the damper.

6. Friction material

The most desirable attribute of a friction material for a torsional damper is a constant coefficient of friction (mu), under both static and dynamic conditions.

If the mu value is much higher for the static friction, the damper will not begin to work until a high vibratory torque level is achieved, and then once it slips, the damping effort (friction torque) process is speculative, as the cotton discs are not rigid, and there must almost certainly be some "rolling" of the main threads, in addition to both coulomb (friction) and viscous (oil shear) damping. The relative slip is very small, and became even less on the later smaller displacement and higher speed engines. However, the damper was quite successful, even if, like some other Royce designs, expensive to manufacture.

7. The "Eagle" and other aero engines

Royce clearly understood a good deal about torsionals by 1914, when he started the design of the Eagle V-12 engine. Two things about this engine are different from the car engine; first the absence of a massive flywheel, rigidly connected to one end of the crankshaft, and secondly, the fact that the firing frequency is now the 6th order, even though the crankshaft still has 6 throws.

Because of the relatively high power and speed of the engine, a directly connected propeller was not feasible. Royce therefore designed a very clever planetary reduction gear, and incorporated

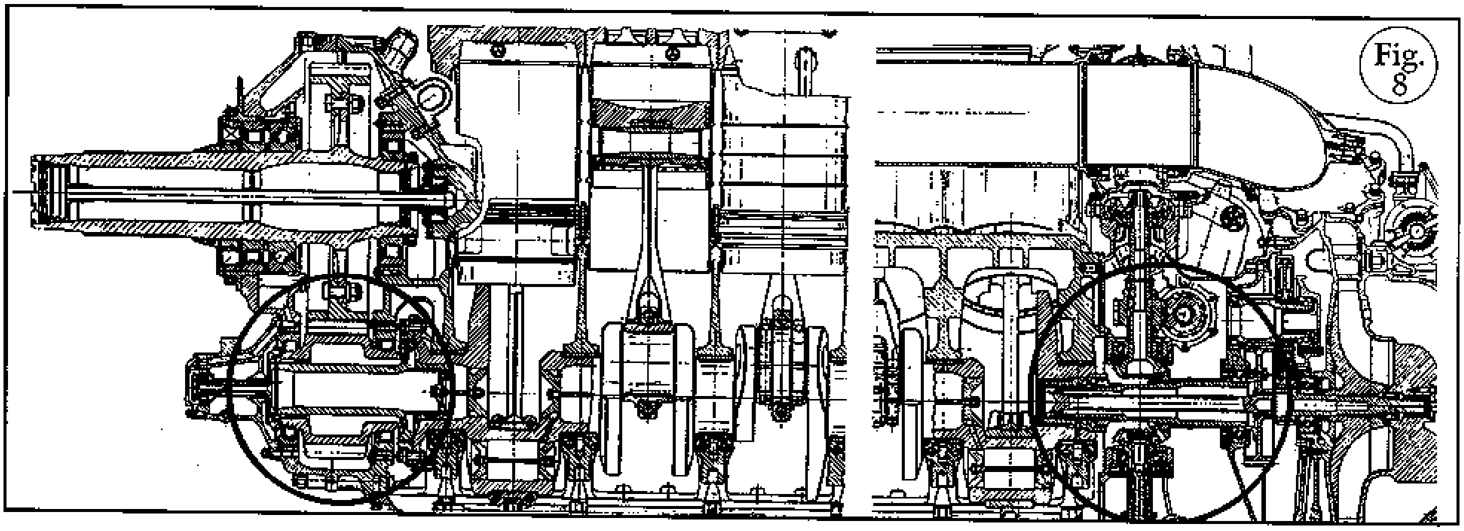


Fig. 8

the principle of a slipping clutch within this gear, in the anchorage of the sun wheel, Fig. 7. At the other end of the crankshaft, the drive for the camshafts, magnetos and other accessories was taken off by means of a spring drive, similar in design to the spring drive he had already patented in 1911 for the 40/50 HP. Royce incorporated both proven elements for control of torsional vibration in the design, which certainly contributed to the outstanding reliability of these early engines.

The planetary gear with friction clutch was only a limiting device, as it clearly must not slip under conditions of normal propeller torque, however it would slip during momentary high torque conditions, which could be experienced when accelerating the engine through the speed range, encountering critical speeds on the way which would otherwise have produced damaging vibratory torque.

With the next generation of aero engines, starting with the design of the Kestrel in 1925, a brilliant solution was found for both ends of the crankshaft, an early and outstanding example of the "center node" principle. If the crankshaft is not encumbered with a massive, rigidly attached flywheel inertia, its first mode of torsional vibration will be to twist in opposite directions about the middle of the shaft, rather than twisting in one direction along the whole length. When this happens, the cylinder firing impulses do not excite this mode, as half the cylinders add to the twisting motion, but the other half oppose the motion, thus cancelling the effect. This dramatically reduces the severity of vibration, and accounts for the ability of this design to reach the astonishing levels of speed and power output with a lightweight crankshaft demonstrated in the later Merlin.

To achieve this effect, each end of the crank must be isolated from driven inertia via a "soft" coupling arrangement, similar in principle to the spring drive. This was achieved reliably in the Kestrel and all the subsequent designs of V12 aero engines by means of a double quill shaft drive to the propeller reduction gear, and a torsion bar spring drive to the camshafts

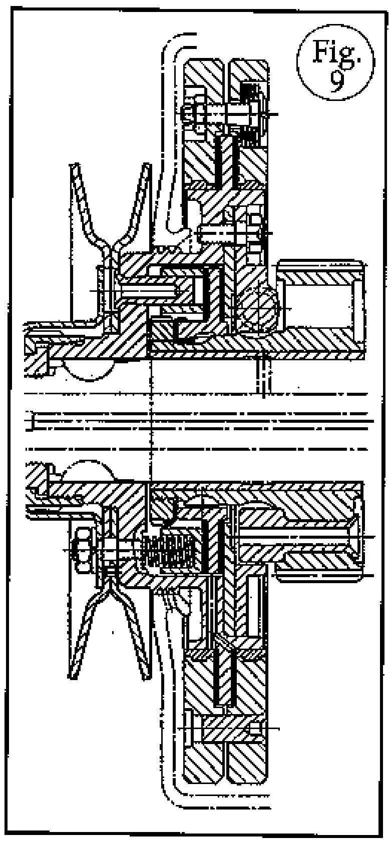


Fig. 9

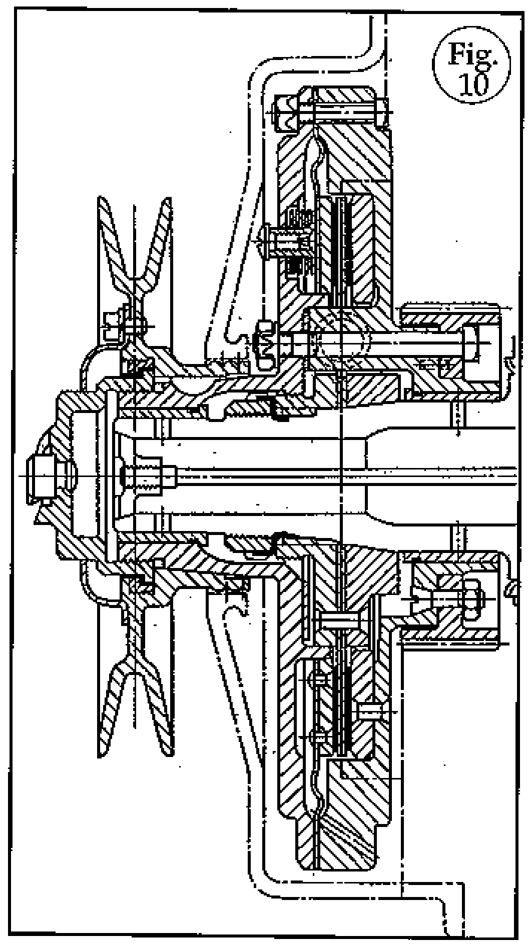


Fig. 10

and supercharger, Fig 8. While it may seem counterintuitive to relate the massive propeller at one end to the small, lightweight supercharger impeller at the other end, in fact they both have similar effective inertias due to their relative speeds. All inertias and spring rates are corrected to crankshaft speed when calculating torsional effects, and the correction is to multiply the actual value by the speed ratio squared.

Then the propeller inertia is reduced by a factor of about 5, and the supercharger impeller inertia is increased by a factor of about 80.

8. Further developments of the car damper

From this point on we will follow the development of the Royce damper related to the small 6-cylinder engine, which had a long period of continuous development, from 1919 to 1959, described in detail in S. H. Grylls' splendid paper, "The History of a Dimension." (I. Mech. E., 8/10/63).

The first production damper arrangement for the small engine series was a modification of the 1911 patent design incorporating a small additional friction damping device for the spring-driven gear assembly, Fig 9, also used for a time on the larger Phantom engines.

As will be seen, this was a compact device, but with a large number of parts, and therefore expensive to manufacture and repair.

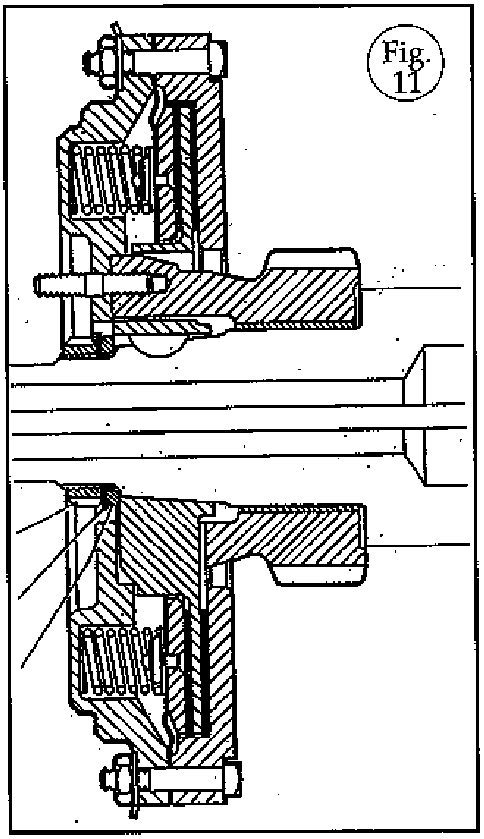


Fig. 11

However, it incorporates for the first time the important principle of attaching the crankshaft pinion directly to a slipping inertia mass, which changes its behaviour.

A further, significant refinement in both cost and effectiveness was made in the dampers for both the Phantom and the small engines by what was called the "low inertia" damper design, Fig. 10, which incorporated the damping function for both the gear and the crankshaft in one friction element, by the simple expedient of attaching the gear directly to the main inertia member, and retaining the spring drive essentially as a self-centering device to retain accurate camshaft timing.

The primary benefit expected, judging by the name given to the new damper, was the reduction of fixed inertia at the nose of the crankshaft, which increased the 3rd order critical speed of the crankshaft, a major step forward to higher safe engine speeds. It was also an improvement in two other aspects; the reduction of complexity, and increased slipping inertia mass.

The final configuration of the Royce damper is shown in Fig 11, the Mk VI Bentley (or Silver Wraith) engine, in production from 1947 to 1959, in its final "Bench"-type simplification, which was introduced after about a year of production.

The "Bench" appellation indicated that the complete damper could be assembled on the bench and then fitted as a unit to the engine, a labor-saving step from the previous design which required the damper components to be assembled on the crankshaft.

9. Current Status

The final version of the Royce damper remains unique in its field, and was applied to a total of over 13,000 post-war Rolls-Royce and Bentley motor cars, most of which are still in existence, and likely to remain so.

However, the machinery which produced the original woven canvas friction discs is no longer in production, and a substitute, either for the complete damper, or for the friction discs, seems to be needed, in order to retain the original engine

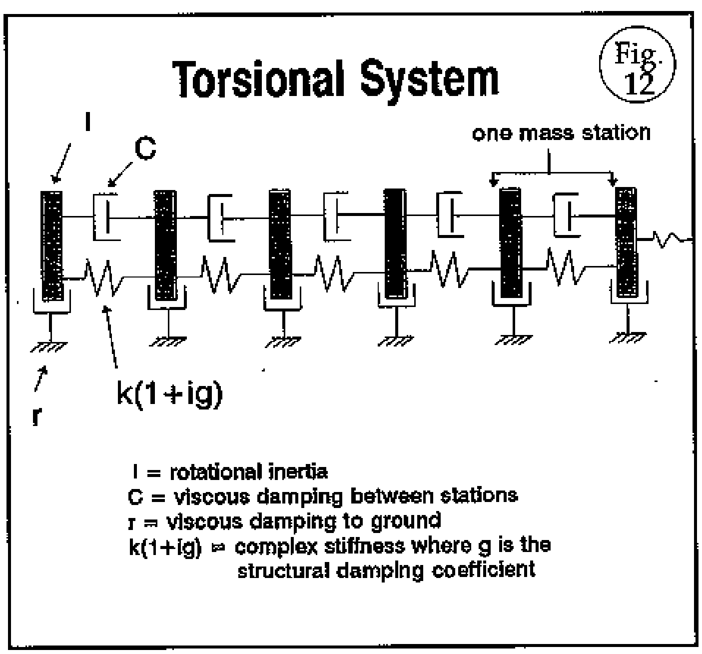


Fig. 13

A

Organic Lining — Specification HO101

HO101 is a non asbestos friction material. Structural stability and remarkable durability are achieved by incorporating Kevlar fibers bound in heat resistant resins and combined with friction stabilizers.

It can be modified to meet a great number of applications. HOERBIGER has developed a proprietary grooving pattern for structural stability and high cooling capacity. Other grooving patterns are also available upon request.

Color: Avocado

This lining was specifically designed for powershift applications. It possesses a high tolerance of poor lubrication and rough mating surfaces while retaining temperature absorption and exceptional durability. Due to the low ratio of static to dynamic friction it is most suitable for heavy duty transmissions where friction stability in prolonged use is essential.

Likewise, it is proven in light brakes that require higher frictional properties and extended life.

Thickness	0.35	mm min
Outside Diameter (OD) of friction material	450	mm max

Preferred and Recommended Dimensions

Thickness	0.70	mm
Outside Diameter (OD) of friction material	360	mm

The following friction coefficients can be obtained, depending upon load conditions and oil flow:

Static	0.13 - 0.16
Dynamic	0.11 - 0.14

Max. Pressure :	2.5	N/mm²
Recommended Pressure :	1.0 - 1.5	N/mm²
Max. Power :	6	W/mm²

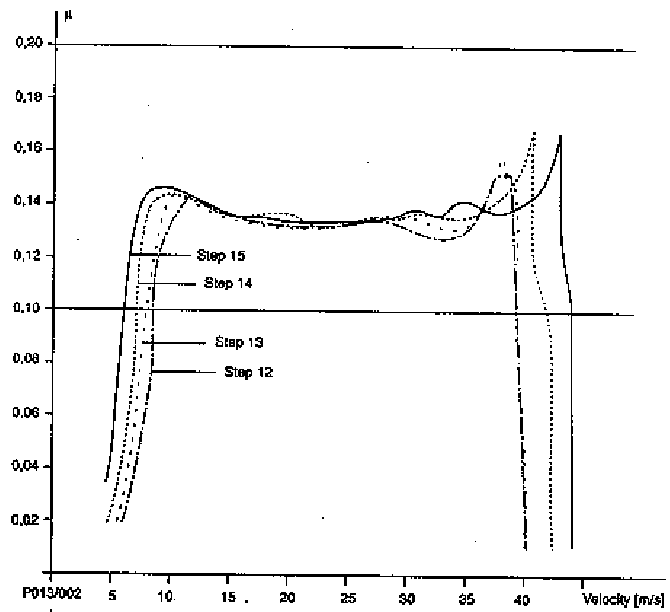
The above values can be increased in case lower frequency of engagements, heavier counter plates, and improved oil flow.

B

Organic Lining — Specification HO101

Step Test: Step 12 to 15

Test conditions		Step			
		12	13	14	15
Unit Energy q_A	J/mm²	0,45	0,49	0,54	0,60
Pressure P_R	N/mm²	0,90	0,80	0,70	0,60
RPM (max.) n	min⁻¹	3450	3600	3800	4000
Frequency of Engagements S_h	h⁻¹	290	285	275	265
Number of Engagements Z_s		5800	5700	5500	5300
Oil Volume $V_{\ddot{o}l}$	l/m²s	0,70			
Friction Surface Areas Z_R		6			
Friction Facing Dimensions		D_A = 226 mm, D_i = 186 mm			



smoothness and mechanical integrity, and to improve the service life of the original damper, which corrodes due to the retention of condensed water by the cotton fibres, especially when cars are infrequently used, and not driven hard.

The last known modification to this damper was made on the last series of 6-cylinder Crewe cars, and noted as "Ferodo washers and drain slots in crankshaft damper." As far as can be ascertained, the change to the washers was only a supplier change, and not a material change.

No details are available to the writer concerning the drain slots, but presumably these were an attempt to cure the most troublesome aspect of the Royce damper, corrosion of the steel plates due to water retention by the cotton washers. In the writers opinion, this approach is problematical, as the only oil supply to the damper is leakage from the front of the crank pinion bearing.

At high engine speeds, when the damper is most active, the internal oil pressure is at least 20 psi due to centrifugal action. If the drain slots are too large, the oil will be centrifuged out faster than the bearing leakage can replenish it. If the drain slots are made too small, they are likely to clog up and be useless. The danger of running the damper dry is significant, since both the performance and the wear rate of the unit could be drastically affected. The writer would appreciate any information on the size and location of these drain slots if any reader has access to this information.

Technical Appendix

1. Damper performance analysis

An analysis has been made of the damping characteristics, both as a matter of record, and as a preliminary to reviewing the effects of adapting a modern damper to these post-war engines, such as the B61 type external rubber damper or modification of a viscous damper.

A proprietary computer analysis program has been used to examine in detail the vibratory behaviour of the Mk VI engine with the damper, the results of which appear in this Appendix. The program requires a mass-elastic model of the engine with each mass station described mathematically by five quantities, Fig 12.

The Holzer method of tabulation is used to determine the model's natural frequencies and mode shapes, while the forced, damped response section of the program uses a transfer matrix solution, using a single-cylinder forcing function based on the slider crank geometry of the engine, together with its reciprocating weight, cylinder pressure and friction torque values.

The program then calculates the vibratory deflection, torque, velocity and acceleration for each element, and also the relative values between elements, by vector summation of the response to the orders 0.5 to 9th, for each half order increment. The response to each individual order can also be obtained.

2. Issues in modelling the Royce damper

Since the program was designed to deal with only the two modern damper types (viscous and rubber), the principal issue is the accurate representation of the complex damping of the Royce damper, with its combination of coulomb and viscous damping, which unfortunately has a non-linear characteristic, and cannot be handled easily in a linear program.

One of the features of the program is the calculation of the heat generated by a viscous damper. This may then be compared to the heat generated by a Royce damper based on vibratory rpm and slipping torque.

By adjusting the viscous damping coefficient for the Royce damper until a viscous heat value is predicted which matches the calculated friction heat at the same vibratory condition, a very close approximation to the effect of the Royce damper can be obtained for any given speed/load combination.

After reviewing the R-R instruction book and performing some experiments on an actual damper, a value of 22 lb. ft break-away torque was established, and an assumption was made of a slipping/static mu ratio of 85%, (based on modern organic, non-asbestos friction materials, e.g.; Hoerbiger HO101-D, Figs 13A & B).

This gave equal heat values with a viscous damping coefficient of 125 (lbf-in-sec)/radian, which was used in the analysis. It should be noted that the model is not especially sensitive to changes in this value.

The performance of this damper is a revelation, <u>as it reduces the crank gear vibratory motion to an extent not achievable by any modern unit</u>, and stands as a monument to Royce's intuitive genius and patient quest for perfection.

3. The importance of cylinders

The Campbell diagram (Fig. 1), shows that the major criticals of 4-, 6-, and 8-cylinder engines occur at quite different engine speeds. (Early B series)

The 4-cylinder engine doesn't encounter its major critical until 9500 rpm, well beyond the normal speed range, which makes a torsional damper unnecessary on almost all engines of this type.

The 8-cylinder critical occurs at 3100 rpm, therefore avoiding very high inertia forces, which increase as the square of the rpm. This low speed major critical is easily dealt with by a simple damper, and the engine then runs up through the speed range untroubled by severe torsionals. This accounts for the popularity of straight-8 engines for performance and racing between the wars.

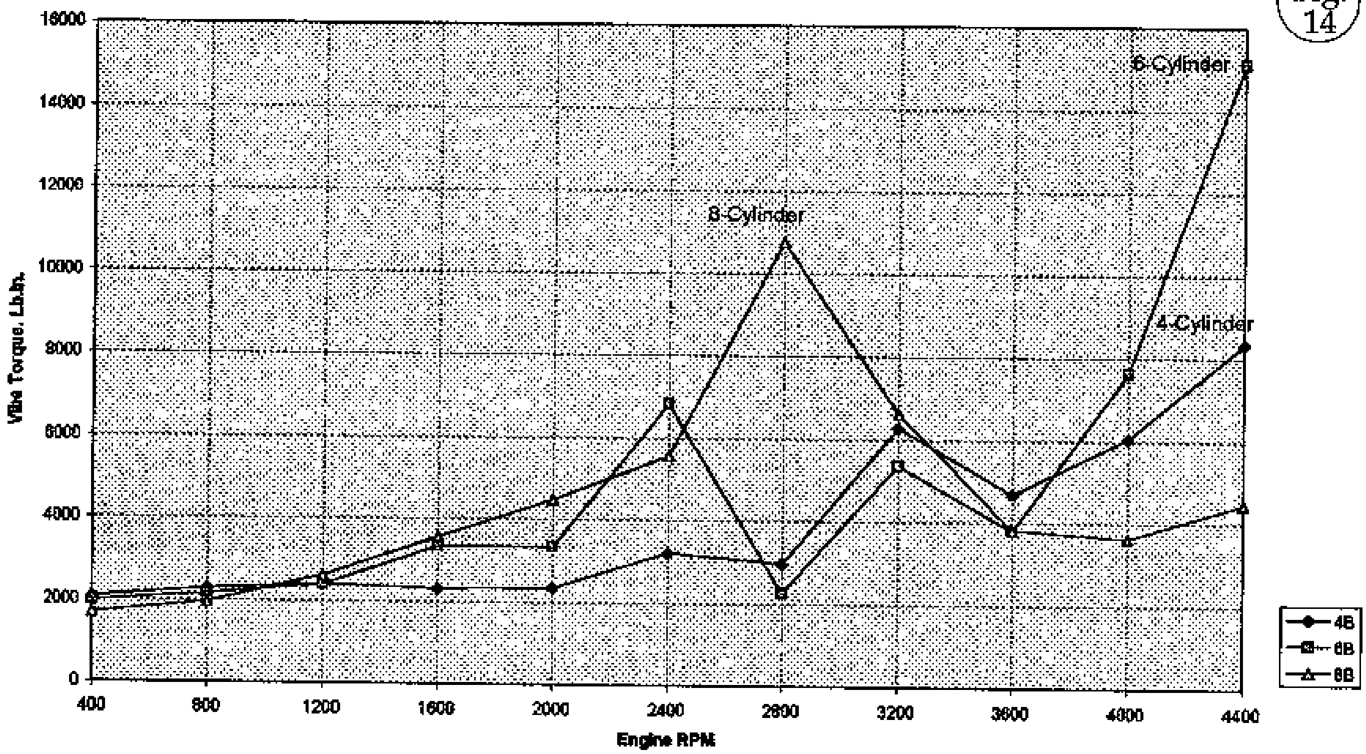
The 6-cylinder critical speed of 5000 rpm is just above the maximum operating speed. In fact it defines the maximum operating speed, as the inertia forces at high piston speeds make the 3rd order vibration levels so severe that normal damping can't reduce them to a tolerable level. This makes high safe speeds very difficult to attain in the traditional long-stroke, in-line 6. (Fig. 14 shows the vibratory torques encountered at full throttle through the speed range without any form of damping for each type of engine. The peaks are the result of the combination of the major and several minor criticals).

It is interesting in this regard to look at the successful engines in the Indy 500, a rigorous test of engine design. Since WWII this arena has been

RPM	400	800	1200	1600	2000	2400	2800	3200	3600	4000	4400
4B	2063	2275	2400	2303	2348	3194	2949	6283	4682	6051	8336
6B	1962	2148	2370	3319	3328	6840	2253	5354	3798	7621	15124
8B	1686	1953	2611	3547	4476	5566	10743	6618	3822	3614	4443

4B,6B,8B, Crank Vibratory Torque, without Damper

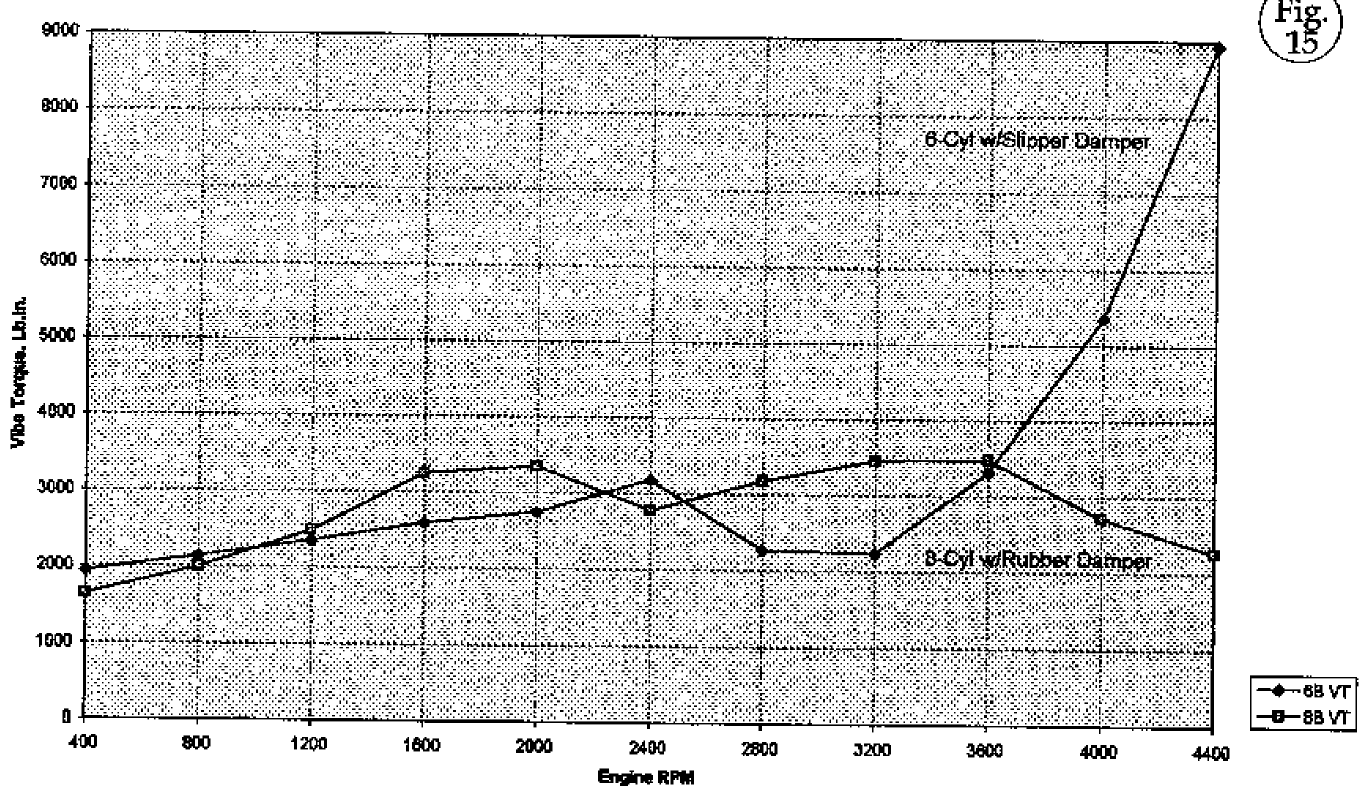
Fig. 14



RPM	400	800	1200	1600	2000	2400	2800	3200	3600	4000	4400
6B VT	1931	2134	2344	2589	2747	3186	2270	2239	3320	5346	8945
8B VT	1638	1986	2482	3247	3348	2784	3177	3462	3481	2714	2263

6B,8B, Crank Vibratory Torques with Damper

Fig. 15



RPM	400	800	1200	1600	2000	2400	2800	3200	3600	4000	4400
6B Hub	188	219	481	1596	2726	4545	4319	4465	4253	5747	10843
6B Ring	196	179	190	497	720	1115	980	965	941	1480	2612
8B Hub	169	211	617	1657	2752	2676	3531	4547	5216	4476	3727

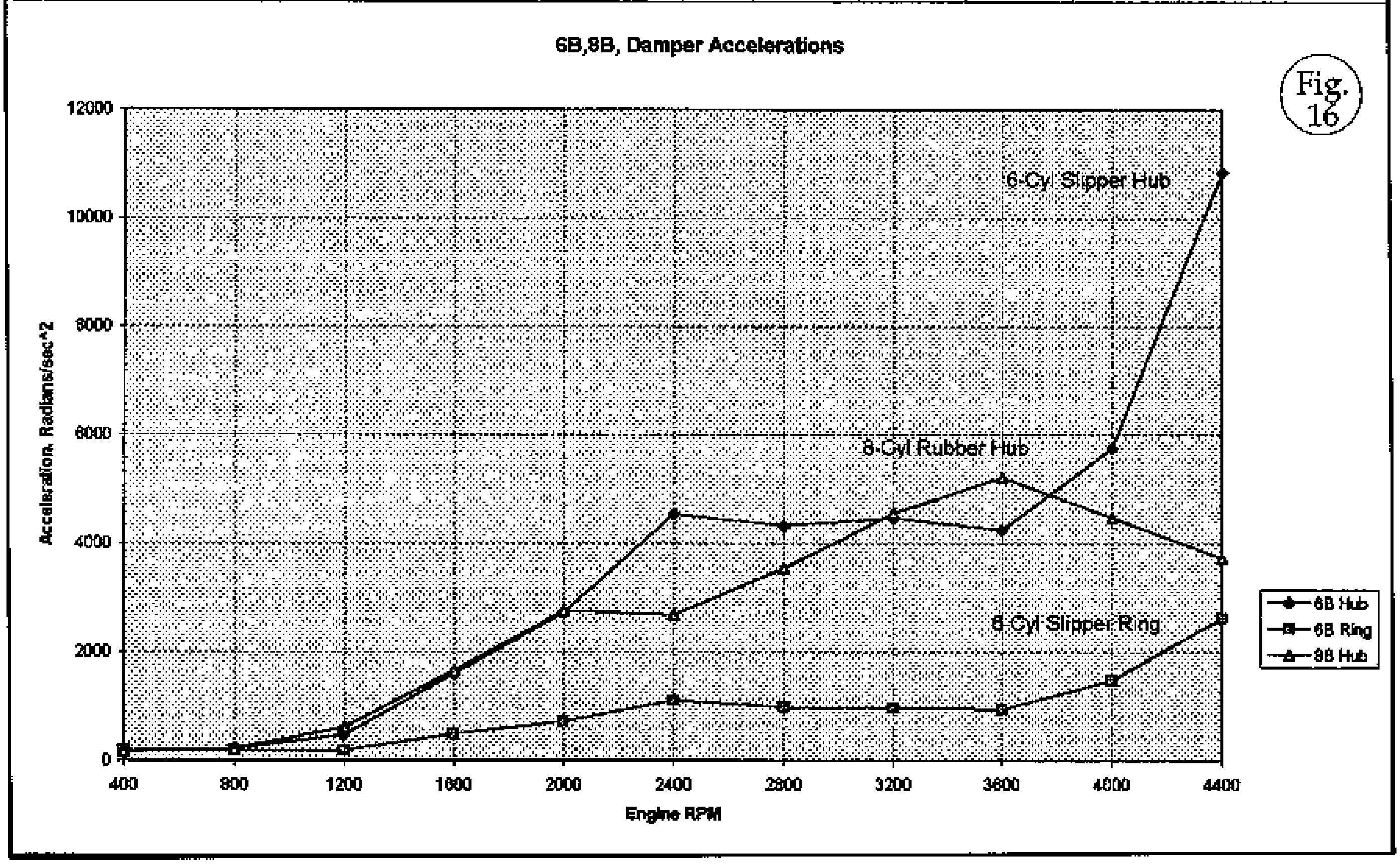


Fig. 16

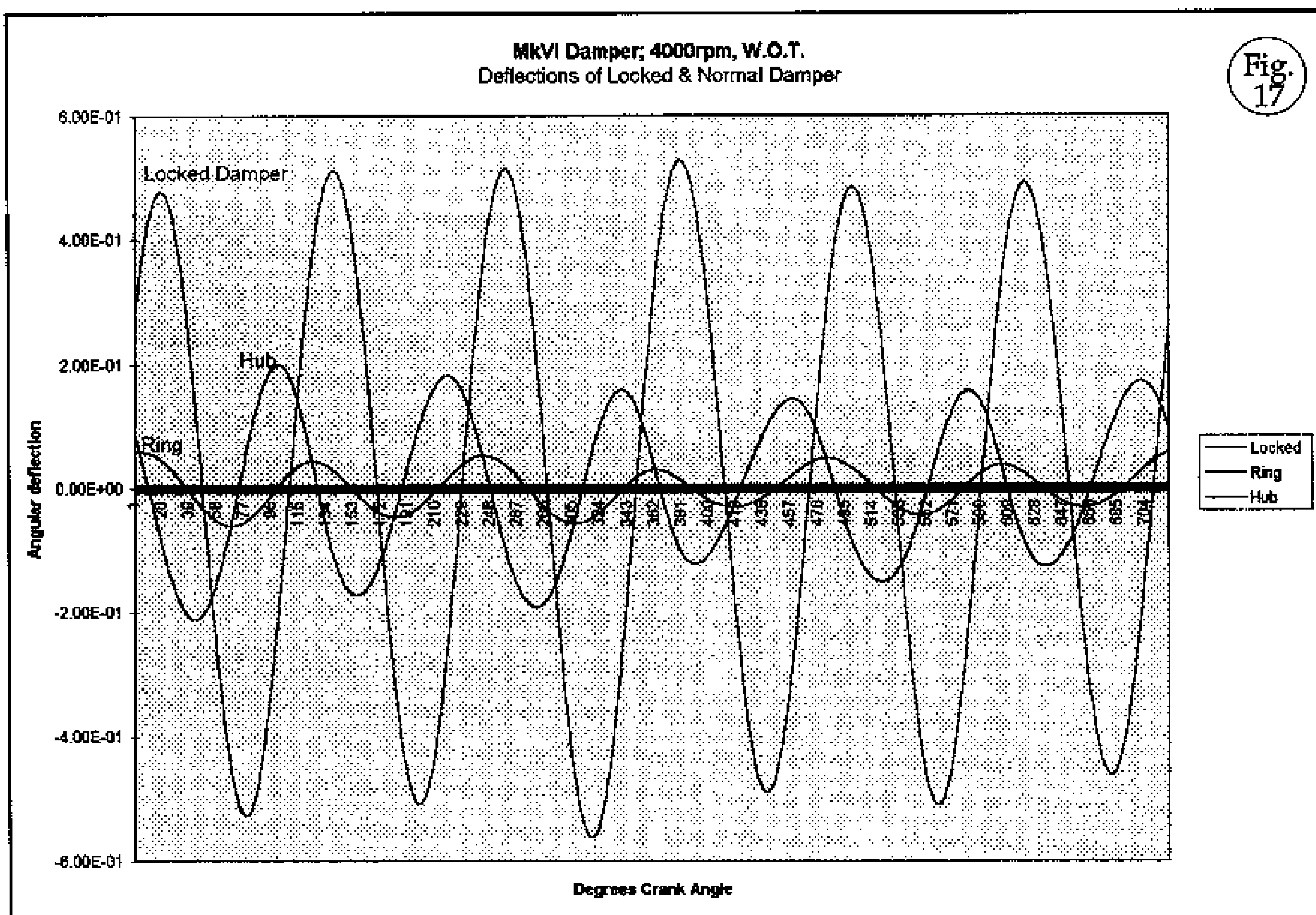


Fig. 17

dominated first by the Meyer-Drake-Offenhauser, a 4-cylinder engine, and then by V-8's.

The only 6-cylinder unit to make a showing has been the Buick V-6, 3.8 liter push rod engine. This elegant design has taken the 6 to new heights of silky performance, with its short, stiff crankshaft, large bore and short stroke. It seems surprising in hindsight that this layout took so long to achieve acceptance.

It is tempting to speculate on what Royce might have done, if, after his short stroke V-8, (one of the earliest of the type, designed in 1905), he had made the leap to a 60 deg. V-6!

4. The final Royce slipper damper

Fig. 15 shows the B6 and B8 with their standard production dampers, the Royce damper in the case of the B6, and a "tuned" external rubber and friction damper in the case of the B8. A tuned damper is one which responds to a narrow range of frequency, in this case the 3000 rpm range, and is only active as the engine passes through this speed range. Its effect is to "split" the undamped torsional peak into two, lower peaks, either side of the original critical speed.

3. The Royce damper is an "untuned" damper (as is the modern viscous damper), which is responsive to angular velocity, not frequency. It is quite effective in reducing the vibratory torques in the crankshaft, bringing them down from 15,000 lb. in. to 9,000 lb. in. at 4,400 rpm, but its real magic lies in the reduction of vibration at the crank gear, when the torsional accelerations of the hub are 10,800 rads/sec^2, but are only 2,600 rads/sec^2 at the inertia ring, which of course, is attached to the gear, Fig.16.

5. "Sticking" dampers and Tufnol

There is an opinion among some vintage Rolls-Royce owners that the Royce damper is subject to "sludging" and even "sticking". There is also an issue of finding an effective replacement for the original cotton duck washers, which form the critical friction element. Without wishing to take any sides in these controversies, there are some analytical data which can shed a light on these areas of concern.

Fig. 17 shows the actual motion of the damper hub and ring at 4,000 rpm, and compares this to the motion which would occur if the damper mechanism were to become completely "locked", that is, no slip at all. A couple of insights arise from this graph. First, how effective the Royce damper is at reducing the critical gear motion, and second, how violent the motion would be, if in fact the damper were to "stick" tight. If the damper were to actually lock solid, the vibratory torque at the interface between the cotton and metal could easily rise to as much as 400 lb. ft. torque at 4000 rpm, compared to the normal slipping torque value of about 20 lb. ft.

It does not seem possible that cotton washers could survive intact under that kind of loading.

It is also instructive to observe the extremely small angular deflections that occur in a working damper, the maximum relative slip between the hub and ring being only 2 tenths of a degree, zero-to-peak displacement. At the outer diameter of the hub disc, the double amplitude motion, peak-to-peak, is therefore only .023" which is considerably less than the thickness of the radial cotton threads in the washers. And this is at a frequency of around 250 Hz, or the pitch of the B note adjacent to middle C on the piano! Such a vibration might be heard, but it is not likely to be felt from the driver's seat.

It is clear that quite deep radial markings are present on the metal faces of a worn damper, and one could surmise that there are two mechanisms at work here, rust and fretting. The propensity of the cotton duck to absorb and retain water seems self evident. Water is always present in the oil of automobile engines to some degree, and the damper is an effective centrifuge, separating the water from the oil, and keeping at least a portion of it within the damper at all times.

Also, the damper is NOT constantly active. Even at full throttle, the damper will not slip until a speed of around 1,600 to 1,800 rpm has been reached. The vibratory torque below these speeds is less than the static friction.

If the engine is decelerated from its maximum speed of 4,000 rpm, closed throttle, the vibratory torques will be insufficient below about 3,600 rpm to slip the damper. Given the tender, loving care that most older R-R cars receive, it is possible that dampers spend much of their life without slipping. Rust will build up at the cotton thread interface, which will then be fretted away when the damper does slip.

Glazing of burnt carbonaceous material on the face of the cotton may make the damper relatively weak, in that its dynamic mu may become much less than its static mu. Which brings us to the question of the replacement materials, Tufnol in the UK and Micarta in the US.

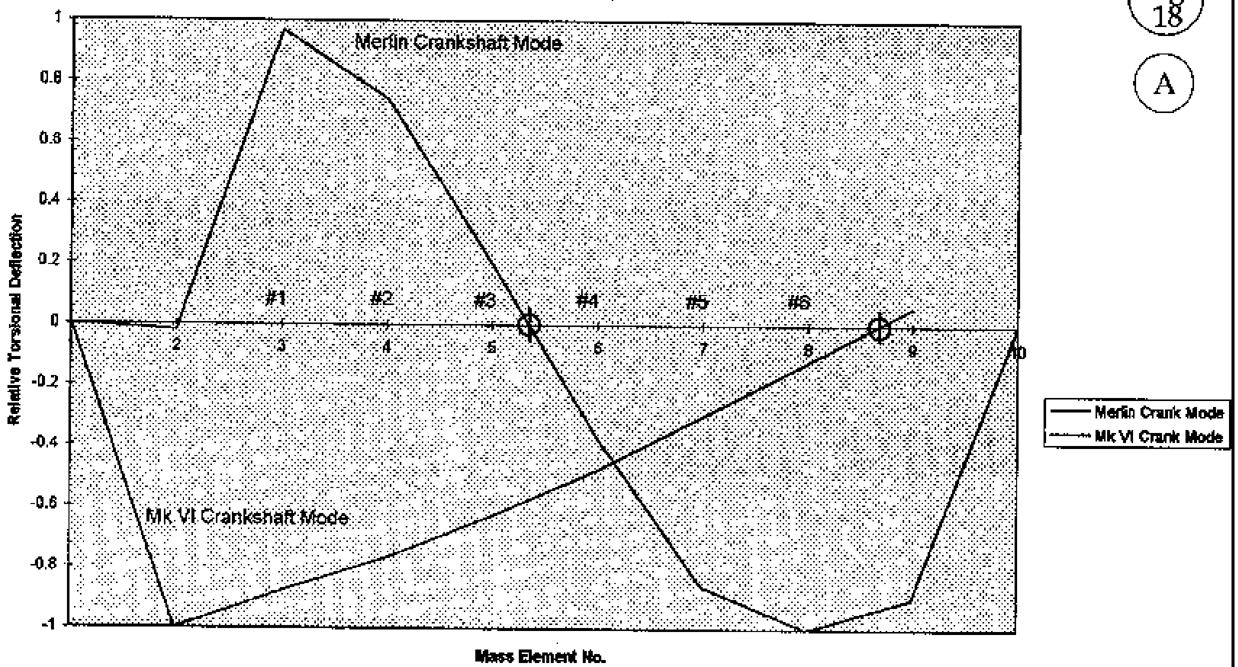
While no laboratory data has been found for these specific materials, a similar type of smooth plastic against steel has been tested to give a low ratio of dynamic mu. Typically only about 33% of the static value in the lubricated condition. Much better materials are available today, developed from non-asbestos, organic mixtures of cellulose,

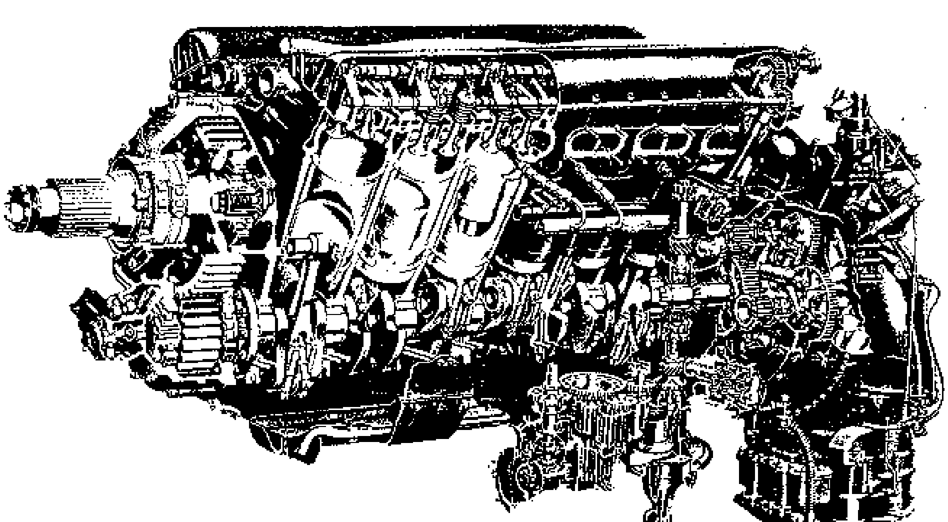
Mass	1	2	3	4	5	6	7	8	9	10
Merlin Crank Mode	0.0001	-0.02	0.97	0.74	0.2	-0.37	-0.86	-1	-0.9	0.006
Mk VI Crank Mode	0.0068	-1	-0.88	-0.77	-0.63	-0.48	-0.3	-0.12	0.06	
Merlin	S'charger	Gears	Pin 1	Pin 2	Pin 3	Pin 4	Pin 5	Pin 6	Red. Gear	Propellor
Mk VI	Damp ring	Hub	Pin 1	Pin 2	Pin 3	Pin 4	Pin 5	Pin 6	FW/Clutch	

Merlin & Mk VI elastic curves

Fig. 18

A



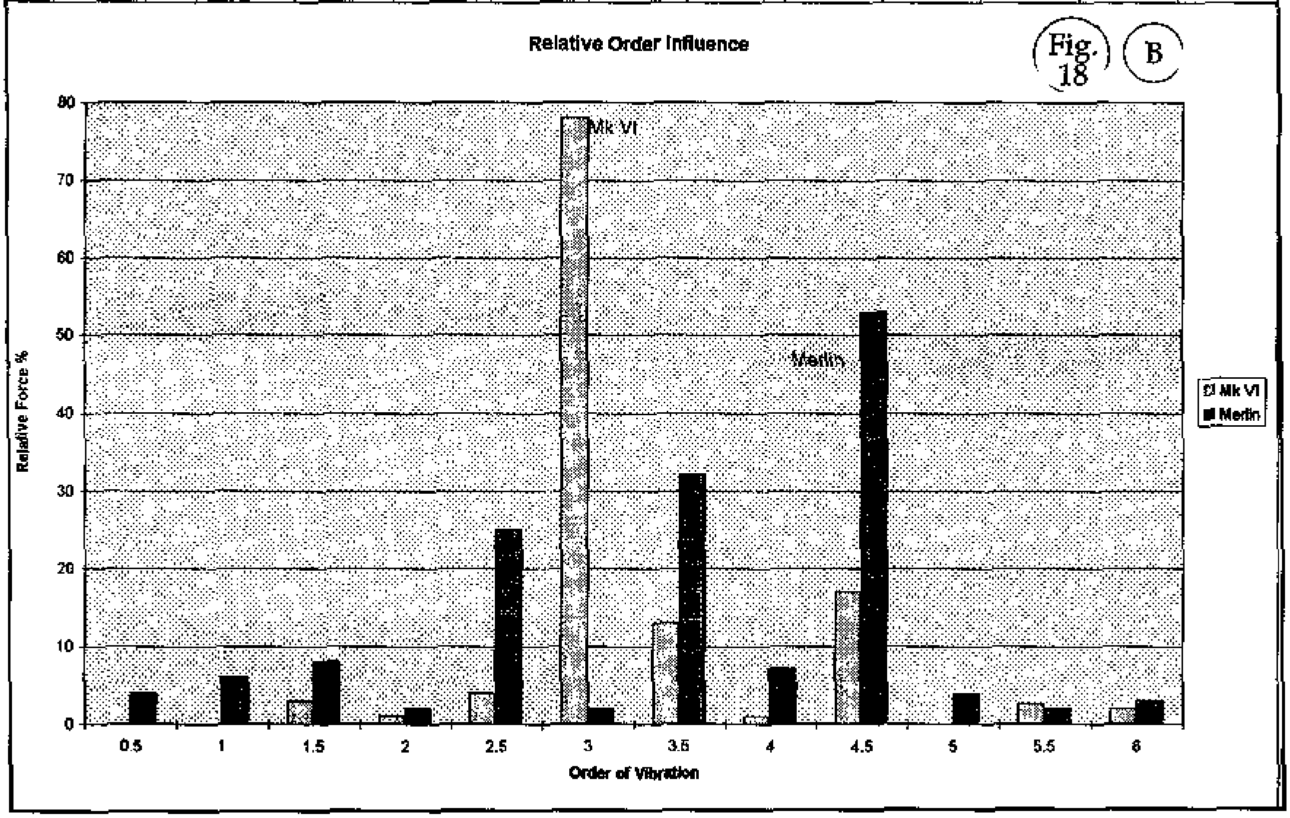


THE MAGNIFICENT MERLIN

blended with metal particles and synthetic fibres, for use in clutches and brakes in mining and construction equipment power-shift transmissions. (Ref. Figs 13A & B. This material absorbs oil, is available in sheets of 1.2 mm thickness, and is easily cut into circles with a sharp pair of dividers. This standard thickness gave a slip torque of 22 lb. ft. on a rebuilt MK. VI damper, and has a dynamic mu value of 85% of static.

The first test unit with the HO101-D material has been installed in Neal Kirkham's 3.5 liter Bentley, B35BL, which had suffered a broken crank from excessive torsional stress in the rear main area. When this engine was stripped, it was found that the damper was "loose" due to wear and partial destruction of the cotton washers, with no friction damping. Several springs in the spring drive were also broken, as they would have been

Order No.	0.5	1	1.5	2	2.5	3	3.5	4	4.5	5	5.5	6
Mk VI	0	0	3	1	4	78	13	1	17	0	2.6	2
Merlin	4	6	8	2	25	2	32	7	53	4	2	3



severely "hammered" without damper friction to control the motion of the inertia wheel.

Since neglect of the damper can lead to crankshaft failure, owners of older cars should consider testing the condition of the damper. In the case of the "low inertia" and post-war 6-cylinder dampers, this can be done by blocking the flywheel ring gear to prevent rotation, and turning the inertia wheel by the cranking dog nut with a torque wrench, when the resistance of the friction (and the spring drive) can be felt. The dog should be turned back & forth a few times and an average torque required to "break-away" the movement recorded. If this is in the range of 18 to 22 lb.ft., then the damper can be considered to be effective, provided that the original cotton washers are still installed, or substitute friction material with proven characteristics.

With the banning of asbestos as a viable friction material, a major research effort was mounted by the major suppliers of transmission clutch linings, and the presently available materials from these specialist suppliers have been carefully blended to ensure excellent performance, together with heat and wear resistance characteristics.

It seems clear that the use of a viscous or rubber damper on the 6-cylinder unit will result in higher gear accelerations, as only the Royce principle of driving the gear from the slipping member gives the lowest acceleration. Whether gear noise would be audible is speculative, but increased gear wear is almost certain to result. The crankshaft stress should be comparable if the substitute damper has a similar heat capacity, and if rubber, is tuned correctly.

The following table shows the relative values of the Royce and a viscous damper, compared with a locked damper. The gear acceleration is in rads/sec^2, and the crank vibratory torque is in lb. in. at No. 7 main journal:

Engine	Locked		Viscous		Royce		
RPM	Accel	V.T.	Accel	V.T.	Accel	V.T.	Order
2400	10759	6840	4545	3186	1115	3186	6th
3200	9091	5354	4465	2239	965	2239	4.5
4400	18229	15124	10843	8945	2612	8945	3rd

The underlined values demonstrate the dramatic effect of the Royce damper in reducing gear acceleration at the 6th order period, where the noise would be most easily heard. Note that the 4400 rpm point is not the peak of the 3rd order critical, but is as near as could typically be reached in top gear, full throttle.

6. The magic of the Merlin

To avoid the very damaging forces of the 3rd and 6th order periods in V-12 aero engines, the center node technique was used to some degree in all Royce's designs. This method has been used by a few other engine manufacturers to minimize the primary firing order criticals, notably Bugatti in his straight-8 car engines, and MTU in their V-series 396 diesel engines.

The trick is to minimize the attached inertia at each end of the crank, which Royce accomplished by quill or torsion bar driving shafts, driven by splines or gear teeth from each end of the shaft, (Ref. Fig 8). The effect of this is shown in Figs 18A and B, which show the elastic curve of the Merlin compared to the Mk VI engine, and also the effect of this shift in node point from the end of the crank to the center of the crank, on the relative exciting force from the various orders at rated speed.

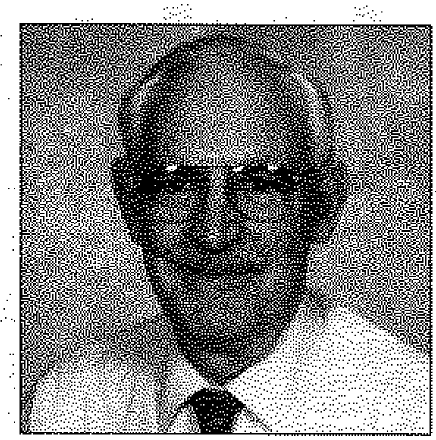
It will be seen that the Mk VI has almost 80% of its excitation from the 3rd order, while the Merlin has almost no excitation from the 3rd, but a mix of forces from the 2.5, 3.5 and 4.5 orders.

Modern engine designers might well take note of this technique, which could be adopted with significant benefit to many commercial designs.

The ability to take very high powers from a lightweight crankshaft without recourse to any form of vibration damper is a notable achievement, one of the most significant, if least appreciated, of the many achievements of Royce and his Company.

John G Crofts.

ABOUT THE AUTHOR



John G. Crofts
Grad. I. Prod. E. (UK). Mem. A.S.M.E. (US)

John G. Crofts joined R-R as an engineering apprentice in 1944. He was following in his father's footsteps; his dad, John H. Crofts, was hired as an apprentice in 1910 by Eric Platford ("P").

Concurrently with his apprenticeship he pursued an academic engineering degree at Derby Technical College and was graduated in the Institute of Production Engineers in 1949. His engineering apprenticeship at R-R was completed that same year and had consisted of four years in the aero engine division and finished with one year in the R-R car division. He continued as a junior design engineer with the car division for four years more.

Eleven years were then spent in Southern Rhodesia (now Zimbabwe) as service engineer and engineering manager with local distributors for R-R and MWM diesel engines. In 1963 he emigrated to the USA to become a design and development engineer with Chicago Pneumatic Tool Co. - engine division. By 1965 he was started on a thirty year career with Cummins Engine Co.; the final fourteen years were spent as the director of application engineering.

Since 1995 he has operated a consulting engineering practice (Midlands Services, Inc.) in Columbus, Indiana providing advice on design and analysis for internal combustion engine shafting, mounting, and performance prediction, including vibration analysis. BH

COMPARISON OF FRICTION MATERIALS IN R-R DAMPER.

Further to the data in "Hose Clamp, Technical Supplement Vol 27-TI, 1998", concerning suitable friction material for substitution in Rolls-Royce and Bentley crankshaft friction dampers details are offered:

1 . A test rig was set up to simulate certain operating conditions in a typical friction damper (early Mk VI type), to c range of alternative friction disc materials in regard to torque at break-away and under running conditions; the effe lubricant types; and the loading pressure required to produce a constant level of running torque.

*** A 14" x 40" center lathe was used, with the rear floating inertia member (see Fig. 11 of reference paper) mou the lathe chuck, and the center member (normally keyed to the crankshaft), forced against the friction disc on test by tailstock while restrained from rotating by a torque wrench of the bending beam type, resting against a support. This because it is a simple device, providing consistent results from test to test, and is very sensitive to vibration from the ' behavior of certain materials.**

*** Between the tailstock and the torque wrench head a small hydraulic cylinder was centered, with a pressure g attached, calibrated from 0 - 1500 p.s.i. This indicated the force required to produce a consistent torque level betwee rotating and fixed elements.**

*** Three types of oil were used for comparison purposes, a) a light hydraulic oil; b) a commonly used automoti oil of 5W-30 SAE grade, with an API duty rating of SJ/SH, and c) a gasoline/diesel engine oil of SAE 15W-40 grade, v API duty rating of CG4/SH with extended life additives.**

2. The friction materials used were 1) original R-R cotton disc, used but in excellent condition. 2) Smooth plastic material (Acco). 3) Soft cardboard. 4) Heavy grade cartridge paper). 5) Aluminum sheet. 6) Raybestos clutch lining. 7) HO 21 material. 8) HO 101 material. 9) HO 121 material. 10) Tufnol replacement disc for a P.11. 11) Tufnol replacement disc for a 31/2 Liter Derby Bentley. (No's 7; 8; & 9 were supplied by Hoerbiger GMBH, and are commercial heavy duty friction materials. No's 10 & 11 are thought to be "Tufnol", a smooth plastic laminated with fabric', from independent R-R/B parts suppliers).

*** The procedure followed was to clean up the two metal surfaces of the old Mk VI damper by polishing with a grinder using fine emery discs. No machining was used, as the original surfaces were flat, but marked with rust groov the original cotton radial strands. These were mostly polished out, but slight witness marks were left, as representing reasonable job of "cleaning up" the components by an owner lacking machine shop facilities.**

continued:

*** The friction material was liberally coated on both sides with fresh lubricant, and the system run at 60 rpm (t speed of the lathe headstock) for several minutes until a temperature of about 120 F on the stationary pressure plate v attained. Data logging then began, with cleaned plates and fresh lubricant applied for each new material sample. This procedure maintained the metal temperature with a preliminary run of 1 minute for each sample.**

*** A rotating torque setting of 20 lb.ft. was maintained by adjusting the force applied by the tailstock, and after minute of running the rotation was stopped, and the chuck barred over with a tommy bar to read the break-away tor The rig was again set in motion to ascertain that the running torque level had not changed.**

3. Results:

The objective was to match as closely as possible the performance of the original R-R Cotton, material 1).

*** Five of the materials tested were disqualified by reason of vibration while running, and/or requiring excessiv (beyond the capacity of the pressure gage) to obtain the standard test torque of 20 lb.ft. These were No's: 2) Smooth p Soft cardboard; 4) Cartridge paper; 5) Aluminum sheet-, and 6) Raybestos clutch lining material.**

*** The most consistent results (many repeat tests were conducted), were obtained with the gasoline/diesel engin SAE grade 15W-40. This is most likely due to the additive package in this lubricant, and also confirms Hoerbigers co that the nature of the oil does affect the results obtained.**

*** The remaining material performances are tabulated below:**

Material	Breaking torque	Running torque	Torque Ratio	Pressure*
1) Cotton	18 lb.ft.	20 lb.ft.	1.11	10.24 p.s.i.
	Very smooth break-away			
7) HO 21	18 lb.ft.	20 lb.ft.	1.11	15.33 p.s.i.
	Very smooth break-away			
8) HO 101	19 lb.ft.	20 lb.ft.	1.05	14.75 p.s.i.
	Smooth break-away			
9) HO 121	19 lb.ft.	20 lb.ft.	1.05	13.59 p.s.i.
	Less smooth break-away, occasional very slight vibration.			
10) P.11 Tuf.	27 lb.ft.	8 lb.ft.	0.29	15.90 p.s.i.
	Harsh break-away, smooth when running.			
11) D.B.Tuf.	22 lb.ft.	10 lb.ft.	0.45	19.75 p.s.i.
	Harsh break-away, smooth when running.			

***Note: The pressure listed is the unit pressure per square inch of the friction material discs, 1-9 = 6.5" x 4.5"; 10 = 6.38" x 4.13"; 11 = 5.88" x 4.0".**

Material thickness = 1,10, 11) 0.062"; 7) 0.050"; 8) 0.048"; 9) 0.030".

The Hoerbiger materials can be obtained in several thicknesses.

VIBRATION DAMPER MATERIAL ANALYSIS

Summary:

This is the second addendum to the paper "Rolls-Royce & Torsional Vibrations", Volume 27-Tl of "H
Clamp", the news letter of the Pacific North-West Region, RROC, published in 1998.

It complements the first addendum, which was a listing of the physical performance attributes of a series of friction m
with the purpose of ranking them against the original cotton duck material supplied by Rolls-Royce for many years for their
crankshaft dampers used on 6-cylinder car engines from 1912 to 1956. Two additional materials have been tested in the same
the first batch of 10 alternative materials, compared to an original R-R cotton duck disc.

Comparisons:

The two additional materials are compared to the original and the two best alternatives previously teste
the same test rig, a 14 center lathe, and the same damper components, with the most consistent lubricant found on the first se
tests, Shell Rotella'T' 15W-40 SAE grade.'

Material	Breaking Torque	Running Torque	Torque Ratio	Pressure
1) Cotton	18 lb.ft.	20 lb.ft.	1.11	10.24 p.s.i.
Very smooth break-away				
7) HO 21	18 lb.ft.	20 lb.ft.	1.11	15.33 p.s.i.
Very smooth break-away				
8) HO 101	19 lb.ft.	20 lb.ft.	1.05	14.75 p.s.i.
Smooth break-away				
12) Fiennes	18 lb.ft.	20 lb.ft.	1.11	17.10 p.s.i.
Very smooth break-away				
13) Ferodo	24 lb.ft.	20 lb.ft.	0.83	19.2 p.s.i.
Violent vibrations under all conditions.				

The sample 12) was furnished by Fiennes Engineering of Clanfield Mill, G.B., and gave a very similar performance to the tw
samples, the original cotton, and H021 material from Hoerbiger.

*The difference between these two alterative materials is insignificant performance-wise, but it would help to underst
behavior of the material at high temperatures. The H021 material is specifically developed for use in spring dampers for loc
clutches in automatic transmissions, and has a very high temperature & wear resistance, consistent with this type of usage.*

(continued)

The porous Polypropylene material from Fiennes may not have been bench-tested for high-temperature wear, or fric co-efficient stability characteristics, bearing in mind that it needs to be able to operate successfully, with a long life, under th following conditions-

Between steel rubbing surfaces of 200 to 250F, with an additional heat input rate of 2.5 watts1square inch of frictio when running at 4, 000 engine rpm. These are the conditions in an engine at high speed during summer.

The Ferodo material was taken from an R-R servo lining replacement, based on information from James Pate RROC, who has anecdotal . information from members of the RREC, about the mysterious Modification note contai Bulletin BB.234, for the Z series of 6-cylinder cars produced at Crewe.

These were the last of the 6-cylinder engines produced by Rolls-Royce, and the note says "Ferodo washers and slots in the crankshaft vibration damper", from chassis B-212-ZY onwards.

There is conflicting information regarding this modification, as Hermann Albers of Albers, Rolls-Royce in Zi told the writer that he had a sample of a cotton duck washer with the word 'Ferodo' clearly stamped on it. However t anecdotal information states that the material first used for this modification was the same material as is used for the linings, and the slots were cut in the friction face of the material to allow of oil ingress across the face, as the material highly absorbent like the cotton, and would tend to wipe dry without the oil slots.

The test of this material was done using a servo liner, which has rivet holes around the face, thus allowing oil t distributed across the working area, but the vibratory characteristics were very severe, smooth running could only b by very gentle application of load, and severe vibration was immediately precipitated by any load above about 10 lb.f torque, and also when testing break-away torque. The loading required to reach an acceptable torque was also highe other satisfactory materials tested, and would probably have required spring modifications in the damper.

It may be that the original Ferodo material was found to be unsatisfactory in service or production. and was r Ferodo by a more suitable material?

ROYCE SLIPPER DAMPER MATERIAL TESTS:

This is the 3rd addendum to "R-R & Torsional Vibrations", Vol. 27-T1 of the Hose Clamp magazine.

Two additional friction materials have been tested in an effort to give R-R owners and repairers a satisfactory substitute with known properties, for the original R-R cotton discs.

The two new materials were tested in the same way as the previous units, including the original R-R cotton disc, which is listed for comparison.

Material	Breaking Torque	Running Torque	Torque Ratio	Pressure
R-R Cotton	18 lb.ft.	20 lb.ft.	1.11	10.24 p.s.i.
	Very smooth break-away			
14) N-411	18 lb.ft.	20 lb.ft.	1.11	12.29 p.s.i.
	Very smooth break-away			
15) JDDW	27 lb.ft.	20 lb.ft.	0.74	13.93 p.s.i.
8385	Smooth break away, very slight vibration when running.			

Both materials were supplied by S.K.Wellman Co. of Canada, address:
606, Rivermere Road, Unit 10,
Concord, ON L4K2H6
Canada. Contact: Mr. Paul Douglas, Tel: 905-669-1301 Fax: 905-669-4124

The N-411 material is smooth, dark, porous material, normally used as a friction braking disk for heavy-duty truck clutch brakes. This is a very rugged application with marginal, if any lubrication. This material is close to the R-R cotton in all respects, requiring perhaps slightly higher spring pressure. It is marked "Texon" and is 1/16" thick, the same as the original cotton.

The JDDW 8385 is a woven and bonded material used for emergency brakes required by law in Europe on farm tractors when used on public highways. This material is not suitable as a replacement due to the lower torque ratio, the higher pressure requirements and the more aggressive action.

A heavy-duty gasoline/diesel engine lubricant meeting SAE CF/SJ requirements, of 15W-40 grade, (Valvoline Premium Blue 2000), was used for this test, (which is a desirable lubricant for classic R-R/Bentley engines), and the friction discs were soaked in the oil for two hours @ 45F before the test, to assure representative performance. Metal temperature during the testing was approx. 125F.

John G. Crofts
Principal engineer