

Fig. 4 — Schematic view of one piece driveline with double Cardan joints

In the section on the *Cardan or Hooke Universal Joint*, it is shown that with respect to the bearing supports, the secondary couple forces have static and dynamic components. The static force is constant in size and direction and is normal to the plane of the joint angle, while the dynamic force of equal magnitude rotates at twice driveshaft speed. However, with respect to the rotating shaft of which the centering device is a part, the couple forces stay in the plane of the coupling yoke and vary from a maximum value in one direction to a maximum value in the opposite direction and back again, once per revolution. These maximum values always occur normal to the plane of the joint angle.

For simplicity, it is assumed that the joint angles between the coupling yoke and each end supporting yoke in Fig. 3 are exactly equal.

Therefore, the force F in lb on the centering device is:

$$F = \frac{2T \sin \theta}{L}$$

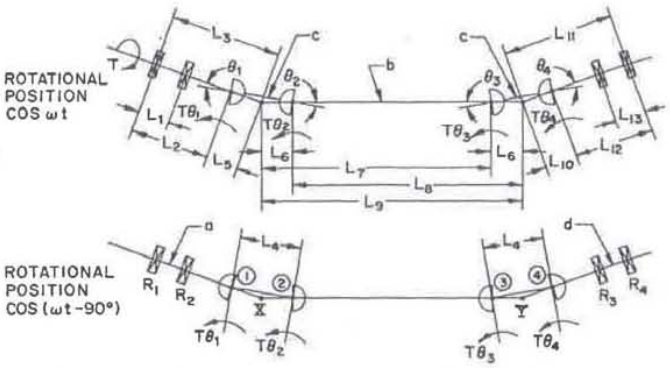


Fig. 5 — Secondary couples in one piece driveline with double Cardan joints

where:

T = Torque, lb-in

θ = Joint angle between coupling yoke and one of the end supporting yokes, deg

L = Coupling yoke length, in

When two double Cardan joints are used in a passenger car, one piece driveline installation as illustrated schematically in Fig. 4, the couples forces on the supports at the transmission and differential ends become quite complex. Fig. 5 shows this same driveline in two rotational positions 90 deg apart with the secondary couples resulting from the applied torque, T. For this instal-

NOMENCLATURE

a = Transmission output shaft
b = Driveshaft
c = Coupling yoke of double Cardan joint
d = Pinion shaft
R_1 = Transmission extension bearing
R_2 = Transmission extension bearing
R_3 = Pinion shaft bearing
R_4 = Pinion shaft bearing
L_1, L_2, etc. = Typical length dimensions
① = Input joint of front double Cardan joint
② = Output joint of front double Cardan joint
③ = Input joint of rear double Cardan joint
④ = Output joint of rear double Cardan joint
X = Front double Cardan joint centering ball
Y = Rear double Cardan joint centering ball
θ_1 = Joint angle of joint ①
θ_2 = Joint angle of joint ②
θ_3 = Joint angle of joint ③
θ_4 = Joint angle of joint ④

T = Driveline torque assumed clockwise when viewed from front of car
$T\theta_1$ = Secondary couple produced by joint ① (For small angles, the sine, tangent, and radian are nearly equal. Therefore, for simplicity, the radian is used.)
$T\theta_2$ = Secondary couple produced by joint ②
$T\theta_3$ = Secondary couple produced by joint ③
$T\theta_4$ = Secondary couple produced by joint ④
F_{1a}, $F_{2b \to a}$, etc. = Typical couple forces on bearing support or centering ball. The subscript notation of F_{1a} indicates that the force results from couple $T\theta_1$ originating at joint ① and is reacted on shaft a. $F_{2b \to a}$ indicates that the force results from couple $T\theta_2$ originating at joint ②, is reacted on shaft b, and then transferred to shaft a by way of the centering ball X.
F_R = Resultant force on bearing support or centering ball
F_S = Static force on bearing support
F_D = Dynamic force on bearing support or centering ball

PROBLEM DATA

T = 16 000 lb-in WOT torque
θ_1 = 3°18′31″ = 0.05775 rad (1 rad = 57.296°)
θ_2 = 3°16′29″ = 0.05715 rad
θ_3 = 4°37′2″ = 0.08058 rad
θ_4 = 4°38′58″ = 0.08115 rad
L_1 = 2.57 in
L_2 = 5.93 in
L_3 = 7.549 in
L_4 = 3.25 in
L_5 = 1.619 in
L_6 = 1.636 in
L_7 = 55.954 in
L_8 = 55.954 in
L_9 = 57.59 in
L_{10} = 1.625 in
L_{11} = 9.105 in
L_{12} = 7.48 in
L_{13} = 3.60 in

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lation, no offset is assumed in the top or plan view. Therefore, the elevation or side views, as shown, are true views of the driveline.

The following illustrative problem for a wide-open throttle (WOT), forward start, or maximum tractive effort (wheel slip) condition, shows the bearing support forces in such a driveline.

Couple Forces on Bearing R_1

Couple T θ_{1a} (The subscript notation indicates that the couple originates at joint ① and is reacted on shaft a.)

$$F_{1a} = \frac{T\,\theta_1}{L_1}$$

$$= \frac{16\,000 \times 0.05775}{2.57}$$

$$F_{1a} = 359.533 \text{ lb}$$

Couple T $\theta_{2b \to a}$ (The subscript notation indicates that the couple originates at joint ②, is reacted on shaft b, and then transferred to shaft a by way of the centering ball X.)

$$F = \frac{T\,\theta_2}{L_9}$$

$$F_{2b \to a} = \frac{F\,(L_3 - L_1)}{L_1} = \frac{T\,\theta_2\,(L_3 - L_1)}{L_1\,L_9}$$

$$= \frac{16\,000 \times 0.05715\,(7.549 - 2.57)}{2.57 \times 57.59}$$

$$F_{2b \to a} = 30.761 \text{ lb}$$

Couple T $\theta_{3b \to a}$

$$F = \frac{T\,\theta_3}{L_9}$$

$$F_{3b \to a} = \frac{F\,(L_3 - L_1)}{L_1} = \frac{T\,\theta_3\,(L_3 - L_1)}{L_1\,L_9}$$

$$= \frac{16\,000 \times 0.08058\,(7.549 - 2.57)}{2.57 \times 57.59}$$

$$F_{3b \to a} = 43.372 \text{ lb}$$

Couple T $\theta_{1c \to a}$

$$F = \frac{T\,\theta_1}{L_4}$$

$$F_{1c \to a} = \frac{F\,(L_2 - L_1)}{L_1} = \frac{T\,\theta_1\,(L_2 - L_1)}{L_1\,L_4}$$

$$= \frac{16\,000 \times 0.05775\,(5.93 - 2.57)}{2.57 \times 3.25}$$

$$F_{1c \to a} = 371.702 \text{ lb}$$

Couple T $\theta_{2c \to a}$

$$F = \frac{T\,\theta_2}{L_4}$$

$$F_{2c \to a} = \frac{F\,(L_2 - L_1)}{L_1} = \frac{T\,\theta_2\,(L_2 - L_1)}{L_1\,L_4}$$

$$= \frac{16\,000 \times 0.05715\,(5.93 - 2.57)}{2.57 \times 3.25}$$

$$F_{2c \to a} = 367.840 \text{ lb}$$

Couple T $\theta_{1c \to b \to a}$

$$F = \frac{T\,\theta_1}{L_4}$$

$$F' = \frac{FL_8}{L_9} = \frac{T\,\theta_1\,L_8}{L_4\,L_9}$$

$$F_{1c \to b \to a} = \frac{F'\,(L_3 - L_1)}{L_1} = \frac{T\,\theta_1\,L_8\,(L_3 - L_1)}{L_1\,L_4\,L_9}$$

$$= \frac{16\,000 \times 0.05775 \times 55.954\,(7.549 - 2.57)}{2.57 \times 3.25 \times 57.59}$$

$$F_{1c \to b \to a} = 535.157 \text{ lb}$$

Couple T $\theta_{2c \to b \to a}$

$$F = \frac{T\theta_2}{L_4}$$

$$F' = \frac{FL_8}{L_9} = \frac{T\,\theta_2\,L_8}{L_4\,L_9}$$

$$F_{2c \to b \to a} = \frac{F'\,(L_3 - L_1)}{L_1} = \frac{T\,\theta_2\,L_8\,(L_3 - L_1)}{L_1\,L_4\,L_9}$$

$$= \frac{16\,000 \times 0.05715 \times 55.954\,(7.549 - 2.57)}{2.57 \times 3.25 \times 57.59}$$

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$$F_{2c \to b \to a} = 529.597 \text{ lb}$$

Couple $T\,\theta_{3c \to b \to a}$

$$F = \frac{T\,\theta_3}{L_4}$$

$$F' = \frac{F\,L_6}{L_9} = \frac{T\,\theta_3\,L_6}{L_4\,L_9}$$

$$F_{3c \to b \to a} = \frac{F'\,(L_3 - L_1)}{L_1} = \frac{T\,\theta_3\,L_6\,(L_3 - L_1)}{L_1\,L_4\,L_9}$$

$$= \frac{16\,000 \times 0.08058 \times 1.636\,(7.549 - 2.57)}{2.57 \times 3.25 \times 57.59}$$

$$F_{3c \to b \to a} = 21.833 \text{ lb}$$

Couple $T\,\theta_{4c \to b \to a}$

$$F = \frac{T\,\theta_4}{L_4}$$

$$F' = \frac{FL_6}{L_9} = \frac{T\,\theta_4\,L_6}{L_4\,L_9}$$

$$F_{4c \to b \to a} = \frac{F'\,(L_3 - L_1)}{L_1} = \frac{T\,\theta_4\,L_6\,(L_3 - L_1)}{L_1\,L_4\,L_9}$$

$$= \frac{16\,000 \times 0.08115 \times 1.636\,(7.549 - 2.57)}{2.57 \times 3.25 \times 57.59}$$

$$F_{4c \to b \to a} = 21.987 \text{ lb}$$

All of these couple forces must now be combined into an equation representing the resultant force on bearing R_1 due to the secondary couples in the driveline. The couple force directions are assumed positive (+) for those being applied left transversely and negative (−) for those being applied right transversely in the car when viewed from the rear. Therefore, the resultant force on bearing R_1 is:

$$F_R = -\frac{T\,\theta_1}{L_1}\cos \omega t + \frac{T\,\theta_2\,(L_3 - L_1)}{L_1\,L_9}\cos \omega t$$

$$+ \frac{T\,\theta_3\,(L_3 - L_1)}{L_1\,L_9}\cos \omega t + \frac{T\,\theta_1\,(L_2 - L_1)}{L_1\,L_4}\cos (\omega t - 90 \text{ deg})$$

$$+ \frac{T\,\theta_2\,(L_2 - L_1)}{L_1\,L_4}\cos (\omega t - 90 \text{ deg})$$

$$- \frac{T\,\theta_1\,L_8\,(L_3 - L_1)}{L_1\,L_4\,L_9}\cos (\omega t - 90 \text{ deg})$$

$$- \frac{T\,\theta_2\,L_8\,(L_3 - L_1)}{L_1\,L_4\,L_9}\cos (\omega t - 90 \text{ deg})$$

$$- \frac{T\,\theta_3\,L_6\,(L_3 - L_1)}{L_1\,L_4\,L_9}\cos (\omega t - 90 \text{ deg})$$

$$+ \frac{T\,\theta_4\,L_6\,(L_3 - L_1)}{L_1\,L_4\,L_9}\cos (\omega t - 90 \text{ deg})$$

This resultant force must now be expressed in terms of both static and dynamic force component equations. The static force equation is simply one half of the resultant force and is:

$$F_S = -\frac{T\,\theta_1}{2L_1} + \frac{T\,\theta_2\,(L_3 - L_1)}{2L_1\,L_9}$$

$$+ \frac{T\,\theta_3\,(L_3 - L_1)}{2L_1\,L_9} + \frac{T\,\theta_1\,(L_2 - L_1)}{2L_1\,L_4}$$

$$+ \frac{T\,\theta_2\,(L_2 - L_1)}{2L_1\,L_4} - \frac{T\,\theta_1\,L_8\,(L_3 - L_1)}{2L_1L_4\,L_9}$$

$$- \frac{T\,\theta_2\,L_8\,(L_3 - L_1)}{2L_1\,L_4\,L_9}$$

$$+ \frac{T\,\theta_3\,L_6\,(L_3 - L_1)}{2L_1\,L_4\,L_9}$$

$$+ \frac{T\,\theta_4\,L_6\,(L_3 - L_1)}{2L_1\,L_4\,L_9}$$

$$= -\frac{359.533}{2} + \frac{30.761}{2} + \frac{43.372}{2}$$

$$+ \frac{371.702}{2} + \frac{367.840}{2} - \frac{535.157}{2}$$

$$- \frac{529.597}{2} + \frac{21.833}{2} + \frac{21.987}{2}$$

$$= -179.7665 + 15.3805 + 21.686$$

$$+ 185.851 + 183.920 - 267.5786$$

$$- 264.7985 + 10.9165 + 10.9935$$

$$F_S = -283.396 \text{ lb}$$

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The dynamic force equation is, however, affected by the location of the couples with regard to the two rotational position views in Fig. 5. If the view of rotational position cos ωt is assumed as the reference view, then the signs of all the couple forces originating in the view of rotational position cos (ωt − 90 deg) must be changed to their opposite sign. Also, as in the static force equation, all couple forces must be divided by two. Therefore, the dynamic force is:

$$F_D = -\frac{T\,\theta_1}{2L_1} + \frac{T\,\theta_2\,(L_3 - L_1)}{2L_1\,L_9} + \frac{T\,\theta_3\,(L_3 - L_1)}{2L_1\,L_9}$$

$$- \frac{T\,\theta_1\,(L_2 - L_1)}{2L_1\,L_4} - \frac{T\,\theta_2\,(L_2 - L_1)}{2L_1\,L_4}$$

$$+ \frac{T\,\theta_1 L_8\,(L_3 - L_1)}{2L_1\,L_4\,L_9} + \frac{T\,\theta_2 L_8\,(L_3 - L_1)}{2L_1\,L_4\,L_9}$$

$$- \frac{T\,\theta_3 L_6\,(L_3 - L_1)}{2L_1\,L_4\,L_9} - \frac{T\,\theta_4 L_6\,(L_3 - L_1)}{2L_1\,L_4\,L_9}$$

$$= -\frac{359.533}{2} + \frac{30.761}{2} + \frac{43.372}{2}$$

$$+ \frac{371.702}{2} + \frac{367.840}{2} - \frac{535.157}{2}$$

$$- \frac{529.597}{2} + \frac{21.833}{2} + \frac{21.987}{2}$$

$$= -179.7665 + 15.3805 + 21.686$$

$$- 185.851 - 183.920 + 267.5785$$

$$+ 264.7985 - 10.9165 - 10.9935$$

$$F_D = -2.004 \text{ lb}$$

The individual couple forces, resultant force, static force, and dynamic force on bearings R_2, R_3, and R_4 are computed in a manner similar to those on bearing R_1. The following tabulation summarizes the static and dynamic forces on the bearing supports.

Bearing	F_S	F_D
R_1	−283.396	−2.004
R_2	+244.9215	+2.2145
R_3	−262.7205	−1.1685
R_4	+301.197	+0.9560

As in any force system, a condition of equilibrium must be satisfied. A summation of the above forces must equal zero. Since the equations used are approximations, the summation, therefore, approximately equals zero as follows:

$$\sum F_S = 0$$

$$- 283.396 - 244.9215 - 262.7205 + 301.197 = 0$$

$$- 546.1165 + 546.1185 = 0$$

$$+ 0.002 \approx 0$$

$$\sum F_D = 0$$

$$- 2.004 + 2.2145 - 1.1685 + 0.9560 = 0$$

$$- 3.1725 + 3.1705 = 0$$

$$- 0.002 \approx 0$$

In the above summary of the forces on the bearing supports, it is shown that the dynamic forces F_D are nearly zero as they should be because the double Cardan joints in the driveline are near constant velocity joints. These forces would be eliminated entirely if true constant velocity joints were used. The couple forces on the supports would then be reduced to the static forces F_S only.

Further computations of this illustrative problem are made to determine the forces on the centering balls X and Y.

Couple Forces on Centering Ball X

Couple $T\,\theta_{2b}$

$$F_{2b} = \frac{T\,\theta_2}{L_9}$$

$$= \frac{16\,000 \times 0.05715}{57.59}$$

$$F_{2b} = 15.878 \text{ lb}$$

Couple $T\,\theta_{3b}$

$$F_{3b} = \frac{T\,\theta_3}{L_9}$$

$$= \frac{16\,000 \times 0.08058}{57.59}$$

$$F_{3b} = 22.387 \text{ lb}$$

Couple $T\,\theta_{1c \to b}$

$$F = \frac{T\,\theta_1}{L_4}$$

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$$F_{1c \to b} = \frac{FL_8}{L_9} = \frac{T\,\theta_1\,L_8}{L_4\,L_9}$$

$$= \frac{16\,000 \times 0.05775 \times 55.954}{3.25 \times 57.59}$$

$$F_{1c \to b} = 276.231 \text{ lb}$$

Couple $T\,\theta_{2c \to b}$

$$F = \frac{T\,\theta_2}{L_4}$$

$$F_{2c \to b} = \frac{FL_8}{L_9} = \frac{T\,\theta_2\,L_8}{L_4\,L_9}$$

$$= \frac{16\,000 \times 0.05715 \times 55.954}{3.25 \times 57.59}$$

$$F_{2c \to b} = 273.361 \text{ lb}$$

Couple $T\,\theta_{3c \to b}$

$$F = \frac{T\,\theta_3}{L_4}$$

$$F_{3c \to b} = \frac{FL_6}{L_9} = \frac{T\,\theta_3\,L_6}{L_4\,L_9}$$

$$= \frac{16\,000 \times 0.08058 \times 1.636}{3.25 \times 57.59}$$

$$F_{3c \to b} = 11.269 \text{ lb}$$

Couple $T\,\theta_{4c \to b}$

$$F = \frac{T\,\theta_4}{L_4}$$

$$F_{4c \to b} = \frac{FL_6}{L_9} = \frac{T\,\theta_4\,L_6}{L_4\,L_9}$$

$$= \frac{16\,000 \times 0.08115 \times 1.636}{3.25 \times 57.59}$$

$$F_{4c \to b} = 11.349 \text{ lb}$$

All of these couple forces must now be combined into an equation representing the resultant force on the centering ball. As in the case of the bearing support forces, the assumed sign convention for the couple force directions must be followed. Therefore, the resultant force on the centering ball X is:

$$F_R = -\frac{T\,\theta_2}{L_9}\cos \omega t - \frac{T\,\theta_3}{L_9}\cos \omega t + \frac{T\,\theta_1\,L_8}{L_4\,L_9}\cos(\omega t - 90 \text{ deg}) + \frac{T\,\theta_2\,L_8}{L_4\,L_9}\cos(\omega t - 90 \text{ deg}) - \frac{T\,\theta_3\,L_6}{L_4\,L_9}\cos(\omega t - 90 \text{ deg}) - \frac{T\,\theta_4\,L_6}{L_4\,L_9}\cos(\omega t - 90 \text{ deg})$$

Since the couple forces on the centering ball in the previous equation are with respect to the rotating shaft, they stay in the planes of their respective yokes and vary from a maximum value in one direction to a maximum value in the opposite direction, and back again, once per revolution. These maximum values always occur normal to the plane of the joint angle. Therefore, when individual couple forces reach their maximum values in rotational position cos ωt, they are zero in rotational position cos (ωt − 90 deg). On the other hand, when individual couple forces are at maximum values in rotational position cos (ωt − 90 deg), they are zero in rotational position cos ωt. Hence, individual couple forces are present which are 90 deg of rotation apart. Thus, the previous general equation can be restated as two dynamic force equations where each equation yields a peak force on the centering mechanism, which are oppositely directed. The maximum dynamic forces on the centering ball X are:

Rotational Position cos ωt

$$F_D = -\frac{T\,\theta_2}{L_9} - \frac{T\,\theta_3}{L_9}$$

$$= -15.878 - 22.387$$

$$F_D = -38.265 \text{ lb}$$

Rotational Position cos (ωt − 90 deg)

$$F'_D = \frac{T\,\theta_1\,L_8}{L_4\,L_9} + \frac{T\,\theta_2\,L_8}{L_4\,L_9} - \frac{T\,\theta_3\,L_6}{L_4\,L_9} - \frac{T\,\theta_4\,L_6}{L_4\,L_9}$$

$$= 276.231 + 273.361 - 11.269 - 11.349$$

$$F'_D = 526.974 \text{ lb}$$

In order to more readily understand these dynamic forces, the vector diagram in Fig. 6 (a) shows them acting on the centering ball X, when viewed from the rear of the car. The two sweep lines or vectors describe two circles when rotating, and are out of phase by 90 deg and, therefore, when one vector reaches its maximum

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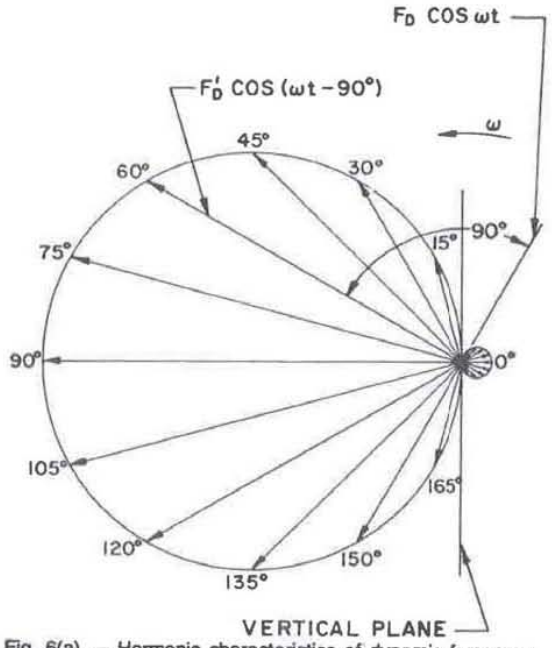


Fig. 6(a) — Harmonic characteristics of dynamic forces on centering ball X

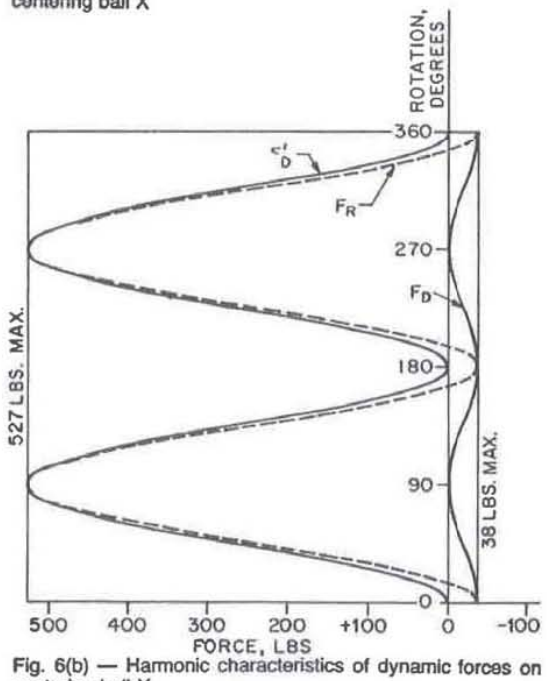


Fig. 6(b) — Harmonic characteristics of dynamic forces on centering ball X

value, the other is zero. These rotating vectors trace force curves having harmonic wave forms when projected, as shown in Fig. 6(b). The two individual dynamic force curves can then be combined, if desired, into a curve representing the resultant dynamic force acting on centering ball X.

The individual couple forces, resultant force, and dynamic forces on centering ball Y are computed in a manner similar to those on centering ball X. Only the equations of the forces on centering ball Y will be listed.

Couple Forces on Centering Ball Y

Couple $T\theta_{2b}$

$$F_{2b} = \frac{T\theta_2}{L_9}$$

Couple $T\theta_{3b}$

$$F_{3b} = \frac{T\theta_3}{L_9}$$

Couple $T\theta_{1c \to b}$

$$F_{1c \to b} = \frac{T\theta_1 L_6}{L_4 L_9}$$

Couple $T\theta_{2c \to b}$

$$F_{2c \to b} = \frac{T\theta_2 L_6}{L_4 L_9}$$

Couple $T\theta_{3c \to b}$

$$F_{3c \to b} = \frac{T\theta_3 L_7}{L_4 L_9}$$

Couple $T\theta_{4c \to b}$

$$F_{4c \to b} = \frac{T\theta_4 L_7}{L_4 L_9}$$

The resultant force on the centering ball Y is:

$$F_R = \frac{T\theta_2}{L_9}\cos\omega t + \frac{T\theta_3}{L_9}\cos\omega t$$

$$+ \frac{T\theta_1 L_6}{L_4 L_9}\cos(\omega t - 90\text{ deg}) + \frac{T\theta_2 L_6}{L_4 L_9}\cos(\omega t - 90\text{ deg})$$

$$- \frac{T\theta_3 L_7}{L_4 L_9}\cos(\omega t - 90\text{ deg}) - \frac{T\theta_4 L_7}{L_4 L_9}\cos(\omega t - 90\text{ deg})$$

The maximum dynamic forces on the centering ball Y are:

Rotational Position cos ωt

$$F_D = \frac{T\theta_2}{L_9} + \frac{T\theta_3}{L_9}$$

Rotational Position cos (ωt − 90 deg)

$$F'_D = \frac{T\theta_1 L_6}{L_4 L_9} + \frac{T\theta_2 L_6}{L_4 L_9}$$

$$- \frac{T\theta_3 L_7}{L_4 L_9} - \frac{T\theta_4 L_7}{L_4 L_9}$$

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The following tabulation summarizes the dynamic forces on centering balls X and Y.

Centering Ball	Maximum F_D	Maximum F'_D
X	−38.265	+526.974
Y	+38.265	−757.521

These results include the effects of both joints as well as the connecting shaft. The dynamic forces on these balls can also be computed with the previously described equation, $F = 2T \sin\theta/L$, which was developed for the joint only and results in approximate values.

Effect of Centering Device Location

The location of the centering device with respect to the two Cardan joint centers of the double Cardan joint determines the operating characteristics of the joint. This location provides for equal angles of the two halves of the joint and theoretically correct constant velocity at one joint angle condition (angular range in the case of the generated ball design) only. It is desirable to locate the centering device so as to produce a minimum inequality of these individual joint angles over the full angular operating range of the joint.

Curve A of Fig. 7 represents a specific double Cardan joint having the ball center of the centering mechanism so located to produce constant velocity at a 13°17′44″ joint angle only in addition to the zero angle position. The curve shows the error or difference between the individual angles up to 20 deg joint angle. At an 8 deg joint angle for example, the error or difference is 0°2′4″ and this makes the double Cardan joint equivalent to a Cardan joint operating at an angle of 0°31′ approximately. At a 20 deg joint angle, the difference is 0°10′20″, which is equivalent to a Cardan joint operating at an angle of 1°51′ approximately. These joint angle errors or differences are considered reasonable for a double Cardan joint which must operate at continuous angles up to approximately 13 deg, and with instantaneous angles up to or slightly above 20 deg in a driveline installation.

Curve B of Fig. 7 represents another double Cardan joint having the ball center of the centering mechanism located exactly midway between the two Cardan joint centers. This produces constant velocity at zero degree joint angle only. At all other angles, as the joint angle increases, the error or difference between the individual angles also increases. At an 8 deg joint angle for example, the difference is 0°1′10″ and this makes the double Cardan joint equivalent to a Cardan joint operating at an angle of 0°24′ approximately. At a 13°17′44″ joint angle, which is the CV condition for the joint represented by curve A, the difference is 0°5′24″, which is equivalent to a Cardan joint operating at an angle of 1°6′ approximately. At a 20 deg joint angle, the difference is 0°18′24″, which is equivalent to a Cardan joint operating at an angle of 2°29′ approximately.

Therefore, it is advisable to locate the centering device ball center so it will produce a minimum inequality of individual joint angles over the full angular operating range of the joint as illustrated by curve A. Further, it is desirable to select component tolerances which will control the variation of the true CV location within acceptable limits from the nominal condition illustrated by curve A.

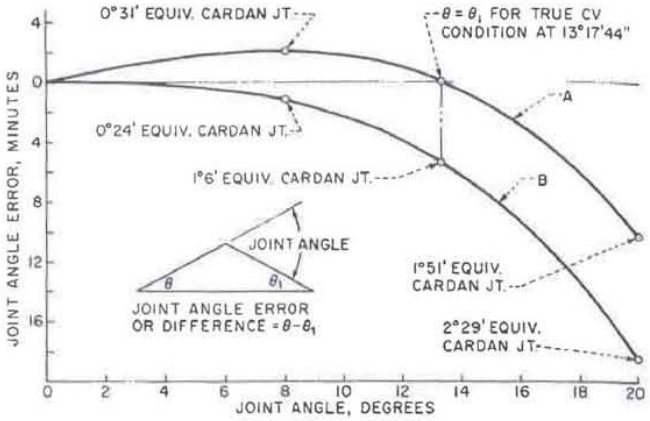


Fig. 7 — Difference in joint angles

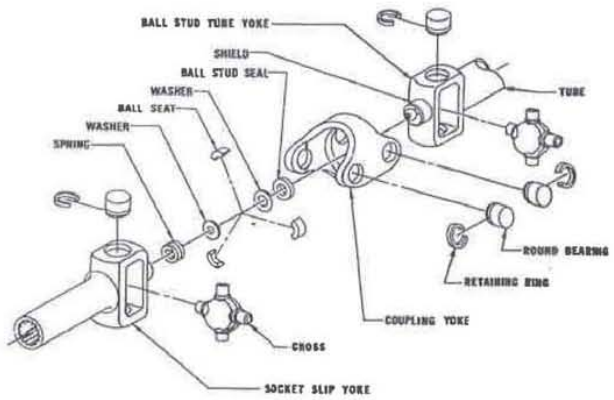


Fig. 8 — Double Cardan universal joint

Centering Device Function

The function of the centering device is to provide internal support so that the double Cardan joint is self-supporting and self-aligning. It must carry the physical weight and unbalance forces of the driveshaft. Also, the centering device must be capable of dividing nearly equally the total joint angle between the two Cardan joints. Further, it must be capable of oscillating through twice the operating joint angle at a frequency of once per revolution. In addition, the centering device must be capable of absorbing the secondary couple forces arising from the two angulated halves of the joint when torque is transmitted. Since these forces are equal and symmetrically opposed to one another, they are effectively cancelled because of the supporting action of the centering device. This does result, however, in high oscillating forces acting upon the centering mechanism normal to the plane of the joint angle as illustrated in Fig. 6. Finally, the centering device must be able to allow in and out movement and withstand axial forces during joint angle changes.

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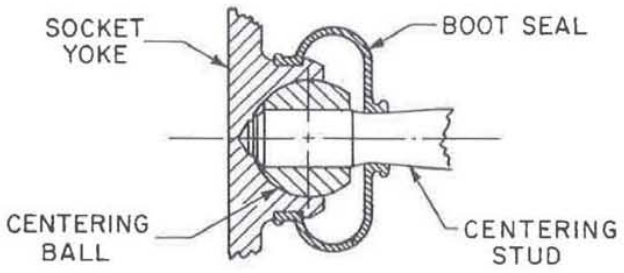


Fig. 9 — Centering ball and stud for axle driveshaft applications

Design

The various components of a typical double Cardan joint, shown in an exploded view in Fig. 8, must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter, *Basic Driveline Requirements*. Initially, the centering device as well as the bearing and cross configurations are determined. Then, the yoke constructions can be designed to meet the various joint application parameters.

For purposes of standardization and economic reasons, it is desirable to incorporate existing bearing and cross components which are being used in a comparable size Cardan joint. When a new bearing and cross configuration is required, the design procedure for these components is described in the section, *Cardan or Hooke Universal Joint*.

Centering Device

The centering device is generally a ball and socket type of mechanism which is basically a self-aligning bearing. The design of the various load carrying components with regard to shape, size, and strength are determined to a large degree by the magnitudes of the physical weight, unbalance, and secondary couple forces acting on the centering mechanism, and the desire to have a self-supporting joint. Venting provisions must be provided to avoid internal pressure build-up, which can cause seal damage and loss of lubricant. This pressure build-up can take place during assembly of the joint as well as during joint angle changes when relative endwise motion of centering device components exists. The two basic types of ball and socket construction which generally have been used are:

1. Centering ball and stud
2. Ball stud and seat

The centering ball and stud shown in Fig. 9 consists of a ball partially enclosed in a socket so that only pivoting or self-aligning motion of the ball can take place. The stud protrudes into a cylindrical hole passing through the ball. This provides support for the double Cardan joint and allows the necessary endwise motion of the stud relative to the ball during joint angle changes. This simple form of centering device has been widely used in double Cardan joints for front wheel drive applications, because it permits the higher joint angles required at the steerable wheel position. With this centering mechanism, the joint will transmit constant velocity at zero and at the design joint angle only. At all other angles, it is nearly a constant velocity joint.

For high speed driveshaft applications this centering device must be more complex, as shown in Fig. 10, in order to withstand the higher oscillatory speeds and additional vibration problems. A full complement of needle rollers, when incorporated between the ball and stud, improves the relative motion characteristics between the socket bearing, ball, and stud. A compression spring, when incorporated to preload the ball against the socket bearing cavity, compensates for clearances due to manufacturing and wear, and thereby further improves operation and life of the joint. A minimum spring load should be used because if it is too high, excessive socket wear and heat will result from the added friction.

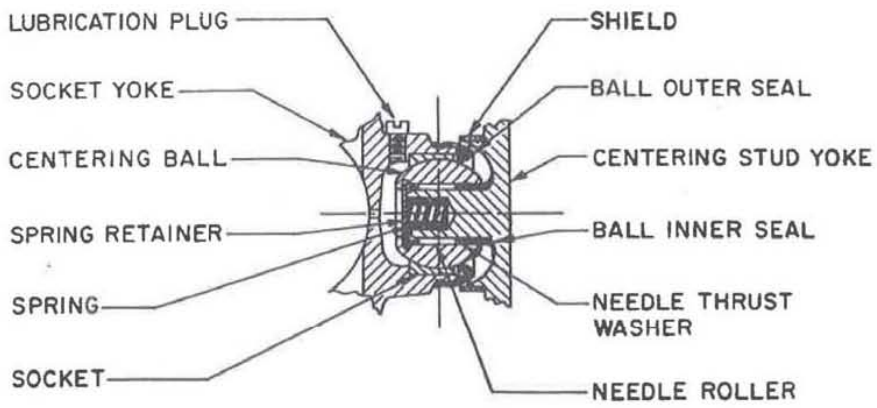


Fig. 10 — Centering ball and stud for driveshaft applications

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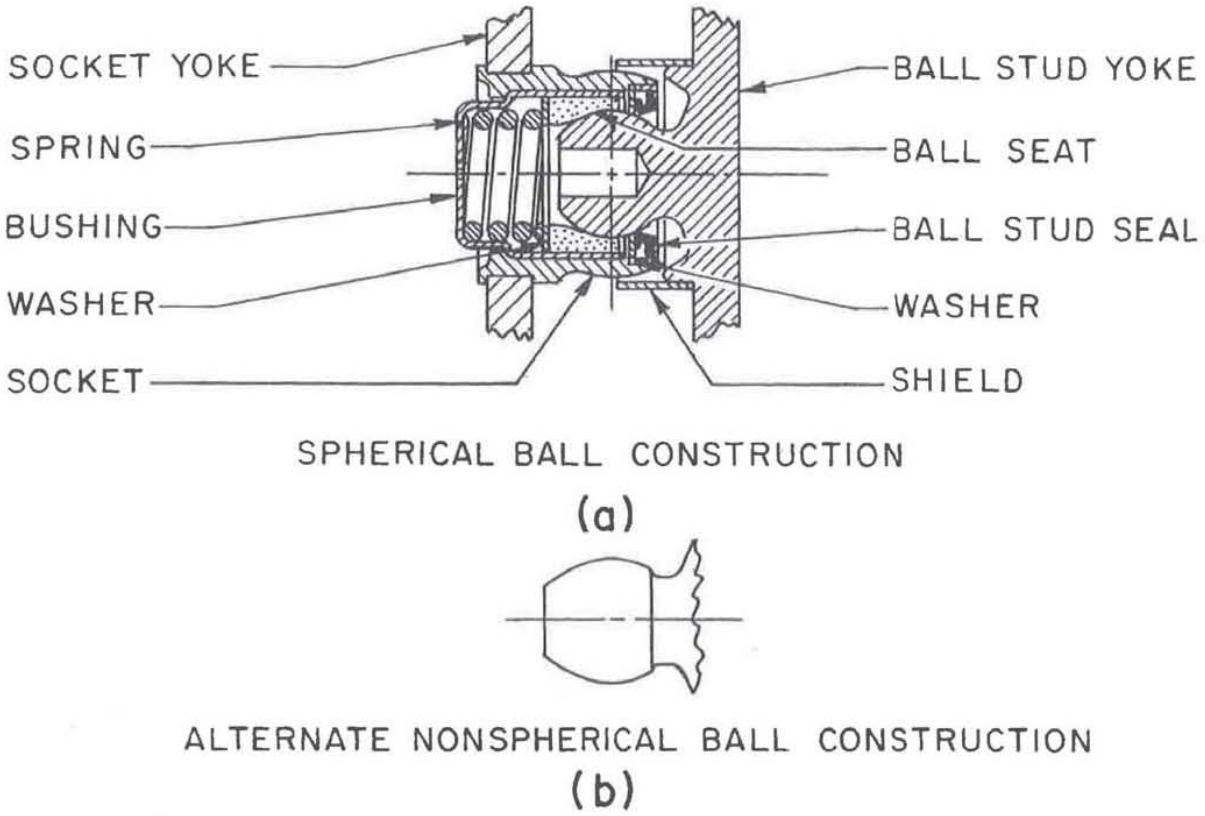


Fig. 11 — Ball stud and seat for driveshaft applications

Because of the various relative motions occurring between centering mechanism components, lubricant must be provided. In order to maintain a sufficient supply of lubricant and to prevent the ingress of dirt, water, and other contaminants, this mechanism must be adequately sealed. For wheel drive applications, a boot type seal surrounding the centering device, as shown in Fig. 9, may be adequate if the joint operates in the open. This boot seal is a flexible member and must be capable of operating under varying environmental conditions at various speeds and joint angles. If the entire joint operates in an enclosed environment, a seal may not be required. For high speed driveshaft applications, a minimum swing diameter of the joint is often a design requirement. Therefore, the clearance space surrounding the centering device is generally at a minimum also. This is especially true when the joint is angulated to its maximum design angle and can, therefore, preclude the use of a boot type seal. Also, unless special design features are employed, a simple boot seal may allow the lubricant to be spun out of the socket cavity and into the seal area and thus cause overheating of the centering device. Hence, a lip type seal is generally pressed onto, or into, the end of the socket and seals the ball as shown in Fig. 10. In addition, a special lip type seal is required to seal the annular space created by the needle rollers between the ball bore and stud.

The second basic type of centering device used in double Cardan joints is the ball stud and seat construction shown in Fig. 11. It does not permit the high joint angles required in steerable wheel drive applications and, therefore, restricts the joint to smaller angles such as are found in drivelines.

The ball can be spherical or a generated non-spherical shape, while the seat construction can be a one piece or a multipiece segmented design. When the spherical ball is used, the joint will transmit constant velocity at zero and at the design joint angle only. At all other angles, it is nearly a constant velocity joint. When using a non-spherical ball, having its surface generated by the revolution of a compound curve of the proper shape, the joint can transmit constant velocity over an angular range from zero to a desired design angle. Beyond this range, it transmits nearly constant velocity like the joint with a spherical ball centering device.

A compression spring is required to hold the seat or seats against the ball, otherwise the joint will not be self-supporting. Also, the spring is necessary to compensate for wear between the ball and seat, or seats, and to allow endwise motion.

As in the case of the previously described centering device, the magnitude of the spring force can have a major influence on joint function and should be controlled as required. This spring force reveals itself as joint angulation torque which is the torque required to angulate the joint over its angular range through zero joint angle, and has a useful purpose in predicting joint performance and life.

When the integral ball moves from its normal contact position with respect to the seat or seats, it is said to *unseat* and thereby

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cause the joint to be nonself-supporting at that instant. This characteristic is true for both the one piece and the multipiece segmented seat design. The multipiece segmented seat design has the advantage over the one piece seat in that the segmented seats wedge themselves between the ball and the socket bushing bore and benefit from the added frictional resistance. Therefore, a smaller spring force is required to hold the seats against the ball as compared to the one piece seat design for a given driveline torque, joint angle, and seat contact angle.

Also, in a multipiece seat design, it is desirable to use three seats since they provide three point contact with the ball at approximately 120 deg intervals. The action of the three seats is similar to a three-jawed chuck providing positive centering.

Fig. 12 shows a centering mechanism having three segmented seats contacting the spherical surface of the ball at 15 deg. These seats have an outer cylindrical surface, while the inner surface is conical for simplicity. If desired, this inner surface can also be curved. The seats are held against the ball in the socket bore by a flat back-up washer and a spring.

In the previous illustrative problem of a one piece driveline with double Cardan joints, the secondary couple forces acting on the front and rear joint centering balls were also computed. Because of the higher joint angle at the rear joint, a higher load resulted there. This force was 757 lb acting on the ball normal to the plane of the 9°16′ joint angle at 16 000 lb-in WOT torque in forward start, as shown in Fig. 12.

The spring force necessary to prevent *unseating* of this ball for a given seat contact angle, driveline torque, and joint angle can be computed. The axial force imposed on the seats by the ball is:

$$F_a = F_n \tan 15 \text{ deg} = 757 \times 0.26795$$

$$F_a = 203 \text{ lb}$$

The sum of the frictional force between the seats and the bore, and the spring force, must equal or exceed this axial force for no movement of the seats away from the ball as follows:

$$F_a \leqq F_f + F_s$$

As a result of an actual test of a centering mechanism assembly, the coefficient of friction between the segmented seats and bore was found to be approximately 0.13. Therefore, the frictional force developed is:

$$F_f = \mu F_n = 0.13 \times 757 = 98 \text{ lb}$$

The spring force is then:

$$F_s = F_a - F_f = 203 - 98 = 105 \text{ lb}$$

This is the spring force which must be exerted on the seats in order to prevent *unseating* of the ball during the WOT forward start condition.

From these computations, it is apparent that the contact angle between the ball and seat should be kept as low as possible without causing such severe wedging or jamming of the seat between the ball and socket bore, which may impede the required

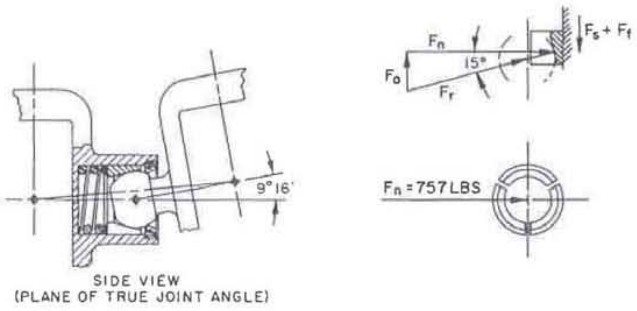


Fig. 12 — Centering mechanism with three segmented seats at WOT start condition load

smooth movement of the seat along the bore during joint angle changes. This minimum contact angle must be somewhat greater than the limiting friction angle determined by the contacting member materials and surface conditions.

In these computations for the required spring force, the assumed forward start condition would cause the reaction torque of the rear axle differential to rotate the pinion shaft or input end in a clockwise direction when viewed from the left side of the vehicle, and result in a decreasing rear joint angle. This tendency to reduce the rear joint angle can be noted when referring to Fig. 4. Even though the vehicle incorporated a rear suspension which counteracted and controlled to a satisfactory degree the pinion windup and rear end squat during forward start, the resulting decrease in joint angle produced the centering mechanism force system shown in Fig. 12.

In studying Figs. 4 and 12, it can be seen that when the vehicle is subjected to a reverse start condition, the tendency would be for the rear joint angle to increase due to pinion winddown and hence the centering ball and seats would move in the opposite direction. The direction of the seat-to-bore friction force reverses resulting in a lower driveline torque where *unseating* of the ball can occur. During these forward or reverse start conditions at relatively high torque, this unseating of the ball is identified as a *click* type of disturbance. This *unseating* of the centering ball can also occur in certain passenger cars operating at relatively high speed, road load conditions where it has been described as a *buzz* type of disturbance. This disturbance can result at certain drivetrain resonant frequencies where the buzz has been found to be torque and joint angle sensitive. Therefore, the selection of the optimum spring force required in a given joint centering mechanism will depend on an evaluation of the various operational modes encountered in a given vehicle application.

The maximum value of the normal ball force can also be used to determine the required neck size of the ball stud. In addition, the Hertz contact stress between the ball and seats can be computed.

Since the seat or seats move in and out relative to the socket, and the ball moves in an oscillatory motion relative to the seat or seats, the centering device must be provided with lubricant and be adequately sealed, as in the case of the previously described centering device. Either the entire mechanism must be sealed with a boot type seal or the socket can be sealed with a lip type seal, which contacts the surface of the ball.

Previously, it was indicated that the clearance space surrounding the centering device is often at a minimum. Therefore, a lip type seal pressed into the bore of the socket is most widely used as

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shown in Fig. 11. Because the ball moves axially in and out relative to the socket bore during joint angulation, the seal must be capable of operating on a variable diameter. It is highly desirable to locate the sealing surface near the equator of the ball so that when in and out motion occurs, the change in diameter will be at a minimum. An additional reason for locating the sealing surface near the ball equator is that this is near the center of oscillation of the joint, where relative motion between the seal and ball is at a minimum also.

Yokes

The input and output yokes can assume various constructions of drivable attaching means, depending on the specific application. These yokes and the coupling yoke of the double Cardan joint must be designed to provide for ease of assembly of the joint components as well as to provide the desired joint angular capability and strength requirements. Because of the complex shaped yokes, a computer stress and deflection analysis procedure using the finite element method can be employed. Also, experimental stress analysis techniques may be used to eliminate or reduce the areas of high stress concentration. As in the case of Cardan joint components, it is highly desirable to provide for *balanced* yoke and cross deflection so that the needle rollers will not be pinched at either end of the bearing or cross trunnion, but are loaded uniformly along their active length.

The coupling yoke is a connecting double yoke member and should be short and compact with the swing diameter as small as possible so that the rotational inertia effects can be kept to a minimum. This is desirable since the coupling yoke does not rotate at constant velocity as has been shown. The minimum possible swing diameter is generally dictated by the maximum joint angle requirement as well as the joint capacity or size.

In order to eliminate the possibility of damage to the centering mechanism due to over-angulation during operation or handling, it is advisable that the permissible maximum joint angle be determined by interference of the yokes. Surfaces functioning as angulation stops can be incorporated at appropriate locations on the yokes as shown in Fig. 13.

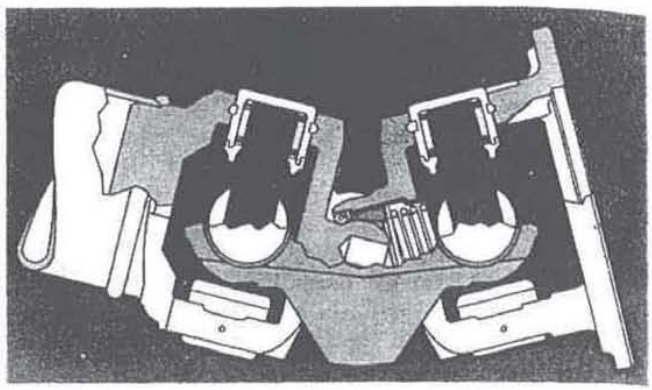


Fig. 13 — Double Cardan joint with angulation stops

Material Selection

The selection of optimum materials for the double Cardan joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages. Also, the type of centering mechanism employed and its detail design can have a major influence on material selection for some joint components. The materials and heat treatments used for the various bearing and cross components are the same as described in the section on the *Cardan or Hooke Universal Joint*.

The various joint yokes can be made of different materials depending on their specific functional and design requirements. Those yokes welded during assembly operations are generally machined from forgings made of resulphurized or plain medium carbon steel which are heat treated to a typical range of BHN 228-255. Other yokes which are not subsequently welded to a connecting member, such as a tube, are frequently made from either pearlitic malleable or nodular iron castings. These castings when heat treated to a typical range of BHN 241-269 for pearlitic malleable iron and BHN 241-285 for nodular iron, allow the machining of yokes having the equivalent strength of forged steel components. The coupling yoke is often made from such typical castings.

When the centering ball and stud type of centering device is used, the centering ball is usually made of chromium steel which is heat treated to a typical R_c60 minimum before finish grinding. In a joint for wheel drive applications, this ball functions in a socket yoke as shown in Fig. 9 and which is generally made from a steel forging. The inside surface of the socket is induction hardened to an adequate case depth and a typical R_c50-55 surface hardness range before finish grinding. The ball in turn is supported on a centering stud which is usually made of plain low carbon or low carbon alloy steel. This stud is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum before finish grinding.

In a joint for high speed driveshaft applications, however, this ball functions in a more complex mechanism as shown in Fig. 10. The ball is retained in a socket mounted in a cavity of the socket yoke. The socket is generally made of a resulphurized low carbon steel which is usually cadmium plated for good wear properties and provided with lubrication grooves on its spherical inside surface. The socket yoke can be made from a steel forging or from a malleable or nodular iron casting. The pivoting ball is supported by the stud portion of the centering stud yoke through a full complement of needle rollers. These needle rollers are made of chromium steel which are hardened to a typical range of R_c60-64 and finish lapped. The yoke is usually made from a steel forging and the stud is induction hardened to an adequate case depth and a typical R_c50-55 surface hardness range before finish grinding. The spring used in the more complex centering device is generally made from carbon spring steel wire. The elastomeric material of the boot or lip type seal is usually a synthetic rubber which must exhibit specific resistance to the action of petroleum-based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application.

The ball stud and seat type of centering mechanism components shown in Fig. 11 can be made of several different material

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combinations depending on the detailed design features. Generally, the material selected for the seat or seats will dictate the material required for the yoke having the integral centering ball. For example, if the seat material is heat treated powdered iron, it is possible to use conventional yoke forging steel for the ball stud yoke. This material allows induction hardening of the ball to an adequate case depth and a typical R_c45-52 surface hardness range before finish grinding. This hardness range is satisfactory for a ball operating on a hardened powdered iron seat.

If the seat material is chromium steel and heat treated to R_c60 minimum before finish grinding, the forging steel used for the ball stud yoke should have the ability to allow induction hardening of the ball to an adequate case depth and a typical R_c58 minimum surface hardness before finish grinding. This may require steel of special composition while maintaining good machinability.

Because the seat or seats move axially in the bore of the socket during joint angle changes, a smooth hardened bearing surface must be provided. Induction hardening of the bore to an adequate case depth and a typical R_c45-52 surface hardness range before finish grinding is usually satisfactory. It is also possible to eliminate this hardening operation on the socket yoke bore if a hardened steel bushing is pressed into place and finish ground. The use of this hardened bushing insert also allows the substitution of a pearlitic malleable or nodular iron casting in place of conventional yoke forging steel for the socket yoke if desired. Since the spring is a very important component in this type of centering device, it is generally made from carbon spring steel wire.

In the segmented seat version of this centering device, a washer is required between the seats and the spring so that uniform pressure is exerted by the spring on all seats and in turn on the ball. To prevent distortion of this steel washer, it can be hardened. Depending on the particular design, it may also be desirable or necessary to have a plain, low carbon steel washer located between the seat or seats and the seal. The function of this washer is to prevent seal damage by the spring-loaded seat or seats prior to final assembly or during over-angulation of the joint. The elastomeric material of the lip type seal is usually the same as for the seal described in the previous centering device and must meet similar performance requirements.

Lubrication

The lubricant used in the bearing and cross assemblies of the double Cardan joint is generally the same as that used in a Cardan joint for these same components in similar applications. This lubricating grease is described in the section on the *Cardan or Hooke Universal joint*.

For the centering mechanism, a lubricating grease having an NLGI grade number 1 is usually used. A typical lubricant of this type generally consists of a refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

Typical basic performance requirements of a lubricating grease which must be met and its various properties are described in the *SAE Handbook* under *Automotive Lubricating Greases*. In addition to providing for optimum joint life by reducing friction and wear at all bearing surfaces, the lubricant used must perform satisfactorily in all environmental conditions of the joint application. The joint bearing and cross assemblies, as well as the centering mechanism, may be lubricated for life or provided with lubrication plugs or fittings for periodic replenishment of the lubricant.

Rating Procedure

The double Cardan joint can be rated on the same basis as the Cardan joint where the term *bearing factor* is used in rating and comparing joint sizes. In addition, the joint can also be rated as described in *Appendix I*. It is highly desirable for reasons of interchangeability and manufacturing economics that the bearing and cross components be the same for Cardan and double Cardan joints of identical capacity rating or size. Further, it was shown in the section on the design of the centering device that this mechanism must be capable of allowing the maximum required joint angle and speed as well as withstand the forces resulting from the maximum transmitted torque.

Applications

The double Cardan joint has found widespread usage in drivelines and axle driveshafts as well as in other drive system applications. For passenger car driveline applications, it has been found that a Cardan joint will operate satisfactorily with a minimum of disturbances at continuous operating joint angles up to about 4½ deg. At approximately 5 deg and higher, a constant velocity joint such as the double Cardan joint has been found necessary in many installations. This joint will allow continuous operating angles up to approximately 13 deg in automotive drivelines. This is nearly three times the angle which would be allowed by a Cardan joint performing the same function. The upper limit of this continuous operating range is determined by a combination of the permissible oscillatory motions and bearing forces in the centering device, as well as in the two Cardan joints. These motions and forces combine to accelerate wear and create frictional heat which affect joint life. For high speed driveshaft applications, the maximum angular design capability of the joint constructions commonly used is approximately 24 deg. Double Cardan joints have been used in both one and multipiece driveline arrangements. In multiple joint drivelines it has been often found necessary to substitute a double Cardan joint for a Cardan joint, especially next to the shaft support bearing, to reduce secondary couple excitation to an acceptable level.

In front wheel axle driveshaft applications, the double Cardan joint has been used as the outboard joint where there has been a need for a constant velocity joint providing a relatively high joint angle capability. The maximum angular design capability of the joint construction commonly used in axle driveshafts is approximately 45 deg.

The normal design operating angle of the joint will depend on

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the specific applications while the maximum intermittent operating angle must be less than the maximum angular capability for a given joint design. For live, rigid front axle applications, normal operating angles have ranged from 0-1½ deg, while the maximum intermittent operating angle has been limited generally by the steering knuckle design.

For independent wheel suspension axle driveshafts such as employed in passenger car applications, normal operating angles as high as approximately 6 deg have been used, while the maximum intermittent operating angle during suspension and steering movements must not exceed the maximum angular capability of the joint.

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Section 3.2.3
Double Cardan Universal Joint
(Trunnion Pin Type)

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

Introduction

The patent literature shows that the concept of the trunnion pin type of double Cardan universal joint was developed as early as 1903 and that patents on various applicable features were issued through the mid 1930s. Its first known vehicle usage began in the late 1920s as the outboard joint in an early American front wheel drive passenger car. Since that time, it has been used as the outboard joint in German trucks and other heavy-duty highway and off-highway, all wheel drive vehicles for both commercial and military applications.

The double Cardan universal joint (trunnion pin type), shown in Fig. 1 in a typical design, is a near constant velocity universal joint which consists of two trunnion pin type Cardan universal joints whose trunnion yokes are connected by a coupling yoke with internal supporting and centering means, and has intersecting shaft axes. At the design joint angle and at zero, the instantaneous angular velocity ratio is unity while at other joint angles, it is near unity.

This joint, which is similar to the conventional double Cardan joint, also has the same kinematic advantages as that joint. Its other advantages are generally the same as for the double Cardan joint. Also, its design configuration permits a relatively rugged construction. The principal disadvantages of this joint are its relatively higher swing diameter and greater manufacturing cost when compared with the double Cardan joint.

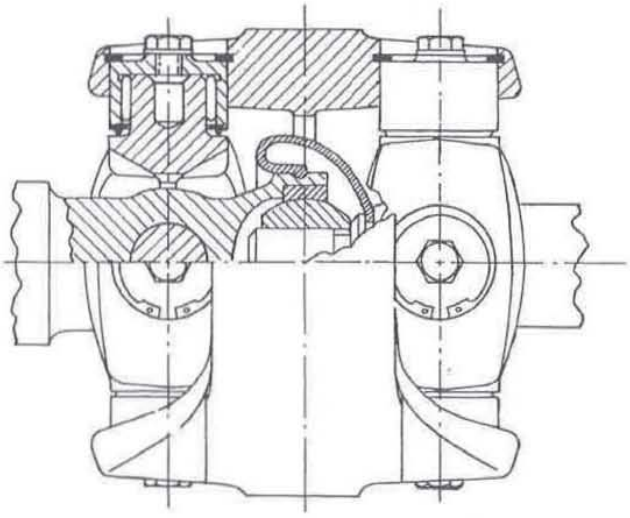


Fig. 1 — Double Cardan universal joint trunnion pin type

Kinematics

The trunnion pin type of double Cardan joint has the same kinematic characteristics as the double Cardan joint.

Effect of Centering Device Location

The location of the centering device with respect to the two trunnion pin type Cardan joint centers of the trunnion pin type of double Cardan joint determines the operating characteristics of the joint. This location provides for equal angles of the two halves of the joint and theoretically correct constant velocity at one joint angle condition (angular range in the case of the generated ball design) only. As described in the section on the *Double Cardan Universal Joint*, the kinematic characteristics of the joint are thus affected and hence, it is desirable to locate the centering device so as to produce a minimum inequality of these individual joint angles over the full angular operating range of the joint.

In some joint constructions, no internal supporting and centering means is employed. In those joints the supporting and centering function is incorporated external of the joint such as in the open steering knuckle of the front axle, for example. The kinematic characteristics are thus determined by the position of the joint relative to the external supporting and centering mechanism.

Centering Device Function

The function of the centering device of the trunnion pin type of double Cardan joint is the same as described in the section on the *Double Cardan Universal Joint*.

Design

The various components of a typical trunnion pin type of double Cardan joint must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the centering device as well as the bearing and cross configurations are determined; then, the yoke constructions can be designed to meet the various joint application parameters.

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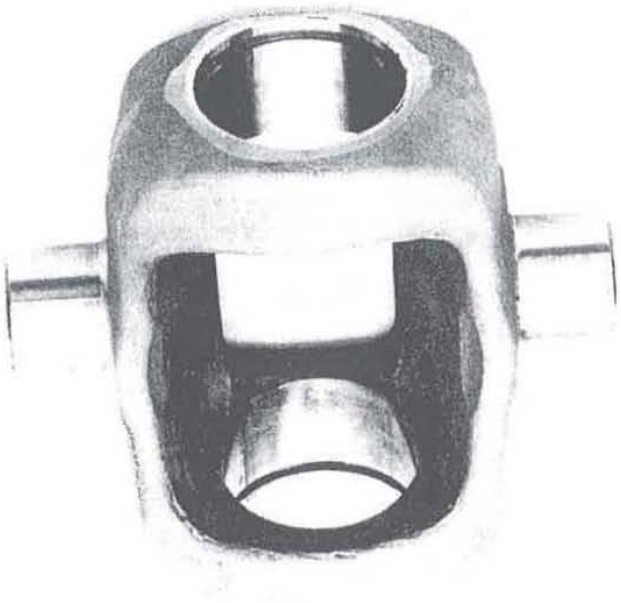


Fig. 2 — Trunnion yoke

Whenever possible, it is desirable to incorporate existing bearing components which are being used in a comparable size Cardan joint. When a new bearing configuration is required, the design procedure for these components is described in the section on the *Cardan or Hooke Universal Joint*.

Centering Device

The centering device of the trunnion pin type of double Cardan joint is generally a ball and socket type of mechanism which is basically a self-aligning bearing featuring a centering ball and stud construction. This construction, and its sealing means, is described in the section on the *Double Cardan Universal Joint*.

Yokes and Other Joint Members

Depending on the specific design of the trunnion pin type of double Cardan joint, the yokes can assume various constructions. The coupling yoke is described in the section on the *Double Cardan Universal Joint*.

The trunnion yoke, shown in Fig. 2, is a yoke with a pair of opposing trunnions. In this joint, two trunnion yokes are used and serve as unconventional cross members. The trunnions connect to the coupling yoke or coupling plate, while the yoke cross holes are connected to trunnion pins which are located in both input and output shafts.

In the sections on the *Cardan or Hooke* and *Double Cardan Universal Joints*, the design of yokes is discussed in varying detail. The procedures for designing the trunnion pin as well as the input and output shaft configurations are described in the section on the *Ball and Trunnion Universal Joint*.

Material Selection

The selection of optimum materials for the trunnion pin type of double Cardan joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The coupling yoke is generally made from a plain medium carbon steel forging which is heat treated to a typical hardness range of BHN 255-285. The trunnion yoke is also usually made from a plain medium carbon steel forging which is heat treated to a typical hardness range of BHN 255-285. This material permits induction hardening of the trunnions to an adequate case depth and a typical R_c60 minimum surface hardness to provide the required wear characteristics and bending strength. The trunnions are then finish ground.

The input and output shafts are generally made from plain medium carbon steel forgings which are induction hardened to an adequate case depth and a typical R_c60 minimum surface hardness on the centering stud and socket areas as required to provide the necessary strength and wear resistance. The shaft sections themselves are induction hardened to an adequate case depth and a typical surface hardness range of R_c44-54 or higher, to provide the required torsional strength.

The trunnion pin is usually made of low carbon alloy steel which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum and finish ground. The necessary case depth is determined by the joint size and its application requirement and is essentially the same as discussed for the Cardan joint cross trunnion, and is also explained in *Appendix A*. The centering ball is generally made of chromium steel which is heat treated to a typical R_c60 minimum and finish ground.

The elastomeric material of the boot seal is usually a synthetic rubber which must exhibit specific resistance to the action of petroleum-based fluids, and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The materials and heat treatments used for the various bearing components are the same as described in the section on the *Cardan or Hooke Universal Joint*.

Lubrication

The lubricant used in the bearing and trunnion yoke assemblies of the trunnion pin type of double Cardan joint is generally the same as that used in the bearing and cross assemblies of a Cardan and double Cardan joint. This lubricating grease is described in the section on the *Cardan or Hooke Universal Joint*. The lubricant used in the centering mechanism is usually the same as that described for the double Cardan joint's centering device.

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

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Rating Procedure

The trunnion pin type of double Cardan joint can be rated on the same basis as the Cardan and double Cardan joints where the term *bearing factor* is used in rating and comparing joint sizes. In addition, the joint can also be rated as described in *Appendix I*.

Applications

The trunnion pin type of double Cardan joint has found its greatest usage as the outboard joint in axle driveshafts of front drive vehicles where there has been a need for a constant velocity joint providing a relatively high joint angle capability. The maximum angular design capability of this joint is approximately 40 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle must be less than the maximum angular capability for a given joint design. For live, rigid front axle applications, normal operating angles have ranged from 0-1¾ deg, while the maximum intermittent operating angle has been generally limited by the steering knuckle design. For independent wheel suspension axle driveshafts, such as those incorporating the typical two-joint halfshaft construction, normal operating angles as high as approximately 6 deg have been used, while the maximum intermittent operating angle during suspension and steering movements must not exceed the maximum angular capability of the joint.

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Section 3.2.4
Double Cardan Universal Joint
(Twin Ball and Centering Plate Type)

F. C. Nagele — Rockwell International
E. R. Wagner, P.E. — Saginaw Steering Gear Div., GMC

Introduction

The twin ball and centering plate type of double Cardan universal joint was patented by Geisthoff in the late 1960s and has been used primarily for slow speed applications such as in agricultural implement power take-off driveshafts.

The double Cardan universal joint (twin ball and centering plate type), shown in Fig. 1 in a typical design, is a constant velocity universal joint which consists of two Cardan universal joints whose crosses are connected by a two-piece flanged coupling yoke with internal supporting and centering means, and has intersecting shaft axes. At all operating angles within the design range, the instantaneous angular velocity ratio is unity.

The two ball centering device divides equally the total joint angle between the two joint halves. This joint transmits constant velocity because the second joint compensates for the fluctuations in angular velocity created by the first joint. Also, this joint is similar to the conventional double Cardan joint with the principal difference being in the design of the centering mechanism.

The principal disadvantage of this joint is the friction producing orbiting characteristic of the centering device's socket and centering plate when operating at an angle. The joint is therefore restricted to relatively low rotational speeds.

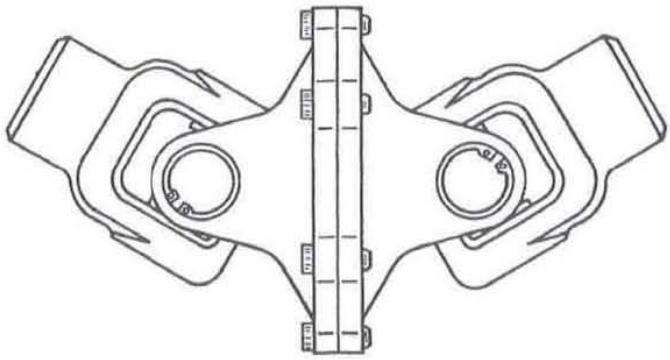


Fig. 1 — Double Cardan universal joint (twin ball and centering plate type)

Kinematics

Velocity Variation and Torsional Excitation

The twin ball and centering plate type of double Cardan joint transmits true constant velocity at all operating joint angles within the design range. Since there is no output velocity variation from this joint, the torsional excitation is effectively cancelled within the joint itself.

Inertia Excitation

In a twin ball and centering plate type of double Cardan joint, the coupling yoke rotates at nonuniform speed when the input angular velocity is constant. Since the rotational inertia of the coupling yoke is accelerated and decelerated through nonuniform motion, oscillating torque loads are impressed on the driveline system. This is known as inertia excitation, and its magnitude depends on the size of the joint angle and the inertia of the coupling yoke. The detailed procedure for analyzing these inertia effects is discussed in the section on the *Cardan or Hooke Universal Joint*.

Secondary Couple Excitation

The analysis of secondary couples for a twin ball and centering plate type of double Cardan joint is the same as for a double Cardan joint. Therefore, the equation for the resultant couple force at each of the centering device ball centers is also the same. A detailed analysis is shown in the section on the *Double Cardan Universal Joint*.

Centering Device Function

The function of the centering device is to provide internal support so that the twin ball and centering plate type of double Cardan joint is self-supporting and self-aligning and divides the joint angle equally between the two halves of the joint. It must have the physical strength to support the weight and unbalanced force of the attached driveshaft and the forces resulting from the secondary couples.

Design

The various components of a typical twin ball and centering plate type of double Cardan joint, shown in a partially disassembled view in Fig. 2, must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by

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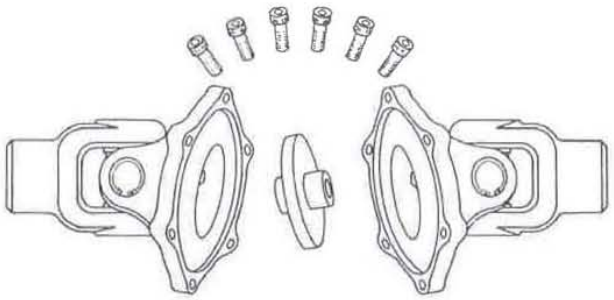


Fig. 2 — Partially disassembled twin ball and centering type of double Cardan universal joint

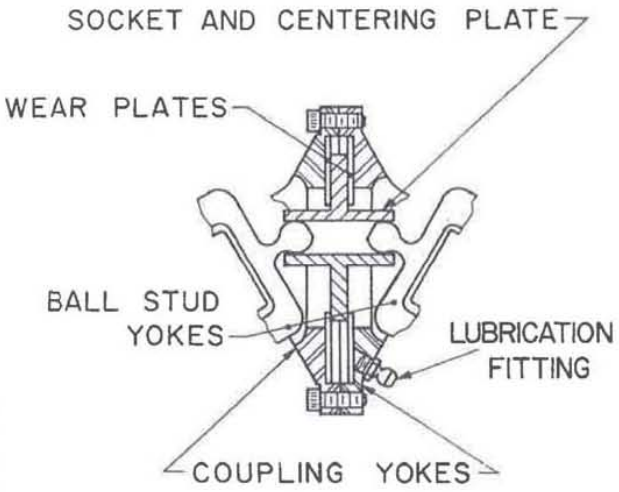


Fig. 3 — Centering device design

various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the centering device as well as the bearing and cross configurations are determined; then the yoke constructions can be designed to meet the various joint application parameters.

For purposes of standardization and economic reasons it is desirable to incorporate, whenever possible, existing bearing and cross components which are being used in a comparable size Cardan joint. When a new bearing and cross configuration is required, the design procedure for these components is described in the section on the *Cardan or Hooke Universal Joint*.

Centering Device

The centering device of the twin ball and centering plate type of double Cardan joint, shown in Fig. 3 in a typical construction, allows a substantially higher operating joint angle than the conventional double Cardan joint. Joint angles as high as approximately 70 deg are possible. The higher operating joint angles are possible because the centering mechanism is controlled by two ball studs which are integral with the input and output yokes and are called ball stud yokes. These ball studs symmetrically located about the joint center engage the disc type socket and centering plate which in turn is supported through wear plates in the two-piece coupling yoke. When the joint operates at an angle, the centering mechanism must orbit within the space provided in the coupling yoke assembly. For the joint to operate without undesirable noise, dimensional tolerances must be selected so that clearances between the ball and socket bore and between the centering plate, wear plates, and coupling yoke are held to a minimum.

The wear plates, socket and centering plate, and balls are lubricated through a passage in the coupling yoke. Although they can be used, ball seats and seals are not shown in the centering device of the design configuration shown in Fig. 3.

Yokes

Depending on the specific application of the twin ball and centering plate type of double Cardan joint, the input and output yokes or ball stud yokes can assume various constructions of drivable means of attachment. The coupling yoke is a two piece design consisting basically of two flange type yokes which are bolted back to back and contains a cavity for the centering device. In the sections on the *Cardan or Hooke* and *Double Cardan Universal Joints*, the design of yokes is discussed in detail.

Material Selection

The selection of optimum materials for the twin ball and centering plate type of double Cardan joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The input and output yokes with the integral ball studs are generally made from plain medium carbon steel forgings, which are heat treated to a typical hardness range of BHN 255-285. This material permits induction hardening of the balls to an adequate case depth and a typical R_c60 minimum surface hardness to provide the required strength and wear resistance. The balls are then finish ground.

The two piece coupling yoke can be made of the same or similar forging steel as the input and output yokes and heat treated to a typical range of BHN 255-285. If desired, the two piece coupling yoke can also be made from either pearlitic malleable or nodular iron castings. These castings when heat treated to a typical range of BHN 241-269 for pearlitic malleable iron and BHN 241-285 for nodular iron, allow the machining of yokes having the equivalent strength of forged steel components.

The wear plates and the socket and centering plate are usually made of a carburizing grade of steel which are surface hardened to an adequate case depth and a typical R_c60 minimum to provide the required wear characteristics and are then finish ground. The materials and heat treatments used for the various bearing and cross components are the same as described in the section on the *Cardan or Hooke Universal Joint*.

Lubrication

The lubricant used in the bearing and cross assemblies of the twin ball and centering plate type of double Cardan joint is

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generally the same as that used in a Cardan and double Cardan joint for these same components. This lubricating grease is described in the section on the *Cardan or Hooke Universal Joint*.

For the centering mechanism, lubricating grease having an NLGI grade number 2 is usually used. A typical lubricant of this type generally consists of a refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant. The centering device is lubricated through a lubrication fitting and channel provided in the coupling yoke.

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

Rating Procedure

The twin ball and centering plate type of double Cardan joint can be rated on the same basis as the Cardan and double Cardan joints where the term *bearing factor* is used in rating and comparing joint sizes. In addition, the joint can also be rated as described in *Appendix I*.

Applications

The twin ball and centering plate type of double Cardan joint has been used primarily in agricultural implement power take-off driveshafts where there has been a need for a constant velocity joint providing a relatively high joint angle capability. The maximum angular design capability of this joint is approximately 70 deg. The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle must be less than the maximum angular capability for a given joint design. It can also be used in other applications where the torque and joint angle requirements are in the same general range, and where the operating speed does not exceed approximately 1000 rpm. The maximum operating speed possible is limited by the frictional characteristics of the centering mechanism, which results in heat buildup, and by the forces resulting from the orbiting of the socket and centering plate.

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Section 3.2.5
Tracta Universal Joint

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

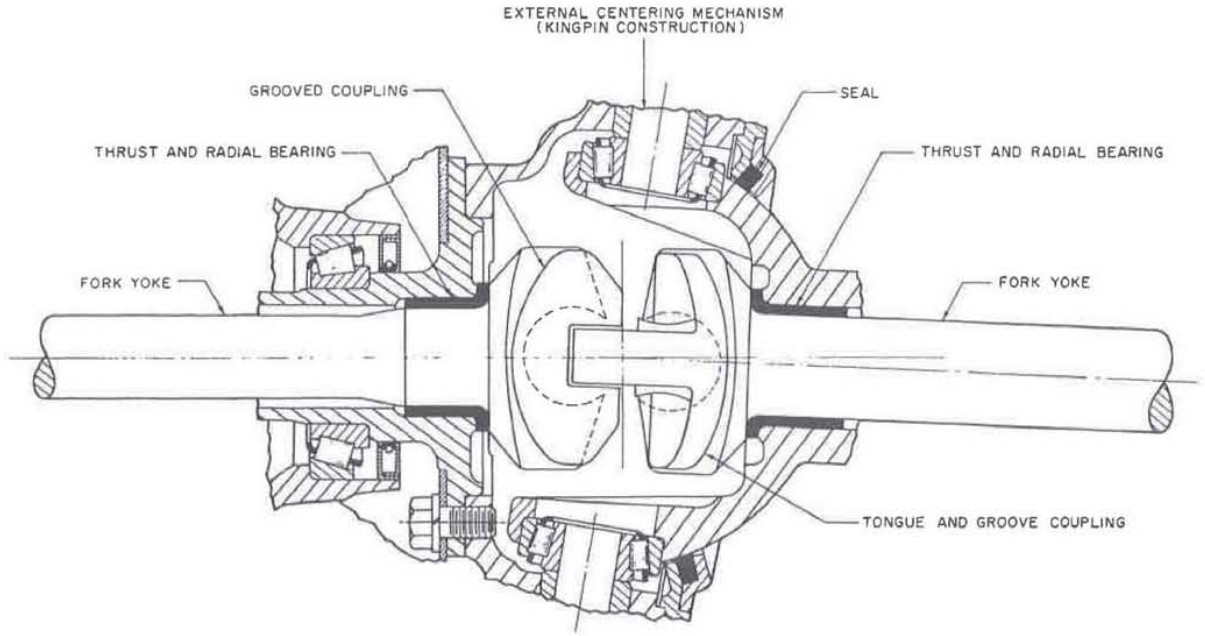


Fig. 1 — Tracta universal joint

Introduction

The Tracta universal joint was patented by Fenaille in the late 1920s. At that time it was used as the outboard joint in a French front wheel drive passenger car built by Fenaille and Gregoire, and for which the joint was named. After that it was applied to other European front drive vehicles and still later was used in some American all wheel drive military vehicles. In addition, the joint has found limited usage in other applications.

The Tracta universal joint, shown in Fig. 1 in a typical design, is a near constant velocity universal joint which consists of two yokes drivable connected through plane bearing surfaces to two intermediate couplings similarly engaged and requires external supporting, centering, and sealing means. Besides transmitting approximate constant velocity, the joint is simple, rugged, and compact and has a relatively high torque capacity for a given swing diameter. Also, it has a relatively high joint angle capability.

Its disadvantages, however, severely restrict its usage in axle driveshaft applications. For example, when operating at high joint angles and torque loads for extended periods of time, the friction due to the sliding bearing surfaces result in heat buildup which can cause failure. Therefore, under heavy torque loads, the joint can function satisfactorily at a continuous operating joint angle of 0 deg only. Also, the joint requires external supporting, centering, and sealing means.

Kinematics

As previously indicated, the Tracta joint transmits nearly true constant velocity. The slight deviation from a true constant velocity condition is due to positioning errors of the joint within the supporting and centering structure. When the geometry of the joint and its supporting and centering mechanism is assumed correct, then it can be proven that the joint can transmit motion uniformly.

Fig. 2 shows the joint displaced at a joint angle, and in its two principal rotational positions, which are 90 deg apart. The driving and driven fork yokes are supported so that their rotational axes will intersect at all times at a point equidistant from the centers of the intermediate hemispherically shaped members, which are the tongue and groove coupling and the grooved coupling. The tongue and groove sliding connection of the two intermediate components permits these members to adjust themselves continu-

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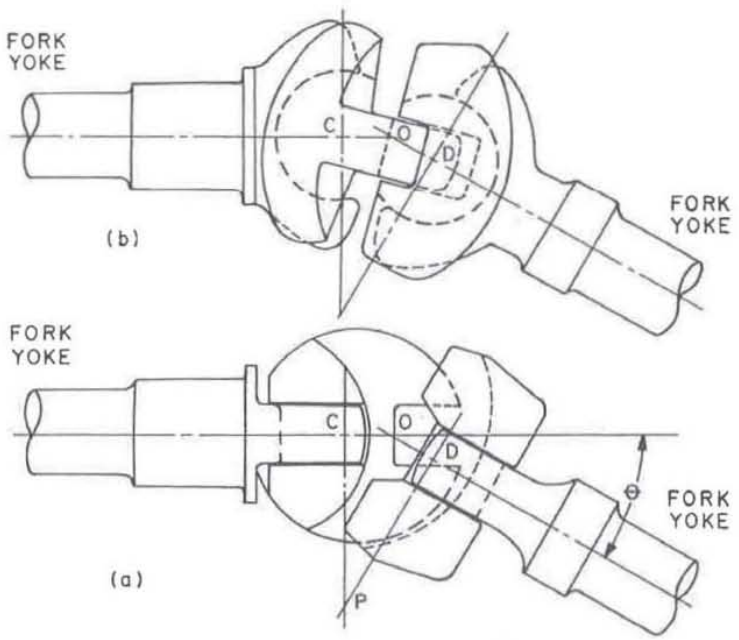


Fig. 2 — Tracta joint in two principal rotational positions 90 deg apart

ously with respect to the angular displacement of their respective mating fork yokes and requires that the pivotal axes of these components must always be in the same plane.

In Fig. 2(a), the intersection of the rotational centerlines of fork yokes A and B is represented by point O. Points C and D represent the centers of the tongue and groove coupling and the grooved coupling respectively, and as indicated, they must be equidistant from point O. Through these centers C and D are drawn the pivoting axes of the intermediate hemispherically shaped members. These axes also represent the pivoting axes of the fork yokes and are perpendicular to the rotational centerlines of the fork yoke shafts. Because of the alignment of the hemispherical components by means of the tongue and groove, these axes will always be in the same plane, and if extended will intersect at point P. The length CO equals the length DO, and angle PCO equals angle PDO. Therefore, the conditions of symmetry are met and the joint transmits constant angular velocity at this instant.

Fig. 2(b) shows the joint rotated through 90 deg and as previously mentioned, points C and D represent the centers of the intermediate members which are equidistant from point O. In this rotational position the pivotal axes through centers C and D are parallel, perpendicular to the joint angle plane, and may then be considered to intersect at infinity. Just as with the rotational position shown in Fig. 2(a), this rotational position also occurs twice per revolution. Since the pivotal axes are in the same plane in this rotational position as they are in all other rotational positions, constant angular velocity is transmitted by this joint.

As illustrated in Fig. 2, the intermediate members occupy different positions at the two principal rotational positions of the joint which are 90 deg apart. This characteristic occurs twice during each revolution and therefore results in a cyclic unbalance. Although this becomes significant at relatively high rotational speeds, it can in most cases be ignored at axle driveshaft speeds.

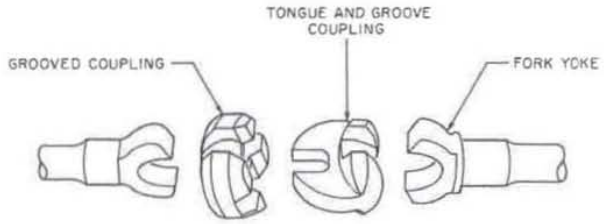


Fig. 3 — Tracta universal joint

Design

The various components of a typical Tracta joint, shown in an exploded view in Fig. 3, must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the joint configuration can be designed to meet the various application parameters.

The fork yoke is a fork-shaped member with plane surfaces engaging the mating groove of an intermediate coupling and with drivable means of attachment. Usually, the fork yoke is integral with the connecting shaft member. The fork ends subtend an angle greater than 180 deg so as to be self-locking when assembled to their mating intermediate couplings. The tongue and

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groove coupling is a hemispherically-shaped intermediate member with tongue and groove located normal to each other. The groove mates with one of the fork yokes, while the tongue engages a groove of another intermediate member. This member is the grooved coupling which is a hemispherically-shaped intermediate member with two grooves located normal to each other. Since one of the grooves provides for engagement of the tongue, the other groove mates with the second fork yoke. In this manner, all four joint members are joined together and are thus able to transmit torque at various joint angles within the design range.

Since all the engaging bearing surfaces must slide relative to each other under load, these surfaces must be adequate in area to provide for acceptable levels of bearing stress. During assembly of the joint, the intermediate components are usually assembled as matched sets to limit the torsional lash. Generally, all machining of the joint components including grinding is done prior to heat treatment. The external supporting, centering, and sealing means for this joint are dependent upon the specific application and will, therefore, not be discussed here.

Material Selection

The selection of optimum materials for the Tracta joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The fork yokes are usually made from medium carbon alloy steel forgings which are heat treated to meet the torsional strength requirements and provide the necessary surface hardness of the fork plane bearing surfaces. A typical heat treatment procedure is to through harden the fork yokes to R_c45-55 and then retemper the shaft and splined ends to R_c41-46 to prevent brittleness.

The tongue and groove coupling and the grooved coupling are generally made from low carbon alloy steel forgings which are heat treated to meet the required torsional strength and provide the necessary surface hardness of the plane bearing surfaces. A typical heat treatment procedure of these two components is to carburize them to a suitable case depth and then surface harden them to R_c58 minimum.

After heat treatment, the four joint components are usually coated with a fine grain conversion coating of manganese iron phosphate. Besides reducing the friction, the phosphate coating acts as a "wear-in" or "break-in" agent for the mating sliding surfaces.

Lubrication

The Tracta joint generally operates submerged in lubricant inside the closed steering knuckle of a vehicle's front live axle steerable wheel. This typical lubricant is either a lubricating mineral oil of SAE viscosity number 90 or 140, or a lubricating grease. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

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

Rating Procedure

The Tracta joint can be rated as described in *Appendix I*.

Applications

The Tracta joint has found usage in the closed steering knuckle of live, rigid front axles of all wheel drive highway and off-highway commercial and military vehicles where there has been a need for a constant velocity joint providing a relatively high joint angle capability. The maximum angular design capability of this joint is in the range of 40-45 deg, which is greater than the angular capability of the closed steering knuckle. In these rigid front axle applications, the normal design operating angles have ranged from 0-1½ deg, while the maximum intermittent operating angle has been limited by the steering knuckle design.

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Section 3.2.6
Tripot Universal Joint
(End Motion Type)

J. H. Dodge — Oldsmobile Div., GMC
E. R. Wagner, P.E. — Saginaw Steering Gear Div., GMC

Introduction

The patent literature shows that the concept of the end motion type of tripot universal joint was developed as early as 1910 and that patents on various applicable features were issued through the early 1970s. Its first known vehicle usage began in the early 1960s as the inboard joint in French front wheel drive vehicles. In 1967, it was similarly applied to two American front drive passenger cars and since that time, this joint has been used in numerous European front drive vehicles. It has also been used as the axle driveshaft inboard and outboard joints in some independently sprung European rear drive vehicles.

The tripot universal joint (end motion type), shown in Fig. 1 in a typical design, is a constant velocity universal joint, radially self-supported and permitting axial movement which consists of a housing drivably connected to a shaft through three equally spaced trunnion mounted balls. This joint and some of its components are similar to the nonconstant velocity ball and trunnion or bipot joint which, however, has only two drive elements.

The principal advantages of this joint are its ability to transmit constant velocity and require a relatively low axial thrust force during end movement when compared with a conventional sliding spline or most other end motion type of joints. This low axial thrust force characteristic eliminates the undesirable noise problems often associated with conventional splines subjected to relatively high torque applications. Another advantage of this joint is that it operates at relatively cooler temperatures when compared with other ball type constant velocity joints in similar applications. In addition, it can be manufactured economically by offering the versatility and flexibility to design various housing, spider, and bearing configurations to meet the functional and durability requirements for a particular application.

A principal disadvantage of this joint is its orbiting characteristic when operating at an angle. This characteristic is directly related to joint size and operating angle. With respect to the plane of the spider, the geometric center of the shaft orbits in a circular path about the joint center with a fully supported housing being assumed. This orbital path moves with constant velocity at three times per revolution of the joint and in the same direction. Therefore, this joint is generally limited to relatively low speed and low joint angle applications and has to be used in conjunction with another universal joint or self-aligning bearing in the drive system. Since the orbiting motion produces inertia effects, orbiting of the relatively large and heavy housing would be undesirable in comparison with a similar motion by the smaller and lighter shaft member. Hence, the joint should always be oriented in a driveshaft layout so that the housing is the supported member while the shaft is free to orbit as required. Also, the maximum joint angle required in an application can limit the axial travel available within the joint.

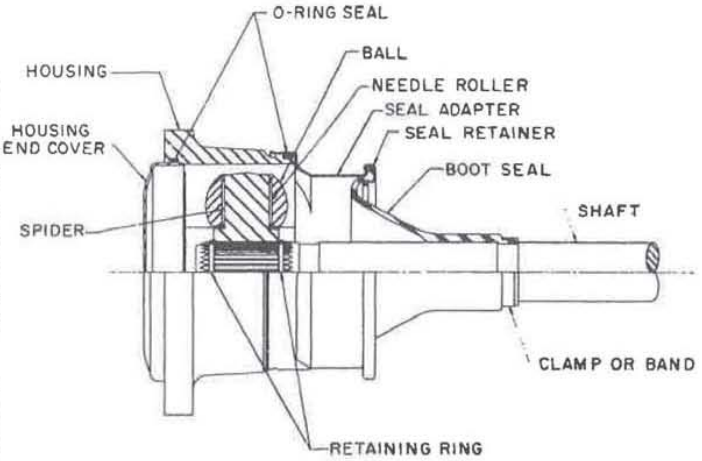


Fig. 1. — Tripot universal joint (end motion type)

Kinematics

NOMENCLATURE

The kinematic characteristics of the end motion type of tripot joint are more complex than one might suspect in a cursory examination. In the following motion and force analysis, the joint is assumed to be of the type having balls which can move endwise along their trunnion axes with their locations determined by the equally spaced axial ball bores whose centers are at the same radius from the housing rotational axis. Also, the spider trunnion centerlines are assumed to be equally spaced and in the same plane. The equations of motion and force are developed according to the following nomenclature:

ω = input angular velocity of driving member (assumed uniform in the analysis)
ω_1 = output angular velocity of driven member
θ = joint angle
β = angle of rotation of driving member
β' = angle of rotation of driven member
T = input torque on driving member (assumed uniform in the analysis)

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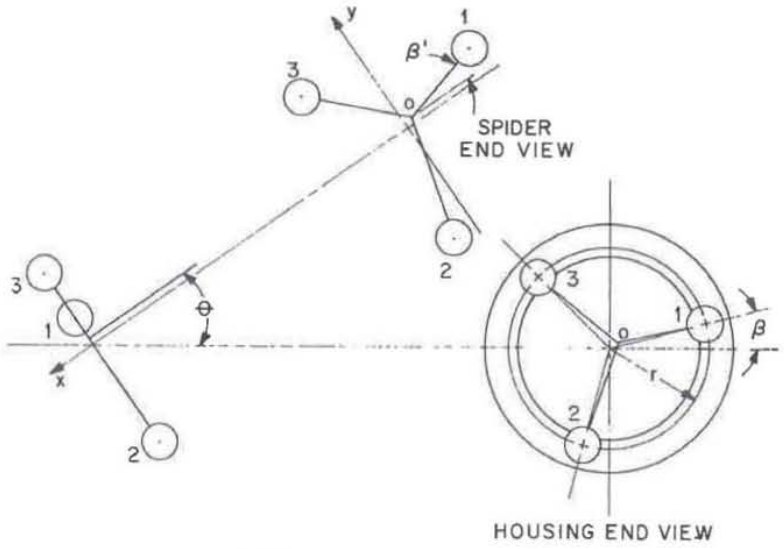


Fig. 2 — Tripot joint angular relationship

T_1 = output torque on driven member
C = secondary couple on driving member
C_1 = secondary couple on driven member
1, 2, 3 = individual balls, ball centers or spider trunnions
F_1, F_2, F_3 = forces normal to respective trunnions in housing end view
D = housing ball bore circle diameter
r = housing ball bore circle radius
r_0 = eccentricity or offset of spider geometric center in spider end view (radius of circular orbiting path) and in housing end view (major axis radius of elliptical orbiting path)
r_0' = eccentricity or offset of spider geometric center in housing end view (minor axis radius of elliptical orbiting path)
r_1 = instantaneous center distance from joint center in plane of spider trunnions
r_1', r_2', r_3' = instantaneous center radii of balls 1, 2, and 3, respectively
v_1, v_2, v_3 = velocity vectors of balls 1, 2, and 3, respectively, for driving member
v_1', v_2', v_3' = total velocity vectors of balls 1, 2, and 3, respectively, of driven member
v_1'', v_2'', v_3'' = velocity vectors normal to instantaneous center radii r_1', r_2', and r_3', respectively
v_1''', v_2''', v_3''' = velocity vectors normal to spider trunnions 1, 2, and 3, respectively
$v_1'^v, v_2'^v, v_3'^v$ = sliding velocity vectors of balls relative to spider trunnions parallel to respective trunnion axes
$\frac{v_1''}{r_1'} = \frac{v_2''}{r_2'} = \frac{v_3''}{r_3'}$ = constant (required condition for constant angular velocity)

Motion Characteristics

The end motion type of tripot joint is shown schematically in Fig. 2 in a general angular relationship. A closed analytical solution using x and y coordinates of the ball centers proves that the joint transmits constant angular velocity, since the angles of rotation β and β' of the driving and driven members, respectively, are always equal.

By means of another approach, it is possible to analytically determine the various motion characteristics. For simplicity, only two specific orientations or principal positions of rotation will be used to develop these interesting characteristics. Fig. 3 illustrates the first principal position of the joint, rotational position (a), to be analyzed. The housing is assumed to be the driving member and also fully supported.

The transmission of motion from driving to driven members is through the balls, which are numbered 1, 2, and 3, clockwise. Ball 1 is in the plane of joint angle θ. Due to the three equally spaced spider trunnions and the three equally spaced housing ball bores, an eccentricity or offset of the driven member or spider geometric center from the driving member axis must take place as shown when the driven member is shifted to joint angle θ from zero joint angle. The joint center is then defined as the point where the housing rotational axis intersects the plane of the spider trunnions.

In the spider end view, this eccentricity is:

$$r_0 = \frac{D(1 - \cos\theta)}{4\cos\theta}$$

while in the housing end view, it is:

$$r_0' = \frac{D(1 - \cos\theta)}{4}$$

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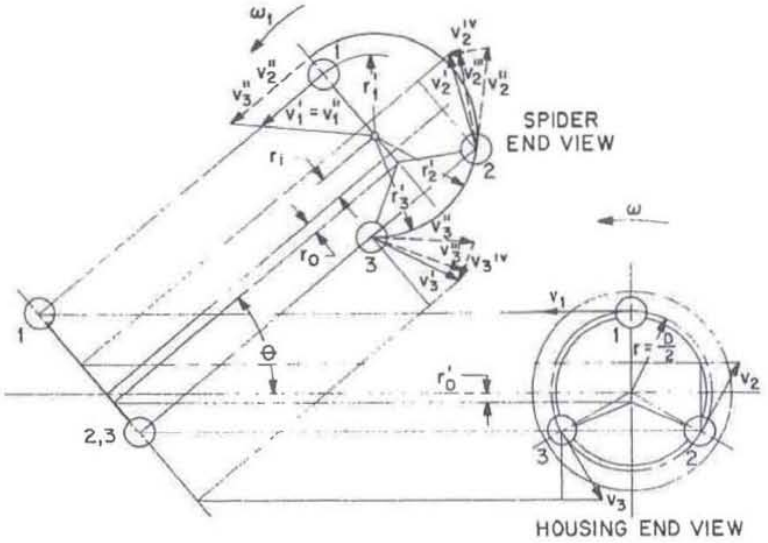


Fig. 3 — Motion analysis for rotational position (a)

The housing or driving member is assumed rotating counterclockwise at constant input angular velocity ω. Ball 1 lies in the plane of the joint angle. In terms of the housing ball bore circle radius r and the input angular velocity ω, the linear speed of this ball with respect to both driving and driven member axes of rotation is:

$$v_1 = r\,\omega$$

With respect to the driving member rotational axis, vectors v_2 and v_3 of balls 2 and 3, respectively, must equal v_1.

A study of the end view of the driven member indicates that the output angular velocity ω_1 must be determined from the instantaneous center and not from the spider geometric center. In Fig. 3, the instantaneous center must be located in the plane of the joint angle due to the symmetrical disposition of the trunnions for balls 2 and 3 about the joint angle plane. In determining the output angular velocity of the driven member, the linear velocity vectors used must be velocities of points on the trunnions at each of the three ball centers. At ball 1, the velocity vector v_1 of the ball for the driving member coincides with and is equal to the velocity vector v_1' of the trunnion for the driven member. When r_1' is assumed equal to r, then the distance of the instant center from the joint center is:

$$r_i = \frac{D(1 - \cos\theta)}{2\cos\theta}$$

This instantaneous center of the spider is, therefore, diametrically opposite its geometric center and offset from the joint center by a distance twice that of the geometric center.

Based on the assumed position of the driven member instant center, an analysis of the various velocity vectors of balls 2 and 3 is made. By construction, it can be proven that the assumed location of the instant center is correct and that the velocity to instant center radius relationship at points 1, 2, and 3 on the trunnions is constant as follows:

$$\frac{v_1''}{r_1'} = \frac{v_2''}{r_2'} = \frac{v_3''}{r_3'}$$

Therefore, the driven member rotates at the same angular velocity as the driving member at this instant; namely,

$$\omega_1 = \omega$$

Fig. 4 illustrates the second principal position of the joint, rotational position (b), to be analyzed. When viewed in the housing end view, this position is 30 deg of rotation counterclockwise from rotational position (a), shown in Fig. 3. Ball 2 is perpendicular to the plane of the joint angle. The spider geometric center is again eccentric to the joint center. In both spider and housing end views, this offset is:

$$r_o = \frac{D(1 - \cos\theta)}{4\cos\theta}$$

However, it has rotated 90 deg, while the joint has rotated only 30 deg. Then, by analysis as described for rotational position (a), it can be shown that the driven member rotates at the same angular velocity as the driving member at this instant; namely,

$$\omega_1 = \omega$$

The analysis also indicates that the spider instant center distance from the joint center is again:

$$r_i = \frac{D(1 - \cos\theta)}{2\cos\theta}$$

It has also rotated through 90 deg and is again located diametri-

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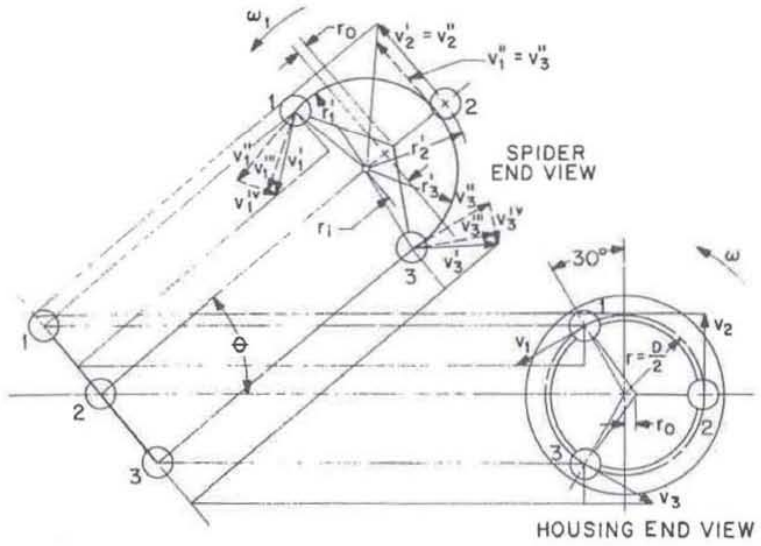


Fig. 4 — Motion analysis for rotational position (b)

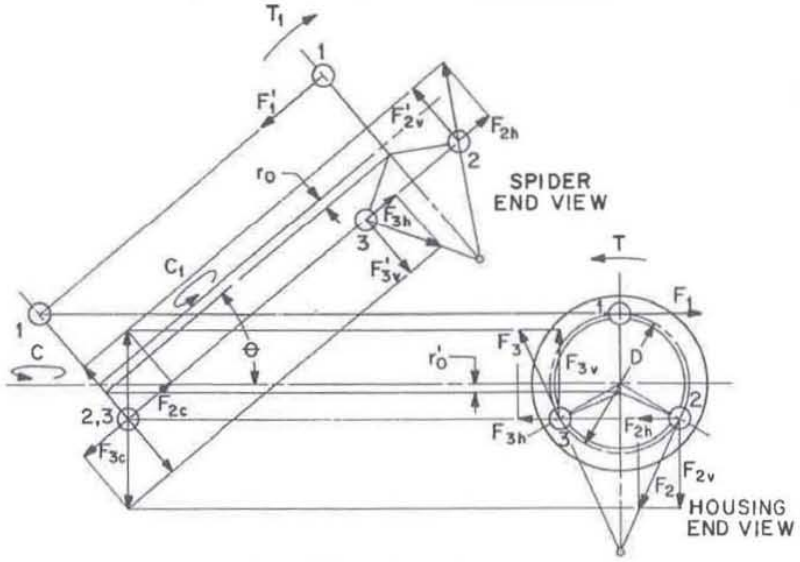


Fig. 5 — Force analysis for rotational position (a)

cally opposite its geometric center from the joint center at a distance twice that of the geometric center.

An analysis for any other position of joint rotation reveals that the joint transmits constant output angular velocity when the input angular velocity is uniform.

The eccentricity of the spider geometric center describes an orbiting path when the joint rotates. With respect to the housing end view, this path is an ellipse having a major axis radius of:

$$r_o = \frac{D(1 - \cos\theta)}{4\cos\theta}$$

normal to the plane of the joint angle, and a minor axis radius of:

$$r_o' = \frac{D(1 - \cos\theta)}{4}$$

in the plane of the joint angle. With respect to the spider end view, this path is a circle having a radius of:

$$r_o = \frac{D(1 - \cos\theta)}{4\cos\theta}$$

in the plane of the trunnions. The spider geometric center rotates in the direction of joint rotation, but at a constant angular velocity of three times the spider angular velocity. Since the instantaneous center is at all times diametrically opposite the spider geometric center, this point also rotates at a constant angular velocity of three times the spider rotational speed.

Force Characteristics

In order to determine the force characteristics, the end motion

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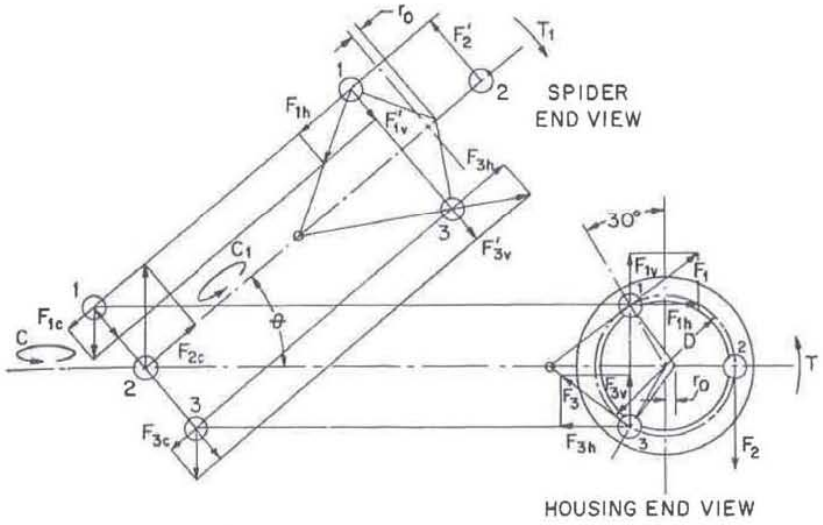


Fig. 6 — Force analysis for rotational position (b)

type of tripot joint will be shown in the same two positions of rotation used in the motion analysis. Fig. 5 illustrates the first principal position of the joint, rotational position (a), to be analyzed. With input torque T applied to the housing or driving member as shown, a resisting torque equal to the output torque T_1 must be applied to the spider or driven member in the opposite direction. The transmission of forces from driving to driven members is through the balls, which are numbered 1, 2, and 3 clockwise. With friction and inertia ignored, the force at each ball must be perpendicular or normal to its corresponding trunnion axis, as shown in the housing and spider end views. The resultant force vectors at balls 2 and 3 in both end views intersect at points which are located on lines normal to the resultant force at ball 1. By setting the moment of F_1 about the housing end view intersection point equal to the driving member torque, the force F_1 can be determined. Then the remaining forces F_2 and F_3 can be obtained by resolving these forces into horizontal and vertical components. It will be noted that F_2 and F_3 are not equal to F_1. Further, by resolving the individual forces into the spider end view, and summing moments, it can be shown that the driving member torque equals the driven member torque and thereby implies that the driving and driven member angular velocities are equal at this instant.

A study of the force components normal to the plane of the joint angle indicates the presence of a secondary couple of the same magnitude on both driving and driven members. These secondary couples are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

The solutions for the resultant forces and their components are summarized as follows:

$$F_1 = \frac{4\,T \cos\theta}{3\,D(1+\cos\theta)}$$

$$F_{2v} = F_{3v} = \frac{2T}{\sqrt{3}\,D(1+\cos\theta)}$$

$$F_{2h} = F_{3h} = \frac{2T\cos\theta}{3\,D(1+\cos\theta)}$$

$$F_2 = F_3 = \frac{2T\sqrt{3+\cos^2\theta}}{3\,D(1+\cos\theta)}$$

$$F_{2v}' = F_{3v}' = \frac{2T\cos\theta}{\sqrt{3}\,D(1+\cos\theta)}$$

$$F_{2c} = F_{3c} = \frac{2T\sin\theta}{\sqrt{3}\,D(1+\cos\theta)}$$

Fig. 6 illustrates the second principal position of the joint, rotational position (b), to be analyzed. A force analysis again shows that the driving member torque equals the driven member torque, which again implies that instantaneously the driving and driven member angular velocities are also equal. The secondary couples on both driving and driven members are again:

$$C = C_1 = T \tan \frac{\theta}{2}$$

Also, as in rotational position (a), the torque is not equally distributed on the balls. The solutions for the resultant forces and their components are summarized as follows:

$$F_2 = \frac{4T}{3\,D(1+\cos\theta)}$$

$$F_{1h} = F_{3h} = \frac{2T\cos\theta}{\sqrt{3}\,D(1+\cos\theta)}$$

$$F_{1v} = F_{3v} = \frac{2T}{3\,D(1+\cos\theta)}$$

$$F_1 = F_3 = \frac{2T\sqrt{1+3\cos^2\theta}}{3\,D(1+\cos\theta)}$$

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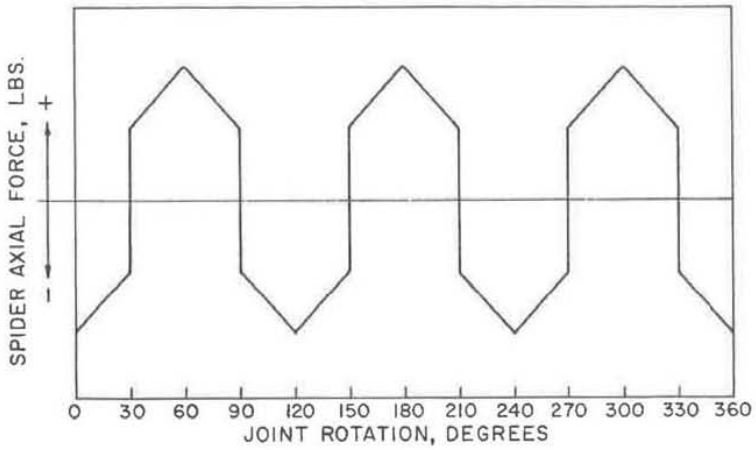


Fig. 7 — Typical theoretical dynamic end thrust

$$F_2' = \frac{4T \cos \theta}{3\,D(1+\cos \theta)}$$

$$F_{1v}' = F_{3v}' = \frac{2T \cos \theta}{3\,D(1+\cos \theta)}$$

$$F_{1c} = F_{3c} = \frac{2T \sin \theta}{3\,D(1+\cos \theta)}$$

$$F_{2c} = \frac{4T \sin \theta}{3\,D(1+\cos \theta)}$$

In practical applications, the joint tends to produce a cyclic or shudder-type disturbance at a frequency of three times shaft speed. This disturbance caused by internal frictional effects is related to joint angle, rotational speed, torque, and joint size. The two sources of friction are sliding of the ball on its spider trunnion and combined sliding and rolling of the ball in its housing ball bore. These friction forces are directed along the trunnion axes, and their overall effects can be evaluated most readily by a computer program. The principal effect of friction is to produce a cyclic axial force or dynamic end thrust along the spider rotational axis. The typical wave form of this force is shown in Fig. 7 as a modified square wave with a frequency of three times rotational speed.

The computer analysis will also show relatively small nonuniform output torque and speed characteristics as well as secondary couple variations. These effects are considered relatively unimportant and insignificant when compared with the axial shudder-type of disturbance which may be undesirable in certain applications. Thus, when frictional characteristics are considered, the tripot joint may be classified theoretically as a quasi-constant velocity universal joint. However, it should be indicated that in the computer analysis, a joint angle was used which was substantially higher than possible in a practical joint where sufficient end motion capability is also desired. Therefore, in a practical sense, the end motion type of tripot joint is considered a constant velocity joint.

Comparison with Other Universal Joints

The end motion type of tripot joint is a distinct ball type of constant velocity joint in that the transmission of motion and force is not through the constant velocity or homokinetic plane, but through a plane which is always normal or perpendicular to the rotational axis of one of the members. This member can be either the driving or driven member. Also, this characteristic regarding the location of the plane of motion and force transmission applies to the fixed center type of tripot joint. All other ball type constant velocity joints utilize the principle of maintaining driving contact through the constant velocity or bisecting angle plane.

In the end motion type of tripot joint, the secondary couple on both driving and driven members is constant at T tan $\theta/2$ for all positions of joint rotation. In all other ball type constant velocity joints which maintain driving contact through the constant velocity or bisecting angle plane, this couple on both driving and driven members is also constant at T tan $\theta/2$.

For the Cardan and ball and trunnion or bipot joints, the secondary couple varies on the driving member from a maximum value of T tan θ to zero twice per shaft revolution with respect to bearing supports, while simultaneously on the driven member, the couple varies from zero to a maximum value of T sin θ twice per revolution with respect to bearing supports. In the section on the *Cardan or Hooke Universal Joint,* it was shown that the resultant couple forces react on bearing supports with a complex excitation, since the driving and driven member couples are 90 deg out-of-phase with each other. The result is equivalent to a static force, constant in size, and normal to the joint angle plane, plus a dynamic or rotating force with a frequency of two cycles per revolution. It is the dynamic force which produces a vibratory excitation on the bearing supports.

In the end motion type of tripot and the other ball type constant velocity joints, the couple forces react as static forces only on the bearing supports and, therefore, do not produce any vibratory disturbances at these points, if geometry errors and friction are ignored. Also, for analysis purposes of universal jointed drive systems, it is of interest that for a given disposition of the joint

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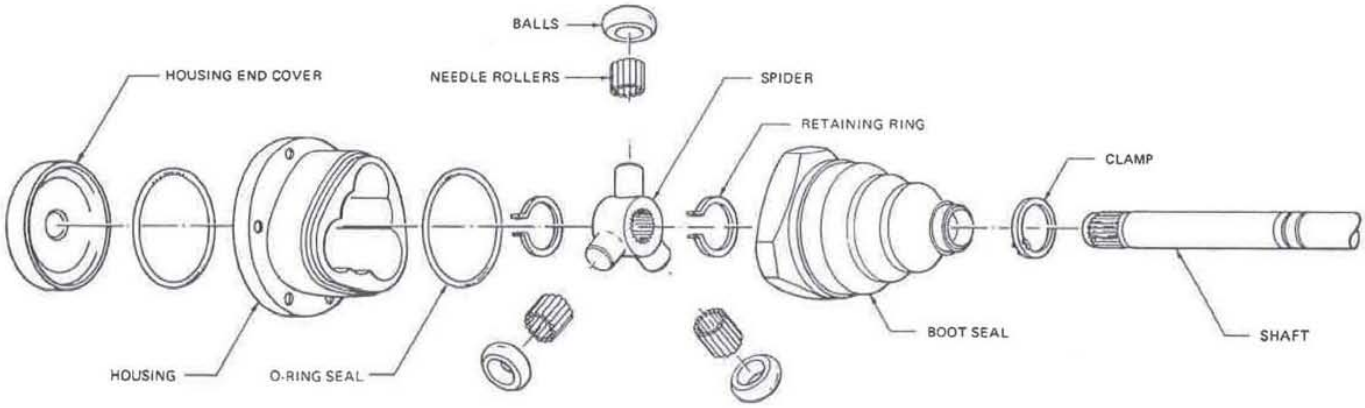


Fig. 8 — Tripot universal joint (end motion type)

angle and torque direction, the secondary couples of all joints are sensed in the same direction and located normal to the joint angle plane. In addition, like the ball and trunnion joint, the same internal geometry which provides for angular motion also allows simultaneous axial movement in the end motion tripot joint.

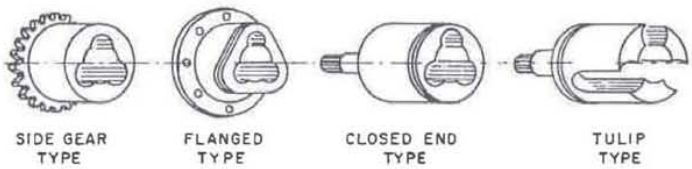


Fig. 9 — Typical tripot joint housing constructions

Design

The various components of a typical end motion type of tripot joint, shown in an exploded view in Fig. 8, must be designed to provide the necessary joint angularity, axial travel, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the housing, spider, needle roller, and ball configuration can be designed to meet the various joint application parameters.

Proper considerations for designing the joint bearing elements are necessary for adequate joint life. The ball-to-housing bore wear can be minimized by the selection of optimum comparable component materials, dimensions, Hertz contact stresses, surface hardnesses, and surface finishes or coatings, as well as an optimum hardness gradient of the housing bore case depth. In addition, an optimum lubricant must be utilized which is compatible with the overall ball-to-housing bore design selected, and also must meet the requirements of the joint application. The optimized design should provide for an adequate precession of the balls in the housing bores and of the needle rollers relative to the pin and the ball bore. This precession characteristic is also described in the section on the *Cardan or Hooke Universal Joint*.

The needle roller length-to-diameter ratio employed is usually in the upper half of the range, or somewhat higher than used in a typical Cardan joint. The needle roller bearing load distribution and Hertz contact stress computation procedure is discussed in the section on the *Cardan or Hooke Universal Joint*. For the entire bearing configuration, the permissible Hertz contact stress, including the associated computation procedure for the bearing configuration, is also shown in *Appendix A*. With regard to needle roller bearing design, the selection of optimum size and tolerances as well as diametral and circumferential clearances is generally the same as for a Cardan, or ball and trunnion joint in similar applications. *Appendix C* illustrates a typical bearing computation procedure for determining these clearances.

The housing is a member with three equally spaced, partly cylindrical axial ball bores on the cylindrical inner clearance surface, and with drivable means of attachment. In the tulip type housing, these ball bores have open circular segments on the periphery as well as on the inner clearance surface. Fig. 9 shows four typical housing constructions employed. The spider is a member with three equally spaced trunnions in the same plane and with internally splined drivable means of attachment. The spider is generally held in position on the splined shaft with one or two retaining rings.

The ball is a partly spherical-shaped member which pivots and transmits torque from the housing to the shaft through needle rollers and the spider, and permits axial movement. Instead of being true spherical, the outer profile surface of the ball also may be made toroidal in shape as discussed in the section on the *Ball and Trunnion Universal Joint*. In a typical design, the total clearance between the ball and housing ball bore may be in the range of approximately 0.001-0.005 in.

The use of needle rollers as rolling elements between the spider trunnion and ball provides for ease of angulation and end motion of the joint while transmitting torque. Due to centrifugal force effects during rotation, various retention means have been employed to maintain the needle rollers in their proper radial position between the ball and the spider trunnion. A thrust shoulder either in the ball bore at its outer end, and also at its inner end, if required, or on the spider trunnion outer and inner ends, will

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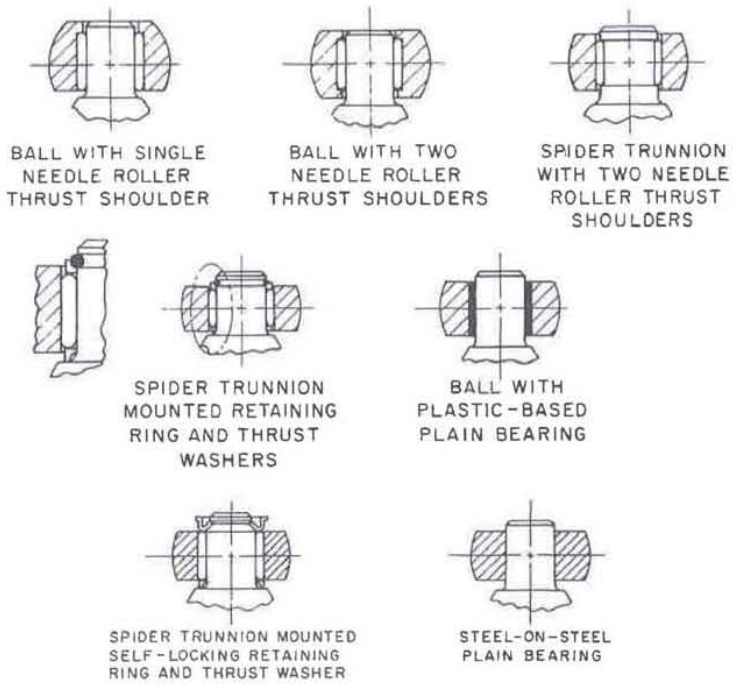


Fig. 10 — Typical tripot joint bearing constructions

provide this function. A self-locking type, or a somewhat more conventional type of retaining ring, mounted on the spider trunnion outer end with a thrust washer at one or both ends of the needle rollers, will also maintain them in their proper radial position. Fig. 10 shows these typical needle roller retention designs for various bearing constructions. Also shown is a bearing design which incorporates a low friction, plastic-based plain bearing in the ball bore in place of needle rollers, and has had limited usage. In addition, a plain bearing configuration is shown where the ball bore itself serves as the bearing and has been employed in some applications where the effects of higher frictional forces can be tolerated.

The boot seal is a flexible member, and in performing its principal function of keeping the lubricant in the foreign matter out of the joint, it must be capable of operating under varying environmental conditions at various speeds, joint angles, and end motion positions. Because of the various housing constructions used, the method of attachment of the boot seal to a particular housing is generally determined by its design configuration. When the flanged or closed end type of housing is used, the seal is usually connected to the housing with a seal adapter as shown in Fig. 1, and held in position with a seal retainer, clamp, or band. In addition, a spiral wire type of clamp or a garter spring have also been used as the retention means. An O-ring seal is generally used between the housing and seal adapter. The band is a ring-like metallic or nonmetallic member, while the clamp is a thin metallic or nonmetallic adjustable band. Usually, the band and clamp are metallic with the former magnetically formed in place. Also, in some flanged or closed end housing designs, the boot seal has been held directly in position on the housing with a band or clamp, without the aid of a seal adapter. Generally, a housing end cover closes the flanged housing, while sealing at this location is accomplished as required with an O-ring or gasket. When necessary, a vent hole can be provided in the cover if the axial stroke of the joint is relatively high. Without the vent, the large change in internal volume adversely affects seal life with certain boot designs.

When the tulip type of housing is employed, a housing cover must be used similar to the design shown for the roller and trunnion joint. This is a member which encloses the housing and provides for attachment of the boot seal. Usually, the housing cover is coined in position on the housing, and an O-ring provides for sealing between these two members. The basic difference between the housing covers used for the roller and trunnion and tripot joint tulip type housings is, that for the tripot joint the cover inside surface is not required to provide the radial supporting means for the joint. The boot seal is generally held in position with a seal retainer, band, conventional clamp, or a spiral spring wire type of clamp. Whenever the flanged end, closed end, or tulip type of housings are used, the seal is usually held in position on the shaft with various retention means such as a metallic band magnetically formed in place, a stretchable nonmetallic band, a clamp, or a garter spring. Also, in some instances no seal retention means are used at all at the shaft when joint venting is necessary due to a relatively high axial stroke.

In some applications the housing has also been made integral with the differential side gear. The side gear type of housing construction can be designed so that the differential lubricating oil also serves as the joint lubricant and the boot seal does not rotate. Since the boot seal is a nonrotating design, a bearing and a lip-type oil seal must be incorporated for the rotating shaft. The boot seal is held in position on the differential housing side with a

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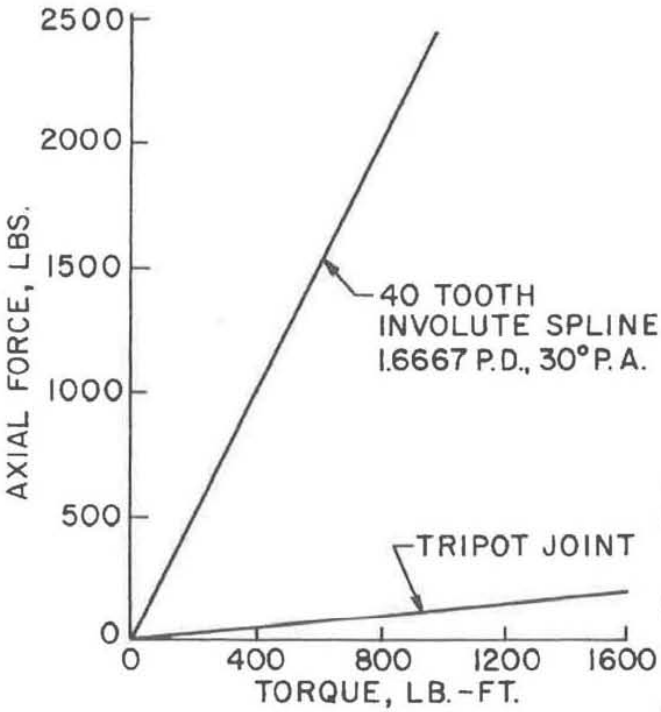


Fig. 11 — Comparison of typical axial force characteristics

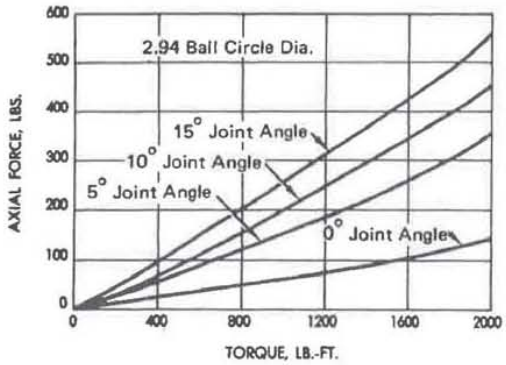


Fig. 12 — Tripot joint axial force characteristics

seal retainer, small bolts, and lock washers, as shown for the sliding block and trunnion joint.

The side gear type of housing construction can also be designed to use conventional joint lubricant. A plug at the inner end of the side gear housing and a lip-type oil seal located between the differential housing and the side gear housing outer surface provide the necessary sealing means to prevent the differential lubricating oil from entering the joint, and keep it in the differential. In order to employ a rotating boot seal, the side gear housing must be long enough to allow attachment of the seal. This seal is generally held in position on the housing and shaft with clamps.

To prevent accidental disassembly of the joint, the maximum axial stroke can be limited by an internal stop feature. Some of these typical stop means which have been employed are the seal adapter shown in Fig. 1, and commonly used with flanged or closed type housings, and the housing cover which is utilized with the tulip type housing. When the side gear type housing is used with a nonrotating boot seal configuration, the seal and seal retainer can be designed to limit the axial travel. When the boot seal is mounted directly on a housing, the seal itself may control the maximum axial movement. The maximum angular design capability of the joint for its entire axial stroke range can be determined by plotting a chart as described in *Appendix G*.

As previously indicated, a principal advantage of the end motion type of tripot joint is that it requires a relatively low axial thrust force when transmitting torque during end movement, when compared with conventional sliding splines or most other end motion type joints. This is illustrated in the typical curves shown in Fig. 11 where a tripot joint is compared with an involute spline under static (nonrotating) conditions. Fig. 12 shows the static (nonrotating) axial force characteristics at various joint angles for a typical joint.

Material Selection

The selection of optimum materials for the end motion type of tripot joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The housing is generally made from a plain medium carbon steel forging or formed blank. It is induction hardened to an adequate case depth and a typical R_c60 minimum surface hardness in the ball bores, central clearance bore, and other selected areas to provide the required wear characteristics and torsional strength. The ball bores may be finish machined prior to induction hardening, or they may be finish ground after induction hardening. The case depth in the ball bores must be sufficient to meet the contact load requirement explained in *Appendix A*. This case depth is relatively deeper when a toroidal ball is used than when a spherical ball is utilized. Also, the housing can be made from a pearlitic nodular iron casting and require essentially the same processing procedure described for the plain medium carbon steel housing. In addition, the housing can be made from a low carbon alloy steel forging or formed blank, which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to meet the functional and durability requirements. The ball bores are usually finish ground after carburizing and hardening.

The spider is usually made from a low carbon alloy steel forging which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. The trunnions are then finish ground. The necessary case depth of the spider trunnion is determined by the joint size and its application requirement, and is essentially the same as discussed for the cross trunnion in the section on the *Cardan or Hooke Universal Joint* and is also explained in *Appendix A*. When using a plastic-based plain bearing in the ball bore instead of needle rollers, it is possible to use a plain medium carbon steel for the spider, which is induction hardened to an adequate case depth and surface hardness on the

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trunnions, to provide the required wear characteristics and bending strength. The trunnions are then finish ground.

The balls are generally made of chromium steel which are hardened to a typical range of R_c62-65 and finish ground or finish lapped. The balls can also be made of low carbon alloy steel which are carburized to an adequate case depth and surface hardened to a typical R_c60 minimum before finish grinding. In addition to providing for an adequate surface hardness and case depth in the housing ball bores, the bore wear can sometimes be reduced by modifying the ball surface finish by, for example, lapping after finish grinding. Also, a phosphate type coating can be applied to the housing if desired. The needle rollers are made of chromium steel which are hardened to a typical range of R_c60-64 and finish lapped. The retaining rings are usually made of carbon spring steel.

The elastomeric material of the boot seal is generally a synthetic rubber which must exhibit specific resistance to the action of petroleum-based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The seal adapter, seal retainer, housing end cover, housing cover, and clamps are usually made of plain low carbon steel. The metallic bands are generally aluminum or copper. The nonmetallic bands are usually synthetic rubber with thin rectangular sectioned bands located at the housing end and with thick rectangular or other geometric sectioned bands employed at the shaft end. The garter springs are generally made from high quality spring steel wire, such as typical music wire, while the spiral spring wire clamps are usually made from conventional spring steel wire. The O-ring seal material is generally a synthetic rubber, while the gasket is usually made from gasket paper. When the nonrotating boot seal is used, the bearing operating on the rotating shaft can be made of various typical plain bearing materials and the oil seal is generally of the elastomeric lip type.

Lubrication

The end motion type of tripot joint has a relatively large capacity for lubricant and is generally lubricated for life with an NLGI grade number 1 or 2 lubricating grease. A typical lubricant of this type usually consists of a refined mineral oil and lithium or calcium 12 hydroxy stearate soap. When the joint operates within the differential, it may be lubricated if desired, with the same lubricating mineral oil used in the differential which is generally of SAE viscosity number 90. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

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

Rating Procedure

The end motion type of tripot joint can be rated as described in *Appendix I*.

Applications

The end motion type of tripot joint has found wide usage in axle driveshafts of front and rear drive vehicles where there has been a need for a constant velocity joint providing a relatively large axial movement capability. Depending on the axial stroke requirement of an application, the maximum angular design capability of this joint is approximately 25 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle throughout its entire axial stroke range must be within the limit lines of the angular capability chart for a given joint design as described in *Appendix G*. For relatively large passenger car applications, normal operating angles as high as approximately 4 deg have been used, while normal operating angles as high as approximately 7 deg have been employed in relatively small passenger car applications. It has been found that if generally low operating angles are maintained, objectionable shudder-type of disturbances are not created.

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Section 3.2.7
Tripot Universal Joint
(Fixed Center Type)

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

Introduction

The fixed center type of tripot universal joint was patented by Orain in the early 1960s and patents on various applicable features were issued through the early 1970s. Its first known vehicle usage began in the late 1960s as the outboard joint in a French front wheel drive passenger car. Since that time, it has been used in other front drive vehicles.

The tripot universal joint (fixed center type) shown in Fig. 1 in a typical design, is a self-supported constant velocity universal joint, which consists of a housing drivably connected to a forked shaft through three equally spaced trunnion mounted balls and retained as an assembly by a retaining clip. The joint center is maintained in a substantially fixed axial location by means of a spring loaded centering device.

The principal advantage of this joint is its ability to transmit constant velocity with a relatively high joint angle capability. A principal disadvantage of this joint is its orbiting characteristic about the joint center when operating at an angle. This characteristic is directly related to joint size and operating angle. This orbital path moves with constant velocity at three times per revolution of the joint and in the same direction. Therefore, this joint is generally limited to relatively low speed applications and has to be used in conjunction with another universal joint or self-aligning bearing in the drive system. Since the orbiting motion produces inertia effects, orbiting of the relatively large and heavy housing would be undesirable in comparison with a similar motion by the smaller and lighter shaft member. Hence, the joint should always be oriented in a driveshaft layout so that the housing is the supported member while the shaft is free to orbit as required. Another limitation of this joint is that only three balls are available to transmit the torque load, while for similar applications, the typical Rzeppa joint, for example, has six drive balls sharing in the transmission of torque.

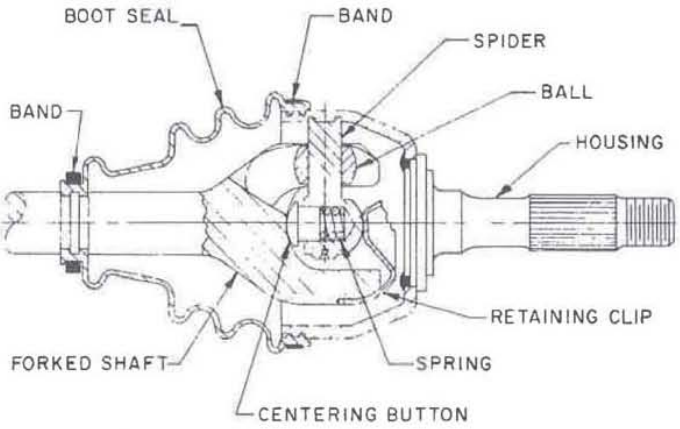


Fig. 1 — Tripot universal joint (fixed center type)

Kinematics

The fixed center type of tripot joint transmits true constant velocity when friction is ignored, and like the end motion type of tripot joint, it is a distinct ball type constant velocity joint in that the transmission of motion and force is not through the constant velocity or homokinetic plane, but through a plane which is always normal to the rotational axis of one of the members. This member can be either the driving or driven member. All other ball type constant velocity joints utilize the principal of maintaining driving contact through the constant velocity or bisecting angle plane. In the section on the *Tripot Universal Joint (End Motion Type)* the kinematic analysis which also applies to the fixed center type of tripot joint is described in detail.

A distinct peculiarity of both of these tripot joints is the orbiting characteristics when operating at an angle. When the typical wheel drive application is assumed where the housing is fully supported and also the driven member, the shaft or driving member must orbit when the joint rotates. With respect to the shaft end view, this path is an ellipse having a major axis radius of:

$$r_0 = \frac{D\,(1 - \cos\theta)}{4\cos\theta}$$

normal to the plane of the joint angle, and a minor axis radius of:

$$r_0' = \frac{D\,(1 - \cos\theta)}{4}$$

in the plane of the joint angle.
where:

D = Shaft fork end bore circle diameter
θ = Joint angle

The forked shaft center at the joint geometric center rotates in the direction of joint rotation at a constant angular velocity of three times the joint angular velocity.

When transmitting torque at an angle, a secondary couple is produced on both driving and driven members of the joint. As in the end motion type of tripot joint, the couple forces react as static nonvibrating forces only on the bearing supports if inertia and

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friction are ignored. The secondary couples are a function of the torque and joint angle only. These couples are of the same magnitude on both driving and driven members and are normal to the joint angle plane. For a given torque direction and disposition of the joint angle, both couples are sensed in the same direction. The values of these couples are the same as for the end motion type of tripot joint and are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

where:

C = Secondary couple on driving member
C_1 = Secondary couple on driven member
T = Torque
θ = Joint angle

Design

The various components of a typical fixed center type of tripot joint, shown in a disassembled view in Fig. 2, must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the housing configuration including the spider and balls, as well as the forked end of the shaft, can be designed to meet the various joint application parameters. Proper design procedures of the joint bearing configuration are necessary for adequate joint life as described in the section on the *Tripot Universal Joint (End Motion Type)*. The permissible Hertz contact stress, including the associated computation procedure for the bearing configuration, is shown in *Appendix A*.

The housing is a bell-shaped member retaining a spider with balls rotatably and slidably mounted on three equally spaced trunnions located in the same plane and with drivable means of attachment. The balls are partly spherical-shaped members which pivot and transmit torque from the housing to the shaft through the spider. The forked shaft is a member with tulip-shaped prongs incorporating three equally spaced partly cylindrical axial ball bores and with drivable means of attachment.

The retaining clip is a three pronged spider-like member which retains the universal joint as an assembly. The partial cut open view in Fig. 3 illustrates a typical manner in which the end lugs on the retaining clip prongs secure the forked shaft to the housing as an assembly by means of a retention feature located on the tulip-shaped prongs. The centering button is a thrust loaded member which locates the joint center. Its spherical-shaped surface contacts the inside surface of the forked shaft. The stem of the button is piloted in a hole located in the spider body. The spring located in this hole may be either mechanical or elastomeric in design and preloads the button against the forked shaft causing the joint to have a substantially fixed center of angulation.

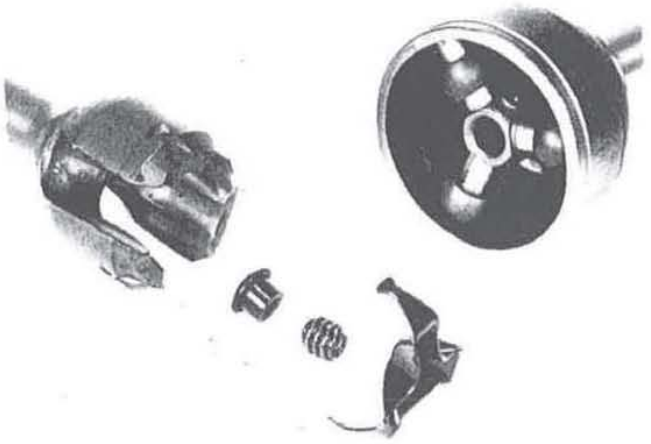


Fig. 2 — Tripot universal joint (fixed center type)



Fig. 3 — Typical assembly retention feature of fixed center tripot joint

The boot seal is a flexible member and in performing its principal function of keeping the lubricant in and foreign matter out of the joint, it must be capable of operating under varying environmental conditions at various speeds and joint angles. The seal is generally held in position on the housing with a metallic band magnetically formed in place or with a spiral spring wire clamp, while a stretchable nonmetallic band usually holds it in position on the shaft.

Material Selection

The selection of optimum materials for the fixed center type of tripot joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The housing is generally a fabricated assembly consisting of a short shaft member welded to a formed housing member. The shaft member is usually made of plain medium carbon steel, which is induction hardened to an adequate case depth and surface hardness to provide the required torsional strength. The formed housing member is made of plain low carbon steel.

The spider is generally made from a low carbon alloy steel forging which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the necessary

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strength and wear characteristics. The trunnions are then finish ground.

The balls are usually made of chromium steel which are hardened to a typical range of R_c62-65 and finish ground. The forked shaft is generally made of a low carbon alloy steel. After machining and finish forming the three equally spaced partly cylindrical axial ball bores, the forked shaft is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. The case depth in the ball bores must be sufficient to meet the contact load requirement explained in *Appendix A*.

The retaining clip is generally made of carbon spring steel. The centering button is usually made of low carbon alloy steel which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum. The spring is generally made from either carbon spring steel wire or of synthetic rubber.

The elastomeric material of the boot seal is usually a synthetic rubber, which must exhibit specific resistance to the action of petroleum based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The metallic band holding the seal to the housing is generally aluminum; while the nonmetallic band holding the seal to the shaft is usually a thick rectangular or other geometric sectioned synthetic rubber. The spiral spring wire clamp is generally made from spring steel wire.

Lubrication

The fixed center type of tripot joint is generally lubricated for life with an NLGI grade number 1 or 2 lubricating grease. A typical lubricant of this type usually consists of a refined mineral oil and calcium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

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

Rating Procedure

The fixed center type of tripot joint can be rated as described in *Appendix I*.

Applications

The fixed center type of tripot joint has found usage as the outboard joint in axle driveshafts of front drive vehicles where there has been a need for a constant velocity joint providing a relatively high joint angle capability. The maximum angular design capability of this joint is approximately 45 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle must be less than the maximum angular capability for a given joint design. For passenger car applications, normal operating angles as high as approximately 5 deg have been used in independent wheel suspension axle driveshafts.

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Section 3.2.8
Rzeppa Universal Joint

F. F. Miller — Spicer Universal Joint Div., Dana Corp.
D. W. Holzinger — Spicer Universal Joint Div., Dana Corp.
E. R. Wagner, P.E. — Saginaw Steering Gear Div., GMC

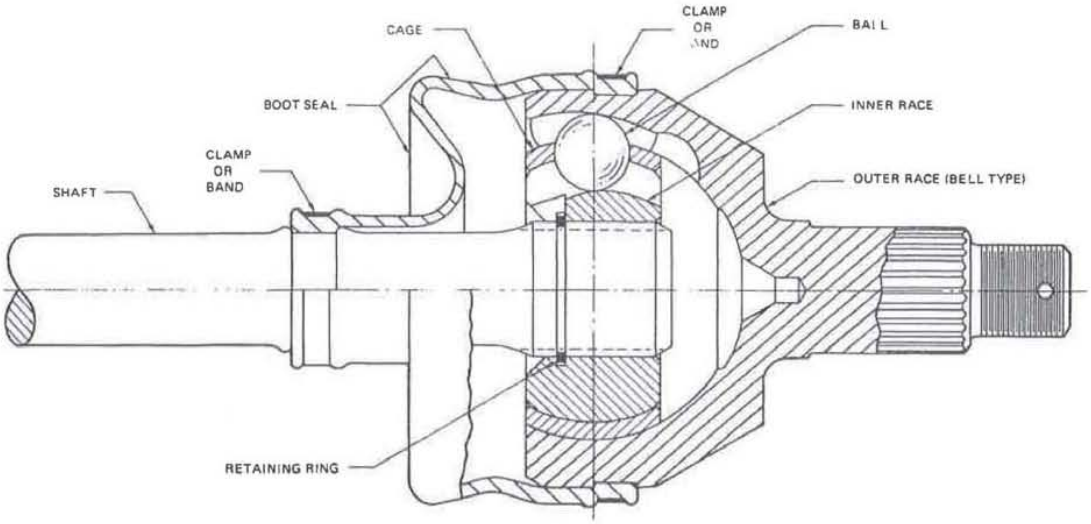


Fig. 1 — Rzeppa universal joint (bell type)

Introduction

The development of the Rzeppa universal joint is shown in the patent literature through a series of patents issued to Rzeppa from the mid 1920s to the mid 1930s. Since that time, additional patents have been granted to others on further improvements. Its first known vehicle usage began in 1936 as the outboard and inboard joints in an early American front wheel drive passenger car. Since that time, it has found wide acceptance throughout the world as the most commonly used outboard joint in front drive vehicles. It has also been used in off-highway equipment, aircraft, marine, industrial, and stationary drive system installations.

The Rzeppa universal joint, shown in Fig. 1 in a typical design, is a self-supported constant velocity universal joint which consists of an outer and inner race drivably connected through balls positioned in the constant velocity plane by axially offset meridionally curved grooves, and maintained in this plane by a cage located between the two races.

A principal advantage of this joint is its ability to transmit constant velocity at a relatively high joint angle when the bell type outer race construction is used. Also, it is rugged and compact and has a relatively high torque capacity for a given swing diameter. In addition, the joint has the thrust capability to withstand relatively high externally imposed axial forces of an intermittent nature. These axial forces are commonly produced by a conventional sliding spline or end motion type joints when shaft length changes are required during vehicle suspension movements.

Kinematics

As previously indicated, the Rzeppa joint transmits true constant velocity. The drive balls are positioned in the half angle or constant velocity plane by the action of the axially offset meridionally curved grooves of the outer and inner races. These grooves are generated from centers symmetrically located about the joint center. This action is illustrated in Fig. 2 where the joint is shown in an angulated position. The balls are maintained in the constant velocity plane by a cage.

Fig. 3 schematically shows that all balls are located in a common plane Z-Z and have their theoretical centers at an equal distance from the driving and driven member axes X-X and Y-Y, respectively, and thus the transmission of constant velocity is effected. Fig. 4 schematically illustrates further the principle shown in Fig. 2 where the balls are positioned in the constant velocity plane by the groove camming action when the joint is angulated to any joint angle θ. When line X is positioned to any angle θ about joint center C, point A, which represents the ball groove generation center for the inner race, moves to A′, forming the triangle ABA′, which is exactly one-half of the joint angle ACA′ since lines AC and CB are equal. Therefore, the plane of the balls is in the bisecting angle or constant velocity plane.

It is of interest to note that all of the balls in the joint, when it is displaced at an angle, are not at the same distance from the joint center C as shown in Fig. 2. With the outer race groove radius generated from the horizontal outer race axis, the upper ball will

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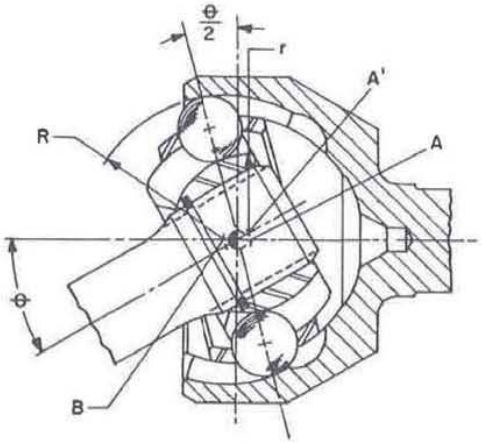


Fig. 2 — Rzeppa universal joint in angulated position

be at a greater outward radius from the outer race centerline than the ball shown in the opposite lower groove. A section taken through the balls at the position shown in Fig. 2 indicates that these balls lie on an ellipse, and that the upper three balls are farther from the joint center than the lower three balls, as shown in Fig. 5. The off-center condition exists because the groove generating point is displaced at a distance from the joint center C in order to effect the camming action necessary for proper ball location.

The location of center D of the ellipse major axis M-M is a function of the joint angle. In Fig. 4, point D will be displaced one-half the distance that the groove generating center will displace for any given joint angle. The ellipse minor axis N-N is the ball circle diameter at zero joint angle. Since the grooves are generated on meridional lines, movement along the grooves will bring the balls closer or farther apart circumferentially, and provision for this movement has to be provided in the cage by elongating the cage windows, as shown in Fig. 6.

When transmitting torque at an angle, a secondary couple is produced on both driving and driven members of the joint. Since driving contact is maintained through the constant velocity or bisecting angle plane, the couple forces react as static nonvibrating forces only on the bearing supports. The secondary couples are a function of the torque and joint angle only. These couples are of the same magnitude on both driving and driven members, and they act in a direction normal to the joint angle plane. For a given torque direction and disposition of the joint angle, both couples are sensed in the same direction. The values of these secondary couples are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

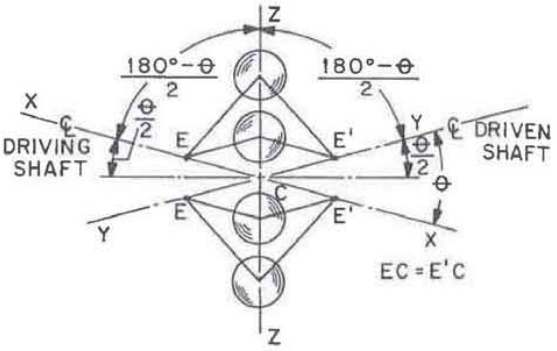


Fig. 3 — Diagram illustrating principle of transmission of constant velocity

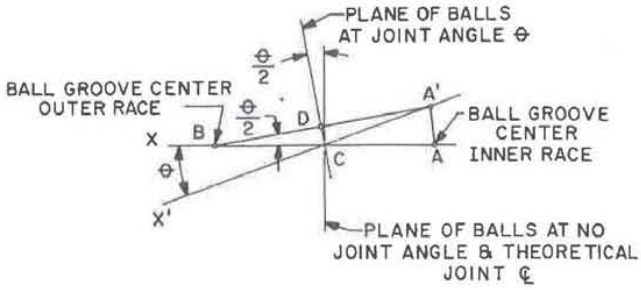


Fig. 4 — Diagram illustrating positioning of balls in bisecting angle plane

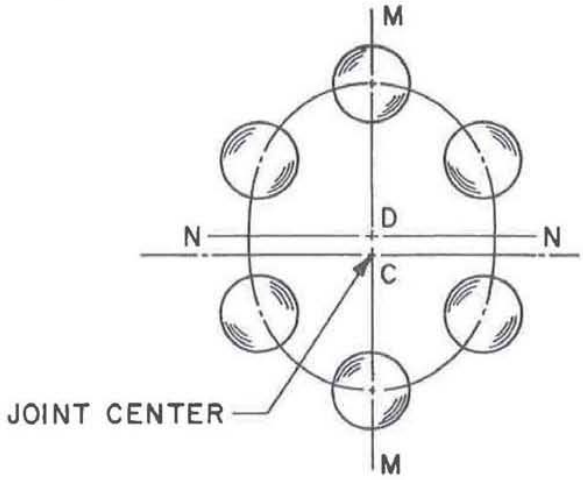


Fig. 5 — Section through plane of balls with joint in angulated position

where:

C = Secondary couple on driving member
C_1 = Secondary couple on driven member
T = Torque
θ = Joint angle

The solution for the values of these secondary couples is shown in *Appendix H* for a simplified case of a ball type constant velocity joint in which driving contact is maintained at all times in the constant velocity plane.

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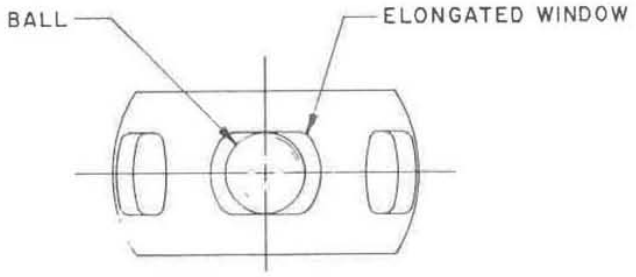


Fig. 6 — Elongated cage windows

Design

The various components of a typical Rzeppa joint, shown in an exploded view in Fig. 7, must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the outer and inner race and cage configuration with an optimized ball size and ball circle diameter can be designed to meet the various joint application parameters.

The outer race is either of the bell or disc type construction. Fig. 7 shows a typical bell type outer race, while Fig. 8 shows a Rzeppa joint with a typical disc type outer race. The bell type outer race is a bell-shaped member with axially offset, meridion-

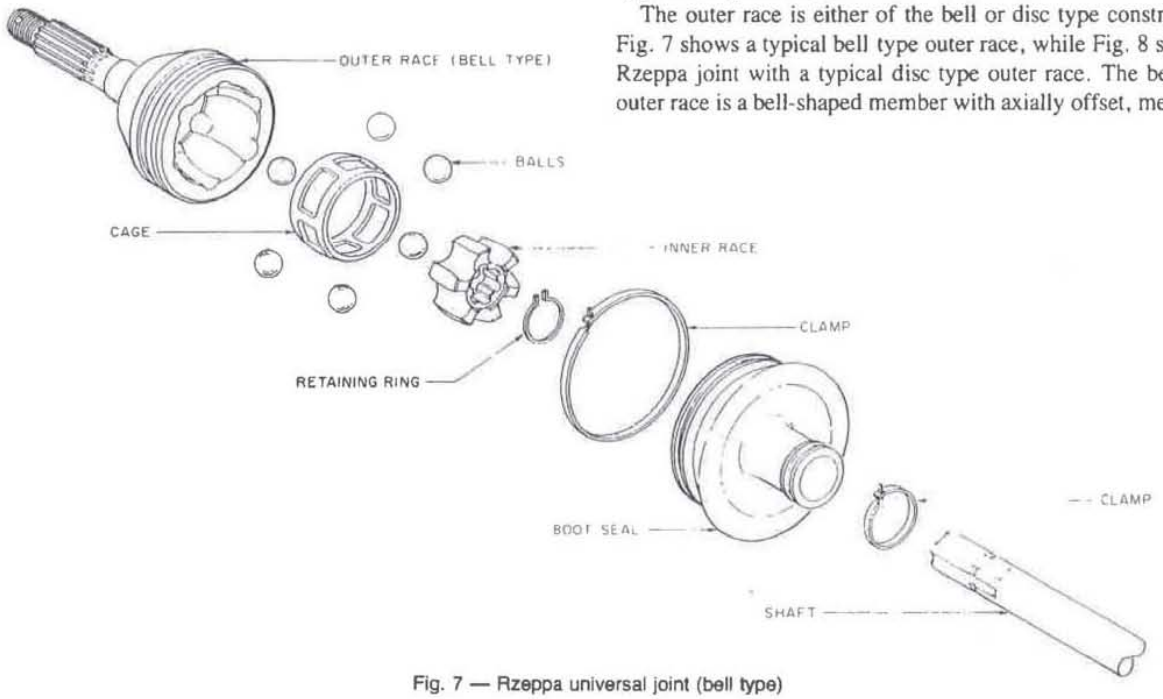


Fig. 7 — Rzeppa universal joint (bell type)

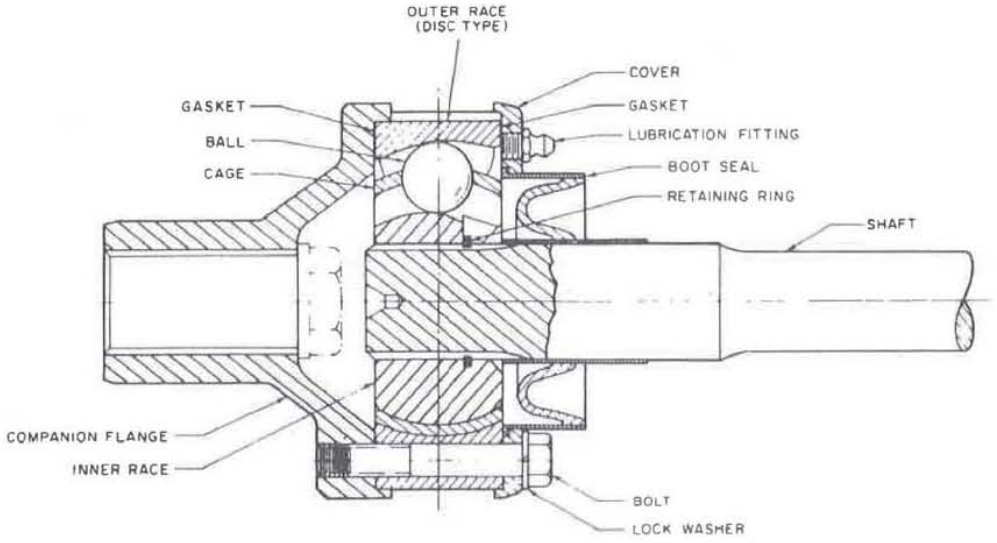
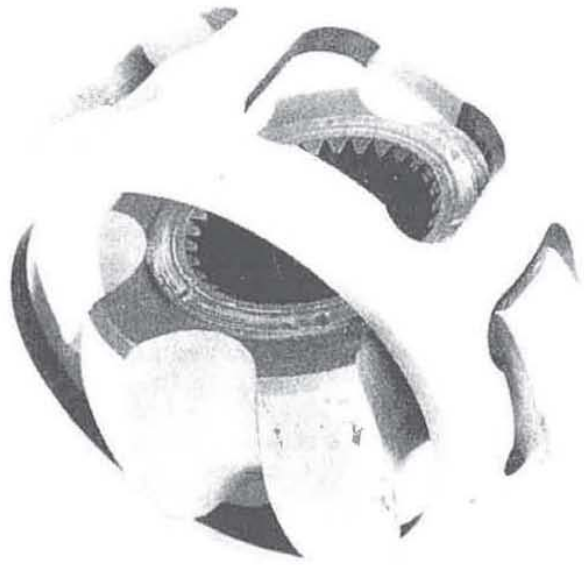


Fig. 8 — Rzeppa universal joint (disc type)

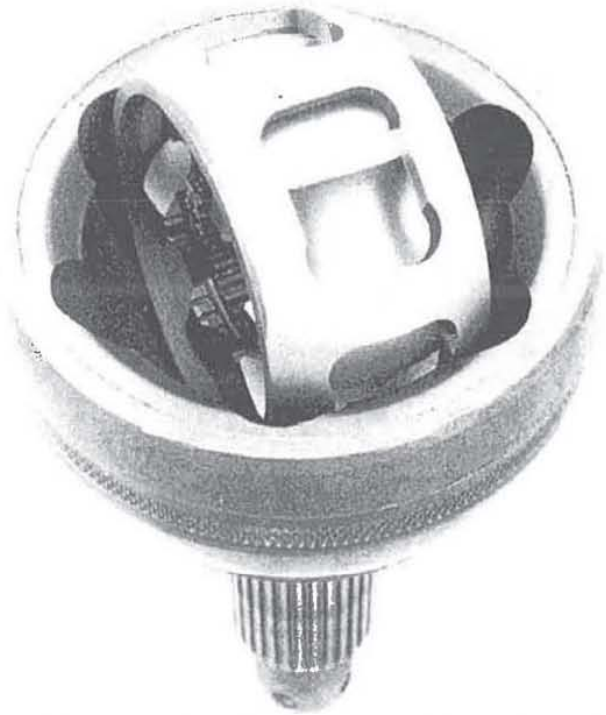
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A

Fig. 9 — Assembly procedure

B



C



ally curved ball grooves on the partly spherical inner bearing surface and with drivable means of attachment. The disc-type outer race is an annular member with axially offset, meridionally curved ball grooves on the partly spherical inner bearing surface and with drivable means of attachment. The bell type outer race construction allows a substantially higher joint angle capability than the disc type construction. The bell type outer race design can provide a maximum joint angle capability of approximately 45 deg, while the disc-type outer race permits approximately 18 deg of maximum angular capability.

The inner race is an annular member with axially offset meridionally curved ball grooves on the partly spherical outer bearing surface, and with internally splined drivable means of attachment. The inner race is generally held in position on the shaft spline with a retaining ring or rings. The cage is a ring-like member with concentric outer and inner, partly spherical bearing surfaces, and with a circumferential series of openings or windows for maintaining the balls in a common plane. A minimum number of three balls can be used in a joint, however, the usual number used is six. These balls are the intermediate drive members of the joint and are assembled with an interference fit between the cage window flats to prevent undesirable ball noise when substantial torque must be transmitted.

Proper proportioning of the outer and inner race and cage components is required so that these members can be assembled. Fig. 9 shows the required assembly sequence for the joint. In order to assemble the cage on the inner race, as shown in Fig. 9(a), it is generally necessary to either have two opposing cage window openings longer than the inner race width, or it may be possible to require only one longer window opening. Also, it is possible to have one lobe or all lobes on the inner race, shorter than the cage windows. Fig. 9(b) shows the inner race and cage

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being inserted into the outer race. The ball size is important not only for the necessary torque-carrying capability of the joint; but it also determines the size of the clearance openings in the cage and outer race for ease of assembly. Since opposite openings are generally required for assembly, six balls are usually used for the joint. Fig. 9(c) shows the balls, as well as the cage and inner race, being assembled into the joint. The length of the ball grooves in the outer race must be long enough not only for the maximum joint angle, but also provide for the additional angular swing allowance required for assembly of all the balls.

The functional and wear life characteristics of the joint are determined to a significant degree by the ball groove geometry. The axial offset location of the ball groove generating point is important for a smoother angular movement of the joint when it is not rotating. Since the axially offset ball grooves act as cam surfaces, frictional locking between the balls and grooves can occur unless the cam surfaces have a sufficient divergence or funnel angle to prevent this. It has been found that an included groove funnel angle of approximately 15-17 deg is required for smooth angulation without encountering friction locking when passing thru 0 deg or when starting from the 0 deg joint angle position. During joint rotation, with roll imposed on the balls, smooth joint action can be obtained with approximately one-half of the static funnel angle requirement. This reduced funnel angle is adequate, for example, in an application where the joint operates at a relatively high fixed joint angle or angular range and does not pass thru 0 deg. If the funnel angle is made larger than necessary for smooth operation of the joint, the forces exerted by the balls on the cage, and in turn on the outer and inner race spherical bearing surfaces, will become excessive. Fig. 10 illustrates how the axially offset ball groove generating points produce the divergence or funnel angle, which is a function of the ball circle radius and the offset generating points for both the outer and inner race ball grooves. These ball groove generating points are equally spaced from the joint center.

The two ball groove cross sections in common use are the circular arc and elliptical arc configurations shown in Fig. 11. For the circular arc groove form, the groove radius-to-ball diameter ratio is approximately 0.501, while for the elliptical arc groove form this ratio is a typical 0.510. This ratio is also called the conformity ratio of the grooves. The contact or pressure angle ranges from 30-45 deg for the elliptic arc groove form, with 45 deg being commonly used. For a given load, this contact angle, together with the groove conformity selected, control the size and shape of the contact areas of the balls with their mating grooves. Hence, the use of the elliptic arc groove form permits the placement of the ball-to-groove contact point away from the groove edge. For the elliptic arc groove configuration, the balls are loaded in compression while in the circular arc groove configuration, the balls are subjected to a shear-type loading. The ratio of ball rolling-to-sliding motion, which is also a characteristic of angular contact bearings, can be controlled by the selection of the optimum contact angle and conformity ratio. The permissible Hertz contact stress in the ball grooves, including the associated computation procedure, is shown in *Appendix A*.

The sealing means employed in a particular joint depends primarily on the application and environment. When the joint operates in the open, a boot seal or other type of seal can be used.

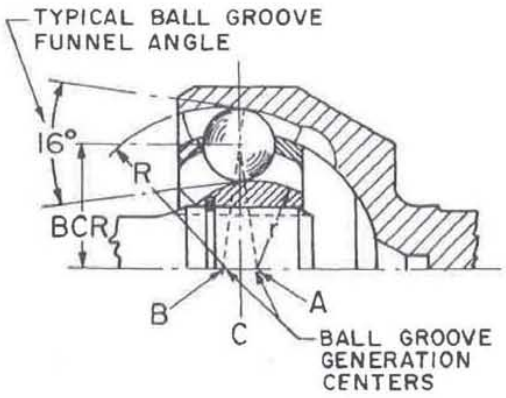


Fig. 10 — Diagram illustrating ball groove funnel angle

The boot seal, which is a flexible member, is most commonly used, and in performing its principal function of keeping the lubricant in and foreign matter out of the joint, it must be capable of operating under varying environmental conditions at various speeds and joint angles. The seal is generally held in position on the outer race and shaft with clamps which are thin metallic or nonmetallic adjustable bands. Bands, which are ring-like metallic or nonmetallic members, have also been used to retain the seal ends. Usually, the clamps and bands are metallic and the latter are either crimped or magnetically formed in place.

Material Selection

The selection of optimum materials for the Rzeppa joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The outer race is generally made from a low carbon alloy steel forging or formed blank. It is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. The ball grooves, spherical inner surface, and other required surfaces are then finish ground. The case depth in the ball grooves must be

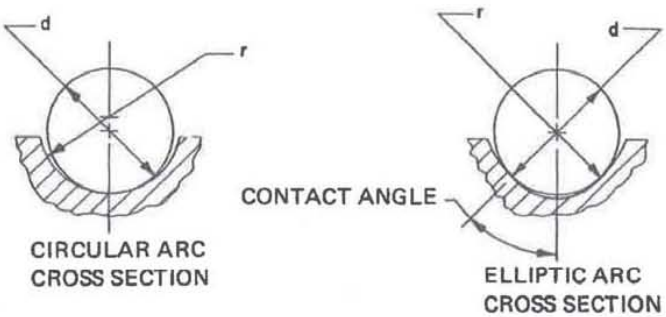


Fig. 11 — Typical ball groove forms

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sufficient to meet the contact load requirement explained in *Appendix A*. The outer race can also be made of medium carbon alloy or plain medium carbon steel, induction hardened to an adequate case depth and a typical R_c60 minimum surface hardness in selected areas before finish grinding to meet the functional and durability requirements. For a bell type outer race, the drivable means of attachment consists usually of a stub shaft which has a threaded end in conjunction with spline. This threaded end is generally tempered to a selected hardness to prevent brittleness and improve toughness.

The inner race is usually made of low carbon alloy steel. After broaching the spline and before finish grinding the ball grooves and spherical outer surface, the inner race is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. As in the case of the outer race, the case depth in the ball grooves must be sufficient to meet the contact load requirement.

The cage is generally made of low carbon alloy steel which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the necessary wear characteristics and strength. The spherical outer and inner surfaces and the window flats are then finish ground.

The balls are made of chromium steel which are hardened to a typical range of R_c62-65 and finish lapped. The retaining ring or rings are usually made of carbon spring steel.

The elastomeric material of the boot seal is generally a synthetic rubber which must exhibit specific resistance to the action of petroleum-based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The clamps are usually made of plain low carbon or stainless steel, while the bands are generally made of either aluminum or copper.

Lubrication

The Rzeppa joint is generally lubricated for life, and the type of lubricant used is dependent on the application and environment as well as the sealing means employed. For axle driveshaft applications where a conventional boot seal is employed, a lubricating grease having an NLGI grade number 1 is usually used. A typical lubricant of this type generally consists of a uniform blend of refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant. Molybdenum disulfide is a common extreme pressure additive. For closed steering knuckle constructions incorporating a final drive, such as a typical planetary gear reduction with the Rzeppa joint rotating at approximately three to five times wheel speed, or for conventional closed steering knuckle designs incorporating direct drive, a typical lubricating mineral oil of SAE viscosity number 90 or 140 is recommended.

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

Rating Procedure

The Rzeppa joint can be rated as described in *Appendix I*.

Applications

The bell type of Rzeppa joint has found its greatest usage as the outboard joint in axle driveshafts of front drive vehicles where there has been a need for a constant velocity joint providing a relatively high joint angle capability. The maximum angular design capability of this joint is approximately 45 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle must be less than the maximum angular capability for a given joint design. For live rigid front axle applications, normal operating angles have ranged from 0-1½ deg, while the maximum intermittent operating angle has been limited generally by the steering knuckle design.

For independent wheel suspension axle driveshafts, such as employed in passenger car applications, normal operating angles as high as approximately 6 deg have been used; while the maximum intermittent operating angle during suspension and steering movements must not exceed the maximum angular capability of the joint.

The disc type of Rzeppa joint has found limited usage as the inboard joint of vehicle axle driveshafts since end motion capability must be provided by the inner race spline or by some other telescoping means. Major uses of this joint, however, have been in aircraft, marine, and industrial applications, as well as power take-off drives.

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Section 3.2.9
Ball Spline Rzeppa Universal Joint

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

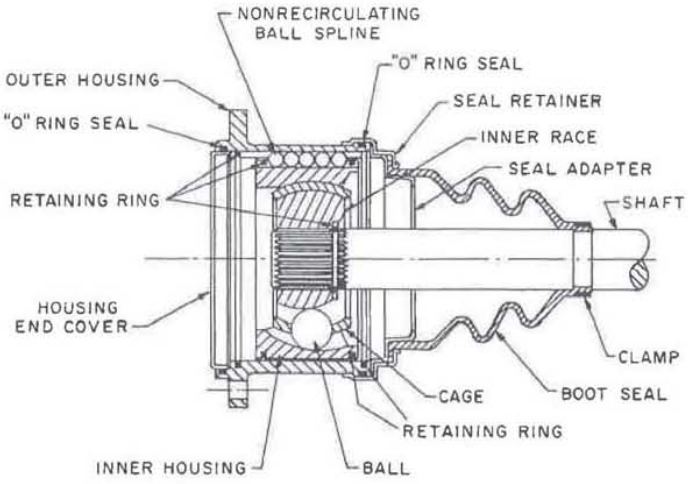


Fig. 1 — Ball spline Rzeppa universal joint

Introduction

The ball spline Rzeppa universal joint was developed in Europe where its first known vehicle usage began in 1960 as the inboard joint in an Italian front wheel drive passenger car. In the mid-1960s it was also similarly used briefly in an American front drive car.

The ball spline Rzeppa universal joint, shown in Fig. 1 in a typical design, is a disc type Rzeppa universal joint which permits axial movement by means of an integral ball spline. Basically, this joint differs from other end motion joints in that the same internal geometry providing for angular motion does not also allow axial motion.

The principal advantage of this joint is its ability to transmit constant velocity and require a relatively low axial thrust force during end movement when compared with conventional sliding splines or most other end motion type joints. The major disadvantage of this joint is that its manufacturing cost is relatively high. A further limitation is that its axial motion capability is relatively small when compared with most other end motion type joints.

Kinematics

The kinematics of the ball spline Rzeppa joint are the same as for the Rzeppa joint, which transmits true constant velocity. The internal geometry which provides for angular motion, however, does not allow axial movement and, therefore, this motion must be accommodated by a separate ball spline device.

Design

The various components of a typical ball spline Rzeppa joint, shown in an exploded view in Fig. 2, must be designed to provide the necessary joint angularity, axial travel, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the disc type Rzeppa joint configuration can be designed to meet the various joint application parameters described in the section on the *Rzeppa Universal Joint*, except that a ball spline mechanism surrounding the joint itself must be incorporated in the construction.

Generally, the ball spline is of the nonrecirculating rather than the recirculating type, because of its simplicity. It requires less space and can be manufactured more economically. The principal components of this ball spline are an outer housing, inner housing, balls as the rolling elements, retaining rings, and sealing means. The outer housing is an annular member with axial ball spline grooves on the cylindrical inner surface and with drivable means of attachment. The inner housing is a disc type Rzeppa outer race with axial ball spline grooves on the cylindrical outer surface. Because of the generally large ball circle diameter of the balls, the number of balls required is usually not dictated by the necessary load carrying ability, but by the desire to have stability in the rolling element spline mechanism. Due to the inherent design characteristics of the joint and nonrecirculating ball spline configuration, the amount of pure rolling end motion of the ball

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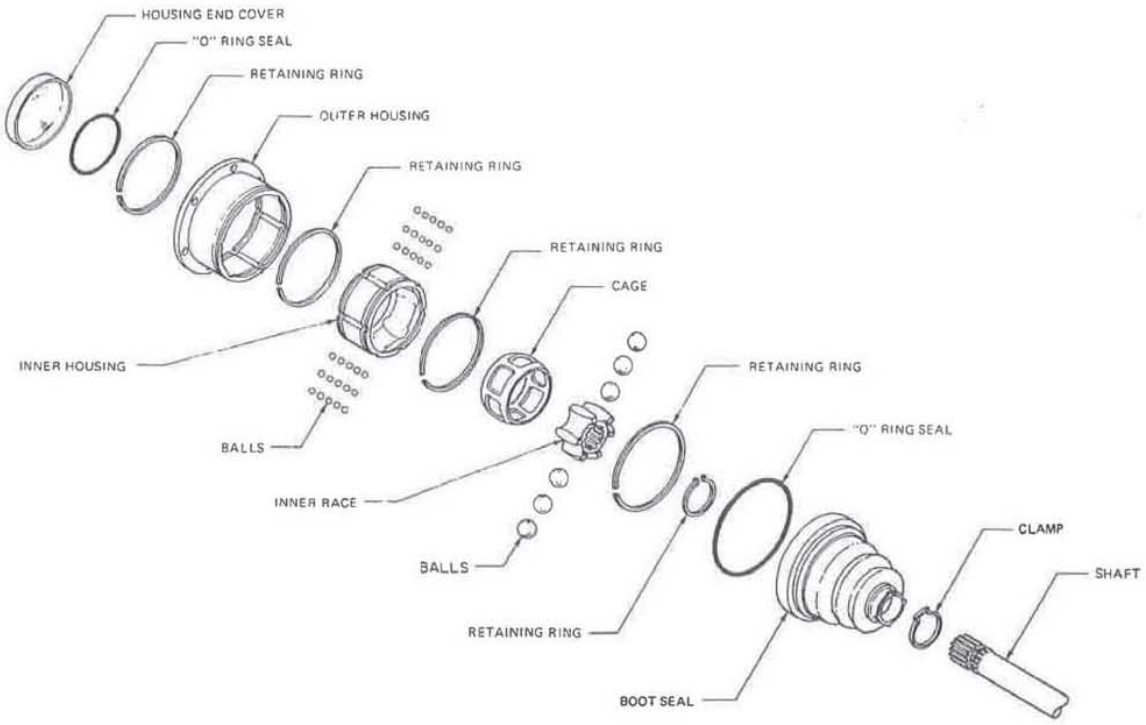


Fig. 2 — Ball spline Rzeppa universal joint

spline is generally limited. Usually, retaining rings located on the cylindrical inner surface of the outer housing and on the cylindrical outer surface of the inner housing are used to control the total axial motion, combined rolling, and sliding, and prevent accidental disassembly of the ball spline. The ball spline is further discussed in the chapter on *Splines and Axial Motion Devices*. The maximum angular design capability of the joint for its entire axial stroke range can be determined by plotting a chart as described in *Appendix G*.

Since the joint and ball spline use the same lubricant, a common sealing means such as a boot seal can be employed. The boot seal is a flexible member and, in performing its principal function of keeping the lubricant in and foreign matter out of the joint and ball spline, it must be capable of operating under varying environmental conditions at various speeds, joint angles, and end motion positions. Generally, the seal is connected to the outer housing with a seal adapter and held in position with a seal retainer or a clamp. An O-ring seal is usually used between the outer housing and seal retainer. However, the boot seal can also be held directly in position on the outer housing with a metallic band magnetically formed in place or with a clamp. Generally, the seal is held in position on the shaft with a metallic band magnetically formed in place or with a clamp. Usually, a housing end cover closes the flanged end of the outer housing, while sealing is generally accomplished between the outer housing and end cover with an O-ring.

As previously indicated, the principal advantage of the ball spline Rzeppa joint is that it requires a relatively low axial thrust force during end movement when compared with conventional sliding splines or most other end motion type joints. Fig. 3 shows

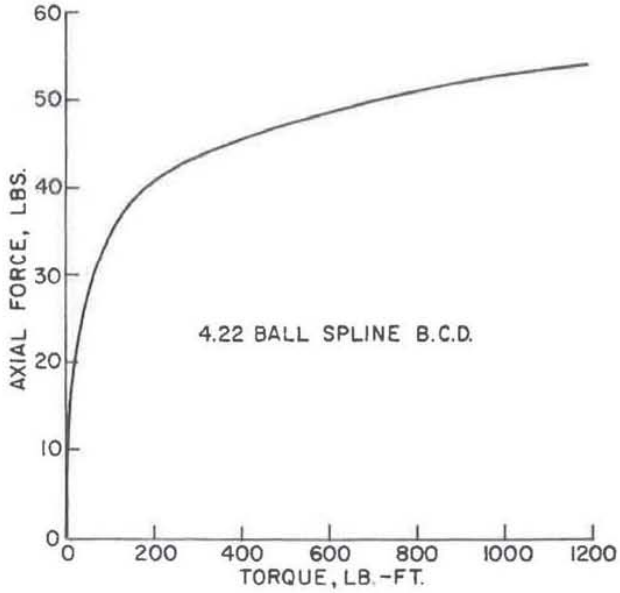


Fig. 3 — Ball spline Rzeppa joint axial force characteristics

the static (nonrotating) axial force characteristics for a typical joint.

Material Selection

The selection of optimum materials for the ball spline Rzeppa joint components is based on ultimate strength, fatigue strength,

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and other functional and durability requirements for a given application, as well as economical processing advantages.

The materials for the components comprising the disc type Rzeppa joint itself are the same as those used in a conventional Rzeppa joint in a similar application. Since the inner housing for the ball spline also serves as the outer race of the Rzeppa joint, this member is generally made from a formed blank or forging of low carbon alloy steel, and is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. The ball grooves for the joint and ball spline and the spherical inner surface are then finish ground. The case depth in the ball grooves must be sufficient to meet the contact load requirement explained in *Appendix A*.

The outer housing is usually made from a low carbon alloy steel forging, which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. The ball grooves are then finish ground. As in the case of the inner housing, the case depth in the ball grooves must be sufficient to meet the contact load requirement.

The ball spline balls are made of chromium steel, which are hardened to a typical range of R_c62-65 and finish lapped. The retaining rings are generally made of carbon spring steel.

The elastomeric material of the boot seal is usually a synthetic rubber, which must exhibit specific resistance to the action of petroleum-based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The seal adapter, seal retainer, housing end cover, and clamps are generally made of plain low carbon steel. The bands are usually made of either aluminum or copper, and the O-ring seal material is generally a synthetic rubber.

Lubrication

The ball spline Rzeppa joint is generally lubricated for life with an NLGI grade number 1 lubricating grease and is the same as usually used in a conventional Rzeppa joint for similar applications. A typical lubricant of this type generally consists of a refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant. Molybdenum disulfide is a common extreme pressure additive.

Typical basic performance requirements of a lubricating grease and its various properties are described in the *SAE Handbook* under *Automotive Lubricating Greases*. In addition to providing for optimum joint and ball spline life by reducing friction and wear at all bearing surfaces, the lubricant used must perform satisfactorily in all environmental conditions of the joint application.

Rating Procedure

The ball spline Rzeppa joint can be rated as described in *Appendix I*.

Applications

The ball spline Rzeppa joint has found limited usage as the inboard joint in axle driveshafts of front drive vehicles where there has been a need for a constant velocity joint providing end motion capability with relatively low axial force characteristics. Depending on the axial stroke requirement of an application, the maximum angular design capability of this joint is approximately 25 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle throughout its entire axial stroke range must be within the limit lines of the angular capability chart for a given joint design as described in *Appendix G*. For passenger car applications, normal operating angles as high as approximately six deg have been used in independent wheel suspension axle driveshafts.

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Section 3.2.10
Weiss Universal Joint
(End Motion Type)

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

Introduction

The end motion type of Weiss universal joint was patented by Weiss in 1936 and its first known vehicle usage began in 1937 as the inboard joint in an early American front wheel drive passenger car. Since 1961, it has also been used in some French front drive cars. Another design version of this joint with a parallel pairs groove construction was patented by Altmann in the mid 1960s and has been used as the axle driveshaft inboard and outboard joints in some independently sprung German rear drive cars.

The Weiss universal joint (end motion type), shown in Fig. 1 in a typical design, is a constant velocity universal joint, radially self-supported and permitting axial movement which consists of two yokes drivable connected through balls positioned and maintained in the constant velocity plane by inclined intersecting straight grooves, located in radial planes or by parallel pairs of inclined intersecting straight grooves symmetrically located about radial planes. Fig. 2 shows the joint with its major drive components in an angulated position.

The principal advantage of this joint is its ability to transmit constant velocity and simultaneously accommodate axial motion. Another advantage is that the joint can provide a relatively higher intermittent joint angle in an axle driveshaft when compared with other end motion joints used for this same application.

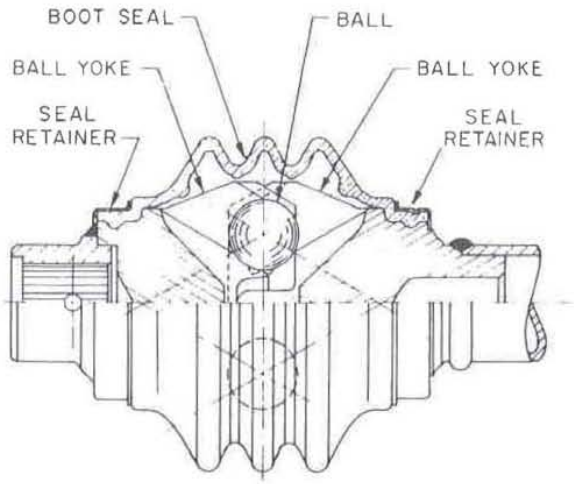


Fig. 1 — Weiss universal joint (end motion type)

A major disadvantage of the joint is that only one half of the drive balls transmit torque in either direction of rotation. Thus the swing diameter is not as small as with other ball type constant velocity joints where all the balls share in the torque load. Also, its

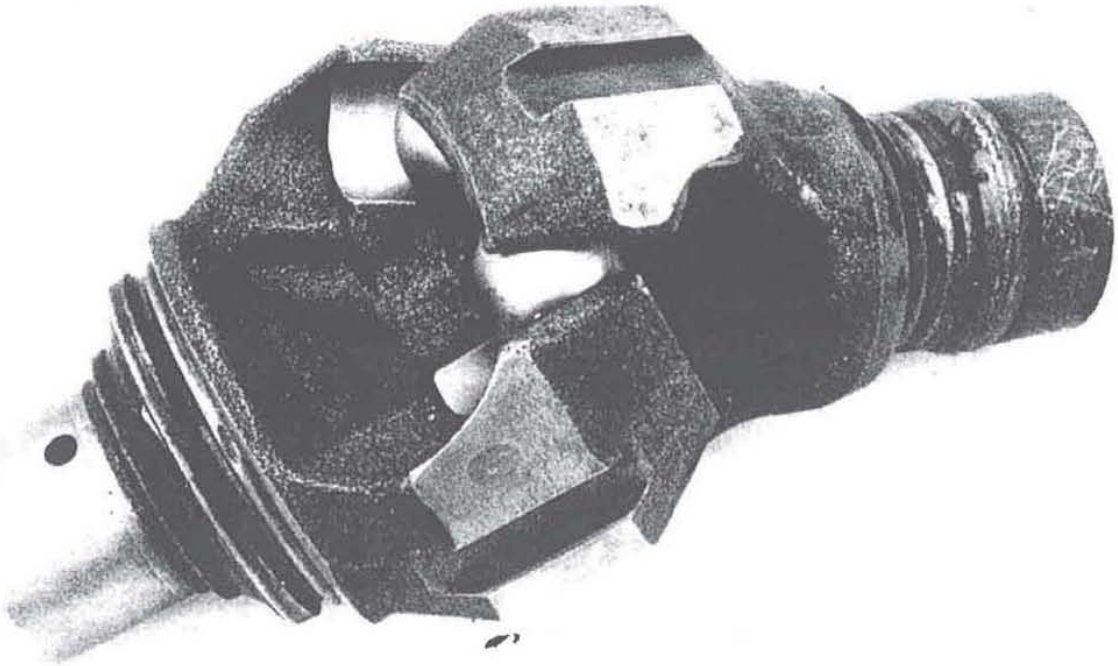


Fig. 2 — Angulated view of Weiss joint (end motion type)

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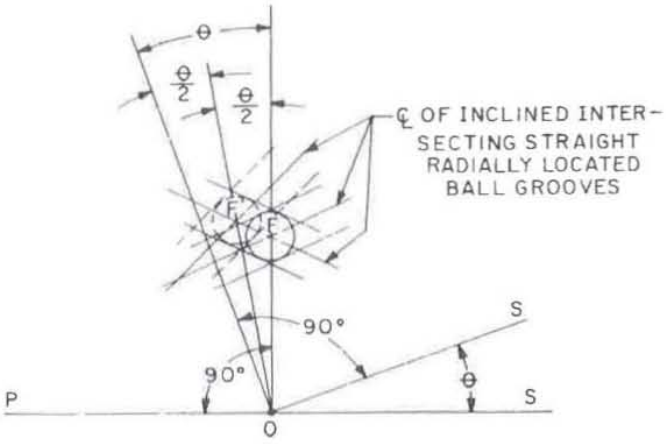


Fig. 3 — Diagram illustrating positioning of balls in bisecting angle plane

cost to manufacture is relatively high. A further limitation of this joint is its axial stroke capability when compared with some other end motion type joints.

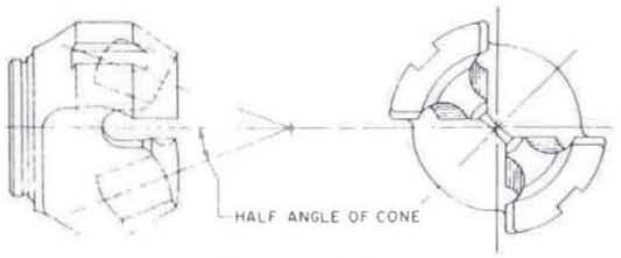


Fig. 4 — Ball yoke

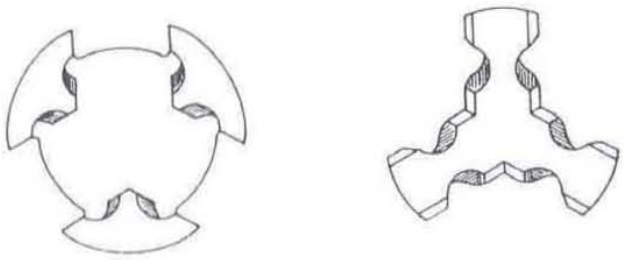


Fig. 5 — End view of ball yokes with parallel pairs of ball grooves

Kinematics

As previously indicated, the end motion type of Weiss joint transmits true constant velocity and simultaneously permits axial motion. The same internal geometry which provides for angular motion also allows axial movement. The drive balls are positioned and maintained in the constant velocity plane without a cage which is required, for example, in the well known Rzeppa joint. The positioning of the balls is accomplished solely by the action of the inclined intersecting straight ball grooves located in radial planes or by parallel pairs of inclined intersecting straight ball grooves symmetrically located about radial planes on the side of the ball yoke ears. The positioning of these balls in the constant velocity or bisecting angle plane is shown in Fig. 3.

When transmitting torque at an angle, a secondary couple is produced on both driving and driven members of the joint. As in all other ball type constant velocity joints which maintain driving contact through the constant velocity or bisecting angle plane, the couple forces react as static nonvibrating forces only on the bearing supports. The secondary couples are a function of the torque and joint angle only. These couples are of the same magnitude on both driving and driven members and are normal to the joint angle plane. For a given torque direction and disposition of the joint angle, both couples are sensed in the same direction. The values of these secondary couples are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

where:

C = Secondary couple on driving member

C_1 = Secondary couple on driven member

T = Torque

θ = Joint angle

The solution for the values of these secondary couples is shown in *Appendix H* for a simplified case of a ball type constant velocity joint in which driving contact is maintained at all times in the constant velocity plane.

Design

The various end motion type of Weiss joint components must be designed to provide the necessary joint angularity, axial travel, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the ball yoke configuration with an optimized ball size and mean ball circle diameter can be designed to meet the various joint application parameters.

The ball yoke is a yoke-shaped member with inclined straight ball grooves located in radial planes or by parallel pairs of inclined straight ball grooves symmetrically located about radial planes on the side of the yoke ears and with drivable means of attachment.

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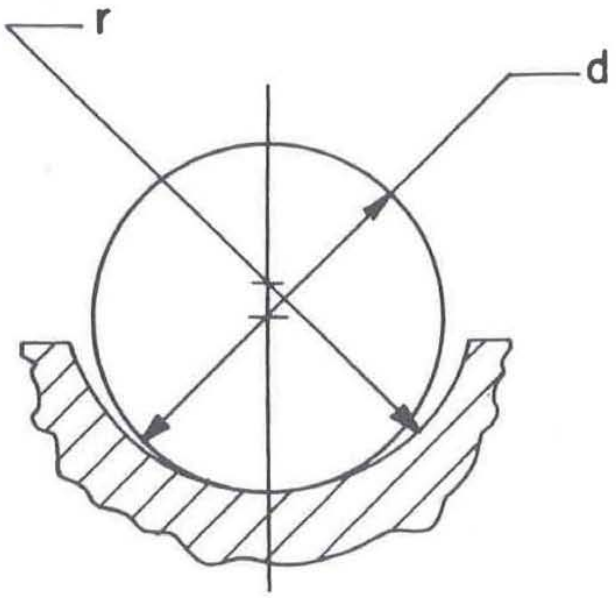


Fig. 6 — Cross section of circular arc ball groove

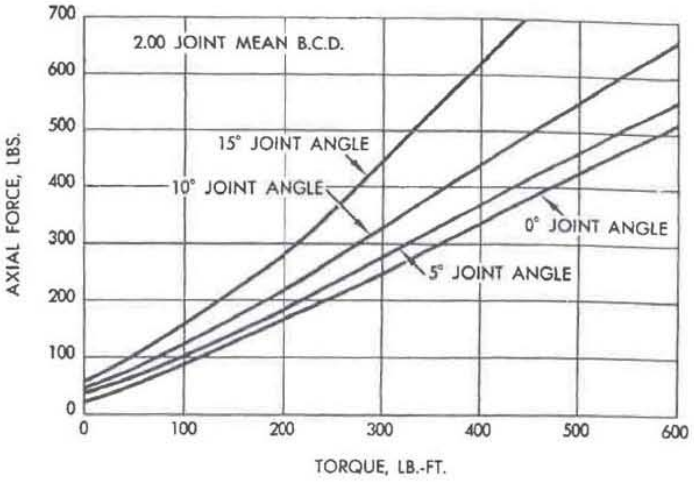


Fig. 7 — Weiss joint axial force characteristics

The typical design version with inclined straight grooves is shown in Fig. 4, while Fig. 5 illustrates the end view of the two ball yokes comprising the driving and driven members of a typical joint with the parallel pairs construction of inclined straight grooves. In typical joint designs, the ball yokes may be either welded to the connecting shaft or held in position on the shaft spline with a retaining ring or rings.

As in all other ball type constant velocity joints, the design of the ball groove geometry determines to a significant degree the functional and wear life characteristics of this joint. In typical constructions of the ball yoke, the half angle of the cone formed by the radiating ball grooves is in the range of 18-25 deg. This inclination of the ball grooves must be sufficient to position the balls in the constant velocity plane and, especially, to prevent the mating crossed grooves from becoming parallel at the maximum joint angle and thus allow escape of the balls. The required inclination of the ball grooves, therefore, will limit or determine the maximum joint axial stroke capability. The cross section of the ball grooves is a circular arc groove as illustrated in Fig. 6 with the groove radius-to-ball diameter ratio having a typical range of 0.5005-0.5020. As previously indicated, a characteristic of this joint is that only one half of the balls transmit torque in either direction of rotation. Therefore, an even number of balls must be used with four being the minimum number possible. Generally, four or six balls are used in a typical joint construction, which are subjected to a compression type of loading. The permissible Hertz contact stress in the ball grooves, including the associated computation procedure is shown in *Appendix A*.

The boot seal is a flexible member and in performing its principal function of keeping the lubricant in and foreign matter out of the joint, it must be capable of operating under varying environmental conditions at various speeds, joint angles, and end motion positions. For the joint design shown in Fig. 1, the seal must also withstand the centrifugal force effects of the entire lubricant mass and is usually held in position on both ball yokes with seal retainers.

Another design version of the joint employing the parallel pairs groove geometry on the ball yokes has used a two piece stamped housing configuration which is mounted on one yoke and surrounds the greater portion of the joint. This housing design aids in retaining the lubricant inside the joint and reduces the centrifugal force effects of the entire lubricant mass on the seal. O-rings are generally used to seal the two piece housing connection and its attachment to the yoke. The housing configuration is designed to permit the required maximum joint angularity and axial stroke and to prevent accidental disassembly of the joint. The boot seal is usually held in position on the housing and on the second ball yoke with clamps.

This housing design could also be used for the joint construction, shown in Fig. 1, to prevent accidental disassembly; otherwise other axial travel limitation means must be employed. The maximum angular design capability of the joint for its entire axial stroke range can be determined by plotting a chart as described in *Appendix G*. The static (nonrotating) axial force characteristics during end movement at various joint angles are shown in Fig. 7 for a typical end motion type of Weiss joint.

Material Selection

The selection of optimum materials for the end motion type of Weiss joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The ball yokes are usually made from low carbon alloy steel forgings which are carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. The ball grooves are usually finish ground after heat treatment, however, they can also be finish milled or broached prior to carburizing if desired. The case depth in the ball grooves must be sufficient to meet the contact load requirement explained in *Appendix A*.

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The balls are made of chromium steel which are hardened to a typical range of R_c62-65, and finish lapped. The retaining rings, when used to hold the ball yokes in position on the shaft spline, are generally made of carbon spring steel.

The elastomeric material of the boot seal is usually a synthetic rubber, which must exhibit specific resistance to the action of petroleum-based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The clamps or seal retainers are generally made of plain low carbon steel. When a stamped housing configuration is employed, the housing members are usually made from plain low carbon steel stampings and the O-ring seal material is generally a synthetic rubber.

Lubrication

The end motion type of Weiss joint is generally lubricated for life with a lubricating mineral oil of SAE viscosity number 80, 90 or 140. An NLGI number 1 grade lubricating grease consisting of a refined mineral oil and lithium 12 hydroxy stearate soap has also been used. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant for the grease, molybdenum disulfide has been a common extreme pressure material.

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

Rating Procedure

The end motion type of Weiss joint can be rated as described in *Appendix I*.

Applications

The end motion type of Weiss joint has found usage in axle driveshafts of front and rear drive vehicles where there has been a need for a constant velocity joint providing axial movement and permitting a relatively high joint angle capability. Depending on the axial stroke requirement of an application, the maximum angular design capability of this joint is approximately 30 deg. This angular capability is generally much greater than that required during extreme suspension movements of most vehicles.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle throughout its entire axial stroke range must be within the limit lines of the angular capability chart for a given joint design as described in *Appendix G*. For passenger car applications, normal operating angles as high as approximately 5 deg have been used in independent wheel suspension axle driveshafts.

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Section 3.2.11
Weiss Universal Joint
(Fixed Center Type)

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

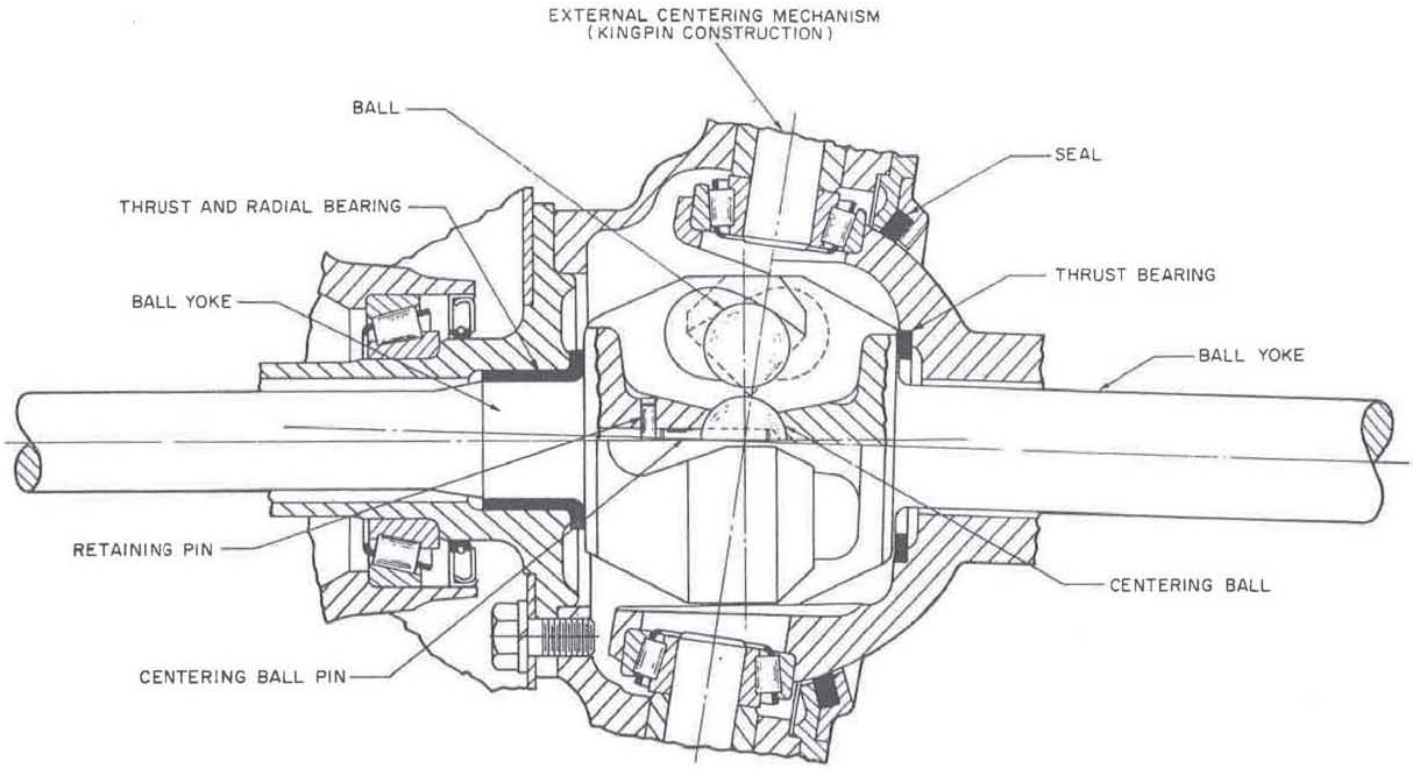


Fig. 1 — Weiss universal joint (fixed center type)

Introduction

The fixed center type of Weiss universal joint was patented by Weiss in 1925, and its first known vehicle usage began in 1937 as the outboard joint in an early American front wheel drive passenger car. Since that time, it has been used in highway and off-highway all wheel drive vehicles for both commercial and military applications.

The Weiss universal joint, (fixed center type) shown in Fig. 1 in a typical design, is a constant velocity universal joint, radially self-supported, which consists of two yokes drivably connected through balls positioned and maintained in the constant velocity plane by axially offset intersecting curved grooves located in radial planes and retained as an assembly by a centering ball. An external seal and axial support means are required.

The principal advantage of this joint is its ability to transmit constant velocity with a relatively high joint angle capability. The joint, however, has inherent disadvantages which restrict its usage. For example, only one half of the drive balls transmit torque in either direction of rotation. Thus, the swing diameter is not as small as with other ball type constant velocity joints where all the balls share in the torque load. Also, external sealing and axial supporting means are required. In addition, its cost to manufacture is relatively high.

Kinematics

As previously indicated, the fixed center type of Weiss joint transmits true constant velocity. The drive balls are positioned and maintained in the constant velocity plane without a cage, which is required, for example, in the well known Rzeppa joint. The positioning of the balls is accomplished solely by the action of the axially offset intersecting curved ball grooves located in radial planes on the side of the ball yoke ears. The groove offsets

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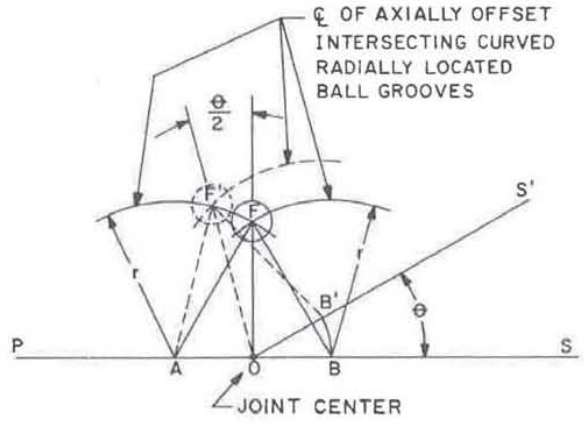


Fig. 2 — Diagram illustrating positioning of balls in bisecting angle plane

are symmetrically located on opposite sides of the joint center. The positioning of these balls in the constant velocity or bisecting angle plane is shown in Fig. 2 where a geometric proof can be derived.

When transmitting torque at an angle, a secondary couple is produced on both driving and driven members of the joint. As in all other ball type constant velocity joints which maintain driving contact through the constant velocity or bisecting angle plane, the couple forces react as static nonvibrating forces only on the bearing supports. The secondary couples are a function of the torque and joint angle only. These couples are of the same magnitude on both driving and driven members and are normal to the joint angle plane. For a given torque direction and disposition of the joint angle, both couples are sensed in the same direction. The values of these secondary couples are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

where:

C = Secondary couple on driving member

C_1 = Secondary couple on driven member

T = Torque

θ = Joint angle

The solution for the values of these secondary couples is shown in *Appendix H* for a simplified case of a ball type constant velocity joint in which driving contact is maintained at all times in the constant velocity plane.

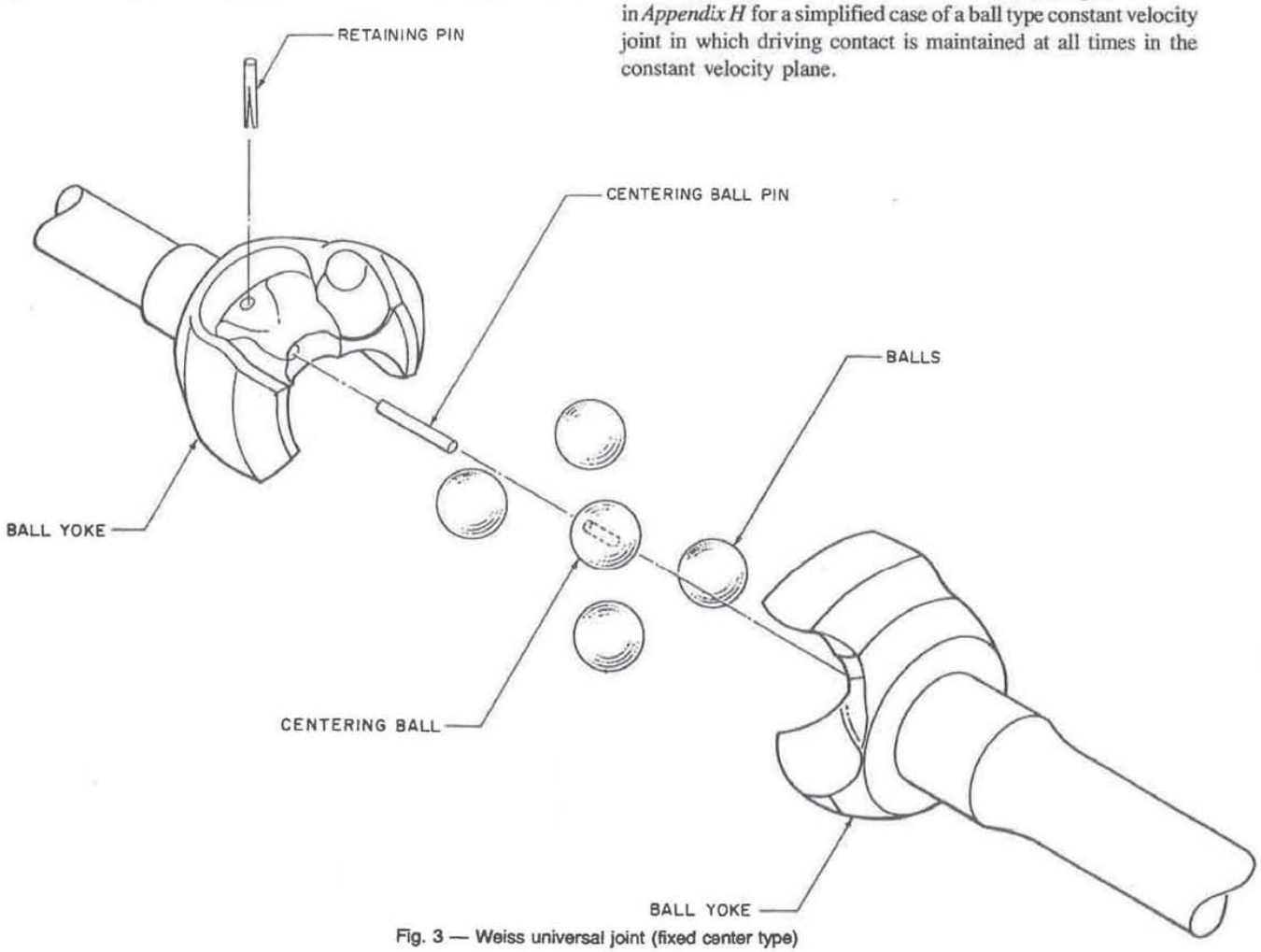


Fig. 3 — Weiss universal joint (fixed center type)

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Design

The various components of a typical fixed center type of Weiss joint, shown in an exploded view in Fig. 3, must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the ball yoke configuration with an optimized ball size and ball circle diameter can be designed to meet the various joint application parameters.

The ball yoke is a yoke-shaped member with curved ball grooves located in radial planes on the side of the yoke ears and with drivable means of attachment. Usually, the ball yoke is integral with the connecting shaft member.

As in all other ball type constant velocity joints, the design of the ball groove geometry determines to a significant degree the functional and wear life characteristics of this joint. In typical constructions of the ball yoke, the offset or inclination angle of the curved ball groove radius center from the joint center in the radial plane is in the range of 20–30 deg. This inclination angle must be sufficient to position and maintain the balls in the constant velocity plane. This angle can be expressed by either angles AFO or BFO, as shown in Fig. 2.

The cross section of the ball grooves is a circular arc groove as illustrated in Fig. 4 with the groove radius-to-ball diameter ratio having a typical range of 0.5005 – 0.5020. As previously indicated, a characteristic of this joint is that only one half of the balls transmit torque in either direction or rotation. Therefore, an even number of balls must be used with four being the minimum number possible. Generally, four or six balls are used in a typical joint construction, which are subjected to a compression type of loading. The permissible Hertz contact stress in the ball grooves, including the associated computation procedure, is shown in *Appendix A*.

The centering ball is a ball which centers the universal joint, absorbs end thrust, and has a hole for a pin that positions the universal joint components as an assembly. The centering ball pin is a pin which positions the centering ball, while the retaining pin is a pin which positions the centering ball pin in the pilot hole of the centering ball. The need for the centering ball pin and retaining pin in keeping the joint together is required only until it is installed in its operating enclosure. Therefore, in recent designs, these pins have been eliminated and the joint is simply held together by wire or by other means until installation, thus eliminating the need for a hole in the centering ball. The external seal and axial support means required for this joint are dependent upon the specific application and will therefore not be discussed here.

Material Selection

The selection of optimum materials for the fixed center type of Weiss joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given applicaion, as well as economical processing advantages.

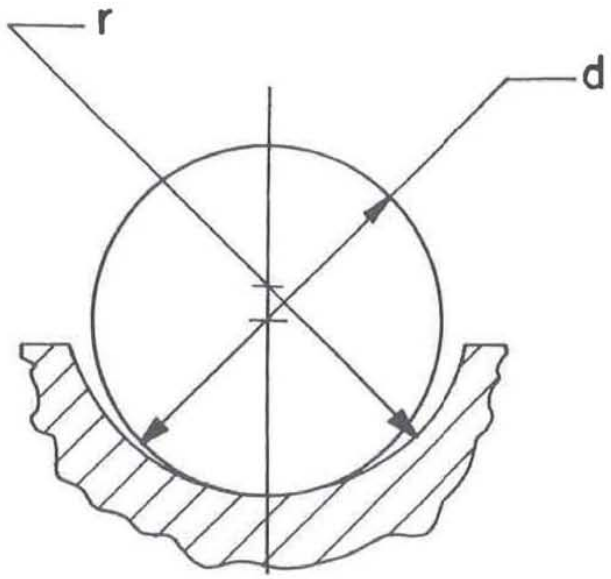


Fig. 4 — Cross section of circular arc ball groove

The ball yokes are usually made from low carbon alloy steel forgings which are carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the required wear characteristics and torsional strength. The ball grooves are generally finish milled before carburizing; however they can also be finish ground after heat treatment if desired. The case depth in the ball grooves must be sufficient to meet the contact load requirement explained in *Appendix A*.

Both the balls used for the transmission of torque and the centering ball are made of chromium steel which are hardened to a typical range of R_c62-65 and finish lapped. The centering ball pin is usually made of a through hardening steel, while the retaining pin is generally made of carbon spring steel or low carbon cold drawn steel.

Lubrication

The fixed center type of Weiss joint generally operates submerged in lubricant inside the closed steering knuckle of a vehicle's front live axle, steerable wheel. This typical lubricant is either a lubricating mineral oil of SAE viscosity number 90 or 140, or a lubricating grease. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

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

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Rating Procedure

The fixed center type of Weiss joint can be rated as described in *Appendix I*.

Applications

The fixed center type of Weiss joint has found usage in the closed steering knuckle of live rigid front axles of all wheel drive, highway and off-highway commercial and military vehicles, where there has been a need for a constant velocity joint providing a relatively high joint angle capability. The maximum angular design capability of this joint is in the range of 30–40 deg, which is greater than the angular capability of the closed steering knuckle. In these rigid front axle applications, the normal design operating angles have ranged from 0-1½ deg, while the maximum intermittent operating angle has been limited by the steering knuckle design.

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Section 3.2.12
Cross Groove Universal Joint

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

Introduction

The cross groove universal joint was patented by Faure in Europe and by Mazziotti and Sharp in the United States in the early 1960s. Its first known vehicle usage began in the mid 1960s as the axle driveshaft inboard and outboard joints in independently sprung German rear drive passenger cars, and as the inboard joint in German front wheel drive cars. Since that time, it has been applied to numerous other front and rear axle driveshaft applications. Also, this joint has been used in some passenger car driveline applications.

The cross groove universal joint, shown in Fig. 1 in a typical design, is a constant velocity universal joint, radially self-supported, which consists of an outer and inner race drivably connected through balls located in circumferentially spaced straight or helical grooves alternately inclined relative to the rotational axis. The balls are positioned in the constant velocity plane by an intersecting groove relationship and maintained in this plane by a cage located between the two races. The joint permits axial movement since the cage is not positionably engaged to either race.

The principal advantage of this joint is its ability to transmit constant velocity and simultaneously accommodate axial motion. Also, it is relatively economical to manufacture. A limitation of this joint is its generally smaller axial stroke capability when compared with some other end motion type joints.

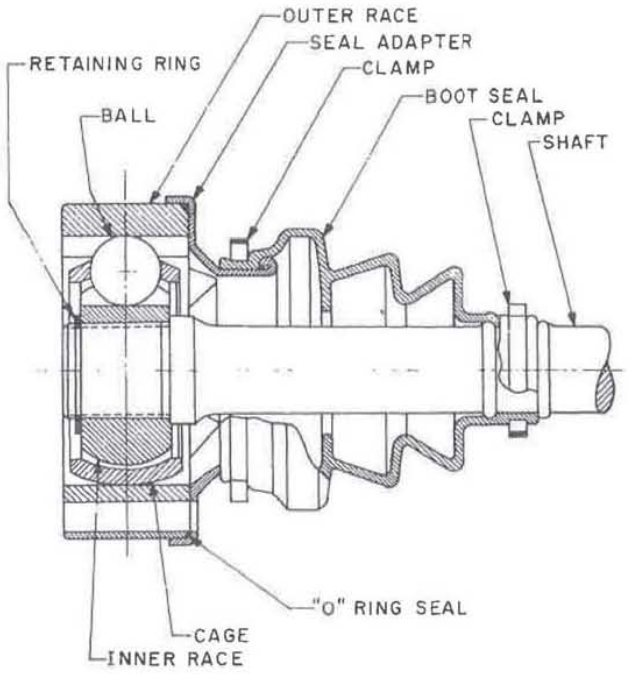


Fig. 1 — Cross groove universal joint

Kinematics

As previously indicated, the cross groove joint transmits true constant velocity and simultaneously permits axial motion. The same internal geometry which provides for angular motion also allows axial movement. The drive balls are positioned in the constant velocity plane by the action of the crossed circumferentially spaced and alternately inclined straight or helical ball grooves and maintained in this plane by a cage. The positioning of these balls in the constant velocity or bisecting angle plane is shown in Fig. 2.

When transmitting torque at an angle, a secondary couple is produced on both driving and driven members of the joint. As in all other ball type constant velocity joints which maintain driving contact through the constant velocity or bisecting angle plane, the couple forces react as static nonvibrating forces only on the bearing supports. The secondary couples are a function of the torque and joint angle only. These couples are of the same magnitude on both driving and driven members and are normal to the joint angle plane. For a given torque direction and disposition of the joint angle, both couples are sensed in the same direction.

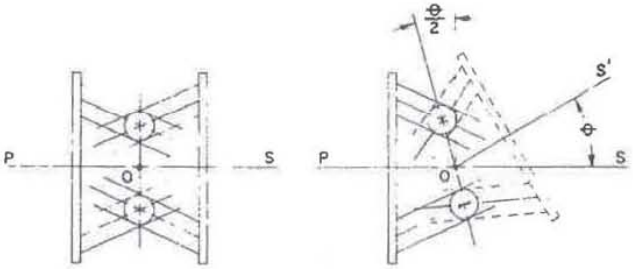


Fig. 2 — Diagram illustrating positioning of balls in bisecting angle plane

The values of these secondary couples are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

where:

C = Secondary couple on driving member

C_1 = Secondary couple on driven member

T = Torque

θ = Joint angle

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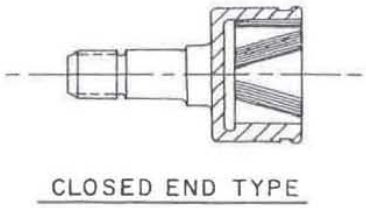


Fig. 3 — Typical cross groove joint outer race constructions

The solution for the values of these secondary couples is shown in *Appendix H* for a simplified case of a ball type constant velocity joint in which driving contact is maintained at all times in the constant velocity plane.

Design

The various cross groove joint components must be designed to provide the necessary joint angularity, axial travel, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the outer and inner race and cage configuration with an optimized ball size and ball circle diameter can be designed to meet the various joint application parameters.

The outer races shown in Fig. 3 illustrate the three typical constructions in use. The outer race is a member with circumferentially spaced straight or helical ball grooves alternately inclined on the cylindrical inner surface and with drivable means of attachment. Fig. 4 shows an end view of a typical outer race describing the orientation of the circumferentially spaced and alternately inclined grooves. The inner race is an annular member with circumferentially spaced straight or helical ball grooves alternately inclined on the partly spherical or conical outer clearance surfaces and with internally splined drivable means of attachment. The inner race is held in position on the shaft spline with a retaining ring or rings.

The cage is a ring-like member with concentric outer and inner cylindrical, or either partly spherical or conical surfaces, and with a circumferential series of openings or windows for maintaining the balls in a common plane.

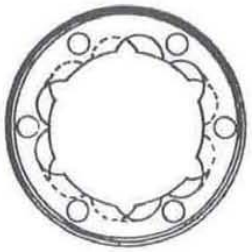


Fig. 4 — End view of typical cross groove joint outer race

When the joint is under static unloaded conditions, no means are required to maintain the cage concentric with the outer and inner races. Therefore, the effect of gravity may cause the cage to move radially into contact with one or both of the races. However, when torque is transmitted by the joint, the alternate balls are urged in opposite axial directions by the ball grooves. Opposite axial movement of these alternate balls is prevented by the cage, which maintains the balls in a common plane. Hence, the opposing axial forces tend to centralize the cage relative to the outer and inner races.

Because of design intent, or due to dimensional tolerances, the cage may lightly contact either the outer and/or inner races. Since the joint provides end motion, the balls positioned in the grooves of the two races and the cage must move axially relative to both races. Therefore, contact of the cage with the outer and/or inner races is not required for positioning of the balls or for proper functioning of the joint. In some ball type joints, the cage is used to position the balls in the constant velocity plane. In such joints, bearing surfaces must be provided between the cage and both races so that the positioning function of the cage can be accomplished.

When the commonly used cage design with partly spherical outer and inner surfaces is utilized, as shown in Fig. 1, its outer surface is in light contact with the cylindrical bore of the outer race. The cage inner surface limits the amount of axial stroke available in the joint by contacting the partly spherical or conical outer clearance surfaces of the inner race during extreme positions of end motion, and thus acting as an internal stop.

It is also possible to provide for somewhat increased axial travel by using various combinations of spherical, conical, or cylindrical surfaces for the cage and the outer surface of the inner race. Further, by proper selection of the cage and inner race configuration, a cage can be developed which is an inverse design of that previously described by providing instead, light contact with the outer surface of the inner race and clearance with the outer race cylindrical bore.

Depending on the inner race and cage configuration used, it may be necessary to utilize another joint member such as a seal adapter or other means, which will limit the maximum axial stroke and thus prevent accidental disassembly of the joint. The balls are assembled with an interference fit between the cage window flats to prevent undesirable ball noise when substantial torque must be transmitted. The window length is determined by the joint end motion and angularity requirements.

As in all other ball type constant velocity joints, the design of

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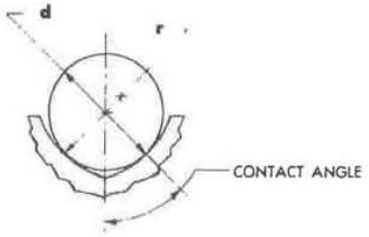


Fig. 5 — Cross section of elliptic arc ball groove

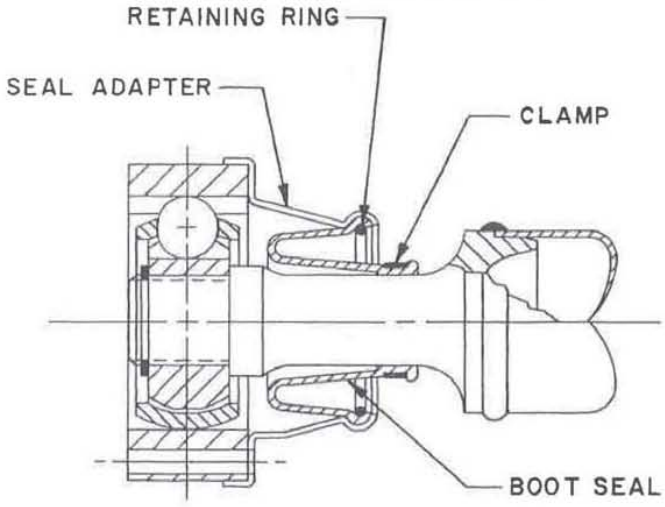


Fig. 6 — Boot seal configuration for high speed driveline application

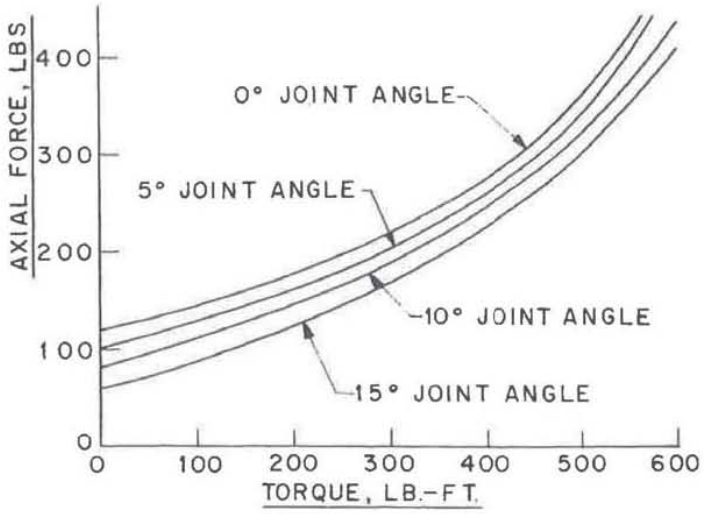


Fig. 7 — Cross groove joint axial force characteristics

the ball groove geometry determines to a significant degree the functional and wear life characteristics of this joint. In typical constructions of the races, the angle of the alternately inclined straight or helical ball grooves with respect to the race axis is in the range of 13-19 deg. This inclination of the ball grooves must be sufficient to position the balls in the constant velocity plane and especially to prevent two of the diametrically opposite mating crossed grooves from becoming parallel at the maximum operating joint angle. If it were not for the plane of the cage being defined and held by the remaining crossed or intersecting grooves, ball control in the two aligned grooves would be lost. In practice, it is generally desirable that the joint angle which results in two aligned grooves be somewhat higher than the maximum operating joint angle for a given application. The maximum possible inclination is governed by the circumferentially spaced and alternately inclined ball grooves converging toward each other at the ends of the races. The required inclination of the ball grooves, therefore, will limit or determine the maximum joint axial motion capability. The maximum angular design capability of the joint for its entire axial stroke range can be determined by plotting a chart as described in *Appendix G*.

The cross section of the ball grooves is usually an elliptical arc groove, as illustrated in Fig. 5, with the groove radius-to-ball diameter ratio having a typical range of 0.505-0.520, and the contact angle ranging from 30-45 deg. With this elliptical arc groove configuration, the balls are subjected to a compression type of loading. An even number of balls must be used in the joint with four being the minimum number possible. Six balls, however, are the usual number used in a typical joint construction. The permissible Hertz contact stress in the ball grooves, including the associated computation procedure, is shown in *Appendix A*.

The boot seal is a flexible member, and in performing its principal function of keeping the lubricant in and foreign matter out of the joint, it must be capable of operating under varying environmental conditions at various speeds, joint angles, and end motion positions. The boot seal design, shown in Fig. 1, is for a typical axle driveshaft application, while Fig. 6 illustrates a boot seal configuration for a typical high speed driveline application. Generally, the seal is connected to the outer race with a seal adapter and held in position with a clamp or retaining ring. An O-ring or a gasket seal is usually used between the outer race and seal adapter. Generally, a clamp is employed to hold the boot seal in position on the shaft. Usually, an outer race end cover closes the flanged type outer race and may also be used, if desired, with the disc type outer race while sealing is generally accomplished, as required, with an O-ring or a gasket. The static (nonrotating) axial force characteristics during end movement at various joint angles are shown in Fig. 7, for a typical cross groove joint.

Material Selection

The selection of optimum materials for the cross groove joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application as well as economical processing advantages.

The outer race is generally made of plain medium carbon steel. After finish broaching the ball grooves and cylindrical bore, the outer race is induction hardened to an adequate case depth and a typical R_c60 minimum surface hardness in the ball groove and bore area to provide the required wear characteristics and torsional strength. The case depth in the ball grooves must be

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sufficient to meet the contact load requirement explained in *Appendix A*.

The inner race is usually made of plain medium carbon steel. After finish broaching the ball grooves and spline, the inner race is induction hardened to an adequate case depth and a typical R_c60 minimum surface hardness in the ball grooves and on the outer surface to provide the required wear characteristics and torsional strength. As in the case of the outer race, the case depth in the ball grooves must be sufficient to meet the contact load requirement. The inner race can also be made of low carbon alloy steel which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to meet the functional and durability requirements.

The cage is generally made of low carbon alloy steel which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the necessary wear characteristics and strength. The window flats are then finish ground. When the partly spherical section cage design is used, the outer surface is also commonly finish ground. The balls are made of chromium steel which are hardened to a typical range of R_c62 - 65 and finish lapped. The retaining rings are usually made of carbon spring steel.

The elastomeric material of the boot seal is generally a synthetic rubber, which must exhibit specific resistance to the action of petroleum-based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The seal adapter, outer race end cover, and clamps are usually made of plain low carbon steel. The O-ring seal material is generally a synthetic rubber, while the gasket is usually made from gasket paper.

Lubrication

The cross groove joint is generally lubricated for life with an NLGI grade number 1 lubricating grease. A typical lubricant of this type usually consists of a refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant. Molybdenum disulfide is a common extreme pressure additive.

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

Rating Procedure

The cross groove joint can be rated as described in *Appendix I*.

Applications

The cross groove joint has found usage in axle driveshafts of front and rear drive vehicles where there has been a need for a constant velocity joint providing end motion capability. In addition, this joint has also been used in vehicle main drivelines. Depending on the axial stroke requirement of an application, the maximum angular design capability of this joint is approximately 25 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle throughout its entire axial stroke range must be within the limit lines of the angular capability chart for a given joint design as described in *Appendix G*. For passenger car applications, normal operating angles as high as approximately 6 deg have been used in independent wheel suspension axle driveshafts, while normal operating angles as high as approximately 4 deg have been employed in main drivelines.

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Section 3.2.13
Double Offset Universal Joint
(End Motion Type)

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

Introduction

The end motion type of double offset universal joint was patented by Devos in the mid 1960s and its first known vehicle usage began shortly thereafter as the axle driveshaft inboard joint in a German rear drive passenger car with a swing axle independent suspension. Since that time, it has been applied as the inboard joint in numerous Japanese, British, and other front wheel drive cars, and also as the axle driveshaft inboard and outboard joints in some rear drive cars. Also, this joint has been used in some passenger car driveline applications.

The double offset universal joint, (end motion type) shown in Fig. 1 in a typical design, is a constant velocity universal joint, radially self-supported, which consists of an outer and inner race drivably connected through balls located in axially straight grooves. The balls are positioned and maintained in the constant velocity plane by a cage with axially offset spherical surfaces located between the two races. The joint permits axial movement since the cage is positionably engaged only by the inner race.

The principal advantage of this joint is its ability to transmit constant velocity and simultaneously accommodate axial motion. The axial stroke capability of this joint is relatively large when compared with other end motion type constant velocity joints. Also, it is relatively economical to manufacture.

The principal disadvantage of this joint is the relatively greater sliding encountered by the balls in the grooves when axial movement is required. This increased sliding characteristic is due to the cage being positionably engaged on the inner race and thus requiring the balls to move the entire distance of the axial stroke instead of one half of the distance of the relative axial displacement between the outer and inner races of some other typical end motion ball type constant velocity joints. Hence, this increased sliding produces a relatively higher axial thrust force.

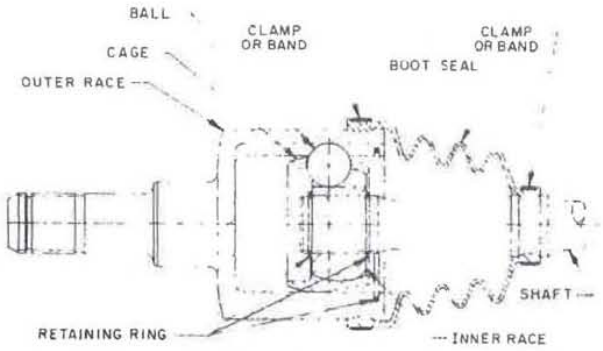


Fig. 1 — Double offset universal joint (end motion type)

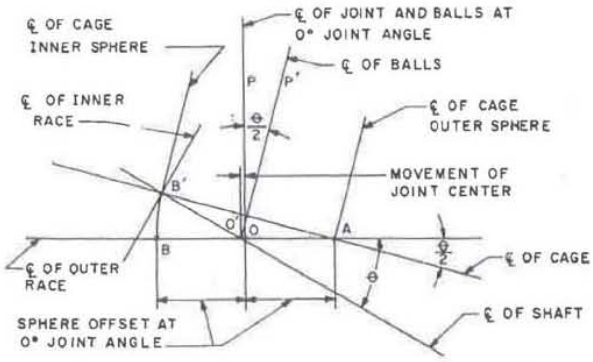


Fig. 2 — Diagram illustrating positioning of cage and balls in bisecting angle plane

Kinematics

As previously indicated, the end motion type of double offset joint transmits true constant velocity and simultaneously permits axial motion. The same internal geometry which provides for angular motion also allows axial movement. The drive balls are positioned and maintained in the constant velocity plane by a cage which has axially offset outer and inner partly spherical surfaces. The centers of these spherical surfaces are offset an equal distance on opposite sides of the plane of the ball centers along the joint axis and the cage, therefore, acts as a cam in moving the balls into the bisecting angle plane. Fig. 2 illustrates the positioning of the cage and balls in the constant velocity or bisecting angle plane.

When transmitting torque at an angle, a secondary couple is produced on both driving and driven members of the joint. As in all other ball type constant velocity joints which maintain driving contact through the constant velocity or bisecting angle plane, the couple forces react as static nonvibrating forces only on the bearing supports. The secondary couples are a function of the torque and joint angle only. These couples are of the same magnitude on both driving and driven members and are normal to the joint angle plane. For a given torque direction and disposition of the joint angle, both couples are sensed in the same direction. The values of these secondary couples are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

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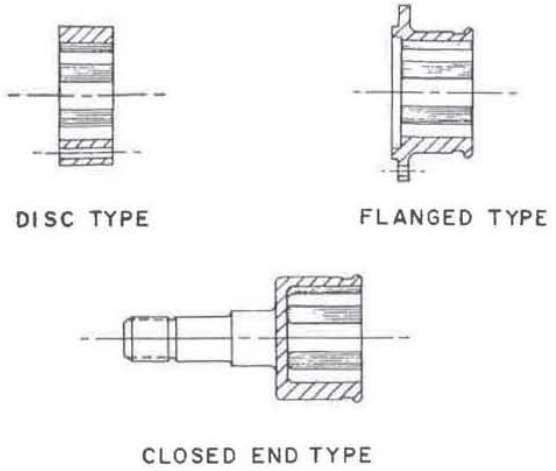


Fig. 3 — Typical double offset joint outer race constructions

where:

C = Secondary couple on driving member

C_1 = Secondary couple on driven member

T = Torque

θ = Joint angle

The solution for the values of these secondary couples is shown in *Appendix H* for a simplified case of a ball type constant velocity joint in which driving contact is maintained at all times in the constant velocity plane.

Design

The various end motion type of double offset joint components must be designed to provide the necessary joint angularity, axial travel, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the outer and inner race and cage configuration with an optimized ball size and ball circle diameter can be designed to meet the various joint application parameters.

The outer races shown in Fig. 3 illustrate three typical constructions in use. The outer race is a member with axially straight ball grooves on the cylindrical inner bearing surface and with drivable means of attachment. The inner race is an annular member with axially straight ball grooves on the partly spherical outer bearing surface and with internally splined drivable means of attachment. The inner race is held in position on the shaft spline with retaining rings. The cage is a ring-like member with axially offset outer and

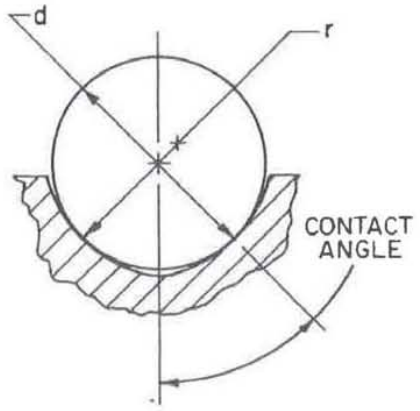


Fig. 4 — Cross section of elliptic arc ball groove

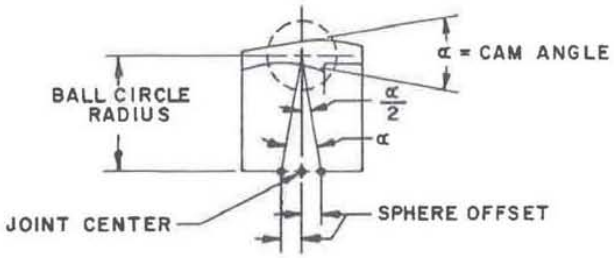


Fig. 5 — Diagram illustrating cam angle formed by cage offset spheres with respect to ball circle radius and joint center

inner partly spherical bearing surfaces and with a circumferential series of openings or windows for maintaining the balls in a common plane. A counterbore is generally provided at one end to permit assembly of the inner race in the cage. The outer spherical diameter of the cage acts as a bearing surface with the outer race cylindrical bore, while the cage inner spherical diameter provides for bearing engagement with the outer spherical surface of the inner race. The balls are assembled with an interference fit between the cage window flats to prevent undesirable ball noise when substantial torque must be transmitted. The window length is determined by the joint angularity requirement.

As in all other ball type constant velocity joints, the design of the ball groove geometry determines to a significant degree the functional and wear life characteristics of this joint. The cross section of the ball grooves is usually an elliptical arc groove as illustrated in Fig. 4 with the groove radius-to-ball diameter ratio having a typical range of 0.505-0.520 and the contact angle ranges from 30−40 deg. With this elliptical arc groove configuration, the balls are subjected to a compression type of loading.Three or more balls must be used with six being the usual number in a typical joint construction. The permissible Hertz contact stress in the ball grooves, including the associated computation procedure, is shown in *Appendix A*.

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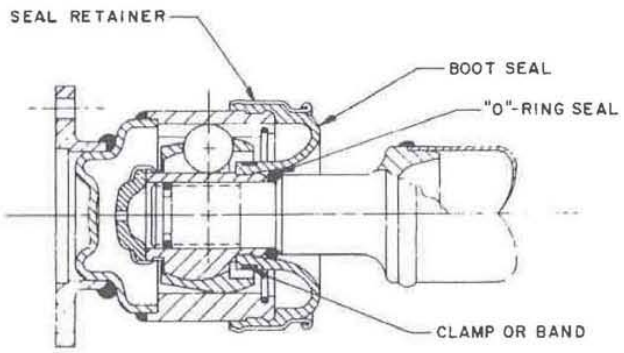


Fig. 6 — Boot seal configuration for high speed driveline application

The cage has axially offset outer and inner partly spherical bearing surfaces, which cause it to function like a cam when the joint is angulated. With respect to the ball center, this offset can be expressed as a cam or wedge angle as shown in Fig. 5. This angle ranges from 17–22 deg in a typical joint.

The boot seal is a flexible member and in performing its principal function of keeping the lubricant in and foreign matter out of the joint it must be capable of operating under varying environmental conditions at various speeds, joint angles, and end motion positions.

The boot seal design shown in Fig. 1 is for a typical axle driveshaft application, while Fig. 6 illustrates a boot seal configuration for a typical high speed driveline application. For axle driveshaft applications, the seal is generally held in position on the outer race and shaft with metallic bands, which are magnetically formed in place, or with clamps. However, when the disc type of outer race is employed, a seal adapter similar to the adapter design shown for the cross groove joint is usually used to connect the boot seal to the outer race with the seal held in position with a clamp. An O-ring or a gasket is then generally used between the outer race and seal adapter. For high speed driveline applications, the boot seal is usually held in position on the outer race with a seal retainer because of the centrifugal force effects created by the seal itself and the lubricant mass. Generally, an outer race end cover closes the flanged type of outer race and may also be used, if desired, with the disc type of outer race while sealing is accomplished between the outer race and end cover, as required, with an O-ring or a gasket.

To prevent accidental disassembly of the joint, the maximum axial travel can be limited with an internal stop feature. Some of these typical stop means which have been used are a retaining ring as shown in Figs. 1 and 6, a separate stamped ball stop attached to the outer race end, a seal adapter, and staking of the ball groove ends. Staking of the ball grooves must be accomplished prior to heat treatment and is used only with the flanged housing design since the joint can then be assembled thru the open flanged end. The maximum angular design capability of the joint for its entire axial stroke range can be determined by plotting a chart as described in *Appendix G*.

As previously indicated, the principal disadvantage of the end motion type of double offset joint is the relatively high axial thrust force during end movement when compared with most other end motion ball type constant velocity joints. Fig. 7 shows the static

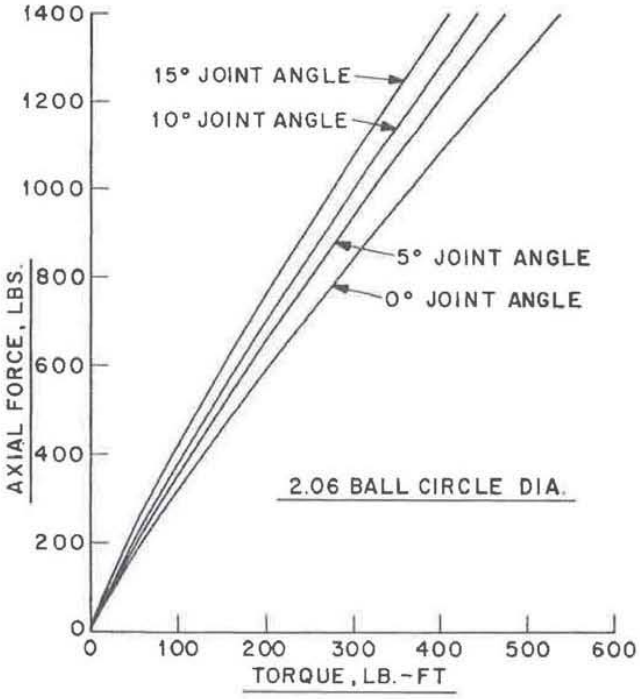


Fig. 7 — Double offset joint axial force characteristics

(non-rotating) axial force characteristics at various joint angles for a typical joint.

Material Selection

The selection of optimum materials for the end motion type of double offset joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The outer race is generally made of low carbon alloy steel. When the outer race is made from a forging or tubular bar stock, the ball grooves and cylindrical bore are usually finish broached prior to carburizing and hardening. If the outer race is formed, the ball grooves and cylindrical bore can be finish formed prior to carburizing and hardening. This heat treatment must provide an adequate case depth and a typical R_c60 minimum surface hardness to obtain the required wear characteristics and torsional strength for the outer race. The case depth in the ball grooves must be sufficient to meet the contact load requirement explained in *Appendix A*.

The inner race is generally made of low carbon alloy steel. The ball grooves may be finish ground after carburizing and hardening along with the spherical outer surface or they may be finish broached along with the spline prior to carburizing and hardening. This heat treatment must provide an adequate case depth and a typical R_c60 minimum surface hardness to obtain the required wear characteristics and torsional strength for the inner race. As in the case of the outer race, the case depth in the ball grooves must be sufficient to meet the contact load requirement.

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The cage is usually made of low carbon alloy steel which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the necessary wear characteristics and strength. The spherical outer and inner surfaces and the window flats are then finish ground.

The balls are made of chromium steel which are hardened to a typical range of R_c62-65 and finish lapped. The retaining rings are generally made of carbon spring steel.

The elastomeric material of the boot seal is usually a synthetic rubber, which must exhibit specific resistance to the action of petroleum based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The seal adapter, seal retainer, outer race end cover, and ball stop are generally made of plain low carbon steel. The clamps are usually made of either plain low carbon or stainless steel, while the bands are generally made of aluminum. The O-ring seal material is usually a synthetic rubber and the gasket is generally made from gasket paper.

Lubrication

The end motion type of double offset joint is generally lubricated for life with an NLGI grade number 1 lubricating grease. A typical lubricant of this type usually consists of a refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant. Molybdenum disulfide is a common extreme pressure additive. Another lubricant which has also been used is a synthetic non-petroleum grease with a lithium soap thickener and various additives, and having an NLGI grade number 0.

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

Rating Procedure

The end motion type of double offset joint can be rated as described in *Appendix I*.

Applications

The end motion type of double offset joint has found usage in axle driveshafts of front and rear drive vehicles where there has been a need for a constant velocity joint providing a relatively large axial movement capability. In addition, this joint has also been used in vehicle main drivelines. Depending on the axial stroke requirement of an application, the maximum angular design capability of this joint is approximately 25 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle throughout its entire axial stroke range must be within the limit lines of the angular capability chart for a given joint design as described in *Appendix G*. For passenger car applications, normal operating angles as high as approximately 6 deg have been used in independent wheel suspension axle driveshafts, while normal operating angles as high as approximately 5 deg have been employed in main drivelines.

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Section 3.2.14
Double Offset Universal Joint
(Fixed Center Type)

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

Introduction

The fixed center type of double offset universal joint was patented by Rzeppa in 1936 and its first known vehicle usage began in the early 1970s in steering shafts of some American trucks and motorhome vehicles.

The double offset universal joint, (fixed center type) shown in Fig. 1 in a typical design, is a self-supported constant velocity universal joint, which consists of an outer and inner race drivably connected through balls located in axially straight grooves. The balls are positioned and maintained in the constant velocity plane by a cage with axially offset spherical surfaces located between the two races.

The principal advantage of this joint is its ability to transmit constant velocity with a relatively high joint angle capability. Also, it is relatively economical to manufacture.

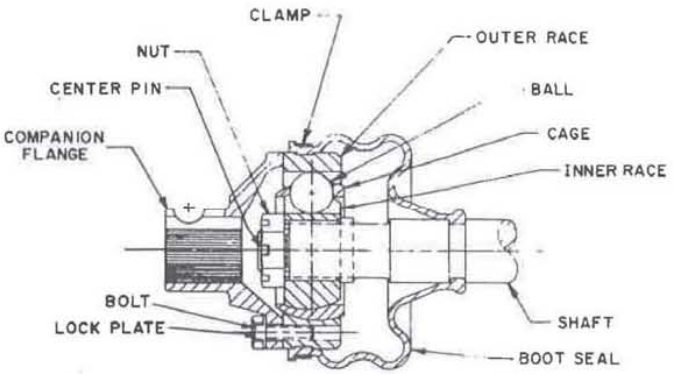


Fig. 1 — Double offset universal joint (fixed center type)

Kinematics

As previously indicated, the fixed center type of double offset joint transmits true constant velocity. The drive balls are positioned and maintained in the constant velocity plane by a cage which has axially offset outer and inner partly spherical surfaces. The centers of these spherical surfaces are offset an equal distance on opposite sides of the plane of the ball centers along the joint axis and the cage, therefore, acts as a cam in moving the balls into the bisecting angle plane. In the section for the *Double Offset Universal Joint (End Motion Type)*, this principle of movement of the cage and balls into the constant velocity or bisecting angle plane is shown in an illustration.

When transmitting torque at an angle, a secondary couple is produced on both driving and driven members of the joint. As in all other ball type constant velocity joints which maintain driving contact through the constant velocity or bisecting angle plane, the couple forces react as static nonvibrating forces only on the bearing supports. The secondary couples are a function of the torque and joint angle only. These couples are of the same magnitude on both driving and driven members and are normal to the joint angle plane. For a given torque direction and disposition of the joint angle, both couples are sensed in the same direction. The values of these secondary couples are:

$$C = C_1 = T \tan \frac{\theta}{2}$$

where:

C = Secondary couple on driving member

C_1 = Secondary couple on driven member

T = Torque

θ = Joint angle

The solution for the values of these secondary couples is shown in *Appendix H* for a simplified case of a ball type constant velocity joint in which driving contact is maintained at all times in the constant velocity plane.

Design

The various fixed center type of double offset joint components must be designed to provide the necessary joint angularity, strength, and life requirements for a given application. The ultimate strength of the joint must be safely in excess of the maximum applied torque, which can be developed by various loading modes discussed in the chapter on *Basic Driveline Requirements*. Initially, the shaft size is determined as outlined in the chapter on *Axle Driveshafts*. Then the outer and inner race and cage configuration with an optimized ball size and ball circle diameter can be designed to meet the various joint application parameters.

The disc type outer race shown in Fig. 1 illustrates the typical construction in use. The outer race is an annular member with axially straight ball grooves on the partly spherical inner bearing

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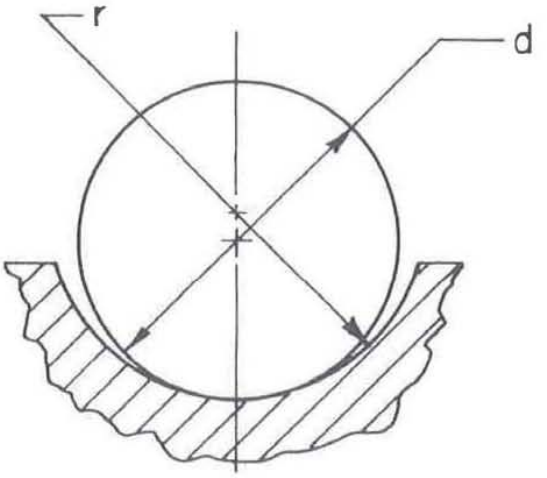


Fig. 2 — Cross section of circular arc ball groove

surface and with drivable means of attachment. The inner race is an annular member with axially straight ball grooves on the partly spherical outer bearing surface and with internally splined drivable means of attachment. The inner race is held in position on the shaft spline with a retaining ring or rings or with a nut.

The cage is a ring-like member with axially offset outer and inner partly spherical bearing surfaces and with a circumferential series of openings or windows for maintaining the balls in a common plane. A chamfer or cylindrical diameter, if needed, at one or both ends of the inner spherical diameter allows for assembly of the inner race in the cage. Also, the edge contour of the cage windows and a minimized thickness of the disc outer race provides for the clearance to assemble the cage in the outer race. The outer spherical diameter of the cage acts as a bearing surface with the outer race inner spherical surface, while the cage inner spherical diameter provides for bearing engagement with the outer spherical surface of the inner race. When the torque load is relatively light, such as in a steering shaft application, the balls may be assembled with a loose fit between the cage window flats rather than with an interference fit. The loose fit thus reduces the steering handwheel effort. The length of the windows is determined by the required joint angularity and the ability to assemble the joint components.

As in all other ball type constant velocity joints, the design of the ball groove geometry determines to a significant degree the functional and wear life characteristics of this joint. The cross sectional configuration of the ball grooves used in a specific steering shaft application has been a typical circular arc groove as illustrated in Fig. 2 with the groove radius-to-ball diameter ratio at 0.502. With the circular arc groove configuration, the balls are subjected to a shear type of loading. An elliptical arc type of groove can also be used if desired, and the resulting angular contact would subject the balls to a compression type of loading. Three or more balls must be used with six being the usual number in a typical joint construction. The permissible Hertz contact stress in the ball grooves, including the associated computation procedure, is shown in *Appendix A*.

The cage has axially offset outer and inner partly spherical bearing surfaces which cause it to function like a cam when the joint is angulated. With respect to the ball center, this offset can be expressed as a cam or wedge angle. In the section for the *Double Offset Universal Joint (End Motion Type)*, this cam angle is described in an illustration.

For a given steering shaft application where the joint angle varied from approximately 20-45 deg in one direction only, and did not pass thru 0 deg, this cam angle was 10 deg. If it is desired to accommodate higher operating speeds, reduced joint angles, and over-center angular motion, the offset of the cage spherical surfaces must be increased in order to provide the required larger cam angle. The cam angle range for such a typical joint is shown in the section on the *Double Offset Universal Joint (End Motion Type)*.

The boot seal is a flexible member and in performing its principal function of keeping the lubricant in and foreign matter out of the joint, it must be capable of operating under varying environmental conditions at various speeds and joint angles. A clamp is generally used to hold the seal in position on the outer race, and may also be used at the shaft end, if required.

Material Selection

The selection of optimum materials for the fixed center type of double offset joint components is based on ultimate strength, fatigue strength, and other functional and durability requirements for a given application, as well as economical processing advantages.

The outer race is generally made of low carbon alloy steel. The ball grooves are usually finish broached prior to carburizing and hardening, while the inner spherical surface is finish ground after carburizing and hardening. This heat treatment must provide an adequate case depth and a typical R_c60 minimum surface hardness to obtain the required wear characteristics and torsional strength for the outer race. The case depth in the ball grooves must be sufficient to meet the contact load requirement explained in Appendix A.

The inner race is generally made of low carbon alloy steel. The ball grooves may be finished ground after carburizing and hardening along with the spherical outer surface, or they may be finish broached along with the spline prior to carburizing and hardening. This heat treatment must provide an adequate case depth and a typical R_c60 minimum surface hardness to obtain the required wear characteristics and torsional strength for the inner race. As in the case of the outer race, the case depth must be sufficient to meet the contact load requirement.

The cage is usually made of low carbon alloy steel which is carburized to an adequate case depth and surface hardened to a typical R_c60 minimum to provide the necessary wear characteristics and strength. The spherical outer and inner surfaces and the window flats are then finish ground. The balls are made of chromium steel which are hardened to a typical range of R_c62-65 and finish lapped.

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The elastomeric material of the boot seal is generally a synthetic rubber, which must exhibit specific resistance to the action of petroleum based fluids and permit the seal to function satisfactorily for any environmental operating condition of the joint application. The clamp is usually made of plastic or plain low carbon steel.

Lubrication

The fixed center type of double offset joint is generally lubricated for life with an NLGI grade number 2 lubricating grease. A typical lubricant of this type usually consists of a refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

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

Rating Procedure

The fixed center type of double offset joint can be rated as described in *Appendix I*.

Applications

The fixed center type of double offset joint has found usage in vehicle steering shaft applications where there has been a need for a constant velocity joint operating at relatively high joint angles. In these applications, the joint operates at relatively slow speeds and low torque loads. The maximum angular design capability of this joint is in the range of approximately 40-45 deg.

The normal design operating angle of the joint will depend on the specific application, while the maximum intermittent operating angle must be less than the maximum angular capability for a given joint design. For steering shaft applications, the relatively continuous operating angle is limited by and must be less than this maximum angular capability.

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Tubing

Chapter 4

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Chapter 4
Tubing

E. R. Wagner, P.E. — Saginaw Steering Gear Div., GMC
L. W. Szymanski — Lift Parts Mfg., Inc.

Design Parameters

Since a tube is the connecting tubular member of a driveshaft, certain design parameters or requirements must be known in order to select the proper tubing size for a specific driveline application. These design parameters are as follows:

Maximum Applied Torque — In general, this torque is considered to be the maximum installed net engine torque multiplied by the appropriate converter stall, transmission low gear, auxiliary transmission, transfer case, front or rear differential or tandem rear axle ratios as applicable. In some instances, however, wheel slip may control the maximum torque which a driveshaft experiences. Furthermore, shock torque as compared to steadily applied torque can be significantly higher in some applications and must not be neglected when determining tube size. The maximum torsional loads which can be developed by various loading modes in a driveshaft or any of its components are discussed in the chapter on *Basic Driveline Requirements*.

Maximum Operating Speed — The maximum driveline operating speed must be known so that the minimum allowable critical speed of the driveshaft can be established. The procedure for determining the critical speed of a driveshaft with a given size tubing is shown in the chapter on *Critical Speed*.

Driveshaft Length, Center to Center — This length is usually governed by the critical speed requirements. Generally, the maximum shaft length for one piece automotive drivelines has been limited to approximately 60 in and is discussed further in the chapter on *Critical Speed*.

Minimum Underbody or Other Vehicle Clearances — Due to suspension motions and other driveline movements, these minimum clearances may restrict all or part of the driveshaft tube diameter. In some instances this can adversely affect the critical speed and may necessitate other design approaches such as the use of a two-piece driveline construction.

Bending Stiffness — Under certain operating conditions, first and second mode resonant bending or beaming frequencies are imposed on the driveline as indicated in the chapters on *Critical Speed* and *Vibration Control*. It is possible to alter somewhat these resonant beaming frequencies by selecting an optimum tube size and wall thickness that may achieve a more desirable transverse stiffness. For example, the bending stiffness may be increased by increasing the tube diameter and reducing the wall thickness of the tubing, or by decreasing the shaft length. The increased transverse stiffness will result in a higher critical speed and reduced shaft flexure. Stiffening individual driveline components doesn't always raise the resonant beaming frequency of the driveline and if it does, it is not substantial. However, once the powerplant is made more rigid, a stiffer driveshaft will contribute to a higher resonant frequency. The bending stresses resulting from shaft flexure at the resonant beaming frequencies must be well within the elastic limit of the tube material.

Torsional Stiffness — The driveline serves as a relative torsionally elastic subsystem in a typical vehicle power train system, and in its simplest form, it is a torsional spring element connecting two masses of relatively large rotating inertias. In a conventional vehicle, the engine, clutch and transmission are located at one end, while the differential, axle shafts and driving wheels are at the other end. These would be replaced by other forms of inertia in axle driveshafts or halfshafts, interaxle driveshafts, and other drive system arrangements.

This system can be subject to a large variety of disturbing forces depending on the specific installation and are discussed in the chapter on *Vibration Control*. Specifically, the driveshaft may be a source of excitation, for torsional vibrations act as a path for disturbances from other sources at certain response frequencies, and, provide isolation when properly tuned in the critical response ranges.

The driveshaft resonant torsional frequency can be altered or detuned, that is raised or lowered somewhat by selecting an optimum tube size and wall thickness which may achieve the desired torsional stiffness. For example, the torsional spring rate may be increased by increasing the tube diameter and reducing the wall thickness of the tubing or by decreasing the shaft length. It may be possible to reduce somewhat the torsional rate by using smaller diameter tubing, provided that the maximum applied torque and critical speed limitations are met. A multi-piece driveline may be used to avoid the critical speed condition, and if needed, to reduce the torsional rate.

In more critical applications, it may be necessary to incorporate a device which will provide a relatively greater torsional flexibility and a more desirable degree of damping than is possible with a conventional tubular shaft. For such applications, a rubber element tube has often been used and consists of an inner tube and an outer tube joined by the torsionally resilient rubber element. This device can vary considerably in design and is discussed in detail in the chapter on *Vibration Control*. Whatever means is employed to achieve the desired torsional compliance and damping must meet the previously discussed design parameters for a specific application.

As previously indicated, it may be required that the driveshaft, of which the tube is only one member, provide a desired torsional rate or rigidity. Thus, it will be necessary to analytically estimate the overall torsional rigidity of the proposed shaft design. This 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 shaft. The tubular portion of

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the shaft length may be of constant diameter and wall thickness, or it may have a reduced diameter at one or both ends. Also, the shaft could be tubular for a portion of its length and solid for the remainder of the distance. Generally, a universal joint at each end and a splined device for allowing end motion would comprise a typical driveshaft assembly. In some driveshaft designs, this axial movement is accommodated by a splined connection inboard of the joints, while in other constructions it is located outboard of the joints. In still other designs, the axial motion also may be accommodated by one or both joints.

The torsional rigidities of the individual universal joints and the splined member can be determined with reasonable accuracy only by laboratory test. Torsional stiffnesses of the various tubular or solid sections of a driveshaft can be readily computed. The computation procedure for determining the overall torsional rigidity of a driveshaft is shown in *Appendix F*.

Torsional Strength and Twist

Circular Tubing

For a round tube, 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:

T = Torque, lb-in

Z_p = Polar section modulus, in^3

d_o = Outside diameter, in

d_i = Inside diameter, in

The angle of twist in radians per unit of length for a round tube is:

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

where:

G = Modulus of rigidity or modulus of elasticity in shear, psi

I_p = Polar moment of inertia of the cross section, in^4

For thin wall tubes, I_p reduces to a simpler approximate equation:

$$I_p = \frac{\pi d_m^3 t}{4}$$

where:

d_m = Mean diameter with respect to midpoint of tube wall thickness or the average of the outside and inside diameters, in

t = Wall thickness, in

Thus, the approximate angle of twist in radians per unit of length for a round tube is:

$$\phi = \frac{4T}{\pi d_m^3 tG}$$

Non-Circular Tubing

Based on the hydrodynamic fluid or shear flow, or membrane analogies for torsion, thin-walled tubes of any shape and with varying wall thickness can be analyzed rather simply for the magnitude of the shearing stresses caused by a torque applied to the tube. The shear stress at any point of the tube is:

$$S_s = \frac{T}{2At}$$

where:

A = Area enclosed by centerline or mean periphery of wall's contour, in^2

t = Wall thickness at that point of the tube, in

This equation may be used to compute the approximate values of shear stresses in tubular members under torsion if the thickness of the wall is small, variations in thickness are not abrupt, and there are no reentrant corners at the area of the tube under consideration. The thinner the wall of the tube, the more accurate will be the computed value of the stress. When this equation is applied to a thin wall round tube, the computed shear stress is approximately the same as the value obtained with the conventional torsion equation previously stated for round tubing.

Square and rectangular tubing is commonly used in agricultural implement power take-off driveshafts. Fig. 1 shows a rectangular tube cross section.

The torsional shear stress for the short sides is:

$$S_s = \frac{T}{2bht}$$

and for the long sides the torsional shear stress is:

$$S_s = \frac{T}{2bht_1}$$

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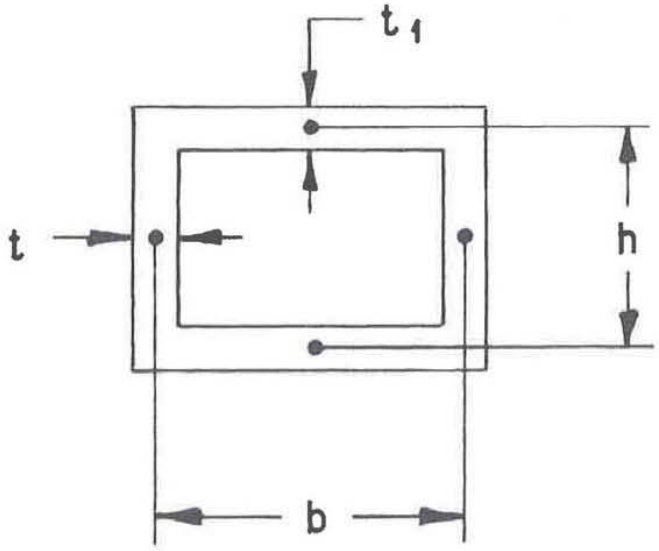


Fig. 1 — Rectangular tube

where:

b = Long side length between centerlines of the opposite wall thicknesses, in
h = Short side length between centerlines of the opposite wall thicknesses, in
t = Short side wall thickness, in
t_1 = Long side wall thickness, in

There will be higher shear stresses at the inner corners due to stress concentrations unless fillets of fairly large radii are provided. In practice, square and rectangular tubing are generally made with rounded corners.

By considering the strain energy of torsion, the angle of twist per unit of length for non-circular tubular members can be readily computed and is expressed as follows:

$$\phi = \frac{S_s L}{2AG}$$

where:

L = Length of centerline or mean periphery of wall's contour, in

This same equation can also be derived from the membrane analogy.

For the rectangular tubing section shown in Fig. 1, the angle of twist in radians per unit of length is:

$$\phi = \frac{bt + ht_1}{2tt_1b^2h^2} \cdot \frac{T}{G}$$

For a tube with a square cross section of uniform wall thickness, the angle of twist in radians per unit of length is:

$$\phi = \frac{T}{b^3tG}$$

where:

b = Length of each side, in

If desired, it is also possible to develop from a consideration of the strain energy of a twisted tubular member an equation for the effective polar moment of inertia of generalized tubular sections. This equation is:

$$I_p \text{ (eff.)} = \frac{4A^2}{\sum \frac{L}{t}}$$

which can be used in place of I_p in the conventional equation for the angle of twist in radians per unit of length as follows:

$$\phi = \frac{T}{GI_p \text{ (eff.)}}$$

This equation yields for rectangular and square tubing the same equations previously determined for these respective sections. When the strain energy of torsion is applied to thin wall round tubing, the equation for the angle of twist in radians per unit of length is the same as the approximate equation previously stated for a round tube.

It was previously indicated that square and rectangular tubing is commonly used in agricultural implement power take-off driveshafts. Its use in these applications is predicated by the desire to provide for a rugged, simple and economical telescoping means by mating square or rectangular tubing in sliding contact with square or rectangular bar shafts, respectively. The equations for computing square and rectangular bar shafts are shown in the chapter on *Splines and Axial Motion Devices*.

Tubing Material

In general, steel is used to make driveshaft tubing. However, depending on the previously described design parameters, it may also be possible to use other materials for the connecting tube. Typical round driveshaft quality steel tubing is made from cold rolled plain low carbon sheet or strip stock suitable for welding and which has a typical hardness range of R_b80-95.

At the minimum hardness of R_b80, the minimum ultimate tensile stress of the sheet or strip stock is 65 000 psi, and the minimum torsional shear yield stress of the round welded tubing is 38 000 psi. This yield stress value corresponds to 0.1% offset when using the offset method. This is the stress at which the material under load reaches a permanent set or plastic strain of 0.1%. The Johnson's apparent elastic limit (JAEL) method is also used to determine the yield stress. This method determines the stress at which the rate of deformation is 50% greater than the

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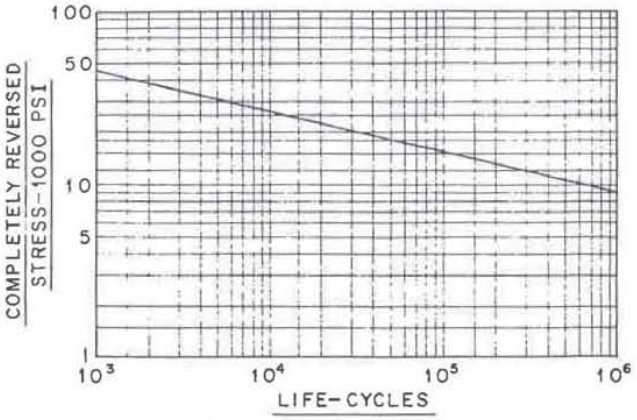


Fig. 2 — S-N curve for round welded steel tubing

initial rate of deformation and results in a lower yield stress value than the 0.1% offset method. Both of these methods used to determine the yield stress are described in detail in *Appendix L*.

Generally, steel tubing is attached to other driveshaft members by welding and depending on this welding operation, the resultant torque capacity can be as low as 80% of the value before welding. This reduction in torque capacity takes place in the heat affected zone adjacent to the weld. Therefore, a torsional shear design stress of 33 000 psi is usually recommended in most automotive applications. In extremely severe applications where shock torque may be induced by various loading modes described in the chapter on *Basic Driveline Requirements*, a suitable safety factor should be applied to this design stress as required.

It is also possible to predict the fatigue life of tubing at various stress levels with an S-N curve. Fig. 2 illustrates such a typical curve based on B_{10} fatigue test data for round driveshaft quality welded steel tubing at completely reversed torsional loading.

In some agricultural implement and various other power take-off applications, non-circular tubing such as square and rectangular tubes are often used. These non-circular tubes are generally made from round tubing by swaging to the desired cross sectional shape. The allowable design stress to be used for non-circular tubing will depend on the specific material used and the particular mechanical process used to form the tubing.

For any given application under consideration, the tube, whether round or non-circular in cross section, must be designed to safely transmit the maximum applied torque and perform safely at the intended maximum operating speed.

Welding

Steel tubing is generally manufactured in a tube mill from sheet or strip stock which is fed continuously through forming rolls. Then the butted tube passes through the electric resistance welding section of the mill and into the cutoff mechanism, which severs the tube into convenient lengths for handling. The weldment at the seam must be uniform and without discrepancies for the entire length of the tube. A non-destructive electrical test can be used to detect any weld defects.

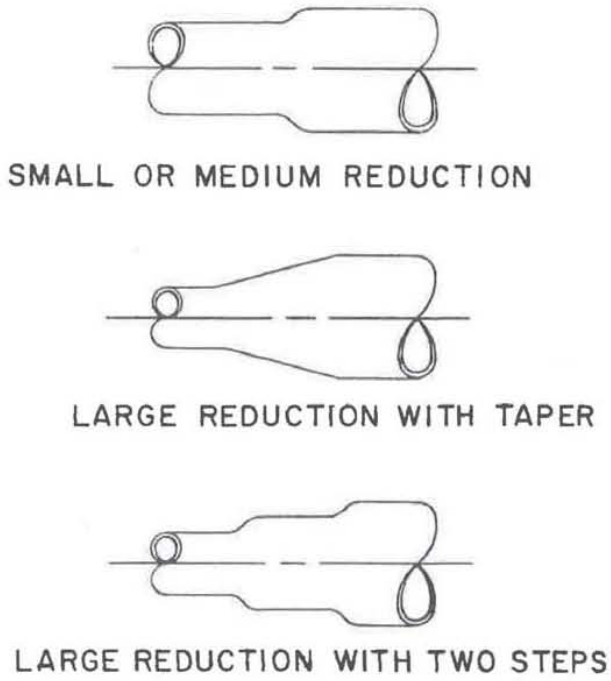


Fig. 3 — Typical swaging configurations

Swaging

A swaged tube is a tube with one or both ends having a smaller diameter than the middle portion. Generally, the amount of reduction is governed by the basic tube size and wall thickness as well as the swaging means employed. Usually a single step swage is used, however, if a significant reduction is required a double step configuration instead of a single step swage can be used if desired. These typical swaging configurations are illustrated in Fig. 3. The minimum swaged diameter permissible is that required to transmit the maximum driveline torque. While obtaining the desired vehicle clearances, a swaged tube can result in a minimum reduction of critical speed if the swaged length is not excessive.

Generally, it is desirable to swage a tube so that it can be attached to existing driveshaft components such as tube or weld yokes and thereby reduce manufacturing costs. Also, this provides for an overall reduction in driveshaft weight and minimizes driveline rotational inertia effects.

Size and Tolerances

Due to generally rigid design requirements, the dimensional tolerances for driveshaft quality welded steel tubing differs from conventional mechanical tubing. Table 1 lists typical tolerances for the more commonly used sizes.

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TABLE I — TYPICAL DRIVESHAFT TUBING TOLERANCES

Tube Diameter, In	Tolerances				
	Wall Thickness, In	Outside Diameter, In	Ovality T.I.R. Max., In	Straightness T.I.R. Max. Runout,* In	Wall Thickness,** In
1.750-1.875	0.109-0.120	±0.0035	0.005	0.010	±0.003
1.750-1.875	0.083-0.095	±0.003	0.004	0.008	±0.0025
1.750-1.875	0.065	±0.004	0.005	0.008	±0.0025
2.000-2.250	0.083-0.134	±0.003	0.004	0.010	±0.003
2.000-2.250	0.065	±0.004	0.005	0.010	±0.003
2.281-2.500	0.095-0.134	±0.004	0.005	0.010	±0.003
2.281-2.500	0.065-0.083	±0.004	0.006	0.010	±0.003
2.531-3.000	0.095-0.134	±0.004	0.005	0.012	±0.003
2.531-3.000	0.065-0.083	±0.005	0.006	0.012	±0.003
3.031-3.500	0.095-0.134	±0.005	0.007	0.012	±0.003
3.031-3.500	0.065-0.083	±0.006	0.009	0.015	±0.003
3.531-4.000	0.095-0.134	±0.007	0.010	0.020	±0.0035
3.531-4.000	0.065-0.083	±0.010	0.015	0.025	±0.0035
4.031-5.000	0.095-0.238	±0.015	0.013	0.025	±0.004
4.031-5.000	0.065-0.083	±0.020	0.040	0.030	±0.005
5.031-6.000	0.095-0.238	±0.020	0.030	0.030	±0.0055
5.031-6.000	0.065-0.083	±0.025	0.040	0.040	±0.005

The gage variation around the circumference of any one tube exclusive of the weld is 0.003 in. Driveshaft tubing is usually supplied flash-in with the ends sized for a sufficient distance on each end to permit the proper insertion of the mating member; or the flash may be removed for the entire length of the tube to a maximum height of 0.010 in.

NOTE: The above table covers lengths up to and including 60 in.
*Excluding Ovality
**Strip Stock
Sheet stock is also used and has wall thickness tolerances approximately double the strip stock tolerances shown.

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Splines and Axial Motion Devices

Chapter 5

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Chapter 5
Splines and Axial Motion Devices

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

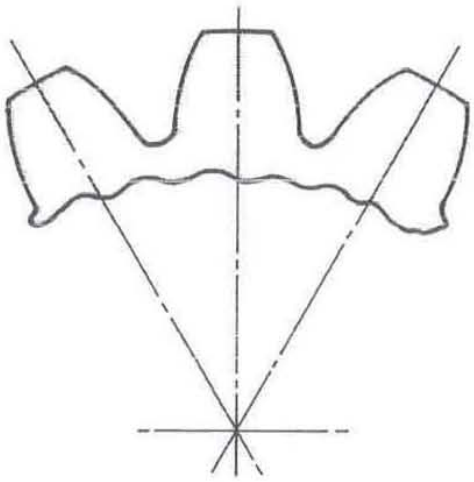


Fig. 1 — Involute spline

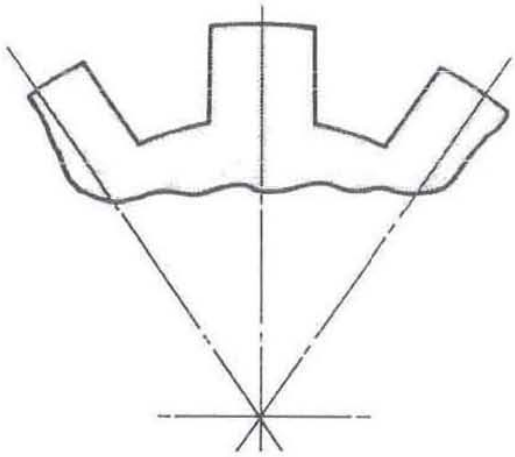


Fig. 2 — Parallel side spline

Introduction

A spline is a machine element consisting of integral keys (spline teeth) or keyways (spaces) equally spaced around a circle or portion thereof. It is generally considered as having a pressure angle less than 45 deg, and can be either straight or helical. A serration is the same as a spline except generally considered as having a pressure angle of 45 deg or greater. It can also be straight or helical.

The basic function of a spline is to transmit the torque requirements of the driveline installation from one drive member to another. Also, in many applications it must be capable of allowing for length changes due to vehicle geometry motions. Although torque is the principal loading mode on driveline splines, forces due to secondary couple effects resulting from universal joint action and shaft dynamic unbalance must not be neglected. This is especially true for short driveshafts operating at relatively high joint angles. Therefore, the spline section must have the ultimate strength necessary to safely transmit the maximum applied torque, including shock effects and other forces, and have a stress level which will provide for adequate fatigue life. The maximum torsional loads which can be developed by various loading modes on a driveshaft or any of its components, are discussed in the chapter on *Basic Driveline Requirements*. Also, in a spline which is required to slip under high torque loads, especially, severe galling or scoring conditions can occur due to axial thrust forces which can be relatively high. Thus, care must be taken to select splines with a sufficiently large diameter which will result in a satisfactorily low level of contact pressure between spline teeth. Typical alternate design procedures are to apply a low friction surface coating on the spline or incorporate a rolling element spline mechanism. This results in lower thrust forces on end supporting structures and bearings.

The design of various spline tooth forms has progressed through an evolution. The motivation for this progress was the constant need for improved function, lower cost, ease of manufacture, clearer specifications, and improved inspection concepts which ensure that the assemblies will function to the design intent. In vehicle drivetrain applications, the two most commonly used spline tooth forms have been the involute spline shown in Fig. 1 and the parallel side spline shown in Fig. 2. Typical methods of manufacturing various spline forms have been broaching, milling, shaper cutting, hobbing, swagging, cold roll forming, and grinding.

Involute Splines

An involute spline is a spline with teeth which have an involute profile and is the spline form most generally used because it offers a variety of advantages. It has a relatively high torque capacity due to its strong tooth configuration and is self-centralizing. The involute spline is economically manufactured and can be efficiently inspected. Its mating characteristics can be closely predicted and controlled.

The various involute spline tooth proportions with their associated tolerances are shown in the standard on *Involute Splines and Inspection, ANSI B92.1 – 1970*. The two most commonly

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used fits described in this standard are:

1. Major diameter fit
2. Side fit

In the major diameter fit, the mating members contact at the major diameter for centralizing. The sides of the teeth act as drivers and the minor diameter is designed with clearance. This type of fit requires grinding which increases cost and therefore should not be specified unless necessary. It is usually used where the shaft spline has excessive runout due to heat treatment distortion and the mating part is required to operate with a minimum runout. This type of fit is often used between the driveshaft slip yoke and transmission output shaft of passenger cars.

In the side fit spline, the mating members contact on the sides of the teeth at the pitch diameter only with the major and minor diameters designed with clearances. The teeth sides act as drivers and centralize the mating splines.

The standards on splines include pressure angles of 30, 37.5, and 45 deg. Splined members with these pressure angles all have the same effective fit (backlash) characteristics. The pressure angle should be chosen to satisfy function and economy. If the assembly requires that the members slide under load or if one or both of the parts have a thin-walled tubular section, then function dictates the use of a 30 deg pressure angle to reduce sliding effort and bursting forces.

If the member is to deliver torque only, and if bursting forces are not detrimental, then economy would dictate the use of a 45 deg pressure angle. Cold roll forming 45 deg pressure angle tooth forms result in far greater tool life than for 30 deg pressure angle teeth.

The 37.5 deg pressure angle spline is a compromise between the 30 deg and 45 deg pressure angle tooth forms. Its application is suggested where the shaft material is too hard to consider use of a 30 deg pressure angle and function prohibits use of a 45 deg pressure angle.

Involute splines can also be classified as:

1. Sliding or slip splines
2. Zero backlash (interference fit) splines

Sliding or slip splines are commonly used to provide for relative axial or end motion capability between connecting driveshafts or between a driveshaft and another drivetrain member. In passenger car and light-duty truck slip yoke applications for drivelines, the ratio of the average engaged spline length to the spline pitch diameter has been in the typical range of 2.25-2.75. Fig. 3 shows the axial force characteristics of a typical sliding spline.

Zero backlash (interference fit) splines are commonly used to fixedly attach a driveshaft to another driveshaft or drivetrain member. The ratio of the engaged spline length to the spline pitch diameter has been in the typical range of 0.80-1.10. Generally, a ratio of approximately 1.00 is recommended for equal strength materials.

The two most common methods of achieving a zero backlash (interference) fit have been to broach the internal spline with a straight design and by specifying either a taper or a helix angle on the external spline. In both of these external spline designs, the mating components show no interference in the plane of rotation at the start of assembly and thus assemble about ⅓-½ of the

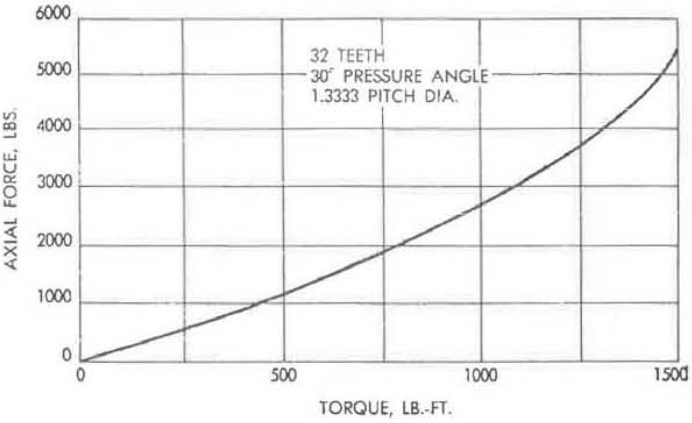


Fig. 3 — Involute sliding spline axial force characteristics

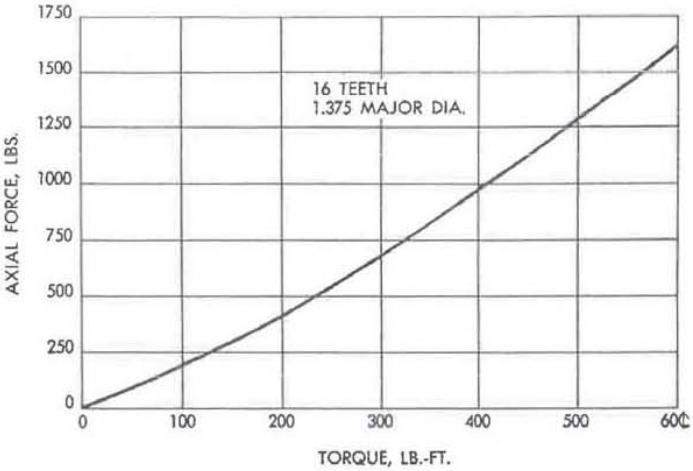


Fig. 4 — Parallel side sliding spline axial force characteristics

engagement length before interference is encountered. Depending on the application, this interference must be sufficient to prevent relative rotational movement under high torque loads and thus cause fretting failure of the splined connection.

For the tapered external spline, the circular tooth thickness increases along the spline length and results in a tapered spline. In typical applications, a change in circular tooth thickness of 0.002-0.006 in/in of length has been satisfactory.

For the external spline with the helix angle, the standard procedure has been to provide a helix angle that will give 0.004-0.006 in windup within the effective length of engagement. This will usually result in 0.002-0.004 in effective interference. Typical helix angles in the range of 0 deg 15 min to 0 deg 40 min have been used.

Parallel Side Splines

A parallel side spline is a spline in which the sides of the driving teeth are parallel. The number of teeth used in parallel side spline are 4, 6, 10, 16 or 32, with 10 and 16 teeth most commonly used.

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This spline form is used extensively in off-highway and agricultural equipment.

An important disadvantage of this tooth form is that it is not tool oriented. For example, a hob made for a 10 tooth spline can only cut a spline with ten teeth and only to the diameter and tooth thickness for which it was designed; whereas, an involute spline hob can cut many different involute splines.

When torque is applied to the parallel side spline, it rotates into driving position with its driven member and initial contact is made at the tip of the external spline teeth only. This usually brings about early wear in that area and by the time the assembly has worn into a full flank bearing condition, an excessively loose fit will result.

Also, the parallel side spline is often designed to permit very small root radii. The resulting high stress raisers often prove to be the starting point of shaft failure.

Parallel side spline can be either sliding or permanent fit splines and are used in applications similar to the usage described for sliding or zero backlash (interference fit) involute splines. In passenger car and light-duty truck slip yoke applications for drivelines, the ratio of the average engaged spline length to the spline major diameter has been in the typical range of 2-3. In slower speed applications this ratio has been smaller. Fig. 4 shows the axial force characteristics of a typical sliding spline. For permanent fit splines, a ratio of the engaged spline length to the spline major diameter of approximately one is generally recommended for equal strength materials.

Typical tolerances for sliding and permanent fit splines at the shaft support bearing location are shown in Tables 1 and 2 respectively. These can be used as a guide in typical design applications. Also, the *SAE Handbook* shows tolerances for parallel side splines with various numbers of teeth. While sliding spline tolerances always have a loose or clearance condition in the tolerances, the permanent fit spline tolerances range from loose to tight (interference).

TABLE I — SLIDING SPLINE TOLERANCES OF EXTERNAL AND INTERNAL SPLINED SLIP YOKE AT SHAFT SUPPORT BEARING

	Tolerances, In
Major Diameter	0.0025L-0.0010L
Side Fit	0.0040L-0.0010L

L = Loose (Clearance)
Clearance before phosphate type coating on external splined member.

TABLE 2—PERMANENT FIT SPLINE TOLERANCES OF EXTERNAL SPLINED NON-SLIP SHAFT WITH END YOKE AT SHAFT SUPPORT BEARING

	Tolerances, In
Major Diameter	0.0040L-0.0000L
Side Fit	0.0015L-0.0015T

L = Loose
T = Tight (Interference)

Stress Equations

The spline stress analysis is performed by computing the principal stresses involved using appropriate equations. The nomenclature used in the various equations is illustrated in Fig. 5 and also described in the following nomenclature list of symbols.

NOMENCLATURE

- T = Torque, lb-in
- Z_p = Polar section modulus, in^3
- D = Pitch diameter, in
- D_o = External major diameter, in
- D_{ri} = Internal major diameter, in
- D_{re} = External minor diameter, in
- D_i = Internal minor diameter, in
- L = Length of spline engagement, in
- N = Number of teeth
- t = Circular tooth thickness, in
- ϕ = Pressure angle

Although the involute spline form is shown, the equations can also be applied to parallel side splines. For the principal stresses, the appropriate equations are expressed as follows:

1. Torsional Shear Stress for Basic Shaft —
The greatest single factor in a spline selection is the basic shaft diameter required to safely transmit the maximum applied torque of a given application. For a basic shaft where this diameter corresponds to the external spline minor diameter D_{re}, the torsional shear stress for a solid round shaft is:

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

The external minor diameter D_{re} of the shaft can be determined by applying an acceptable stress level for the proposed material keeping in mind heat treatment, stress concentrations, and the fatigue life expected.

For a hollow round shaft having external teeth with a bore of D_x, the torsional shear stress is:

$$S_s = \frac{T}{Z_p} = \frac{16\ T\ D_{re}}{\pi\ (D_{re}^4 - D_x^4)}$$

For a hollow round hub having internal teeth with an outside diameter D_{xo}, the torsional shear stress is:

$$S_s = \frac{T}{Z_p} = \frac{16\ T\ D_{xo}}{\pi\ (D_{xo}^4 - D_{ri}^4)}$$

Should a restraint limit the outside diameter of the hub D_{xo}, it will be necessary to start with this restraining diameter and by applying a suitable stress level, the major diameter D_{ri} of the internal spline can be determined.

In these computations, the basic shaft diameter required at the spline section of the shaft only is considered. However, in a typical shaft configuration with a varying geometric design along its length, sections other than the spline section may dictate the minimum diameter permissible at a particular shaft section and thus affect spline size. This is due to the stress concentration

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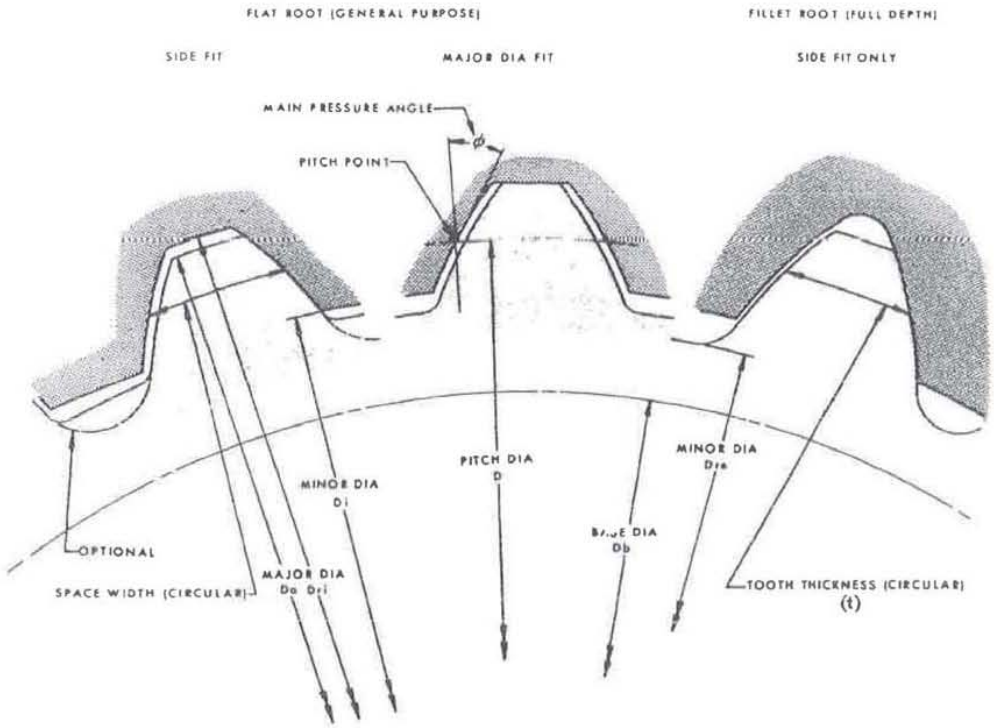


Fig. 5 — Spline nomenclature symbols and terms

effects resulting from common stress raisers such as shoulder fillets, spline runout, and undercut relief diameters. Also, in axle driveshafts, in particular, retaining ring and boot seal grooves for universal joints must be considered. These varying geometric features of a typical shaft are also discussed in the chapter on *Axle Driveshafts*, while *Appendix D* shows values of stress concentration factors for some common stress raisers.

2. Shear Stress in Spline Teeth —
When assuming that the teeth shear at the pitch line and with 100% teeth in contact, the shear stress is:

$$S_s = \frac{2T}{DtLN}$$

When assuming that the teeth shear at the minor diameter D_i of the internal spline and with 100% teeth in contact, the shear stress is:

$$S_s = \frac{2T}{D_i(t_x)\ LN}$$

where:

t_x = Tooth thickness at D_i

The above equations may be modified according to the application and tolerances where there is justification that all teeth will not be in contact.

3. Compression Stress in Spline Teeth —
When assuming 100% teeth in contact, the compressive stress is:

$$S_c = \frac{4T}{D\ (D_o - D_i)LN}$$

This equation also may be modified according to the application and tolerances where there is justification that all teeth will not be in contact.

4. Hoop or Bursting Stress in Hub —
The hoop or bursting stress induced in a hub of diameter D_e surrounding an internal involute spline is:

$$S_t = \frac{2T \tan \phi}{\pi\ D\ (D_e - D_{ri})L}$$

Material Selection

Based on a stress analysis of the proposed splined connection, suitable materials and heat treating processes can be selected. This selection of materials and heat treatment for the splined members depends not only on the ultimate strength, fatigue strength, wear life, and other functional requirements, but also on economical processing advantages.

Internal splined members can be made from plain or resulphurized medium carbon steel forgings or bar stock, which are heat treated to a typical hardness range of BHN 228-269. If they

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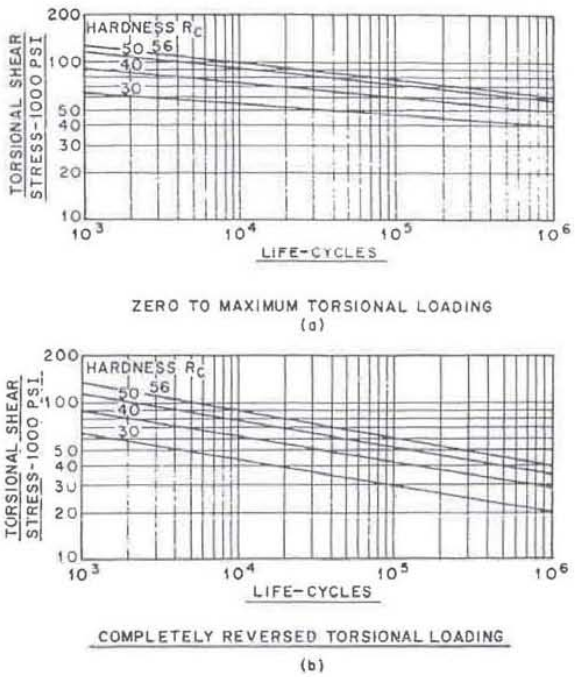


Fig. 6 — S-N curves for spline

are not subsequently welded to another driveline member, internal splined components such as slip yokes or end yokes are frequently made from either pearlitic malleable or nodular iron castings. These castings when heat treated to a typical range of BHN 241-269 for pearlitic malleable iron and BHN 241-285 for nodular iron allow the machining of components having the equivalent strength of forged steel members. Also, in some applications, formed components made of plain low carbon steel are used without subsequent heat treatment.

As in the case of internal splined members, external splined components assume various design configurations. Since the spline is usually designed to be stronger than the shaft, the most common type of failure is a fatigue failure in torsional shear in the area where the spline runout or blending with the shaft occurs. In order to achieve the necessary torsional strength of the basic shaft size for an external splined section, these components generally must be either through or surface hardened, depending on the choice of material. 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 or resulphurized medium carbon or medium carbon alloy steels can be induction hardened. The induction hardening process allows the selection of a relatively high case depth and a typical surface hardness range of R_c44-54 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. Both surface hardening procedures can provide for significant improvements in strength over the through hardening process with induction hardening being capable of yielding the greater increase in strength and fatigue life. With regard to spline tooth distortion (generally growth of tooth thickness) of surface hardened components, it has usually been found that induction hardening yields the least distortion, while carburizing results in the greatest distortion. This difference in distortion is primarily due to the characteristics of these two heat treatment processes.

The limiting stress levels for the external and internal splined components can be determined for the proposed material depending on the heat treatment, stress concentrations, and the expected fatigue life. The ultimate tensile and shear stresses, as well as the tensile and shear yield stresses, can be estimated for through hardened steel as shown in the chapter on *Axle Driveshafts*.

When a shaft and external splined member is carburized and 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 on the shaft section size-to-case depth ratio obtained and the geometry of the components. For surface hardened splined shafts, the governing factors on the allowable levels of nominal shear stress are discussed briefly in the chapter on *Axle Driveshafts*. The actual estimated stress levels to be used for surface hardened components should be based primarily on previous static ultimate and yield torque test data of geometrically similar components.

It is also possible to predict the fatigue life of a spline at various stress and hardness levels with the use of S-N curves. Such curves, based on B_{10} fatigue test data, are shown in Fig. 6. Fig. 6(a) illustrates typical curves for zero to maximum torsional loading, while Fig. 6(b) shows typical curves for completely reversed torsional loading. Zero to maximum loading conditions are most often found in power take-off applications and in cases where an engine is connected to a transmission with a driveshaft. The completely reversed loading condition is most commonly found in vehicle drivelines, which are generally located between the transmission and differential or final drive unit. Also subjected to this reversed loading characteristic are axle driveshafts.

Surface Treatments

Various surface treatments, including some proprietory types, have been applied to sliding splines to reduce the axial slip force characteristics and provide for protection from corrosion. Usually, the surface treatment is applied only to the external splined member after heat treatment.

The most commonly used surface treatment has been a fine grain conversion coating of manganese iron phosphate where an average thickness of 0.0001-0.0003 in per side has been generally added to the external spline. In parallel side spline constructions, the major diameter can be finish ground to produce the desired diametral clearance and centrality. Besides reducing the friction and providing for protection from corrosion, the phosphate coating acts as a "wear-in" or "break-in" agent for the spline. The coating provides for separation of steel-on-steel sliding surfaces and is porous enough to hold and retain some lubricant. Imperfections in the surfaces are sheared away without promoting permanent damage. Therefore, initial galling and wear problems are

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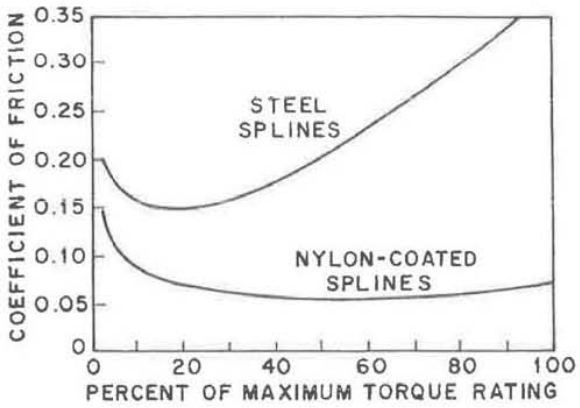


Fig. 7 — Comparison of coefficient of friction versus torque between lubricated steel and nylon coated splines

forestalled. Problems associated with the use of phosphate coatings are the additional tolerances and clearances required for the material, and in due time, the coating wears off.

In order to obtain a greater reduction of axial slip force, a nonmetallic coating such as nylon has been used. Nylon is one of the strongest and stiffest of the thermoplastics and is well known for its good friction and wear properties. It is tough and abrasion resistant and has a high resistance to oils and a high heat distortion temperature. It can be used with friction reducing fillers such as teflon, molybdenum disulfide, and graphite. The nylon coated spline requires no design changes other than a slight reduction in thickness and major diameter of the external steel spline. It has good lubricant retention characteristics. It eliminates "spline grunt", offers some vibration dampening, and because of the load equalizing effect of the relatively elastic nylon coating, offers some increase in fatigue life of the spline. Friction tests for steel on nylon with lubricant show a significant reduction of the coefficient of friction for the same combination without lubricant. Therefore, lubricant is recommended with nylon coated splines.

The reasons for the improved performance of the nylon coated spline can be summarized as follows:

1. The dissimilar materials in contact prevent galling and seizing.
2. The slightly orange peel type surface of the material acts as closely spaced pockets for the lubricant. Because of the relatively low modulus of elasticity of the material, as compared to steel, the lubricant becomes entrapped and sealed in these pockets when the material is compressed by torque loads.
3. The difference between the coefficient of friction of steel and nylon increases with torque because of the tendency of the high spots on the steel surfaces to break through the lubricant film. The nylon conforms elastically to permit a greater actual contact surface without breaking through the lubricant film. Fig. 7 shows a comparison of the coefficient of friction versus torque for both plain steel and nylon coated splines with lubrication.
4. The lower coefficient of friction of the nylon causes less heat buildup within the splined members and prevents rapid deterioration of the lubricant. Fig. 8 shows a comparison of the spline temperature versus torque in terms of spline pressure between the steel and nylon coated splines.

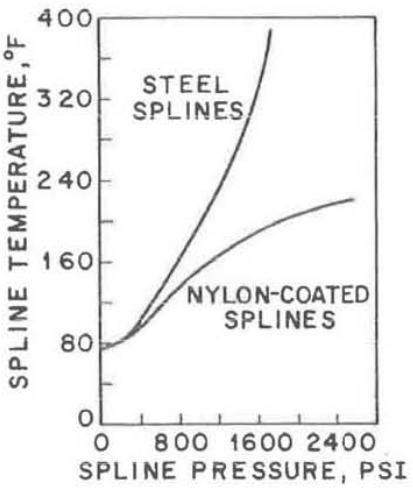


Fig. 8 — Comparison of spline temperature versus spline pressure between steel and nylon coated splines

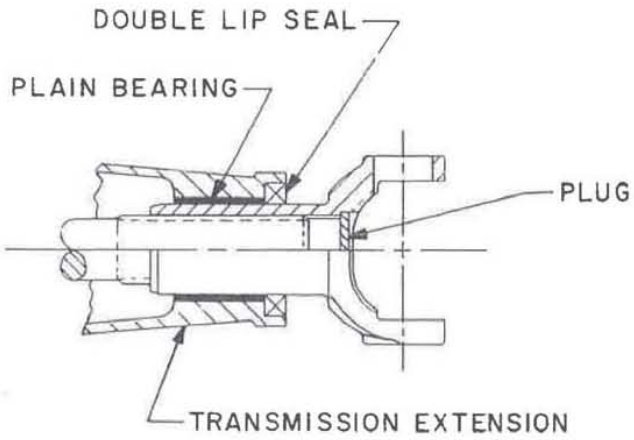


Fig. 9 — Conventional slip yoke sealing design at transmission output shaft

Lubrication and Sealing

In driveline installations, sliding splines are used extensively. A sliding spline can be described as a type of positive displacement pump. Therefore, these spline constructions must be provided with sealing means and be either lubricated for life at assembly, or a lubrication plug or fitting must be included for periodic replenishment of the lubricant.

The use of splines encompasses a very wide range of applications and the selection of the proper lubricant depends on the torque load, axial motion, and environmental characteristics of

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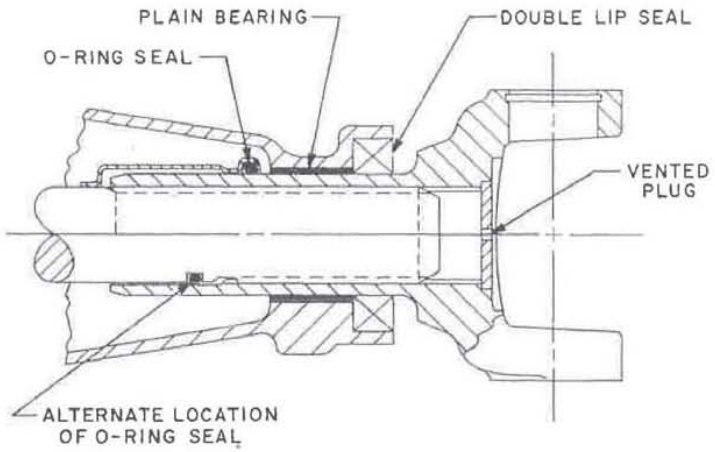


Fig. 10 — Alternate slip yoke sealing designs at transmission output shaft

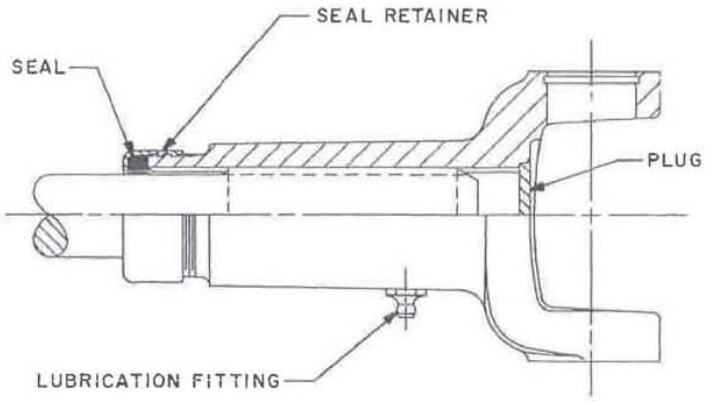


Fig. 11 — Slip yoke sealing design at hub end

the specific application. A lubricating grease having an NLGI grade number 2 is extensively used due to its ability to withstand relatively heavy loads or extreme pressures. A typical lubricant of this type generally consists of a refined mineral oil with an appropriate soap thickener such as lithium, calcium complex, or sodium. Extreme pressure additives with graphite or molybdenum disulfide being common materials, and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

Typical basic performance requirements of a lubricating grease which must be met and its various properties are described in the *SAE Handbook* under *Automotive Lubricating Greases*. In addition to providing for optimum spline life by reducing friction and wear at all bearing surfaces, the lubricant used must perform satisfactorily in all environmental conditions of the spline application. Also, in some vehicle transmission designs, the transmission fluid may serve as the spline lubricating means.

The method of sealing employed for a splined connection generally depends on its specific design as well as its application. In the case of an internal splined slip yoke used at the transmission output shaft of a vehicle, a double lip type of seal has been generally used on the ground outer bearing and sealing surface of the splined hub as shown in Fig. 9. Also, in some applications, a boot seal has been employed along with the lip seal to protect it from dust, dirt, and other foreign matter. Usually the body end of the slip yoke hub is sealed with a plug to retain the spline lubricant. When transmission fluid serves as the lubricating means, a spline tooth in the slip yoke is usually removed to facilitate the free back and forth flow of the fluid during stroking.

In some transmission constructions, an O-ring may also be required in addition to the lip seal, to seal the transmission fluid. Depending on the particular design, this O-ring seal may either contact the ground outer bearing and sealing surface of the hub or it may be positioned in a groove on the transmission output shaft and seal in the ground counterbore end of the hub as shown in Fig. 10. The use of the O-ring seal necessitates a vented plug instead of a solid plug in the slip yoke. In these sealing designs employing an O-ring seal, the spline is lubricated with grease at assembly.

When the internal splined slip yoke does not have a ground outer bearing and sealing surface, the sealing means is incorporated at the hub end as shown in Fig. 11. This construction is often used in one piece driveshafts having inboard slip or in multi-piece driveshafts adjacent to the shaft support bearing. This seal may be a felt, cork neoprene or synthetic rubber design usually held in position with a cap or seal retainer which is threaded onto the hub or held by other means. The sealing is usually accomplished on

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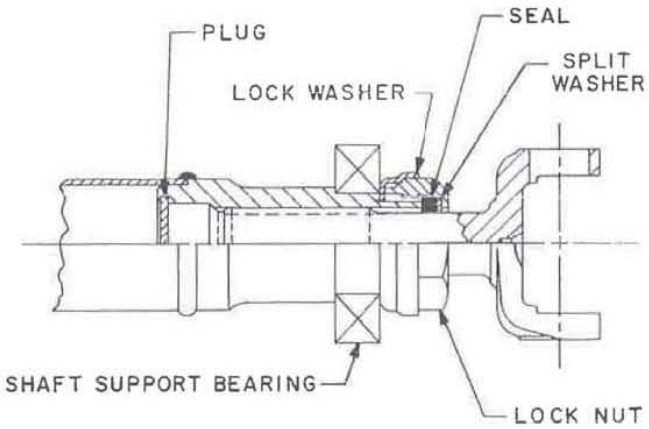


Fig. 12 — Seal design for external splined slip yoke

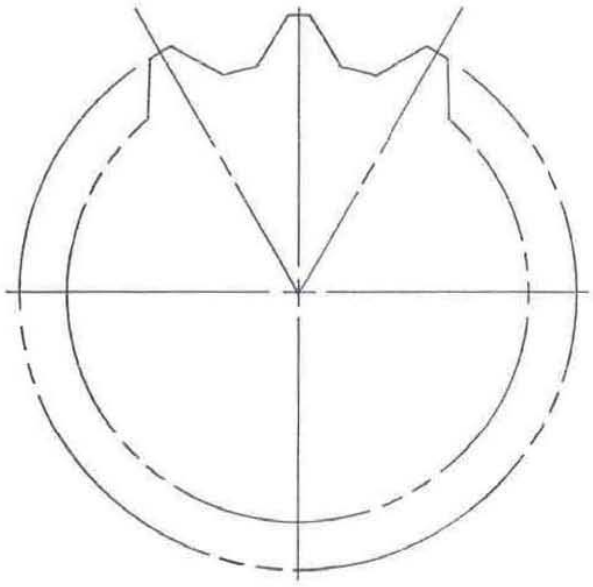


Fig. 13 — Truncated star serrations

the shaft surface adjacent to the external spline. However, in some designs the felt or cork neoprene type seal may also be formed to fit the external spline tooth contour where the sealing is thus accomplished.

This hub end sealing method described for the internal splined slip yoke can also be used with the external splined slip yoke and internal splined slip shaft construction shown in Fig. 12. This construction is also often used in inboard slip one piece or in multi-piece driveshafts.

Miscellaneous Serrations, Splines and Axial Motion Devices

Steering Shaft Serrations and Splines

In low torque and very slow speed applications such as universal jointed steering shafts, wide usage is made of modified or

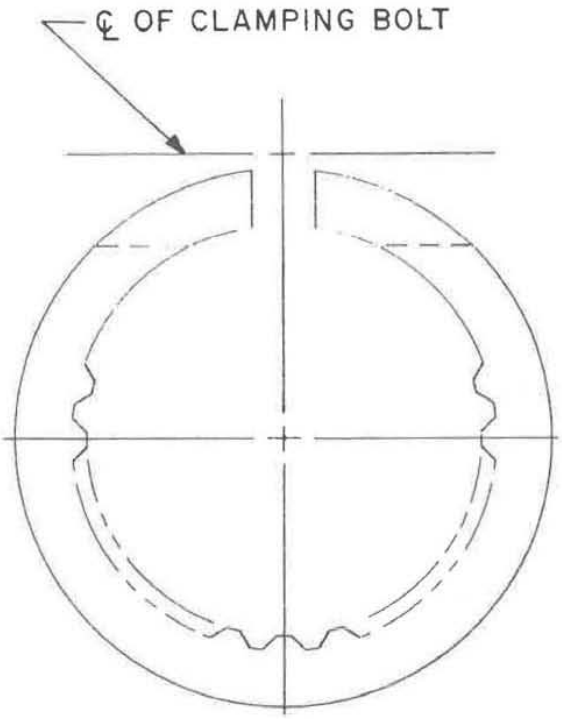


Fig. 14 — Hub end view of partial circle of internal truncated star serrations

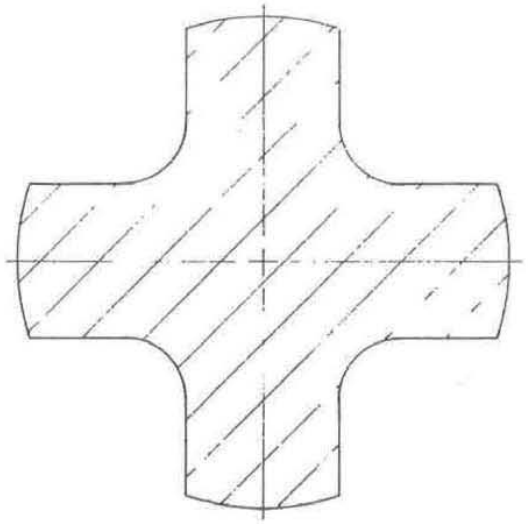


Fig. 15 — Special four tooth parallel side spline

truncated star type serrations with straight sides to connect mating components. Generally, a full circle of external serrations are used on the shaft member as shown in Fig. 13. When the mating member is held to the shaft with a clamping bolt or with a clamp and bolt, which are common practice, then only a partial circle of internal serrations are used as shown in Fig. 14. This permits the partial serrated hub of a universal joint yoke, housing, or flange to be attached tightly to the shaft and thus eliminate the circumferential and axial lash. Usually, the number of serrations used per circle is quite large and may be in the typical range of 36 or 48.

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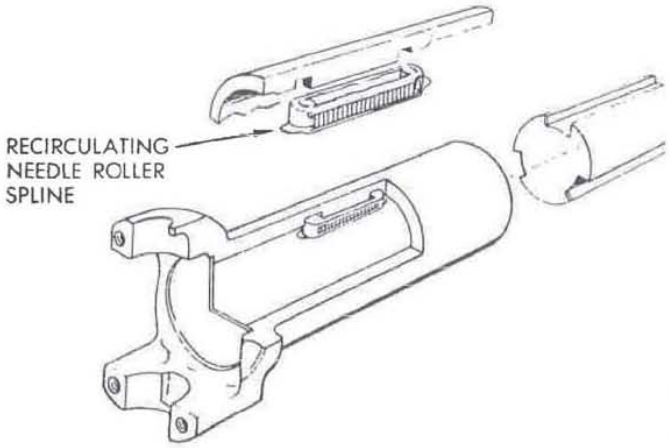


Fig. 16 — Slip yoke with recirculating needle roller spline

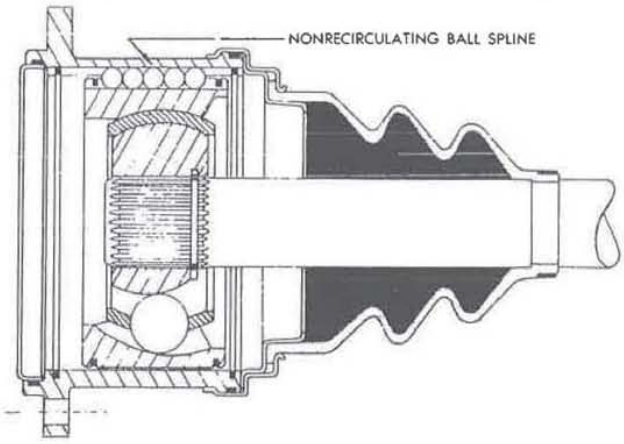


Fig. 17 — Ball spline Rzeppa universal joint

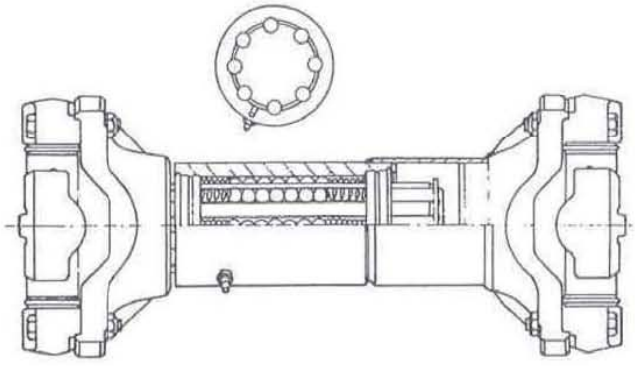


Fig. 18 — Nonrecirculating ball spline with spring return feature in a driveshaft

In some vehicle steering shaft applications, sliding involute spline are used to accommodate relative movement between the body and frame, for example, when tilting the cab of a tilt cab truck. Also, in other vehicles various special spline forms other than conventional spline are used to provide for end motion capability. Fig. 15 shows such a spline form in which the four tooth external spline is coated with nylon to provide for a lash and

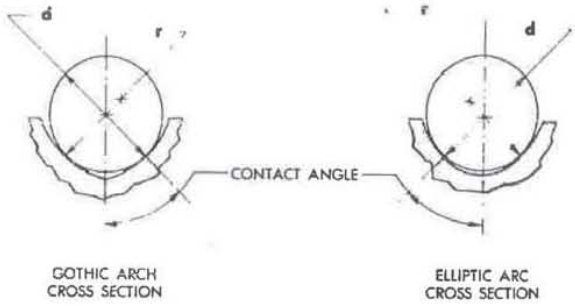


Fig. 19 — Typical ball spline ball groove forms

noise free connection. With a steel surface on one component mating with a nylon coated surface on the other component, fretting damage to the spline is eliminated by the dissimilar materials during small movements between the body and frame of the vehicle.

These sliding spline designs must be provided with sealing means as well as proper lubrication. Because of the less severe nature of steering shaft applications, the seal constructions can be the simpler versions of those used in drivelines. The lubricant selected can be the same as or similar to that used in driveline sliding spline applications, except with the addition of tacking agents in some applications to reduce noise resulting from clearances.

Rolling Element Splines

Various rolling element spline mechanisms have been designed to reduce the axial force required to slip a spline while transmitting torque. While basically higher in cost, they offer numerous advantages which can justify their usage.

They show inherently much lower coefficients of friction not only while moving, but also on breakaway or start, and result in lower thrust pulses and forces on connected components and bearings. The rolling action avoids galling and subsequent interference of mating sliding components and the resultant extremely high thrust forces. Therefore, drivetrain component failures are less apt to occur and it may even become possible to consider lighter components in the connected units. Generally, these devices can be lubricated for life at assembly and not require further attention or maintenance. However, in some applications relubrication means are provided.

Rolling element splines may employ rollers, needle rollers, or balls as the rolling elements. The roller and needle roller types can be either of the recirculating or nonrecirculating design with the recirculating type allowing a greater length of pure rolling action depending solely on the length of the raceways on the internal or shaft member. Fig. 16 shows a recirculating needle roller spline adapted to a driveshaft slip yoke. The ball spline can also be either of the recirculating or nonrecirculating configuration. Fig. 17 shows a nonrecirculating design used in a ball spline Rzeppa universal joint, while Fig. 18 shows another nonrecirculating design with a return spring feature used in a driveshaft application. In a recirculating spline design, the rolling elements theoretically can always have true rolling motion regardless of the amount of axial stroke required by the rolling element spline

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mechanism. During pure rolling, the rolling elements will move one half of the distance of the relative axial motion of the two raceway members of the spline mechanism. Therefore, when designing a nonrecirculating spline device, sufficient raceway length must be provided, if possible, to allow rolling motion of the rolling elements. Even if adequate raceway length is provided, some skidding or sliding of the rolling elements will be encountered. The location of the individual rolling elements relative to each other and the axial travel stops in both directions of a given raceway, as well as their location or orientation relative to the rolling elements in the other raceways, cannot be controlled without a device such as a separator element or a return spring feature shown in Fig. 18.

Generally, the ball groove form in the nonrecirculating design ball spline is either the Gothic arch or elliptical arc cross sectional configuration as shown in Fig. 19. A contact angle of approximately 45 deg is usually used. The 45 deg contact angle provides for best dimensional control and results in optimum life and performance due to minimum ball and groove contact stresses and all operating conditions. Also the rotational lash characteristics can be controlled without changing the contact angle. From the cross section of the ball groove forms, it can be seen that there is clearance space for small foreign particles that may enter the ball races. Purging of contaminates from the races is thus facilitated by the Gothic arch or elliptical arc groove form. The groove radius-to-ball diameter or r/d ratio is generally in the typical range of 0.51-0.55. This ratio is also called the conformity ratio of the grooves. When the Hertz contact stress is not of importance, a straight sided 90 deg V-shaped groove has also been used in certain applications. In recirculating type ball splines, the Gothic arch groove configuration is generally used, and depending on the particular design of the external and internal members, often only a half or one side of the groove is used. In general, it is desirable to use a ball-to-groove conformity which will provide the minimum contact area required for the permissible Hertz contact stress level. The smaller contact area resulting from less conformity or wrap around of the groove on the ball reduces sliding and scuffing between the ball and groove produced by the larger and longer contact area, and thus improves ball wear life even though the contact stress is higher.

The number and size of the rolling elements required is generally determined by the torque load transmitted as well as the type of material and surface hardening procedure employed for the raceways. The permissible Hertz contact stress in the raceways, including the associated computation procedure, is shown in *Appendix A*.

In the case of nonrecirculating rolling element splines, all rolling elements are active load carrying elements. However, in recirculating designs, only a fraction of the total number of rolling elements (less than one half) are active load carrying elements in a given direction of rotation. The raceways of typical rolling element spline members are surface hardened to provide the necessary surface hardness and case depth for optimum wear life, which requires a surface hardness of R_c60 minimum and an adequate case depth as determined by the contact load requirement of the application as explained in *Appendix A*. Low carbon alloy steel components can be carburized, while plain medium carbon steel components allow induction hardening to meet the functional and durability requirements. Usually, the raceways are finish ground after surface hardening. However, in some designs, raceways have also been finish broached or finish formed prior to surface hardening. The finish lapped rolling elements are generally made of chromium steel with balls hardened to a typical range of R_c62-65, while rollers and needles are hardened to a typical range of R_c60-64.

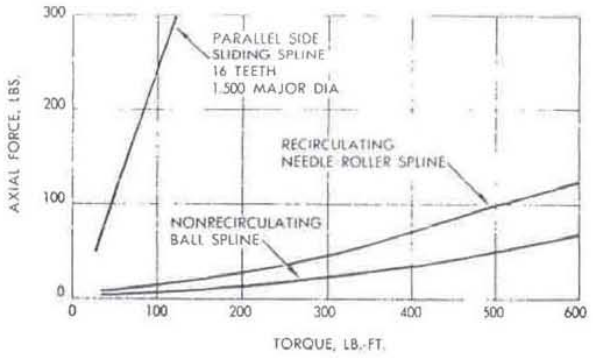


Fig. 20 — Comparison of rolling element spline axial force characteristics with a sliding spline

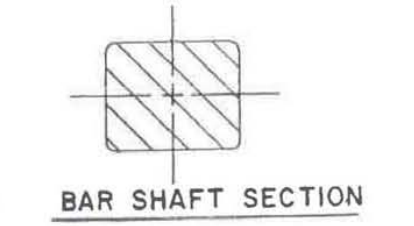
In a given design, it is desirable to have a minimum of three raceways to provide for adequate centralization between the driving and driven members. Also, in the axial direction it is desirable to have an adequate length necessary for optimum design. The length of the rolling element assembly must in all cases be long enough to provide for adequate stability, otherwise other stablizing means must be provided. Therefore, in many designs more rolling elements than those necessary to transmit the required torque load have been used. Also, in some constructions non-load carrying spacer elements have been employed to achieve the same result.

Rolling element spline constructions must be provided with sealing means and lubrication. Boot seals are commonly used as well as other typical seal designs employed in conventional sliding splines. Also, in some ball spline designs a wiper type seal has been employed. The type of lubricant selected may vary depending on the specific spline construction and application. A lubricating grease having an NLGI grade number 1 is extensively used due to its ability to withstand relatively heavy loads. A typical lubricant of this type generally consists of a refined mineral oil and lithium 12 hydroxy stearate soap. Extreme pressure and other additives are incorporated, as required, to impart to or enhance specific properties or performance characteristics of the lubricant.

Typical basic performance requirements of a lubricating grease, which must be met and its various properties are described in the *SAE Handbook* under *Automotive Lubricating Greases*. In addition to providing for optimum spline wear life, the lubricant used must perform satisfactorily in all environmental conditions of the spline application.

The axial force characteristics of some typical rolling element splines are shown in Fig. 20. Their relatively low thrust forces point out the principal advantage of rolling element splines when compared with a conventional sliding spline. For a typical ball spline, the coefficient of friction is as low as 0.005 approxi-

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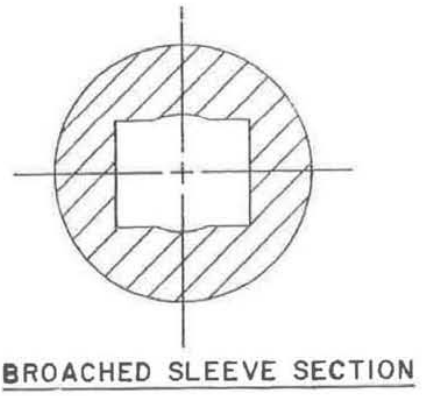


Fig. 21 — Typical rectangular section shaft telescoping components

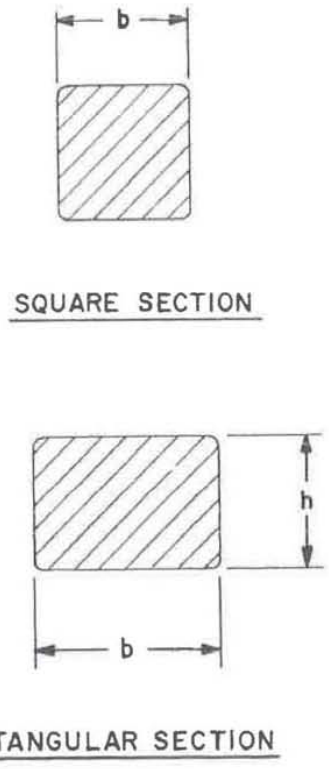


Fig. 22 — Square and rectangular bar shafts

mately, and for a typical roller or needle roller spline, it is approximately 0.017.

Agricultural Implement and Other Power Take-Off Axial Motion Devices

In agricultural implement and various other power take-off applications, numerous types of sliding and rolling element axial motion or telescoping devices have been employed in the driveline systems. Some of the simplified forms of sliding members which have been used are square, rectangular, or hexagonal bar shafts and grooved driveshaft tubing. Fig. 21 shows a typical rectangular section shaft mating with a driveshaft sleeve member having a broached rectangular hole. In addition to using broached or extruded sleeves as mating members for telescoping driveshafts, square and rectangular tubing are also commonly employed with square and rectangular bar shafting, respectively. When square or hexagonal bar shafts are used, phasing means, such as a weld tack or pin mating with a groove or grooves in the broached hole or on the bar shaft, must be provided to insure correct phasing of the Cardan universal joint yokes. The torsional strength of square and rectangular bar shafts depend on the specific type of material used and heat treatment method employed. Generally, these shafts are assembled with an interference fit in mating broached holes of yokes, and retained axially by welding. Fig. 22 shows the cross sections of typical square and rectangular bar shafts. The torsional shear stresses of shafts with these sections can be determined by the following conventional torsion equations:

$$S_s = \frac{T}{Z_p}$$

where:

T = Torque, lb-in

Z_p = Polar section modulus, in^3

For a square bar shaft:

$$Z_p = \frac{5b^3}{24}$$ (S_s is maximum at midpoint of each side)

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where:

b = Length of each side, in

then the torsional shear stress is:

$$S_s = \frac{24T}{5b^3}$$

For a rectangular bar shaft the equation for the polar section modulus in simplified form is:

$$Z_p = \frac{5b^2h^2}{15b+9h}$$ (S_s is maximum at midpoint of each longer side)

where:

b = Length of longer side, in

h = Length of shorter side, in

then the torsional shear stress is approximately:

$$S_s = \frac{(15b + 9h)\,T}{5b^2h^2}$$

The torsional shear stress of square and rectangular bar shafts can also be computed by using the exact Saint Venant equation which is:

$$S_s = \frac{T}{\alpha bh^2}$$

where:

α = Numerical factor depending upon the ratio b/h shown in Table 3.

TABLE 3 — NUMERICAL FACTORS FOR RECTANGULAR CROSS SECTION PROPORTIONS

$\frac{b}{h}$ =	1.00	1.50	1.75	2.00	2.50	3.00	4.00	6.00	8.00	10.00	∞
α =	0.208	0.231	0.239	0.246	0.258	0.267	0.282	0.299	0.307	0.313	0.333
β =	0.141	0.196	0.214	0.229	0.249	0.263	0.281	0.299	0.307	0.313	0.333

The maximum shear stress given by this equation occurs at the middle of each side of a square cross section and at the middle of each of the longer sides of a rectangular cross section. For a square cross section, the same value of shear stress is obtained by both the conventional torsion and Saint Venant equations. For a rectangular cross section, however, the conventional torsion equation results in an error of the shear stress not exceeding approximately 4% from the exact value obtained with the Saint Venant equation.

The angle of twist in radians per unit of length for square and rectangular bar shafts can be computed by using the Saint Venant equation which is:

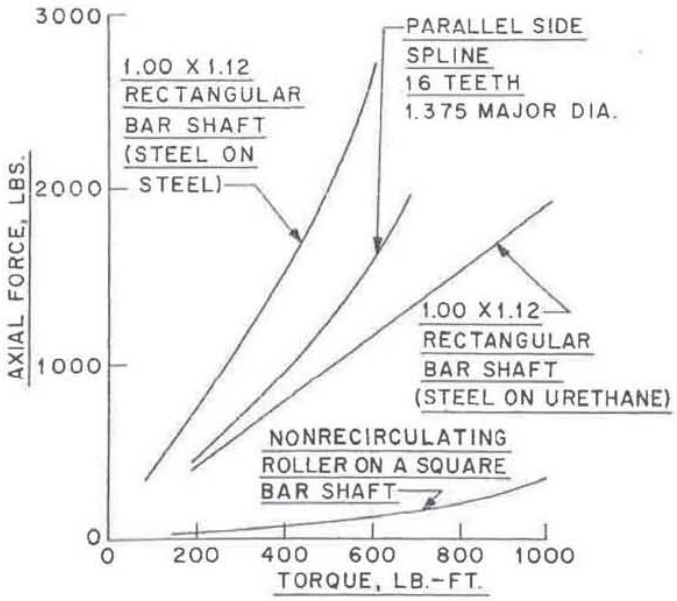


Fig. 23 — Comparison of axial force characteristics of typical agricultural implement telescoping devices with a conventional sliding spline

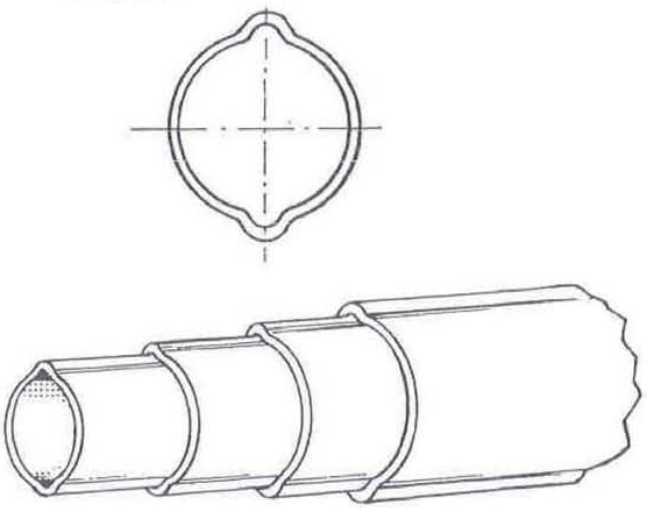


Fig. 24 — Grooved telescoping tubes

$$\phi = \frac{T}{\beta bh^3G}$$

where:

β = Numerical factor depending upon the ratio b/h shown in Table 3

G = Modulus of rigidity or modulus of elasticity in shear, psi

The torsional strength and stiffness properties of the square and rectangular bar shafts, as well as almost any other form of noncircular cross section shaft can also be determined by the membrane or soap film analogy, and provide results of reasonably close approximation. The equations for mating square and rectangular

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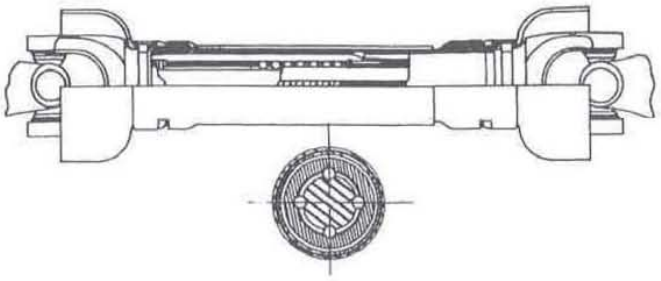


Fig. 25 — Nonrecirculating ball spline in a power take-off driveshaft

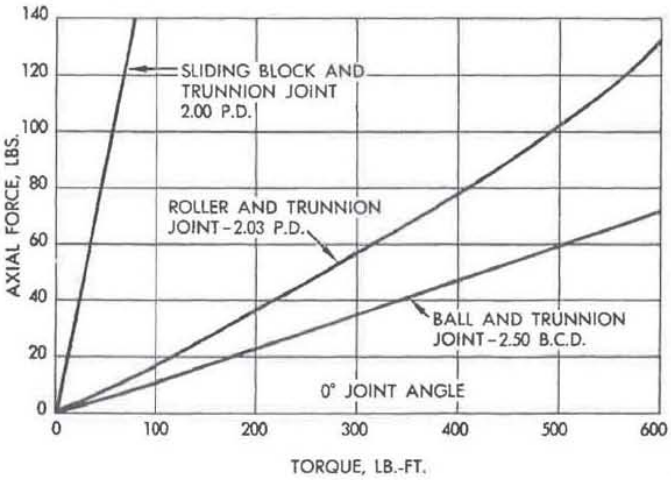


Fig. 26 — Axial force characteristics of end motion type non-constant velocity joints

tubing are shown in the chapter on *Tubing*. Fig. 23 shows the axial force characteristics of two typical rectangular bar shaft telescoping devices compared with a conventional sliding spline. It can be seen that a reduction of thrust force is obtained when a low friction material such as urethane is used on one of the mating sliding members. In this case, the internal surface of the steel sleeve is coated with a urethane lining. Fig. 24 shows a typical grooved tubular design with the transmission of torque accomplished through the integral driving grooves or ridges of progressively telescoping tubes. In order that the tubes have equal strength, their area polar moment of inertias must be equal. This is accomplished by increasing the wall thickness of the inner tubes. Usually no sealing means is provided with these axial motion devices, however, in some applications a lubrication fitting is employed for relubrication.

Rolling element splines which were previously described have also been widely used in agricultural implement and other power take-off telescoping mechanisms. Fig. 25 shows a typical non-recirculating ball spline used in a power take-off driveshaft application. A lubrication fitting for relubrication and sealing means are provided. Also shown is a typical telescoping safety shield used on power take-off driveshafts incorporating various types of axial motion devices. Nonrecirculating rollers have also been employed in rolling element axial motion designs. The axial force characteristics are shown in Fig. 23 for such a typical mechanism where the rollers operate on a square bar shaft. A significant

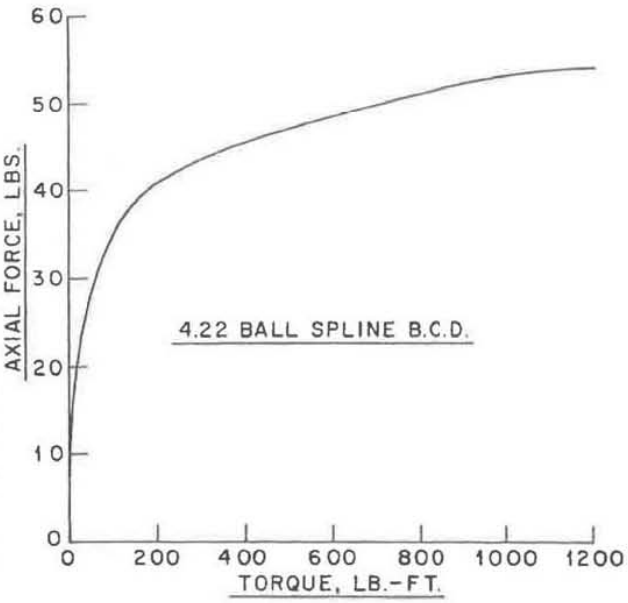


Fig. 27 — Ball spline Rzeppa joint axial force characteristics

reduction in thrust forces of this device when compared with the rectangular bar shafts and a conventional sliding spline, illustrates the principal advantage of rolling element splines.

End Motion-Type Universal Joints

In certain installations, it may be desirable or even necessary to combine axial movement with universal joint action. Non-constant as well as constant velocity universal joints are available that simultaneously permit axial movement and which replace the function of separate sliding or rolling element splines. These joints are discussed in varying detail in the chapter on *Universal Joints*.

The ball and trunnion, roller and trunnion, and sliding block and trunnion universal joints are nonconstant velocity types. The static (nonrotating) axial force characteristics of such typical joints are shown in Fig. 26.

The ball spline Rzeppa, tripot, Weiss, cross groove, and double offset universal joints are constant velocity types. The static (nonrotating) axial force characteristics of such typical joints are shown in Figs. 27-31, respectively. Depending on their specific design characteristics, end motion type universal joints under dynamic or rotating conditions, however, will exhibit widely varying and generally lower axial forces which are affected by rotational speed, torque, joint angle, and internal joint fit-up.

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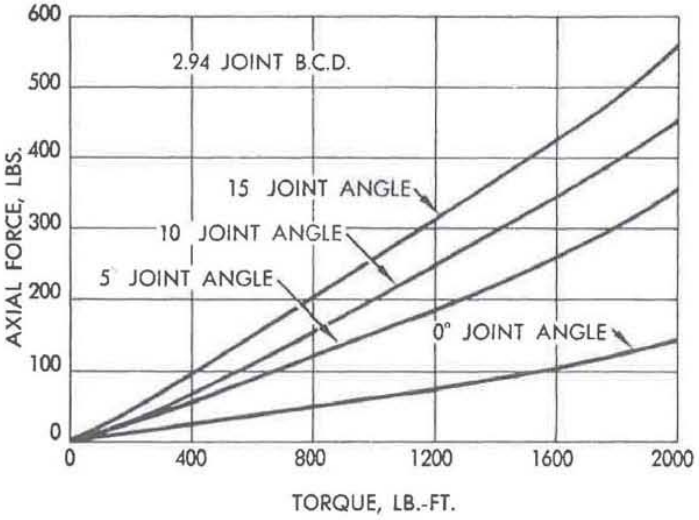


Fig. 28 — Tripot joint axial force characteristics

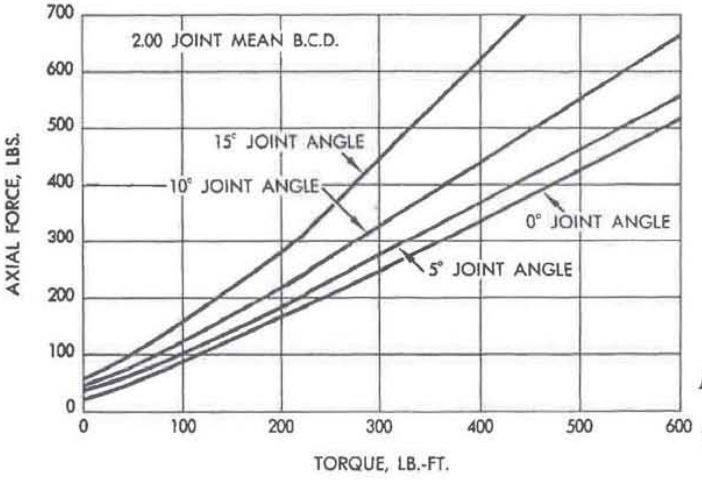


Fig. 29 — Weiss joint axial force characteristics

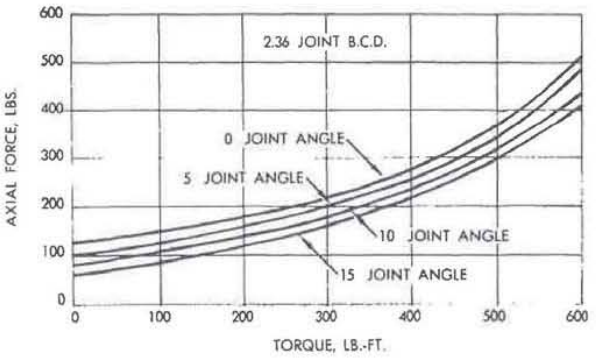


Fig. 30 — Cross groove joint axial force characteristics

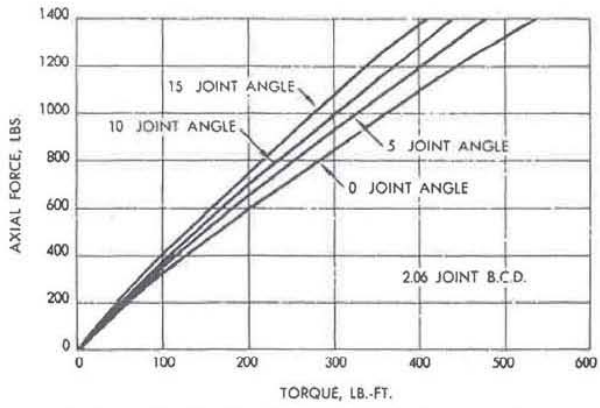


Fig. 31 — Double offset joint axial force characteristics

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Shaft Support Bearings

Chapter 6

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Chapter 6
Shaft Support Bearings

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

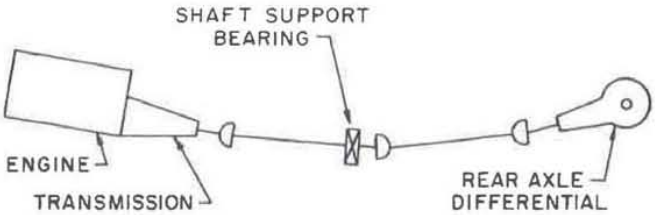


Fig. 1 — Schematic view of two piece driveline with shaft support bearing

Introduction

In driveline arrangements with three or more universal joints, the driveline must be supported at one or more intermediate points, in addition to the end supports, as illustrated schematically in Fig. 1. Such devices are called shaft support bearings.

In vehicle driveline systems, these support bearings must also be of special design in order to reduce to an acceptable level the transmission of driveline disturbances to the frame or body structure and passenger compartment. This important function is in addition to that of supporting the driveline weight and resisting the forces resulting from the maximum applied torque.

Design Considerations

In vehicle installations, the shaft support bearing usually consists of a rolling element bearing isolated in rubber and with a bracket configuration for attaching it to the vehicle structure. Generally, a sealed, radial type ball bearing is used and, depending on the application, certain functional specifications may be required to ensure that the bearing operates satisfactorily for any environmental requirements of the driveline.

Contingent on the overall design configuration of a particular support bearing, associated components such as grease retainers, bearing shields, or flingers are incorporated as required. Usually, a waterproof lubricant is packed around the sides of the bearing itself for protection from corrosion. To prevent the loss of this lubricant due to centrifugal force effects during rotation, grease retainers are used. In addition, a bearing shield or flinger in front of the support bearing and also at the rear in some designs, is used to protect not only the bearing, but in some cases, the rubber as well, from damage by foreign objects. In some designs, separate grease retainers are not used since their function is incorporated in the bearing shields. However, in other designs, a separate grease retention means may not be required because the shape of the support bearing configuration adequately retains the protective lubricant.

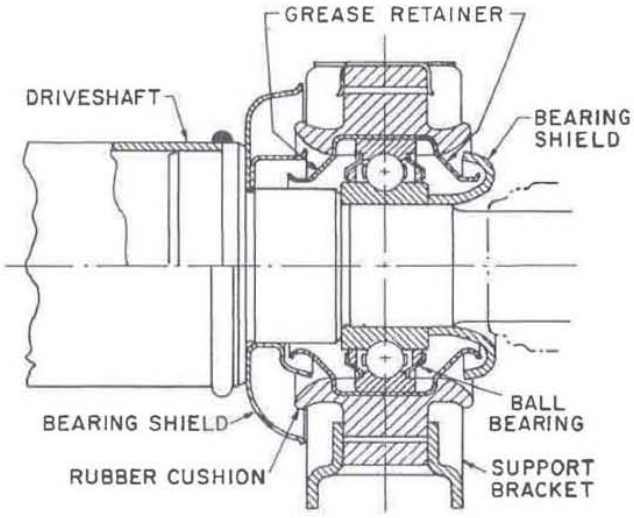


Fig. 2 — Typical truck shaft support bearing

Fig. 2 shows a typical shaft support bearing with associated components for a truck driveline application. In applications that are less critical with regard to the transmission of driveline disturbances, rolling element bearings in conventional support constructions such as typical pillow blocks or flanged housings are utilized. In some of these non-vehicle applications, the driveline may be operating either in a generally horizontal, inclined, or even in a vertical position. Thus, the support bearing must meet the requirements of the specific application. In a typical vertical cargo pump driveline installation, for example, the support bearing must bear the weight of both the suspended driveshaft and pump.

For vehicle applications, the support bearing is an important mounting for frequency response characteristics. The manner in which it is attached to the vehicle, whether directly to the underbody or to the frame, can have a considerable effect on the support bearing's capability of providing optimum isolation and reducing the transmission of disturbances to an acceptable level in a given application. In order to inhibit the transmitted vibrations produced by the driveline, support bearings with relatively low natural frequencies and spring rates are desirable. This requirement is illustrated in Fig. 3 where the relative forces transmitted, (transmissibility) by two support bearings with elastomers of different natural frequencies, are shown. It should be noted that the support bearing with a high natural frequency transmits high forces at high speeds and that a very significant reduction in transmissibility is obtained by using a low frequency type support bearing. A support bearing with low frequency and low damping characteristics will reduce transmissibility and provide good isolation at higher shaft speeds where driveline unbalance or first order excitation

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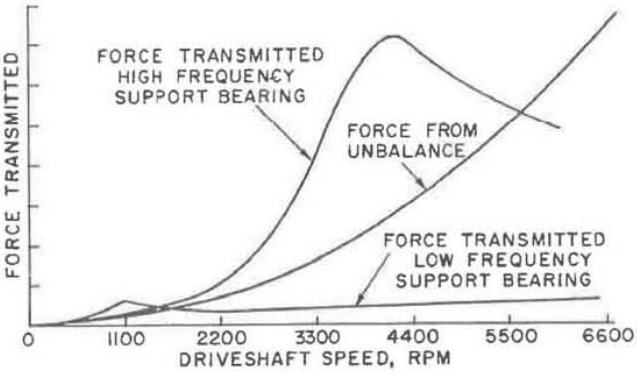


Fig. 3 — Typical shaft support bearing transmissibility characteristics

may be severe. These higher frequency vibrations can be both felt and heard. The unbalance disturbing forces on the support bearing are vehicle speed sensitive and are directly proportional to the square of the shaft rotational speed. Due to an excessively loose spline fit, unbalance excitation also may, in some instances, be torque sensitive because the spline centers itself differently under torque than under no torque. When this excitation occurs on a support bearing with a relatively low natural frequency, it is generally too small to induce vibration in the vehicle.

The support bearing, however, must also be designed to minimize the transmission of universal joint secondary couple force reactions, which result from second order excitations. This excitation level is related to the operating and phase angles of the universal joints and the transmitted torque, and occurs at lower rotational speeds and with higher driveline torques. When subjected to this vibration, the vehicle experiences a shudder type disturbance especially during low speed acceleration at approximately 20 Hz (cps). Shudder is thus a low frequency vibration generally occurring under 20 mph, and can be felt more than heard. This disturbance is not only vehicle speed sensitive, but also torque sensitive and rear suspension jounce (or load) sensitive.

The resultant force on the support bearing caused by secondary couple forces reacts with a combination of static and dynamic components where the dynamic component rotates at twice shaft speed. The magnitude of the couple forces at a given joint angle condition of the driveline are directly proportional to torque and; therefore, the disturbances increase directly with the transmitted torque. In the section on the *Cardan or Hooke Universal Joint*, the characteristics and effects of secondary couple forces and excitation are discussed in detail. Since the support bearing has a well defined resonant frequency, it is quite sensitive to secondary couple excitation when the driveline is rotating at a speed equivalent to one half of the support bearing's response frequency. A low frequency, low damping support bearing required to produce good isolation at high speeds will often respond to these couple forces at low speeds. Support bearings of this type usually have a critical or natural frequency of 1000-2000 cpm and; therefore, they will be excited at 500-1000 rpm driveline speed.

In this section on the *Cardan or Hooke Universal Joint*, it is shown that secondary couple forces cannot be eliminated entirely,

Table 1 — Maximum Dynamic Couple Forces

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

because of the necessity to compromise on joint angle selection between curb and design loading conditions for rear drive vehicles employing a typical live rigid axle. Some rear drive vehicles also use an independent suspension system at the rear in addition to the commonly used front independent suspension. In this vehicle configuration, the driveline geometry remains relatively constant and is not affected by suspension movement and changes in load because the final drive is frame mounted. Thus, when a two piece driveshaft is required in such a vehicle, the support bearing design relative to secondary couple excitation is simplified since the necessity to compromise on joint angle selection between curb and design loading does not exist.

Experience on vehicle installations with shaft support bearings has shown that the resultant dynamic couple force on the support bearing should be maintained under a certain maximum value in order to preclude objectionable shudder. In Table 1 are shown the limits on the oscillating force which have been found to be generally acceptable at maximum transmitted torque. These rotating couple forces act twice per driveshaft revolution on the support bearing. Since these forces are torque sensitive, the vehicle wide open throttle start or acceleration modes of the driveline are usually most important.

Besides responding to driveline produced excitations, the support bearing can also be sensitive to other vehicle vibrations. Throughout the low, medium, and high speed ranges of the vehicle, the driveshaft may act as a transmitter or response mechanism for disturbances emanating from other sources in the vehicle. However, when a support bearing is inserted in the driveline, such disturbances may be especially objectionable at certain response frequencies of the sensitive support bearing. These vibrations may originate from the engine and transmission, engine belt driven accessories, exhaust system, rear axle, frame and body structure, or suspension system components such as brakes, wheels, and tires. Furthermore, these excitations may occur during various driving modes such as slow accelerations constant speed cruising, float, and slow deceleration or coast. Depending on the specific nature of these various vibrations, they may be classified into sensitivity categories such as vehicle speed sensitive, engine speed sensitive, torque sensitive, rear suspension jounce (or load) sensitive, or into various possible combinations of these categories. Also, they may be only felt, or only heard, or both. Therefore, various corrective measures or design procedures have to be used to eliminate or minimize such objectionable disturbances at their sources, if possible. Otherwise, appropriate vibration control means or devices have to be incorporated at optimum locations either in the drivetrain or elsewhere in the vehicle.

Design features of the support bearing have a considerable

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effect on its acceptability in a given installation. Due to second order driveline excitation, the driveshaft will be displaced depending on the magnitude of the static and dynamic components of the resultant secondary couple force reactions on the support bearing, as well as the spring rate and geometry of the support bearing rubber member.

In order to meet the functional requirements, the support bearing must provide for low radial stiffness, some axial compliance, and a means for controlling the extreme radial movement by means of integrally molded rubber ribs or projections. These molded ribs or other design features, which control extreme radial deflection, thereby tend to increase the radial stiffness of the support bearing in the fully deflected position.

Generally, the design configuration of the rubber member is such that the elastomer is loaded principally in shear, especially during the initial or most of the deflected range. When compared to compression loading, a lower spring rate results for a given deflection range with the rubber loaded in shear. Furthermore, the most desirable damping characteristics are achieved in shear rather than in compression loading. By increasing the durometer hardness of the rubber, the increase of dynamic over static moduli properties are not as great with shear loading as they are with compression loading. Therefore, the dynamic characteristics of the support bearing are not as adversely affected by a shear loaded design as they are by a compression loaded design. The damping or internal friction of rubber approximates very closely pure viscous damping or fluid friction and it is customary to perform computations on that approximation.

Usually, molded natural rubber is used as the elastomeric material in the support bearing because of its desirable inherent properties. Sometimes it is compounded to provide maximum resilience (efficiency) and therefore, result in low hysteresis (damping). Depending on the design configuration of the rubber member, typical hardnesses in the range of 45-70 Shore A durometer have been found satisfactory. A driveline with a particular support bearing design should be evaluated in an actual vehicle in order to select the minimum rubber hardness which will provide for optimum attenuation of disturbances, as well as a satisfactory maximum operating speed. In order that the support bearing performs adequately in a particular application as intended, and to provide a means for maintaining a quality control audit of manufactured components, it is desirable that certain properties such as dynamic spring rate and either resilience or dynamic damping coefficient be specified at a given ambient temperature. The radial dynamic spring rates of some typical support bearings have been in the relatively wide range of 300-1500 pounds per inch, while efficiencies (resilience) have ranged from 75% for truck support bearings, to nearly 90% for passenger car support bearings. Also, the rubber used must meet certain environmental requirements, of which ozone resistance is a typical example.

In addition to fulfilling the functions of supporting the driveline weight and reducing the transmitted vibrations previously described, the support bearing and its attaching means to the vehicle must also have the strength to withstand the severe secondary couple forces which can result from relatively high momentary shock or impact torque loads impressed on the driveline.

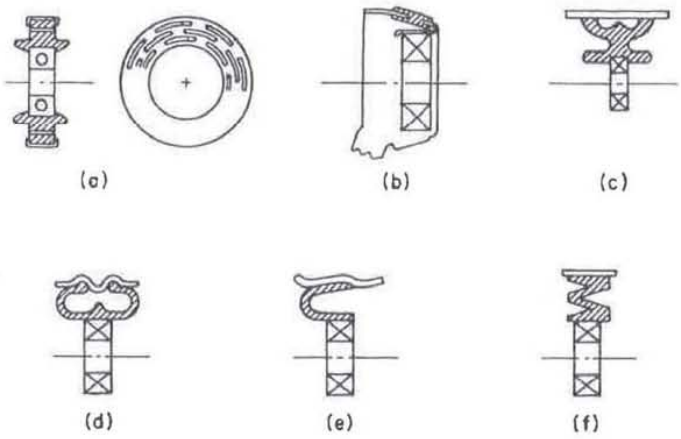


Fig. 4 — Typical shaft support bearing configurations

These forces on the support bearing may be considerably higher than those applied during wide open throttle start or acceleration. In addition, during these relatively high shock or impact torque loading conditions, the driveline sliding spline generally may not slip as required during rear axle pinion torque reaction (windup or winddown) and suspension movement. Therefore, these relatively high instantaneous forces on the support bearing tend to cause the driveline to buckle or "jackknife" were it not for the compliance permitted at the front by the engine and transmission rubber mounts, and at the rear by the suspension rubber bushings of a typical vehicle. Fig. 4 shows typical shaft support bearings and illustrates the large variation of design configurations possible in driveline applications and thus explains the relatively wide range of rubber hardnesses and dynamic spring rates which have been used.

Computations can be performed which determine analytically the first and second order disturbing or exciting vehicle speeds attendant with the use of shaft support bearings. These appropriate computation procedures are illustrated in two typical driveline problems. The first covers the design of a support bearing for a typical large passenger car application, while the second covers the design of a support bearing for a typical light duty-truck installation. Each support bearing configuration is designed to provide optimum functional characteristics for its particular application.

Typical Large Passenger Car Shaft Support Bearing

In the two piece driveline installation for a typical large passenger car, a Cardan joint, double Cardan joint and Cardan joint are used in series, and the shaft weight at the shaft support bearing is 23 lb. The hardness of the natural rubber used in the support bearing is 50 ± 5 durometer. The support bearing's radial dynamic spring rate is 700-1180 lb/in, while its efficiency is 80-88% at 73°F.

The mean dynamic spring rate is:

$$k = \frac{700 + 1180}{2} = \frac{1880}{2}$$

$$k = 940 \text{ lb/in}$$

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The mean efficiency is:

$$\eta = \frac{80 + 88}{2} = \frac{168}{2}$$

$$\eta = 84\%$$

and thus the damping factor is:

$$\zeta = \frac{100 - 84}{100} = 0.16$$

The undamped natural frequency is:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{kg}{W}}$$

where:

k = Spring rate, lb/in

W = Weight, lb

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

Hence:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{940 \times 386.4}{23}}$$

$$f_n = 20.00 \text{ Hz (cps)}$$

The damped natural frequency is:

$$f_{n(D)} = \sqrt{1 - \zeta^2}\,(f_n)$$
$$= \sqrt{1 - (0.16)^2} \times 20.00$$

$$f_{n(D)} = 19.49 \text{ Hz (cps)}$$

$$F_{n(D)} = 19.46 \times 60 = 1169.40 \text{ cpm}$$

This critical or natural frequency is within the 1000-2000 cpm range for typical support bearings. Thus, the support bearing will respond to driveline unbalance or first order excitation at 1169 rpm shaft speed. The secondary couples excite the support bearing when the driveshaft is rotating at one half of the support bearing's basic response frequency. Therefore, the second order driveline excitation will occur at 585 rpm shaft speed.

For this particular vehicle, N/V the ratio of driveshaft speed in rpm to car speed in mph is 37. Thus, the first and second order vehicle disturbing or exciting speeds are:

First Order

$$V = \frac{1169}{37} = 32 \text{ mph}$$

Second Order

$$V = \frac{585}{37} = 16 \text{ mph}$$

Typical Light-Duty Truck Shaft Support Bearing

In the two piece driveline installation for a typical light-duty truck, three Cardan joints are used in series and the shaft weight at the shaft support bearing is 22 lb. The hardness of the natural rubber used in the support bearing is 60 ± 5 durometer. The support bearing's radial dynamic spring rate is 1300 ± 200 lb/in, while the dynamic damping coefficient is 4 lb-s/in at 73°F.

The critical damping coefficient is:

$$c_c = 2\sqrt{km} = 2\sqrt{\frac{kW}{g}}$$

$$= 2\sqrt{\frac{1300 \times 22}{386.4}}$$

$$c_c = 17.21 \text{ lb-s/in}$$

The damping factor is:

$$\zeta = \frac{c}{c_c}$$

where:

c = damping coefficient, lb-s/in

$$\zeta = \frac{4}{17.21} = 0.23$$

The percent damping is:

Percent damping = $100\,\zeta = 100 \times 0.23 = 23.00\%$

The undamped natural frequency is:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{kg}{W}}$$

$$= \frac{1}{2\pi}\sqrt{\frac{1300 \times 386.4}{22}}$$

$$f_n = 24.05 \text{ Hz (cps)}$$

The damped natural frequency is:

$$f_{n(D)} = \sqrt{1 - \zeta^2}\,(f_n)$$
$$= \sqrt{1 - (0.23)^2} \times 24.05$$

$$f_{n(D)} = 23.39 \text{ Hz (cps)}$$

$$f_{n(D)} = 23.39 \times 60 = 1403.40 \text{ cpm}$$

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This critical or natural frequency is within the 1000-2000 cpm range for typical support bearings. Thus, the support bearing will respond to driveline unbalance or first order excitation at 1403 rpm shaft speed. The secondary couples excite the support bearing when the driveshaft is rotating at one half of the support bearing's basic response frequency. Therefore, the second order driveline excitation will occur at 702 rpm shaft speed.

For this vehicle, N/V the ratio of driveshaft speed in rpm to vehicle speed in mph can vary from 37-46 depending on the rear axle ratio and tire size selected for a particular vehicle. Thus, the first and second order vehicle disturbing or exciting speeds are:

First Order

$$\text{For } \frac{N}{V} = 37$$

$$V = \frac{1403}{37} = 38 \text{ mph}$$

$$\text{For } \frac{N}{V} = 46$$

$$V = \frac{1403}{46} = 30 \text{ mph}$$

Second Order

$$\text{For } \frac{N}{V} = 37$$

$$V = \frac{702}{37} = 19 \text{ mph}$$

$$\text{For } \frac{N}{V} = 46$$

$$V = \frac{702}{46} = 15 \text{ mph}$$

Hence, first order disturbances will occur in the 30-38 mph vehicle speed range and second order disturbances will occur in the 15-19 mph speed range.

In the preceding computations for the shaft support bearing designs of the two vehicle applications considered, it is noted that damping does not appreciably affect the natural frequency of the support bearing. Therefore, since low damping is generally desired, the effect of damping may be neglected in approximate computations.

Thus far the discussion on shaft support bearings considered only the application of typical multipiece driveshaft arrangements where, in addition to providing for damping of first and second order driveline produced excitations, the support bearing also is required to support a portion of the shaft's weight at an intermediate location. In some passenger cars with independent rear suspension, one piece driveline arrangements have also been used which required the need of a shaft support bearing for whip control. In these vehicle drivetrain arrangements, the driveline geometry remains constant or relatively constant and is not affected by suspension movement or change in load because the final drive is frame mounted.

These arrangements have incorporated either a typical open driveshaft design of a relatively small shaft diameter in proportion to its length with a universal joint at each end, or a driveshaft of similar geometric proportions operating inside a torque tube. In the open driveshaft construction which is of rather limited usage, the maximum shaft diameter may be limited by the necessary clearance required between the shaft and vehicle underbody, while in the torque tube installation, the maximum shaft size is dictated by the required annular clearance space inside the torque tube. The torque tube construction may be of the flanged end type which serves as a structural member and rigidly connects the engine and transmission with the rear axle housing, and thus no universal joints are used with the driveshaft. Generally, this driveshaft has a splined means of attachment at each end.

The full length torque tube was also once commonly used in passenger car drivetrain arrangements with rear unsprung live axles. In addition to the driveshaft transmitting torque from the engine and transmission to the live axle final drive in such a system, the torque tube must resist driving and braking thrust forces from the rear wheels to the vehicle through a commonly called torque ball design at its front end. This torque ball also provides for angular movement during suspension motion since the rear end of the torque tube is rigidly flange connected to the final drive housing. Thus a universal joint is required in the driveshaft at the torque ball, while the rear end of the shaft is attached to the final drive pinion shaft with a splined connection. When clearance space beneath the underbody and vehicle unsprung weight are not of importance, the torque tube may be made large enough to allow a relatively normal size tubular driveshaft to operate inside it which was once common practice. However, when reduced clearance space and weight are necessitated by vehicle design, the driveshaft diameter may become relatively small in proportion to its length.

Depending on the vehicle application parameters, the driveshaft may be either of solid bar or tubular shaft construction when the open driveline or torque tube configurations are employed. Although the tubular shaft is larger in diameter than a solid bar shaft for a given application, it can provide for a reduction in weight as well as an increase in torsional and transverse stiffness.

When these relatively small diameter drivelines are employed, one or more rubber isolated support bearings are required to permit the shaft to operate safely above its critical speed and to reduce the transmissibility of unbalanced forces. Both critical speed and unbalance are first order excitations. In the chapter on *Critical Speed*, it is shown that the critical speed of a shaft is affected by its area moment of inertia, weight, length, and material modulus of elasticity. Hence, depending on the particular design, these drivelines of relatively small diameter must operate not only above their lowest critical speed but may also have to operate above higher orders or harmonics of the fundamental or

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lowest critical speed. Also, when such a shaft incorporates typical Cardan universal joints at each end, the effect of the twice per revolution secondary couple excitation on critical speed must be considered. This excitation can produce an effect like a critical speed at one half of the lowest or fundamental critical speed. Therefore, such driveline designs must be analyzed and evaluated with regard to the mode shapes of the lateral and vertical bending modes for the various possible orders of excitation to determine the optimum location of the support bearing along the shaft length. Positioning a support bearing at a nodal point would obviously be least desirable. This study may also indicate the necessity to use more than one support bearing. In an open driveline arrangement, the support bearing is generally attached to the frame, while in a torque tube design the support bearing is positioned inside the torque tube at its optimum longitudinal location.

In addition to full length torque tube designs, modified or short torque tube constructions have also been used in some passenger car drivetrain systems. Generally, in these vehicle drive system layouts, the front portion of the driveline is a typical open tubular shaft with a universal joint at each end. The rear end of this open shaft is attached to a shaft which is usually of solid bar construction and operates inside a short torque tube. The torque tube is rigidly flange connected to the final drive housing of the rear unsprung live axle, while the rear end of the bar shaft is attached to the final drive pinion shaft with a splined connection. The front end of the bar shaft protruding from the torque tube must be supported with a support bearing which is attached to the frame. Since no torque ball is incorporated in this modified torque tube construction, suspension control members must be included in the vehicle chassis layout to transfer the driving and braking thrust forces from the rear wheels to the frame.

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Driveshaft Attachment Methods

Chapter 7

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Chapter 7
Driveshaft Attachment Methods

E. R. Wagner, P.E. — Saginaw Steering Gear Div., GMC
D. W. Holzinger — Spicer Universal Joint Div., Dana Corp.

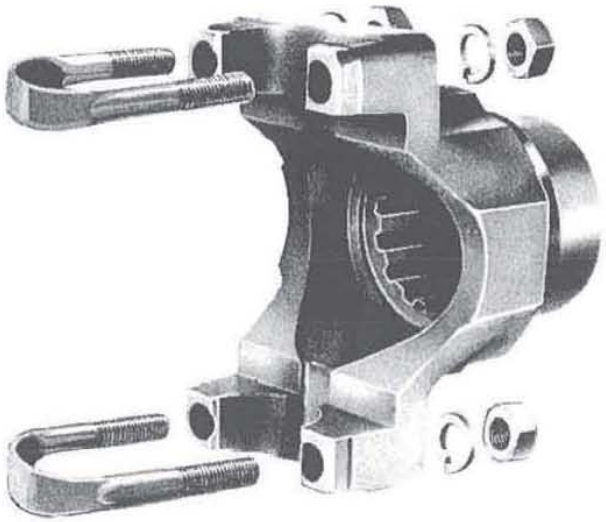


Fig. 1 — End yoke (U-bolt type)

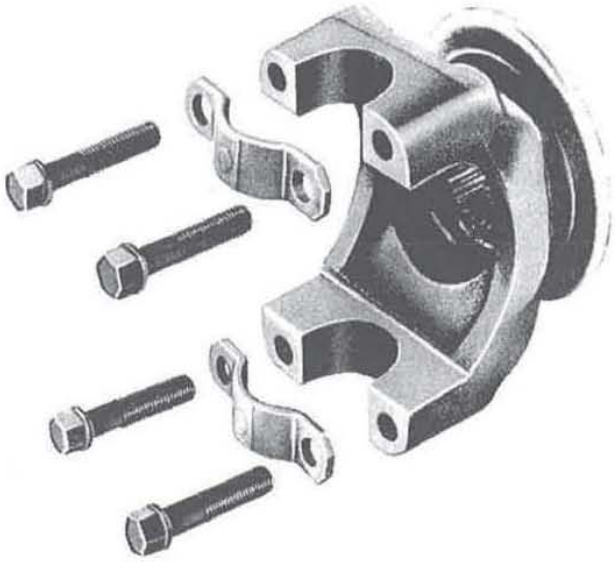


Fig. 2 — End yoke (strap type)

Introduction

The attachment of a driveshaft to other drivetrain members in a vehicle or other installation can be accomplished in a number of ways. The methods of attachment chosen or required should be designed with sufficient strength to withstand the forces which may result from the functional and durability requirements of a given application. Also, the ease of assembly and disassembly, for servicing when required, should be considered when selecting a particular design for an application. The most common methods of attachment of a driveshaft with Cardan or double Cardan joints have been with end yokes, flange yokes, splines of various designs, and shaft support bearings. Driveshafts using other types of universal joints are generally attached with splined, flanged, or disc type connections.

End Yoke

An end yoke is a yoke which attaches a driveshaft to another drivetrain component. The end yoke assumes numerous constructions with the specific design being dependent to a large degree on the application requirement. The U-bolt and strap type constructions with a splined hole in the hub and using round bearings, shown in Figs. 1 and 2, respectively, are commonly used in passenger car and light-duty truck applications. The wing bearing type construction with a splined hole, shown in Fig. 3, is commonly used in heavy-duty trucks and off-highway vehicles.

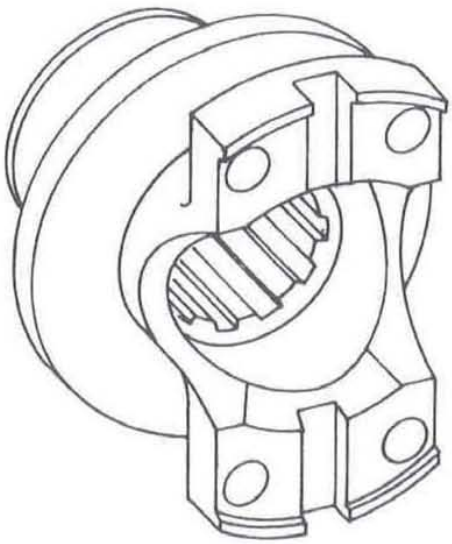


Fig. 3 — End yoke (wing bearing type)

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Fig. 4 — End yoke (quick-disconnect type)

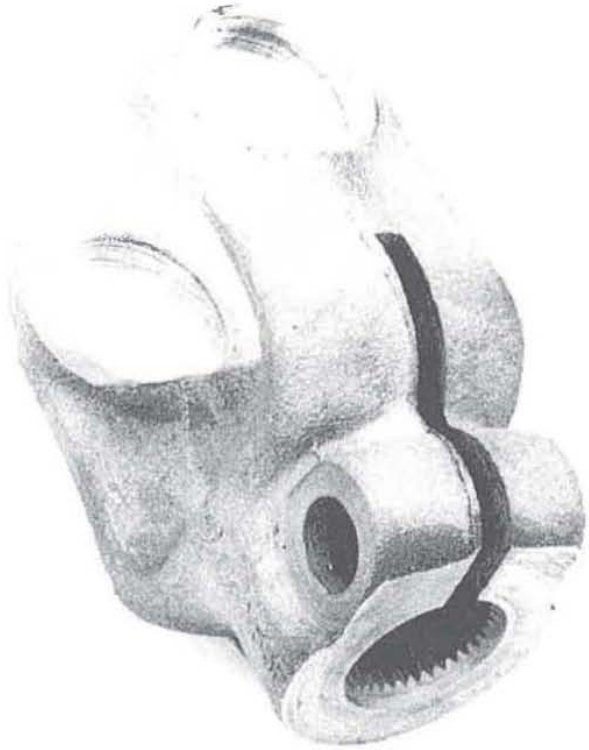


Fig. 5 — End yoke (clamp bolt type)

In all of these applications, the yoke is usually retained with a nut or bolt and washer at the splined connection when attached to another driveline or drivetrain component. The thread size of these fasteners as well as for the U-bolt, bolt, or cap screw fasteners, retaining the round or wing bearings to the yokes must be of sufficient size and strength to withstand the forces which may result from the functional and durability requirements of a given application. The quick-disconnect type construction with a splined hole, as shown in Fig. 4, is generally used in power take-off drive systems. The clamp bolt type construction with a partially serrated hole, shown in Fig. 5, is most commonly used in very slow speed applications such as steering shaft systems.

Other end yoke constructions with a splined hole or with a straight round hole and keyway are retained with a set screw or a nut. An end yoke can also have a tapered round hole and keyway and be retained with a nut. Still other end yoke constructions have square, rectangular, or hexagonal holes which are usually set screw retained. The end yoke is also described in the section on the *Cardan or Hooke Universal Joint*.

Flange Yoke

A flange yoke is a yoke which attaches a driveshaft to a companion flange. The flange yoke is shown in Fig. 6 in two alternate constructions. These two types are the commonly used circular and rectangular configurations. For clarification purposes, the mating companion flange for each type is also shown. The yoke is held to the flange with bolts, and a pilot diameter is provided to maintain concentricity. The bolt size required is dictated by the clamping force necessary to withstand the forces

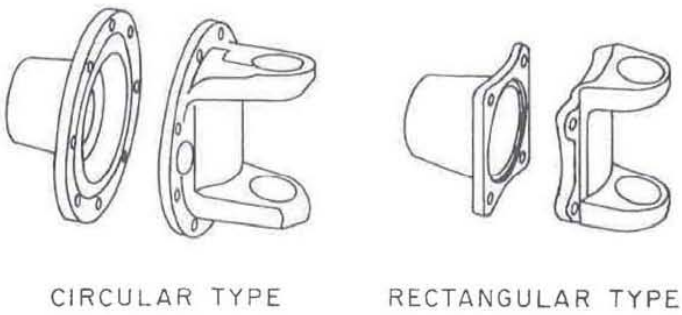


Fig. 6 — Flange yoke

which may result from the functional and durability requirements of a given application. The flange yoke is used in a wide variety of driveshaft applications, and is also described in the section on the *Cardan or Hooke Universal Joint*.

Splines

Splines are commonly used to attach or connect a driveshaft to another drivetrain component. In passenger car applications, for example, the two-joint outboard slip driveshaft configuration is commonly used in a typical sliding or slip connection with the transmission output shaft. When multipiece driveline configurations are employed, the individual shaft sections are generally fastened together with either slip spline or interference, or permanent fit spline connections. In vehicle axle driveshaft applications, both slip and interference, or permanent fit splines are also employed as dictated by the application requirements. The

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