Design Parameters

The design of an accessory driveshaft requires consideration of the application parameters for the intended accessory drive application. Application parameters and established practices for driveshafts, in general, are described in detail in the chapter on *Basic Driveline Requirements*. The various parameters which apply specifically to accessory driveshafts are as follows:

Frequency of Equipment Operation

The frequency of operation of the accessory equipment, whether intermittent or continuous, can significantly influence the design of the driveshaft, for adequate life or durability. A knowledge of this frequency of operation is especially important in the selection or design of the driveshaft's universal joints and slip splines which must have adequate capacity to meet the life requirements.

Torque

The accessory driveshaft may be subjected to various torsional loading modes during the performance of the accessory's function. Generally, most applications involve some degree of loading above the normal torque required to operate the accessory equipment. The degree of this above normal loading is influenced not only by the operating characteristics of the particular equipment but also by the type of PTO required and its location in the vehicle's power train or drivetrain system.

For example, any equipment driven directly by a typical reciprocating engine, such as a gasoline or Diesel engine, is subject to various engine induced torsional shock loads described previously for engine crankshaft and engine flywheel PTO's. These shock effects decrease in severity with an increase in the number of cylinders, since the engine torque or firing impulses per revolution are related to the number of cylinders. Reciprocating driven equipment, such as compressors and pumps, may also induce severe loadings of this type.

Inertia loading is another form of above normal loading which can be developed by certain equipment, such as booms and winches, when lifting a load and in many variable speed drives. Severe shock loading can be induced by equipment that is stopped or slowed down suddenly, such as earthboring augers. Further, severe shock loads can be imposed by clutches which are often incorporated in accessory drive systems to engage and disengage equipment from the power source.

Therefore, the entire accessory drive system must be carefully analyzed so that the maximum applied torque on the driveshaft can be determined. Appropriate shock load factors can be used as a means of estimating this maximum applied torque. These shock load factors for a particular application should be based upon judgement and experience.

Speed and Universal Joint Angles

The operating speed of the driveshaft depends on the required input speed of the driven accessory equipment. The critical speed of the driveshaft must be higher than the maximum design rotational speed of the accessory drive system.

For a typical one piece driveshaft connecting a PTO with the driven equipment, the output shaft of the PTO and the input shaft of the driven equipment should be parallel or approximately parallel for an optimum installation, so that equal or near equal universal joint angles are obtained. Since Cardan universal joints are commonly used on accessory driveshafts, the two yokes on the connecting shaft should also be properly phased. The permissible size of the joint angles depend on the operating speed. As the operating speed decreases downward, the joint angle may increase, and as the speed increases, the joint angle must decrease.

Table 1 may be used as an application guide for determining permissible operating angles for an application. In this table, the angles and speeds are continuous conditions for a one piece, two-joint driveshaft with equal angles. Also, these angles and speeds are based on a maximum acceleration limit of 1000 rad/s^2.

Table I—Allowable Operating Joint Angles

Driveshaft Speed, rpm	Maximum Operating Angle, Deg
5000	3°30′
4500	3°50′
4000	4°20′
3500	5°00′
3000	5°45′
2500	6°55′
2000	8°40′
1500	11°30′

Length Changes

Provisions for driveshaft length changes or axial movements are required for varying operating and loading conditions in an accessory application, as well as those length changes that are necessary to facilitate assembly and disassembly. Additional length changes are also necessary to accommodate vehicle build tolerances and to compensate for the effects of frame deflection, rough terrain, and temperature change. All of these length change requirements may be provided with a conventional slip spline.

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Agricultural Implement Driveshafts

Chapter 13

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Chapter 13
Agricultural Implement Driveshafts

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Introduction

In agricultural implement applications, a driveshaft employing typical Cardan joints is the most commonly used means for transmitting power from the tractor power take-off (PTO) shaft to the implement's input shaft. It is also the most efficient as well as the most economical drive system or power line for performing this function.

The driveshaft characteristics relative to function, strength, and durability must meet certain requirements as specified in industry recommendations and standards published in the *SAE Handbook* and the ASAE (American Society of Agricultural Engineers) *Agricultural Engineers Yearbook*.

There are three basic types of agricultural implements which utilize tractor power for the performance of their specific functions. These three types of implements are:

Trailing, Towed, or Pull Type Implement

This implement has wheels which support a portion of its weight, while the tractor drawbar is used for partial weight support as well as the hitch point. This hitch point serves as the articulating connection between the tractor and the trailing or pull type implement and permits a relative angular movement about that point. The typical pull type implement can change its position continuously relative to the tractor, such as when turning and when traversing hills and depressions of the terrain.

Fig. 1 shows a typical tractor and pull type implement combination in a 90 deg turn. Some applications require maximum operating joint angles as high as 60 deg and shaft length changes for as much as 16 in when a tractor and pull type implement combination changes from a straight ahead operation to a 90 deg turn.

Joint angles as high as 90 deg are encountered when some implements are changed to transport configuration. Since the driveshaft does not rotate in this position, a 90 deg hinge capability designed into the yokes meets this requirement. The maximum joint angle at which a Cardan joint can operate is approximately 62 deg.

Balers, mower-conditioners, and forage harvesters are typical examples of pull type implements which require tractor power to perform their specific functions.

Mounted or Three-Point Hitch Type Implement

The total weight of this implement is supported by the three-point free link hitch of the tractor in association with a hydraulic power operated lift. The linkage of this typical three-point hitch consists of one upper link and two lower links, each articulated to the tractor and implement at opposite ends in order to connect the implement to the tractor. Fig. 2 illustrates a typical three-point hitch for a tractor.

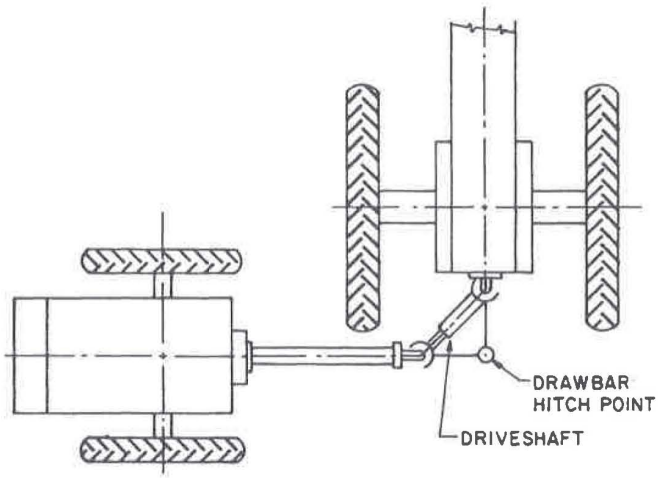


Fig. 1 — Tractor and pull type implement in a 90 deg turn

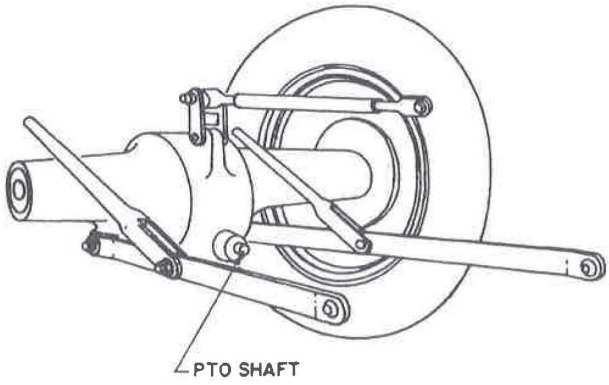


Fig. 2 — Three-point hitch for a tractor

Implements mounted on the three-point hitch of the tractor do not undergo the relative movement due to turning but are subject to the vertical position changes only as adjustments are made for various implement heights to satisfy operating conditions as well as for transport. Fertilizer spreaders, side-mounted mowers, and crop sprayers are typical examples of three-point hitch type implements which require tractor power to perform their specific functions.

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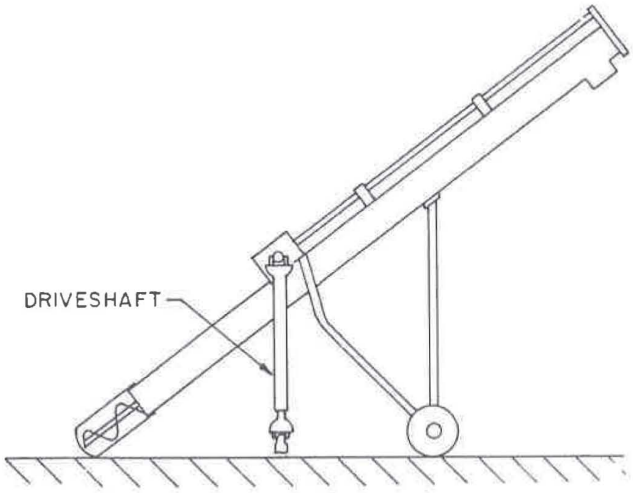


Fig. 3 — Stationary implement (grain auger)

Stationary Type Implement

This implement is located in a fixed position relative to the tractor while performing its function. Fig. 3 shows a typical stationary implement (grain auger) that may be driven by the tractor's PTO shaft. The position of this type of implement is not necessarily held relative to the tractor by means of the drawbar or the three-point hitch. Proper positioning of the tractor relative to the implement is important to obtain efficient power transmission. The operator must be instructed to properly align the tractor relative to the implement in order to achieve minimum joint angles that are required for satisfactory cancellation of torsional disturbances and to maintain the design length of the telescoping driveshaft. Crop blowers, grain conveyors, and irrigation pumps are other typical examples of stationary type implements which require tractor power to perform their specific function.

Therefore, whatever type of implement is employed, the driveshaft must transmit to the implement the power that is required for the performance of the intended function. Further, depending on the type of implement used, the necessary driveline geometry may vary widely while operating (joint angle and shaft length changes), or it may be fixed.

The two standard tractor PTO shaft speeds are 540 or 1000 rpm and the implements are designed to operate at one of these two speeds. The driveshaft is generally furnished as part of the implement and is, therefore, designed specifically to meet the requirements of a given implement. Consideration must be given to the fact that the implement may be used with a tractor having power available which far exceeds the requirements of the implement. Therefore, overload protection should be provided if there is a possibility that the implement may become clogged or be overloaded. For the tractor-implement combination under consideration, a driveline geometry should be selected which will provide the required performance with minimum vibration resulting from the universal joints.

Driveline Arrangements and Constructions

The widely varying application requirements for agricultural implement driveshafts necessitate many different design arrangements and constructions. In general, the following basic driveline arrangements and constructions are used to transmit power to the implement:

Two-Joint Telescoping Driveshaft

The two-joint telescoping driveshaft shown in Fig. 4 is used for either three-point hitch or short tongue pull type implements and as the front telescoping shaft in a three-joint driveline arrangement for pull type implements. In addition, this type of driveshaft is also used for stationary implements.

Three-Joint Driveline (Fixed Length Front Shaft)

The three-joint driveline arrangement illustrated in Fig. 5 in a side or elevation view may be employed with pull type implements operating at a PTO shaft speed of 540 rpm. The typical driveline construction consists of a fixed length front shaft located over the hinge point, a shaft support bearing mounted in a pivoting type pedestal, and a telescoping rear shaft.

Three-Joint Driveline (Telescoping Front Shaft)

The three-joint driveline arrangement shown in Fig. 6 in a side or elevation view is normally used with pull type implements operating at a PTO shaft speed of 1000 rpm. The typical driveline construction consists of a telescoping front shaft located over the hinge point, a shaft support bearing mounted in a rigid pedestal, and a fixed length rear shaft.

Standard Application Dimensions

Standardized dimensional specifications for the tractor PTO shaft and the hitch point have been adopted. These specifications will permit an implement, together with its universal jointed driveshaft, to be used with any tractor of a suitable size and operating at the proper PTO shaft rotational speed. These standardized dimensions for tractors are described in the *SAE Handbook* and also in references published by the ASAE. The basic data associated with the standard tractor PTO shaft are listed in Table 1 for reference.

Table 1 — Standard Tractor PTO Shaft Data

	PTO Shaft Speed, rpm		
	540	1000	
Nominal Shaft and Spline Size, In	1.375	1.375	1.750
Spline Type	Parallel Side	Involute	Involute
Number of Teeth	6	21	20
Diametral Pitch	—	16/32	12/24
Pressure Angle	—	30 deg	30 deg
Horizontal Distance from PTO Shaft End to Drawbar Hitch Point, In	14.0	16.0	20.0

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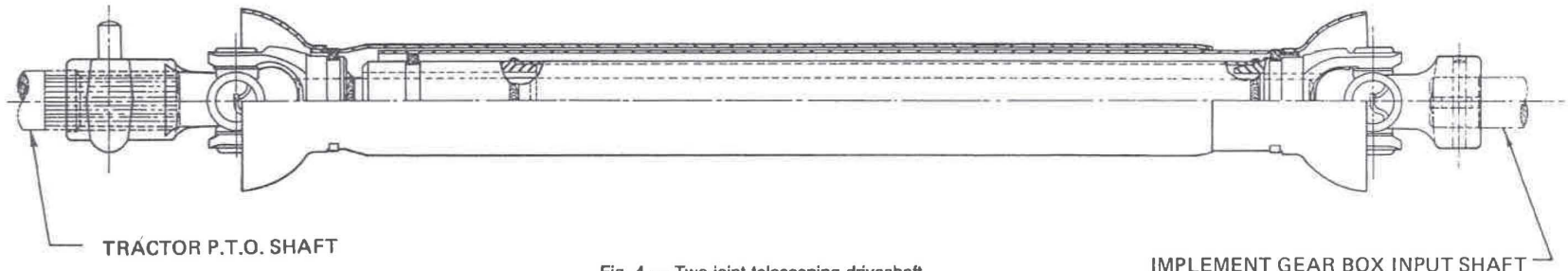


Fig. 4 — Two-joint telescoping driveshaft

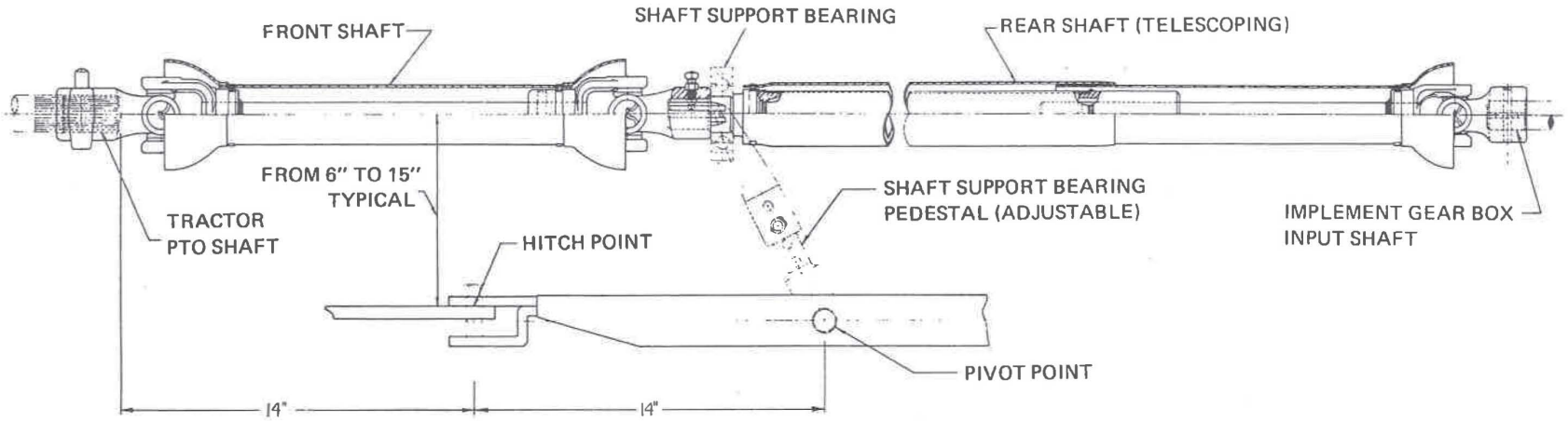


Fig. 5 — Three-joint driveline with fixed length front shaft

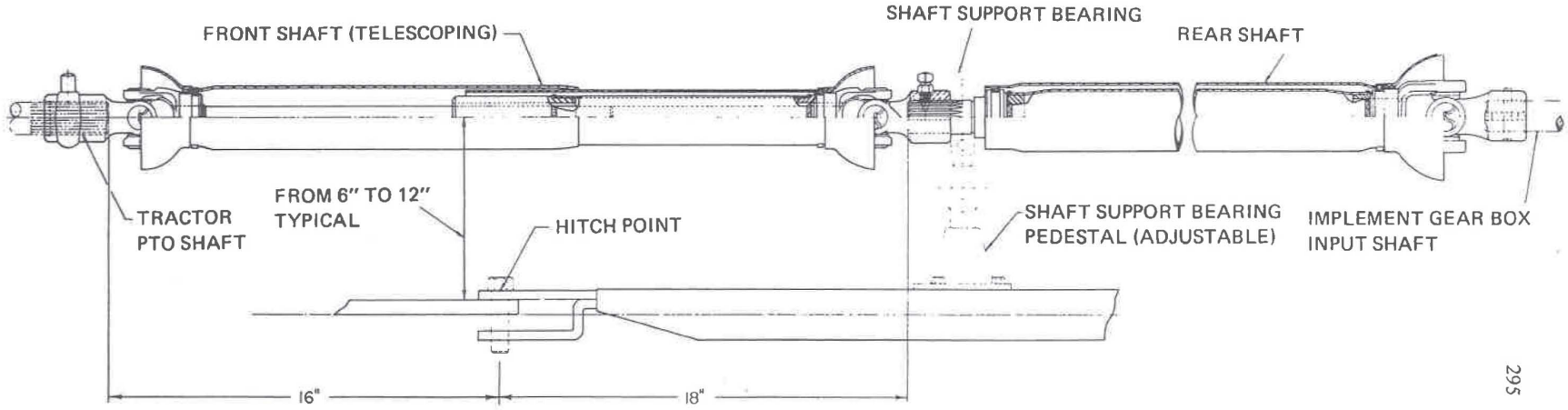
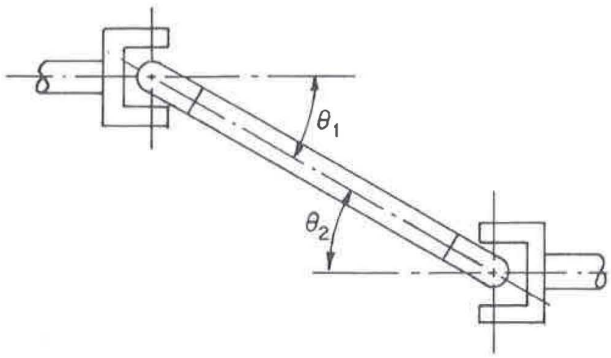
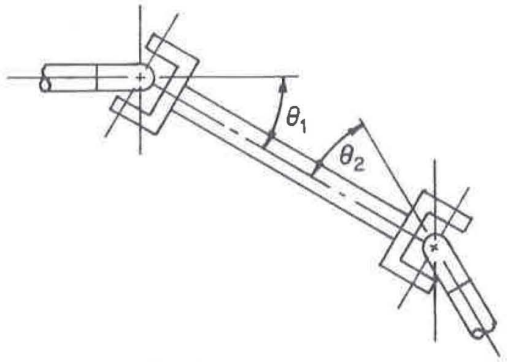


Fig. 6 — Three-joint driveline with telescoping front shaft

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(a) PARALLEL SHAFTS ARRANGEMENT



(b) INTERSECTING SHAFTS ARRANGEMENT

Fig. 7 — Two-joint driveshafts with all shafts in a common plane

Design Parameters

The design of an agricultural implement driveshaft requires consideration of the application parameters. The application parameters and established practices for driveshafts are described in detail in the chapter on *Basic Driveline Requirements*. The various parameters which apply specifically to agricultural implement driveshafts are as follows.

Driveline Operating Geometries

In the section on the *Cardan or Hooke Universal Joint*, the kinematic analysis of that joint shows that when it operates at an angle, nonconstant velocity output motion results. The nonconstant velocity characteristics produce fluctuating speeds and torques which induce disturbances at frequencies that are twice the driveshaft rotational speed. These disturbances are related to torsional, inertia, and secondary couple excitations reacting on the driveshaft and its supporting members.

When a second joint, operating at the same angle and with the proper yoke phase relationship, is added to the driveshaft system, the velocity variation between input and output shafts can be cancelled. Fig. 7 shows the two basic, two-joint driveshaft arrangements which can produce cancellation of this velocity variation. These two driveshaft arrangements are called the parallel and the intersecting shaft arrangements and are described in detail in the section on the *Cardan or Hooke Universal Joint* and also in *Appendix L*.

The velocity variation produced by these two driveshaft arrangements can be expressed in terms of a resultant joint angle called the equivalent angle. In other words, the two-joint driveshaft can be replaced by a single universal joint operating at this angle and expressed mathematically as follows:

$$\theta_{eq} = \sqrt{\theta_1^2 - \theta_2^2}$$

where:

θ_{eq} = Equivalent angle of single joint, deg or rad

θ_1 = Joint angle of first joint, deg or rad

θ_2 = Joint angle of second joint, deg or rad

In this equation, the smaller joint angle is subtracted from the larger joint angle in order to avoid the square root of a negative number. When the joint angles of the two joints are equal, the equivalent angle is zero and, therefore, constant velocity is transmitted.

Even though this constant velocity condition results in zero torsional disturbance at the driveshaft output shaft, the connecting shaft operates with nonuniform velocity and because of its mass, will induce an inertia disturbance. This disturbance is affected by the joint angle size and the inertia of the connecting shaft.

In practical driveshaft systems, however, equal joint angles are not always possible and optimum cancellation can be obtained by proper selection of joint angles and phasing of joint yokes as shown in the section on the *Cardan or Hooke Universal Joint*.

Optimum cancellation of disturbances can also be achieved in multi-joint drivelines as well as in two-joint driveshafts. The resultant torsional and inertia effects of two or more Cardan joints operating in series in the driveline can be expressed in terms of the equivalent angles. These equivalent angles provide a convenient and useful method of expressing acceptable disturbance limits which have been obtained through practical experience and, therefore, permit the design of a satisfactory driveline geometry for an application.

Agricultural implements are divided basically into the pull type, three-point hitch, and stationary categories. The driveline from the tractor to the implement, especially for the pull type, can have many different operating geometries depending on the specific application. The joint angles will vary continuously and over a wide range depending on the turning radius of the tractor-implement combination and the contour of the terrain.

The location of the tractor PTO shaft relative to the hitch point, either drawbar or three-point hitch, is standardized and permits an implement to be used with any make or model tractor having the required 540 or 1000 rpm PTO shaft speed. Implements, however, are single purpose machines and have input shaft to hitch point relationships which vary considerably and are not always in line with the tractor PTO shaft in the plan view, but are offset from each other. The nonstandard input shaft to hitch point relationships and the offset are necessary due to the implement design

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configuration. When a pull type implement, with a nonstandard in line input shaft to hitch point distance, is used with a tractor having a standard PTO shaft to hitch point distance, unequal joint angles are produced which are the same in both right and left turns. However, when this implement has also an offset design of the input shaft relative to the tractor PTO shaft, the unequal joint angle relationship in the right turn differs from that produced in the left turn.

The horizontal distance of the tractor PTO shaft relative to the hitch point and the horizontal distance of the implement input shaft relative to the hitch point is commonly referred to as the hitch point spacing or split.

This characteristic is expressed as a 14-14 in split, 16-16 in split, 16-18 in split, etc. The first number designates the horizontal distance from the tractor PTO shaft end to the hitch point, and the second number designates the horizontal distance from the implement input shaft end to the hitch point.

In the case of an ideal driveline geometry, the distance from the tractor PTO shaft to the hitch point is the same as the distance from the implement input shaft to the hitch point. Also, the tractor PTO shaft is in line with the implement input shaft. This ideal geometry produces equal universal joint angles for any tractor-implement position during a turn. However, very few applications have ideal driveline geometries. Generally, the lack of sufficient driveshaft telescoping length or implement configuration necessitates deviation from the ideal geometry of equal joint angles. Because of the relatively low operating speeds of agricultural implement drivelines, certain acceptable limits on the torsional and inertia vibrations resulting from the nonconstant velocity characteristics of the Cardan joints will permit some degree of deviation from the ideal geometry.

Figs. 8, 9, and 10 illustrate schematically three driveline geometries in the plan view which are commonly found in pull type implement applications. The three-joint driveline shown in Fig. 8 is for a 1000 rpm application. The front shaft telescopes and the rear shaft is a fixed length. The distances from the tractor PTO shaft to the hitch point and from the implement input shaft to the hitch point are both the standard 16 in dimensions. The resulting driveline geometry is ideal and it may be used in applications where sufficient telescoping capability is available, usually in the smaller size driveshafts. These smaller driveshafts use Cardan joints which are also proportionately smaller and require less overall joint length and, therefore, permit a greater length for the telescoping members in the connecting shaft.

Fig. 9 illustrates a three-joint driveline for a 540 rpm application. In this case, the distances from the tractor PTO shaft to the hitch point and from the implement input shaft to the hitch point are both the standard 14 in dimensions. The distance from the tractor PTO shaft to the hitch point is not sufficient to obtain the telescoping length required for a complete 90 deg turn. Therefore, in this case, the front shaft is a fixed length and telescoping capability is incorporated in the rear shaft which is normally of sufficient length to accommodate the required length change. It is noted in Fig. 9 that this driveline configuration will produce a driveline geometry which is not ideal and that some torsional disturbance will result. Equal joint angles occur only at a 57.3 deg turn angle. The driveline geometry, however, can be considered acceptable. As previously stated, practical experience has indicated that satisfactory performance can be achieved when certain limits on the torsional and inertia disturbances of this geometry are not exceeded.

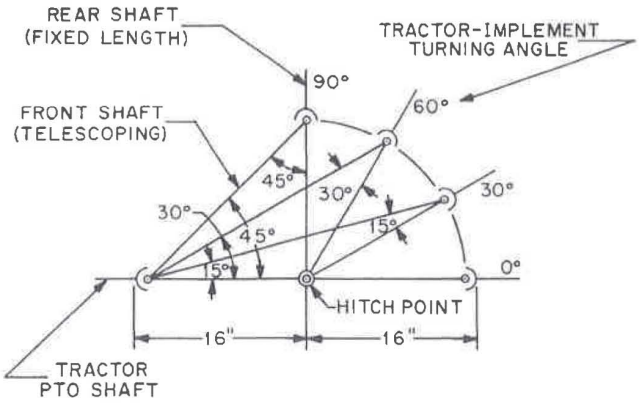


Fig. 8 — Driveline geometry for three-joint driveshaft (16-16 in hitch point split)

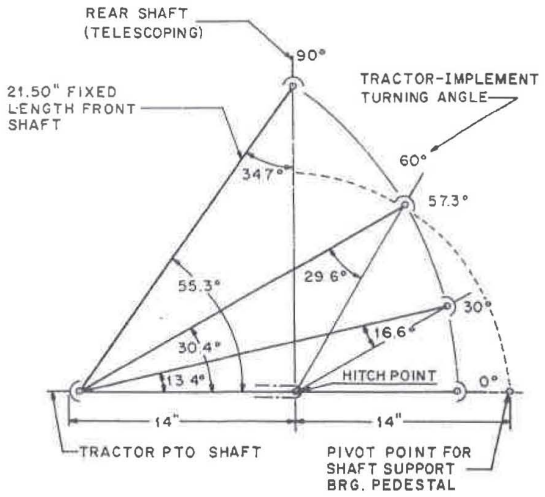


Fig. 9 — Driveline geometry for three-joint driveshaft (14-14 in hitch point split)

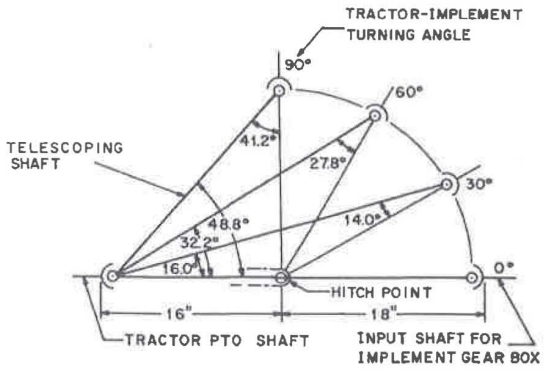


Fig. 10 — Driveline geometry for two-joint driveshaft (16-18 in hitch point split)

Fig. 10 shows a driveline geometry which is a modification of the ideal geometry illustrated in Fig. 8 where the distance from the implement input shaft to the hitch point is increased from 16 to 18

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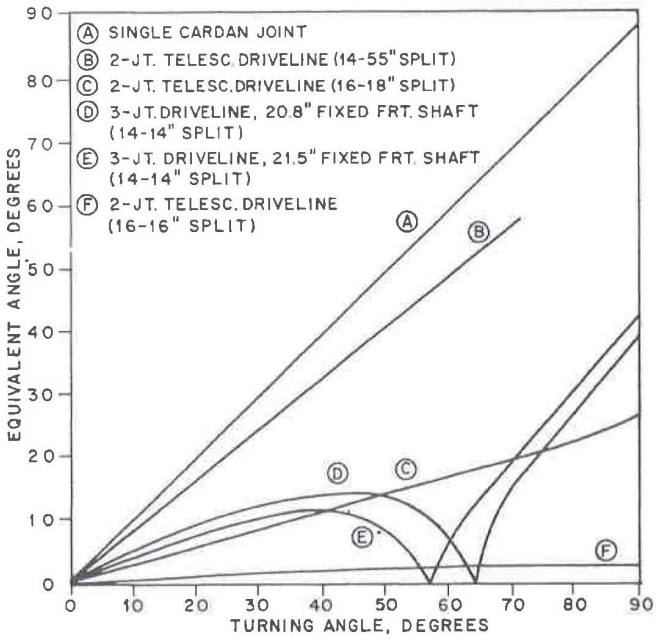


Fig. 11 — Equivalent angle characteristics

in. The two-joint driveline shown is for a 1000 rpm application. The additional length obtained by this modification is generally adequate to provide the necessary telescoping capacity for all but the largest sizes of driveshafts, while the resulting driveline geometry can be considered acceptable.

Fig. 11 illustrates in a graphical manner the equivalent angle characteristics at various turning angles for the previously described driveline geometries, as well as for other typical driveline geometries. All of these driveline geometries are shown compared to a single Cardan joint which is represented by line A. A "theoretically" ideal driveline geometry produces complete cancellation of torsional disturbances throughout the entire tractor-implement turning angle and results in a line which would coincide with the abscissa of the graph. These two lines are the extreme limits for any driveline installation, with all other possibilities located between these two limits.

Curves B through F illustrate the equivalent angle characteristics of the typical driveline geometries used in most applications. Curve B represents a two-joint telescoping driveline with a 14-55 in hitch point split or spacing for use in a 540 rpm application. The driveline geometry resulting from this hitch point split produces equivalent angle characteristics during a full 90 deg turn which is very close to that of a single universal joint and is not recommended for large turns of the tractor-implement combination. Curve C represents a two-joint telescoping driveline with a 16-18 in split for use in a 1000 rpm application. The near ideal driveline geometry resulting from this split as shown in Fig. 10 produces satisfactory equivalent angle characteristics during a full 90 deg turn.

Curves D and E represent three-joint drivelines with fixed front shafts and telescoping rear shafts for use in 540 rpm applications. The geometry for the driveline of Curve E is illustrated in Fig. 9. Curves D and E have three common characteristics as follows:

1. The equivalent angle is zero at some point during a 90 deg turn.
2. The values of the equivalent angle increase rapidly beyond this zero point.
3. The value of the equivalent angle changes considerably with a small change in the fixed length.

Curve F represents a two-joint telescoping driveline with a 16-16 in split for use in a 1000 rpm application. As shown in Fig. 8, the driveline geometry resulting from this split is ideal. In practice, such an ideal driveline geometry would produce a slight error in joint angles and it can therefore, be referred to as a "practical" ideal geometry. As illustrated by curve F, the 16-16 in split results in nearly total cancellation of the torsional disturbance and would be ideal for all 1000 rpm applications. Unfortunately, when the larger size universal joints are employed with this split, there is insufficient telescoping space available between joint centers in conventional driveshafts for a full 90 deg turn. Where a 90 deg turn is not required, the 16-16 in split arrangement can be used for the larger joint sizes.

Large tractors (standard 1.750 in PTO shaft) with the 20 in distance from the PTO shaft to the hitch point will accommodate the larger universal joints, while allowing sufficient telescoping length. Implements designed for use with these tractors provide for a 20-20 in split of the hitch point which results in a driveline geometry and equivalent angle characteristics very similar to the 16-16 in split represented by curve F.

In practice, the variation in implement types as well as their usage and requirements allows the various driveline operating geometries to be used successfully in the different applications. As an example, implements with relatively low rotating inertias, such as a typical fertilizer spreader, can perform satisfactorily with a high torsional velocity fluctuation; whereas, a relatively high rotating inertia implement, such as a typical baler, requires a relatively constant speed input.

Disturbance Level Limits

As previously stated, certain maximum levels of torsional and inertia disturbances can be tolerated in agricultural implement applications. Practical limits for these disturbances have been established with the maximum angular acceleration being the design parameter used to express these limits. In the section on the *Cardan or Hooke Universal Joint,* a useful approximate equation for the maximum angular acceleration of the joint was derived and is expressed as follows:

$$\alpha_{1(max)} = \theta^2\omega^2$$

where:

$\alpha_{1(max)}$ = Angular acceleration of driven yoke, rad/s^2

θ = Joint angle, rad

ω = Angular velocity of driving yoke, rad/s

This equation indicates that the maximum angular acceleration is affected by the joint angle and rotational speed only. Fig. 12 has

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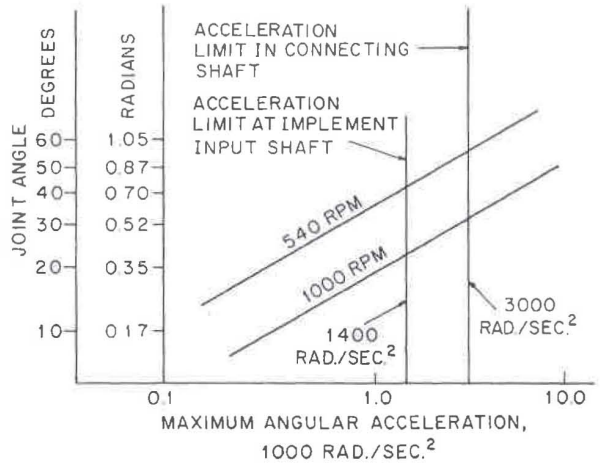


Fig. 12 — Practical limits of maximum angular acceleration for normal operation

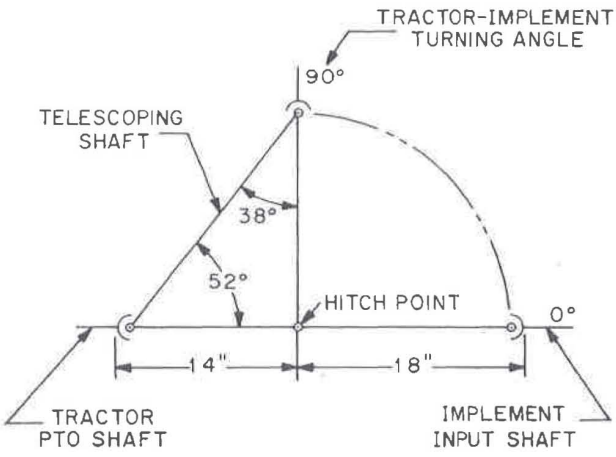


Fig. 13 — Driveline geometry for two-joint driveshaft (14-18 in hitch point split)

been developed with the use of this equation and shows the effect of joint angle and maximum angular acceleration on shaft rotational speed. The 540 and 1000 rpm constant rotational speed curves illustrated in this figure are the two standard tractor PTO speeds. Also shown in this figure are the limits of the maximum angular acceleration as determined from experience. These practical limits of maximum angular acceleration for the normal or straight ahead operating position of the implement are 1400 rad/s² (torsional disturbance) at the implement input shaft, and 3000 rad/s² (inertia disturbance) in the connecting shaft or shafts of the driveline.

Some applications produce angular acceleration levels in the driveshaft connecting member, when cornering with an implement of up to approximately 7000 rad/s². This angular acceleration, however, is a momentary condition and the resulting inertia vibration during this time is unavoidable and tolerated. The angular acceleration in the connecting shaft while cornering, is reduced in many applications because of the necessity to reduce tractor speed (through engine speed reduction) during the turning maneuver.

Higher acceleration levels entering the implement at the input shaft can also be accommodated by introducing a slip clutch, over-running clutch, or a torsional resilient device in the driveline to allow the rotational inertia of the implement to maintain a relatively constant velocity. The tendency of the driveshaft speed fluctuations to change the speed of the implement's rotational inertia is thus minimized by a slippage, lag, or torsional compliance.

The tolerance levels of implements to torsional speed fluctuations vary due to the rotating inertia and the total mass of the specific implements. The angle required between the tractor and implement during a turn is also important in the selection of an acceptable driveline operating geometry.

Fig. 12 is a useful graph for analyzing driveline operating geometries of typical applications for acceptable levels of torsional and inertia vibrations. The following three problems are representative of typical applications.

Application Problem 1

The driveline geometry shown in Fig. 13 in the plan view is for a two-joint telescoping driveshaft in a 540 rpm application with a 14-18 in hitch point split. The tractor PTO shaft and the implement input shaft are assumed in line in the side view. Determine the acceptability of this driveline geometry with regard to torsional and inertia disturbances for both the straight ahead and the 90 deg turn positions.

Straight Ahead Position—Since no joint angles are present in either the side or the plan views, constant velocity is transmitted at the implement input shaft and, therefore, no angular acceleration exists in the driveline and at the implement input shaft.

90 Degree Turn Position (Right and Left Turn)—The torsional equivalent angle is:

$$\theta_{eq} = \sqrt{\theta_1^2 - \theta_2^2}$$

where:

θ_1 = Joint angle of first joint, deg

θ_2 = Joint angle of second joint, deg

$$\theta_{eq} = \sqrt{(52)^2 - (38)^2}$$

$$\theta_{eq} = 35.50 \text{ deg}$$

In Fig. 12, the practical limit of the torsional equivalent angle at 540 rpm for the normal operation position is approximately 38 deg. Therefore, the 90 deg turn position, which is a momentary operation, will not produce excessive torsional disturbance at the implement input shaft and is tolerated for most applications.

For this driveline geometry, the inertia excitation produced by the connecting or center shaft can be expressed in terms of an

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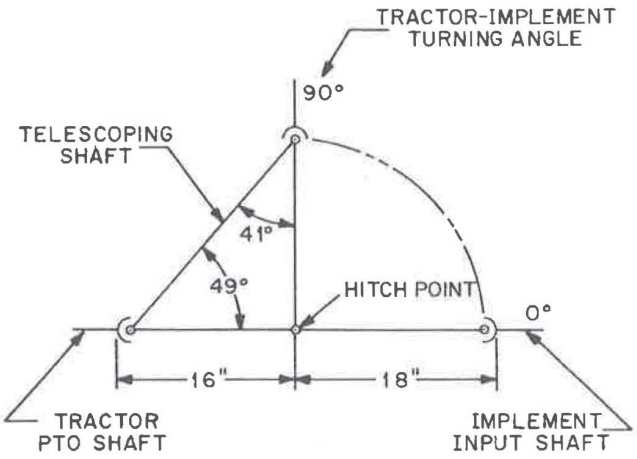


Fig. 14 — Driveline geometry for two-joint driveshaft (16-18 in hitch point split)

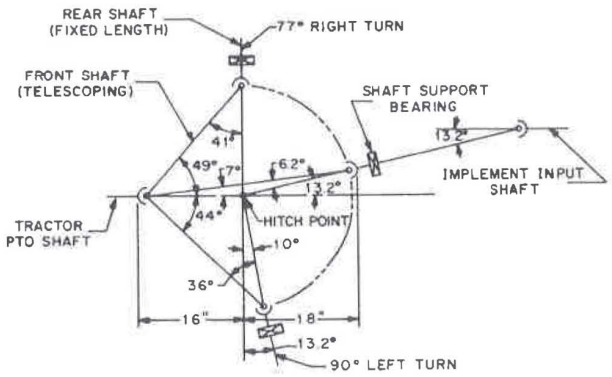


Fig. 15 — Driveline geometry for three-joint driveshaft (16-18 in hitch point split)

inertia equivalent angle as explained in detail in the section on the *Cardan or Hooke Universal Joint*. In the analysis of agricultural implement driveshafts, only the inertia drive equivalent angle is considered. For the two-joint driveshaft in this problem, the inertia drive equivalent angle is the joint angle of the first joint which is 52 deg.

In Fig. 12, the practical limit of the inertia equivalent angle at 540 rpm for the normal operating position is approximately 55 deg. Therefore, the 90 deg turn position, which is a momentary operation, will not produce excessive inertia disturbance and is tolerated for most applications.

Application Problem 2

The driveline geometry shown in Fig. 14 in the plan view is for a two-joint telescoping driveshaft in a 1000 rpm application with a 16-18 in hitch point split. The tractor PTO shaft and the implement input shaft are assumed in line in the side view. Determine the acceptability of this driveline geometry with regard to torsional and inertia disturbances for both the straight ahead and the 90 deg turn positions.

Straight Ahead Position—Since no joint angles are present in either the side or the plan views, constant velocity is transmitted at the implement input shaft and, therefore, no angular acceleration exists in the driveline and at the implement input shaft.

90 Degree Turn Position (Right and Left Turn)—The torsional equivalent angle is:

$$\theta_{eq} = \sqrt{\theta_1^2 - \theta_2^2}$$

where:

θ_1 = Joint angle of first joint, deg

θ_2 = Joint angle of second joint, deg

$$\theta_{eq} = \sqrt{(49)^2 - (41)^2}$$

$$\theta_{eq} = 26.83 \text{ deg}$$

In Fig. 12, the practical limit of the torsional equivalent angle at 1000 rpm for the normal operating position is approximately 20 deg. Therefore, the 90 deg turn position will produce excessive torsional disturbance. However, since the operation during the turn is momentary, the resulting disturbance level at the implement input shaft is tolerated.

For the two-joint driveshaft in this problem, the inertia drive equivalent angle is the joint angle of the first joint which is 49 deg. In Fig. 12, the practical limit of the inertia equivalent angle at 1000 rpm for the normal operating position is approximately 30 deg. Therefore, the 90 deg turn position will produce excessive inertia disturbance. However, since the operation during this turn is momentary, the resulting disturbance level is tolerated.

Application Problem 3

The driveline geometry shown in Fig. 15, in the plan view, is for a three-joint driveline with a telescoping front shaft in a 1000 rpm application and with a 16-18 in hitch point split. No joint angles are assumed in the side view. The driveline phasing is a parallel-crossed arrangement where the front shaft yokes are phased parallel or in line and the rear shaft yokes are phased crossed or at 90 deg. This phasing arrangement was chosen because it is the optimum phasing arrangement for minimum disturbances when the shaft support bearing is located on the rear shaft. It is the practice in agricultural implement three-joint drivelines to locate this shaft support bearing on the rear shaft. The various possible phasing arrangements for three-joint drivelines are described in the section on the *Cardan or Hooke Universal Joint*.

Determine the acceptability of this driveline geometry with regard to torsional and inertia disturbances for the straight ahead and the right and left turn positions as shown.

Straight Ahead Position—The torsional equivalent angle for the three-joint driveline is:

$$\theta_{eq} = \sqrt{\theta_1^2 - \theta_2^2 - \theta_3^2}$$

$$= \sqrt{(7)^2 - (6.2)^2 - (13.2)^2}$$

$$\theta_{eq} = 12.79 \text{ deg}$$

In Fig. 12, the practical limit of the torsional equivalent angle at

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1000 rpm for the normal operating position is approximately 20 deg. Therefore, the torsional disturbance level produced by this driveline is acceptable.

For the three-joint driveline, the inertia drive equivalent angle is:

$$\theta_D = \sqrt{2\theta_1^2 - \theta_2^2}$$

$$= \sqrt{2(7)^2 - (6.2)^2}$$

$$\theta_D = 7.72 \text{ deg}$$

In Fig. 12, the practical limit of the inertia equivalent angle at 1000 rpm for the normal operating position is approximately 30 deg. Therefore, the inertia disturbance level produced by this driveline is acceptable.

77 Degree Right Turn Position—The torsional equivalent angle is:

$$\theta_{eq} = \sqrt{(49)^2 - (41)^2 - (13.2)^2}$$

$$\theta_{eq} = 23.36 \text{ deg}$$

The practical limit of the torsional equivalent angle at 1000 rpm for the normal operating position is approximately 20 deg. Therefore, the 77 deg right turn position will produce some excessive torsional disturbance. However, since the operation during this turn is momentary, the resulting disturbance level is tolerated.

The inertia drive equivalent angle is:

$$\theta_D = \sqrt{2(49)^2 - (41)^2}$$

$$\theta_D = 55.87 \text{ deg}$$

The practical limit of the inertia equivalent angle at 1000 rpm for the normal operating position is approximately 30 deg. Therefore, the 77 deg right turn position will produce excessive inertia disturbance. However, since the operation during this turn is momentary, the resulting disturbance level is tolerated.

90 Degree Left Turn Position—The torsional equivalent angle is:

$$\theta_{eq} = \sqrt{(44)^2 - (36)^2 - (13.2)^2}$$

$$\theta_{eq} = 21.58 \text{ deg}$$

Since the practical limit of the torsional equivalent angle for the normal operating position is approximately 20 deg, the 90 deg left turn position will produce some excessive torsional disturbance. However, because the operation during this turn is momentary, the resulting disturbance level is tolerated.

The inertia drive equivalent angle is:

$$\theta_D = \sqrt{2(44)^2 - (36)^2}$$

$$\theta_D = 50.75 \text{ deg}$$

Since the practical limit of the inertia equivalent angle for the normal operating position is approximately 30 deg, the 90 deg left turn position will produce excessive inertia disturbance. However, because the operation during this turn is momentary, the resulting disturbance level is tolerated.

These typical problems illustrate the need for selecting an acceptable driveline operating geometry for a given application. The trend to implements with higher horsepower requirements, higher inertia, and the ability to turn sharp corners will necessitate almost total cancellation of torsional speed fluctuations at the implement input shaft.

The hitch point spacings available for tractor-implement combinations have been either the standard 14-14 in (540 rpm), or the 16-16 in (1000 rpm) splits. Ideal geometries of equal joint angles are, therefore, possible in applications where telescoping driveshafts can be used which provide for the required extended and collapsed lengths. Thus, complete cancellation of the torsional speed fluctuations at the implement input shaft is achieved.

The higher horsepower driveshafts, however, are too large to fit into the ideal length established by these standards. The resulting joint angles are, therefore, not equal, and some torsional disturbance is produced.

Two methods have been used to obtain ideal geometries with equal angles for implements with higher horsepower requirements. In the first method, the distances from the hitch point to both the tractor PTO shaft and the implement input shaft are increased by means of hitch extensions. An extension of three inches will result in an additional six in of telescoping distance being made available for the driveshaft length to produce equal joint angles. These equal joint angles will provide complete cancellation of torsional speed fluctuations for the higher horsepower driveshafts. This modification of the hitch point spacing will increase the bending moment on the hitch. Care should be exercised so that the resulting force on the hitch point does not exceed a standardized industry recommendation.

A second method used to increase the telescoping capability in a driveshaft is to incorporate an additional tubular member or members in the connecting shaft. Such a design approach provides for a double telescoping or multi-telescoping feature which produces the length necessary to obtain equal joint angles. Therefore, complete cancellation of torsional speed fluctuations can be obtained in an application using a smaller standard hitch point split arrangement.

Critical Speed

The two rated or standard tractor PTO shaft speeds are 540 and 1000 rpm. Agricultural implements are designed to operate at one of these two rated speeds. In addition, consideration must be given to the possibility of exceeding these rated speeds before the maximum possible operating speeds are limited by some controlling or governing means. Therefore, the driveshaft must be capable of performing safely for sustained periods of time at the peak operating speed of the intended application.

Fundamental Critical Speed—The safe operating speed capability of a driveshaft requires that its critical speed must be higher than its maximum operating speed. It is defined as the rotational speed which coincides with the natural bending resonant frequency of the shaft. The subject is discussed in detail in the chapter on *Critical Speed*. It has been shown that the fundamental or lowest critical speed of a shaft is affected by its area moment of inertia, weight, length, material modulus of elasticity,

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and dimensional clearances.

For purposes of analysis, agricultural implement driveshaft can be grouped into three basic types as follows:

Two-Joint Driveshaft or Front Shaft of Three-Joint Driveshaft—This type of shaft includes the front fixed length shaft of a 540 rpm three-joint driveshaft, the 1000 rpm two-joint telescoping driveshaft, and the front telescoping shaft of a 1000 rpm three-joint driveshaft. This type of shaft has a maximum length that is dictated by hitch point distance requirements from the tractor PTO shaft and the implement input shaft. These hitch point distances are necessary in order to provide a driveline operating geometry that will produce an acceptable cancellation of torsional disturbances.

The length restriction, together with the maximum operating speed of either the 540 or the 1000 rpm application, necessitates a certain area moment of inertia for a typical steel driveshaft. The commonly used steel bar shafts or tubes, (square or round) when properly sized for torsional strength for their particular applications, will have critical speed levels that are, in general, well above the 1000 rpm operating speed, which is the higher of the two standard tractor PTO shaft speeds.

Rear Shaft of Three-Joint Driveshaft—This type of shaft includes the rear telescoping shaft of a 540 rpm three-joint driveshaft and the rear fixed length shaft of a 1000 rpm three-joint driveshaft. The rear shaft does not have the same length restrictions as the front shaft of a typical three-joint driveline in order to achieve acceptable cancellation of torsional disturbances.

Therefore, the critical speed of this shaft should be analyzed to ensure that it is higher than the maximum operating speed. The critical speed can be increased by simply using larger diameter tubing without increasing the wall thickness. If it is not practical to use larger diameter tubing in an application, the required critical speed can also be obtained by using multiple shorter length shafts supported by shaft support bearings. Solid bar shafts can also be utilized for the rear shaft or shafts after the optimum placement of the shaft support bearings has been determined.

Stationary Implement Driveshaft—This type of shaft is generally a two-joint telescoping driveshaft used to operate a stationary implement which is located in a fixed position relative to the tractor.

A stationary implement driveshaft may operate in one of the following two modes:

1. Transmit power to a stationary implement when attached to the tractor at the hitch connection.
2. Transmit power to a stationary implement when not rigidly attached to the tractor.

For a stationary implement connected to the tractor, the maximum driveshaft length is determined by the tractor-implement geometry. Since a stationary implement not connected to the tractor may have a driveshaft of any practical length, a maximum length should be established based on the requirements of the implement and its design position relative to the tractor.

It is recommended that the design length be properly indicated in the instruction manual of the implement. A positive method of preventing over-extension of the driveshaft is to design a stop in the telescoping members to prevent an extension beyond the design length. In either case, the proposed shaft design must be analyzed to ensure that the critical speed is above the maximum operating speed.

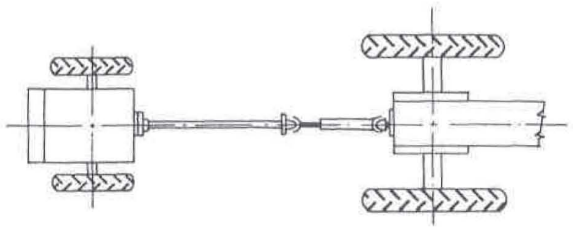


Fig. 16 — Tractor and implement operating in line

The implement in the operating environment must be properly aligned relative to the tractor to obtain minimum and approximate equal joint angles at both joints, and in the non-hitched application, not to exceed the recommended design length of the telescoping driveshaft.

Effect of Secondary Couples—In typical agricultural implement applications and at universal joint angles, power from the tractor is transmitted at a relatively higher torque than in most other applications. Therefore, the effect of Cardan joint secondary couples on both the connecting shaft and on the supports must be analyzed. These secondary couple effects are discussed in detail in the section on the *Cardan or Hooke Universal Joint*. Secondary couples produce vibrations which occur whenever a Cardan joint operates at an angle and transmits torque. These secondary couples tend to deflect or bend the connecting shaft and their magnitude increases with an increase in joint angle and torque load.

Since secondary couple disturbances occur twice per shaft revolution, they can produce a vibration of the shaft at one half of the rotating speed of the fundamental critical speed of the driveshaft previously described. Therefore, for each driveline design and operating geometry, the fundamental and the half critical speeds should be analyzed. The fundamental critical speed should be greater than the maximum operating speed and the half critical speed, when applicable, should not occur within the normal operating speed range.

Connecting Shaft Members

The connecting shaft members of the driveshaft must be capable of transmitting the maximum applied torque and also allow the length changes required for a given application. In typical pull type agricultural implements, the driveshaft length varies from a minimum when the implement corners to a maximum when the implement and tractor are in the straight ahead position. Since the hitch or hinge point is located below the driveline, an additional lengthening and shortening occurs when operating on undulating terrain such as over hills and through depressions.

Telescoping requirements can best be established by analyzing a layout of a specific tractor-implement combination in the various positions. The maximum extended length of the driveshaft occurs for a position which is a combination of Figs. 16 and 17, where the tractor and implement are shown in line and driving over a hill. The minimum collapsed length occurs for a position which is a combination of Figs. 1 and 18, where the tractor and implement are illustrated making a maximum turn while operat-

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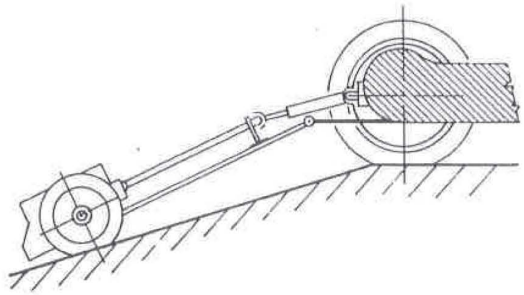


Fig. 17 — Tractor and implement operating over a hill

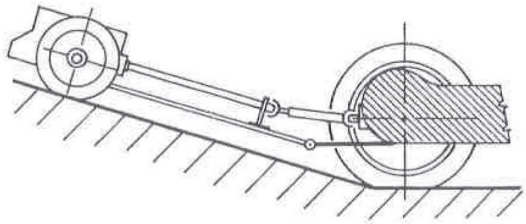


Fig. 18 — Tractor and implement operating in a depression

ing in a depression. Also, Fig. 17 shows that if the vertical articulating angle between the tractor and implement becomes excessive, there is a possibility that the hitch point may strike the driveline.

Therefore, the driveline geometry for a particular application must be carefully analyzed so that sufficient telescoping capability is available for the connecting shaft members of the proposed driveshaft. This telescoping capability must provide adequate engagement of the connecting shaft members in the maximum extended length for proper operation. Also, it must be capable of achieving the minimum collapsed length before the fully collapsed or solid length is attained.

The typical driveline geometries shown in Figs. 8, 9, and 10 illustrate the need for driveshafts which can accommodate length changes. Also, these geometry layouts show that the required location of the telescoping member in either the front shaft or the rear shaft in three-joint drivelines depend on various application requirements. These application requirements are the rated PTO shaft operating speed, standardized location of the hitch point from both the tractor PTO shaft and the implement input shaft, implement power needs, and the turning angle dictated by the farming operation performed.

The requirement of shaft length changes of up to approximately 16 in in some applications is generally achieved with a square or rectangular bar shaft engaging a steel sleeve. The relatively low cost of this steel-on-steel construction offsets its major disadvantage of producing relatively high telescoping forces under torque load.

This steel-on-steel construction is generally adequate for most light load applications, especially for those cases where the torque load requirements are pulsating, allowing slip to occur during the low torque portion of the load cycle. Applications which have a steady high torque load requirement, normally those requiring the higher horsepower driveshafts such as forage harvesters and pull type combines, cannot accommodate the higher telescoping forces resulting from the typical steel-on-steel design end motion devices.

These higher telescoping forces, especially during a sharp turn, impose relatively high bending forces on the tractor PTO shaft and on the implement input shaft. In addition, the pedestal mounting for the shaft support bearing required when a three-joint driveline is used, must also withstand relatively high bending forces during a sharp turn. Various means have been used to reduce telescoping forces in these applications below those obtained with a square or rectangular bar engaging a steel sleeve. They are described in detail in the chapter on *Splines and Axial Motion Devices*, and a comparison of their typical axial thrust force characteristics during end motion is shown.

Some of these more commonly used telescoping methods are:

1. Splines
2. Square or rectangular bar shaft in a plastic sleeve
3. Rolling element devices

Generally, pull type implements with a continuous torque in excess of a certain load level may require a low friction telescoping shaft which reduces the axial thrust. Applications requiring over 60 horsepower at 540 rpm may need connecting shafts with telescoping force characteristics below that of a typical square bar shaft and steel sleeve construction.

Typical stationary or fixed position implements such as crop blowers, grinder mixers, and irrigation pumps, where telescoping under load is not required, can use the more economical square bar shaft and steel sleeve construction. Stationary implements that are not attached to the tractor to limit the maximum telescoping length of the driveshaft, may require connecting members which will not pull apart or separate. A feature such as this can prevent the operator from exceeding the design length limitation of the driveshaft. This feature can be accomplished in many ways. A typical simple design provides a stop on the bar shaft which contacts the inside edge of the sleeve.

In addition to the previously described conventional single telescoping constructions used for a connecting shaft, some applications may require a double telescoping or multi-telescoping feature in the connecting shaft members. Such a multi-telescoping mechanism may be necessary in order to obtain an ideal driveline geometry of equal joint angles and thus provide cancellation of torsional disturbances as described in the *Disturbance Level Limits* section of this chapter. A typical multi-telescoping mechanism consisting of grooved tubes is shown in the chapter on *Splines and Axial Motion Devices*. The use of progressively telescoping grooved tubes in such a design provides not only for an increase in the maximum extended length, but also a decrease in the fully collapsed length. The construction of this design is more complicated than a conventional telescoping shaft since a method of centralizing the intermediate tube or tubes must also be incorporated. A disadvantage of this design is the introduction of additional clearances in the assembly which may contribute to unbalance vibration problems.

Torque

The agricultural implement driveshaft may be subjected to various torsional loading modes as the implement performs its func-

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tions. Since the torsional loading modes may vary considerably in a given implement application, the torque requirements are best determined by means of a load or duty cycle. In a typical duty cycle, the torque requirements are expressed as a function of time.

Some implements have a pulsating type of load cycle (for example, balers) while other implements have a steady type of load cycle (for example, windrowers). Still other implements have an erratic load cycle (for example, rotary mowers and mixer mills). The characteristics of these duty cycles can also vary since they will change with the type of crop being worked and the rate at which this crop is processed through the implement.

A true duty cycle can only be established through actual field testing of an instrumented tractor-implement combination. If this procedure is not possible, a load cycle can be computed in a manner similar to that shown in the chapter on *Basic Driveline Requirements*.

Torque Overload Protection

Agricultural implements are designed for a loading condition or duty cycle which is dependent upon the specific work being performed. The design load can be exceeded by inadvertently jamming the implement or by subjecting the implement to abnormal working conditions. Jamming an implement, for example, will result in the driveline being subjected to the full torque available at the tractor PTO shaft.

Since the tractor is not necessarily matched to the implement, the available torque at the tractor PTO shaft can be far greater than that required to perform a particular function. Therefore, jamming an implement, either by the crop or by a foreign object, can cause an abnormally high torque load to be transmitted through the driveshaft and the implement drivetrain.

Protection from such abnormal overloads can be achieved with various devices such as friction type clutches, spring overload ratchet mechanisms, or shear elements. Fig. 19 illustrates a typical friction type overload clutch, while Fig. 20 shows a typical ratchet type overload clutch. These devices are normally adjustable within their design range, through the use of tension bolts or an adjustment nut as shown.

Various types of shear elements such as bolts, pins, and keys designed to shear at a predetermined load, are not adjustable. However, the torque to shear is dependent upon the shear strength of the shear element and its distance from the rotational center of the shaft. Fig. 21 illustrates a nomograph for typical shear plate drives showing the ultimate shear torque for various combinations of bolt sizes and strengths, and distances of the bolt from the rotational center.

Fig. 22 shows a nomograph for typical shear pin drives where the ultimate shear torque can be determined for various combinations of pin sizes and strengths, and shaft diameters. Fig. 23 illustrates a nomograph for typical keyed connections where the ultimate shear torque can be determined for various combinations of key widths and lengths, and shaft diameters.

The general practice has been to have the shear occur at approximately 70% of the static torsional yield point of the weakest member of the drivetrain. In the case of the bolt, the unthreaded diameter should be of sufficient length to ensure that its threaded portion is not in the shear plane.

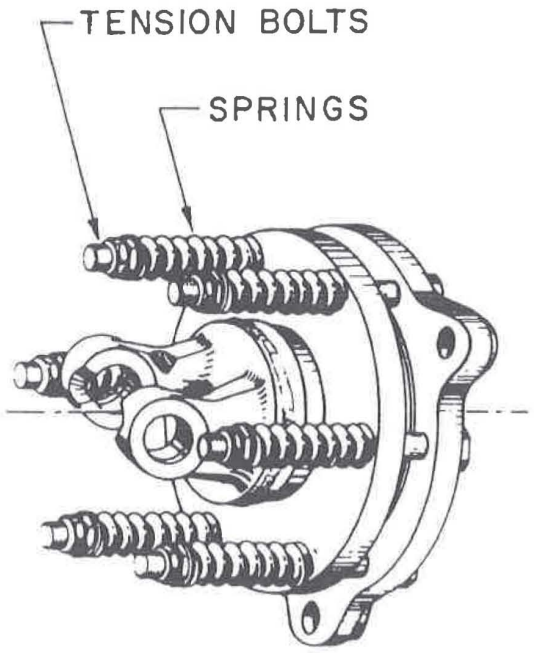


Fig. 19 — Friction type overload clutch

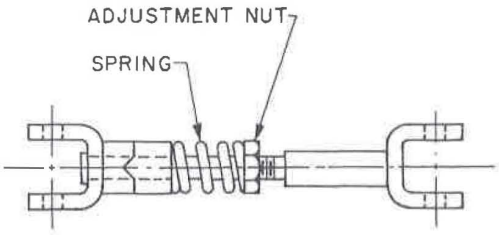


Fig. 20 — Ratchet type overload clutch

Driveshaft Size Selection

The Cardan joint is the most common universal joint used in agricultural implement applications. In the section on the *Cardan or Hooke Universal Joint*, it is shown that this joint must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. As described in *Appendix I*, the universal joint's static torque capacity is the maximum permissible statically applied torque at a zero degree joint angle which is safely below the ultimate strength of the joint. The maximum torque experienced in the application must not exceed this static torque capacity. In the case of agricultural implements, the level of this maximum torque is generally determined by the torque overload protection device in the drive system. In addition, the joint bearing configuration must have the dynamic capacity to transmit power for a required period of time at various torques, speeds, and joint angles.

Some manufacturers of Cardan joints have established static and dynamic capacity ratings for their various sizes of universal joints. It is good practice to consult manufacturer's publications for available universal joint sizes, capacity ratings, and other pertinent functional data which may be useful in selecting a joint

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NOMOGRAPH VALUES REPRESENT THE AVERAGE ULTIMATE TORQUE AT WHICH A BOLTED SHEAR PLATE WILL FAIL.

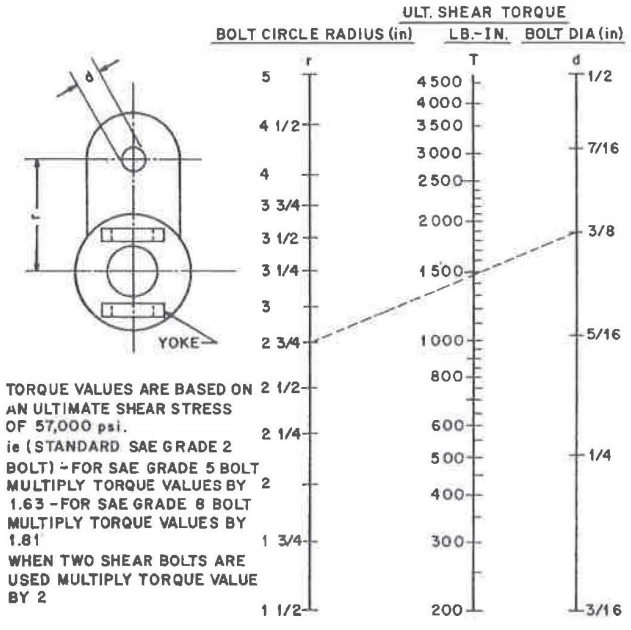


Fig. 21 — Shear torque nomograph for shear plate drives

NOMOGRAPH VALUES REPRESENT THE AVERAGE ULTIMATE TORQUE AT WHICH A SHEAR PIN WILL FAIL.

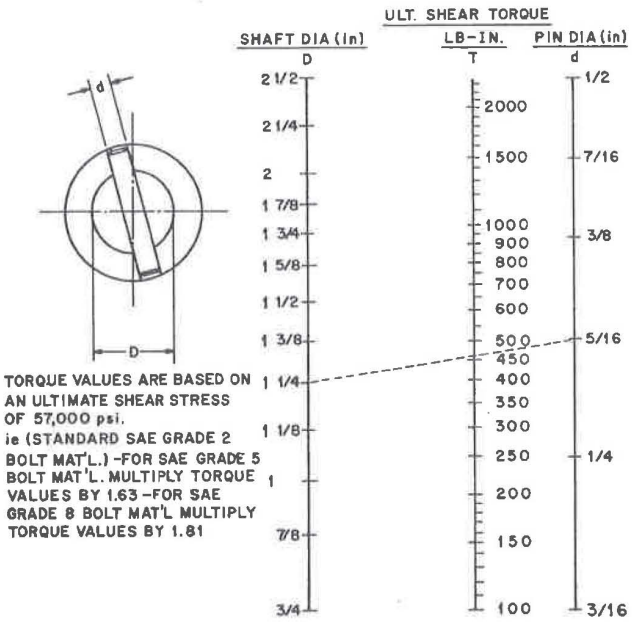


Fig. 22 — Shear torque nomograph for shear pin drives

for a specific application.

Most agricultural implement drivelines, however, are sized according to ASAE recommendation R331.2, which separates these drivelines into various usage or application categories of

NOMOGRAPH VALUES REPRESENT THE AVERAGE ULTIMATE TORQUE AT WHICH A 1.00 in. LONG KEY WILL FAIL.

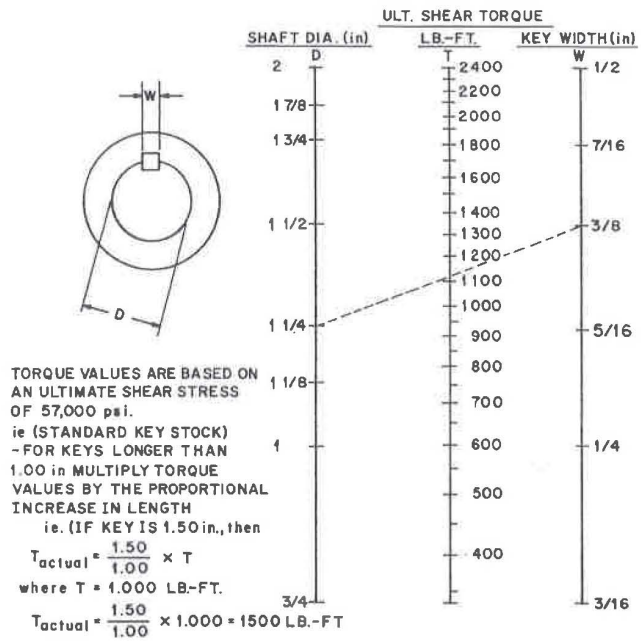


Fig. 23 — Shear torque nomograph for keyed connections

increasingly larger horsepower and torsional load ratings. For each of the application categories, the torsional load test specifications or ratings are listed under appropriate headings as follows:

1. Static Torque Requirements
   a. Cardan joint assemblies consisting of the cross, bearings and yokes
   b. Connecting members consisting of tube, sleeve, and shaft
2. Dynamic Torque Requirements
   *a. Constant angle bearing durability test for Cardan joints
   b. Fully reversed torsional fatigue test for Cardan joints and connecting members

*In this test the Cardan joints shall operate with a 10 deg joint angularity under a constant torque load for 1 000 000 revolutions at a speed of 540 rpm without failure. The B_{10} life level as defined by the AFBMA (Anti-Friction Bearing Manufacturers Association), is used to evaluate the load ratings for the joint crosses and bearings. The B_{10} life is defined as the life that 90% of a statistical sample of universal joint bearings will reach or exceed.

Therefore, based on the maximum torque experienced in a given application and the duty cycle required in normal operation, the driveshaft's universal joints and connecting members can be sized according to these torsional load ratings established for the various application categories.

The life expectancy of the Cardan joint can be determined for a given application by using the constant angle bearing durability torque test loads listed in ASAE recommendation R331.2 for various application categories. These torque loads are B_{10} values and are called the dynamic torque ratings.

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As shown in *Appendix B*, the life for rolling element bearings is inversely proportional to the third power of the load for ball bearings and the 10/3 power of the load for roller or needle roller bearings. From these basic relationships, convenient empirical life equations have been developed and published in AFBMA standards. These equations in general form are:

$$L = \left(\frac{C}{P}\right)^3 \text{ million revolutions for ball bearings}$$

and

$$L = \left(\frac{C}{P}\right)^{\frac{10}{3}} \text{ million revolutions for roller or needle roller bearings}$$

where:

L = The B_{10} life in millions of revolutions under the constant bearing load P

C = The load which will yield a B_{10} life of one million revolutions and is called the basic dynamic load rating

From these basic empirical life equations, a commonly used equation has been developed for agricultural implement driveshaft design to compute the average joint life in millions of revolutions. This equation is:

$$L_a = 5\ 000\ 000 \left(\frac{T_r}{T_a}\right)^3$$

where:

L_a = Average joint life, revolutions

T_r = Dynamic torque rating, lb-in

T_a = Actual torque for a particular operating condition, lb-in

This equation yields the approximate average joint life for a particular operating condition consisting of a given torque load, a rotational speed of 540 rpm, and a 10 deg joint angle. As defined by AFBMA, the basic dynamic capacity rating yields one million revolutions of B_{10} life. The average life is approximately five times the B_{10} life and, therefore, the constant of 5 000 000 is used in the equation. Also, for roller or needle roller bearings, the exponential power would be 10/3. The cubic exponent has been utilized in the equation because it yields a more conservative average life value.

When the joint angle for a particular operating condition is not 10 deg, the average life obtained must be corrected by multiplying it by an angle factor shown in Fig. 24. This correction factor is necessary since the life of the joint is affected significantly by the joint angle.

Rotational speed also affects the life of the joint in a particular operating condition. Since the speed factor does not change substantially between 540 and 1000 rpm, it is disregarded in the computations. The resulting average joint life can then be converted into hours.

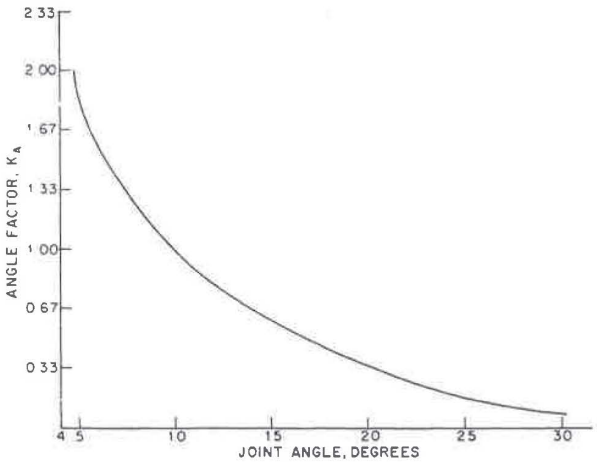


Fig. 24 — Joint angle correction factors

In actual practice, very few applications have a constant operating load condition as described in a previous section on *Torque*. Therefore, joint life is best determined by means of a load or duty cycle as shown in the chapter on *Basic Driveline Requirements*. A load cycle for an application is usually composed of two or more operating load conditions which represent fractions or percentages of total operating time of the load cycle.

For each of the individual operating load conditions, the joint life can be determined as previously explained. Then, the total life expectancy of the joint in revolutions or hours can be predicted as follows:

$$\text{Total Life Expectancy} = \frac{1}{\frac{N_1}{L_1} + \frac{N_2}{L_2} + \frac{N_3}{L_3} + \ldots + \frac{N_m}{L_m}}$$

where:

N_1 = Fraction of total time at operating load condition 1

L_1 = Life expectancy at operating load condition 1 as determined by previous average life equation, revolutions or hours

Safety Shields or Guards

The main PTO driveline from the tractor to the implement is located in an exposed area. In order to protect against inadvertent personal contact with the rotating driveshaft, a safety shield or guard must be used to cover the shaft as specified in industry standards published by the SAE and ASAE.

The safety shielding of the driveline, including all universal joints, shall be adequate to prevent the operator, or any observer, from coming in contact with positively driven rotating members of the driveline. In addition, this protection shall be adequate when operating during turns. The safety shielding shall be integral with, and journaled on the rotating members of the driveline.

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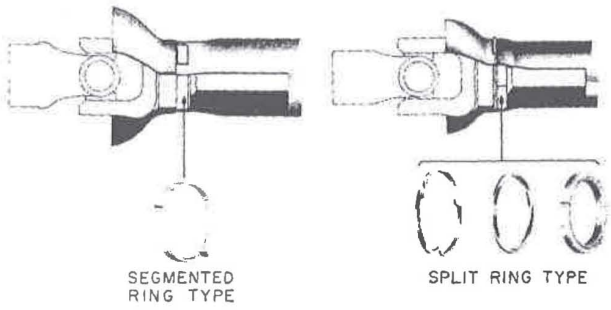


Fig. 25 — Bearings for safety shields

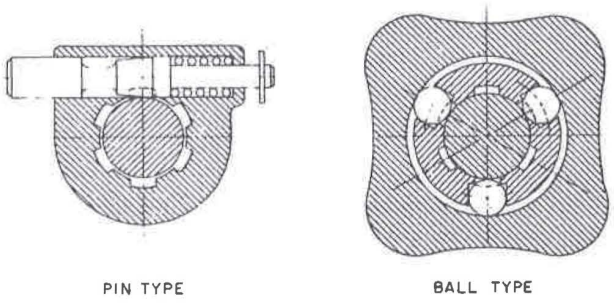


Fig. 26 — Quick-disconnect designs

It is shown in this chapter in the section on *Connecting Shaft Members* that the driveshaft length varies from a minimum when the tractor and implement make a maximum turn while operating in a depression, to a maximum when the tractor and implement are in a straight ahead position and operating over a hill. The safety shielding must be capable of accommodating these extremes of telescoping without separating or reaching a fully collapsed position. In the extreme extended position, the safety shielding, because it is not a torque transmitting component, does not require the same amount of overlap of mating members that is required by the telescoping driveshaft.

The two basic types of safety shielding are:

1. The safety shield that rotates with the driveshaft until an obstruction is encountered at which time it stops, while the driveshaft continues to rotate inside the shielding.
2. The safety shield that is stationary at all times with the driveshaft rotating inside the shielding. In this design, both ends of the shield are secured to the tractor and implement, generally with a chain.

The rotating type of safety shield is used in North America. Since relative rotation between the shield and driveshaft is infrequent, the plain bearings used are generally of a thermoplastic material having a comparatively low coefficient of friction. The bearings normally contact the universal joint yokes at each end of the connecting shaft in grooves which permit relative rotation when required, and also position the safety shield members on the driveshaft. Fig. 25 illustrates two types of bearings for safety shields used in North American applications. These two types of bearings are the segmented and split ring designs as shown.

Shaft Attachment

The attachment requirements for a driveline in an application are described in the Application Parameters and Established Practices section of the chapter on *Basic Driveline Requirements*. Also, the various ways in which this attachment can be accomplished are shown in the chapter on *Driveshaft Attachment Methods*. The three locations for attachments are as follows:

Tractor PTO Shaft — The agricultural implement driveshaft is designed for a specific implement and as such, is normally furnished as a part of the implement. The necessity of frequent coupling of the implement to the tractor requires rapid and relatively simple means of performing this operation. The most common means of attaching the driveline to the tractor PTO shaft is with a quick-disconnect type end yoke, which may have either a pin or balls engaging a groove on the splined shaft. For these quick-disconnect designs, the pin and balls retract for connecting or disconnecting and are locked into the groove when connected. Fig. 26 shows these two basic types of attaching means. Other modified attaching devices are available which contact the contour of the PTO shaft groove and thus decrease the stress between the groove and locking means.

Implement Input Shaft — Since the driveline is designed for and is a part of the specific implement, it is seldom disconnected from it. Therefore, the shaft attaching means at the implement end is different from that at the tractor end. In general, an end yoke is used at the implement end which is retained on the splined implement input shaft with a typical conical point set screw fitting into a mating conical hole on the shaft. A common lock nut is used to lock the set screw in position. Typical bolts or roll pins extending through the end yoke hub and shaft are also commonly employed. In addition, many other types of fastening means are used especially in two-joint driveshafts where the implement input may be a gear box, a pulley shaft, or a long round bar shaft. Various end yoke constructions are also shown and described in the chapter on *Driveshaft Attachment Methods* and in the section on the *Cardan or Hooke Universal Joint*.

Shaft Support Bearing Pedestal — When the driveline has three universal joints instead of two, it must be supported at an intermediate point with a shaft support bearing. This support is in addition to the end supports located at the tractor PTO shaft and at the implement input shaft. A typical shaft support bearing design for agricultural implement driveshafts is illustrated in Figs. 5 and 6, where the bearing is mounted to the implement in a pivoting and rigid type pedestal, respectively. As compared with typical automotive application requirements, the support bearing used with agricultural implement driveshafts is, in general, not rubber isolated. This type of bearing is described in the chapter on *Shaft Support Bearings* and the procedure for analyzing the secondary couple loads on the support bearing is discussed in detail in the section on the *Cardan or Hooke Universal Joint*. Also, recommended splined shaft sizes for these pedestal type bearings have been adopted and are described in references published by the ASAE.

Torsionally Resilient Driveshafts

The introduction of a torsionally resilient device such as a

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typical rubber element tube in an agricultural implement driveshaft, may aid in solving the following types of power transmitting problems:

1. Reduction of peak torques by the storage of energy: One such example is starting the rotation of a relatively large rotational inertia mass in an implement by rapid clutch engagement. A rubber element tube driveshaft will store some of the available energy and thus decrease the peak torque transmitted through the shaft.
2. Altering the natural frequency of a two-mass system: An example of such a system is a multiple knife rotating drum which produces a disturbing frequency that coincides with the natural frequency of the system and causes a resonance condition to occur. A rubber element tube driveshaft may be incorporated which can change the frequency of the system and, therefore, avoid resonance in the operating range.
3. Damping torsional vibration: An example of such a system is a two-joint driveshaft operating with unequal joint angles. Disturbing torque pulses produced by nonconstant velocity universal joint action occur twice per shaft revolution and, if they coincide with the natural frequency of the system, very high peaks will result. A rubber element tube driveshaft may be incorporated which will have a spring rate compatible with the system and, thereby reduce and dampen noise transfer.

The application and design considerations of torsionally resilient driveshafts are described in the Rubber Element Tubes section of the chapter on *Vibration Control*.

Auxiliary Drives

Driveshafts are also used to transmit power for auxiliary operations within some implements and are an integral part of these implements. Typical examples of such auxiliary operations are feed roller drives in forage harvesters and multiple blade drives in rotary cutters.

These universal jointed driveshafts require neither the large operating angles nor the long telescoping capability necessary in the main PTO driveshaft between the tractor and implement. The operating speeds for auxiliary operations can differ from the two standard PTO speeds and depend on the particular application. Auxiliary driveshafts when located in exposed areas should be adequately shielded. Where space and application requirements permit, it is advantageous to use as many of the main PTO driveshaft components as possible in the auxiliary drive system for standardization purposes.

Lubrication

In agricultural implement applications, Cardan universal joints as well as various types of sliding and rolling element axial motion or telescoping devices have been employed in the driveline systems. Both the universal joints and the telescoping mechanisms must be either lubricated for life at assembly or lubrication plugs or fittings must be included for periodic replenishment of the lubricant as required. The implement manufacturer's operating instructions should be consulted with regard to recommended lubricating procedures.

Typical lubricating grease for the Cardan joint is described in the section on that joint. For splines and various other telescoping mechanisms, typical lubricating greases are discussed in the chapter on *Splines and Axial Motion Devices*.

Typical basic performance requirements of lubricating greases which must be met and their various properties are described in the *SAE Handbook* under *Automotive Lubricating Greases*. In addition to providing for optimum life by reducing friction and wear at all bearing and sliding surfaces, the lubricants used must perform satisfactorily in all environmental conditions of the application.

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Axle Driveshafts

Chapter 14

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Chapter 14
Axle Driveshafts

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

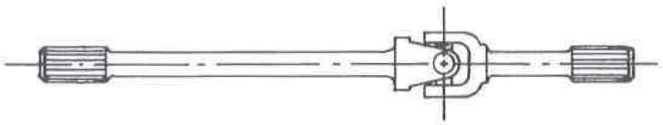


Fig. 1 — Steerable wheel axle driveshaft (live rigid axle type)

Introduction

The discussion of wheel driveshafts in this chapter will be limited to those used in universal jointed axle driveshaft constructions which have been used in front wheel, rear wheel, or all wheel drive passenger cars, trucks, and other highway, as well as off-highway vehicles. These applications may include an axle shaft with only one universal joint, or the shaft may have joints at both ends such as in the common halfshaft construction. The chapter on *Universal Joints* discusses in varying detail joints suitable for these applications. Depending on the type of suspension adopted for the vehicle, provision may have to be made for permitting an axial length change of the shaft while it is transmitting torque. This length change may be effected within the joint or joints themselves, or in the shaft itself by a separate splined or other end motion device. Generally, these axle driveshaft applications encompass one of the following three shaft constructions:

1. Steerable wheel axle driveshaft (live rigid axle type). In this application, only one universal joint is used in the live rigid axle shaft construction, as shown in Fig. 1, and is of the fixed center type. The joint may be either self-supporting or non-self-supporting depending on the axle and steering knuckle design features. The steering knuckle can be either of the open or closed constructions, as shown in Figs. 2 and 3, respectively.
2. Steerable or non-steerable wheel axle driveshaft (independent wheel suspension type). In this application, two universal joints are used in the halfshaft construction, as shown in Fig. 4. Depending on the independent wheel suspension design, the self-supporting joints may be either of the fixed center or end motion type at one or both ends. In addition to being the most common axle driveshaft design employed in vehicles with independently sprung driving wheels, this halfshaft construction is also used with de Dion rigid axles which are sometimes utilized.
3. Non-steerable wheel axle driveshaft (independent wheel suspension type). In this application, one universal joint is used in the halfshaft construction as shown in Fig. 5. The self-supporting joint may be either of the fixed center or end motion type, depending on the independent wheel suspension design. The suspension construction has been known as the swing axle type, once commonly used and now nearly obsolete because of its generally undesirable vehicle stability and handling characteristics.

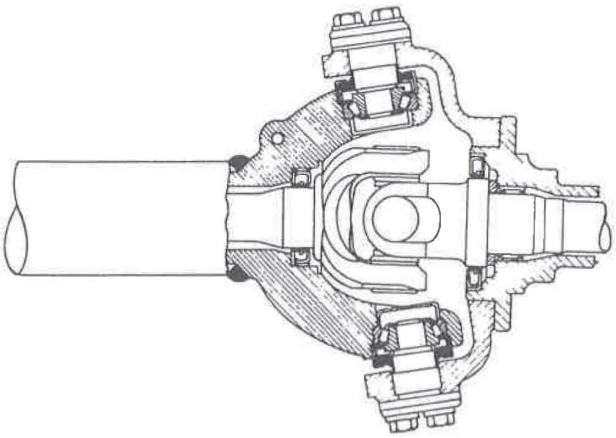


Fig. 2 — Open steering knuckle construction

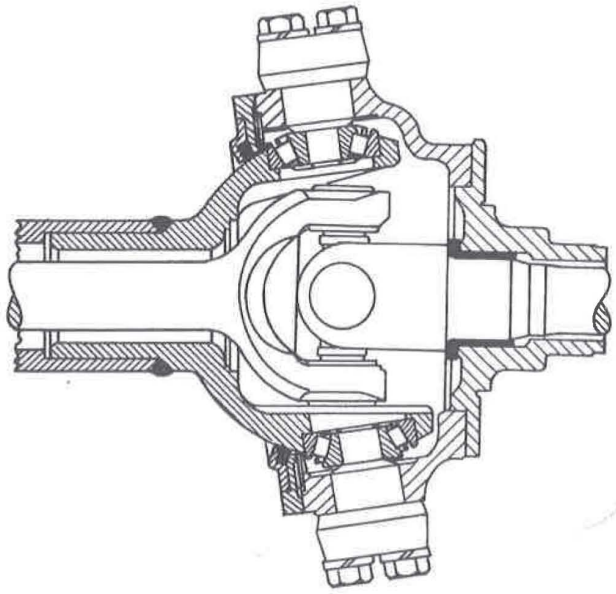


Fig. 3 — Closed steering knuckle construction

Design Parameters

The design parameters of universal joined axle driveshafts like the

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main or longitudinal driveline depend on the requirements established for a particular vehicle application. These parameters are as follows:

Engine, Transmission, and Final Drive Location

In a rear wheel drive vehicle where the engine and transmission are at the front, the rear frame mounted or sprung differential or final drive unit can readily be located to provide equal length axle driveshafts to the driving wheels as is also the custom in conventional unsprung live rigid axle constructions. In the case of an all wheel drive layout such as is used in a typical four-wheel drive vehicle, the required location of the front driveline to provide clearance as it passes the transmission and engine to the front axle results in an offset location of the front differential. In front live rigid axle design, the right and left hand driveshafts are, therefore, generally of unequal length. When an independent suspension is incorporated in such a vehicle, these shafts require two universal joints, and in order that they can be the same length, a longer output shaft from the differential must be used on one side than the other.

The engine, transmission, and final drive can also be combined as a unit and located at the front such as in a front wheel drive vehicle or positioned at midship or rear locations in a rear wheel drive vehicle. In these combined power train arrangements, the engine may be mounted either longitudinally or transversely. The overall design configuration of the power train will determine whether the right and left hand axle driveshafts can be the same length. When one halfshaft is longer than the other, a difference in torsional rigidity and transverse stiffness exists if the shafts are of the same diameter. Further, the directional stability of the vehicle may be affected by such shafts when they are subjected to high torque load applications or changes in torque.

In addition, differences in joint angles and shaft length changes or axial stroke occur in unequal length shafts. If this shaft length difference is considerable, problems with respect to excessive angularity of the joints on the shorter shaft could arise. Also, in front drive installations with independent suspension, this shaft length difference can result in limitations on the wheel steer angles and the reactions at the steering wheel may be affected. These characteristics must be studied for acceptability with regard to the various suspension and steering movements of the independently sprung wheels.

Further, in front drive vehicles, the final drive output shafts are generally ahead of the steerable wheel centerlines so that the best use is made of the total available angular movement of the outboard universal joints. This angle of trail of the shafts provides the largest steering lock angles, and thus a minimum turning circle for the vehicle, because the Ackerman steering effect of the front wheels produces unequal angular deflection of the inside and outside wheels during turns. Also, this angle of trail or rearward sweep of the shafts ensures that both outboard and inboard universal joints never operate at zero degree joint angles for any possible vehicle loading conditions during all continuous straight ahead driving modes.

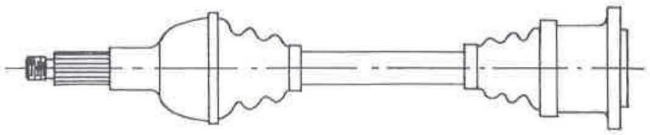


Fig. 4 — Steerable or non-steerable wheel axle driveshaft (independent wheel suspension type)

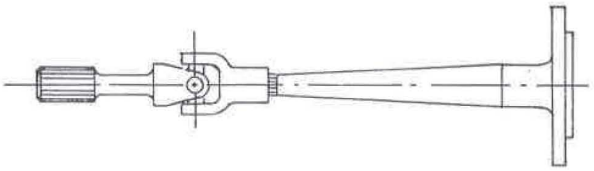


Fig. 5 — Non-steerable wheel axle driveshaft (independent wheel suspension type)

Maximum Applied Torque and Brake Location

Since the axle driveshafts are located between the vehicle differential or final drive unit and the driving wheels, the shaft torque is not only higher than in the vehicle's main driveline, but the load intensities due to shock and impact can also be higher, depending on the drivetrain arrangement. These higher shock and impact factors can especially occur in vehicles where the entire power train is located at the same end of the vehicle as the driving wheels, such as in front engine-front drive or rear engine-rear drive arrangements.

In these installations, the power flow path from the engine to the driving wheels is relatively short and torsionally stiff when compared to the more commonly used rear drive vehicle drivetrain arrangements where the engine and transmission are located at the front while the final drive unit is at the rear. In this more commonly used vehicle drive system, a relative torsionally elastic driveshaft connects the engine-transmission unit to the final drive unit, which in turn is connected to the driving wheels with either live rigid or universal jointed axle shafts which also are relative torsionally elastic members. Thus, this more commonly used power train arrangement has a much longer power flow path and usually provides greater torsional flexibility than either a front engine-front drive or rear engine-rear drive installation. Therefore, shock and impact factors on the various drivetrain components are generally not as severe as on similar components in front engine-front drive or rear engine-rear drive vehicles.

When large high powered vehicles incorporate power trains with relatively short power flow paths such as in typical front drive layouts, a torsionally stiff drive system is required to transmit the relatively high torque. The lack of sufficient system torsional compliance can result in a power-on to power-off clunk in the vehicle due to normal lash or clearances in the drivetrain components. If desired,

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or necessary, it is possible to incorporate a suitable damping means in the drive system to reduce to a tolerable level or eliminate this undesirable noise.

The location of the brakes and their type can also dictate the maximum torque which the shafts must be capable of transmitting. The maximum torsional loads which can be developed by various loading modes in an axle driveshaft or any of its components are discussed in the chapter on *Basic Driveline Requirements*.

When outboard or wheel end unsprung brakes are used, which is the common practice, the maximum transmitted torque is generally determined by the maximum engine delivered torque. This torque is the maximum installed net engine torque multiplied by the appropriate converter stall, transmission low gear, transfer case, and front or rear final drive ratios as applicable. In some installations, however, wheel slip may control the maximum torque which an axle driveshaft experiences. In some applications, shock torque as compared to steadily applied torque can be significantly higher and must not be neglected when determining shaft and universal joint size as shown in the chapter on *Basic Driveline Requirements*. Thus, braking torque does not affect the axle shafts of a vehicle equipped with outboard brakes.

When the vehicle has inboard located (chassis mounted or sprung brakes), the maximum engine delivered torque, with brakes applied, does not reach the axle driveshafts. These shafts, however, must resist the full effects of the braking torque. Braking torque as compared to maximum engine delivered torque through an automatic transmission (stall torque), for example, is instantaneous in nature and depending on the type of brakes, can have a relatively high shock or impact factor. In an inboard braking system, the decelerating effects of the wheel rotational inertia also must be considered with the braking torque. Therefore, inboard located brakes can significantly affect shaft size as well as universal joint size. However, the maximum torsional loads which can be developed by various other possible loading modes must not be neglected when determining shaft and joint size, as shown in the chapter on *Basic Driveline Requirements*.

Shaft Speed

Unlike the main driveline, critical speed generally does not have to be considered in most axle driveshafts due to their location between the differential and the driving wheels where the speed is reduced, and because of their generally short length. In a rear drive vehicle, if the differential does not have a torque biasing or limited slip feature, design considerations may have to be included, however, in some universal joint rotating boot seal designs. These design considerations also have to be employed in front axle driveshafts since a locking differential is not used at the steerable wheel end of a vehicle. When one wheel is spinning free due to loss of traction and with the opposite wheel remaining stationary, the resulting momentary or prolonged higher rotational speed may cause the seal to "balloon" excessively and interfere with other vehicle components.

Bending Stiffness

It was previously stated that unlike the main driveline, critical speed is in general not of importance in most axle driveshaft applications. However, in some vehicle installations, and especially where the right and left hand shafts are of unequal length and if this length difference is considerable, resonant bending or beaming frequencies may be present at certain operating conditions due to the longer shaft. These resonant beaming frequencies can be altered by selecting an optimum shaft size that may achieve a more desirable transverse stiffness. It is possible to raise these resonant beaming frequencies by simply increasing the transverse stiffness. For example, when the shorter shaft is a solid shaft, the longer shaft can be either a larger diameter solid shaft or a still larger diameter part tubular shaft to stiffen this shaft and thus effect a more balanced transverse stiffness for the two shafts. This increased transverse stiffness will reduce beaming excitation and result in a higher critical speed.

In some applications, it may be desirable or necessary to lower the resonant beaming frequency of the longer shaft below the exciting frequency of the drivetrain system. For unequal length axle shafts of the same diameter, this may be accomplished by clamping a mass damper on the longer shaft midway between the outboard and inboard universal joints. This concentrated weight of suitable size reduces the shaft's resonant bending frequency or critical speed.

The resonant bending frequency of the longer shaft may also be lowered by reducing its diameter. However, this reduction in shaft diameter is limited by the minimum size required to transmit the maximum applied torque. A still further lowering of the resonant beaming frequency of this reduced size longer shaft can be accomplished by attaching a mass damper as previously stated. Whatever means are used to lower the longer shaft's resonant beaming frequency, its strength and critical speed characteristics must be of such values which will permit the shaft to function safely at all required operating conditions.

Torsional Stiffness

Like the main driveline, the axle driveshaft serves as a relative torsionally elastic member in the drivetrain system and in its simplest form it is a torsional spring element connecting two masses of relatively large rotating inertias. This system can be subject to disturbing forces depending on the specific installation. When the drive axle layout, for example, involves unequal shaft lengths, and if this length difference is considerable, resonant torsional frequencies may be present at certain operating conditions due to undesirable torsional deflection relationships between the two shafts. It is possible to raise the resonant torsional frequencies above the exciting frequencies by simply reducing the twisting or windup of the longer shaft. For example, when the shorter shaft is a solid shaft, the longer shaft can be

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either a larger diameter solid shaft or a still larger diameter part tubular shaft to stiffen this shaft and thus effect a more balanced torsional deflection characteristic between the two shafts, if this is desired. Also, as previously indicated, the use of a larger diameter solid shaft or part tubular shaft will reduce beaming disturbances by stiffening the shaft in bending. When it is desired to increase torsional compliance by reducing shaft size, the minimum shaft size selected must be capable of transmitting the maximum applied torque and perform safely at the maximum operating speed previously described, as well as not result in undesirable beaming excitation at certain operating conditions.

In a given application, as previously indicated, an axle driveshaft may be required to provide a desired torsional rate or rigidity. Thus, it will be necessary to analytically estimate the overall torsional stiffness of the proposed design. This axle driveshaft is actually an assembly of a number of flexible elements in series, which are not of uniform diameter and cross section throughout the length of the assembly. The shaft itself may be of a constant or varying solid diameter, or it may be tubular or part tubular. In general, the tubular or part tubular construction has ends of reduced solid diameters, which provide for the attachment of universal joints with splined connections. Depending on the application, the axle driveshaft may be required to provide for axial length changes which may be incorporated in one or both joints or this function can be provided by a separate splined, or other axial motion mechanism.

The torsional rigidities of the individual universal joints and the separate axial motion member, if used, can be determined with reasonable accuracy only by laboratory test. Torsional stiffnesses of the various solid or tubular sections of the shaft can be readily computed if desired. The computation procedure for determining the overall torsional rigidity of an axle driveshaft is shown in *Appendix F*.

Wheel Located Final Drive Unit

Some vehicle applications, in addition to having a gear reduction in the conventional final drive and differential unit, have another gear reduction located at the wheel. Due to the location of this final drive unit, the axle shaft, therefore, rotates at a speed higher than wheel speed. This higher speed must be considered in selecting the axle shaft size with regard to critical speed, beaming disturbances, and torsional excitation.

Rear Suspension Type

Certain types of independent rear suspensions require that the axle driveshaft serve a dual purpose. In addition to transmitting the driving torque, the axle shafts also either serve as structural members or provide for a partial correction of annoying and excessive vehicle body pitching motions during acceleration or braking. These dual functioning axle driveshafts are described as follows:

Axle Driveshaft as a Suspension Structural Member—The axle driveshaft can be incorporated as a structural member in certain types of independent rear suspensions by serving as a transverse control arm. With the shaft functioning as a fixed length control arm, the necessity of sliding splines or an end motion type of universal joint or joints is eliminated. Fixed center type of universal joints are, therefore, required which can withstand the shaft thrust forces. The magnitude of these axial forces and whether they are tensile or compressive will depend on the suspension design and the severity of the cornering forces imposed on the vehicle. The vehicle acceleration and braking forces are transmitted to the frame by means of another suspension member or members. By having the axle shaft serve as one of the suspension structural members, it is possible to reduce the unsprung weight at the rear of the vehicle and increase ground clearance.

Axle Driveshaft with Anti-Squat and Anti-Lift Characteristics—Rear drive vehicles equipped with certain types of independent rear suspension display a marked tendency to squat during acceleration and rise during braking due to weight transfer. Corrective means can be provided between the differential and the driving wheel, which results in an increase or decrease in the effective length of the axle driveshaft in response to the transmitted torque. This increase or decrease in effective shaft length is accomplished by the torque responsive axial thrust forces. The suspension by its geometry makes use of this extension and reduction in effective length to induce inward or outward forces to substantially balance the downward and upward forces exerted on the suspension by the rear portion of the vehicle under the influence of acceleration and braking. This feature has been commonly called anti-squat and anti-lift, respectively.

The corrective means to accommodate this variable shaft length can be provided, for example, by an end motion device in the driveshaft separate from the universal joint or joints. This end motion device can be a typical rolling element spline mechanism with raceways disposed at the proper helix angle to provide the necessary torque responsive axial thrust forces.

It is also possible to incorporate this corrective means within an end motion nonconstant or constant velocity universal joint or joints to provide the extension and reduction of the shaft effective length. Certain end motion ball type constant velocity joints, in particular, can allow their ball groove geometries to be so designed to provide the necessary torque responsive axial thrust forces required to increase or decrease the shaft effective length.

"Floating" Rear Axle Shaft

Often rear double jointed halfshafts incorporate end motion capability in both joints in order to provide the necessary axial stroke or telescoping required during the entire range of suspension travel. However, in such a construction the shaft member can "float", which is to say that if no

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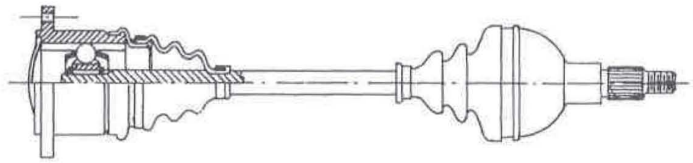


Fig. 6 — Axle driveshaft (solid bar shaft design)

special measures are taken, the shaft can move freely in an axial direction. The position of this shaft is, hence, undetermined and may be in any unstable position where it can be subjected to the influence of inertia or any external forces and thus the shaft may wander to an undesirable end position where excessive wear may occur depending on the specific universal joint design.

In some halfshafts employing certain types of end motion joints, it has been possible to maintain the shaft member in a central or normal position by means of separate devices such as springs located within both joints. Also, it is possible with those and other types of end motion joints to normally maintain the shaft at one end position with a spring device at the opposite end and provide satisfactory functioning of the driveshaft.

With still other end motion joint constructions, however, it is desirable to maintain the shaft normally in a central position from where it may be displaced, as required, with respect to the joints during suspension motion. It is possible in some of these joint constructions to provide this movement for the shaft and its return to the central position without the aid of springs. A characteristic can be incorporated in the cooperating ball groove geometry of the inner and outer members of these joints. This characteristic exerts on the "floating" shaft member a progressively increasing restoring force which seeks to return the shaft member to the central or normal position after the shaft has moved from this latter position under the influence of certain disturbing axial forces.

Joint Angle and Axial Stroke

There are numerous factors which can affect the magnitude and frequency of the joint angle and shaft length changes in a typical axle driveshaft. These factors are power train layout, location of axle driveshaft (front or rear), suspension type, and installation tolerances.

It was previously described how the layout of the engine, transmission, and final drive in a vehicle, whether in a separated or combined unit construction, can affect joint angles and shaft length changes of the axle driveshafts. When a typical double jointed halfshaft is used at the rear, the joint angle and shaft length changes result only from the periodic wheel movement determined by the independent suspension system geometry. However, when such an axle driveshaft and an independent suspension system are incorporated at the front, both joint angles and the axial length are affected by the steering as well as the suspension geometry. In addition, movements of the power train relative to the frame will further affect joint angles and shaft length. When a live rigid axle is employed at the front, the joint angle changes encountered are only those caused by the steerable wheel.

An axle driveshaft which permits shaft length changes for the necessary suspension, steering, and power train movements must also provide additional axial motion capability for installation or build tolerances. For a given end motion joint design used in an axle driveshaft, the maximum intermittent operating angle throughout its entire axial stroke range must, therefore, be within the limit lines of the angular capability chart as described in *Appendix G*.

Clearances

The allowable space for the axle driveshaft universal joint or joints is generally determined by the design of the various vehicle underbody, frame, engine, transmission, final drive, and suspension components. The various movements of the suspension as well as the steering motions in the case of steerable wheels may especially dictate the swing diameter available for the joint at a specific location.

Environment

The vehicle application requirements as well as the location of the axle driveshaft beneath the vehicle and its proximity to the engine, transmission and final drive, or the driven wheel, can result in the universal joint or joints being subjected to varying environmental operating conditions. When elastomeric boot seals are used on universal joints, they may be especially vulnerable to flying objects such as stones and dirt, in addition to being exposed to wide temperature variations while performing their sealing function. When a universal joint is used at the inboard end, for example, its closeness to the engine, transmission and final drive unit, and the exhaust system, such as in a front wheel drive vehicle, can result in the joint being subjected to relatively high temperatures in certain power train systems.

Shaft Construction and Attachment Methods

Usually, axle driveshafts are made of round solid or bar shaft construction, as shown in Fig. 6 in a typical halfshaft design. However, they can also be made of tubular or part tubular design, as illustrated in Fig. 7 and 8, respectively. Generally, the tubular or part tubular construction is used in some axle designs to effect ease of attachment of certain universal joints or other drivetrain components to the tube ends, reduce cost and weight, expedite manufacturing, or to meet certain design parameters or requirements of the vehicle application.

The attachment of joints or other drivetrain components to axle shafts can be effected in a number of ways. When solid axle shafts are used, these joints or other components are usually attached with splined torque-transmitting connections and held in position with retaining rings. In some instances, a joint component may also be formed integral

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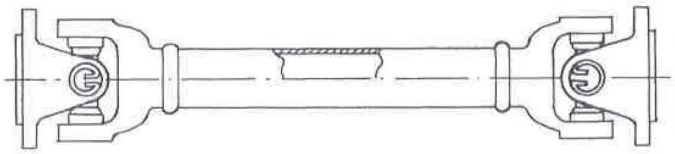


Fig. 7 — Axle driveshaft (tubular shaft design)

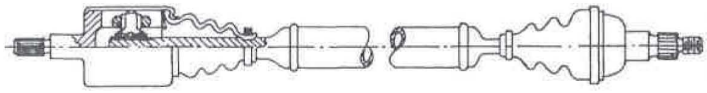


Fig. 8 — Axle driveshaft (part tubular shaft design)

with the shaft end. In the case of tubular shafts or those of part tubular construction, joint components or stub shafts are generally welded to the tubes. Also, some tubular shaft constructions have had their ends hot upset forged or formed by other means to a solid shaft configuration so that joint components can be attached with splined connections.

After assembly of the universal joints to the axle shafts, the axle driveshaft assemblies can in turn be fastened to other drivetrain components in a variety of ways. The splined torque transmitting connection is frequently used in conjunction with a threaded nut, retaining ring, retaining pin, or a compression spring as the retention means. Still another means of attachment is to shape the joint housing or outer member as a disc or with a flange so that the joint can be held to its mating member with bolts. In the case of the disc type outer member, it has also been connected to its mating member with splines located on the disc outer surface. When a Cardan joint or joints are used in axle driveshaft applications, the common main driveline attachment methods, such as splined connections, flange yokes, U-bolts, or wing bearings, are also employed. In the chapter on *Driveshaft Attachment Methods* these various attachment means are also discussed.

Torsional Strength and Twist

In addition to the principal loading mode, which is torsional, bending and axial thrust forces also may be imposed on the axle driveshaft due to the different possible shaft configurations employing one or two universal joints of various types in either a fixed or variable shaft length construction. Bending forces, for example, can be imposed on the shaft by universal joint secondary couple effects. Axial thrust forces can be developed when the shaft in a fixed length construction serves as a suspension structural member, or when a shaft with a variable length construction uses either fixed center joints with separate end motion means, such as sliding splines, or an end motion joint or joints. These types of loading may or may not be present at the same instant, and if any of them are present simultaneously, the combined or resultant stresses that occur can be higher than those computed for any individual loading mode. Thus, the size of the shaft, including its method of attachment to mating drivetrain components, can be affected to a considerable degree by the various loading modes developed by the type of suspension, universal joints, or separate end motion means, such as sliding splines, selected for the particular application. Splines are also commonly used to fixedly attach joints to the shaft itself and to other connecting power train members. The chapter on *Splines and Axial Motion Devices* can aid in the selection and computation of suitable size splines. The magnitude and characteristics of universal joint secondary couple effects depend on the particular type of joint used and are described in the kinematics section for the appropriate joint under consideration.

The torsional strength of the shaft is determined not only by the shaft size and geometry, but also by the type of material used and the heat treatment method employed. Other processing procedures, such as cold rolling, roll forming of spline, and shot peening can further affect shaft strength.

Round Solid Shaft

For a round solid shaft, the nominal shear stress in torsion has a maximum value at the outermost fiber and is:

$$S_s = \frac{T}{Z_p} = \frac{16T}{\pi d^3}$$

where:

T = Torque, lb-in
Z_p = Polar section modulus, in^3
d = Shaft diameter, in

The nominal torsional shear stress S_s applies to a shaft without a point of stress concentration or stress raiser. When a shaft has a stress raiser, the maximum or effective stress S_s' is affected by the geometric or theoretical stress concentration factor K_t and is expressed as follows:

$$S_s' = K_t S_s$$

Therefore, it is possible to compute the maximum or effective stress from static ultimate and yield torque test data for a given shaft. The yield torque can be determined either by the offset or the Johnson's apparent elastic limit (JAEL) method. In the offset method, the yield torque is the torque at which the shaft reaches a specified permanent set or plastic strain. A value of 0.2% has been widely accepted. The JAEL method determines the torque at which the rate of deformation is 50% greater than the initial rate of deformation and results in a lower yield torque value than the 0.2% offset method. Both of these methods used to determine the yield stress are described in detail in *Appendix L.*

The torsional shear yield stress thus determined from test data can be used with sufficient accuracy to predict the strength of other shaft designs having the same material and heat treatment, as well as the approximate same size and geometric construction. For a given shaft design with a

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varying geometric configuration along its length, the critical section governing the shaft strength is determined by this maximum stress which is dependent on the combined effect of the effective diameter at this particular section and the stress concentration factor. *Appendix D* shows values of K_t for a grooved shaft and for a shaft with a shoulder fillet. Values of K_t for other common stress raisers can be found in the references listed in *Appendix D*.

For a round solid shaft, the angle of twist in radians per unit of length is:

$$\phi = \frac{T}{GI_p} = \frac{32T}{\pi d^4 G}$$

where:

G = Modulus of rigidity or modulus of elasticity in shear, psi
I_p = Area polar moment of inertia, in^4

Round Tubular Shaft

For a round tubular shaft, the torsional shear stress has a maximum value at the outermost fiber and is:

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

where:

d_o = Outside diameter, in
d_i = Inside diameter, in

The angle of twist in radians per unit of length for the round tubular shaft is:

$$\phi = \frac{T}{GI_p} = \frac{32T}{\pi(d_o^4 - d_i^4)\,G}$$

Shaft Material

The selection of the optimum steel for the solid or tubular axle shaft material depends not only on the previously described design parameters which dictate the ultimate strength, fatigue strength, and other functional requirements, but also on economical processing advantages. In general, solid axle shafts are made from bar stock. When the shaft is made integral with one of the universal joint driving members or when a stub shaft is required for typical attachment by welding to a tubular member or shaft, the component is usually made from a forging or formed blank. In some passenger car axle driveshaft designs in particular, tubular type shafts are also utilized. These tubes can be made from typical driveshaft quality steel tubing discussed in the chapter on *Tubing* when Cardan joints, for example, are employed in an axle shaft serving as a suspension structural member in certain types of independent rear suspensions. In other axle shaft constructions, various other types of welded or seamless steel tubing with suitable mechanical properties are also used. Whatever type of material and heat treatment procedure is utilized, the shaft, whether solid or tubular, must be designed to safely transmit the maximum applied torque of a given application.

In solid axle shaft design, it is possible to provide for a wide range of ultimate yield and fatigue strengths depending on the type of steel used and the heat treatment method employed. Medium carbon alloy steels can be through hardened to suitable hardness ranges and may result in acceptable levels of strength for some applications. Low carbon alloy steels can be carburized to a suitable case depth and surface hardened to a typical R_c60 minimum. Plain medium carbon or medium carbon alloy steels can be induction hardened. The induction hardening process, which is most commonly used in axle driveshaft applications, allows the selection of a relatively high case depth and a suitable surface hardness such as a typical range of R_c45–54 or higher. For optimum surface and case hardening response in the induction hardening process, the core structure must be initially developed to a satisfactory hardness range depending on the type of steel used. Both surface hardening procedures can provide for significant improvements in strength over the through hardening process, with induction hardening being capable of achieving the greater increase in strength and fatigue life while resulting in the least distortion of the shaft during heat treatment. This is due primarily to the difference in characteristics of the two surface hardening processes. Often the shafts are also shot peened for additional fatigue strength and life.

The limiting design stress levels for a given size shaft can be determined for the proposed material depending on the heat treatment process, stress concentrations resulting from specific geometric features, and the expected fatigue life. For example, when a through hardening steel is selected, the ultimate and yield strengths in shear can be easily estimated. For steel the ultimate tensile strength is a function of material hardness and can be determined as follows:

$$S_{ut} = 500Bhn$$

where:

S_{ut} = Ultimate tensile strength
Bhn = Brinell hardness number

The ultimate strength in shear can then be approximated as $0.8S_{ut}$ for ferrous materials. Also, the tensile yield and shear yield strengths vary somewhat with material hardness. Fig. 9 shows a plot of average tensile yield strength as a percent of the ultimate tensile strength. Fig. 10 shows a curve of the average shear yield strength versus hardness in Rockwell "C" values. If desired, it is also possible to estimate the average shear yield strength from the average tensile yield strength for ductile materials by using the following relationship based on the distortion energy theory:

$$S_{ys} = 0.58\ S_{yt}$$

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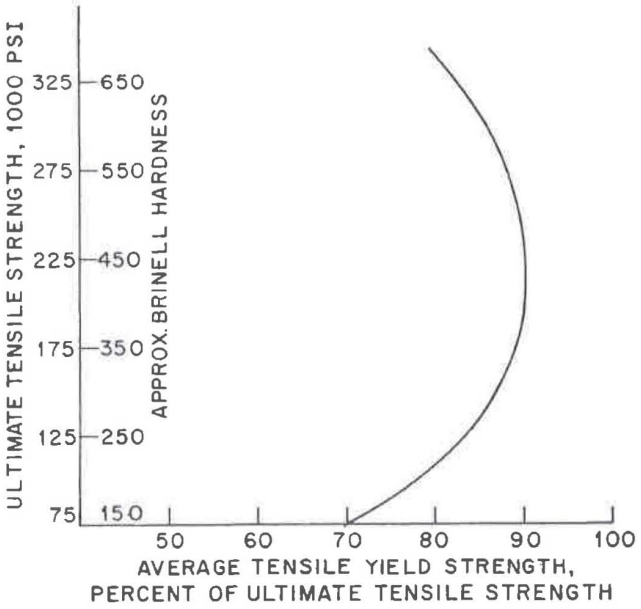


Fig. 9 — Average tensile yield strength as a percent of ultimate tensile strength

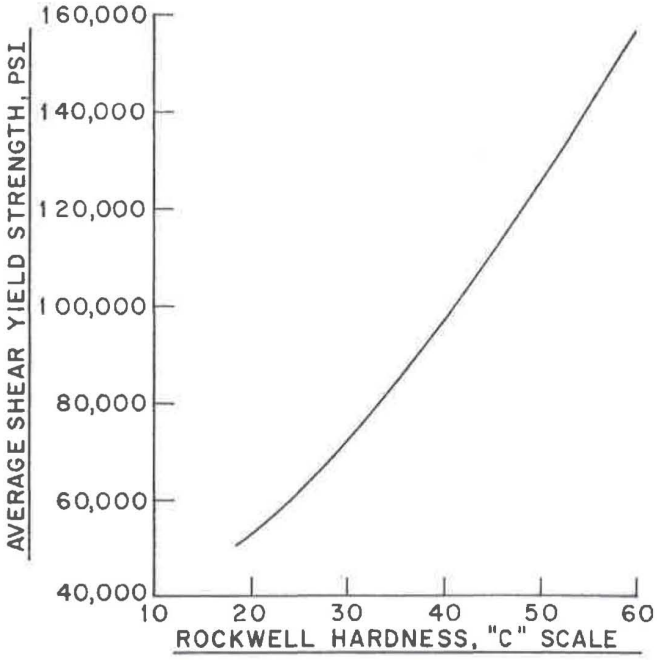


Fig. 10 — Average shear yield strength versus Rockwell "C" hardness

where:

S_{ys} = Shear yield strength
S_{yt} = Tensile yield strength

When a shaft is carburized and surface hardened, it is possible to gain 20-40% or more in strength over a through hardened member and when induction hardened an even greater improvement in strength can be realized. The increase in strength gained by surface hardened components is dependent upon the shaft section size-to-case depth ratio obtained and the geometry of the components.

The limiting design stress levels to be used for each surface hardening procedure should be based primarily on laboratory static torque tests where the ultimate and yield strengths in torsion can be found for various size shafts with similar geometric configurations and proportions. The failures in typical test samples will occur in the weakest portion of the shaft which is governed generally by the nature of the stress concentration at that point. Thus, at the torsional yield loads, the corresponding nominal shear stresses can be computed at various critical points along the shaft. If the stress concentration factor at a given point is known, the actual maximum or effective stress at this point can be determined. In general, it is the practice to refer to a basic shaft size without a stress concentration factor when selecting the proper shaft diameter. Usually, this basic shaft diameter corresponds to the spline minor diameter, although the pre-rolling or rolling blank diameter of cold formed splines which approximately equals their pitch diameter has also been used.

For example, when computing the basic shaft size of an induction hardened shaft with no stress concentration considered in a typical passenger car front drive application, the allowable nominal torsional shear design yield stress can range from 110 000 psi for a 1.290 in diameter shaft to 140 000 psi for a 0.720 in diameter shaft. This typical design yield stress range depends not only on shaft size, but also on utilizing the proper material, case depth and induction hardening process.

In other on-highway or off-highway applications requiring larger size shafts, lower levels of stress should be used. These lower stress levels are necessary in the larger shafts, not only because of the relatively shallower case depths in proportion to shaft section size obtained by induction hardening, but also because of the more severe usage where significant shock effects can occur. Actual fatigue test data for an axle driveshaft can be used to determine an S-N curve, and from this curve it is possible to estimate fatigue life at various stress levels for a given shaft or other shafts which are sufficiently similar in geometric construction.

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Nomenclature and Terminology

Chapter 15

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Chapter 15
Nomenclature and Terminology

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Basic Driveshaft Terms

Driveshaft—An assembly of one or two universal joints connected to a solid or tubular shaft member. (Figs. 1 and 2).

Driveline—An assembly of one or more driveshafts with provisions for axial movement, which transmits torque and/or rotary motion at a fixed or varying angular relationship from one shaft to another. (Fig. 3)

Driveshaft Length, Center to Center—The distance between the outermost universal joint centers on a driveshaft. On driveshafts with fixed centers, it is the nominal dimension, while on driveshafts with variable length centers, it is the compressed and extended lengths. (Fig. 1)

Slip—The permissible length of axial travel. (Fig. 1(b))

Phase Angle—The relative rotational position of the universal joint yokes on a driveshaft or driveline. (Fig. 1(a))

Critical Speed—The speed at which the rotational speed coincides with the transverse natural vibration frequency of the driveshaft. (Refer to chapter on *Critical Speed*)

Balancing—A procedure by which the mass distribution of a rotating body is checked and, if necessary, altered in order to ensure that the vibration at a frequency corresponding to an operating speed at the supporting bearings is within specified limits. (Refer to chapter on *Balancing*)

Driveshaft Types

Two-Joint Outboard Slip Driveshaft—A driveshaft or part of a driveline with a universal joint at each end. Axial movement is provided outboard of joint centers. (Fig. 1(a) and Fig. 2)

Two-Joint Inboard Slip Driveshaft—A driveshaft or part of a driveline with a universal joint at each end. Axial movement is provided inboard of joint centers. (Fig. 1(b)).

Single Joint Coupling Shaft—The coupling member or members of a multiple joint driveline which consists of a universal joint, tube, shaft support bearing, and a slip or fixed spline shaft. (Fig. 1(c) and Fig. 3)

Driveshaft Components

This listing of component definitions includes only the more common components other than universal joints which may be used in typical driveshafts. Various universal joints and their components that are utilized in driveshaft applications are defined under a following heading on universal joints.

Tube—The tubular connecting member of a driveshaft. (Fig. 2)

Swaged Tube—A tube with one or both ends having a smaller diameter than the middle portion. (Refer to chapter on *Tubing*)

Tube Diameter—The outside diameter of a tube. (Fig. 1(a))

Rubber Element Tube—Consists of an inner tube and an outer tube joined by the torsionally resilient rubber element. (Fig. 2)

Rubber Element—Consists generally of a number of compressed elastomeric rings or a molded elastomeric section. (Fig. 2)

Liner—Consists of a sound and vibration deadening material added to the inside surface of a tube. (Fig. 2)

Torsional Damper—A tuned mechanical device which consists generally of an inertia ring attached to a drivetrain component by means of an elastomeric inner ring. (Fig. 2)

Shaft Support Bearing—Consists usually of a rolling element bearing isolated in rubber and a bracket configuration for attachment. (Fig. 3)

Spline—A machine element consisting of integral keys (spline teeth) or keyways (spaces) equally spaced around a circle or portion thereof. (Fig. 5)

Basic Universal Joint Terms

Universal Joint—A mechanical device which can transmit torque and/or rotary motion from one shaft to another at fixed or varying angles of intersection of the shaft axes. (Refer to chapter on *Universal Joints*)

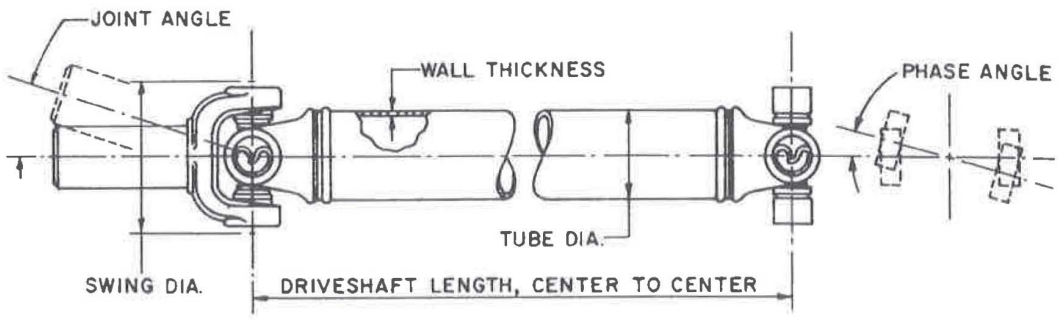
Nonconstant Velocity Universal Joint—A universal joint which transmits motion with a variation in angular velocity between the output and input members when operating at joint angles greater than zero. The average angular velocity ratio is unity. (for example: the Cardan or Hooke joint)

Constant Velocity or CV Universal Joint—A universal joint which transmits motion with an angular velocity ratio of unity between output and input members. (for example: the Rzeppa joint)

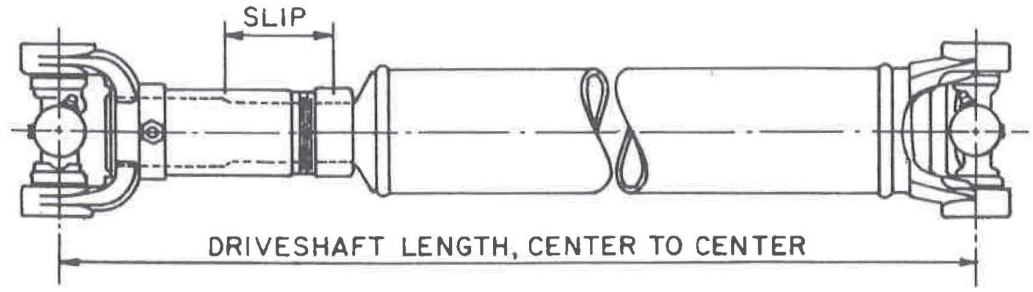
Near Constant Velocity Universal Joint—A universal joint which transmits motion with an angular velocity ratio of unity when operating at the design joint angle and at zero. When operating at other angles, the angular velocity ratio is near unity. (for example: the double Cardan joint)

Self-Supporting Universal Joint—A universal joint supported by internal means. (for example: the Cardan or Hooke joint)

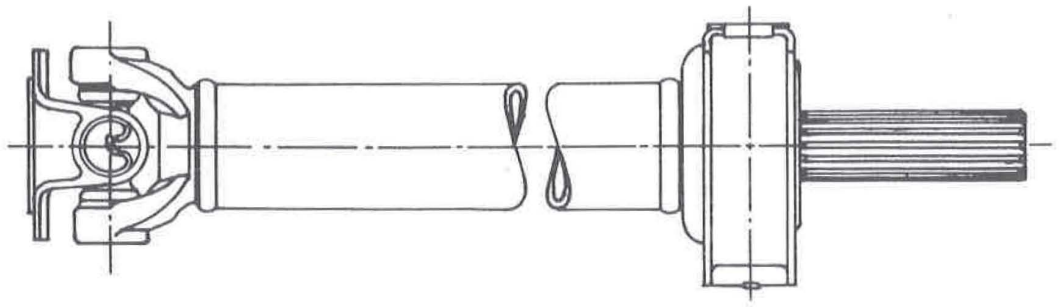
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(a) TWO-JOINT OUTBOARD SLIP DRIVESHAFT



(b) TWO-JOINT INBOARD SLIP DRIVESHAFT



(c) SINGLE JOINT COUPLING SHAFT

Fig. 1 – Basic driveshaft constructions

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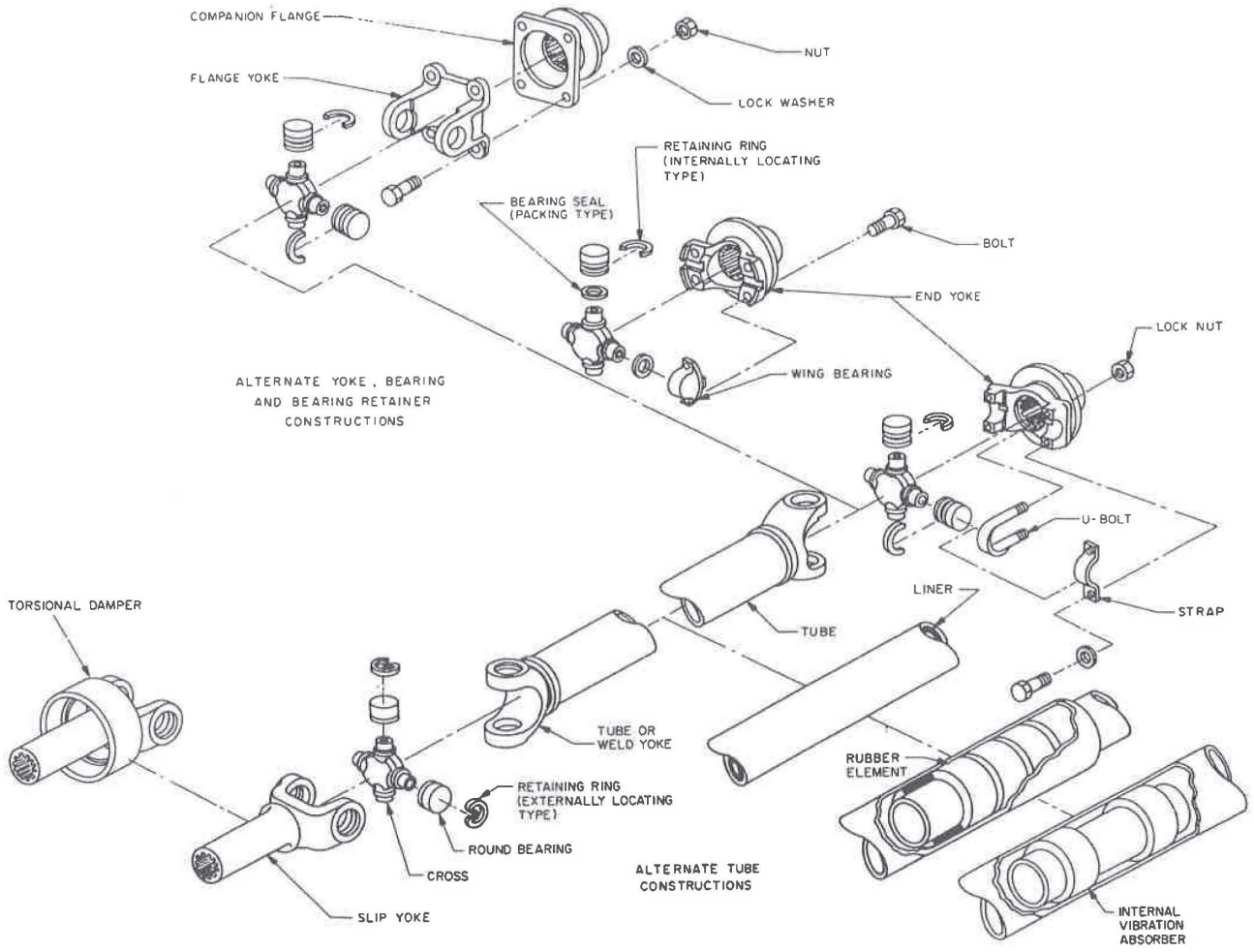


Fig. 2 — Two-joint outboard slip driveshaft

Nonself-Supporting Universal Joint—A universal joint which requires an external means of support. (for example: the Tracta joint)

Fixed Center Universal Joint—A universal joint which maintains the joint center at a fixed location and can resist axial thrust forces. (for example: the Cardan or Hooke joint)

End Motion Universal Joint—A universal joint which permits relative axial movement between input and output members resulting in a variable joint center location. (for example: the cross groove joint)

Joint Angle—The acute angle described by the intersection of the rotational axes of the input and output members of a universal joint and measured in the plane described by these axes. (Fig. 1(a))

Swing Diameter—The maximum diameter of the circular path described by a rotating universal joint. (Fig. 1(a))

Constant Velocity (or Homokinetic) Plane—The plane described by all contact points of a universal joint which produces an angular velocity ratio of unity between the output and input members. This plane bisects the obtuse angle formed by the input and output member rotational axes and is perpendicular to the plane containing these axes. (Refer to chapter on *Universal Joints*)

Torsional Equivalent Angle—The joint angle of a single nonconstant velocity universal joint which produces the same speed variation as a driveline with two or more nonconstant velocity universal joints. (Refer to chapter on *Universal Joints*)

Inertia Equivalent Angle—The joint angle of a single nonconstant velocity universal joint which describes the inertia effects of a driveline. (Refer to chapter on *Universal Joints*)

Inertia Drive Equivalent Angle—The joint angle of the first nonconstant velocity universal joint in a two-joint system. In a three- or multiple joint system, it is the joint angle of a single nonconstant velocity universal joint which produces the same maximum acceleration as the maximum of the sum of the accelerations of all but the last nonconstant velocity universal joint in the driveline. (Refer to chapter on *Universal Joints*)

Inertia Coast Equivalent Angle—The joint angle of the last nonconstant velocity universal joint in a two-joint system. In a three-or multiple joint system, it is the joint angle of a single nonconstant velocity universal joint which produces the same maximum acceleration as the maximum of the sum of the accelerations of all but the first nonconstant velocity universal joint in the

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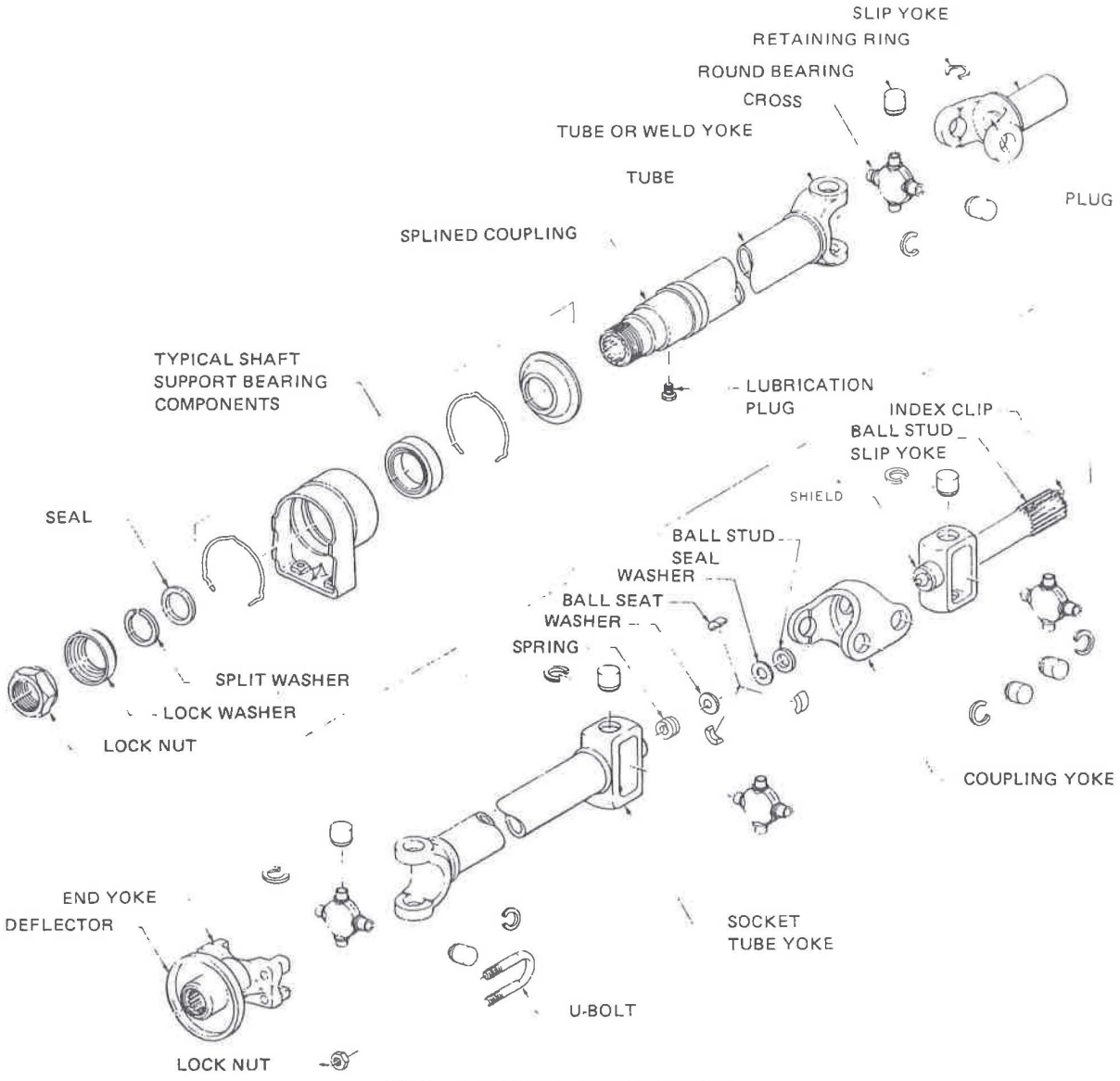


Fig. 3 — Two piece driveshaft or driveline

driveline. (Refer to chapter on *Universal Joints*)

Secondary Couple—A bending moment on the driving and driven members of a universal joint produced by an angular change in the direction of torque. It is a function of torque and joint angle. In a nonconstant velocity universal joint, the bending moment on the driving member oscillates from zero to maximum, while simultaneously on the driven member the moment varies from maximum to zero, twice per revolution. In a constant velocity universal joint, the bending moment on both driving and driven members is constant for all positions of joint rotation. (Refer to chapter on *Universal Joints*)

Bearing Factor—A size characteristic for comparing the various size Cardan and double Cardan type universal joints relative to bearing capacity. It is the product of the projected needle roller bearing area on the cross trunnion times the torque radius. (Refer to chapter on *Universal Joints*)

Universal Joints

The definitions of the following universal joints with their respective components and pertinent terms are listed in the same order as these joints are described in the chapter on *Universal Joints*. To aid in understanding these definitions, reference should be made to the illustrations shown in the applicable universal joint section, except as noted for the Cardan joint definitions.

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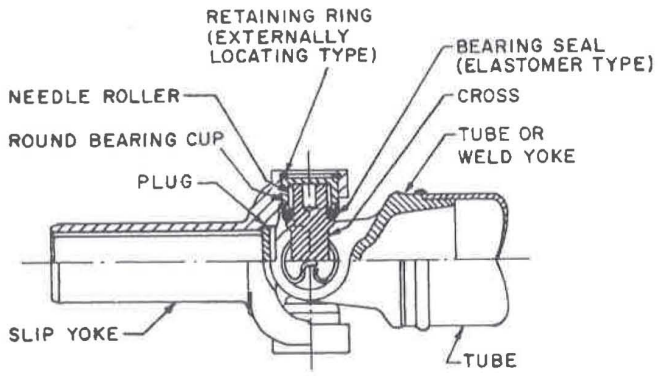


Fig. 4 — Cardan universal joint

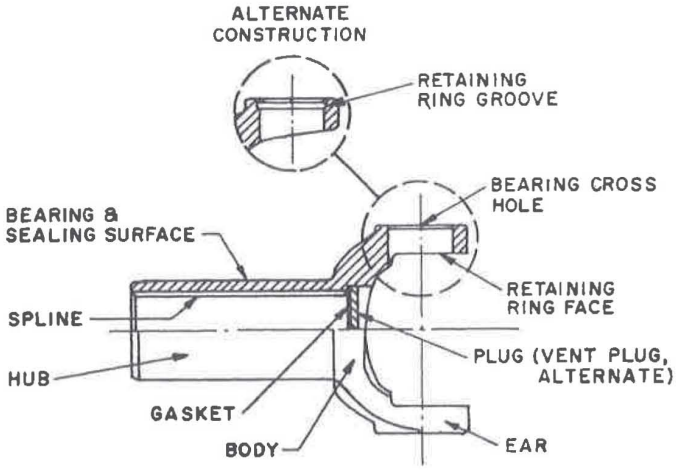


Fig. 5 — Slip yoke

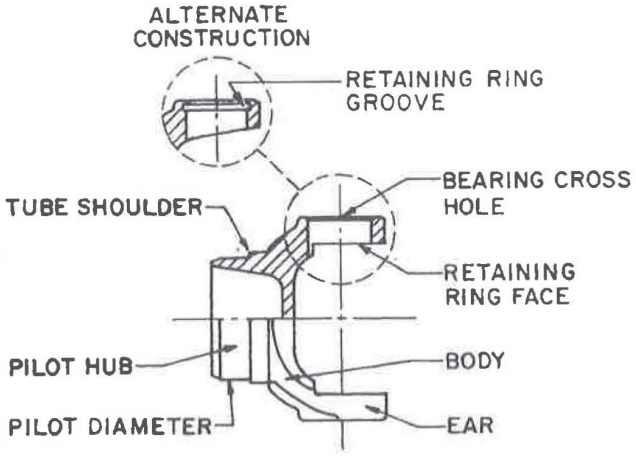


Fig. 6 — Tube or weld yoke

Nonconstant Velocity Types

Cardan or Hooke Universal Joint

Cardan or Hooke Universal Joint—A nonconstant velocity universal joint which consists of two yokes drivably connected by a cross through four bearings. (Figs. 2 and 4)

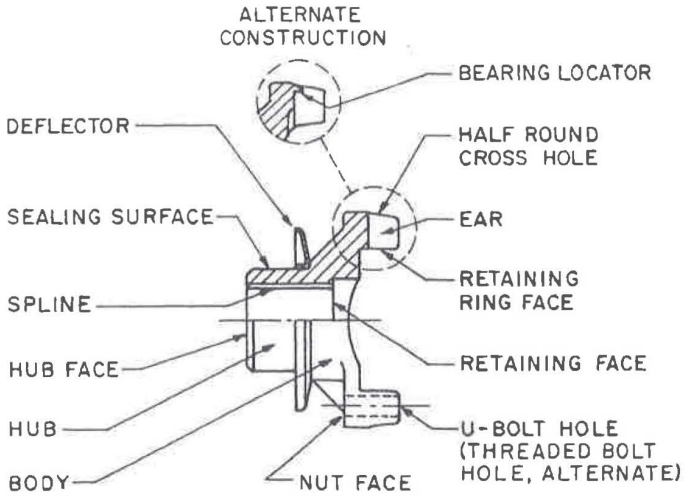


Fig. 7 — End yoke (round bearing type)

Yoke—The basic torque and/or motion input and output member with drivable means of attachment. (Fig. 2)

Slip Yoke—A yoke which accommodates axial movement (Figs. 2 and 5)

Tube or Weld Yoke—A yoke with a piloting hub for attachment to a tube or other shaft member. (Figs. 2 and 6)

End Yoke—A yoke which attaches a driveshaft to another drivetrain component. (Figs. 2, 7, and 8)

Flange Yoke—A yoke which attaches a driveshaft to a companion flange. (Figs. 2 and 9)

Companion Flange—A flange member that fixedly attaches a driveshaft to another drivetrain component. (Figs. 2 and 10)

Ear—One of two projecting parts of a yoke symmetrically located with respect to the rotational axis. (Fig. 5)

Hub—The central part of a member used for attachment to another member. (Fig. 5)

Bearing Cross Hole—A through hole in each ear of a yoke used to locate a round bearing. (Fig. 5)

Half Round Cross Hole—A semicircular hole located on the end of each ear of some end yoke designs used to locate a round bearing. (Figs. 2 and 7)

Bearing Locator—A projection in a half round cross hole, of some end yoke designs, used to locate a round bearing with respect to the yoke centerline. (Fig. 7)

Retaining Ring Groove—A groove used to locate a retaining ring. (Figs. 5 and 12)

Retaining Ring—A removable member used as a shoulder to retain and position a round bearing in a hole. (Figs. 2, 4, and 11)

U-Bolt—A clamping type bolt with two threaded parallel legs used to retain a round bearing in some end yoke designs. (Figs. 2 and 11)

Round Bearing—Consists of a round bearing cup with needle rollers generally held in place by a needle roller retainer or a bearing seal. (Figs. 2, 11, and 12)

Round Bearing Cup—A cup-shaped member used as the bearing bore of a round bearing and for positioning the thrust end of a cross trunnion. (Figs. 4 and 12)

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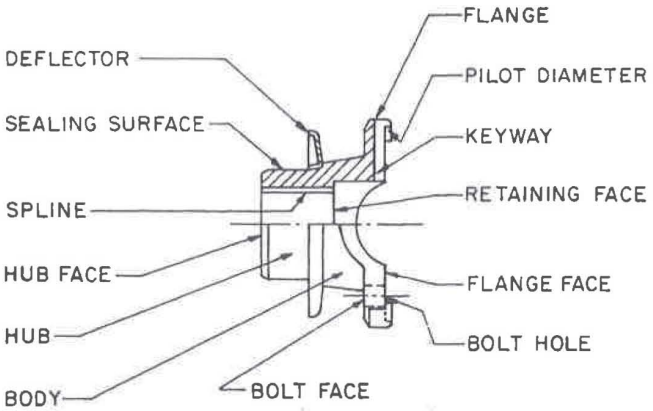


Fig. 8 — End yoke (wing bearing type)

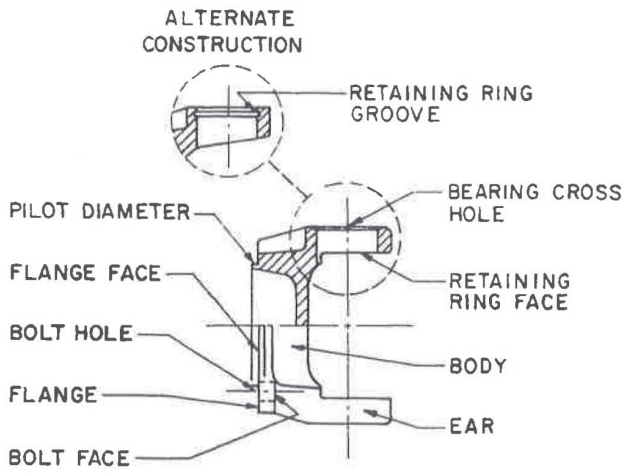


Fig. 9 — Flange yoke

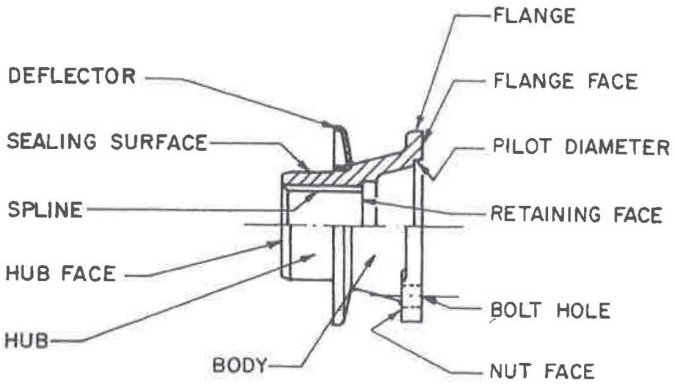


Fig. 10 — Companion flange

Needle Roller—One of the rolling elements of a bearing. (Figs. 4 and 12)

Needle Roller Retainer—A member used to retain needle rollers in a bearing. (Figs. 12 and 14)

Bearing Seal—A generally flexible member which prevents the escape of lubricant from or entry of foreign matter into a bearing. (Figs. 4 and 13)

Dust Shield, Deflector—A protective member whose function is to exclude foreign objects. (Figs. 7 and 13)

Seal Retainer—A member used to hold a bearing seal in position on the bearing. (Fig. 13)

Wing Bearing—Consists of a wing bearing cup with needle rollers generally held in place by a needle roller retainer or a bearing seal. (Figs. 2, 11, and 14)

Wing Bearing Cup—A member with a key and projecting wings used as the bearing bore of a wing bearing and for positioning the thrust end of a cross trunnion. The low wing bearing type cup has thin flanged wings for attachment. The high wing bearing type cup has thick flanged wings for attachment. The delta wing bearing type cup has delta-shaped flanged wings for attachment. (Figs. 11 and 14)

Round Bearing (Retainer Plate Type)—Consists of a round bearing with a generally integral retainer plate for retaining and positioning the bearing in a yoke. (Fig. 11)

Bearing Retainer (Star Type)—A member with locking prongs on an arched rim used for retaining and positioning a round bearing in a yoke. (Fig. 11)

Strap—A narrow plate type member used to retain a round bearing in some end yoke designs. (Figs. 2 and 11)

Retaining Ring Face—A surface used for positioning a round bearing with a retaining ring. (Figs. 5 and 12)

Thrust Face—The closed end of a bearing used as a thrust surface. (Figs. 12 and 14)

Cross—The intermediate drive member which has four equally spaced trunnions in the same plane. (Figs. 2, 4, and 15)

Trunnions—Anyone of the four projecting journals of a cross. (Fig. 15)

Trunnion Diameter—The diameter of a projecting journal of a cross. (Fig. 15)

Thrust End—The end of the cross trunnions used as a thrust surface. (Fig. 15)

Ball and Trunnion Universal Joint

Ball and Trunnion Universal Joint—A nonconstant velocity universal joint, radially self-supported and permitting axial movement, which consists of a housing drivably connected to a ball head shaft through a pair of diametrically opposed trunnion mounted balls.

Housing—A member with two diametrically opposed, partly cylindrical axial ball bores on the cylindrical inner clearance surface and with drivable means of attachment.

Ball—A partly spherical-shaped member which pivots and transmits torque from the housing to the ball head shaft through needle rollers and the pin and permits axial movement.

Ball Head Shaft—A member which provides for the transversely located pin and has drivable means of attachment.

Pin—A member pressed through the ball head shaft which provides trunnions for the needle rollers, balls, centering buttons, and associated components.

Centering Button—A thrust loaded member which locates the ball head shaft assembly during its various movements and radially retains the needle rollers.

Spring Washer—A member which maintains contact of the centering buttons with the housing bore radial thrust surfaces.

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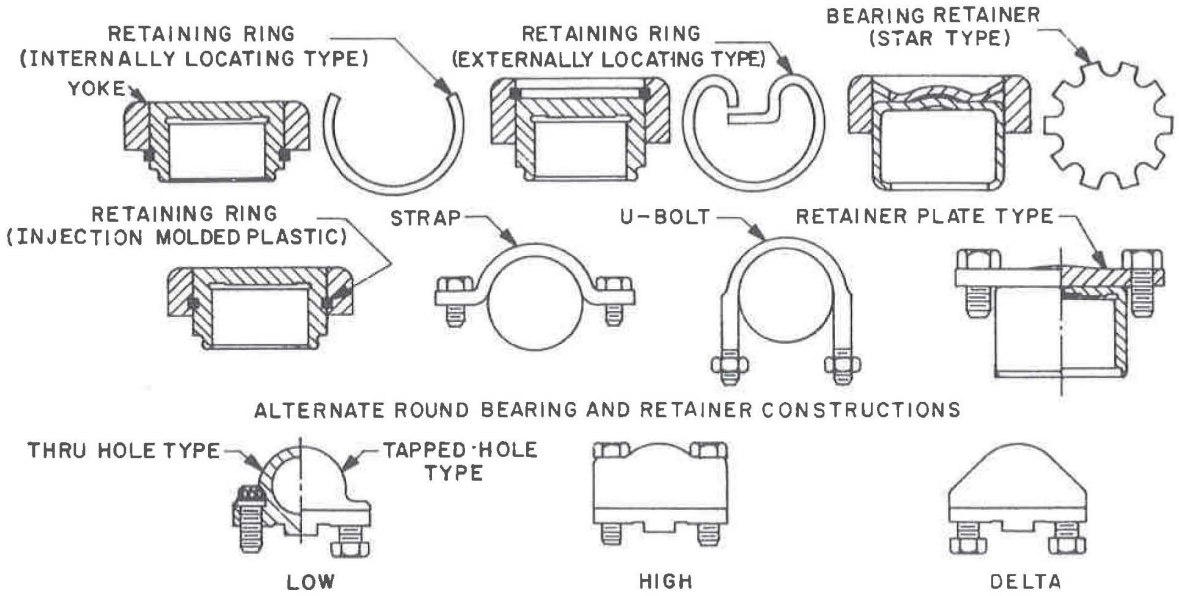


Fig. 11 — Bearing and retainer types

Spacer Washer—A member which radially positions the needle rollers inside the ball.

Boot Seal—A flexible member which prevents the escape of lubricant from or entry of foreign matter into the universal joint.

Clamp—A thin metallic or nonmetallic adjustable band used to hold the boot seal in position on the universal joint.

Housing End Cover—A member which protects the universal joint from contamination and loss of lubricant during shipping and handling.

Roller and Trunnion Universal Joint

Roller and Trunnion Universal Joint—A nonconstant velocity universal joint, radially self-supported and permitting axial movement, which consists of a housing drivably connected to a

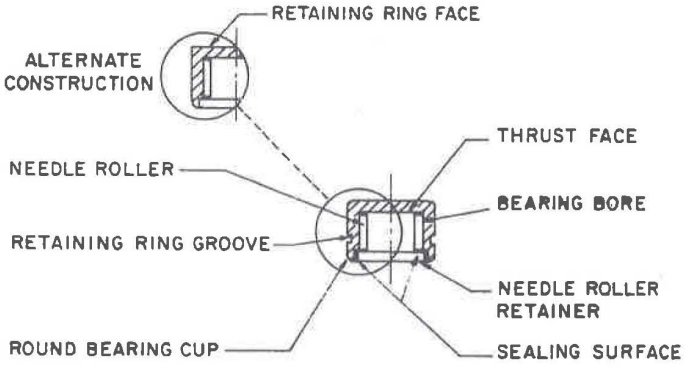


Fig. 12 — Round bearing

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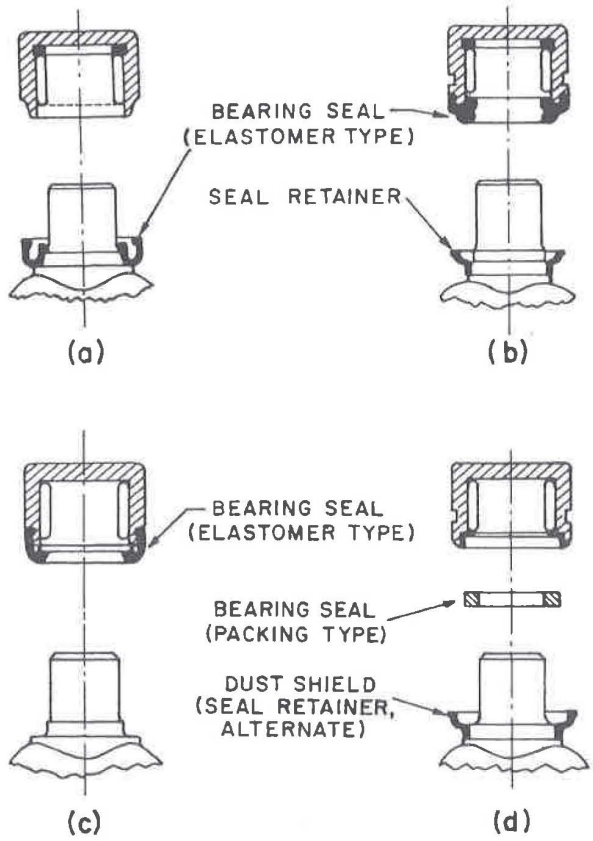


Fig. 13 — Bearing seal types

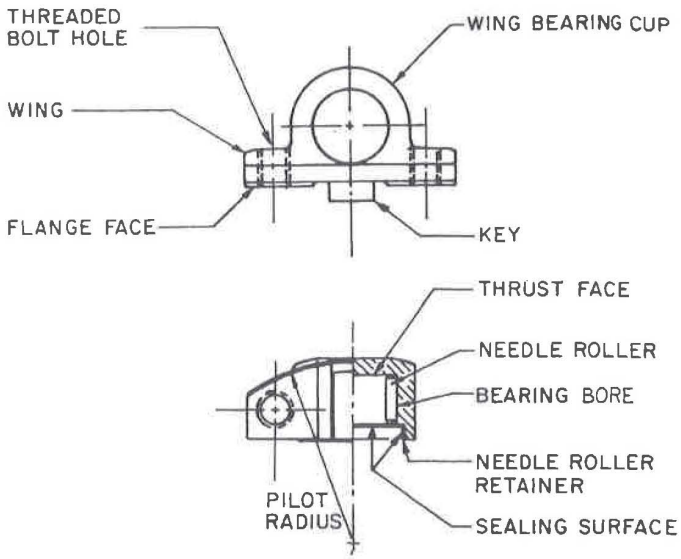


Fig. 14 — Wing bearing (low wing bearing type)

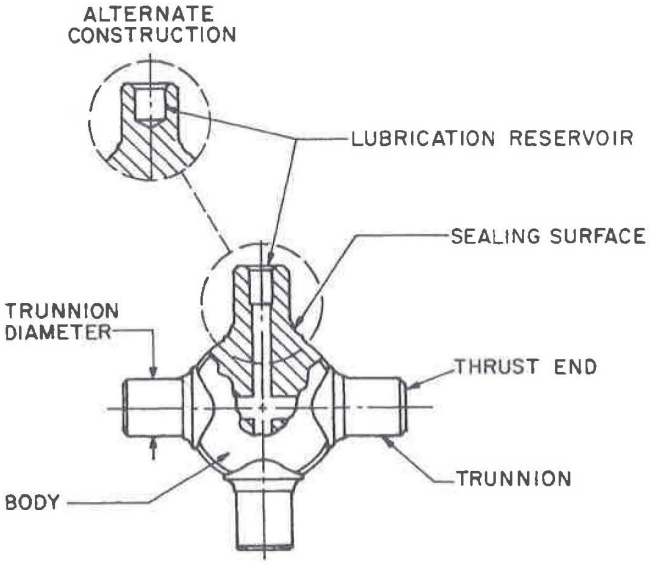


Fig. 15 — Cross

shaft through a pair of diametrically opposed trunnion mounted rollers.

Housing—A member with two diametrically opposed axial roller slots on the cylindrical inner clearance surface and with drivable means of attachment. In the tulip type housing, these roller slots extend through its periphery.

Roller—A cylindrical shaped member which pivots and transmits torque from the housing to the shaft through the needle rollers and pin, and permits axial movement.

Housing Cover—A sleeve-like member which encloses the tulip type housing and provides for attachment of the boot seal and bearing surfaces for radial support of the universal joint.

Pin—A member pressed transversely through the shaft which provides trunnions for the needle rollers and rollers.

Seal Adapter—A member used to connect the boot seal to the housing.

Seal Retainer—A member used to hold the boot seal in position on the seal adapter or housing.

Sliding Block and Trunnion Universal Joint

Sliding Block and Trunnion Universal Joint—A nonconstant velocity universal joint, radially self-supported and permitting axial movement, which consists of a housing drivably connected to a shaft through a pair of diametrically opposed trunnion mounted bearing blocks.

Housing—A member with two diametrically opposed axial bearing block slots on the cylindrical inner clearance surface and with drivable means of attachment.

Bearing Block—A square or rectangular shaped member which pivots and transmits torque from the housing to the shaft through the pin, and permits axial movement.

Pin—A member pressed transversely through the shaft which provides trunnions for the bearing blocks.

Rubber Coupling

Rubber Coupling—A mechanical device with an intermediate resilient elastomeric drive element or elements which provides for angular misalignment, absorbs impact and shock forces, dampens drivetrain disturbances, and functions generally with nonconstant velocity universal joint characteristics.

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