splined connection used must be of sufficient size and strength to withstand the forces which may result from the functional and durability requirements of a given application. Various types of splined constructions such as involute, parallel side, and rolling element splines, as well as square and rectangular bar shaft and other configurations are described in the chapter on *Splines and Axial Motion Devices*.

Shaft Support Bearings

When a driveline arrangement has three or more universal joints, the driveline must be supported at one or more intermediate points with a shaft support bearing in addition to the end supports. These shaft support bearings are generally of special design and must withstand the forces which may result from the functional and durability requirements of a given application. These special design characteristics and the various possible support bearing configurations are described in the chapter on *Shaft Support Bearings*.

Ball Type Universal Joint Attachment Means

Driveshafts with various ball type universal joints, in particular, are generally attached to other drivetrain components with splined, flanged, or disc type connections. These connections are usually designed integral with the housing or outer race member and are shown in the sections describing these various joints and are also discussed in the chapter on *Axle Driveshafts*. The attachment means must be designed to withstand the forces which may result from the functional and durability requirements of a given application.

Fastener Tightening Torques

In Table 1, typical tightening torque values are tabulated for various types and sizes of fasteners and recommended for the positions or locations indicated. Some applications, however, may require other tightening torque ranges in order to meet the installation functional and durability requirements and operating characteristics. In any event, the fastening means should be properly applied and excessive tightening torques should be avoided to prevent fastener failure at high rotational speeds. The bolts, cap screws, and U-bolts used are generally made of medium carbon alloy steel and conform to the SAE grade 8 bolt specifications. It will be noted that the torque values listed for U-bolts are somewhat lower than normally used with the indicated thread sizes. These lower torque values are required to prevent bearing cup distortion, which can result in premature joint bearing failure from the use of the higher torque values or from the application of improper tightening procedures.

TABLE I — TYPICAL RECOMMENDED TIGHTENING TORQUES

Position or Location	Fastener Size	Torque (lb-ft)
Flange Yoke Bolts	5/16-24 U.N.F. Thd.	22 - 26
	3/8-24 U.N.F. Thd.	40 - 48
	7/16-20 U.N.F. Thd.	63 - 75
	1/2-20 U.N.F. Thd.	97 - 116
	5/8-18 U.N.F. Thd.	194 - 232
	11/16-18 N.S. Thd.	262 - 315
	3/4-16 U.N.F. Thd.	341 - 409
	7/8-14 U.N.F. Thd.	543 - 652
End Yoke U-Bolts	5/16-24 N.F. Thd.	14 - 17
	3/8-24 N.F. Thd.	20 - 24
	7/16-20 U.N.F. Thd.	32 - 37
Bearing Retainer Plate Bolts or Cap Screws	5/16-24 N.F. Thd.	17 - 24
	3/8-24 U.N.F. Thd.	32 - 42
	7/16-20 U.N.F. Thd.	50 - 66
Bearing Cap and Strap Bolts or Cap Screws	5/16-24 U.N.F. Thd.	17 - 24
	1/2-20 U.N.F. Thd.	77 - 103
	9/16-18 U.N.F. Thd.	110 - 147
	3/4-16 U.N.F. Thd.	271 - 362
	1-12 U.N.F. Thd.	744 - 844
Wing Bearing Bolts or Cap Screws	5/16-24 U.N.F. Thd.	22 - 26
	3/8-24 U.N.F. Thd.	40 - 48
	7/16-20 U.N.F. Thd.	63 - 75
	1/2-20 U.N.F. Thd.	97 - 116
Stub Shaft Bolts	3/8-24 N.F. Thd.	21 - 32
	7/16-20 N.F. Thd.	34 - 50
	1/2-20 U.N.F. Thd.	52 - 78
	5/8-18 N.F. Thd.	104 - 156
	3/4-16 N.F. Thd.	182 - 273
Shaft Support Bearing (Center Bearing) Stub Shaft Bolts	3/4-16 U.N.F. Thd.	120 - 160
Stub Shaft End Nuts (Self-Locking Type)	7/8-16 N.S. Thd.	350 - 387
	1-20 N.S. Thd.	550 - 600
	1-1/4-18 N.S. Thd.	700 - 750
	1-1/2-18 N.E.F. Thd.	1000 - 1500
	1-3/4-16 N.S. Thd.	1000 - 1500
Stub Shaft End Nuts (Cotter Pin Type)	Torque nuts to the mean value specified for self-locking type stub shaft end nuts. If the cotter pin cannot be assembled with ease, rotate the nut to the nearest slot and then assemble the cotter pin.	

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Balancing

Chapter 8

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Chapter 8
Balancing

E. R. Wagner, P.E. — Saginaw Steering Gear Div., GMC

Theory

It is well known that a rotating body will not exert any variable disturbing action on the bearing supports when the axis of rotation coincides with one of the principal axis or axes of symmetry of the body. Due to the practical limits of manufacturing, the center of gravity or mass center of a rotating driveshaft, in all but rare instances, does not exactly coincide with its axis of rotation. This rotating eccentric mass center produces an unbalanced centrifugal force proportional to the square of the rotational speed, causing deflection, stresses, and vibration which may result in component failure, objectionable noise, and ride feel for a vehicle's occupants.

Balancing can be defined as the process of altering the distribution of mass in a rotating body to eliminate or minimize vibration at the supporting bearings. Unbalance is the product of weight and displacement and the customary unit for its measurement is the ounce-inch. It is the product of the weight in ounces and the radial distance in inches that the unbalance weight is from the center or axis of rotation.

A shaft may be in static or dynamic unbalance, or both. It can be shown that a statically balanced shaft is not necessarily dynamically balanced. In Fig. 1(a) is shown a shaft with a single disc which is in static unbalance. The unbalance of this single disc can be detected by allowing the disc to rotate about its shaft rotational axis between two horizontal and parallel knife edges. The disc will rotate and come to rest with the heavy side on the bottom. This type of unbalance is called static unbalance since it can be detected by static as well as by rotating means.

It is possible to statically balance the single disc by an additional weight equal to the unbalance but opposite to the original eccentricity. By inspection, it can be seen that the single disc will now be not only statically balanced, but also in dynamic balance. This dynamic balance is evident since the two centrifugal forces cancel each other. In fact, the balancing weight need not be equal to the unbalance weight. It is merely necessary to place the balancing weight at a radial distance such that the centrifugal force developed will be equal to the centrifugal force of the dynamic unbalance. Therefore, in contrast to static unbalance, which can be measured by nonrotating as well as rotating means, dynamic unbalance can only be detected by rotating the shaft and disc system. Also, the system must be rotated at a speed high enough to assure dynamic motion.

When two discs are placed on a shaft as shown in Fig. 1(b), the problem becomes rather complex. Arbitrarily, the arrangement of the unbalance indicated in the illustration is such that the system is statically balanced. Upon rotation, however, the system will be dynamically unbalanced since a bending moment is developed in the shaft. Because of the bending moment, the shaft will rotate in a deflected configuration and whirling will result. Therefore, the problem of the two discs as illustrated in Fig. 1(b) is one of static balance and dynamic unbalance. The two-disc system can be balanced by the same procedure employed in balancing the single disc. However, it is seldom convenient to balance such a system in the planes in which the unbalances occur.

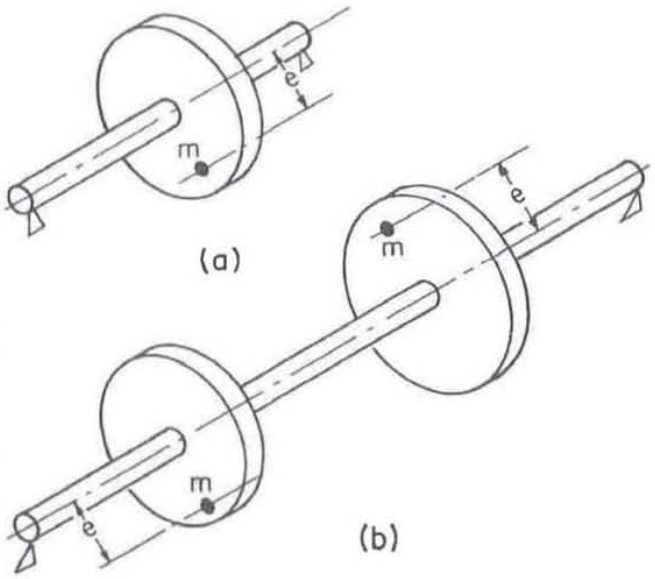


Fig. 1 — Single disc and two disc shaft systems

Fig. 2 shows a general case of a shaft with two discs where the system is both statically and dynamically unbalanced. The unbalanced forces P and Q can always be eliminated by the addition of two correction weights in any two parallel planes of rotation. First, the unbalanced force P can be replaced by two parallel forces Pa/L and Pb/L. In a similar manner, Q can be replaced by two parallel forces Qc/L and Qd/L. The two forces in each plane can then be combined into a single resultant force respectively, R_1 and R_2, that can be balanced by a single correction weight as shown. The two correction weights, C_1 and C_2, introduced in the two parallel planes completely balance P and Q, and the system is statically and dynamically balanced. This is so since there are neither unbalanced forces nor unbalanced moments when the system is in rotation. Therefore, any system that is dynamically balanced is also statically balanced, but the converse may not be true.

Unbalance Effects

Unbalance forces on a typical driveshaft are the results of many individual unbalance effects. These unbalance effects can be

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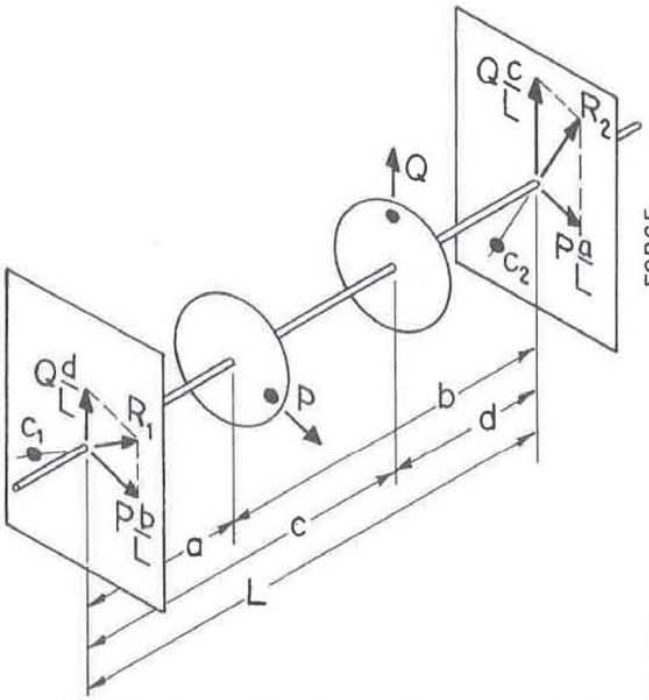


Fig. 2 — System of two discs statically and dynamically unbalanced

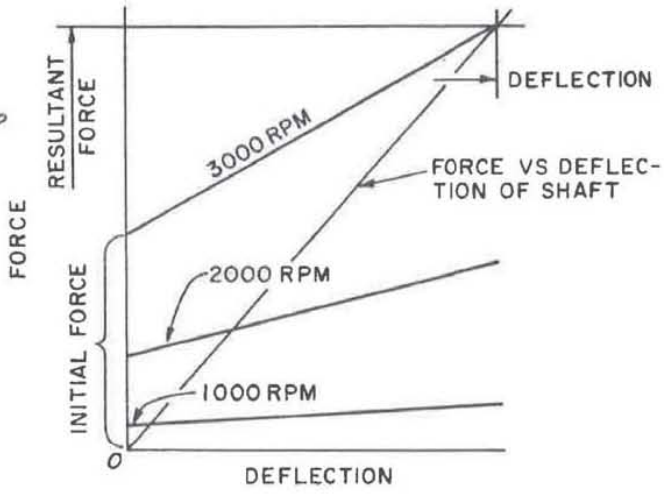


Fig. 3 — Unbalance forces versus deflection

listed as follows:

1. Non-symmetry of rotating parts such as universal joint yokes which can be essentially heavier on one side than the other or tubing which can have a varying wall thickness around the circumferential section. These and other examples of the same type can produce unbalance effects.
2. The driveshaft can be supported off center on its supporting members. The driveshaft depends upon these adjoining structures or units for its support.
3. The driveshaft itself may not be straight. Straightness or the amount of runout defines the extent of this condition.
4. Operating clearances of universal joints, splines, and other components will allow the members that they support to move off center.
5. The driveshaft itself is an elastic member. When subjected to some unbalance forces, it will tend to deflect until the spring restoring forces as a beam are in equilibrium with the resultant unbalance force.

These five basic unbalance effects contribute to the dynamic unbalance. The resultant effect of these conditions can be defined as two basically separate phenomena as follows:

1. An initial unbalance force consisting of an off center relationship, support runout, shaft runout, and looseness.
2. An additional unbalance force resulting from deflections of the driveshaft as an elastic member until an equilibrium condition exists.

Fig. 3 shows a representation of these two conditions. In this figure, an element of the driveshaft is considered. This element is subjected to some spring restoring force which will move back toward center whenever it is deflected. This spring restoring force is a function of the transverse stiffness of the driveshaft itself.

In Fig. 3, force is plotted versus deflection. A straight line from the origin is defined as the deflection rate of this element. This rate defines the restoring force relative to units of deflection. Plotted in this figure is a series of lines illustrating the effects of an initial unbalance force at zero deflection being supplemented by the effects of the increment of mass being moved off center. The point of equilibrium is reached when the restoring force equals the resultant unbalance force. This equilibrium point defines the actual unbalance which exists in the driveshaft.

Some statements which can be made from this figure are:

1. As shaft speed increases, there will be a greater difference between the total unbalance force and the initial unbalance force.
2. The initial force itself is comprised of several elements. The effect of these elements changes with speed.
3. Some shaft speed will exist where the slopes of the two lines are the same. Under this condition, the initial amount of unbalance would increase at the same rate as the spring restoring force increases with deflection. This condition is a representation of the critical speed.
4. The total runout or deflection in a driveshaft changes with speed. A measurement of this deflection change is also a measurement of the unbalance change.

Balancing Parameters

Various factors which influence dynamic balance in a driveshaft must be controlled to provide reasonable limits of unbalance for the entire operating speed range. These parameters which must be kept under control are as follows:

1. Runout of the driveshaft itself is the most significant parameter relative to the way in which unbalance is effected by various operating speeds. Over a wide speed range, especially at high speeds, this parameter is the most important single factor in dynamic balancing.
2. Operating clearances of universal joints, splines, and other components must be maintained as close as possible. These

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clearances allow the driveshaft to move off center during rotation and actually add to the effective runout.

3. Driveshaft components that are non-symmetrical have a lesser effect on dynamic balance. Yokes and most other components subjected to this non-symmetry are generally located on the shaft ends close to where unbalance corrections can be made. These components at the ends are not subject to the deflection changes at increasing speed to the extent that they would be if they were located at or near the midpoint of the driveshaft length. A notable exception exists where tubing has variations in wall thickness. It is very important to maintain uniform wall thickness around a diameter so that those elements of tubing near the mid-length of the shaft will have small initial unbalance.
4. Any dynamic balancing operation on a driveshaft should be performed with the shaft's mass center supported very close to its rotational axis. Corrections can be made and a reasonable limit of unbalance achieved. This operation can be completely nullified if the driveshaft is mounted in supporting yokes or flanges which have large runouts. These runouts cause the shaft to rotate off center and eliminate much of the benefit that a dynamic balancing operation is meant to achieve.

Balancing Limits

In any balancing procedure, the balance specification must include all of the previously described parameters. The specification must establish limits of driveshaft runout since smaller runout limits result in better balancing. The specification must also define limits on fits and clearances and limits of symmetry which must be maintained. Also, accuracy limits in supporting the driveshaft in the balancing machine must be specified. Most important is that the balancing procedure must show a calibration method for measuring the magnitude of the force on the supports.

There is generally little relationship between the summation of unbalance forces on the supports and the amount of correction necessary to balance a driveshaft commercially. The balancing machine indicates the total effect of all of the unbalance variables. The unbalance correction is limited to those elements which are correctable. It is impossible to correct for unbalance resulting from operating clearances and the magnifying effect such as from centrifugal force. These uncorrectable unbalance effects are those associated with the weight of the driveshaft and clearances involved. Operating clearances in splines and universal joint bearings can be maintained at approximately the same limit for any size or type of driveshaft. Hence, the limit to which dynamic balance may be achieved is a function of the weight of the driveshaft itself. Dynamic balance limits in manufacturing should be established, therefore, as a function of the driveshaft weight supported by each end and subject to operating clearances. Components solidly attached to the balancing machine are not affected by operating clearances and, thus, do not influence the unbalance limit considerations.

Balancing Machines

Driveshafts are usually dynamically balanced in machines which rotate the shaft at its operating speed or at some appropriate speed within its operating range, and determine the location and amount of the unbalance force at each of the supporting bearings. Generally, driveshafts which rotate faster than automotive vehicle wheel speed require dynamic balancing.

The most widely accepted dynamic balancing machines are those which rigidly mount the supporting heads and measure unbalance forces through sensitive electronic circuits. These machines essentially measure the force in the same manner as strain gage or other comparable instrumentation would measure any oscillating stress. The comparatively rigid supporting heads in these balancing machines require that the driveshaft deflect as a simple beam. This requirement appears to be the best compromise when performing a dynamic balancing operation. Admittedly, it would be preferable to support the driveshaft on heads which have the same resilient characteristics that exist in its operating environment. Since this condition is not possible to achieve, the comparatively rigid support is used.

Balancing Methods

Conventional Balancing Procedure

The most common driveshaft dynamic balancing method is the conventional balancing procedure which is basically that described previously in conjunction with Fig. 2. Balancing is generally the final operation performed in the assembly of a driveshaft.

Prior to the actual balancing operation, the driveshaft is checked for straightness and, if necessary, corrected to allowable maximum runout limits to reduce the amount of balance correction required. The allowable maximum runout limits are dependent upon the particular driveshaft design and the nature of its application.

Generally, for automotive two-joint driveshafts not exceeding 60 inches in length, the runout must not exceed certain allowable maximum limits when mounted between fixed centers. These limits are affected by the tube size and wall thickness used. In the chapter on *Tubing*, a table is shown which lists the allowable runout, straightness, and ovality of various tube sizes. This table can be used as a guide to assist in determining a practical allowable maximum runout specification. For example, a typical passenger car two-joint driveshaft design with a 3.00 in diameter tube specifies that the runout or straightness deviation must not exceed 0.015 in total indicator reading, excluding 0.009 in tube ovality, when mounted between fixed centers. This runout is measured at the midpoint of the tube length as well as near both ends of the tube. Depending on the particular driveshaft design, similar allowable maximum runout limits are used for automotive multi-joint driveshafts.

In the conventional balancing procedure, the two correction planes required for dynamic balancing are selected as close to each end of the shaft as practical so that the amount of balance

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correction and balance time necessary is minimized. The permissible amount of residual unbalance in each of these planes is expressed in ounce-inches and is determined by the application and the magnitude of the vibration which can be tolerated.

Unbalance correction of the driveshafts is generally accomplished by projection welding to the shaft, suitable size steel weights at the proper angular locations of the correction planes. For automotive driveshafts, typical correction weights range in size from 0.10 to 1.38 oz. The projection welding of weights to the shaft is a rapid operation and can be easily adapted to being performed on the balancing machine. It can be readily incorporated into a completely automatic cycle, especially if the correction weights are metered out and sheared off from strip stock as required. A metering operation overcomes the accuracy limitation imposed by precut weights of various sizes. In selecting the point or plane of correction, the proposed correction weight should not interfere with the function of the shaft nor should it distort or weaken it.

For passenger car and light-duty truck driveshafts, it has generally been the practice to balance between 3000 and 3500 rpm. Police car shafts are usually balanced at higher speeds such as 4300 rpm. Medium and heavy-duty truck and other vehicle applications may require other speed ranges. The permissible unbalance for passenger car and light-duty truck shafts has been approximately 0.50 oz-in maximum at each end and also at the center in the case of a two piece shaft. However, the general practice has been to initially balance these shafts to within approximately 0.25 oz-in maximum. Then, if the balance is rechecked with the shafts oriented in the balancing machine 180 deg from their original position, or if they are rechecked on another balancing machine, the unbalance limit should be within the permissible 0.50 oz-in maximum at the original specified balancing speed.

For medium and heavy-duty truck applications and other heavy duty driveshafts, the permissible unbalance has been approximately 1.00 oz-in maximum for each 10 lb of shaft weight per end. Multipiece shafts are used extensively in medium and heavy-duty truck applications. Therefore, because of the generally large overall length and heavy weight of these drivelines, they are usually balanced in one piece sections, for convenience, to the permissible unbalanced specification.

Inboard Balancing (Overbalancing) Procedure

The inboard balancing method, which is used to a limited extent, differs from the conventional balancing procedure in that the balance weights are attached at approximately one quarter of the shaft length from each end of the driveshaft as shown in Fig. 4. The optimum location of the balance weights was found by a theoretical derivation to be actually 0.23L, where L is the joint center-to-joint center shaft length. Determining the balance weight location by empirical means agrees quite closely with the theoretical location.

The inboard balancing method also differs from the conventional balancing procedure in that it permits the use of tubing with a higher runout. This balancing procedure depends, in fact, on the tubing having a controlled amount of curvature (runout) and with the balance weights placed on the inside of this curvature as shown in Fig. 4.

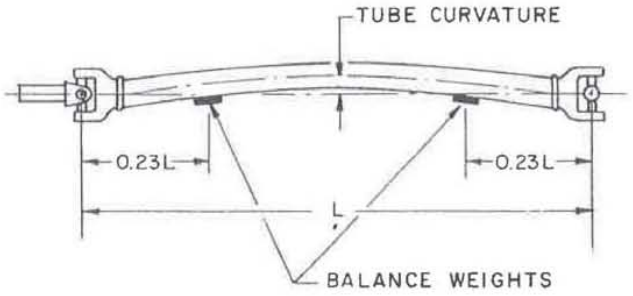


Fig. 4 — Inboard balanced driveshaft

When an inboard balanced driveshaft is properly balanced, it is said to be "overbalanced" or "thru" balanced. A shaft balanced in this manner has the balance or correction weights located on the inside of the tube curvature or runout as previously stated and these weights are of sufficient size to "pull" the shaft thru center at the designated balancing speed. For example, when a correction weight is added on the inside of the bend in the tube and the shaft is then rotated, centrifugal force tends to straighten the tube. The addition of larger weights will further reduce the tube curvature and this runout will become zero when the proper amount of weight correction has been added. A further increase in size of the correction weights will cause the shaft centerline to move past the rotational axis centerline and the tube curvature will be in the opposite direction. When this dynamic runout condition occurs at the designated balancing speed, the driveshaft is said to be "overbalanced" or "thru" balanced.

When the shaft has not yet reached the zero dynamic curvature or straightened condition, it is described as "underbalanced". Therefore, in order for an inboard balanced driveshaft to be properly balanced, it should be overbalanced. This overbalanced condition will permit a given driveshaft to operate at a higher critical speed level than when the shaft is underbalanced.

For a typical passenger car size driveshaft, the maximum allowable runout of the tube at any point is approximately 0.040 in total indicator reading with the manufacture of the tube controlled so that the runout is not less than approximately 0.020 in total indicator reading. The weld seam, because of its added mass, is held preferably in circumferential location at the inside of the bend in order to assist in achieving static balance of the tube. The driveshaft, if it is to be inboard balanced at both ends, is overbalanced at 3500 rpm to a typical unbalance range of from 0.25 oz-in minimum to 0.50 oz-in maximum.

When this balance is rechecked with the shaft oriented at 180 deg from its original position in the balancing machine, or if it is rechecked on another balancing machine, the unbalance limit should be in the range of 0.12 oz-in minimum to 0.38 oz-in maximum at 3500 rpm. Because of underbody clearances, the balance weights on some inboard balanced driveshafts are not placed at their optimum or theoretical axial locations. Frequently, the balance correction at the rear of the shaft is positioned closer to the shaft end than the optimum location.

The inboard balancing technique permits the use of driveshafts which do not require straightening provided that the runout is within the required range previously described. On a balancing machine, an overbalanced shaft tends to operate smoother at

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higher speed, while a conventionally balanced shaft will perform more smoothly in the lower speed range. Driveshafts balanced by the two methods show no noticeable difference in a passenger car when driven at various speeds.

Centrifugal Force Effects

Centrifugal forces developed by the rotation of an unbalanced shaft will result in forced vibration at resonance and also cause fatigue problems. These forces are proportional to the distance of the center of mass from the center of rotation, or eccentricity, to the size of this mass and to the square of the rotational speed. This last factor is of great importance, because it means that when the speed is doubled any centrifugal force due to unbalance is quadrupled. As stated previously, the centrifugal force causes vibration and for a predetermined level of acceptable vibration, if the speed is to be doubled, the unbalance must be reduced to one-fourth of the original magnitude. Fig. 5 shows the centrifugal force characteristics of various levels of unbalance at different shaft speeds and clearly illustrates the rapid increase of centrifugal force.

The general equation for centrifugal force is:

$$F = \frac{W}{g} R \omega^2$$

where:

W = Weight of unbalanced mass

g = Acceleration due to gravity

R = Radius of the mass center from the axis of rotation

ω = Angular velocity

This general equation can be expressed in a more convenient form for use in determining the effect of unbalance. For a given number of ounce-inches of unbalance and a known rotational speed, the centrifugal force is:

$$F = 0.000001776\ WRN^2$$

where:

F = Centrifugal force, lb

W = Weight, oz

R = Radius of weight from axis of rotation, in

N = Rotational speed, rpm

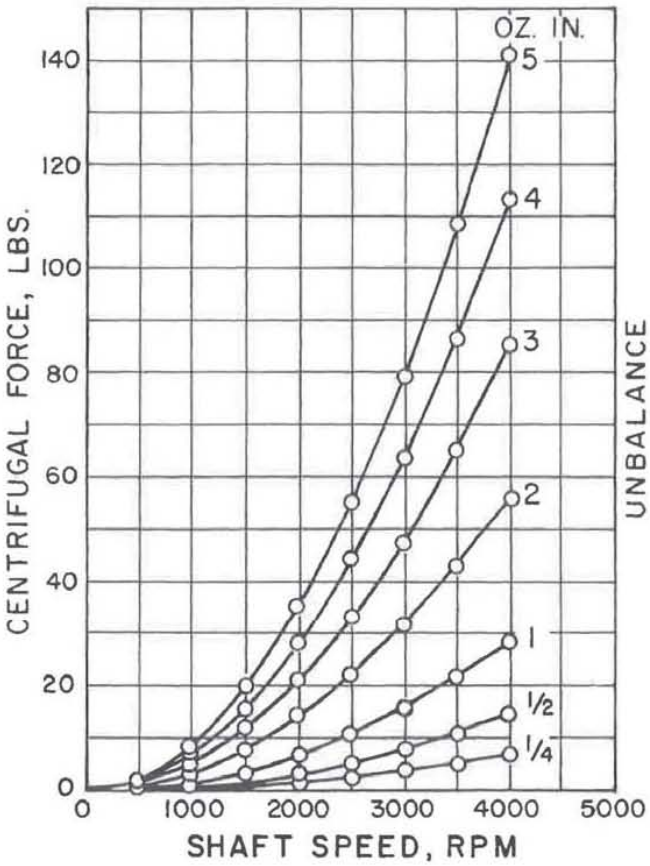


Fig. 5 — Centrifugal force characteristics versus shaft speed for various levels of unbalance

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Vibration Control

Chapter 9

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Vibration Control

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Section 9.1
Basic Vibration — Fundamentals and Terminology

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Introduction

A vibration results from a system (group of components) responding to an alternating excitation with the resulting motion being transmitted either audibly, (can be heard) or physically, (can be felt), to an observer. Vibrations are repetitive, oscillatory motions.

Vibrations are sometimes elusive and difficult to find; however, they do lend themselves to description and definition that fall into patterns and categories that can be analyzed. Prerequisites to such an analysis are a knowledge of vibration in general and the terminology of vibrations in particular. Vehicle vibration problems are perhaps the most difficult to identify and correct. The difficulty arises because the vibration which the driver hears or feels is generally not the source of the problem, but rather a component responding to some external excitation. The source of the vibration may be built into the vehicle and therefore cannot be corrected. In order for a vibration problem to exist, the three basic elements which must be present simultaneously are:

A. A responding system
B. An excitation or exciting force
C. A transmission path (not required when the noise is airborne)

Responding Systems

A responding system is a component or system of components which tends to vibrate at a given frequency, the natural frequency, and in a given type of motion. In order to isolate the responding system, it is necessary to identify what components are capable of providing alternating force inputs or excitations and what systems are likely to respond to these force inputs. Every component of the vehicle is a responding system, that is, every component has a natural frequency. Fortunately, the majority of these components seldom become a problem for the following reasons:

A. The natural frequencies are beyond the range of the exciting force.
B. The system is damped to the extent that no appreciable motion occurs.
C. The systems are physically separated or isolated from the exciting force.

A typical driveshaft and its universal joints can excite various responding systems in a vehicle with first and second order excitations.

Glossary of Terms

Amplitude

The amplitude is the largest value of the displacement of a vibrating member on each side of the mean position.

Antinode

An antinode is a point of maximum amplitude in a vibrating system.

Beaming

Beaming or bending refers to the flexing of any beam-shaped member in the transverse direction, that is, vibrations in a direction perpendicular to the longitudinal axis of the member.

Critical Speed

The critical speed is any speed that produces an excitation frequency which is equal to the natural frequency of the component or responding system. Critical speed produces a resonance and for practical purposes, the terms natural frequency, resonance, and critical speed are synonymous.

Cycle

A cycle of oscillation is the complete sequence of variations in displacement which occur during a period.

Damping

Damping is the dissipation of energy in a vibrating system. Damping can be effected, for example, with mass, friction, hydraulic fluid, or rubber.

Dynamic System

Any system that is in motion is referred to as a dynamic system.

Excitation

An alternating or oscillating force which induces a system to vibrate is an excitation or exciting force.

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Forced Vibration

A forced vibration is one in which an external exciting force of a periodic nature is applied to a system and thus causes the system to vibrate.

Free Vibration

A free vibration is the vibration during which no force is externally applied to the system.

Frequency

Frequency of vibration is the number of cycles occurring in a given unit of time.

Mode

A mode of vibration is a characteristic pattern or shape of the vibrating system. A complex system has many modes and, therefore, will have many modes of vibration.

Natural Frequency

The natural frequency of a body or system is a frequency of free vibration.

Node

A node is the point which is at rest in a vibrating system.

Order or Harmonic

An order or harmonic is the number of excitations or disturbances produced per revolution by a rotating system.

Period

A period is the time required for a periodic motion to repeat itself.

Resonance

Resonance is a condition which exists when the forcing frequency coincides with the natural frequency of a responding system so as to produce a vibration of maximum amplitude.

Responding System

A component or combination of components that vibrate when an exciting force is applied.

Static System

Any system that is at rest is referred to as a static system.

System

A system is any group of attached or interdependent components.

Transmission Path

An element or a system of components which permits a vibration or motion to be transmitted or transferred is called the transmission path.

Vibration

A vibration in general is a repetitive, oscillatory motion.

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Section 9.2
Description of Drivetrain Disturbances

A. H. Berker — Ford Motor Co.

In describing the various vehicle drivetrain disturbances, it is convenient to classify them according to their source of excitation. Basically, there are four vibration generators in the drivetrain of a typical rear wheel drive vehicle with a front located engine. These vibration generators are the powerplant (engine coupled to a transmission), universal joints, driveshaft and rear axle. Therefore, disturbances may be grouped in four separate categories as follows:

Powerplant Induced Disturbances

These disturbances include gear rattle, shudder, clunk, boom, and tip-in-moan, as well as accessory induced disturbances which are described as follows:

Gear Rattle

Gear rattle occurs with manual transmission equipped vehicles during low speed (10-30 mph) acceleration in high gear. Engine firing impulses cause all gearing in mesh to oscillate in a torsional mode through the lash limits of the mating gears. Striking of the gear teeth results in an audible sensation called gear rattle. In typical four cycle engines, torque or firing impulses are related to the number of cylinders as shown in Table 1.

TABLE I — ENGINE TORQUE OR FIRING IMPULSES

Number of Cylinders	Torque or Firing Impulses per Engine Revolution
4	2
6	3
8	4

Shudder

Shudder is a periodic driveline disturbance which is generally experienced at low speed, high torque acceleration. Engine induced shudder occurs at low engine speed in high gear and with a fully engaged clutch. Clutch induced shudder or "judder" usually results from the self-excited vibration (slip-stick) of the clutch plate due to improper clutch linkage modulation. Shudder is further described in this section under *Universal Joint Induced Disturbances* where shudder is caused by nonconstant velocity universal joint secondary couple excitations.

Clunk

Clunk is an impact type of driveline disturbance. It is generally attributed to backlash in transmission or axle gearing, and/or sudden engagement of automatic transmission friction elements.

Clunk, as experienced when a stationary vehicle with running engine is shifted from neutral to drive or reverse, is a static engagement clunk. Whereas clunk, as experienced on tip-in acceleration or tip-out deceleration, is a torque reversal clunk.

Boom

Boom is an audible disturbance experienced during low to medium speed acceleration. Boom may also occur when cruising in the mid-speed range. Generally, boom is associated with vehicle structural resonance and/or drivetrain first mode vertical bending resonance.

Boom is sometimes induced by engine firing impulses over the frequency range of 40-50 Hz.

Tip-In-Moan

Tip-in-moan is an audible disturbance experienced inside a vehicle that is being lightly or moderately accelerated in the speed range of 35-55 mph. This disturbance represents the response of the vehicle as a system to engine firing. However, the primary responding subsystem is the drivetrain which is set into a vibratory motion of relatively large amplitudes when the firing frequency coincides with the drivetrain second mode vertical bending natural frequency. This response is transmitted to the passenger compartment via the rear engine mount and/or the rear suspension.

Others

Under this general heading, a brief reference is made to a sub-group of disturbances where engine accessories act as vibration generators.For example,vibration may be experienced by the normal operation of the compressor or alternator, due to the unbalance in these accessories and due to bracket designs, which may not provide for rigid attachment. Also, pumping noise may be experienced with normal air pump operation that may be objectionable.

Universal Joint Induced Disturbances

Nonconstant velocity type of universal joints will induce disturbances with frequencies that are twice the driveshaft rotational frequency. These disturbances manifest themselves as torsional, inertia, and secondary couple vibrations or excitations and are described in detail in the section on the *Cardan or Hooke Universal Joint*. The double Cardan joint, which is either a near constant or a constant velocity joint depending on its design, may cause clicking or buzzing type of disturbances which are further described in the section on the *Double Cardan Universal Joint*.

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These disturbances associated with the Cardan and double Cardan joints are also briefly described as follows:

Torsional Vibration

The nonuniform rotational velocity of a Cardan joint operating at an angle, produces a torsional vibration. This disturbance also occurs in driveshafts with two or more joints where the resultant difference in joint angles or the equivalent joint angles is excessive, or where the joint yoke phasing is improper.

This disturbance may occur in any continuous operating condition of the driveline, and it is usually frequency and torque sensitive. It very often occurs under full drive or full coast operating modes. Torsional excitation can cause driveline rumble, engine roughness, mid-car shake or gear rattle (manual transmission drivetrains) or a combination of disturbances under high torque load conditions.

Inertia Vibration

This vibration is produced by the oscillating torque loads resulting from the driveshaft inertia being accelerated through nonuniform motion due to excessive joint angles in any continuous operating condition. This disturbance exists in any driveshaft with Cardan joints and occurs even when torsional excitation is eliminated when joint angles are equal in a two-joint system. The level of torsional fluctuation depends upon the joint angle size and the inertia of the connecting shaft member. Inertia vibrations are quite similar to torsional vibrations except that they occur at light or no load and, furthermore, they are frequency sensitive.

Secondary Couple Vibration

Secondary couple vibrations occur whenever a Cardan joint operates at an angle and transmits torque. The magnitude of the disturbance increases with an increase in joint angle and torque load.

These disturbances manifest themselves as vehicle take-off shudder or drive pinion "flutter" at approximately 10 mph (excessive rear joint angle), engine roughness and mid-car shake (excessive front joint angle), and boom "periods" similar to those induced by engine firing. Multipiece driveline installations with intermediate shaft support bearings are quite susceptible to these secondary couple excitations which are torque or load sensitive.

Double Cardan Joint Click

A clicking sound can be caused by the unseating of the centering ball in a double Cardan joint. This clicking sound may occur during relatively high torque vehicle start operations in forward or reverse gear.

Double Cardan Joint Buzz

A buzzing sound can be caused by the unseating of the centering ball in a double Cardan joint. This disturbance may occur in certain vehicles operating usually at relatively high speed conditions. This disturbing characteristic can result at certain drivetrain resonant frequencies where the buzz has been found to be torque and joint angle sensitive.

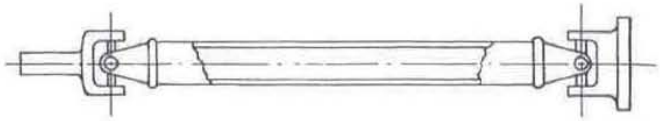


Fig. 1 — Plain steel tubular driveshaft

Driveshaft Induced Disturbances

These disturbances include driveshaft ringing, spline grunt, and drivetrain critical speed. Driveshaft ringing and spline grunt may be attributed to the design of the driveshaft, while drivetrain critical speed is associated with driveshaft unbalance. These disturbances are described as follows:

Driveshaft Ringing

Driveshaft ringing is peculiar to the conventional driveshaft design with a plain steel tube as shown in Fig. 1. The hollow tubing of such a shaft would respond to diverse driveline inputs at approximately 100 Hz to cause an audible metallic disturbance called driveshaft ringing.

Spline Grunt

When a spline is required to slip under torque loads, an objectionable noise may occur in some sliding spline installations. This noise is called spline grunt.

In vehicle driveline applications, this noise may occur especially when the vehicle is started forward, when the driveshaft length must change due to vehicle rear end squat and axle carrier nose windup. In addition, this noise may also occur during sudden severe braking when the driveshaft length must change due to vehicle rear end lift and axle winddown.

Drivetrain Critical Speed

Rotational forces due to unbalance of the driveshaft and/or other components, cause drivetrain vibration at first order driveshaft rotational speeds expressed in rpm. When the driveshaft rotational frequency, in revolutions or cycles per second, corresponds to a bending resonant frequency (usually the first vertical bending mode of the drivetrain), a disturbance can occur. This disturbance is associated with critical speed. Sustained vehicle operation at or near the critical speed may also cause stress failures in the transmission extension and converter or clutch housings.

The critical speed can be determined experimentally on the vehicle. It is also feasible to predict critical speeds analytically by means of a computer technique utilizing the mobility concept, which is described in some advanced vibration references. This method combines the experimentally determined powerplant mobility with the computed driveshaft mobility to generate the drivetrain mobility. A plot of this mobility will yield the first mode and second mode bending resonant frequencies.

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Rear Axle Induced Disturbances

These disturbances are axle noise, start-up groan, axle chuckle, and driveline rumble which are described as follows:

Axle Noise

Axle noise, an audible disturbance which occurs over the frequency range of 350-650 Hz, is generated by the following mechanism. The axle gears are designed to rotate at a constant velocity ratio, that is, perfect conjugation. However, the velocity ratio may be subject to minute variations due to manufacturing compromises. This variation, which is termed conjugation deviation or error, is generally present at single or twice tooth contact frequency. This conjugation deviation causes the drive pinion to speed up and slow down with respect to the ring gear for each tooth engagement and thereby generates drivetrain vibrations usually of very small amplitudes. For example, the axle housing assembly tends to oscillate in a pitching motion. However, when the forcing frequency, at a given vehicle speed, coincides with a bending or a torsional natural frequency of the drivetrain, resonance occurs and axle housing pitching motion is greatly amplified. The signal is then transmitted to the passenger compartment primarily through the rear suspension and the vehicle structure, and secondarily via an airborne path.

Axle noise may be determined by subjective car evaluation. However, computer programs have been devised to predict, in the design stage, the following axle noise characteristics of a vehicle:

1. Problem frequencies (Hz) and corresponding vehicle speeds (mph).
2. Severity (major or minor) and nature (torsional or bending) of the problems.

Start-Up Groan

Start-up groan, which is also known as G-string, occurs as the vehicle starts moving in forward or reverse gear. This disturbance is usually limited to automatic transmission equipped vehicles. Basically, it is a self-excited vibration in the frequency range of 100-150 Hz caused by slip-stick at the drive pinion and ring gear mating tooth surfaces.

Axle Chuckle

Axle chuckle is due to the excessive clearance between the hub of the differential side gear and the bore of the differential case. Axle chuckle is normally experienced when the vehicle is allowed to coast from high speed to creep. However, it may also occur in float and, at worst, in cruise.

Driveline Rumble

The eccentricity of the drive pinion, due to gear pitch line runout, would act like the axle gear teeth conjugation error, thereby causing the axle housing to vibrate in a pitching mode. However, this gear runout will not contribute to axle noise because its frequency is relatively low. On the other hand, this signal is within the range of rear suspension natural frequencies. Consequently, rear suspension resonances would occur at given speeds and the resulting vibration is called driveline rumble.

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Section 9.3
Control of Drivetrain Disturbances

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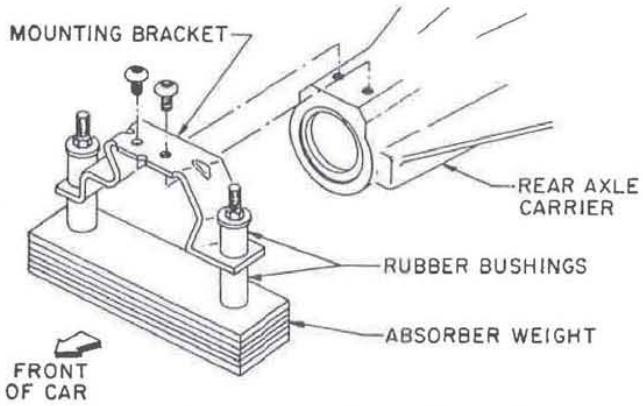


Fig. 1 — Rear axle mounted body boom tuned absorber

In the previous section on *Description of Drivetrain Disturbances*, these disturbances were described under four separate groupings, depending on their source of excitation. Therefore, the various methods for controlling these disturbances are also grouped in the four categories as follows:

Powerplant Induced Disturbances

Clutch Tuning and High Inertia Engine Flywheel

The clutch may be tuned so as to reduce or eliminate engine induced gear rattle and boom periods. Basically, such a tuning involves reducing the rate of the coil springs that are circumferentially located on the clutch hub, and also the clutch friction (Coulomb) damping. The latter can be effected by a redesign of the friction ring or element (between the clutch hub flange and the driven member) and/or a change in friction material. However, the side effects of these modifications should be carefully considered.

Increasing the rotational inertia of the engine flywheel may also be an effective means for reducing or eliminating engine induced gear rattle and boom periods. However, it may sometimes be necessary to combine clutch tuning with a high inertia flywheel in order to effectively control engine induced gear rattle and boom periods.

Engine Firing Induced Body Boom Suppressor

This device consists of a rubber bushing isolated mass attached to the pinion nose of the rear axle carrier with a suitable bracket. The absorber is generally tuned in the frequency range of 40-50 Hz. The absorber weight is usually either a laminated or a cast construction with a typical design of the laminated construction illustrated in Fig. 1.

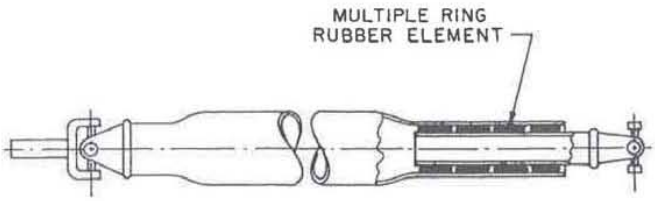


Fig. 2 — Rubber element tube driveshaft

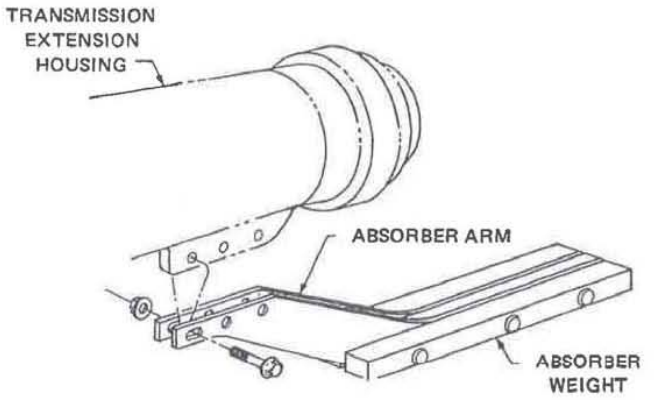


Fig. 3 — Transmission mounted tuned absorber

Rubber Element Tube Driveshaft

A typical design of such a driveshaft is shown in Fig. 2. This special driveshaft is quite effective in reducing gear rattle and clunk and is equally effective as an axle noise attenuator. The design of this shaft is described in detail in the following section on *Rubber Element Tubes*.

Generally, this attenuator consists of four to eight rubber rings bonded by an epoxy type of adhesive or vulcanized to a short inner tube and pressed into a longer outer tube. In this design the drive torque is transmitted entirely through the rubber element. The rubber element is normally positioned at the rear of the driveshaft. In the "reversed" or "inverted" design, the rubber element is located at the front of the driveshaft.

Axle Shafts with Altered Torsional Rates

Changing the axle shaft diameters, and thereby its torsional spring rate, may alleviate the low speed gear rattle, boom, and shudder problems.

Transmission Mounted Tuned Absorber

The absorber as shown in Fig. 3 consists of an arm with one end attached to a weight and the other end to the transmission. In a

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typical design, the arm features one or two steel plate stampings riveted to a laminated weight. Ordinarily, the absorber is bolted to a flange provided on the underside of the transmission extension housing. The device may be tuned to a problem frequency by adjusting the stiffness of the arm (absorber spring) and/or the weight (absorber mass). This type of absorber is considered to be very lightly damped as the damping inherent in the system, or the internal hysteresis, is very low.

The absorber is principally a tip-in-moan attenuator. For that purpose, it is tuned to the drivetrain second mode vertical bending natural frequency in the range of 95-130 Hz.

The absorber is also effective in reducing boom. In this application, the tuning frequency would correspond to the first mode vertical bending natural frequency in the range of 65-95 Hz.

Mode Shaping Technique

A vibrating system will exhibit a characteristic form, envelope, or shape at a given frequency. This is commonly called the mode shape of the system at that frequency. Basically, the mode shaping technique utilizes the concept of mode shapes in the diagnosis of drivetrain disturbances. However, frequencies of interest are confined to that of drivetrain first and second mode bending resonances. Mode shapes are generated by shaking the drivetrain as described in the following test case and measuring the displacement at various stations in the system.

Initially, it was analytically determined for the test case that the first and second mode bending resonant frequencies for the typical drivetrain were 76.5 Hz and 113 Hz, respectively. On the basis of this information, mode shaping is implemented as follows:

The vehicle equipped with that typical drivetrain is set up on a test stand where a constant input force is applied at the rear end of the universal joint slip yoke of the driveshaft at a frequency of 76.5 Hz. The envelope or mode shape is determined by probing along the longitudinal axis of the drivetrain. A typical plot of the relative amplitudes is illustrated as a solid line (graph 1) in Fig. 4. The test then is repeated with a constant force input at 113 Hz, with the resulting plot shown as a dashed line (graph 2) in the same figure.

Examination of these two graphs, 1 and 2, in Fig. 4 would disclose that relatively larger excursions or amplitudes take place along the driveshaft than along the remainder of the drivetrain. Furthermore, the relative amplitudes are zero at some points along the drivetrain. These stationary locations are called nodes or nodal points. It will be noted that for either first or second mode bending resonances of this particular drivetrain, the front node is common and the rearmost node occurs in the proximity of the rear universal joint.

Also, graph 1 would indicate that the middle node M_1 for the first node bending resonance is situated at the rear end of the transmission extension housing. This plot suggests that this nodal point would be the optimum location for the rear engine mount with regard to the boom problem. Such a redesign would reduce the transmission of vibrations to the vehicle structure and, hence, minimize boom. (The association of boom with vehicle structural resonance was previously mentioned.)

Similarly, the relocation of the rear engine mount to M_2 would

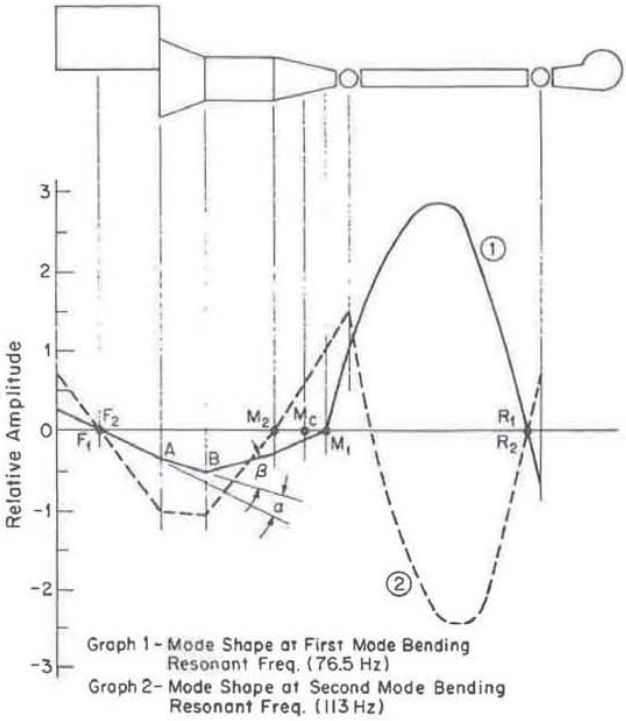


Fig. 4 — Drivetrain mode shapes

tend to attenuate tip-in-moan, a disturbance that is generally associated with the second mode bending resonance, as previously described. However, in the event that boom and tip-in-moan should be equally objectionable, it may be advisable to position the rear engine mount at some location, say M_c, between M_1 and M_2 as a viable compromise.

It is pertinent to point out that the path of vibrations emanating from the drivetrain may be either through the rear engine mount or the rear spring attachments, or both. Consequently, the mode shaping technique will not hold if the vibration path should happen to be solely through the rear spring attachments.

The application of the mode shaping technique to drivetrain critical speed studies is described in this section under *Drivetrain Critical Speed Attenuation*.

Control of Accessory Induced Disturbances

In general, control of inherent unbalance in the rotating members of accessories would alleviate these disturbances. Also, these accessories should be properly attached so as to inhibit vibrations.

Universal Joint Induced Disturbances

The section on the *Cardan or Hooke Universal Joint* discusses in detail the guidelines and procedures for minimizing the torsional, inertia, and secondary couple disturbances associated with driveshafts incorporating such a joint, while the section on the *Double Cardan Universal Joint* describes procedures for minimizing the clicking and buzzing disturbances associated with

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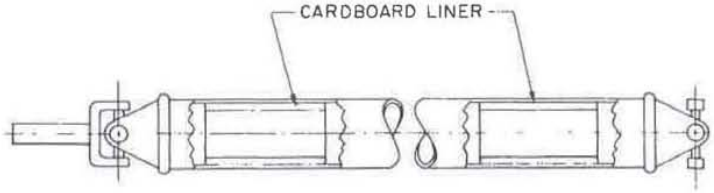


Fig. 5 — Steel tubular driveshaft with two cardboard liners

double Cardan joint designs. Methods for minimizing these universal joint induced disturbances are described briefly as follows:

Torsional Excitation Limits

The effect of two or more Cardan joints operating in series in a driveline can be expressed as an equivalent joint angle. In passenger car and most truck applications, the maximum level of torsional excitation should be maintained under approximately 400 rad/s² angular acceleration in any continuous operating condition for a generally satisfactory driveline installation. This excitation level is comparable to that of an equivalent joint angle of approximately 3 deg operating at 3600 rpm. To reduce torsional vibrations to an acceptable level, the driveline geometry can be altered in a vehicle to change joint angles by revising the engine and pinion nose angles and, therefore, decrease the equivalent joint angle.

Inertia Excitation Limits

For passenger car and light-duty truck applications, the maximum level of inertia excitation should be maintained under approximately 1000 rad/s² angular acceleration in any continuous operating condition for a generally acceptable driveline installation with Cardan joints. This excitation level is comparable to an equivalent angle of approximately 4.75 deg operating at 3600 rpm. Larger size trucks and certain other highway as well as off-highway vehicles, agricultural equipment, power take-off, and stationary installations may generally tolerate higher levels of disturbances.

Inertia vibrations may be reduced by decreasing the equivalent joint angle and/or the size of the driveshaft. For example, a shorter driveshaft incorporating a smaller diameter tube and set at a reduced equivalent angle would be less susceptible to inertia vibrations.

It should be noted that the equivalent joint angle limits specified for acceptable levels of torsional and inertia excitation relate to idealized situations. In practice, the level of these disturbances have sometimes been found to be acceptable with joint angles that are relatively greater than those suggested.

Secondary Couple Excitation Limits

Secondary couple excitation is particularly critical in automotive vehicles with three or more universal joints in the driveline arrangement. These multipiece driveshafts must be supported with one or more intermediate shaft support bearings, which because of their design requirements are quite sensitive to Cardan joint secondary couple and other driveline disturbances. The design of these bearings is described in the chapter on *Shaft Support Bearings*. For generally acceptable passenger car and truck drivelines, the upper limits of the secondary couple oscillating force are shown in Table 1. It should be noted that these rotating couple forces act twice per driveshaft revolution on the shaft support bearing.

TABLE 1 — MAXIMUM DYNAMIC COUPLE FORCES

Installation	Maximum Dynamic (Rotating) Couple Force, Pounds
Passenger Cars	8
Light-Duty Trucks	15
Medium-Duty Trucks	25
Heavy-Duty Trucks	75

Double Cardan Joint Click Attenuation

The clicking sound caused by the unseating of the centering ball in a double Cardan joint may be reduced or eliminated. The unseating of the centering ball can be attenuated by proper design of the centering device components or by restricting the maximum operating joint angle. In the typical centering ball and seat type of construction used in some double Cardan joints, the seat design and the spring load should be optimized to prevent the unseating of the centering ball.

Double Cardan Joint Buzz Attenuation

The buzzing sound caused by the unseating of the centering ball in a double Cardan joint can also be reduced or eliminated. At given vehicle speeds, certain forcing frequencies in the driveline system have been known to induce this disturbance by causing the centering mechanism to respond to that frequency. It has been found that when this exciting frequency is eliminated or shifted, the unseating condition may be attenuated. However, the buzzing problem may be generally eliminated readily by simply increasing the spring load, or by reducing the operating joint angle.

Driveshaft Induced Disturbances

Liners

The liner is generally an oversized tube of spiral wound corrugated cardboard paper cut to nearly the full length of the driveshaft tube. The liner may also be cut in shorter sections for installation at one or both ends of the driveshaft tube. A typical design of the latter arrangement is shown in Fig. 5. Cardboard liners will tend to reduce longitudinal vibrations (driveshaft ringing) in the driveshaft tube. Cardboard liners are further described in the section on *Liners*.

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Control of Spline Grunt

Spline grunt may be eliminated by reducing the axial thrust force required to slip the spline. Solutions to this problem are discussed in the chapter on *Splines and Axial Motion Devices*.

Drivetrain Critical Speed Attenuation

The control of critical speed vibration requires a sufficiently high driveshaft critical speed and/or a low unbalance. This solution is also described in the chapter on *Critical Speed*.

The first requirement for the control of critical speed vibration, namely, high driveshaft critical speed, may be met by revising the bending properties of the drivetrain as follows:

Use of low density driveshaft tubing material, such as aluminum or other material, provided that these materials have adequate mechanical properties to transmit the torque load.

Driveshaft redesign to incorporate:

1. A shorter length tube. This change may be achieved by lengthening the transmission extension housing (while preserving its rigidity) and/or the use of a three-joint, two piece driveshaft.
2. A larger tubing section moment of inertia without increasing the weight per unit of length. Enlarging the outside diameter and reducing the wall thickness of the driveshaft tubing would meet this requirement.

Powerplant (engine coupled to transmission) redesign to provide maximum rigidity and minimum weight. This approach would imply increased rigidity in powerplant components and across their joints or connections, and replacement of cast iron with lower density material components.

In a given drivetrain, driveshaft changes would produce a slight increase in critical speed, while powerplant modifications would result in a significant increase. The reverse may well hold true for another drivetrain. It appears, therefore, that the driveshaft and powerplant must be properly matched so as to obtain the maximum critical speed.

The second requirement for the control of critical speed vibration, namely that of low unbalance, is equally important because, with an unbalanced driveshaft, the problem may be experienced even though the critical speed is greater than the vehicle maximum speed. The greater the unbalance, the lower the speed at which the problem will be felt below the critical speed. Maximum unbalance of the driveshaft should be specified so that the maximum unbalance force throughout the operating speed range will not excite the system.

The shaft should be designed so that the critical speed of the drivetrain is higher than the maximum design operating speed of the vehicle. As a guideline, it is suggested that the critical speed should be approximately 15% above the vehicle maximum speed for driveshafts having an unbalance of 0.50 oz-in or less in passenger car and light-duty truck applications.

By way of an illustration, consider the hypothetical vehicle that was subjected to mode shaping analysis. This analysis was previously described in this section under *Mode Shaping Technique*. Also, it was stated that the drivetrain critical speed was associated with the lowest or first mode bending resonant frequency. This frequency was 76.5 Hz for the vehicle of interest.

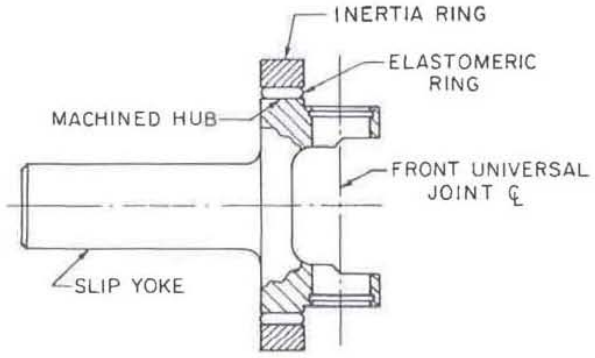


Fig. 6 — Slip yoke mounted tuned absorber

For this investigation, the following assumptions are made:

1. The driveshaft unbalance is not excessive.
2. The ratio of the driveshaft speed to vehicle speed (in high gear) is 50 rpm/mph.
3. The vehicle maximum speed is 120 mph.

These assumptions suggest that:

The critical frequency is 76.5 Hz and therefore the critical speed is:

$$N_c = \frac{76.5 \times 60}{50} = 92 \text{ mph, approximately.}$$

Since the maximum speed is 120 mph, the vehicle will experience vibrations associated with critical speed over the speed range of about 92-120 mph.

Therefore, it is evident that this vehicle would require a drivetrain redesign to render it acceptable for critical speed. Here again, the mode shaping technique will afford a solution to the problem.

Inspection of graph 1 in Fig. 4 would indicate that there are discontinuities or sharp breaks at joints or connections between drivetrain components. Also β, the slope angle at B, is somewhat greater than α at A, or at any other discontinuity. By definition, the slope or the slope angle of the mode shape diagram at a given location is actually a measure of the stiffness or rigidity at that location. Further, the greater the slope, the more flexible is the location. Consequently, it is concluded that, relatively speaking, joint B between the converter housing and the transmission case is highly flexible. Hence, this joint must be made stiffer in order to raise the critical speed above the vehicle maximum speed. This analysis implies that both the converter housing and the transmission case should be redesigned in order to provide maximum rigidity at their connecting joint. An obvious course of action would be to eliminate the joint by designing a converter housing that is integral with the transmission case. Another alternative would be to provide thicker flanges at the joint and additional bolts, and also to adequately gusset the two components in the proximity of their connecting joint.

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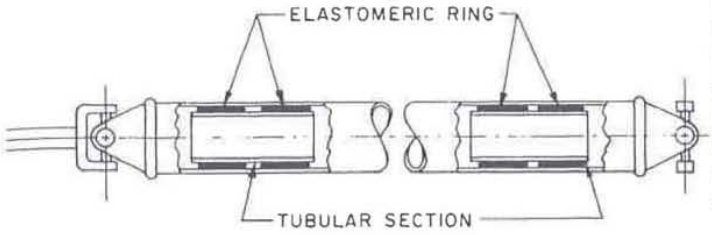


Fig. 7 — Untuned internal absorber with two tubular sections

Rear Axle Induced Disturbances

Axle Noise Attenuators

It was indicated previously that the nature of the axle noise problem may be either torsional or bending. Actually, axle noise may be both a torsional and bending problem since the two modes are coupled through the rear axle assembly. However, the primary cause of a specific resonance relates either to the torsional or bending properties of the drivetrain. Therefore, it is customary to identify a resonance as a "torsional" or "bending" problem and to identify the corresponding attenuator as "torsional" or "bending".

"Torsional" attenuators include tuned absorbers, inertia discs, and rubber element tube driveshafts. "Bending" attenuators include the so-called "untuned" internal absorbers and driveshafts with low bending compliance.

Tuned Absorber—A tuned absorber consists generally of a relatively thin elastomer ring mounted concentrically between a steel or cast iron inertia ring and a cylindrical section of the drivetrain. The section thicknesses of typical elastomer rings have been in the range of approximately 0.09-0.13 in, while inertia ring outside diameters have ranged from 3.50-5.50 in approximately.

The absorber may be mounted on the universal joint slip yoke hub, driveshaft tube, companion flange hub, or axle housing tubes. In typical applications, the slip yoke and companion flange require machined hub diameters in order to accept the tuned absorbers. The slip yoke mounted absorber ring element shown in Fig. 6 has a rectangular cross section. The driveshaft is balanced with the tuned absorber mounted on the slip yoke.

Driveshaft and axle housing tube mountings may be either external or internal. These designs would require closely controlled mounting diameters at these locations. The tuned absorber is further described in the section on *Torsional Dampers*.

Inertia Disc—The inertia disc or flywheel is identical to the tuned absorber except that it does not incorporate an elastomer ring. The disc is either pressed on the machined hub diameter of the slip yoke or companion flange, or cast integral with these components.

The inertia disc is seldom mounted on the driveshaft tube or axle housing tube. Detuning with inertia discs is effected by introducing additional inertia in the drivetrain system.

Rubber Element Tube Driveshaft—A typical design of this device as shown in Fig. 2 was described previously.

Untuned Internal Absorber—Such a device is used primarily to control drivetrain "bending resonances." The absorber as shown in Fig. 7 is a typical design composed of two internal tubular ring sections. This arrangement is identical to the torsional tuned absorber mounted internally on the driveshaft except that the elastomer rings are relatively stiffer. Consequently, the tuning frequency of such a device is the range of 650-750 Hz, which is above the normal 350-650 Hz frequency range of axle noise. For that reason, such an absorber is identified as "untuned".

The function of the elastomer rings is to hold in position the tubular absorber ring. Sometimes the "untuned" internal absorber does not incorporate any elastomer rings. Then the tubular section is press fitted within the driveshaft tube. The arrangement provides an additional inertia to the system and thereby modifies its bending characteristics. The absorber is usually located within the driveshaft at positions where "bending" peaks (antinodes) have been determined to be significant.

Driveshaft with Low Bending Compliance—Such a driveshaft would feature a shorter length, thinner wall tubing, and a larger outside diameter. However, it is mandatory that the powerplant rigidity is not sacrificed to shorten the driveshaft. This design would change the "bending" characteristics of the drivetrain and thereby attenuate axle noise associated with a "bending resonance".

Therefore it is possible to find a sufficiently adequate axle noise attenuator on the basis of car evaluations. However, this approach would be impractical and time consuming for the development of the optimum attenuator. For example, a torsional tuned absorber must be evaluated first at various driveline positions in order to obtain the optimum location. Then further car tests are required to determine the optimum configuration (absorber inertia mass, tuning frequency, and damping factor). Furthermore, if axle noise is related to a "bending" problem, it must be ascertained whether the "untuned" absorber or a "stiffer" driveshaft is preferable. Then if an "untuned" attenuator is selected, further vehicle tests must be performed to determine the optimum weight and optimum locations of the tubular sections.

However, the lengthy vehicle evaluations may be simulated by computer analysis in a much shorter time. Then, only a few vehicle tests need be scheduled for verification and final tuning.

Start-Up Groan Attenuator

This device is similar to a slip yoke mounted torsional tuned absorber that has been tuned to the problem frequency in the range of 100-150 Hz. The section thicknesses of typical elastomer rings used in this type of absorber have been in the range of approximately 0.19-0.28 in. The design of the start-up groan attenuator is further described in the section on *Torsional Dampers*.

Driveline Rumble Attenuator

The design of this attenuator is similar to that of the transmission mounted tuned absorber described previously. However, in this application, the absorber is bolted to the rear axle carrier and nestled along the driveshaft. It is generally tuned in the frequency range of 40-60 Hz.

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Vibration Control

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Section 9.4.1
Rubber Element Tubes

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Introduction

The driveshaft in a conventional vehicle design, serving as the torque transmission device between the primary power source and the driven member, can react torsionally in the following three ways:

1. As an exciter of torsional vibrations.
2. As a transmitter or response mechanism to the disturbances from other drivetrain units.
3. As an isolator when properly tuned to produce a minimum transfer of excitations in critical response ranges.

The rubber element tube is a torsionally resilient device incorporated in the driveshaft to accomplish isolation of these effects. Basically it consists of an inner tube and an outer tube joined by the torsionally resilient rubber element as shown in Fig. 1 in a typical construction. The transmission of torque is through this rubber element, which serves to attenuate disturbances by providing resistance to or by braking the "noise path".

As a torsionally resilient spring located between driving and driven mass inertias, the rubber element tube, by design, can be used to tune the effects of torsional resonances and dampen the effects of energy transfer such as:

1. Peak torque loads delivered by engine misfiring.
2. High kinetic energy torque loads resulting from instantaneous stoppage of a mass.
3. Torque loads resulting from starting to turn a high inertia load.

Rubber element tubes in their various design forms are generally used in conjunction with compatible universal joints and other driveshaft components to provide a torsionally resilient torque transmitting device between a driving and driven member. The rubber element tube, as part of the torque transmitting system or driveshaft, offers the following desirable features:

1. It can be designed to oscillate through relatively large amplitudes.
2. When positioned between universal joint centers, it must transmit only torque and the remaining oscillations without having to flex through high frequency angular misalignment such as required by typical flexible couplings.
3. With its location between universal joints, it is generally positioned away from the sources of high temperatures and the driveshaft attachment problems at each end.
4. With reasonable size proportions, it can usually be designed with an adequately low stress level for relatively long life.

The disadvantages of a driveline incorporating a rubber element tube are as follows:

1. Increased cost.
2. Increased weight.

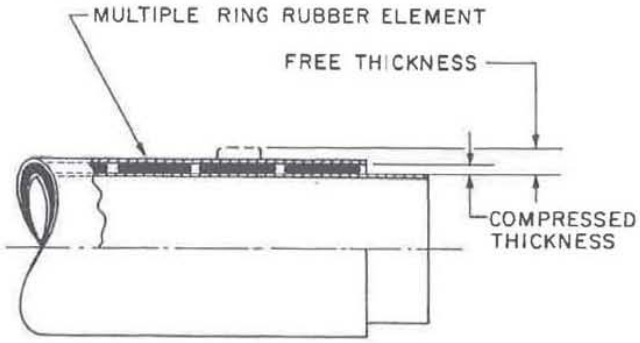


Fig. 1 — Compressed elastomeric ring, rubber element tube construction in a parallel loading arrangement

3. A problem in dynamic balancing may occur if the length of the rubber element used is relatively short.
4. The danger of an improper installation where the torsional resonance is brought down within the operating speed range. The resonant frequency could be well above the operating range without the resilient member or when using a conventional plain tubular driveshaft.

Rubber Element Tube Constructions

Depending on the application requirement, the rubber element tube can be designed in various constructions in order to accomplish its intended purpose. Generally, these constructions are either the *compressed multiple elastomeric ring* or the *molded elastomeric section* designs using round tubing. Another construction which is used is a *square tubular* design incorporating compressed cylindrical elastomeric rollers.

Fig. 1 illustrates a typical *ring type* rubber element tube where the rings are in a parallel torsional spring loading arrangement. This basic design consists of two concentric steel tubes with a number of elastomeric rings compressed between the tubes. The rings are usually bonded by an epoxy type of adhesive or sometimes molded to a generally shorter inner tube and held by friction resulting from the compressive fit with the outer tube. The number of rings required depends on the maximum transmitted torque in conjunction with a length necessary to provide adequate transverse stiffness and a minimum effect on critical speed and dynamic balance. The elastomeric ring sections provide, within practical limits, a means to adjust the torsional spring rate of the rubber element tube, as well as a degree of damping. This type of rubber element tube, being relatively stiff in torsion, is usually used in high speed applications such as passenger car drivelines.

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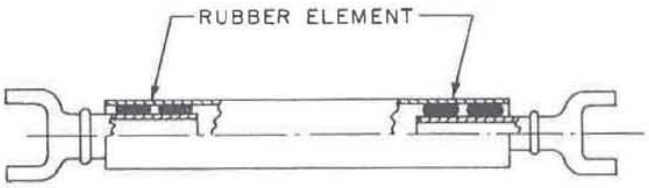


Fig. 2—Compressed elastomeric ring, rubber element tube construction in a combined parallel-series loading arrangement

If a relatively softer torsional rate is desired, the *ring type* rubber element tube can be designed in a combined parallel-series torsional spring loading arrangement as shown in Fig. 2. This typical design is commonly referred to as a *double ended* rubber element tube.

The molded elastomeric section type of rubber element tube can be designed in various constructions. Fig. 3 illustrates such a typical construction where the molded rubber section is employed between an inner tube and a two, or multiple piece segmented outer tube. Shrinking stresses caused by cooling of the bonded elastomer are relieved by using the split shell outer tube design. The molded assembly is then pressed into a solid outer tube which serves as a housing. The resulting reduction of the outer diameter places the rubber in compression. This type of construction provides a greater degree of flexibility in spring rate adjustments for a particular application. It is generally used in low speed applications which produce large amplitudes or vibrations requiring a relatively softer cushion or spring rate. The molded end contour of the elastomeric section is usually scalloped in an elliptical concave shape to permit flexing and windup of the rubber, without destroying the bond between the inner and split shell outer tube.

The size of a rubber element tube increases as the spring rate and torque requirements increase. With an increase in size proportions, the complexity of the design also increases. Fig. 4 shows such a more complex design, which also employs the molded elastomeric section construction. Because of the additional weight, typical bronze plain bearings are added to effectively centralize the rubber element assembly for balance purposes. Provisions for lubrication and sealing of the plain bearings are also incorporated. Lubrication of these plain bearings is required to minimize wear and the detrimental effects of heat generated by oscillation of the rubber element. This design is generally used in an application where the driveshaft is connected directly between the driving and driven mass inertias without an intermediate form of torsional damper.

Rubber element tubes, which are extremely soft and provide a relatively high degree of torsional windup, usually require an integral angulation stop. A typical design of this type is illustrated in Fig. 5. The metallic angulation stop in this design limits the torsional windup to an angle which will not shear and tear the molded rubber.

The square tubular construction with compressed cylindrical elastomeric rollers is comprised of two square, cross-section tubular members, one within the other, or an outer square tube and an inner square bar shaft. A typical design is shown in Fig. 6 in a

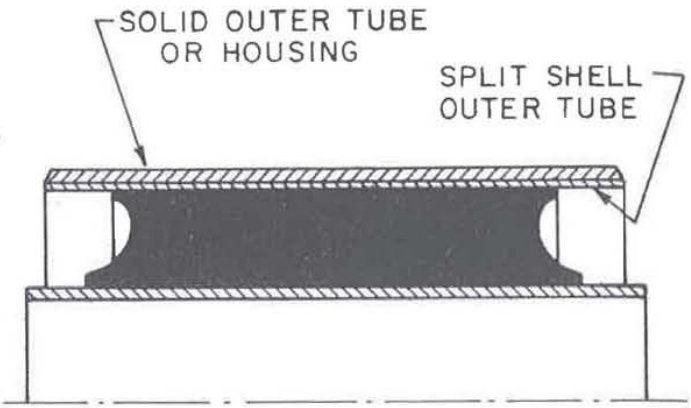


Fig. 3 — Molded elastomeric section, rubber element tube construction

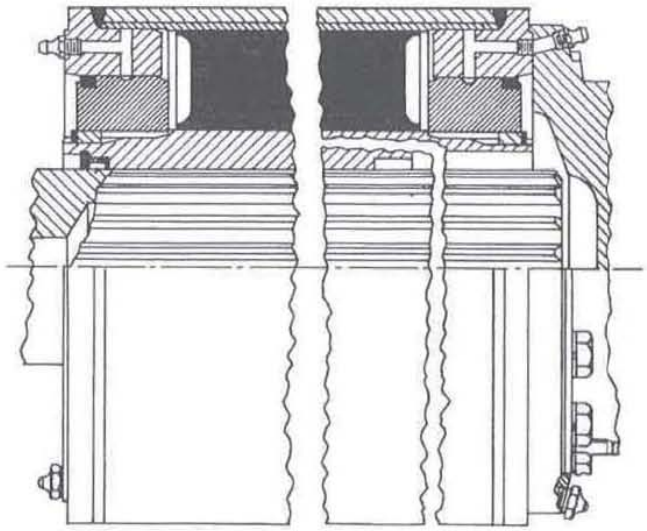


Fig. 4 — Molded elastomeric section, rubber element tube construction with plain supporting bearings

sectional view. The sides of each square member are at 45 deg to those of the other, and a rubber roller of suitable diametral size and length is located in each of the four triangular shaped spaces formed between the two coaxial members. The rubber rollers are most commonly assembled by being initially frozen and then positioned in these triangular shaped spaces. Upon thawing, these rollers increase in size and are placed under compression.

When torque is applied, the rollers roll and are simultaneously deformed. A characteristic of this rubber element tube design is that it provides a progressive increase of the spring rate as the torque load increases. It is generally used in low speed applications.

This design will not be discussed further in the remainder of this section on *Rubber Element Tubes*. Only the compressed ring and the molded rubber element tube constructions will be described in more detail.

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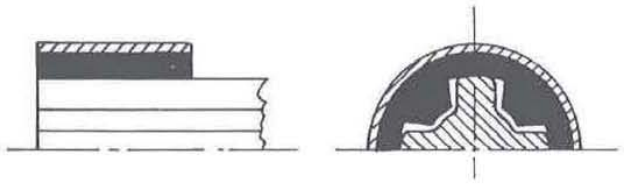


Fig. 5 — Molded elastomeric section, rubber element tube construction with integral angulation stop

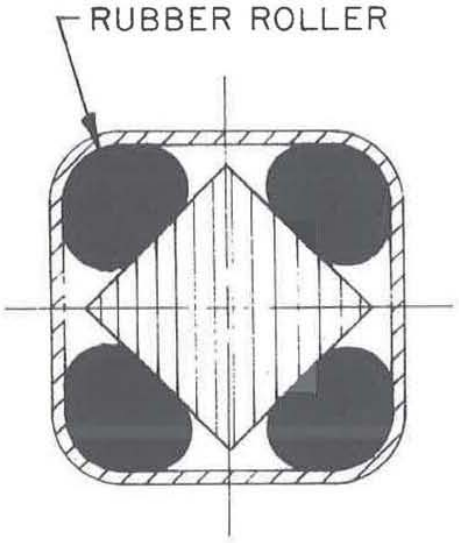
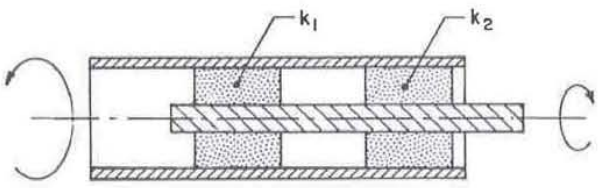
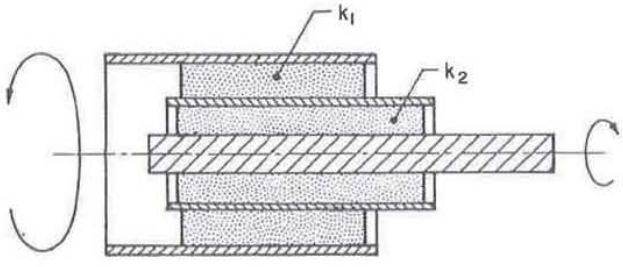


Fig. 6 — Square tubular rubber element, tube construction with compressed cylindrical elastomeric rollers



PARALLEL LOADED TORSIONAL SPRING



SERIES LOADED TORSIONAL SPRING

Fig. 7 — Rubber torsional spring arrangements

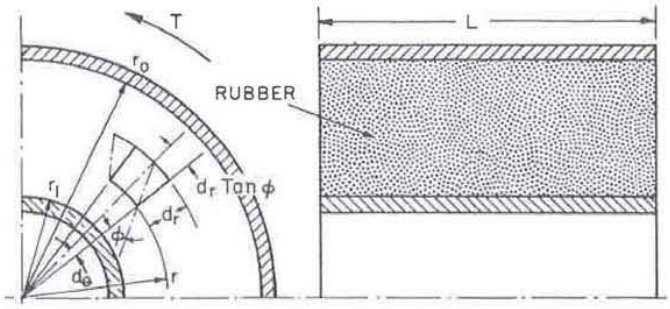


Fig. 8 — Rubber torsional spring relationships under load

Design

Basic Parameters

The design of rubber element tubes is governed by the following basic parameters:

1. Application usage
2. Transmitted torque and speed
3. Compatibility of universal joint and driveshaft size with the transmitted torque and speed
4. Required torsional spring rate
5. Adequate fatigue life
6. Required axial thrust resistance

Elastomeric Torsional Spring Arrangements

The application of a rubber element tube ranges from a noise attenuating device or damper in its simplest form, to a specially designed torsional elastic spring in its more sophisticated form. Regardless of its application intent, the end result with a complement of rubber between an inner and outer tube, produces a torsional spring. As shown in Fig. 7, this torsional spring can be designed basically in either a *parallel* or in a *series* loading arrangement, depending on the application requirement. In addition, a torsional spring can also be designed in a combination of these two arrangements. In the *parallel* torsional spring system, the resultant torsional spring rate is:

$$k_R = k_1 + k_2 + \ldots + k_n$$

where:

k_1, k_2 and k_n = Individual elastomer section torsional spring rates, lb-in /deg or lb-in/rad

In the *series* torsional spring system, the resultant torsional spring rate is:

$$k_R = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \ldots + \frac{1}{k_n}}$$

Elastomeric Torsional Spring Analysis

The basic analytical design characteristics for a cylindrical, elastomeric torsional spring can be developed. Fig. 8 illustrates such a typical rubber spring deflected by a torsional load applied

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about its axis. The torsional load distorts the elastomer in shear. When the length of the rubber L is constant, the shear stress at any radius r is:

$$S_s = \frac{T}{2\pi r^2 L}$$

where:

S_s = Shear stress at radius r, psi

T = Torque, lb-in

r = Any radius, in

L = Length of elastomer, in

The maximum shear stress will occur at the inner radius of the rubber section where $r = r_i$ and is:

$$S_{s(max)} = \frac{T}{2\pi r_i^2 L}$$

The entire elastomeric section can be considered as a continuous shear sandwich without ends and being composed of thin elemental, cylindrical rings of thickness d_r. As the radial thickness of an elemental ring is reduced, the curvature becomes negligible when measured in terms of the thickness d_r, and the similarity to a flat sandwich becomes increasingly apparent. There is no evidence that any such cylindrical ring has any knowledge of, or is influenced by, adjacent elemental rings. Each thin cylinder behaves as though it were the only elastomeric section involved.

For the isolated element shown in Fig. 8, the relative angular deflection contributed by the elemental ring for small deflections $d\theta$ is:

$$d\theta = \frac{d_r \tan \phi}{r}$$

Also, the small angle ϕ determines the distortion of the element and is called the shearing strain. This shearing strain is proportional to the shear stress and inversely proportional to the modulus of rigidity or shear modulus as follows:

$$\phi = \frac{S_s}{G} = \frac{T}{2\pi r^2 LG}$$

where:

G = Shear modulus of rubber, psi

Then by substitution:

$$d\theta = \frac{1}{r} \tan \left[\frac{T}{2\pi r^2 LG} \right] d_r$$

A tangent function can be expressed as a series solution as follows:

$$\text{Tan } X = \left[X + \frac{X^3}{3} + \frac{2X^5}{15} + \ldots \right]$$

Since this series converges rapidly, a reasonable approximation can be made by using the first term only. Using this simplified solution, the angular deflection or torsional windup is:

$$d\theta = \frac{T d_r}{2\pi r^3 LG}$$

$$\theta = \frac{T}{2\pi LG} \left[-\frac{1}{2r^2} \right]_{r_i}^{r_o}$$

$$\theta = \frac{T}{2\pi LG} \left[\frac{1}{2r_i^2} - \frac{1}{2r_o^2} \right]$$

$$\theta = \frac{T}{4\pi LG} \left[\frac{1}{r_i^2} - \frac{1}{r_o^2} \right]$$

where:

θ = Angular deflection, rad

r_i = Inner radius of rubber, in

r_o = Outer radius of rubber, in

The torsional spring rate in lb-in/rad per radian is:

$$k_T = \frac{T}{\theta} = \frac{4\pi LG r_i^2 r_o^2}{r_o^2 - r_i^2}$$

In order to apply the previous equations, design values must be established for the shear stress and shear modulus. Maximum shear stress values in the elastomeric torsional spring can be determined by analyzing its dynamic and static capacity relative to the torque to be transmitted in the intended application.

Torsional Loading Characteristics

Type A Applications — In applications where the rubber element tube is used in a driveshaft between the primary power source and a driven member, the maximum continuous torque output of the power source is generally used as the design parameter for determining the rubber element tube capacity. This procedure assumes that continuous operation at torsional resonance can be avoided. The system is usually tuned below the idling and above the cranking speed of the engine. In installations where torsional resonance occurs above the operating range, a conventional tubular driveshaft may suit the application since a higher spring rate would be needed for tuning the system above the operating range.

Type B Applications — The rubber element tube can also be used in applications where it is subjected to torque multiplication of the power source, such as in a driveshaft located between the transmission and rear axle of a typical vehicle. The maximum computed torque is considered to be the maximum installed net engine torque multiplied by the transmission lowest (highest numerical) gear ratio, and the transmission efficiency factor. If

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the vehicle is equipped with a torque converter driven transmission, the torque multiplication due to the converter must also be included.

In addition, shock loads of significantly higher value can be imposed on the driveshaft depending on the type of application. Maximum torsional loads which can be developed by various loading modes in a driveshaft are discussed in the chapter on *Basic Driveline Requirements*. Since the driveshaft must be capable of safely transmitting the maximum applied torque, appropriate shock factors must be used in determining the ultimate strength design capability of the rubber element tube.

For a given application, a rubber element tube design must have adequate fatigue life. To determine minimum fatigue strength, a multiplied torque value is usually used which is representative of the tractive effort or wheel slip torque in the application. The severity of a particular application will determine the actual torsional load level to be used in evaluating the fatigue life characteristics of a proposed rubber element tube.

Operating Speed

The required maximum operating speed of the driveshaft will also affect the design of the rubber element tube. As previously described, the maximum operating speed can determine whether the molded elastomeric, or the compressed multiple elastomeric ring, rubber element tube may be used. In general, the molded elastomeric section design is limited to the relatively lower operating speed ranges. For relatively high speed applications, such as passenger car drivelines, the compressed multiple elastomeric ring design is usually required because it can provide the necessary transverse stiffness when located near the driveshaft end. This transverse stiffness produces a minimum effect on critical speed and permits dynamic balancing of a shaft which is required for relatively high speed applications.

In comparison with a conventional tubular driveshaft for a particular vehicle application, the critical speed of a rubber element tube design will be reduced because of its nonhomogeneous section characteristics. This critical speed can be determined by laboratory test or by computer analysis. In passenger car applications, the reduction in critical speed has been in the approximate range of 100-200 rpm for typical one piece, rubber element tube driveshafts which were required to perform within the same clearance envelopes as conventional driveshafts.

For typical steel driveshaft tubing, a modulus of elasticity of 29 000 000 psi and a density of 0.283 lb/in^3 have generally been used in critical speed computations. For rubber, however, these characteristics will vary considerably depending on the specific elastomer being employed. For example, for small deformations the modulus of elasticity of typical natural rubber varies from 150 psi at 30 Shore A durometer hardness to 500 psi at 70 Shore A durometer. The density of natural rubber ranges from 0.037 lb/in^3 at 30 durometer to 0.047 lb/in^3 at 70 durometer. At 300% elongation, the modulus of elasticity of natural rubber varies from approximately 700-2200 psi in the 50-70 durometer range. The modulus of elasticity achieved in an elastomer is dependent not only on compounding characteristics, but also on other considerations such as stress and temperature.

These values for density and the latter values for modulus of elasticity have been used with satisfactory results in determining the critical speed of ring type rubber element tube driveshafts with a computer. When using the widely accepted computation methods stated in the chapter on *Critical Speed*, it has been found that the relatively large variations in density and modulus of elasticity of the elastomer have little effect on the critical speed.

Inner and Outer Tube Sizes and Wall Thicknesses

The outside diameters and wall thicknesses of the inner and outer tubes are based on the required application parameters as well as on compatibility with the universal joint size employed. Initially, an inner tube of a given outside diameter and wall thickness is selected which will safely transmit the maximum applied torque for the application and also permit its attachment to the universal joint size used. In general, components of a given universal joint size, such as typical Cardan joint weld or tube yokes are available for attachment to tubes of different sizes and wall thicknesses.

The outer tube size and wall thickness selection is based not only on torque capacity and universal joint size compatibility, but also on a satisfactory critical speed level which will permit the driveshaft to perform safely at its intended maximum operating speed. In addition, the annular space between the inner and outer tubes and its length must be sufficient to permit the incorporation of a rubber element which will meet the required application parameters.

The dimensional tolerances associated with conventional driveshaft quality welded steel tubing are usually used in rubber element tubes. Typical tolerances for the more commonly employed tube sizes are shown in the chapter on *Tubing*. For the ring-type rubber element tubes, the outside diameter of the inner tube and the inside diameter of the outer tube, as well as tube ovality, are quite critical in achieving proper assembly fits, rubber compression, and the resulting torsional spring rate.

Allowable Shear Stresses

For a given application, the selection or requirement of either the molded elastomeric section or the compressed multiple elastomeric ring construction will determine the permissible stress level which can be used in designing the rubber element tube. Laboratory tests have shown that the bond strength at the inner radius is the critical design stress limitation of a rubber element tube. For design computations, allowable stress levels have been determined which permit an adequate fatigue life, as well as the necessary ultimate torsional strength.

Molded Rubber Element Constructions — A typical shear stress ranging from 100-150 psi at the maximum applied torque is generally employed in the design computations for molded rubber element tubes that are used for the previously described type A applications. This stress level is applied at the outer surface of the inner tube and has provided adequate fatigue life for those applications having relatively moderate torque, low speeds, and large amplitudes of vibrations.

Compressed Ring Rubber Element Constructions — A typical shear/compression stress ranging from 450-500 psi at the maximum applied torque is generally employed in the design

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computations for the ring type rubber element tubes that are used for the previously described type B applications. This higher stress level at the outer surface of the inner tube is permissible due to the compression of the elastomeric rings between the inner and outer tubes. For example, in order to transmit the maximum applied torque without slip between the rings and the outer tube, and operate at the relatively high rotational speeds found in passenger car drivelines, the compression is usually in the range of 40-49%.

In addition, vehicle driveshafts must be capable of compensating for variations in length due to suspension travel. This length change is generally accomplished with sliding or slip splines. When this end movement occurs during high torsional load applications on the driveshaft, the resulting axial thrust forces produced by the slip spline may be relatively high. These thrust forces must be resisted by the rubber element design and hence the compression of the elastomer must be sufficient to prevent slipping between the rings and the outer tube.

Depending on the specific requirements, other applications may permit a lower compression level of the elastomer and provide satisfactory function and durability. Also, the geometric proportions of the rubber element design necessary in certain applications, can influence the optimum compression level to be employed. This percent compression can be computed as follows by referring to Fig. 1:

$$\text{Percent compression} = \left(\frac{\text{Free elastomer section thickness} - \text{Compressed elastomer section thickness}}{\text{Free elastomer section thickness}} \right) 100$$

The percent compression used should be sufficient within practical limits to prevent slipping of the outer tube relative to the rings when the rubber element tube is operating under varying environmental conditions encountered in a particular application.

Fatigue Life

The rubber element tube must be capable of functioning satisfactorily in an application for its expected service life. It is possible to estimate its fatigue life at various stress levels with an S-N curve developed from life test data. Fig. 9 illustrates such an approximate load-life curve for a typical rubber element tube. The actual load-life relationship can vary considerably because of the following variables.

Frequency of Torsional Load — If the torsional load cycles are applied at a relatively high frequency, the temperature will increase due to the hysteresis characteristics of the elastomer with a resulting loss in fatigue life. Hence, the load cycles are generally slowed down in a test to keep the temperature rise at reasonably low levels. The time element is quite important in the application of the load and in the recovery of the rubber from the effect of the loading. The load cycle frequency employed depends on the design characteristics of a given rubber element tube and the application requirements. Load cycle frequencies in the typical range of 3-10 cpm have been used with satisfactory results.

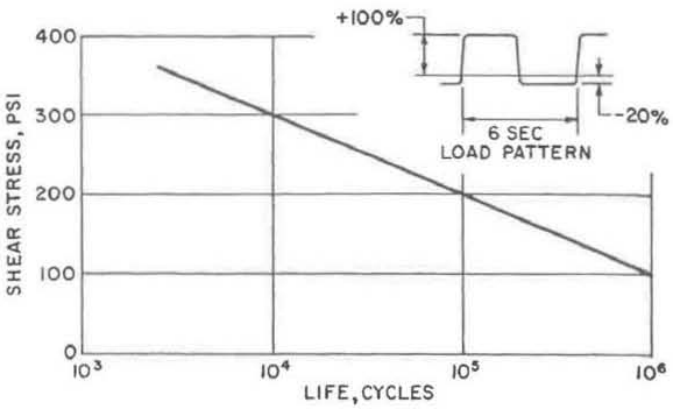


Fig. 9 — S-N curve for rubber element tube

Torsional Load Pattern — The shape of the load-time cycle affects the overall life of the rubber element tube. Maximum fatigue life is obtained with low frequency load and unload cycles.

In a fatigue test for a conventional driveshaft tube, the fully reversed torsional load cycle is commonly employed. This type of loading characteristic is undesirable for a rubber element tube. When a rubber element tube is twisted with a torsional load and then unloaded, not all of the stored energy is released. This unrecovered energy is lost as heat because of the hysteresis effect. Also, during the time that the load is applied, creep occurs in the elastomer and increases with temperature. Also, most of the creep occurs shortly after application of the load. If the rubber element tube is allowed to remain unloaded, it will gradually recover from a portion of this displacement. The unrecovered creep is referred to as permanent set.

In order to determine whether slipping or permanent set occurs in the rubber element tube, the torsional loading cycle is generally performed by applying the major load in the forward drive direction and then unloading by applying a relatively small load in the reverse direction. As shown in Fig. 9, this load pattern approximates a square load wave in both directions of rotation. The loads are usually applied for the same period of time in both directions. The load-time cycle employed depends on the design characteristics of a given rubber element tube and the application requirements. Load patterns consisting of 100% load in the forward direction and typically 15-20% in the reverse direction have been used with satisfactory results.

Rubber Properties — The hardness and strength of the elastomeric material, its bond strength at the point of maximum stress, and the amount of damping, all have an effect on the fatigue life of a rubber element tube. Rubber materials with comparatively low hysteresis and creep characteristics are desirable.

Shear Modulus

It was previously shown in the analysis of an elastomeric torsional spring that the shear modulus is a function of the shearing strain which occurs when the elastomer is distorted in shear. Typical values of shear modulus for rubber have been obtained by

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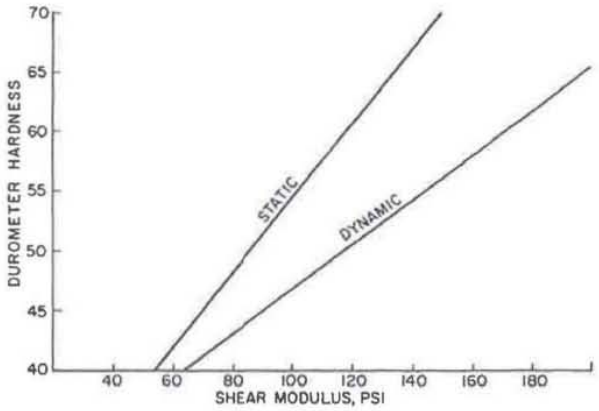


Fig. 10 — Shear modulus characteristics for molded rubber element tubes

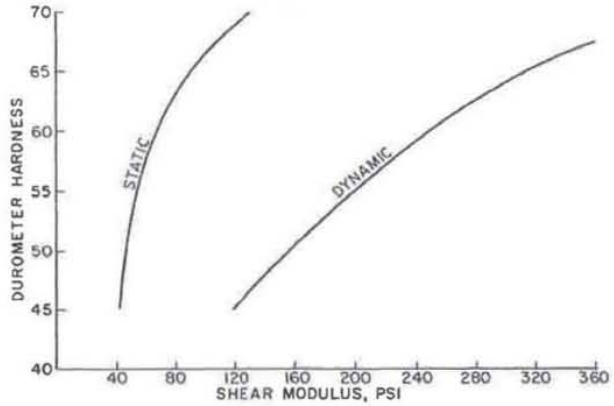


Fig. 11 — Shear modulus characteristics for compressed ring rubber element tubes

empirical means and they can vary considerably depending on the physical properties of the elastomer. These shear modulus values are quite useful for preliminary computations where it is necessary to estimate the torsional spring rate of a proposed rubber element tube design.

Experimentally derived data has shown that the shear modulus increases with increasing durometer hardnesses of the elastomer. Thus, when changing the durometer hardness of a given rubber element tube, the torsional rate is approximately proportional to the ratio of the shear moduli corresponding to the two durometers. The ability to change the elastic characteristics of rubber without altering the dimensions of the component is of great value in tuning a proposed design for an application.

Fig. 10 shows the relationship between the durometer and typical static and dynamic shear moduli for molded rubber element tubes, while Fig. 11 shows a similar relationship for compressed ring, rubber element tubes. Due to the inherent characteristics of an elastomer, the dynamic shear modulus is greater than the static modulus and, therefore, the dynamic torsional rate of a rubber spring is higher than its static rate. In addition, the ratio of the dynamic to the static shear modulus shows a progressive increase with an increase in rubber hardness. This progressive increase of the static to dynamic modulus ratio is more pronounced for ring type than for molded rubber element tubes. The primary importance of the dynamic shear modulus in computations is to establish the natural frequency of the rubber element tube design. With respect to static shear modulus, a value of less than 54 psi (approximately), is generally not recommended for molded rubber torsional springs.

Torsional Spring Rate

The rubber element tube is a torsionally elastic spring, specially designed for a given application. In a simple cylindrical torsion spring such as illustrated in Fig. 7, the spring rate is directly proportional to the rubber length and its shear modulus.

As previously described, the rubber element tube can be designed either in a compressed multiple elastomeric ring, or in a molded elastomeric section construction. Further, the rubber element tube can also be designed, if desired, in either a parallel or a series torsional spring loading arrangement, or in a combination of these two arrangements. Therefore, various spring rate possibilities can be obtained with these constructions and loading arrangements, and their usage depends on the basic driveshaft parameters specified for an application.

When the parallel loading arrangement is employed in a rubber element tube, as shown in Fig. 1, the spring rate is directly proportional to the effective length of the rubber. Hence, when comparing two ring type rubber element tubes, for example, which are identical in design except for the number of equal size rings used in a parallel loading arrangement, the rate of these two torsional springs varies directly with the ratio of the number of rings.

Also, when the durometer hardness of the elastomer is increased in any given rubber element, the spring rate is increased. When changing the rubber hardness from one durometer value to another in a typical design, the rate is proportional to the shear moduli corresponding to these two durometer values.

Ring type rubber element driveshafts have been generally used in certain passenger car applications where increased torsional flexibility was desired over that attainable with a conventional tubular shaft. By using the commonly employed parallel loading arrangement in the rubber element tube design of one-piece driveshafts, for example, typical torsional rates in the relatively wide range of 40-115 lb-ft/deg have been used with satisfactory results in American cars. This torsional rate range has been achieved by utilizing a minimum of two, to as many as approximately eight rings. The actual number of rings used in a given rubber element tube design is determined not only by the desired torsional flexibility, but also by the various other required application parameters for the driveshaft.

By comparison, the torsional rates of typical one piece conventional tubular driveshafts for passenger cars have been in the range of 120-200 lb-ft/deg. This relatively wide range of torsional rates results not only from the various diameters, wall thicknesses, and lengths of tubing employed or required, but also from the type of universal joint design used at each end.

When lower torsional rates have been required in passenger car rubber element tube driveshafts than those obtainable with the common parallel loading arrangement, the combined parallel-series loading arrangement used in the double ended rubber element tube design shown in Fig. 2 has provided rates as low as approximately 10 lb-ft/deg. This parallel-series loading arrange-

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ment results in increased torsional flexibility, but with a reduced torsional load capacity for the driveshaft, and has been generally employed in relatively small size and lower powered European cars.

Usually two rings are used at each end in a parallel loading arrangement, as shown in Fig. 2, to provide the necessary transverse stiffness. These parallel loaded springs are connected to each other in a series loading arrangement and, hence, produce a total torsional spring rate which is less than either of the parallel loaded sections.

Material Selection

The selection of optimum materials for the rubber element tube is based on ultimate strength, fatigue strength, torsional resilience, and other functional and durability requirements for an application, as well as economical processing advantages.

Elastomer

Because of its desirable inherent properties, natural rubber is generally the preferred elastomeric material for the rubber element. However, certain synthetic rubbers can be used when they are properly compounded to provide properties similar to natural rubber. Also, these synthetic elastomers can be compounded to provide special properties to meet the requirements of a specific application. For example, certain environmental conditions may prohibit the use of natural rubber.

Natural rubber with its relatively low internal hysteresis provides an adequate hardness range as well as excellent tensile strength, elongation, vulcanizing properties, adhesion to metals, hot and cold rebound ability, and good compression set. In addition, natural rubber provides excellent resistance to abrasion and creep and good resistance to tearing, heat, and heat aging.

The compounding of elastomeric materials used in rubber element tubes is usually done in order to develop specific physical properties, such as hardness, and to permit its processing in manufacturing equipment. This material hardness is generally in the typical range of 40-70 Shore A durometer. A practical tolerance of plus or minus five points on the durometer hardness requirement is usually allowed for compounding batch variations. Since the hardness is a gage of the rubber stiffness, its value is extremely important in achieving the torsional windup or spring rate desired in a given design.

Methods commonly used for checking rubber hardness have often been very inaccurate. It has been common to find a lack of agreement by as much as five points on the same elastomer sample by various individuals using different durometers. Unfortunately, a five point variation in hardness can be equivalent to 15-20% in actual stiffness of the rubber as measured in a compression deflection test. Consequently, a variation in the actual torsional spring rate of the rubber element tube will result. In order to prevent an excessive and undesirable variation in the spring rate of ring type rubber element tubes used in passenger cars, for example, the tolerance on the durometer requirement has been generally held to a typical plus or minus three points.

Tubing

In general, the inner and outer tubular members of the rubber element tube are made of typical driveshaft quality welded steel tubing. This tubing, which is described in detail in the chapter on *Tubing*, is made from cold rolled, plain, low carbon sheet or strip stock suitable for welding and has a typical hardness range of R_b 80-95.

The outer tube hardness of the ring type rubber element tube is particularly critical. A low hardness will cause the outer tube to yield over each ring during assembly. This yielding is especially noticeable after the tube has been pressed up for a period of time. Yielding of the outer tube due to this "push-up" will result in failure of the assembly to meet the ultimate torque requirement.

Processing and Assembly Procedures

Depending on the application requirements, the rubber element tube can be either a typical compressed multiple elastomeric ring, or a typical molded elastomeric section design. Certain rubber element tube components, because of their construction or design characteristics, are in most cases purchased from molded rubber product manufacturers. In the molded elastomeric section design, the rubber is molded to a specific geometric shape between an inner and an outer tube. In the commonly used ring type rubber element tube, the driveshaft manufacturer may purchase only the elastomeric rings and then assemble the rubber element tube in its entirety, or the inner tube with the attached rings may be purchased. In general, these rings are bonded in place on the tube outer surface with adhesive or they are sometimes molded or vulcanized in place. This multiple ring and inner tube member is then assembled by "press-up" or "push-up" into an outer tube to form the rubber element tube. Since the ring type rubber element tube with bonded natural rubber rings is most commonly used, its processing, assembly, testing, and quality control monitoring procedures are described in greater detail.

A thermosetting adhesive, which is usually an epoxy base type, is used to bond the rubber rings to the inner tube outer surface so that the rings remain in position while the inner member is assembled by push-up into the concentric outer tube. The adhesive must be capable of meeting the torsional strength requirement of the intended application over a typical temperature range of $-40 - +190°F$.

In order to provide a bonded rubber ring and inner tube member which will permit the rubber element tube to meet the application requirements, the mating surfaces should be properly designed and certain processing or cleaning procedures must be employed so that these surfaces are free of all detrimental contamination. In the case of the rubber rings, the molded surface which is to be bonded should be of such a design and surface texture to insure a thin, even adhesive film between the rubber and the tube. Because intimate contact of the mating surfaces is essential during the curing period, the inside diameter of the rubber rings must be sufficiently undersize so that no movement will occur during cure of the adhesive. If the adhesive is to be cured at ambient temperature, a typical interference fit of 0.010-0.020 in has been

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adequate, while an approximate 0.050 in interference fit has been satisfactory in providing for thermal expansion of the rubber during an oven cure process.

Since the bonding surface of rubber rings must be completely free of all contamination such as dust, dirt, oil, mold release agent, and so forth, the rings are generally cleaned and prepared for adhesive bonding by a chlorination process. The bonding surface of the inner tube must also be free of contamination such as dust, dirt, oil, grease, and so forth. Typical processing procedures which can prepare a satisfactory metal surface for adhesive bonding are sand or shot-blasting, alkaline or chlorothene degreasing, and phosphatizing.

The adhesive is applied in a thin and even coat of approximately 0.002-0.005 in thickness to the cleaned tube surface. The cleaned rubber rings are then placed in position on the tube. This assembly procedure is done in such a manner that no adhesive is wiped from its position on the tube and that intimate contact of the mating surfaces is ensured. The adhesive may be cured either in a typical ambient temperature range of 60-100°F for 24 h, or oven cured at a typical temperature of 255°F for one h. The ultimate strength of the bond is reached upon completion of either curing procedure.

In general, the as molded outside diameters of the rubber rings are used when the bonded multiple ring and inner tube member is assembled in the outer tube. Sometimes the ring outside diameters are ground to reduce eccentricity between the ring outer surface and the inside diameter of the inner tube. The effect of this eccentricity in the inner bonded assembly is amplified when the outer tube is assembled and may result in driveshaft balancing problems. The grinding procedure also permits some degree of correction so that the rubber element tube will meet the torsional spring rate requirement. In addition, the grinding process can be used to remove the mold release agent and flash rubber from the ring outside diameters when the rings are molded to the inner tube. The grinding of the ring outside diameters, however, can be detrimental in that the rubber element tube may not meet the ultimate torque and axial thrust load requirements of an application.

The outer tube must have the ends chamfered or deburred on the inside diameter to aid in entry of the rubber rings. Also, prior to the push-up operation where the outer tube is pressed over the multiple ring and inner tube assembly, the inner surface of the outer tube is cleaned in the area which will contact the rubber rings. Usually, a typical washing process provides an adequately clean surface.

The assembly procedure of ring type rubber element tubes is generally accomplished with a typical horizontal hydraulic press. The outer tube is restricted from moving endwise by suitable fixtures. On the end where the multiple ring and inner tube member is to be inserted, a funnel or guide fixture with a tapered inner surface is positioned concentric with the outer tube. The large end of this tapered surface will accommodate the entry of the rubber rings. The conical taper guides, and gradually compresses the rubber rings to a diameter which is slightly smaller than the inside diameter of the outer tube.

To aid in the push-up operation, the multiple ring and inner tube assembly is usually rolled over a felt pad which is saturated with a special process lubricant. This procedure lubricates the outer surface of the rings with a film of lubricant which is a mineral oil of relatively low viscosity such as SAE 10W with special additives. Because of their detrimental effects, ordinary lubricating oils are never used to facilitate the assembly procedure.

During the push-up procedure, the film of lubricating oil oxidizes due to friction and/or heat from compressed air, or it is wiped off the relatively smooth molded outer surface of the rings. When the outside diameters of the rings are ground as previously described, small amounts of lubricant may become trapped in the roughened outer surfaces during press-up. The presence of this lubricant can result in the rubber element tube not being able to meet its functional design requirements.

In the push-up procedure, a hydraulic ram with a suitable nose piece forces the inner tube assembly through the funnel and into the outer tube for a controlled distance. Because of the compression imposed on the rubber rings, they are distorted and hence the inner tube requires an "over-push". The ram is then retracted and another ram moves into the outer tube from the opposite direction and back pushes or back presses the inner tube so that the rubber is in a neutral position. This "back-up" operation avoids a telescoping effect of the rubber element tube since the stretching of the elastomer is eliminated. Rubber element tubes which are not back pushed can be expected to "grow" in overall length by approximately 0.19 in within 12 h after push-up. It is rarely possible to re-push a rubber element tube to a new length unless it is done immediately after the initial push-up operation.

Testing and Quality Control Monitoring Procedures

Because of the inherent characteristics of ring type rubber element tubes, certain testing and quality control monitoring procedures are necessary. These procedures are performed to initially substantiate or qualify design computations relative to torsional spring rate, load capacity, and fatigue life. Then during manufacturing, these design parameters must be audited and maintained with quality control monitoring procedures.

The static torsional spring rate and load capacity tests are performed in a typical torsion test machine. For a given rubber element tube driveshaft, the static spring rate is generally specified as an angular deflection range at a given torsional load for practical reasons. Usually, the procedure is to initially preload the driveshaft with a relatively low torsional load. Then the given torque load is generally applied and released three times at a relatively slow constant loading and unloading rate. The time element is important in the application of stress and in the recovery of the rubber spring from the effect of stressing.

The angular windup at the given torque load is recorded for the third load application and must be within the angular deflection range specified for that torsional load, less the permanent set. Fig. 12 illustrates a typical torsional load versus angular deflection curve for a static torsional rate test performed on a rubber element tube driveshaft. The three load and unload cycles show the

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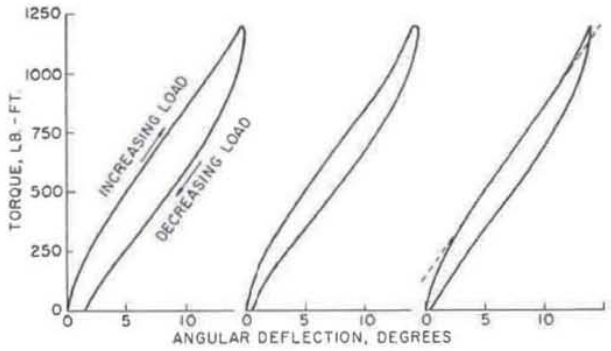


Fig. 12 — Rubber element tube, static torsional rate test

hysteresis characteristics of a typical rubber spring. The area enclosed in each loop represents the energy lost through hysteresis and also is a measure of the elastomer's damping characteristic. It can be seen that for the same torsional load, the unloading strains do not equal the loading strains. The creep that is thus indicated would become smaller if the rubber had been given time to recover. The unrecovered creep is referred to as permanent set.

Fig. 13 shows a typical torsional load versus deflection curve for a static torsional load capacity test conducted on a rubber element tube driveshaft. In this test, both the ultimate and JAEL yield torque loads are determined as well as the driveshaft failure mode. In the case of this specific driveshaft, the inner tube of the rubber element tube assembly failed, by collapsing, as noted on the curve.

Another test procedure which is used to verify design computations is a torsional fatigue test. The fatigue test parameters employed will vary with the design and the application intent of the driveshaft. Previously, it was described how fatigue life is affected to a considerable degree by such variables as the frequency of the torsional load and the shape of the load-time cycle used in the test.

Since the functional and durability characteristics of rubber element tubes are quite sensitive to variations in the elastomer properties, the rubber rings themselves are generally subjected to special quality control monitoring procedures. Variation in rubber properties can occur from one compounded batch to another. Hence, the molded rubber product manufacturer certifies that the rubber rings in a given shipment meet the compound specification. The driveshaft manufacturer, besides inspecting the rings for surface condition and dimensional requirements, also maintains a quality control audit procedure on the durometer hardness and compression set. Although the durometer check does not always correlate exactly with spring rate, it does serve as a warning that a spring rate change may be evident when a durometer hardness change occurs.

Because the rubber rings are under compression in the ring type rubber element tube, a certain maximum level of compression set must not be exceeded when the check is conducted in a specified manner. Excessive permanent set will result in the rubber element not meeting certain design parameters such as the torque load requirement, for example.

For the multiple ring and inner tube assembly, the common

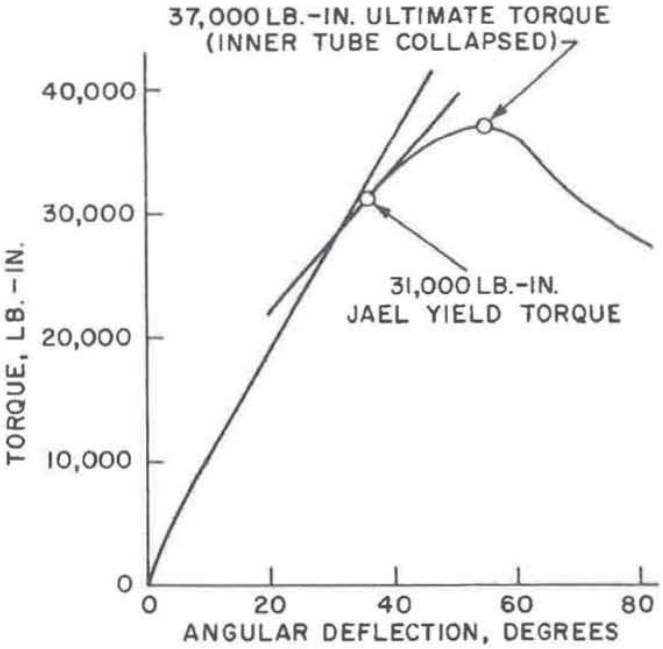


Fig. 13 — Rubber element tube, static torsional load capacity test

practice has been to bond the rings to the tube with an adhesive as previously described. The integrity of this bond must be verified after curing and before assembly of the outer tube. The bond must be capable of withstanding a load producing a specified minimum shear stress which in general has been approximately 300 psi. The ring and inner tube assembly is checked in a fixture which supports a ring so that when an axial load is applied to the tube, the bonded area is subjected to shear loading. Upon application of a load which produces a stress higher than the specified minimum shear stress, the adhesive should shear from the tube or the rubber ring should be torn with some rubber adhering to the tube. If these failure characteristics do not occur, the rubber element tube will not meet the fatigue test requirements and is thus unacceptable.

Applications

The rubber element tube as an integral component of the driveshaft provides added torsional flexibility. This desirable characteristic offers the following advantages for the applications described:

1. The torsional flexibility provides a means to avoid a torsional resonant condition. Typical applications for this feature are:
   a. Driveshaft connection between engine flywheel and dynamometer of engine test dynamometers.
   b. Driveshaft connection between engine and transmission of vehicle midship-mounted transmissions.
   c. Compressor, pump, and fan drives, as well as other drive systems where nonuniform rotating motion occurs.

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2. The torsional flexibility provides an amplitude limitation on impact loads such as starting and stopping inertia loads, rough shifts, and sudden changes in torque which cushions, modifies, and smooths out the transition period. Typical applications for this feature are:
   a. Agricultural implements
   b. Rotary mowers
   c. Construction equipment
   d. Industrial machines and equipment
3. The torsional flexibility provides cushioning and serves as an insulator which tends to prevent clattering, rattling, and contact noises on components with backlash. Typical applications for this feature are:
   a. Isolation of drivetrain disturbances in passenger cars, trucks, and construction equipment, as well as other drive systems.
   b. Any driveshaft acting as a resonator for driving and driven member disturbances in the drivetrain.

Application Problems

Application Problem 1

A new model prototype passenger car has torsional excitation emanating from the rear axle. A conventional plain steel tubular driveshaft connecting the transmission with the rear axle differential is in resonance with the exciting source and thereby amplifies the disturbance. Underbody clearances do not permit the use of a tuned torsional damper, flexible coupling, or other similar device. Due to space limitations, it appears that only a torsional spring such as a rubber element tube can be incorporated in the driveshaft. The design requirements for a suitable rubber element tube are computed based on the following passenger car application parameters:

Engine—Type — V8, four stroke spark ignition
Maximum installed net torque — 320 lb-ft
Maximum speed — 4300 rpm
Transmission—Type — three speed automatic with torque converter
Maximum torque converter ratio — 2.00
Low gear ratio — 2.52
Vehicle Weight—Curb weight — 3600 lb
GVW weight — 4500 lb
Driveshaft—Rubber element tube torsional spring rate — 1.42-2.08 deg/100 lb-ft or 48-70 lb-ft/deg (59 lb-ft/deg mean torsional rate desired).

Based on tests performed with other passenger cars having a similar power train arrangement and the same approximate weight, a predicted shock factor of 1.60 is assumed in computing the maximum developed driveshaft torque. This shock factor was determined for fully loaded vehicles and disregards drivetrain component efficiencies. Hence, the maximum theoretical engine developed stall torque and the predicted shock torque in the driveshaft for this vehicle are:

$$T_{\text{theoretical stall}} = 320 \times 12 \times 2.00 \times 2.52 = 19\,354 \text{ lb-in}$$
$$T_{\text{shock}} = 19\,354 \times 1.60 = 30\,966 \text{ lb-in}$$

Because of the high operating speed and torque requirements, a ring type rubber element tube is proposed for this application. To provide a satisfactory critical speed level, the tubing must be 3.25 in outside diameter × 0.065 in wall, swaged down to 2.75 in outside diameter at both ends in order to operate within the required underbody clearance envelope. Also, the rubber element is to be incorporated in the rear swaged portion of the tube. Hence, the proposed rubber element tube is to use 2.75 in outside diameter × 0.065 in wall outer tubing and 2.25 in outside diameter × 0.095 in wall inner tubing, in order to provide sufficient annular space for the rubber element.

For these proposed tubing sizes of the rubber element tube, the JAEL yield torques are computed as follows using a design shear stress of 33 000 psi. Further, from previous test data of these tube sizes, the corresponding estimated ultimate torque values are also listed.

Outer Tube: 2.75 in O.D. × 0.065 in wall

$$T = \frac{\pi (d_o^4 - d_i^4) S_s}{16\, d_o}$$

where:

T = Torque, lb-in

S_s = Shear stress, psi

d_o = Outer diameter, in

d_i = Inner diameter, in

hence:

$$T = \frac{\pi [(2.75)^4 - (2.62)^4] \times 33\,000}{16 \times 2.75}$$

T = 23 730 lb-in (JAEL yield)

T = 36 500 lb-in (ultimate)

Inner Tube: 2.25 in O.D. × 0.095 in wall

$$T = \frac{\pi [(2.25)^4 - (2.06)^4] \times 33\,000}{16 \times 2.25}$$

T = 21 946 lb-in (JAEL yield)

T = 33 760 lb-in (ultimate)

These torque values for the proposed tubing indicate that the JAEL yield torques are above the driveshaft maximum, theoretical stall torque, while the ultimate torques are safely above the predicted driveshaft shock torque level.

Figs. 14-16 illustrate the proposed ring type rubber element tube with appropriate nomenclature. This nomenclature is described as follows and is used in design computations and in the dimensioning procedure.

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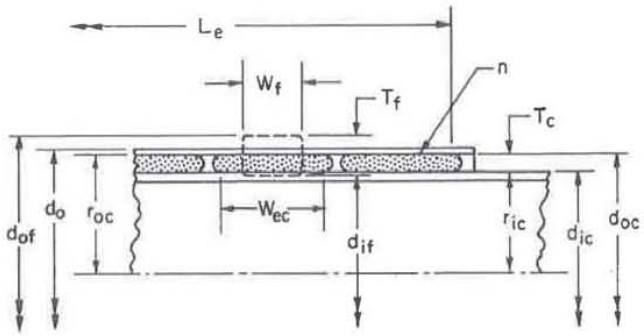


Fig. 14 — Nomenclature for ring type rubber element tube

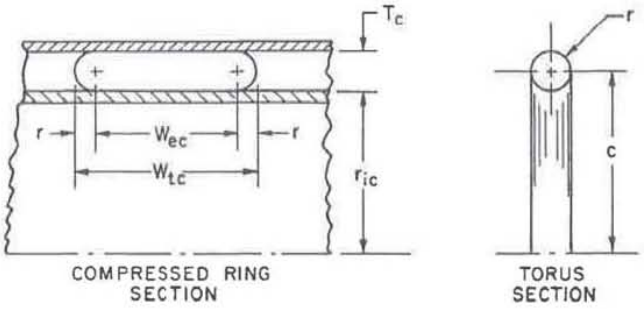


Fig. 15 — Bulging condition of compressed rubber ring

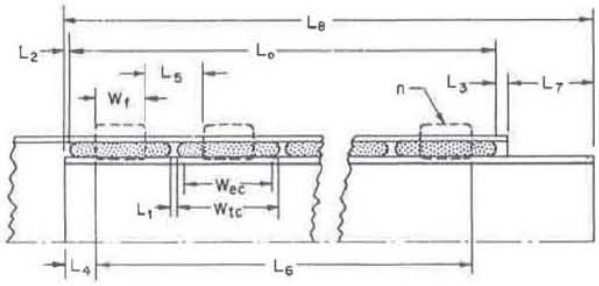


Fig. 16 — Nomenclature for ring type rubber element tube

NOMENCLATURE

L_o = Overall length of rubber element, in
L_e = Total effective length of rubber element, in
L_1 = Space between compressed rubber rings, in
L_2 = Distance from compressed rubber ring to inner tube edge, in
L_3 = Distance from compressed rubber ring to outer tube edge, in
L_4 = Distance from free rubber ring to inner tube edge, in
L_5 = Spacing distance of free rubber rings, in
L_6 = Overall spacing distance of free rubber rings, in
L_7 = Minimum recommended distance of outer tube edge to inner tube weld line, in
L_8 = Overall length of inner tube, in
n = Number of rubber rings
d_o = Outer tube outside diameter, in
d_{oc} = Outer tube inside diameter and also outer diameter of rubber element, in
r_{oc} = Outer tube inside radius and also outer radius of rubber element, in
d_{ic} = Inner tube outside diameter and also inner diameter of rubber element, in
r_{ic} = Inner tube outside radius and also inner radius of rubber element, in
d_{of} = Outside diameter of free rubber ring, in
d_{if} = Inside diameter of free rubber ring, in
r = Radius of torus section, in
c = Mean radius of torus ring, in
T_f = Thickness of free rubber ring, in
W_f = Width of free rubber ring, in
T_c = Thickness of compressed rubber ring, in
W_{tc} = Total width of compressed rubber ring (including bulge effect) in
W_{ec} = Effective width of compressed rubber ring, in
V_f = Total free rubber ring volume, in^3
V_b = Bulge volume of compressed rubber ring, in^3
V_n = Net volume of compressed rubber ring, in^3

Using this nomenclature in appropriate equations, the rubber element design can be determined in the following computation procedure:

The inner diameter of the rubber element is 2.25 in and therefore, the inner radius is:

$$r_{ic} = \frac{d_{ic}}{2} = \frac{2.25}{2} = 1.125 \text{ in}$$

The outer diameter of the rubber element is 2.62 in and, hence, the outer radius is:

$$r_{oc} = \frac{d_{oc}}{2} = \frac{2.62}{2} = 1.31 \text{ in}$$

The design shear stress at the inner radius of the elastomer at maximum applied torque is assumed to be 475 psi. Thus, the required total effective length of the rubber spring is:

$$L_e = \frac{T}{2 \pi r_{ic}^2 S_s}$$

where:

T = Torque, lb-in

S_s = Shear stress, psi

hence:

$$L_e = \frac{30\,966}{2 \pi \times (1.125)^2 \times 475}$$

$$L_e = 8.20 \text{ in}$$

Since the rubber element diametral limits are 2.62 in outer diameter and 2.25 in inner diameter, the compressed thickness of the rubber section is:

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$$T_c = \frac{d_{oc} - d_{ic}}{2}$$

$$T_c = \frac{2.62 - 2.25}{2} = 0.185 \text{ in}$$

The compression of the rubber rings is assumed to be 48% and, therefore, the free rubber ring section thickness is:

$$T_f = \frac{100\ T_c}{100 - 48}$$

$$T_f = \frac{100 \times 0.185}{100 - 48} = 0.356 \text{ in}$$

When a width to thickness ratio of 2.45 for the free rubber ring cross section is assumed, the free ring width is:

$$W_f = 2.45\ T_f$$

$$W_f = 2.45 \times 0.356 = 0.872 \text{ in}$$

Now it is possible to determine the volume of the free rubber ring. Since an ambient temperature cure process is to be used, a diametral interference fit of 0.015 in is assumed between the ring inner diameter and the inner tube outside diameter to prevent ring movement during cure. Therefore, the free ring inner diameter is:

$$d_{if} = d_{ic} - 0.015$$

$$d_{if} = 2.250 - 0.015 = 2.235 \text{ in}$$

and the free ring outer diameter is:

$$d_{of} = d_{if} + 2T_f$$

$$d_{of} = 2.235 + 2 \times 0.356 = 2.947 \text{ in}$$

Hence, the total free rubber ring volume is:

$$V_f = \frac{\pi\ [d_{of}^2 - d_{if}^2]\ W_f}{4}$$

$$= \frac{\pi\ [(2.947)^2 - (2.235)^2] \times 0.872}{4}$$

$$V_f = 2.527 \text{ in}^3$$

The volume of the free rubber and compressed rubber rings must be the same. Because the ends of the rings are unrestrained, a bulging condition will result when the ring is compressed into the outer tube as shown in Fig. 15. The total bulge volume for one compressed ring will be assumed to equal the volume of a torus with one half of the volume at each end of the cylindrical section. This assumption is sufficiently accurate for rings having compressions in the range of 40-49%. The cylindrical portion of the compressed ring between the bulge ends constitutes the effective length of the elastomer. The total bulge volume per ring can be computed with the following equation:

$$V_b = 2\pi^2 cr^2$$

The compressed ring section thickness is 0.185 in and the rubber element inner radius is 1.125 in. From these rubber element dimensions, the necessary torus dimensions are computed as follows:

$$r = \frac{T_c}{2} = \frac{0.185}{2} = 0.0925 \text{ in}$$

and:

$$c = r_{ic} + r = 1.125 + 0.0925$$

$$c = 1.2175 \text{ in}$$

hence:

$$V_b = 2\pi^2 \times 1.2175 \times (0.0925)^2$$

$$V_b = 0.206 \text{ in}^3$$

This bulge volume must be subtracted from the total free rubber ring volume as follows:

$$V_n = V_f - V_b = 2.527 - 0.206 = 2.321 \text{ in}$$

This net volume is the cylindrical portion of the compressed rubber ring which constitutes the effective width of the elastomer. Thus, the effective width of the compressed rubber ring is:

$$W_{ec} = \frac{4\ V_n}{\pi\ (d_{oc}^2 - d_{ic}^2)}$$

$$W_{ec} = \frac{4 \times 2.321}{\pi\ [(2.62)^2 - (2.25)^2]} = 1.64 \text{ in}$$

Therefore, the number of rings required in this rubber element tube is:

$$n = \frac{L_e}{W_{ec}} = \frac{8.20}{1.64} = 5.00 \text{ rings}$$

The desired mean torsional spring rate of the rubber element tube is 59 lb-ft/deg. Since 57.296 deg equal one rad, this spring rate expressed in lb-in/rad is:

$$k_T = 59 \times 12 \times 57.296$$

$$k_T = 40\ 566 \text{ lb-in/rad}$$

Based on this torsional rate, the static shear modulus of the elastomer is:

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$$G = \frac{k_T (r_{oc}^2 - r_{ic}^2)}{4 \pi L_e r_{ic}^2 r_{oc}^2}$$

where:

G = Shear modulus, psi

k_T = Torsional spring rate, lb-in/rad

r_{oc} = Outer radius of rubber element, in

r_{ic} = Inner radius of rubber element, in

L_e = Effective length of rubber element, in

$$G = \frac{40\,566\,[(1.31)^2 - (1.125)^2]}{4 \pi \times 8.20 \times (1.125)^2 \times (1.31)^2} = 81.65 \text{ psi}$$

In Fig. 11 at 81.65 psi static shear modulus, the rubber hardness is 63 durometer. The tolerance for the rubber ring hardness is assumed to be ± 3 durometer points which will meet the spring rate requirements of the application.

With the completion of the design computations, it is then possible to determine the remaining rubber element tube dimensions by referring to Figs. 15 and 16. When only two rings are used in a rubber element tube, the rings must be spaced far enough apart to provide adequate transverse stiffness, otherwise critical speed and balancing problems can result. In this application problem, however, five rings are required in the rubber element tube and, therefore, the transverse stability problem is not present. Hence, it is desirable to select dimensions which will result in a short and compact rubber element tube assembly and which is located as close as practical to the driveshaft end. If the rings are spaced further apart than necessary, the large overlap of the outer and inner tubes results in an excessively heavy driveshaft. This excessive weight would be detrimental to the shaft's critical speed.

The total width of the compressed rubber ring, including bulge effect is:

$$W_{tc} = W_{ec} + 2r = 1.64 + 2 \times 0.0925$$

$$W_{tc} = 1.825 \text{ in}$$

In general, the minimum space between compressed rubber rings is in the typical range of 0.040-0.125 in. For this rubber element tube, this space is assumed to be:

$$L_1 = 0.040 \text{ in}$$

Hence, the overall length of the rubber element is:

$$L_o = n W_{tc} + (n-1) L_1$$

$$L_o = 5 \times 1.825 + (5-1) \times 0.040 = 9.285 \text{ in}$$

The minimum distance from the compressed rubber ring to the inner tube edge is usually in the typical range of 0.100-0.125 in. For this rubber element tube, this distance is assumed to be:

$$L_2 = 0.100 \text{ in}$$

The minimum distance from the compressed rubber ring to the outer tube edge is generally in the typical range of 0.125-0.230 in. For this rubber element tube, this distance is assumed to be:

$$L_3 = 0.230 \text{ in}$$

Since the distance from the compressed rubber ring to the inner tube edge L_2 was assumed to be 0.100 in, the distance from the free ring to the inner tube edge is:

$$L_4 = L_2 + \frac{W_{tc}}{2} - \frac{W_f}{2}$$

$$L_4 = 0.100 + \frac{1.825}{2} - \frac{0.872}{2} = 0.577 \text{ in}$$

Also, since the space between compressed rubber rings L_1 was assumed to be 0.040 in, the spacing distance of the free rubber rings is:

$$L_5 = L_1 + W_{tc} - W_f$$

$$L_5 = 0.040 + 1.825 - 0.872 = 0.993 \text{ in}$$

Therefore, the overall spacing distance of the free rubber rings is:

$$L_6 = n W_f + (n-1) L_5$$

$$L_6 = 5 \times 0.872 + (5-1) \times 0.993 = 8.332 \text{ in}$$

The distance from the outer tube edge of the rubber element tube to the inner tube weld line is usually in the range of 1.00-2.87 in. The minimum recommended distance is 1.00 in, otherwise the rubber element could be damaged from the welding heat. For this rubber element tube, this distance is assumed to be:

$$L_7 = 2.87 \text{ in}$$

Hence, the overall length of the inner tube is:

$$L_8 = L_2 + L_o + L_3 + L_7$$

$$= 0.100 + 9.285 + 0.230 + 2.870$$

$$L_8 = 12.485 \text{ in}$$

This rubber element tube, as designed, can meet the application parameters and will operate in the same clearance envelope as the conventional plain steel tube. The incorporation of this rubber element tube in the driveshaft, however, results in a reduction of the shaft's critical speed. A computer analysis of both the conventional tubular driveshaft and the rubber element tube driveshaft

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indicates that the critical speed is reduced by 143 rpm. When a safe speed factor of 0.85 is used, the resulting safe operating speed is still safely above the maximum speed of 4300 rpm.

Application Problem 2

The drivetrain of a large vehicle consists of an engine and mid-mounted transmission. A typical solid shaft type of driveshaft mounted directly to the engine flywheel and the transmission input shaft by conventional means results in a torsional resonance within the normal operating speed range. Determine the feasibility of a rubber element tube driveshaft that will result in the torsional resonance occurring outside of the normal operating speeds of the vehicle when the following application parameters are given:

Engine—Type — Six cyl, four stroke Diesel
Maximum installed net torque — 850 lb-ft (10 200 lb-in)
Maximum speed — 2500 rpm
Idling speed — 500 rpm
Cranking speed — 150 rpm
Mass polar moment of inertia (including flywheel) — 40.50 lb-in-s^2

Transmission—Type — Hydraulic torque converter and planetary gear unit, midship mounted
Mass polar moment of inertia (effective on the transmission input shaft) — 22.20 lb-in-s^2

Installation Limitations—Length available from flange mounting face to flange mounting face — 16.50 in. Approximate maximum swing diameter clearance — 9.00 in.

The primary concern in this application is to avoid, as much as possible, torsional resonance within the normal operating range. An analysis of the existing driveshaft, indicates an 8.00 in swing diameter and a 16.50 in installed length. The solid shaft effective length is reduced to 13.50 in by allowing for an ineffective length due to the relatively stiff components of the universal joints. To provide an adequate safety factor for peak loads which are estimated at five to one and to obtain satisfactory fatigue life, the torsional shear design stress is to be 10 000 psi for the solid steel shaft. The required shaft diameter is:

$$D = \sqrt[3]{\frac{16T}{\pi S_s}}$$

where:

D = Diameter, in

T = Torque, lb-in

S_s = Shear stress, psi

hence:

$$D = \sqrt[3]{\frac{16 \times 10\,200}{\pi \times 10\,000}}$$

$$D = 1.73 \text{ in}$$

The torsional spring rate of this shaft is:

$$k_T = \frac{T}{\theta} = \frac{GI_p}{L_e} = \frac{\pi D^4 G}{32 L_e}$$

where:

k_T = Torsional spring rate, lb-in/rad

T = Torque, lb-in

θ = Angle of twist, rad

G = Modulus of rigidity or modulus of elasticity in shear, psi (Assume 12 000 000 psi for steel)

I_p = Area polar moment of inertia, in^4

L_e = Effective length, in

thus:

$$k_T = \frac{\pi \times (1.73)^4 \times 12\,000\,000}{32 \times 13.50}$$

$$k_T = 781\,685 \text{ lb-in/rad}$$

Since the drivetrain consists of two rotating inertias connected with a torsional spring, the natural frequency of this two mass system is:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_T (J_e + J_t)}{J_e J_t}}$$

where:

f_n = Natural frequency, (cps) Hz

k_T = Torsional spring rate of shaft, lb-in/rad

J_e = Mass polar moment of inertia of engine, lb-in-s^2

J_t = Mass polar moment of inertia of inertia of transmission, lb-in-s^2

hence:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{781\,685 \times (40.50 + 22.20)}{40.50 \times 22.20}}$$

$$f_n = 37.16 \text{ cps (Hz)}$$

also:

$$f_n = 37.16 \times 60 = 2230 \text{ cpm}$$

and thus the natural frequency of the system corresponds to 2230 rpm shaft speed.

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The firing frequency of the six cyl, four-stroke Diesel engine is three cycles per revolution. Therefore, torsional resonance occurs at one third of the system's natural frequency which is at 743 rpm. If two times resonance is assumed to be a desirable minimum condition, the system can be considered to be unsuitable for operation under 1486 rpm.

With one cylinder misfiring, resonance will occur at 4458 rpm. This condition is above the operating range and would result in a reasonably large amplitude transfer through the entire range. These resonance values cannot be considered satisfactory for this installation.

An ideal situation for this application would have a torsional resonant frequency below the 500 rpm engine idle speed and above the 150 rpm engine cranking speed, or in the vicinity of 300 rpm. This condition will permit engine starting, while quickly passing through the resonant speed to idle, without excessive stresses on any of the components.

Using the assumed resonance speed of 300 rpm, the rubber element tube design can be computed. Since the engine firing frequency is three cycles per revolution, the natural frequency of the system is:

$$f_n = \frac{3N_r}{60}$$

where:

N_r = Resonance speed, rpm

hence:

$$f_n = \frac{3 \times 300}{60} = 15 \text{ cps (Hz)}$$

and thus knowing the desired frequency, the required torsional spring rate can be determined as follows:

$$k_T = \frac{(2\pi f_n)^2 J_e J_t}{J_e + J_t}$$

$$= \frac{(2\pi \times 15)^2 \times 40.50 \times 22.20}{40.50 + 22.20}$$

$$k_T = 127\,375 \text{ lb-in/rad}$$

Since one radian equals 57.29578 deg, the spring rate can also be expressed as follows:

$$k_T = \frac{127\,375}{57.29578} = 2223 \text{ lb-in/deg}$$

or:

$$k_T = \frac{2223}{12} = 185.26 \text{ lb-ft/deg}$$

For practical reasons, the torsional spring rate of the rubber element tube driveshaft is generally specified as an angular deflection at a convenient torque load such as 100 lb-ft as follows:

$$\text{Torsional rate} = \frac{100}{185.26}$$

$$\text{Torsional rate} = 0.54 \text{ deg/100 lb-ft}$$

This spring rate is, therefore, required to provide the assumed 300 rpm resonance speed.

Within the installation's limitation on the swing diameter and an inner tube which is compatible with the size of the universal joints, the following rubber element proportions are assumed for trial computations:

Inner diameter = 4 in
Outer diameter = 8 in

Since the driveshaft is located between the power source and a driven member, and because of the moderate torque and relatively low speeds, a molded rubber element construction is proposed. Thus, the design shear stress is assumed to be 100 psi at 850 lb-ft maximum, installed net engine torque and a rubber hardness of 55 durometer is proposed. The required length of the rubber is:

$$L = \frac{T}{2\pi r_i^2 S_s}$$

where:

L = Length of elastomer, in

T = Torque, lb-in

r_i = Inner radius of elastomer, in

S_s = Shear stress at inner radius, psi

hence:

$$L = \frac{10\,200}{2\pi \times (2)^2 \times 100}$$

$$L = 4.06 \text{ in}$$

This length is the minimum length of rubber necessary to meet the application's torque requirement. The torsional spring rate of this rubber section is:

$$k_T = \frac{4\pi LG r_i^2 r_o^2}{r_o^2 - r_i^2}$$

where:

k_T = Torsional spring rate, lb-in/rad

L = Length of rubber, in

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G = Shear modulus of rubber, psi

r_i = Inner radius of rubber, in

r_o = Outer radius of rubber, in

Because the rubber element required for this application must be tuned to a given resonant speed, the dynamic shear modulus of the elastomer must be used in order to compute the dynamic spring rate. In Fig. 10 at 55 durometer, the dynamic shear modulus is 145 psi for a molded rubber section type of rubber element tube. Thus, the dynamic spring rate is:

$$k_T = \frac{4\pi \times 4.06 \times 145 \times (2)^2 \times (4)^2}{(4)^2 - (2)^2}$$

$$k_T = 39\,455 \text{ lb-in/rad}$$

or:

$$k_T = \frac{39\,455}{57.29578} = 689 \text{ lb-in/deg}$$

or:

$$k_T = \frac{689}{12} = 57.42 \text{ lb-ft/deg}$$

As previously explained, this torsional spring rate can also be expressed as follows for practical reasons:

$$\text{Torsional rate} = \frac{100}{57.42}$$

$$\text{Torsional rate} = 1.74 \text{ deg/100 lb-ft}$$

This torsional rate resulting from the assumed rubber spring proportions results in an excessive torsional windup. The natural frequency of the rubber element tube is:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_T (J_e + J_t)}{J_e J_t}}$$

$$= \frac{1}{2\pi} \sqrt{\frac{39\,455\ (40.50 + 22.20)}{40.50 \times 22.20}}$$

$$f_n = 8.35 \text{ cps (Hz)}$$

Since the engine firing frequency is three cycles per revolution, the resonance speed is:

$$N_r = \frac{8.35 \times 60}{3} = 167 \text{ rpm}$$

This resonance speed is just slightly above the cranking speed and also because the torsional windup is excessive, the computed rubber spring is not acceptable. A stiffer rubber element tube with less torsional windup is required to satisfy the assumed resonance speed of 300 rpm. In order to achieve this desired result, any or all of the following modifications to the elastomeric section can be made:

1. Reduce the rubber outer diameter
2. Increase the rubber length
3. Increase the durometer hardness and thus increase the dynamic shear modulus of the rubber.

Since in this case considerable reduction in torsional windup is necessary, all three of these rubber design parameters will be modified as follows:

$r_o = 3$ in

$L = 9$ in

$G = 155$ psi at 57 durometer from Fig. 10

therefore:

$$k_T = \frac{4\pi LG\, r_i^2\, r_o^2}{r_o^2 - r_i^2}$$

$$= \frac{4\pi \times 9 \times 155 \times (2)^2 \times (3)^2}{(3)^2 - (2)^2}$$

$$k_T = 126\,217 \text{ lb-in/rad}$$

or:

$$k_T = \frac{126\,217}{57.29578} = 2203 \text{ lb-in/deg}$$

or:

$$k_T = \frac{2203}{12} = 183.58 \text{ lb-ft/deg}$$

and for practical reasons this torsional spring rate can be expressed as follows:

$$\text{Torsional rate} = \frac{100}{183.58}$$

$$\text{Torsional rate} = 0.54 \text{ deg/100 lb-ft}$$

Hence, the correct spring rate is obtained with the revised rubber design parameters. Fig. 17 illustrates the geometric proportions of the molded rubber element tube that satisfies the application requirements.

The procedures and computations performed in this problem are directed towards a mid-mounted transmission application. However, the same procedures and analysis can be applied to a wide range of installations involving the same principles such as

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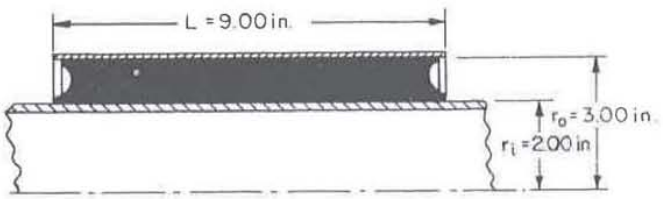


Fig. 17 — Molded rubber element tube

dynamometer, compressor, fan, and pump drives, and so on, where nonuniform rotating motion occurs and a torsional resonance frequency within the operating range is to be avoided.

The basic application parameters required to make an analysis are:

1. Installation length and swing diameter limitations
2. Power Source
   a. Type, number of cylinders and stroke
   b. Maximum torque
   c. Maximum and minimum speed
   d. Idling speed
   e. Cranking speed
   f. Mass polar moment of inertia including flywheel
3. Driven Member
   a. Type
   b. Mass polar moment of inertia, effective on the input shaft

Application Problem 3

The rotating mass of the drivetrain components in a typical agricultural implement is subject to angular accelerations and decelerations during operation of the implement. Some of these speed fluctuations are designed into the implement. For example, a flywheel is used to give up energy for compression of a bale in a typical hay baler. Other fluctuations are accidental, as would be the case if the baling mechanism jammed due to overload or malfunction.

Speed fluctuations of a moving mass indicate some type of energy transfer. The introduction of agricultural implements having larger rotating masses means that relatively larger amounts of this energy could be transferred through the driveshafts, gears, and other torque transmitting components of an implement's drivetrain. These larger amounts of transferred energy show up as higher torque peaks through the system. These torque peaks reach values higher than the nominal torque required for normal operation of the implement.

Torsional Impact — In cases of implement jamming, the kinetic energy of the rotating mass is converted into strain energy in the power transmitting components, resulting in very high torsional impact type of torque loads. The conversion of kinetic energy into strain energy can best be illustrated by an equivalent two mass system as shown in Fig. 18.

In this system, it is assumed that the shaft speed is 500 rpm and the mass moment of inertia of each disc is 50 lb-in-s². The kinetic energy of each disc is:

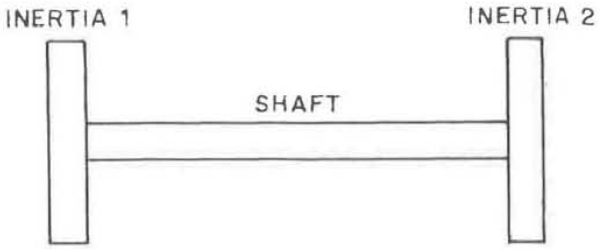


Fig. 18 — Equivalent two mass system

$$K.E. = \frac{1}{2} J \omega^2 = \frac{1}{2} J \left(\frac{2 \pi N}{60} \right)^2$$

where:

K.E. = Kinetic energy, in-lb

J = Mass polar moment of inertia, lb-in-s²

ω = Angular velocity, rad/s

N = Rotational speed, rpm

hence:

$$K.E. = \frac{1}{2} \times 50 \times \left(\frac{2 \pi \times 500}{60} \right)^2$$

K.E. = 68 539 in-lb

For a shaft of uniform cross section subjected to torsional loading, the strain energy in the shaft is:

$$U = \frac{T \theta}{2}$$

where:

U = Strain energy, in-lb

T = Torque, lb-in

θ = Angular deflection, rad

During an instantaneous stoppage of one disc by an external force for the two mass system shown in Fig. 18, the kinetic energy produced by this disc is assumed to be converted to strain energy. The torsional rigidity of the shaft is such that it twists one radian under the torque load. The resulting torque in the shaft is:

$$T = \frac{2U}{\theta} = \frac{2 \times 68\,539}{1}$$

T = 137 078 lb-in

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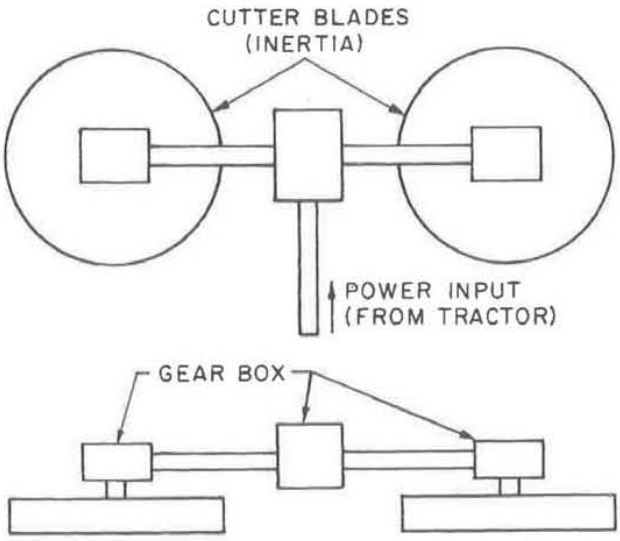


Fig. 19 — Schematic view of double bladed rotary mower

When additional torsional elasticity is introduced into this shaft to obtain a torsional deflection of three radians under the torque load, the maximum torque is:

$$T = \frac{2U}{\theta} = \frac{2 \times 68\,539}{3}$$

$$T = 45\,693 \text{ lb-in}$$

Therefore, a significant reduction in the torsional impact type of torque load results when torsional elasticity is increased in the drive system. A practical example of a case where instantaneous stoppage of mass occurs is a typical double bladed rotary mower as shown schematically in Fig. 19.

When one of the blades stops instantly such as when striking a rock or a stump, the kinetic energy of the other blade is transformed into strain energy in the connecting shafts and gears. As shown in the previous example, a decrease in the spring rate of the shafts between these masses will reduce the torque and consequently the stresses in the shafts and gears of the gear reduction units. For this type of application, as well as others requiring a cushioning effect for torsional impact loads and starting inertia loads, the ring type of rubber element tube provides a means for reducing the detrimental effects of these shock torque peaks.

Torsional Vibration — As with other vehicles, agricultural equipment will, by the nature of their design and use, have inherent vibrations and gear noises, usually as a result of unequal universal joint angles. These disturbing torque pulses, when produced by nonconstant velocity universal joint action, will occur twice per shaft revolution, and if they coincide with the natural frequency of the system, will result in very high peaks. Through analysis of the drivetrain system, a driveshaft can be designed incorporating a rubber element tube that will have a spring rate compatible with the system, thereby reducing and damping noise transfer.

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Section 9.4.2
Torsional Dampers

E. R. Wagner, P.E. — Saginaw Steering Gear Div., GMC
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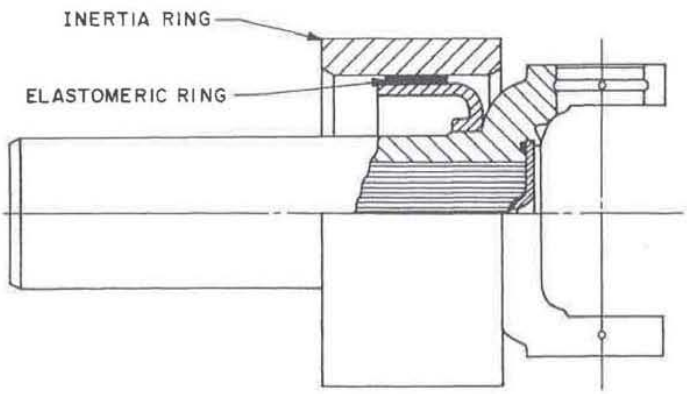


Fig. 1 — Torsional damper

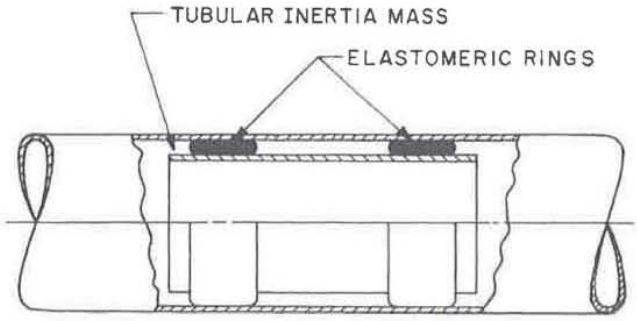


Fig. 2 — Internal vibration absorber

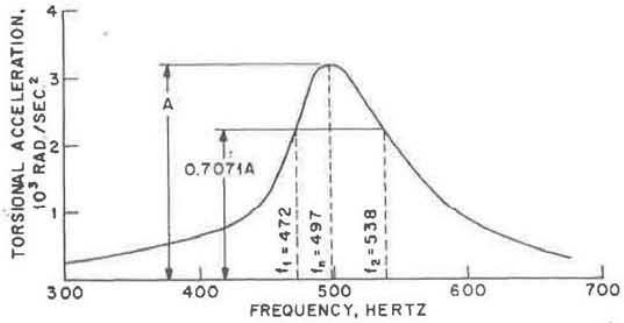


Fig. 3 — Tuned absorber frequency response curve

In conventional passenger cars and trucks, objectionable driveline vibrations may occur at certain speeds and load conditions. Some of these disturbances may be eliminated or reduced to an acceptable level with the incorporation of a torsional dynamic absorber or damper into the main torsional drivetrain system of the vehicle. Such an absorber or damper, shown in Fig. 1, is a tuned mechanical device, which consists generally of an inertia ring attached to a drivetrain component by means of an elastomeric inner ring. This device can also be of the internal vibration absorber (IVA) construction shown in Fig. 2 where a tubular inertia mass is located usually by means of two elastomeric rings inside a driveshaft tube. Both of these devices are based on the fundamental principle of the damped vibration absorber, wherein the amplitude of the main mass, excited by a disturbing force at a given frequency, is reduced by a smaller mass tuned to the impressed or disturbing frequency.

The main torsional system in an automotive vehicle consists of the engine, flywheel, plus clutch, or converter, and transmission inertia masses connected through a relatively flexible driveshaft to the final drive, axle, brake, and drive wheel inertia masses. This elastic system is susceptible to torsional oscillations. On the basis of a computer analysis, it is possible to predict the problem or resonant frequencies and then formulate torsional absorbers with optimum location, spring rate, inertia mass, and damping factor, that is, tuning.

The ideal location for the absorber is at the antinode of the torsional system. This is the point of maximum angular velocity or highest amplitude of oscillation. Here the damper will be most effective. When this optimum location is not feasible or practical, then the damper should be located as far as possible from the nodal point (point of zero motion). At the nodal point, a damper would be ineffective.

When a given tuned damper is forced to oscillate at a constant level of input angular acceleration through an appropriate frequency range which includes its resonant frequency, a frequency response curve is generated. The shape of this curve is based on the damping in the elastomeric element and is a measure of the damper's effectiveness. When viscous damping is assumed to apply, the width of the resonance curve at the half-power points can be determined. These half-power points are referred to as sidebands and are the two frequencies on either side of resonance where the amplitudes are $\sqrt{2}/2$ times the peak amplitude.

Fig. 3 shows a typical frequency response curve at ambient temperature for an axle noise tuned torsional damper where the damping is to be 5-10% of critical damping. In establishing the tuning requirements for this particular damper, it is also desired that the resonant frequency must fall within the temperature band shown in Fig. 4. This damper is required in a passenger car which exhibits axle gear noise at 55 mph. For this particular vehicle, N/V, the ratio of driveshaft speed in rpm to car speed in mph is 42. Also, the number of teeth in the drive pinion is 13. From this vehicle data, it is possible to compute the axle noise frequency which is the same frequency as the pinion and ring gear tooth mesh or contact frequency as follows:

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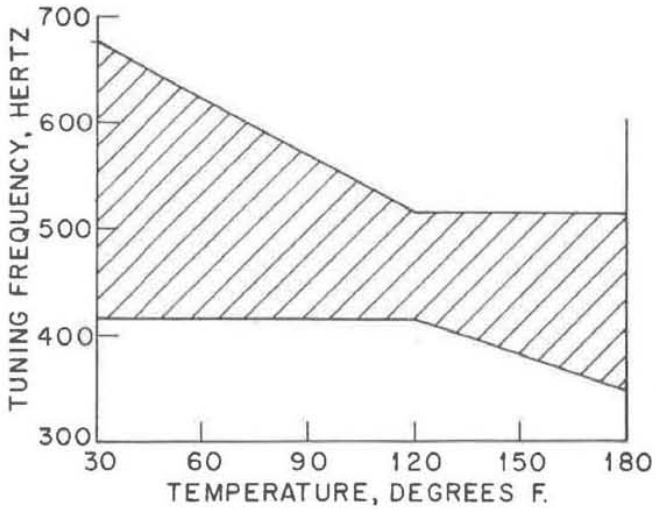


Fig. 4 — Effect of temperature on tuning frequency

$$\text{Axle Noise Frequency} = \frac{55 \times 42 \times 13}{60} = 500 \text{ cps (Hz)}$$

Hence, the resonant or peak frequency of the damper producing the frequency response curve in Fig. 3 may be tuned to the problem frequency of the car. However, in some instances the optimum tuning of a damper has been found to differ from the problem frequency experienced in the car. By analyzing this frequency response curve, it can be determined whether it meets the required damping of the intended drivetrain installation.

The damping factor is:

$$\zeta = \frac{c}{c_c} = \frac{f_2 - f_1}{2f_n}$$

where:

c = Torsional damping coefficient of the elastomeric element, lb-in-s/rad

c_c = Critical torsional damping coefficient, lb-in-s/rad

f_n = Resonant or peak frequency, Hz

f_1 and f_2 = Frequencies on each side of resonance for which the amplitudes are $\sqrt{2}/2$ or 0.7071 times the peak amplitude (called the sidebands or half-power points as previously indicated), Hz

hence:

$$\zeta = \frac{f_2 - f_1}{2f_n} = \frac{538 - 472}{2 \times 497}$$

$$\zeta = 0.0664$$

The percent of critical damping is:

$$\text{Percent damping} = 100\zeta = 100 \times 0.0664 = 6.64\%$$

Thus, this value would meet the damping requirements for the damper. For best performance, a damper should have light damping for which the recommended range of critical damping has been 5-15%.

Generally, a torsional damper is used to control axle noise in the 350-650 Hz frequency range. It has also been utilized to minimize start-up groan or G-string noise in the lower frequency range of 100-150 Hz at ambient temperature.

The rotational mass moment of inertia of the inertia ring of typical dampers have been in the range of approximately 0.008-0.031 lb-in-s^2. The rule of thumb has been that the inertia ratio or the ratio of the absorber inertia to the main inertia should be 1/10:1/12. An abosrber so designed would have a practical size. The main inertia referred to in this ratio is the effective power train inertia at the point in the driveline where the damper is to be located.

The inertia of the damper can be determined experimentally, by trial and error, or analytically by computer analysis. The geometric limits of the inertia ring configuration are usually governed by vehicle packaging requirements. In past practice, the outside diameters of typical inertia rings have ranged from 3.50-5.50 in approximately. Hence, by assuming an annular size which provides the necessary clearances with other drivetrain components and the underbody, and by using the known ring material density, this configuration can be determined with the following equation:

$$J = \frac{W}{g} k^2$$

where:

J = Mass moment of inertia of absorber ring, lb-in-s^2

W = Weight of absorber ring, lb

k = Radius of gyration of absorber ring, in

g = Acceleration due to gravity, 386.4 in/s/s

Generally, the damper elastomer is an integral ring or a rolled up rectangular strip which is assembled in compression in the annular space between the inertia ring and a hub on the driveline.

For typical axle noise and start-up groan tuned absorbers, the section thickness of the compressed elastomeric rings have been in the approximate range of 0.09-0.13 in and 0.19-0.28 in, respectively.

This elastomer is usually rubber, which is in the typical hardness range of 50-70 Shore A durometer, depending on the tuning requirements. In order for the damper to function effectively in a drivetrain installation, the percent compression has been generally in the typical range of 22-30% and can be determined as follows:

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$$\text{Percent compression} = \left(\frac{\text{Free elastomer section thickness} - \text{Compressed elastomer section thickness}}{\text{Free elastomer section thickness}} \right) 100$$

The percent compression should be sufficient to prevent slipping of the inertia ring relative to the hub when operating under varying environmental conditions in a particular installation.

Inertia rings are commonly machined from a cast iron casting. Depending on the design configuration and the size of the inertia mass of the ring, it may be therefore desirable to specify a minimum bursting speed depending on the maximum driveline speed.

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Section 9.4.3
Liners

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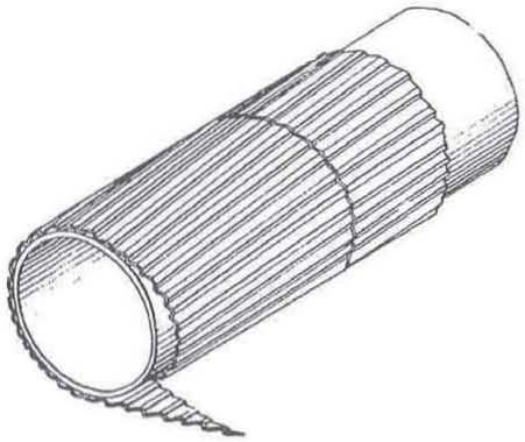


Fig. 1 — Cardboard liner

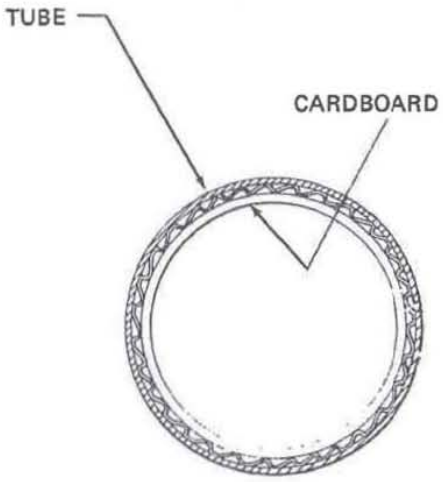


Fig. 2 — Section view of cardboard liner pressed into tube

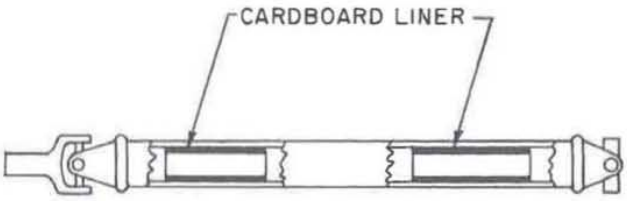


Fig. 3 — Typical tubular driveshaft with two cardboard liners

In certain applications, such as passenger cars for example, driveshafts with plain steel tubing may emit objectionable disturbances which are an inherent characteristic of their design. The most common disturbance is driveshaft ringing, and is an audible metallic disturbance which is caused by the hollow steel tube amplifying longitudinal vibrations. This disturbance is induced by diverse driveline inputs at approximately 100 Hz.

Driveshaft ringing may be reduced to an acceptable level by installing a liner consisting of a suitable sound and vibration deadening material on the inside diameter of the steel tubing. In general, the liner is an oversize tube of adhesive bonded cardboard paper construction which consists typically of a three ply core surrounded by a single ply spiral wound corrugation as shown in Fig. 1. This spiral wound corrugation distorts upon assembly and provides an adequate interference fit with the steel tube inside diameter, as shown in Fig. 2, so that the cardboard liner will not move relative to the tube. Usually, the liner is cut to nearly the full length of the driveshaft tube. Also, the liner may be used in shorter lengths for installation in one or both ends of the tube as illustrated in Fig. 3. The cardboard tube construction, as described, generally produces only a minimal effect on the driveshaft critical speed and dynamic balance, which is less than that resulting from the use of a rubber element tube in the same application.

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Critical Speed

Chapter 10

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Chapter 10
Critical Speed

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Theory

It is well known that rotating shafts at certain speeds can become dynamically unstable and vibrations of large amplitude are likely to occur which may result in fracture. This phenomenon is due to the shaft unbalance forces and is known as critical, whirling, or whipping speed. It is the speed at which the rotational speed coincides with the transverse natural vibration frequency of the shaft.

In spite of careful workmanship, the center of mass of a revolving shaft never coincides exactly with the axis of rotation. Owing to the eccentricity of the mass center, an unbalanced centrifugal force is produced which causes the shaft to whirl and the system to vibrate. The phenomenon would be still more pronounced if the shaft carried a heavy disc at the middle of its length with its center of mass offset at a distance e from the axis of rotation, as shown in Fig. 1 in an idealized system.

Under the influence of centrifugal force, the shaft at the middle of its length will deflect a distance y from its neutral position. This force is balanced by the internal elastic restoring force of the shaft. The internal hysteresis of the shaft material acts as damping on the whirl. A self-excited whirl at the critical speed may build up and the deflection increases to infinity assuming no damping.

The typical driveline is essentially a long slender rotating member and being of elastic material, it has a transverse spring rate. This elastic characteristic allows it to distort when subjected to various transverse vibration conditions.

The driveline depends for its support on some external member. This member, in itself, is usually resiliently supported and, therefore, has some series of response characteristics. The coupling of the resilient long shaft and resilient supports can produce many transverse vibratory modes. It is traditional to think of this driveshaft as having a critical speed which must be avoided. A simple critical speed condition actually exists only when the driveshaft is supported on a completely rigid structure, as shown in Fig. 1. Under this condition, a simple bending mode of vibration based upon rigid supports is possible. In a typical one piece vehicle driveline installation, the driveshaft has a universal joint at each end. It is assumed to be a simply supported beam where the rigid end supporting condition at the joint centers is considered to be a hinged or pinned connection. If the shaft is considered uniform in cross section along its entire length, the critical speed is:

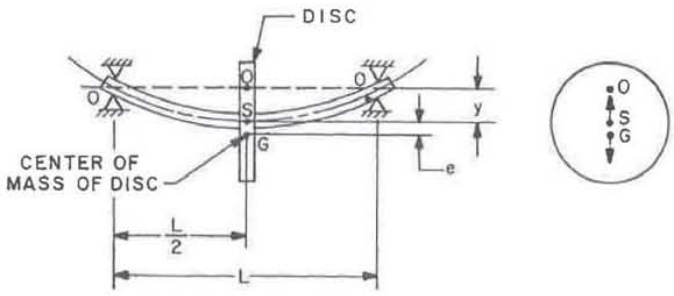


Fig. 1 — Shaft with central unbalanced disc

$$N_c = 30\,\pi \sqrt{\frac{EIg}{WL^3}}$$

where:

N_c = Revolutions per minute, rpm
E = Modulus of elasticity, psi
I = Area moment of inertia, in^4
g = Acceleration due to gravity, in/s^2
W = Total weight of shaft, lb
L = Shaft length between supports (universal joint centers), in

The speed determined by this equation is the lowest critical speed. Other critical speeds are harmonics and occur at 4, 9, 16, and so on multiples of the fundamental. Generally, these higher orders are not of importance in vehicle driveline installations. The half critical speed may be important in certain drivelines due to secondary couple action resulting from nonconstant velocity type universal joints, since the couple force oscillates at a frequency of twice shaft speed.

When assuming a round steel shaft having a modulus of elasticity of 29 000 000 psi and a density of 0.281 lb/in^3, the previous equation can be expressed as follows:

Round Solid Steel Shaft

$$N_c = 4\,705\,000\,\frac{d}{L^2}$$

where:

d = Shaft diameter, in
L = Shaft length between supports (universal joint centers), in

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Round Tubular Steel Shaft

$$N_c = 4\ 705\ 000\ \frac{\sqrt{d_o^2 + d_i^2}}{L^2}$$

where:

d_o = Outside diameter, in
d_i = Inside diameter, in
L = Shaft length between supports (universal joint centers), in

It is apparent that equations can easily be derived for other shaft materials simply by knowing the modulus of elasticity and density of the intended material. It is also possible to revise the equations if shaft sections other than round solid or tubular configurations are proposed. However, it has been found that a thin wall, round tubular section is the most efficient because the critical speed varies directly as the section moment of inertia, and inversely as the mass per unit length.

A typical driveline supported on a resilient structure does not possess the characteristics defined by rigid supports. The deviations from the simple fundamental of vibration characteristics are more pronounced as the supports become relatively softer. A relationship can, therefore, be defined showing the expected transverse characteristics as related to the transverse spring rate of the shaft and to the spring rate of its supports. A term "stiffness ratio" can be used to define this relationship as follows:

$$\text{Stiffness Ratio} = \frac{EI}{kL^3}$$

where:

E = Modulus of elasticity
I = Area moment of inertia
k = Spring rate of the support
L = Shaft length between supports

Transverse vibration characteristics essentially define the response of the system to a periodic exciting force. In discussing transverse vibrations, this exciting force is usually unbalance. Secondary couple loads often produce similar characteristics. As previously indicated, the two per revolution characteristics of secondary couple loads usually have the effect of a force vibration at one half of the rotating speed of basic response from the one per revolution unbalance excitation.

Mounting characteristics can vary over a wide range between the two extremes of the rigid and relatively soft supports. A stiffness ratio of zero indicates completely rigid supports or a fixed-fixed condition. As the stiffness numerical value increases, the supports become progressively more resilient and approach the free-free condition. The actual frequencies that exist will depend upon the relative stiffness of the supports and the driveshaft itself.

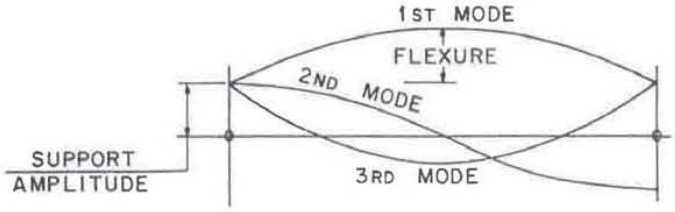


Fig. 2 — Critical speed mode shapes

Actual Critical Speed Mode Shapes

The use of the previous critical speed equations is satisfactory where comparatively stiff support structures are utilized. They are, however, very much in error when extremely resilient supports are employed. For example, the very low rate, low frequency shaft support bearings used in multi-piece driveshafts completely change these critical speed considerations. The introduction of more resilient supports actually introduces additional modes of vibration and has the effect of changing the fundamental transverse frequency. When supported on a very low rate structure, the driveshaft's critical speed characteristics are as follows:

1. The first mode of vibration resembles a bounce frequency. There is some flexure of the driveshaft depending on the relative stiffness, but the ends and center of the shaft are moving in the same direction at the same time.
2. The second mode of vibration of the driveshaft resembles a pitch frequency. In this condition, opposite ends of the shaft are vibrating in opposite directions at the same time and the shaft itself flexes slightly. Its basic motion is pitching around the middle of the driveshaft itself.
3. The third mode of vibration resembles a free-free beam. There is a large amount of flexure within the shaft itself. The ends of the driveshaft are vibrating in a direction opposite that of the center of the shaft resulting in a large degree of shaft flexure.

These three vibration mode shapes are shown in Fig. 2. Some driveshafts may therefore assume critical speed characteristics depending upon the support restraint. With extremely rigid supports, it tends to vibrate as a fixed-fixed beam as described by the previous basic equations. With extremely soft supports, the shaft tends to bounce, pitch, and flex as a free-free beam in that order. The softer the supports, the less flexure will occur within the first two modes and they will essentially be a rigid body bouncing and pitching on flexible supports. The third mode, however, as a free-free beam will involve large amounts of flexure within the driveshaft itself.

The support characteristics can vary over a large range between the two extremes of fixed-fixed and free-free beams. The practical effects of these mounting characteristics on the critical speed are as follows:

1. Driveshafts mounted on relatively stiff supports reasonably well, obey the classic approach of the fixed-

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fixed beam. It is merely necessary to maintain the maximum operating speed well below the fundamental transverse frequency to avoid critical speed.

2. With extremely resilient supports, the first and second modes of vibration are virtually bounce and pitch frequencies. These modes can have the characteristic of low frequency, short duration, and instability which can operate in a virtually vibration free manner. The small forces from unbalance with these low speeds further help the situation.
3. With extremely resilient supports, the third mode of vibration is well above those computed by the classical methods. The third mode can be over twice the basic fundamental frequency when this condition is at an extreme. It must be pointed out that extremely high flexure occurs in this third mode and that it does become the most serious potential failure condition.
4. With elastic supports, driveshafts can be made longer and thinner without entering into critical speed difficulties.

It is generally considered that the critical speed itself is a potential failure situation and is certainly true of the simply supported shaft on comparatively rigid mountings. As the critical speed is approached, there is a large increase in the one per revolution exciting force being transferred to the supports. There is also an increase in flexure stress within the shaft itself. There would normally be much disturbing vibration and the possibility of whipping out of the driveshaft.

When dealing with extremely resilient mountings on even one end of the driveshaft, the situation is much different. The lower modes of vibration will occur at much lower speeds and will involve more motion in the supports than flexure in the driveshaft. Also, there will be transmitted vibrations. At low speeds, these disturbances can be so small as to be virtually vibration free. Therefore, with extremely soft mounts, there is little difficulty operating through the vibratory modes.

With very elastic supports, the third mode of vibration is extremely serious. Since the shaft is tending to oscillate as a free-free beam, there is very high flexural stress occurring. The mounts themselves, although subjected to vibration, are actually moving opposite to their imposed force in acting in a post resonant manner. There will generally be little transmission of vibration to the supports despite the large flexural stress in the shaft itself. Driveshaft whipping out, or whirling in this condition is even more likely than with a fixed-fixed relationship. This critical speed failure combined with the low level of transmitted vibration, produces an instantaneous type of failure with little vibration warning to the vehicle occupants. These failures occur quickly and are explosive in nature. It, therefore, becomes extremely important to avoid any operation at or approaching this mode of vibration, where very soft supports are used.

Shaft Whirl

A characteristic of any vehicle driveline at its critical speed is that of driveshaft elliptical whirl. This whirling is synchronous in nature in that the whirl frequency is the same as the shaft speed and can be defined as an elliptical orbital path superimposed on the rotation of a shaft around its bearing axis. During whirl the shaft bends outward and describes this orbital path.

This whirling condition can also be either forward or reverse in direction, as shown in Figs. 3 and 4, respec-

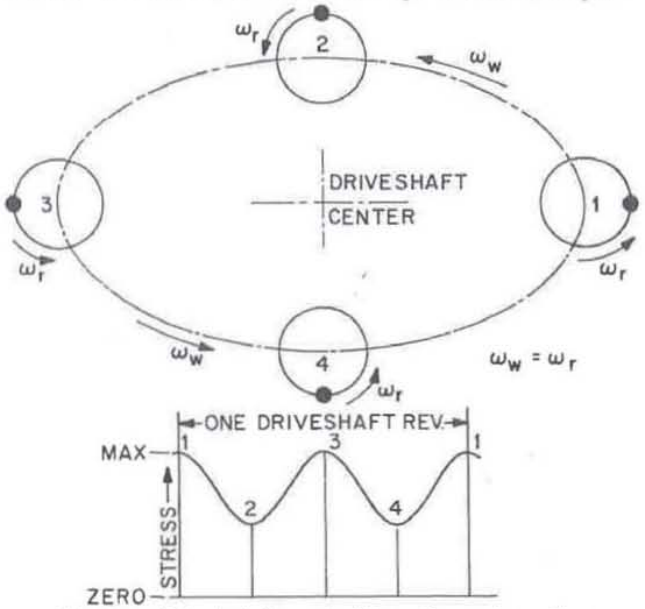


Fig. 3 — Driveshaft forward whirl motion and resulting stress

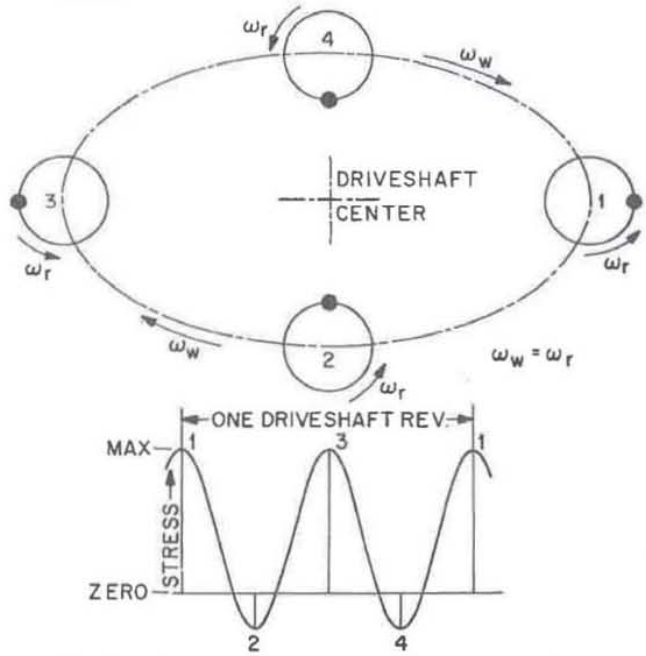


Fig. 4 — Driveshaft reverse whirl motion and resulting stress

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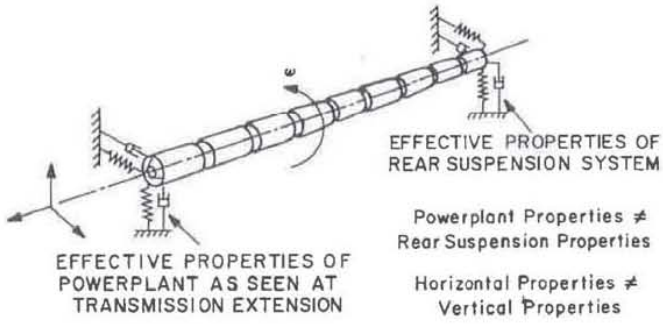


Fig. 5 — Driveshaft end constraints

tively. In forward whirl, the ellipse is generated in the same direction as the direction of shaft rotation, while in reverse whirl, the ellipse is generated opposite to the direction of shaft rotation.

The whirling is elliptical simply because of unsymmetrical driveshaft support stiffnesses such as shown in Fig. 5. The variations in amplitude depend on these vertical and horizontal end conditions. The driveshaft at any one point can be considered in equilibrium. The total deflecting force is equal to the internal restoring force.

In order to more fully understand shaft whirl and the resulting stresses, an analytical analysis is required. The driveshaft is assumed to be rotating with a speed of ω_r in nonsymmetric bearing supports. Also, it is assumed that the centrifugal force produced by shaft unbalance is sufficient to cause the driveshaft to bend such that its center of gravity location is displaced in a horizontal and vertical direction from the bearing axis. This condition is termed shaft whirl, meaning that the driveshaft rotates on its axis with a speed of ω_r, while its center of gravity translates around the bearing axis describing an ellipse at a speed of ω_w. The whirling frequency is the same as that of shaft speed and the whirling motion is elliptical simply because of the non-symmetrical end supports where the rotating centrifugal force vector caused by the unbalance produces different amounts of motion in the horizontal and vertical directions. This type of motion generates an ellipse. The whirl can be either forward or reverse depending upon other system parameters.

When a rotating shaft exhibits a whirling motion, bending stresses are induced in the shaft which can result in varying degrees of alternating-mean stress relationships. Although these stress levels may be below the yield strength of the material, they can result in a fatigue failure in due time. For example, if a point of maximum stress on the driveshaft is taken and if this stress is plotted for one complete revolution of the shaft in forward whirl, as shown in Fig. 3, it can be seen that the stress frequency is twice the rotational frequency of the shaft and there is no stress reversal. The stress, in other words, varies in either all tension or all compression. If now this same point is taken and the stress is plotted for one complete revolution of the shaft in reverse whirl, as shown in Fig. 4, it can be seen that the maximum stress for both conditions is the same, but that the variable stress with reverse whirl is greater. It is also noted that the stress for reverse whirl alternates from tension to compression, resulting in a lower fatigue life than for the same motion in forward whirl. For this comparison, it is assumed that the maximum stresses in both cases are the same. It is possible then that reverse whirl can be detrimental to a driveline, not only because it lowers its fatigue life, but also because it may degenerate into a relatively unstable whirl condition.

Reverse whirl is undesirable as compared to forward whirl if the elliptical motions for both conditions of whirl were identical. In an actual driveline installation, however, these motions are not the same. The reverse motion usually is less than the forward motion. Experimental work has shown that reverse whirl always occurs after forward whirl and might not be as detrimental as originally expected. In fact, there is some empirical evidence that to some extent the reverse whirl may be even desirable.

Agreement with Practical Results

As previously stated, it has been found that the actual critical speeds are somewhat lower than the theoretically calculated values. This is due to the fact that the points of support are not rigid. A vehicle driveshaft, when rotating at high speed, will whirl in the same way as a heavy rope which is being swung around by two persons. If they cease their whirling effort, their hands will, nevertheless, be carried around in a circle. This is also so with the driveshaft supports. The latter include the universal joints, which are fitted to shafts overhanging their bearings. Under the influence of the centrifugal force on the driveshaft, these shafts and bearing supports will flex in the same plane as the driveshaft, thus virtually increasing the distance between supports. The effect, of course, depends on the relative stiffness or resilience and on the overhang of the connected shafts. Additional influences on the reduction of the calculated critical speed can be caused by shaft unbalance, mass distribution in non-homogeneous shafts, and secondary couples. Also, compliance in the universal joints, transmission and engine mounts, shaft support bearings, and axle or suspension pivot bushings can influence the driveshaft critical speed. The influences of these various factors are also discussed in the chapters on *Vibration Control* and *Shaft Support Bearings*.

Safe Operating Speed

Since the actual drivetrain system critical speed is somewhat below the theoretically calculated value for the driveshaft, factors can be used to determine the maximum safe operating speed permissible for a driveshaft in a particular installation. In general, a safe speed factor of 0.85 has been used for passenger car and light-duty truck applications, while 0.75 has been generally accepted for medium and heavy-duty trucks.

These speed factors are associated with vehicle applica-

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tion experience and are related to the following generally accepted driveshaft installation parameters:

Universal Joint Angles	6 deg maximum continuous operating angle
Driveshaft Length	60 in maximum installed center to center distance of universal joints.

Also, these speed factors are based on the following driveshaft unbalance limits:

Passenger Cars and Light-Duty Trucks	0.500 oz-in for each ten lb of weight per end of shaft.
Medium and Heavy-Duty Trucks	1.000 oz-in for each 10 lb of weight per end of shaft.

Therefore, the maximum safe operating speed of a shaft is:

$$N_s = KN_c$$

where:

K = Safe speed factor
N_c = Critical speed, rpm

Successful applications exceeding the above mentioned angle and length parameters have been and are continuing to be made. However, the degree of their success is generally associated with the compatibility of the end supporting members to the weight and length of the driveshaft as indicated previously.

When applying the 0.85 and 0.75 speed factors to shaft lengths in excess of 60 in, it is recommended that caution be exercised. It may be necessary to match the powerplant support to the driveshaft to obtain maximum critical speed capability. Positioning and rigidity of the powerplant supports becomes more significant since the weight of the driveshaft attached to its supporting member may be enough to create a first order response. If sustained vehicle operation is in this critical frequency, failures can result in the transmission extension, converter housing, or clutch housing.

Solutions to failures of this type are:

1. Redesign the powerplant support means and rigidity.
2. Redesign the driveshaft with a greater section moment of inertia without increasing the weight per unit length.
3. Reduce driveshaft weight on supporting member by introducing a two piece driveshaft with a shaft support bearing.

Factors Affecting Shaft Length

The maximum length of a driveshaft is controlled by various factors. Some of these factors are:

1. Critical speed
2. Tubing outer diameter
3. Tubing wall thickness
4. Driveshaft weight
5. Spline slip section between joint centers or outboard of joint centers
6. Driveshaft end supporting member rigidity
7. Precision of rotational axis coinciding with geometric axis.
8. Fit of slip spline member
9. Clearance along axis of cross trunnions

In general, these factors have the following effects on the length:

1. Critical speed increase → length decrease
2. Tubing outer diameter increase without an increase in weight per unit length → length increase
3. Tubing wall thickness increase → length decrease
4. Driveshaft weight increase → length decrease
5. Spline slip between joint centers → length decrease
6. Driveshaft end support stiffness (as attached to solid frame) increase → length decrease
7. Precision of rotational axis increase → length increase
8. Looser spline fit → length decrease
9. Clearance along cross trunnion axis increase → length decrease

Nonuniform Section Shafts

The previous equations for critical speed are based on the shaft having a uniform moment of inertia or section size. For a driveshaft having a variable cross section and moment of inertia along its length due to universal joints, swaged tubing, splined couplings and so forth, and if the shaft material is nonhomogeneous in character, the computations for critical speed are more complicated. For such a shaft, the Rayleigh or Stodola methods can be used, which are described in various mechanical vibration textbooks as indicated in the references. These methods involve a combination of analytical and graphical procedures which can easily be programmed for use on a computer. The most common example of a nonhomogeneous section shaft is the rubber element tube driveshaft. This type of shaft, as well as typical physical properties of the elastomeric section necessary for computations, are described in the chapter on *Vibration Control*.

Critical Speed Charts

Figs. 6-9 are charts of theoretically calculated critical speeds for round tubular steel driveshafts of the more commonly used tube outside diameters and wall thicknesses. When using the values from the charts, the appropriate safe speed factor should be applied to determine the maximum safe operating speed. The charts should be used only when other means of determining the critical speed are not available.

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O. D. Tube		1¾"			2"					2½"				
Wall Thickness, in.		.120	.083	.065	.134	.109	.095	.083	.065	.134	.109	.095	.083	.065
LENGTH IN INCHES	36	8390	8568	8656	9603	9723	9791	9849	9938	12164	12286	12355	12414	12504
	37	7942	8111	8194	9091	9205	9269	9324	9408	11516	11631	11696	11752	11837
	38	7530	7690	7769	8619	8727	8787	8840	8920	10918	11027	11089	11142	11222
	39	7148	7300	7375	8183	8285	8343	8392	8468	10365	10468	10527	10578	10654
	40	6796	6940	7011	7779	7876	7931	7978	8050	9853	9952	10007	10055	10128
	41	6468	6605	6673	7404	7496	7549	7594	7662	9378	9472	9525	9571	9640
	42	6164	6295	6359	7055	7143	7193	7236	7302	8937	9026	9077	9121	9186
	43	5880	6005	6067	6731	6815	6863	6904	6966	8526	8611	8660	8701	8764
	44	5616	5735	5794	6429	6509	6554	6593	6653	8143	8224	8271	8310	8370
	45	5369	5483	5540	6146	6223	6266	6304	6361	7785	7863	7907	7945	8002
	46	5138	5248	5302	5882	5955	5997	6033	6087	7450	7525	7567	7603	7658
	47	4922	5027	5078	5634	5704	5744	5779	5831	7137	7208	7249	7283	7336
	48	4719	4819	4869	5402	5469	5507	5540	5590	6842	6911	6950	6983	7033
	49	4528	4625	4672	5184	5248	5285	5316	5364	6566	6632	6669	6701	6749
	50	4349	4442	4487	4978	5040	5076	5106	5152	6306	6369	6405	6435	6482
	51	4180	4269	4313	4785	4845	4879	4908	4952	6061	6122	6156	6186	6230
	52	4021	4106	4149	4603	4660	4693	4721	4763	5830	5889	5922	5950	5993
	53	3871	3953	3994	4431	4486	4517	4544	4585	5612	5668	5700	5728	5769
	54	3729	3808	3847	4268	4321	4352	4377	4417	5406	5460	5491	5517	5557
	55	3594	3671	3708	4114	4166	4195	4220	4258	5212	5264	5293	5319	5357
	56	3467	3541	3577	3969	4018	4046	4070	4107	5027	5077	5106	5130	5167
	57	3347	3418	3453	3831	3878	3906	3929	3964	4852	4901	4928	4952	4988
	58	3232	3301	3335	3700	3746	3772	3795	3829	4686	4733	4760	4783	4817
	59	3123	3190	3223	3575	3620	3645	3667	3700	4529	4574	4600	4622	4655
	60	3020	3084	3116	3457	3500	3525	3546	3578	4379	4423	4448	4469	4501
	61	2922	2984	3015	3345	3386	3410	3430	3461	4237	4279	4303	4324	4355
	62	2829	2889	2918	3238	3278	3301	3321	3351	4101	4142	4165	4185	4216
	63	2739	2798	2826	3136	3175	3197	3216	3245	3972	4012	4034	4054	4083
	64	2654	2711	2739	3039	3076	3098	3116	3145	3849	3887	3909	3928	3956
	65	2573	2628	2655	2946	2983	3003	3021	3049	3731	3769	3790	3808	3835
	66	2496	2549	2575	2857	2893	2913	2930	2957	3619	3655	3676	3693	3720
	67	2422	2474	2499	2773	2807	2827	2844	2869	3512	3547	3567	3584	3610
	68	2351	2401	2426	2692	2725	2744	2761	2785	3409	3443	3463	3479	3505
	69	2284	2332	2356	2614	2647	2665	2681	2705	3311	3344	3363	3379	3404
	70	2219	2266	2289	2540	2572	2590	2605	2629	3217	3250	3268	3283	3307
	71	2157	2203	2225	2469	2500	2517	2532	2555	3127	3159	3176	3192	3215
	72	2097	2142	2164	2401	2431	2448	2462	2485	3041	3071	3089	3104	3126
	73	2040	2084	2105	2335	2365	2381	2395	2417	2958	2988	3005	3019	3041
	74	1986	2028	2049	2273	2301	2317	2331	2352	2879	2908	2924	2938	2959
	75	....	1974	1994	2213	2240	2256	2269	2290	2803	2831	2847	2860	2881
	76	....	....	....	2155	2182	2197	2210	2230	2729	2757	2772	2785	2806
	77	....	....	....	2099	2125	2140	2153	2172	2659	2686	2701	2714	2733
	78	....	....	....	2046	2071	2086	2098	2117	2591	2617	2632	2644	2664
	79	....	....	....	1994	2019	2033	2045	2064	2526	2551	2566	2578	2597
	80	....	....	....	....	1969	1983	1994	2013	2463	2488	2502	2514	2532
	81	....	....	....	....	....	....	....	1963	2403	2427	2440	2452	2470
	82	....	....	....	....	....	....	....	....	2345	2368	2381	2393	2410
	83	....	....	....	....	....	....	....	....	2288	2311	2324	2335	2352
	84	....	....	....	....	....	....	....	....	2234	2257	2269	2280	2297
	85	....	....	....	....	....	....	....	....	2182	2204	2216	2227	2243
	86	....	....	....	....	....	....	....	....	2132	2153	2165	2175	2191
	87	....	....	....	....	....	....	....	....	2083	2104	2115	2126	2141
	88	....	....	....	....	....	....	....	....	2036	2056	2068	2078	2093
	89	....	....	....	....	....	....	....	....	1990	2010	2021	2031	2046
	90	....	....	....	....	....	....	....	....	....	1966	1977	1986	2001
	91	....	....	....	....	....	....	....	....	....	....	....	....	1957
	92	....	....	....	....	....	....	....	....	....	....	....	....	....
	93	....	....	....	....	....	....	....	....	....	....	....	....	....
	94	....	....	....	....	....	....	....	....	....	....	....	....	....
	95	....	....	....	....	....	....	....	....	....	....	....	....	....
	96	....	....	....	....	....	....	....	....	....	....	....	....	....
	97	....	....	....	....	....	....	....	....	....	....	....	....	....
	98	....	....	....	....	....	....	....	....	....	....	....	....	....
	99	....	....	....	....	....	....	....	....	....	....	....	....	....
	100	....	....	....	....	....	....	....	....	....	....	....	....	....
	101	....	....									....	....	....
	102	....	....									....	....	...
	103	....	....									....	....	....
	104	....	....									....	....	....
	105	....	....	....	....	....	....	....	....	....	....	....	....	....
	106	....	....	....	....	....	....	....	....	....	....	....	....	....
	107	....	....	....	....	....	....	....	....	....	....	....	....	....

Fig. 6 — Tubular driveshaft critical speed

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3″					3½″						4″					
.134	.109	.095	.083	.065	.134	.120	.109	.095	.083	.065	.134	.120	.109	.095	.083	.065
14728	14850	14920	14979	15069	17292	17361	17415	17485	17545	17635	19857	19926	19981	20051	20111	20201
13942	14058	14124	14181	14266	16370	16435	16487	16553	16609	16695	18798	18863	18915	18982	19039	19124
13218	13328	13390	13444	13525	15520	15582	15630	15693	15747	15828	17821	17884	17933	17996	18050	18131
12549	12653	12713	12764	12840	14734	14793	14839	14898	14950	15027	16919	16978	17025	17085	17136	17213
11929	12029	12085	12133	12206	14006	14062	14106	14163	14211	14285	16084	16140	16184	16241	16290	16363
11355	11449	11503	11549	11618	13332	13385	13427	13480	13527	13596	15309	15362	15405	15459	15505	15575
10820	10910	10961	11005	11071	12704	12755	12795	12846	12890	12957	14589	14639	14680	14731	14775	14842
10323	10409	10457	10499	10563	12120	12169	12207	12256	12298	12361	13918	13966	14005	14054	14096	14160
9859	9941	9987	10028	10088	11576	11622	11658	11705	11745	11805	13292	13339	13376	13422	13463	13523
9426	9504	9549	9587	9644	11067	11111	11146	11190	11229	11287	12708	12753	12788	12833	12871	12929
9020	9095	9138	9175	9230	10591	10633	10666	10709	10746	10801	12162	12204	12238	12281	12318	12373
8641	8713	8753	8788	8841	10145	10185	10217	10258	10293	10346	11650	11690	11723	11764	11799	11852
8284	8353	8392	8426	8477	9727	9765	9796	9835	9869	9920	11169	11208	11239	11279	11312	11363
7950	8016	8053	8086	8134	9334	9371	9400	9438	9470	9519	10718	10756	10785	10823	10855	10904
7635	7698	7734	7765	7812	8964	9000	9028	9064	9095	9142	10294	10330	10358	10394	10426	10472
7338	7399	7434	7464	7509	8616	8650	8678	8712	8742	8787	9894	9928	9956	9991	10021	10066
7059	7118	7151	7180	7223	8288	8321	8347	8380	8409	8452	9517	9550	9577	9610	9639	9682
6795	6852	6884	6911	6953	7978	8010	8035	8067	8095	8136	9161	9193	9219	9251	9279	9320
6546	6600	6631	6658	6698	7685	7716	7740	7771	7798	7838	8825	8856	8880	8911	8938	8978
6310	6362	6392	6418	6456	7408	7438	7461	7491	7517	7555	8507	8537	8560	8590	8616	8655
6086	6137	6166	6191	6228	7146	7175	7197	7226	7251	7288	8206	8235	8257	8286	8311	8349
5875	5924	5951	5975	6011	6898	6925	6947	6975	6999	7035	7921	7948	7970	7998	8022	8058
5674	5721	5748	5771	5806	6662	6688	6709	6736	6759	6794	7650	7677	7698	7725	7748	7783
5483	5529	5555	5577	5610	6438	6464	6484	6510	6532	6566	7393	7419	7439	7465	7487	7521
5302	5346	5371	5393	5425	6225	6250	6270	6295	6316	6349	7148	7173	7193	7218	7240	7273
5130	5172	5196	5217	5249	6023	6047	6066	6090	6111	6142	6916	6940	6959	6984	7005	7036
4965	5007	5030	5050	5081	5830	5853	5872	5895	5915	5946	6695	6718	6736	6760	6780	6811
4809	4849	4872	4891	4921	5646	5669	5687	5709	5729	5758	6484	6506	6524	6547	6567	6596
4660	4699	4721	4740	4768	5471	5493	5510	5532	5551	5580	6283	6305	6322	6344	6363	6392
4518	4555	4577	4595	4623	5304	5325	5342	5363	5382	5410	6091	6112	6129	6150	6169	6197
4382	4418	4439	4457	4483	5145	5165	5181	5202	5220	5247	5908	5928	5945	5966	5983	6010
4252	4287	4307	4325	4351	4992	5012	5028	5048	5065	5091	5733	5753	5769	5789	5806	5832
4128	4162	4182	4198	4224	4847	4866	4881	4901	4917	4943	5565	5585	5600	5620	5637	5662
4009	4042	4061	4078	4102	4707	4725	4741	4760	4776	4801	5405	5424	5439	5458	5474	5499
3895	3928	3946	3962	3986	4574	4592	4606	4625	4640	4664	5252	5270	5285	5303	5319	5343
3786	3818	3836	3851	3874	4446	4463	4477	4495	4511	4534	5105	5123	5137	5155	5170	5194
3682	3713	3730	3745	3767	4323	4340	4354	4371	4386	4409	4964	4981	4995	5013	5028	5050
3582	3612	3628	3643	3665	4205	4222	4235	4252	4267	4289	4829	4846	4859	4876	4891	4913
3486	3515	3531	3545	3566	4092	4109	4122	4138	4152	4174	4699	4716	4729	4745	4760	4781
3393	3422	3437	3451	3472	3984	4000	4012	4029	4042	4063	4575	4591	4604	4620	4634	4654
3305	3332	3348	3361	3381	3880	3895	3908	3923	3937	3957	4455	4471	4483	4499	4512	4533
3219	3246	3261	3274	3294	3780	3795	3807	3822	3835	3855	4340	4356	4368	4383	4396	4416
3137	3163	3178	3191	3210	3683	3698	3710	3725	3737	3757	4230	4245	4256	4271	4284	4303
3058	3084	3098	3111	3129	3591	3605	3616	3631	3643	3662	4123	4138	4149	4164	4176	4195
2982	3007	3021	3033	3052	3502	3516	3527	3541	3553	3571	4021	4035	4046	4060	4072	4091
2909	2933	2947	2959	2977	3416	3429	3440	3454	3466	3484	3922	3936	3947	3961	3973	3990
2839	2862	2876	2887	2905	3333	3346	3357	3370	3382	3399	3827	3841	3851	3865	3876	3894
2771	2794	2807	2818	2835	3253	3266	3276	3289	3301	3318	3736	3749	3759	3772	3783	3800
2705	2728	2740	2751	2768	3176	3189	3199	3212	3223	3239	3647	3660	3670	3683	3694	3710
2642	2664	2676	2687	2703	3102	3114	3124	3136	3147	3163	3562	3574	3584	3597	3607	3624
2581	2602	2614	2625	2641	3030	3042	3052	3064	3074	3090	3479	3492	3501	3513	3524	3540
2522	2543	2555	2565	2580	2961	2973	2982	2994	3004	3020	3400	3412	3421	3433	3444	3459
2465	2485	2497	2507	2522	2894	2905	2915	2926	2936	2951	3323	3335	3344	3356	3366	3381
2410	2430	2441	2451	2466	2829	2841	2849	2861	2871	2885	3249	3260	3269	3281	3290	3305
2356	2376	2387	2397	2411	2767	2778	2786	2798	2807	2822	3177	3188	3197	3208	3218	3232
2305	2324	2335	2344	2358	2706	2717	2726	2736	2746	2760	3108	3118	3127	3138	3147	3162
2255	2274	2284	2294	2307	2648	2658	2667	2677	2686	2700	3040	3051	3059	3070	3079	3093
2207	2225	2236	2245	2258	2591	2601	2610	2620	2629	2643	2975	2986	2994	3004	3014	3027
2160	2178	2188	2197	2210	2536	2546	2554	2565	2573	2587	2912	2923	2931	2941	2950	2963
2115	2133	2142	2151	2164	2483	2493	2501	2511	2519	2532	2851	2861	2869	2879	2888	2901
2071	2088	2098	2106	2119	2432	2441	2449	2459	2467	2480	2792	2802	2810	2820	2828	2841
2029	2045	2055	2063	2076	2382	2391	2399	2408	2417	2429	2735	2745	2752	2762	2770	2783
1987	2004	2013	2021	2034	2333	2343	2350	2359	2368	2380	2680	2689	2696	2706	2714	2726
....	1964	1973	1981	1993	2287	2296	2303	2312	2320	2332	2626	2635	2642	2651	2659	2671
....	....	....	....	....	2241	2250	2257	2266	2274	2286	2573	2582	2590	2599	2606	2618
....	....	....	....	....	2197	2206	2213	2221	2229	2240	2523	2532	2538	2547	2555	2567
....	....	....	....	....	2154	2163	2169	2178	2186	2197	2473	2482	2489	2498	2505	2516
....	....	....	....	....	2112	2121	2127	2136	2143	2154	2426	2434	2441	2449	2457	2468
....	....	....	....	....	2072	2080	2087	2095	2102	2113	2379	2388	2394	2403	2410	2421
....	....	....	....	....	2033	2041	2047	2055	2062	2073	2334	2342	2349	2357	2364	2375
....	....	....	....	....	1995	2002	2009	2017	2024	2034	2290	2298	2305	2313	2320	2330
....	....	....	....	....	....	1965	1971	1979	1986	1996	2248	2256	2262	2270	2277	2287

Fig. 7 — Tubular driveshaft critical speed

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PET_AAM_1012-0281

O. D. Tube		4½″							5″						
Wall Thickness, in.		.238	.203	.165	.134	.109	.083	.065	.238	.203	.165	.134	.109	.083	.065
LENGTH IN INCHES	36	21911	22082	22268	22422	22547	22677	22768	24474	24646	24833	24987	25112	25243	25334
	37	20743	20904	21081	21226	21344	21468	21554	23169	23331	23509	23655	23773	23897	23983
	38	19666	19818	19986	20124	20236	20353	20434	21966	22120	22288	22426	22538	22656	22738
	39	18670	18815	18974	19105	19211	19322	19400	20854	21000	21160	21291	21397	21509	21586
	40	17748	17886	18037	18162	18263	18368	18442	19824	19963	20115	20240	20341	20447	20521
	41	16893	17024	17168	17286	17383	17483	17553	18869	19001	19146	19264	19361	19462	19532
	42	16098	16223	16360	16473	16565	16661	16727	17981	18107	18245	18358	18450	18546	18613
	43	15358	15477	15608	15716	15803	15895	15958	17154	17275	17406	17514	17602	17693	17757
	44	14668	14782	14907	15010	15093	15180	15241	16384	16498	16624	16727	16811	16898	16959
	45	14023	14132	14252	14350	14430	14513	14571	15664	15773	15893	15992	16072	16156	16214
	46	13420	13524	13639	13733	13809	13889	13945	14990	15095	15210	15304	15381	15461	15517
	47	12855	12955	13065	13155	13228	13304	13358	14359	14459	14569	14660	14733	14810	14863
	48	12325	12421	12526	12612	12682	12756	12807	13767	13863	13969	14055	14126	14199	14250
	49	11827	11919	12020	12103	12170	12240	12290	13211	13303	13404	13487	13555	13626	13675
	50	11359	11447	11544	11623	11688	11756	11803	12687	12776	12874	12953	13018	13086	13133
	51	10918	11003	11096	11172	11234	11299	11345	12195	12280	12374	12450	12513	12578	12623
	52	10502	10583	10673	10746	10806	10869	10912	11730	11812	11902	11976	12036	12099	12142
	53	10109	10188	10274	10345	10402	10463	10505	11292	11371	11457	11528	11586	11646	11688
	54	9738	9814	9897	9965	10021	10079	10119	10877	10954	11037	11105	11161	11219	11260
	55	9387	9460	9540	9606	9660	9715	9754	10485	10559	10639	10705	10759	10815	10854
	56	9055	9126	9203	9266	9318	9372	9409	10114	10185	10263	10326	10378	10432	10470
	57	8740	8808	8883	8944	8994	9046	9082	9763	9831	9906	9967	10017	10069	10106
	58	8441	8507	8579	8638	8686	8736	8771	9429	9495	9567	9626	9675	9725	9760
	59	8158	8221	8291	8348	8394	8443	8477	9112	9176	9246	9303	9349	9398	9432
	60	7888	7949	8017	8072	8117	8164	8196	8811	8872	8940	8995	9040	9087	9120
	61	7632	7691	7756	7809	7853	7898	7930	8524	8584	8649	8703	8746	8792	8824
	62	7387	7445	7508	7559	7602	7645	7676	8251	8309	8372	8424	8467	8511	8541
	63	7155	7210	7271	7321	7362	7405	7434	7992	8048	8109	8159	8200	8243	8272
	64	6933	6987	7046	7094	7134	7175	7204	7744	7798	7857	7906	7946	7987	8016
	65	6721	6773	6831	6878	6916	6956	6984	7507	7560	7617	7665	7703	7743	7771
	66	6519	6570	6625	6671	6708	6747	6774	7282	7333	7388	7434	7471	7510	7537
	67	6326	6375	6429	6473	6509	6547	6573	7066	7115	7169	7214	7250	7288	7314
	68	6141	6189	6241	6284	6319	6356	6381	6860	6908	6960	7003	7038	7075	7101
	69	5965	6011	6062	6103	6137	6173	6198	6662	6709	6760	6802	6836	6871	6896
	70	5795	5840	5890	5930	5963	5998	6022	6473	6518	6568	6609	6642	6677	6701
	71	5633	5677	5725	5764	5797	5830	5853	6292	6336	6384	6424	6456	6490	6513
	72	5478	5520	5567	5605	5637	5669	5692	6119	6161	6208	6247	6278	6311	6334
	73	5329	5370	5416	5453	5483	5515	5537	5952	5994	6039	6077	6107	6139	6161
	74	5186	5226	5270	5307	5336	5367	5388	5792	5833	5877	5914	5943	5974	5996
	75	5048	5088	5131	5166	5195	5225	5246	5639	5678	5722	5757	5786	5816	5837
	76	4916	4955	4996	5031	5059	5088	5109	5491	5530	5572	5607	5635	5664	5684
	77	4790	4827	4868	4901	4928	4957	4977	5350	5387	5428	5462	5489	5518	5538
	78	4668	4704	4744	4776	4803	4831	4850	5213	5250	5290	5323	5349	5377	5397
	79	4550	4585	4624	4656	4682	4709	4728	5082	5118	5157	5189	5215	5242	5261
	80	4437	4472	4509	4540	4566	4592	4611	4956	4991	5029	5060	5085	5112	5130
	81	4328	4362	4399	4429	4454	4479	4497	4834	4868	4905	4936	4960	4986	5004
	82	4223	4256	4292	4322	4346	4371	4388	4717	4750	4786	4816	4840	4865	4883
	83	4122	4154	4189	4218	4242	4266	4283	4604	4636	4672	4701	4724	4749	4766
	84	4025	4056	4090	4118	4141	4165	4182	4495	4527	4561	4589	4612	4636	4653
	85	3930	3961	3994	4022	4044	4068	4084	4390	4421	4455	4482	4505	4528	4544
	86	3840	3869	3902	3929	3951	3974	3990	4289	4319	4352	4378	4400	4423	4439
	87	3752	3781	3813	3839	3861	3883	3898	4191	4220	4252	4278	4300	4322	4338
	88	3667	3695	3727	3752	3773	3795	3810	4096	4125	4156	4182	4203	4225	4240
	89	3585	3613	3643	3669	3689	3710	3725	4004	4032	4063	4088	4109	4130	4145
	90	3506	3533	3563	3587	3607	3628	3643	3916	3943	3973	3998	4018	4039	4053
	91	3429	3456	3485	3509	3529	3549	3563	3830	3857	3886	3911	3930	3951	3965
	92	3355	3381	3410	3433	3452	3472	3486	3747	3774	3802	3826	3845	3865	3879
	93	3283	3309	3337	3360	3378	3398	3412	3667	3693	3721	3744	3763	3783	3796
	94	3214	3239	3266	3289	3307	3326	3339	3590	3615	3642	3665	3683	3702	3716
	95	3146	3171	3198	3220	3238	3256	3270	3515	3539	3566	3588	3606	3625	3638
	96	3081	3105	3131	3153	3171	3189	3202	3442	3466	3492	3514	3531	3550	3563
	97	3018	3042	3067	3088	3106	3124	3136	3371	3395	3421	3442	3459	3477	3490
	98	2957	2980	3005	3026	3043	3060	3072	3303	3326	3351	3372	3389	3406	3419
	99	2897	2920	2945	2965	2981	2999	3011	3236	3259	3284	3304	3321	3338	3350
	100	2840	2862	2886	2906	2922	2939	2951	3172	3194	3218	3238	3255	3271	3283
	101	2784	2805	2829	2849	2864	2881	2893	3109	3131	3155	3175	3190	3207	3219
	102	2729	2751	2774	2793	2809	2825	2836	3049	3070	3093	3113	3128	3144	3156
	103	2677	2697	2720	2739	2754	2770	2781	2990	3011	3034	3052	3068	3084	3095
	104	2625	2646	2668	2687	2702	2717	2728	2933	2953	2976	2994	3009	3025	3036
	105	2576	2596	2618	2636	2650	2666	2676	2877	2897	2919	2937	2952	2967	2978
	106	2527	2547	2568	2586	2601	2616	2626	2823	2843	2864	2882	2897	2912	2922
	107	2480	2500	2521	2538	2552	2567	2577	2770	2790	2811	2828	2843	2857	2868

Fig. 8 — Tubular driveshaft critical speed

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5½″							6″						
.238	.203	.165	.134	.109	.083	.065	.238	.203	.165	.134	.109	.083	.065
27038	27210	27398	27553	27678	27809	27901	29602	29775	29964	30119	30244	30376	30467
25596	25759	25937	26084	26202	26326	26413	28024	28187	28366	28513	28632	28756	28842
24267	24421	24590	24729	24841	24959	25041	26568	26723	26893	27032	27144	27262	27344
23038	23185	23345	23477	23584	23695	23773	25223	25370	25531	25663	25770	25882	25960
21901	22040	22193	22318	22419	22525	22599	23978	24118	24271	24396	24498	24604	24678
20845	20978	21123	21242	21339	21440	21511	22822	22956	23101	23221	23317	23419	23489
19865	19991	20129	20243	20335	20431	20498	21749	21875	22014	22128	22220	22317	22384
18951	19072	19204	19312	19400	19492	19556	20749	20870	21002	21111	21199	21291	21355
18100	18215	18341	18444	18528	18616	18677	19816	19932	20058	20162	20246	20334	20395
17304	17414	17535	17634	17714	17798	17856	18945	19056	19177	19276	19356	19440	19499
16560	16665	16781	16875	16952	17032	17088	18131	18236	18352	18447	18524	18604	18660
15863	15964	16074	16165	16238	16315	16369	17367	17469	17579	17670	17744	17821	17875
15209	15306	15412	15498	15569	15643	15694	16651	16748	16855	16942	17012	17086	17138
14594	14687	14789	14872	14940	15011	15060	15979	16072	16174	16257	16325	16396	16445
14016	14106	14203	14283	14348	14416	14464	15346	15435	15533	15614	15679	15747	15794
13472	13558	13652	13729	13791	13856	13902	14750	14836	14930	15007	15070	15135	15181
12959	13041	13132	13206	13266	13329	13372	14188	14271	14361	14436	14496	14559	14602
12475	12554	12641	12712	12770	12830	12873	13658	13737	13824	13896	13954	14015	14057
12017	12093	12177	12246	12301	12360	12400	13157	13233	13317	13386	13442	13500	13541
11584	11658	11738	11804	11858	11914	11953	12682	12756	12837	12904	12958	13014	13053
11174	11245	11323	11387	11438	11493	11530	12234	12305	12383	12447	12499	12553	12591
10785	10854	10929	10991	11041	11093	11129	11808	11877	11952	12014	12064	12117	12153
10416	10483	10555	10615	10663	10714	10749	11404	11471	11544	11603	11652	11702	11737
10066	10130	10201	10258	10305	10354	10388	11021	11085	11156	11213	11260	11309	11343
9734	9796	9863	9919	9964	10011	10044	10657	10719	10787	10843	10888	10935	10968
9417	9477	9543	9596	9640	9686	9718	10310	10370	10436	10490	10534	10580	10611
9116	9174	9237	9289	9332	9376	9407	9980	10039	10102	10154	10197	10241	10272
8829	8885	8946	8997	9038	9081	9110	9666	9722	9784	9835	9876	9919	9948
8555	8609	8669	8718	8758	8799	8828	9366	9421	9481	9530	9569	9611	9640
8294	8347	8404	8452	8490	8530	8558	9080	9133	9191	9239	9277	9318	9346
8044	8096	8152	8198	8235	8274	8301	8807	8859	8915	8961	8998	9037	9064
7806	7856	7910	7955	7991	8029	8055	8546	8596	8651	8695	8732	8770	8796
7578	7626	7679	7722	7758	7794	7820	8297	8345	8398	8442	8477	8514	8539
7360	7407	7458	7500	7534	7570	7595	8058	8105	8156	8199	8233	8269	8293
7151	7197	7247	7287	7321	7355	7379	7829	7875	7925	7966	7999	8034	8058
6951	6995	7044	7084	7116	7150	7173	7610	7655	7703	7743	7776	7809	7833
6759	6802	6850	6888	6920	6952	6975	7401	7444	7491	7530	7561	7594	7617
6576	6617	6663	6701	6731	6763	6785	7199	7241	7287	7325	7355	7387	7409
6399	6440	6484	6521	6551	6582	6603	7006	7047	7091	7128	7158	7189	7211
6230	6269	6313	6348	6377	6407	6428	6820	6860	6904	6939	6968	6999	7020
6067	6105	6148	6182	6210	6240	6260	6642	6681	6723	6758	6786	6816	6836
5910	5948	5989	6023	6050	6079	6099	6471	6508	6550	6584	6611	6640	6660
5760	5796	5836	5869	5896	5924	5943	6306	6343	6383	6416	6443	6471	6490
5615	5650	5689	5722	5748	5775	5794	6147	6183	6222	6254	6280	6308	6327
5475	5510	5548	5579	5605	5631	5650	5994	6029	6068	6099	6124	6151	6170
5341	5375	5412	5443	5467	5493	5511	5847	5881	5919	5949	5974	6000	6018
5211	5244	5281	5311	5335	5360	5378	5706	5739	5775	5805	5829	5855	5872
5087	5119	5154	5183	5207	5232	5249	5569	5601	5637	5666	5690	5714	5732
4966	4998	5032	5061	5084	5108	5125	5437	5469	5504	5532	5555	5579	5596
4850	4881	4915	4942	4965	4988	5005	5310	5341	5375	5403	5425	5449	5465
4738	4768	4801	4828	4850	4873	4889	5187	5217	5251	5278	5300	5323	5339
4630	4659	4691	4718	4739	4762	4777	5069	5098	5131	5157	5179	5201	5217
4525	4554	4585	4611	4632	4654	4669	4954	4983	5015	5041	5062	5084	5099
4424	4452	4483	4508	4529	4550	4565	4843	4872	4903	4928	4948	4970	4985
4326	4354	4384	4408	4428	4449	4464	4736	4764	4794	4819	4839	4860	4875
4231	4258	4288	4312	4332	4352	4367	4633	4660	4689	4714	4733	4754	4768
4140	4166	4195	4219	4238	4258	4272	4533	4559	4588	4612	4631	4651	4665
4051	4077	4105	4129	4147	4167	4181	4436	4462	4490	4513	4532	4552	4565
3966	3991	4019	4041	4060	4079	4092	4342	4367	4395	4418	4436	4455	4469
3883	3907	3934	3957	3975	3993	4007	4251	4276	4303	4325	4343	4362	4375
3802	3826	3853	3875	3892	3911	3924	4163	4187	4214	4235	4253	4272	4284
3724	3748	3774	3795	3812	3830	3843	4077	4101	4127	4149	4166	4184	4197
3649	3672	3697	3718	3735	3753	3765	3995	4018	4043	4064	4081	4099	4111
3575	3598	3623	3643	3660	3677	3689	3914	3937	3962	3983	3999	4017	4029
3504	3526	3551	3571	3587	3604	3616	3836	3859	3883	3903	3920	3937	3948
3435	3457	3481	3500	3516	3533	3545	3761	3783	3807	3826	3842	3859	3871
3368	3389	3413	3432	3448	3464	3476	3687	3709	3732	3752	3767	3784	3795
3303	3324	3347	3366	3381	3397	3408	3616	3637	3660	3679	3695	3711	3722
3240	3260	3283	3301	3316	3332	3343	3547	3568	3590	3609	3624	3640	3651
3178	3199	3221	3239	3254	3269	3280	3480	3500	3522	3540	3555	3571	3581
3119	3138	3160	3178	3192	3208	3218	3414	3434	3456	3474	3488	3504	3514
3061	3080	3101	3119	3133	3148	3158	3351	3370	3392	3409	3424	3438	3449

Fig. 9 — Tubular driveshaft critical speed

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*Critical speed values shown in Figs. 6-9 used with the permission of Welded Steel Tube Institute.

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Test Procedures

Chapter 11

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Chapter 11
Testing Procedures

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E. R. Wagner, P. E. — Saginaw Steering Gear Div., GMC

Introduction

The purpose of testing universal joints, other driveshaft components, and complete driveshaft assemblies is to perform a comprehensive evaluation of the individual component or assembly function, durability, and reliability characteristics. The evaluation of a driveshaft by laboratory or vehicle testing is most important, because such tests will yield results which substantiate that the design can or cannot perform satisfactorily in the actual application. Generally, testing procedures are oriented towards an accelerated simulation to define these desired characteristics.

There are many types of laboratory and vehicle tests that can be used to substantiate or predict the performance of a universal joint or driveshaft assembly. Laboratory testing facilities or equipment of various types are available that can be programmed to simulate the actual operating modes to which the component is subjected. However, experience is necessary to plan and organize a laboratory test program and to interpret the results as to their acceptability for the particular requirement.

When a vehicle performance can be measured, a duty cycle set up or analytically established, it is within the scope of existing test facility capability to execute a controlled test program that will duplicate these requirements in the laboratory. For example, a particular test can be performed at fixed conditions of torque, speed, and joint angle, and carried out at several different fixed test conditions. A life curve can be established from this data which can be used to evaluate other operating conditions. Analytical procedures can be used to extrapolate existing known test results and, from this data, a design can be evaluated for a different requirement.

The integrity, control, and physical and mechanical properties of the component being tested can contribute significantly to the value of the test. Therefore, when conducting a test, consideration must be given to the variations in the dimensions and the mechanical properties of the test sample or specimen. Whenever possible, it is good practice to carefully record and identify all applicable specimen characteristics before the test. In addition, a specimen should be thoroughly analyzed metallurgically after the test has been completed. Metallurgical characteristics should be evaluated in conjunction with the test results. Nondestructive metallurgical tests are also beneficial before a specimen is tested. Test facilities should be designed so that the test conditions can be calibrated and, if possible, monitored during the test.

Many different types of tests are necessary to establish that a particular universal joint is an acceptable design for the intended application. Universal joints are applied in many different types of physical and environmental applications. In some applications, the joint is subjected to a steady state torque load at a given speed and joint angle. In others, the torque, speed, and joint angles may all be subject to variation. In still other applications, there are relatively high torque excursions or shock peaks which cause a reduction in bearing and fatigue life of the joint components. Some joints operate in relatively clean environments under normal temperature conditions, while other joints may be required to operate submerged in great depths of sea water. For a given joint, a normal operating environment may be in a relatively hot atmosphere with a heavy concentration of abrasive dust and dirt. This same joint may also be exposed to water, salt, mud, snow, and extreme temperatures ranging from hot to cold.

In general, the purpose of the joint is to transmit power through an angle and withstand exposure to the many adverse environmental conditions. Therefore, it is important when planning or organizing tests that suitable test programs be selected, which by proper simulation of actual operating conditions will produce meaningful results. However, due to the many types of applications and environments, it is economically impossible to conduct simulated laboratory tests for each application. Laboratory test procedures are no substitute for actual vehicle experience, whether obtained under controlled test conditions or in actual field usage. Actual vehicle tests using production components and which incorporate production variables provide the most comprehensive evaluation of the design.

Laboratory Testing

Laboratory tests for complete driveshaft assemblies, universal joints, or other driveshaft components are generally classified in categories and described as follows:

Ultimate Torque Test

In this test, a torque load is applied statically at a controlled rate to a driveshaft assembly or universal joint until a failure occurs. The specific purpose of this test is to identify the failure point in the assembly or component. Fig. 1 illustrates a typical torsion test machine.

Common torsion test machines are designed so that the rate at which the torque load is applied may be controlled as desired. These torsion test machines are also equipped with a means for plotting on graph paper the torque load versus angular deflection. This graph can then be used to establish the yield point by commonly used techniques such as the 0.1% offset method or the Johnson's Apparent Elastic Limit (JAEL) method described in *Appendix L*.

In a typical universal joint ultimate torque test, the yield point is generally difficult to distinguish from the ultimate failure point

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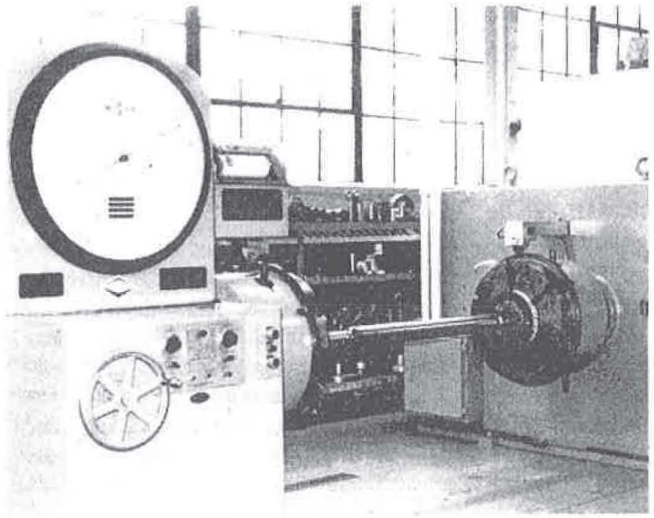


Fig. 1 — Torsion test machine

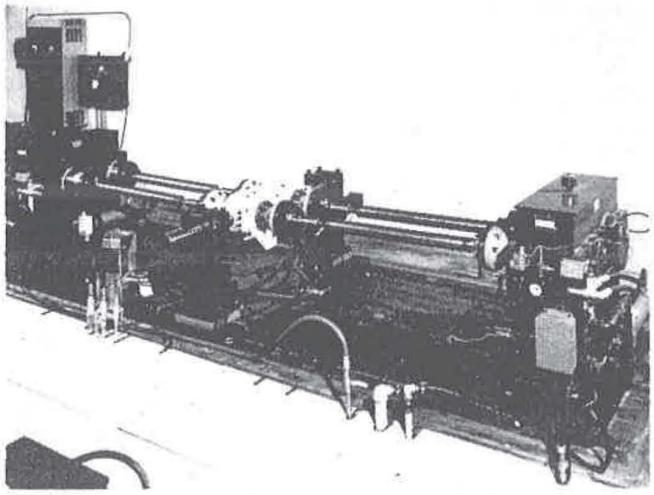


Fig. 2 — Four-square wear test facility

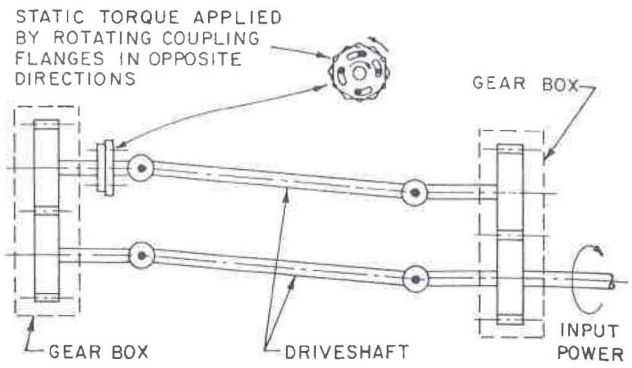


Fig. 3 — Four-square wear test machine with static torque application means

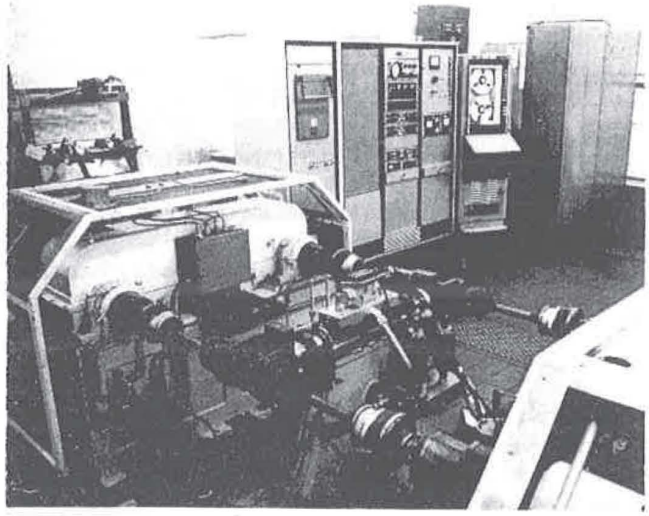


Fig. 4 — Four-square test facility for axle driveshafts

because of the relatively high material hardness and because the two values are nearly identical.

Wear Test

A universal joint has bearing contact surfaces which are necessary for the performance of its required function and are, therefore, subject to wear. These bearings may be of the plain or the rolling element bearing types. In order to investigate or determine the wear characteristics of these bearings, the universal joint must be operated at fixed conditions of torque, speed, and joint angle, or at variable torques, speeds, and joint angles.

A stress analysis and a study of the capabilities of the joint should be made before establishing a test procedure or program. The purpose of the test may be to evaluate the joint design for a specific application or to develop empirical data for future reference. A Weibull analysis procedure can be used to evaluate test results and to establish a life expectancy curve. The confidence level of the analysis or the curve depends on the accuracy and reliability of the test data developed.

The most economical test facility for investigating or evaluating the wear characteristics of a universal joint is a typical four-square test machine shown in Fig. 2. Usually, two or four driveshafts are tested simultaneously in a closed loop arrangement formed by these driveshafts and two gear boxes which are located at each end of the machine. These gear boxes commonly have one-to-one gear ratios or they can have a different ratio as long as both gear boxes have identical ratios. The gear boxes are generally located on a bed plate so that they can be positioned as required to accommodate various driveshaft lengths and joint angles and then locked down. When testing driveshafts with non-constant velocity joints, the joint angles must be equal at both ends or be selected to provide adequate cancellation.

Fig. 3 illustrates schematically a simple two station, four-square configuration. The test load is applied statically by twisting one shaft relative to its mating shaft to a prescribed torque and then locking the two shafts together as shown. In more complex designs, the torque can be applied dynamically through hydraulic or electric circuits. The system is driven by an electric motor located externally from the closed loop. The common four-square facility can test driveshafts at a constant torque, speed, and joint angle.

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Also, there are more elaborate four-square facilities that can test driveshafts at variable torque, speed, and joint angles to simulate actual operating conditions. Generally, these machines are tape-controlled and thus typical vehicle test conditions can be accurately duplicated in the laboratory with a duty cycle obtained from an instrumented vehicle operating on an actual test run or schedule. Such a four-square test facility is shown in Fig. 4 for testing axle driveshafts for front wheel drive vehicles.

It is extremely important that the proper torque load be maintained in the closed loop of the four-square facility. In a simple four-square machine, some load drop may occur as the test progresses. If too great a load drop is permitted, the test accuracy may be questionable. The simplest static load measuring devices are torque wrenches, levers loaded by weights, and beam scales. Torque cells and torque transducers are more elaborate load measuring devices which are installed in the closed loop system to monitor the torque load as desired.

Also, in a typical wear test the temperatures of the universal joints are often monitored to measure the progress of wear. Infrared radiation sensors and various types of pyrometers can be used to measure temperature during the course of the test.

It is good practice to incorporate in the test facility, devices for terminating the wear test. The test can be terminated by load sensing devices which shut off the test facility when the torque either exceeds or drops below a specified range. Also, the test can be terminated when the universal joint temperature exceeds a specified maximum level as monitored by the temperature sensing devices.

In evaluating the results of a typical four-square wear test, the variations in mechanical properties and dimensional tolerances of the specimens can produce a spread in test results which may be confusing. Two different procedures can be used to analyze these test results. One procedure is to specify that the specimen must operate for a given number of hours at fixed conditions of torque, speed, and joint angle with no surface fatigue or damage. Another procedure is to specify that the specimen must operate for a given number of hours at fixed conditions of torque, speed, and joint angle and still be operable at the end of this period.

The most accurate test results are produced with constant conditions of torque, speed, and joint angle, and the data thus obtained is most effectively utilized in establishing useful empirical data. The most accurate life curves are developed when the point of failure is accurately controlled and the mechanical and dimensional characteristics of the specimens are reasonably similar.

In addition to commonly used four-square facilities, a dynamometer setup can also be utilized to wear test universal joints. The test has to be properly controlled and monitored and a procedure has to be defined to evaluate the test results.

Fatigue Test

A driveshaft and its components such as the universal joints are subjected to repeated torsional loads of varying magnitude and direction during actual service. Depending on the specific application, these loading conditions can consist of zero to maximum loads, completely reversed loads, or mean plus alternating loads.

A fatigue test can be conducted so as to simulate the loading conditions encountered. It is preferable to duplicate the loads and stresses developed in the application. However, if the actual loading data is not available, a fatigue test can be performed and the test results used to develop S-N curves that can be useful in predicting fatigue life. To assist in developing S-N curves, analytical and graphical procedures, such as Weibull analyses and Goodman diagrams may be used to analyze the fatigue test data.

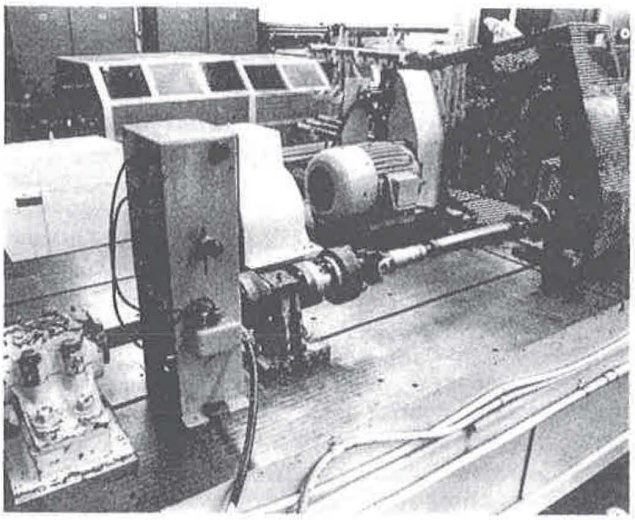


Fig. 5 — Mechanical type fatigue test fixture

A fatigue test should produce meaningful and reliable accelerated test results in a reasonable period of time. Therefore, the fatigue test should be conducted at a reasonable loading frequency. For a completely reversed load type test, 250 cpm has generally been the accepted maximum loading frequency for a typical driveshaft. Load application frequencies higher than approximately 500 cpm may cause excessive heat in the test specimens. For a zero or a slight preload to maximum load type fatigue test, load application frequencies of approximately 1800 cpm can generally be utilized without causing excessive heat build-up in specimens.

The design characteristics of certain driveshafts can also be important in the selection of suitable loading frequencies for a fatigue test. Examples of such driveshafts are those shafts incorporating rubber element tube designs. If the torsional loading cycles are applied at a relatively high frequency to these devices, the excessive temperature increase due to the hysteresis characteristics of the elastomer will result in a loss of fatigue life. Therefore, the load cycles are generally slowed down in a test to maintain the temperature rise at reasonably low levels. As discussed in the *Rubber Element Tubes* section of the chapter on *Vibration Control,* load cycle frequencies in the typical range of 3-10 cpm have produced satisfactory results.

A fatigue test can be performed as desired with numerous devices which may apply the load either with mechanical, electrical, or hydraulic means, or with a system that can be a combination of two of these basic means. A typical mechanical type fatigue test fixture testing a driveshaft as shown in Fig. 5 may consist of simple pedestals positioned on a bed plate to hold the specimen so that it can be oscillated back and forth angularly with a bell crank mechanism located at one end of the test fixture.

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Usually, this mechanism is attached to a flywheel driven by an electric motor through a suitable gear reduction that provides the desired oscillating or cycling speed.

At the opposite end of the test fixture, the oscillating specimen reacts against an adjustable length torsion bar to which it is attached to provide the desired torque load in a suitable angular deflection range. The magnitude of the test load for a given oscillating angle can be varied by changing the effective length of the torsion bar. The desired load application characteristic of the test cycle can be incorporated to simulate any of the three loading conditions previously described simply by changing the lengths of the arms of the bell crank mechanism.

Other fatigue test equipment that can provide the basic means for oscillating or vibrating a specimen are hydraulic rotary actuators, magnetic shakers, Sonntag type constant load testers, resonant machines, and electro-hydraulic test systems. The electro-hydraulic test systems can be quite elaborate. Specimens can be fatigue tested by such equipment which may be programed or tape-controlled. The data on the tape can represent actual field data obtained from a typical instrumented vehicle operating under actual service conditions. Thus, the actual service conditions can be duplicated in the laboratory.

Whatever fatigue test system is used, it is important that an accurate means be provided for measuring and calibrating the applied load. The fatigue test fixture should have permanently installed strain gages or strain gaged devices, such as load or torque cells, an amplifier, and an oscilloscope so that uniform test results can be produced.

In addition, the fatigue test fixture should be designed so that the load measuring system can be checked by means of dead weights. This procedure can be accomplished with a jack and a platform scale.

It is also good practice to incorporate in the test fixture a device for shutting off the fatigue test. This procedure can be accomplished mechanically or by means of strain gages. The strain gage signal can be monitored and, when the stress level drops, the test can be terminated. This procedure is advisable because it prevents catastrophic failures, destruction of possible valuable test evidence, and potential damage to test equipment.

Fatigue testing can be conducted on components or on complete assemblies. A fatigue test of a complete driveshaft assembly, including the end yokes, is preferable.

Whenever possible, it is good practice to magnetic particle inspect all components to be tested. After the test, components that have not failed should be magnetic particle inspected again.

To add more value to the fatigue test, it is suggested that a metallurgical analysis be conducted of the components tested. Metallurgical analyses are costly and time consuming. However, the greater the scope of the analysis, the more meaningful the tests are for evaluation of the test results and for future reference.

Environmental Test

During actual service, the driveshaft may be required to operate in extreme or adverse environmental conditions as dictated in a given application. In order that the vital driveshaft components such as universal joints, slip splines, and shaft support bearings may function satisfactorily in any environmental condition of the application, provision must be made for adequate sealing means.

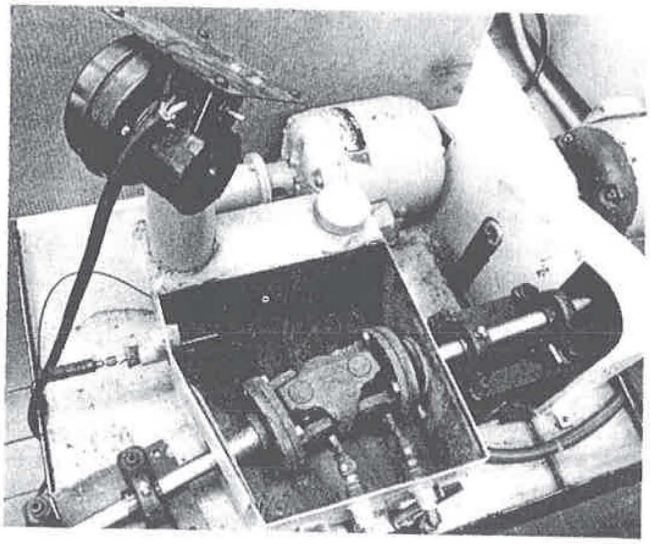


Fig. 6 — Environmental test chamber for double Cardan joint

The effectiveness of the sealing elements can be evaluated by an environmental test. Since there are numerous and widely varying environmental condition possibilities, the necessary environmental test procedure may vary considerably from one application to another. In addition, the functional characteristics of the component may influence the environmental test.

The control and evaluation of an environmental test is difficult and requires practical experience. When a universal joint, for example, operates in a hot abrasive environment, the environmental test should simulate such operating conditions. To investigate the effect of the abrasive environment, the joint should be operated at a proper rotating speed and angle in an enclosure or chamber where the atmosphere would be laden with the abrasive material. The concentration of the abrasive material and the air temperature should be properly controlled.

Also, it may be necessary to vary the temperature of the joint in order to produce the effect of heating and cooling it. This temperature change characteristic can be evaluated by an effective water test procedure where the joint is operated at an angle in a heated environment and then submerged in cold water. The rapid cooling of the joint reduces the pressure inside the joint and permits water to seep through or past the seals.

Fig. 6 illustrates a typical automotive driveshaft type double Cardan joint in an environmental test chamber. The joint is operated without a torque load at a fixed angle, while it is subjected to a test cycle consisting of selected alternating combinations of speed, temperature, and salt spray. The test cycle is repeated until the required total test time has been accumulated. In this test, the joint bearing seals as well as the centering mechanism seal are evaluated.

In addition to evaluating seals on Cardan and double Cardan type joints, the seals used on other types of universal joints, such as typical ball type constant velocity joints, must also be evaluated. Generally, these seals are of the boot seal design and are used on both fixed center and end motion type CV joints. The application requirements and functional characteristics of these CV joints differ considerably from the Cardan and double Cardan

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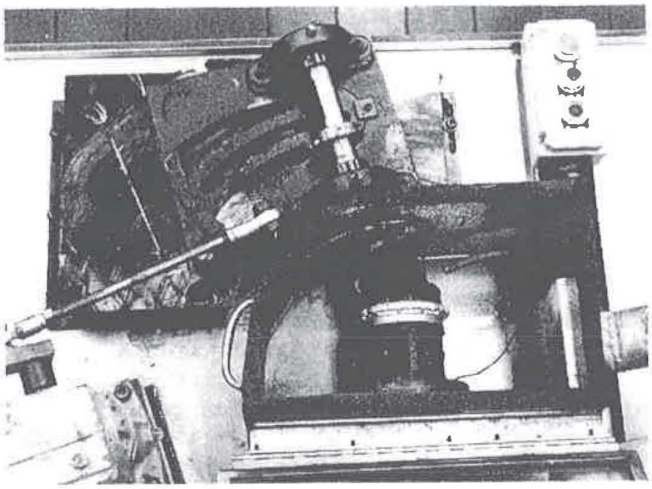


Fig. 7 — Environmental flex life test chamber for end motion type CV joint

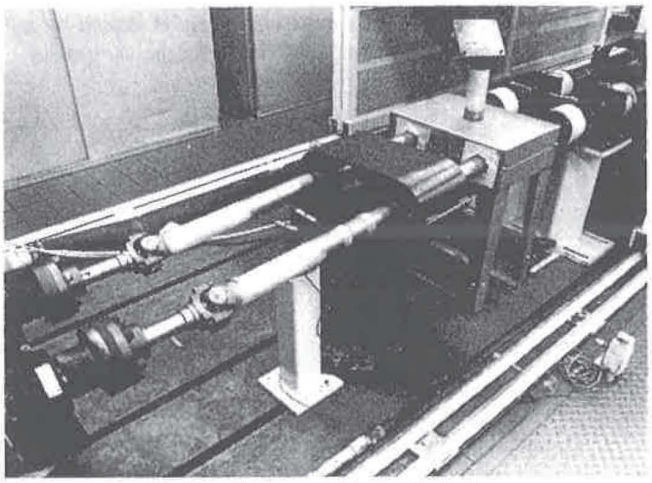


Fig. 8 — Environmental wear test setup

joints. Therefore, the boot seals are usually evaluated by environmental test procedures which also differ considerably.

The fixed center type CV joint is commonly used as the outboard joint in axle driveshafts for front drive vehicles where a relatively high joint angle is required. In a typical environmental test, this joint is rotated at a desired rotational speed and rocked back and forth through its required angular operating range at a given rocking frequency. When operated in this manner, the joint is subjected to selected alternating temperature extremes for given periods of time to comprise a desired test cycle. This test cycle is repeated until the required total test time has been accumulated. Since the boot seal must flex when the joint is performing its functions, this type of test is more correctly described as an environmental flex life test.

The end motion type CV joint is commonly used as the inboard joint in axle driveshafts for front wheel drive vehicles where axial movement and a relatively low joint angle are required. In addition, this joint is generally used at outboard and inboard locations in axle driveshafts for rear wheel drive vehicles with independent rear suspension.

A typical environmental flex life test for an end motion type CV joint is similar to that for a fixed center type CV joint, except that the joint is not rocked through as large an angular operating range. Since this joint must also provide end motion, the required axial stroking of the joint is, therefore, incorporated in the environmental test procedure. Fig. 7 shows a typical end motion type CV joint in an environmental flex life test chamber.

In general, an environmental test is conducted on a driveshaft component, such as a universal joint, without a torque load. Depending on the application requirements, it may be desirable to combine a typical Cardan or double Cardan joint wear test, for example, with an environmental test. Such a test can be described as an environmental wear test where the combined effects of environment and wear can be evaluated. The test is conducted in a typical four-square test facility as shown in Fig. 8 where some of the joints are required to operate enclosed in environmental chambers. Thus, joints can be evaluated for wear simultaneously with and without the effects of environment.

Axial Stroking Test

In this test, the axial thrust force and wear life characteristics of splines and various other axial motion devices can be evaluated. In a typical axial stroking test, the splined member or other axial motion mechanism is stroked at a suitable stroking speed when subjected to various torque loads. In a simple test fixture, the test specimen is not rotated while being tested. The chapter on *Splines and Axial Motion Devices* illustrates typical static (nonrotating) axial thrust force characteristics obtained from such testing procedures.

It is also possible to perform a wear life test in the same test fixture. This testing procedure can include controlled environmental conditions, if desired.

In order to determine dynamic axial thrust force characteristics of axial motion devices, elaborate test equipment is required. A four-square test facility such as shown in Fig. 4, may be used for that purpose. When evaluating typical end motion type universal joints, the dynamic axial thrust force characteristics can be determined for various combinations of torque, speed, and joint angle.

Grease Leakage Spin Test

In this test, the typical Cardan or double Cardan universal joint is operated at various speeds to determine if the joint bearing seals adequately retain the lubricant. A rotating speed can be reached where the centrifugal force effects will cause the lubricant to leak past the sealing elements. The test can be conducted, as desired, at normal or elevated temperatures.

It may also be desirable to evaluate the joint bearing seals by operating the joint at an angle because of the relative movement required between the seals and the bearings during rotation. In order to most effectively evaluate the centering mechanism seal for a double Cardan joint, the joint should be operated at a specified angle.

Boot Seal Spin Test

In this test, the boot seal of a typical ball type CV joint can be evaluated for its radial growth or expansion characteristics caused by centrifugal force at high rotational speeds. The boot seal

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should not interfere with vehicle components while performing its required functions in actual service.

Rubber Element Tube Driveshaft Static Torsional Rate Test

In this test, a rubber element tube driveshaft is installed in a typical torsion test machine. The purpose of the test is to determine the static torsional spring rate. The procedure for conducting this test is described in the *Rubber Element Tubes* section of the chapter on *Vibration Control*.

High Angle Torque Test

In this test, the universal joint, generally a high angle fixed center type, is rotated at an extremely low speed and approximately at or near its maxium operating joint angle. The joint is subjected to a typical test cycle consisting of a period of time at a relatively high torque load, alternating with a period of time at no load. The test cycle is repeated until the required number of cycles have been completed without failure of the joint.

The purpose of the test is to evaluate the strength of the joint while operating at a relatively high joint angle. The outboard universal joint of a typical front drive vehicle may be subjected to such a loading condition when operating at the full steer angle during turning maneuvers.

Vehicle Testing

The evaluation of a driveshaft is best accomplished under actual operating conditions of the specific application. Generally, the type of vehicle tests conducted on driveshafts by vehicle manufacturers depend on whether the particular vehicle is intended for on-highway or off-highway applications, or whether it can be a combination of both.

By means of appropriate testing procedures or schedules, manufacturers subject vehicles and individual components to a comprehensive evaluation of the durability characteristics of the complete vehicle. Since the driveshaft is a power transmitting member of the vehicle drivetrain, it is most effectively evaluated by a durability test schedule designed specifically for the various elements of a typical drivetrain system.

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Accessory Drives

Chapter 12

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Chapter 12
Accessory Drives

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Introduction

An accessory driveshaft is a driveshaft or driveline located between the power take-off (PTO) of mobile equipment and its auxiliary or accessory equipment which is being driven. Trucks and agricultural tractors are the most common examples of on- or off-highway vehicles that utilize accessory drives. The discussion of vehicle engine driven accessory power systems in this chapter will be limited to those systems on trucks which incorporate accessory drives requiring universal jointed driveshafts. The application of PTO drive systems on agricultural tractors and implements is discussed in the chapter on *Agricultural Implement Driveshafts*.

The areas of a truck's powerplant and drivetrain system which are most commonly used for power take-off are as follows:

1. Front of engine crankshaft
2. Engine flywheel
3. Transmission
4. Auxiliary Transmission
5. Driveline between transmission and rear axle

Due to increased mechanization and automation, PTO systems are needed on most trucks to supply power to operate various truck mounted material handling equipment, as well as any other accessory which requires a power source to perform its function. Typical applications which require a PTO system drive are pumps, compressors, concrete mixers, garbage or refuse packers, fire apparatus, hoists, snow plows, winches, dump bodies, tailgate loaders, and many others.

In a basic sense, a PTO is an auxiliary transmission which can be attached to a transmission or another auxiliary transmission. Other special PTO's are available that drive off the engine or are installed directly into the vehicle driveline. PTO systems, except for the flywheel type, are generally installed in trucks by accessory equipment builders. The flywheel PTO is usually installed by the vehicle manufacturer because field installation is difficult.

PTO Types and Their Typical Applications

The type of PTO required, including the design of the accessory driveshaft, is determined by the power demands of the particular accessory equipment under consideration. More specifically, the type of PTO required is determined by whether the chassis-mounted equipment requires power for a short period or for continuous extended operation. Further, some types of PTO's permit the accessories to be operated only when the vehicle is stationary, while other PTO types also allow the accessories to be operated at all vehicle traveling speeds. In addition, the PTO must be able to supply the necessary torque and speed required to power the driven unit. With regard to speed and direction of rotation, some applications may require single speed PTO's while others necessitate multispeed and reversible PTO's. Therefore, the torque and speed requirements of the specific application will dictate the type of PTO necessary and its location in the vehicle's power train or drivetrain system. These requirements will thus influence the design parameters for a driveshaft which must transmit the torque and speed required by the driven unit.

The most commonly used PTO types for truck power train or drivetrain systems and their characteristics are described briefly, together with some of their typical applications. This brief description will provide a basic understanding of the design parameters which must be considered when applying an accessory driveshaft with a specific type of PTO in a given application. These common types of truck PTO's are as follows:

Engine Crankshaft PTO

Taking power off the front of a truck engine has become a common means of driving truck-mounted equipment. This location for a PTO is a desirable power source, because it is available at a reasonably constant rotational speed regardless of vehicle road speed. It is not affected by clutch actuation or transmission gear ratios. Hence, the engine crankshaft PTO allows equipment operation at relatively high levels of performance, when desired, while the vehicle is stationary, or at all traveling speeds. On many truck engines, full engine horsepower is available.

When the accessory driveshaft is connected directly to the engine crankshaft by means of an adapter, it is subjected to engine induced torsional excitations resulting from the engine torque or firing impulses. Also, engine misfiring produces load impulses of a detrimental nature in the drive system.

These torsional resonance conditions within the normal operating range should be avoided, as much as possible, by introducing appropriate damping means such as torsional dampers, or couplings, or rubber element tube driveshafts. On Diesel engines such devices are generally a necessity.

PTO systems connected to the front of the engine crankshaft are used to power various equipment. Concrete mixers, body hoists, refuse packers, aerial ladders and platforms, and fire truck water pumps are typical applications of vehicle mounted equipment.

Fig. 1 shows schematically a driveshaft providing the drive for the pump of a typical hydraulic system which powers a refuse packer. Another method of providing power for vehicle mounted equipment is with an all-mechanical drive system illustrated schematically in Fig. 2. In this typical design, the equipment is connected through a series of driveshafts, with intermittent support shafts and bearings alongside one of the truck frame rails, to the front of the engine crankshaft.

Still another method of driving vehicle mounted equipment

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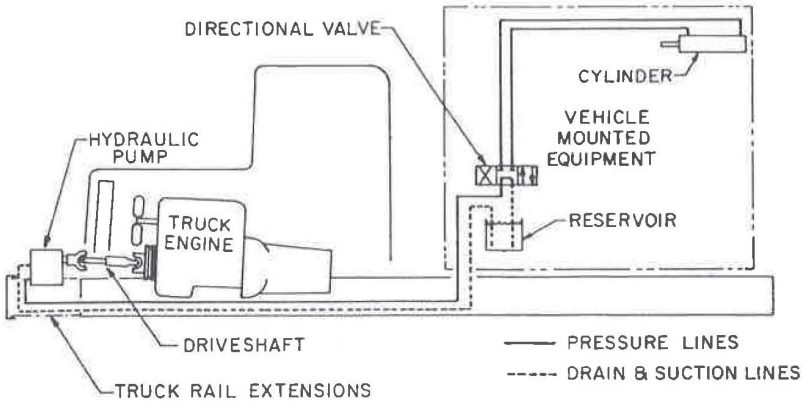


Fig. 1 — Typical engine crankshaft PTO with hydraulic drive system

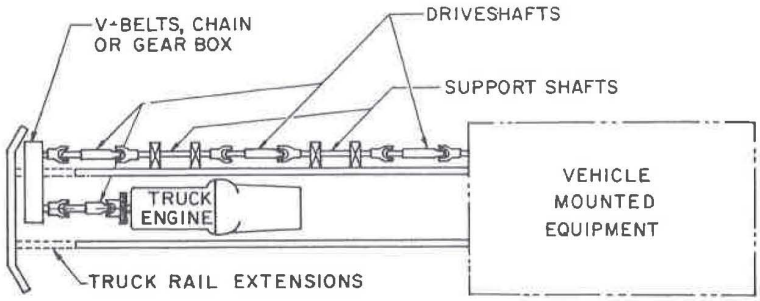


Fig. 2 — Typical engine crankshaft PTO with mechanical drive system

from the front of the engine is by means of direct drive. A typical example of this type of equipment is a centrifugal water pump mounted on the frame rail extensions of a fire truck. The pump is usually combined with a gear reducer which increases the truck engine speed to the desired higher pump operating speed. The pump and gear unit combination is connected to the front of the engine crankshaft with a driveshaft.

Engine Flywheel PTO

As in the case of a PTO connected to the front of the engine crankshaft, a flywheel PTO drive will transmit power whenever the engine is running. Consequently, the vehicle mounted equipment can be operated regardless of whether the vehicle is moving or not. Flywheel PTO units are typically located and mounted between the engine and transmission directly behind the engine flywheel housing. This type of PTO can provide a relatively short driveline system to certain driven equipment under consideration and is less vulnerable to collision damage than a PTO mounted at the front of the engine.

The various engine induced torsional vibration problems, particularly those associated with Diesel engines, must be considered and compensated for in the design of the PTO drive. These torsional resonance conditions and the damping means for controlling these disturbances were previously described briefly for a PTO connected to the front of the engine crankshaft.

Flywheel connected PTO systems have been used for providing power to concrete mixers, blowers for unloading dry cement, construction equipment, and so forth.

Transmission Mounted PTO

The transmission is a common area in a vehicle's drive system for locating a PTO, to direct part of the engine power to accessory equipment. This transfer of engine power is typically accomplished through a gear in the transmission, designed as the PTO driver gear for the PTO.

PTO units are available in many shapes and sizes for mounting to the side, bottom, or top of a truck transmission. They are commonly built to SAE standardized six- and eight-bolt mounting configurations.

With many types of transmission mounted PTO's being manufactured, a wide range of torques and speeds are available to satisfy the truck user's needs in providing power to a wide variety of accessory equipment. By the proper selection of a transmission and PTO combination, the performance and service life of the intended accessory drive can be optimized. Some PTO's can be operated only when the vehicle is stationary, with the transmission in neutral, while other PTO's may be operated also when the truck is in motion.

Transmission mounted PTO's have been used to provide power for hydraulic pumps, fluid transfer pumps, blowers, generators, compressors, cranes, winches, hoists, tailgate loaders, fire truck equipment, refuse and garbage packer bodies, and numerous

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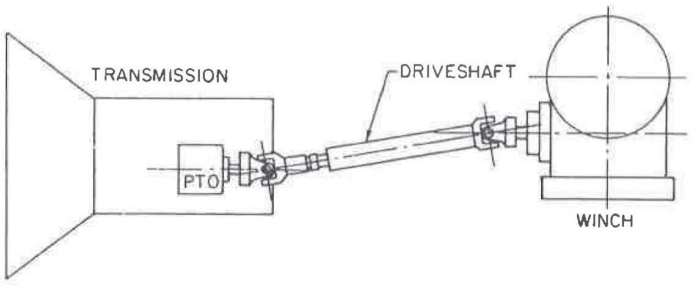


Fig. 3 — Typical transmission side mounted PTO with direct drive arrangement

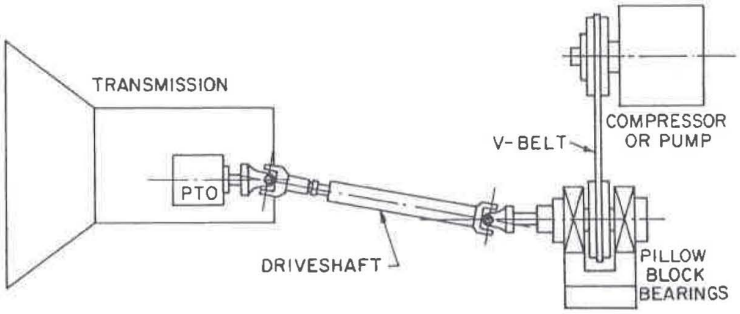


Fig. 4 — Typical transmission side mounted PTO with indirect drive arrangement

other applications. Fig. 3 shows schematically a transmission side mounted PTO driving a winch through a driveshaft in a typical direct drive arrangement. A typical indirect drive arrangement is illustrated in Fig. 4 where a compressor or pump is provided with power from a PTO through a driveshaft and V-belt drive system. The indirect drive arrangement is often used when the speed required by the driven equipment is different than the output speed of the PTO. Since torque varies inversely with the speed, it is essential that the drive system be carefully analyzed so that a correctly rated PTO is used which will provide the necessary torque at the accessory unit input shaft. This brief description for transmission mounted PTO's can also apply, in general, to PTO's mounted on auxiliary transmissions.

Split-Shaft PTO

A PTO, located in the truck driveline between the transmission and rear axle, is commonly referred to as a split-shaft PTO. This type of PTO can supply power simultaneously to both the vehicle's driving wheels and to the driven accessories. Necessary shifting means can be incorporated so that operation of these accessories will not interfere with the normal road operation of the vehicle. The split-shaft PTO is a mechanical designed gear box and its basic function is to divert the engine power from the normal use of propelling the vehicle, to driving some mechanical or hydraulic operated accessory equipment.

The split-shaft PTO has more power transmitting potential than transmission mounted PTO's, for example, since all the ratios of the main transmission as well as the ratios of the PTO itself are available. Therefore, the split-shaft PTO can provide the power required to operate the larger accessories. Some split-shaft PTO units also have two or three openings for mounting other PTO's or shift controls to allow several accessories to be operated with the truck either stationary or traveling. These features provide added versatility to users of vehicle mounted accessory equipment.

Split-shaft PTO's have been used to provide power for winches, post hole diggers, fuel tankers and large compressors, generators, or hydraulic pumps. Fig. 5 illustrates schematically a typical installation of a split-shaft PTO providing power for a hydraulic pump.

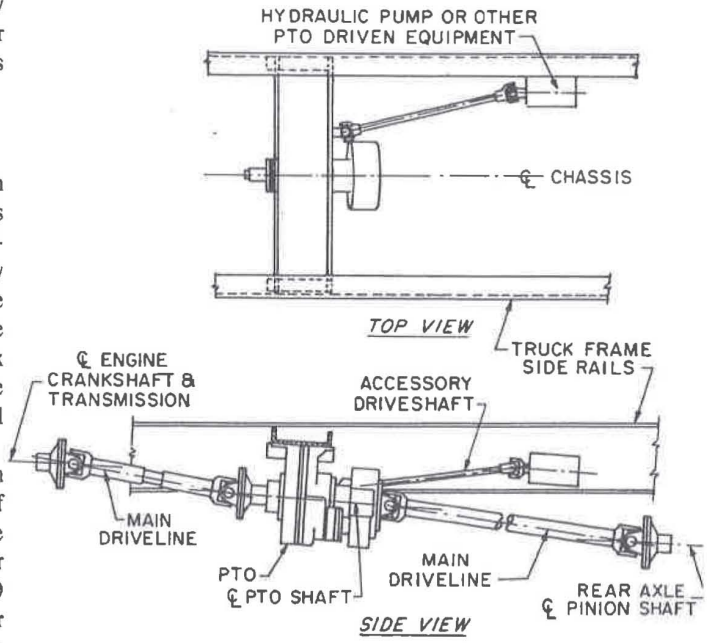


Fig. 5 — Typical split-shaft PTO

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