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# **Automotive Bearing Technology**

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**Introduction**

Automotive ball and roller bearings are mechanical components that support rotating shafts with such low mechanical losses that they are referred to as being "anti-friction" devices. Without these components, automotive vehicles would not perform as well as they do today where they allow engine power to be delivered to the wheels in a very smooth and efficient manner.

Anti-friction bearings include ball, cylindrical roller, and tapered roller bearings. All bearings have an inner ring, an outer ring, a complement of rolling elements, and a separator. The separator spaces the rolling elements evenly around the annulus of the bearing. The rolling elements can be balls, cylindrical rollers, or tapered rollers. The three basic dimensions of anti-friction bearings are the inside diameter (ID) or bore (B), the outside diameter (OD), and the width (W). See Figure 1.

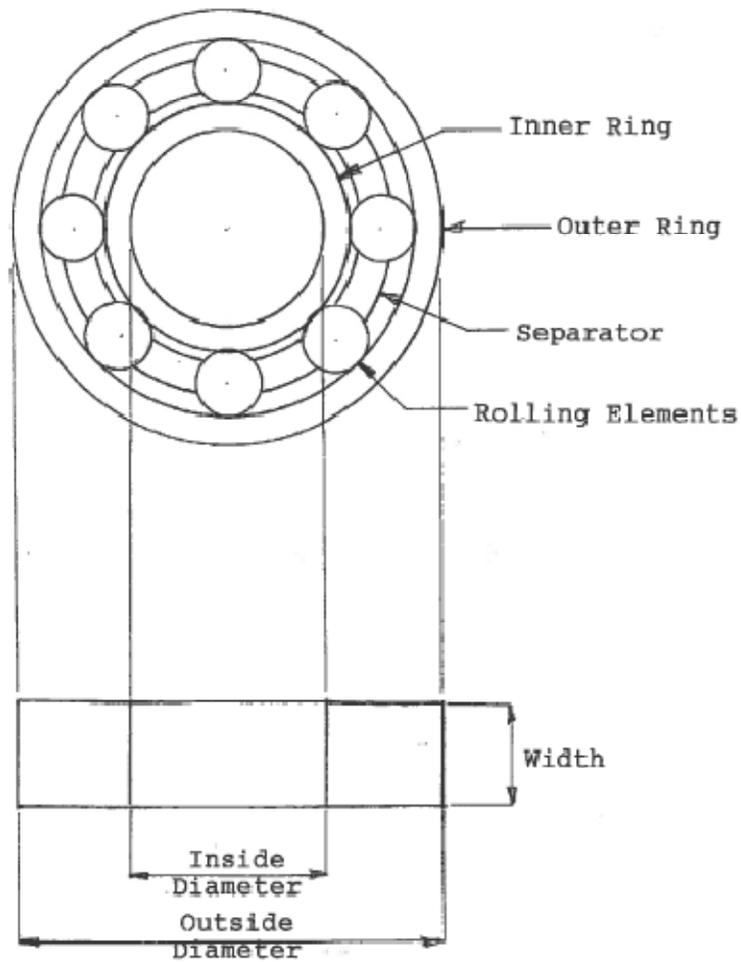
Bearings support two kinds of loading, radial loads and thrust loads. Radial loads act perpendicular to the bearing axis of rotation while thrust loads act parallel to the bearing axis of rotation. Sometimes there are two radial loads acting  $90^\circ$  apart; the Pythagorean Theorem is then used to calculate the resultant load acting on the bearing. See Figure 2.

At the bottom of Figure 2 are two diagrams that show the effect of location on the amount of load that each of two bearings mounted on a shaft supports. The first



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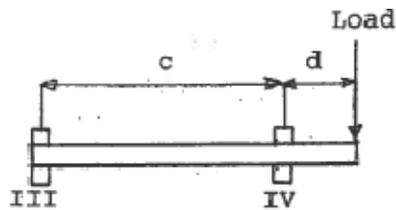
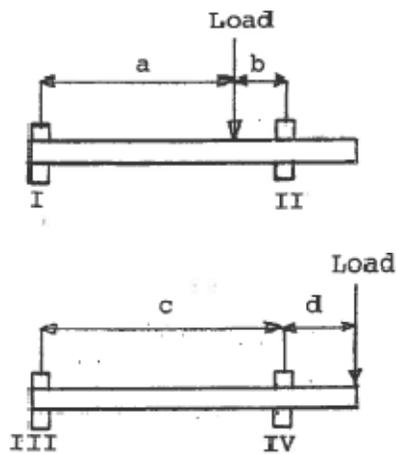
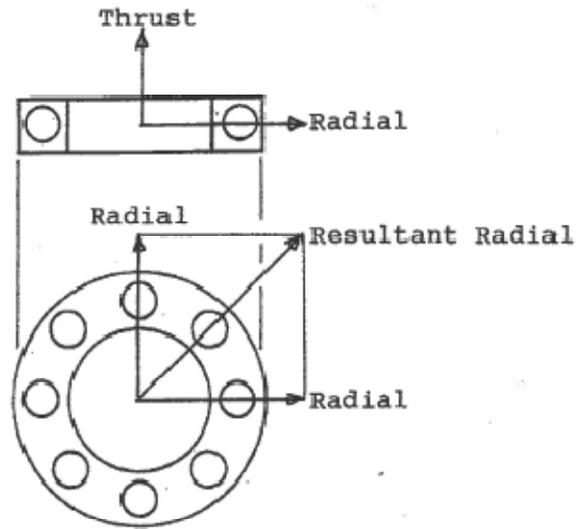
**Figure 1**  
**Anti-Friction Bearing Terminology**





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**Figure 2**  
**Bearing Loads**





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diagram has the load "straddle mounted" between two bearings. Simple beam calculations will show that Bearing II which is closer to the load supports a higher portion of the load than bearing I. In the second diagram, the load is "overhung" from two bearings. Calculations will show that bearing IV, which is closer to the load, supports a quantity that is actually greater than the load itself.

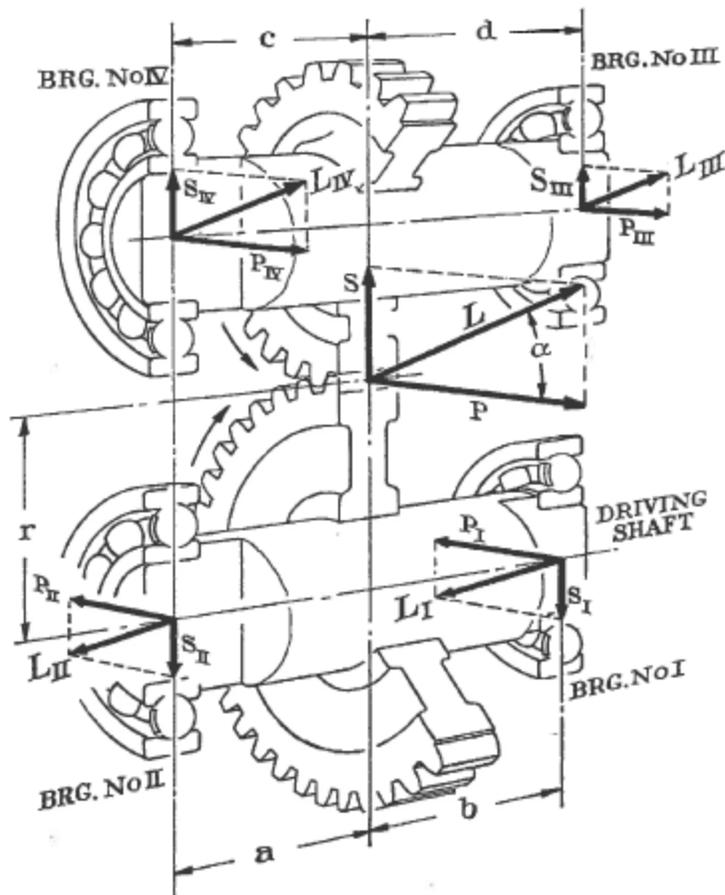
### **Spur Gear Bearings**

One of the most fundamental applications of automotive anti-friction bearings involves supporting spur gears. Spur gear shafts supported by anti-friction bearings are commonly used in vehicle manual transmissions. Figure 3 has two straddle mounted spur gears in mesh that are supported by ball bearings. Spur gears when delivering power produce radial loads only. The radial loads are composed of a tangential force  $P$  and a separating force  $S$  which when combined equal the resultant force  $L$  as shown on Figure 3. This force is reacted by  $P$ ,  $S$ , and the resultant force  $L$  at each of the four bearing locations shown at distances  $a$ ,  $b$ ,  $c$ , and  $d$ . The force  $L$  on each of the four bearings is used to calculate bearing operating life. The steps taken that lead to the calculation of bearing life follow: (The equations are listed in Appendix I.)

- The torque on the driving gear is calculated from the input horsepower and rpm.
- Tangential force  $P$  is calculated from the torque and the driving gear pitch radius.

**Figure 3**

**Spur Gear Bearings**





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- Separating force  $S$  is calculated from the tangential force  $P$  and the gear pressure angle.
- $P$  and  $S$  at each of the four bearing locations are calculated using simple beam equations.
- Resultant bearing load  $L$  is calculated for each location using the Pythagorean Theorem.
- Bearing life is calculated for each location from the load and operating speed. (See Appendix I.)

### **Transmission Bearings**

The upper sketch of Figure 4 has two sets of gears called the "constant mesh" and "second speed" gearsets which are the same as those shown on the lower sketch of Figure 4 in the automotive manual transmission. The first set of gears is the constant mesh gearset and is used to deliver input power to the lower countershaft which drives the second speed set of gears that are used to power the right hand output shaft. The bearing loads of an automotive manual transmission are at their highest in the intermediate gears such as second gear; therefore, this exercise will demonstrate how to calculate the loads on the bearings in Figure 4 with the transmission in second gear. Bearings A and B support the input shaft; bearings C and D support the countershaft; and bearings E and F support the output shaft. Bearing F, shown as a small diameter roller bearing, is located radially between the input and output shafts on the same centerline as the input pinion gear which overhangs its two bearings A and B. The gears



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**Figure 4**

**Transmission Bearings**

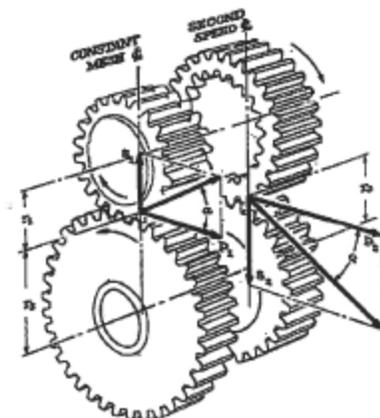


Diagram I.

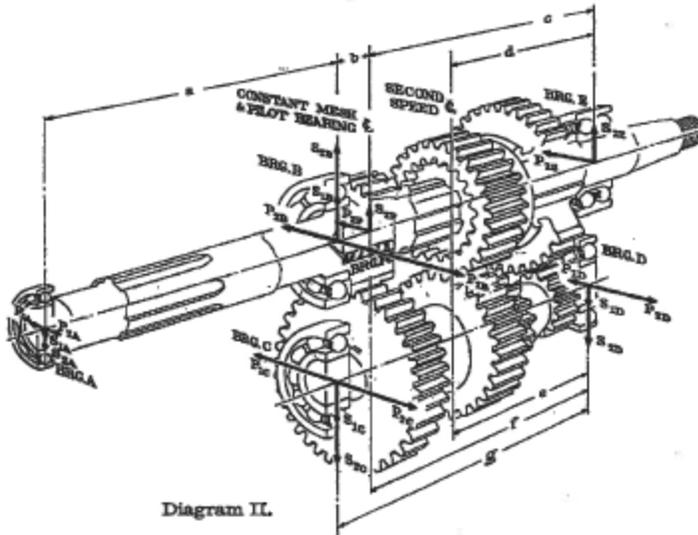


Diagram II.



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on the other two shafts straddle mount their bearings. A listing of the steps leading to the calculation of the bearing loads and life follows: (The equations are shown in Appendix II.)

- $P_1$  and  $S_1$  tangential and separating forces for the first set constant mesh gears are calculated as before.  $P_2$  and  $S_2$  for the second speed gearset are calculated from  $P_1$  and the ratio of  $r_2$  to  $r_3$ .
- $P_{1A}$ ,  $S_{1A}$ ,  $P_{2A}$ , and  $S_{2A}$  loads on bearing A shown on Figure 4 are derived from the overhung load imposed by the constant speed mesh and, also, from the second speed gearset straddle mounted load on the constant speed pinion.
- $P_{1B}$ ,  $S_{1B}$ ,  $P_{2B}$ , and  $S_{2B}$  loads on bearing B shown on Figure 4 are calculated in a similar manner to bearing A above.
- The loads on the remaining four bearings are calculated in a similar manner to that of bearings A and B.
- The resultant load and speed is calculated at each location from which bearing life is determined. (See Appendix II.)

### **Planetary Gear Bearings**

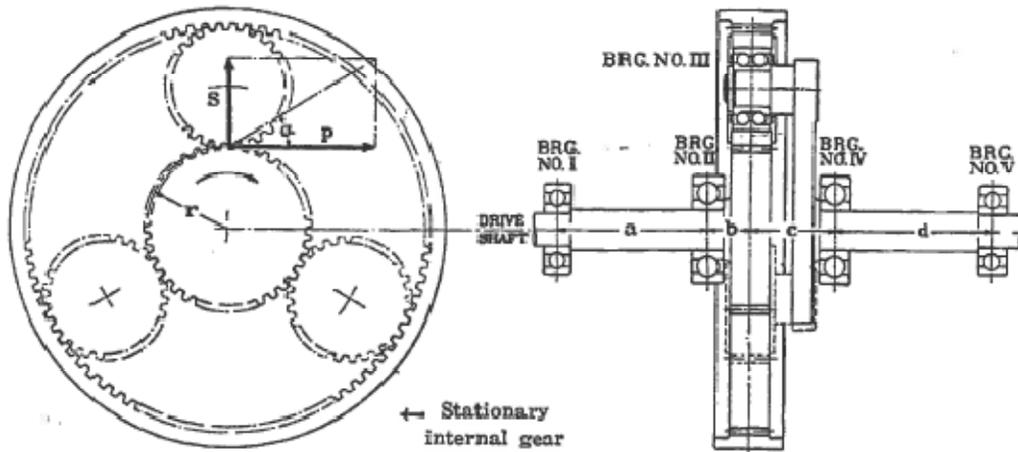
Planetary gears are very versatile design tools that are used in vehicle automatic transmissions. Figure 5 has a sketch of a planetary gearset. The center sun gear drives three planet gears inside a stationary outer ring gear. The planets are held together by a carrier which drives the output shaft. In this mode, the output is a reduction and is used in automatic transmissions as one of the intermediate gears. When the planet carrier is



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**Figure 5**

**Planetary Gear Bearings**





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used to drive the ring gear with the sun gear stationary, the output is an increase in speed and is used as an overdrive gear. When the sun gear is used to drive the ring gear with the carrier stationary, the output is in the opposite direction as the input and is used in vehicles as reverse gear. Compound planetary gearsets with two sun gears, two sets of planets, and one ring gear are used to provide the complete number of gears needed to power automotive vehicles.

The planetary gearset on Figure 5 has two bearings supporting the overhung sun gear shaft, a bearing located under each planet gear, and two bearings supporting the overhung loaded output carrier shaft. When a planetary gearset employs three planets such as shown in Figure 5, the tangential and separating forces cancel leaving bearings I, II, IV, and V unloaded. This can be seen if the P and S vectors shown on the upper planet on Figure 5 were also placed on the other two planets. However, the planet bearings are loaded because of the torque transmitted. With equally divided torque, the load on each planet bearing equals  $2P/3$ . The load is doubled because of the diametrically opposed meshes on each center supported planet gear and divided by three because there are three planets. The separating forces S cancel out.

The speed change of the planetary gear arrangement in Figure 5 with the sun gear driving follows: (N is the rpm of the sun gear.)

$$\text{Output rpm} = N/2[(1+(\text{number of planet teeth}/\text{number sun teeth})$$

### **Wheel Bearings**

Wheel bearings and associated equipment are classified as safety items in the automotive industry because of the consequences involved if a failure occurs and a wheel leaves the vehicle. Because of the seriousness of the situation, extensive engineering development and testing is a never ending task in order to ensure that these products perform without failure for their entire design life or, if a failure should occur, passenger safety is not endangered.

Front wheel bearings on traditional rear drive vehicles are used in closely mounted pairs with one bearing being referred to as the inner (inboard) bearing which is larger and the other being referred to as the outer (outboard) bearing. See Figure 6. Each bearing pair is mounted on a spindle located on each side of the vehicle. For straight ahead driving, each bearing pair supports a radial load equal to one-half the vehicle front weight which is referred to as V.G.R. or "Vertical Ground Reaction" on Figure 6. In addition, with angular contact bearings (either ball or tapered roller), the smaller



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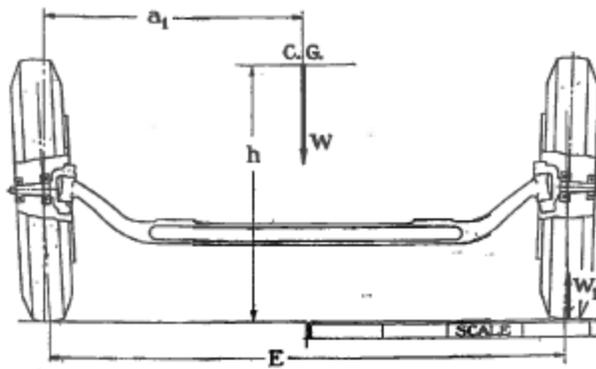
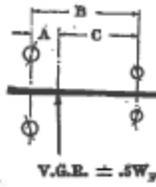
outer bearing supports an induced thrust load imposed by the larger more heavily loaded inner bearing. For a typical ball bearing pair, this thrust load equals  $.454(\text{inner bearing radial load} - \text{outer bearing radial load})$ . Using this information and the information given in the introduction, the equations for the load on the inner and outer



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**Figure 6**

**Wheel Bearings**





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bearings follow:

$$L_I = .5W_F (C/B) \quad L_O = .5W_F (A/B) \quad T_O = .227W_F (C-A)/B$$

$L_I$  is the inner bearing radial load in pounds.  $L_O$  is the outer bearing radial load.  $T_O$  is the outer bearing thrust load.  $W_F$  is the vehicle front weight. A, B, and C are bearing locating dimensions in inches as shown on Figure 6. With the loads now known, bearing life can be calculated for straight ahead driving. In general, the required straight ahead driving life for front wheel bearings is 2000 to 3000 B10 hours on an overloaded vehicle travelling at 100 miles per hour (equivalent to 200,000 to 300,000 road miles). Front wheel bearing loads can be much higher when the vehicle is rounding a corner. The calculation of cornering loads on vehicle front wheel bearings requires an analysis of centrifugal force acting on the front center of gravity of the vehicle which will be one of the topics discussed in another course.

### **Integral Spindle Wheel Bearing**

One of the more recent automotive wheel bearing advances has been the design, development, and mass production of the "Integral Spindle Wheel Bearing" by one of the major U.S. car companies. It was put into production when the automobile companies started manufacturing light-weight front wheel drive vehicles. The new design combines the spindle, hub, and bearings into one lubricated for life and sealed package that bodes well for the new automotive concept of "modular assembly" as it is a one-piece unit that bolts to the vehicle and the wheels bolt to it. The unit is assembled, adjusted, lubricated, sealed, and tested on automatic equipment at the bearing manufacturing plant. Prior to that, wheel bearings and associated parts were individually hand assembled at the automotive manufacturing plants. Figure 7 has a section drawing of an integral spindle drive wheel assembly and Figure 8 has a section drawing of an integral spindle non-drive wheel assembly. The drive wheel assembly has an internal spline into which the drive shaft male spline is inserted. Both designs incorporate integral speed sensors for anti-lock braking systems (ABS).

Figure 9 has a drawing of the front wheel bearing assembly that was used when rear wheel drive vehicles were the prime manufacturing mode in the U.S. Separate tapered roller bearings were assembled onto a spindle and in a hub, lubricated, sealed, and hand adjusted. Figure 10 has a drawing of a double row tapered roller bearing design that was used on low production front drive vehicles before the arrival of the new light-weight front drive vehicles.

It can be seen when comparing the two designs that cost savings are realized with the new integral spindle designs because ball bearing pathways are ground directly onto



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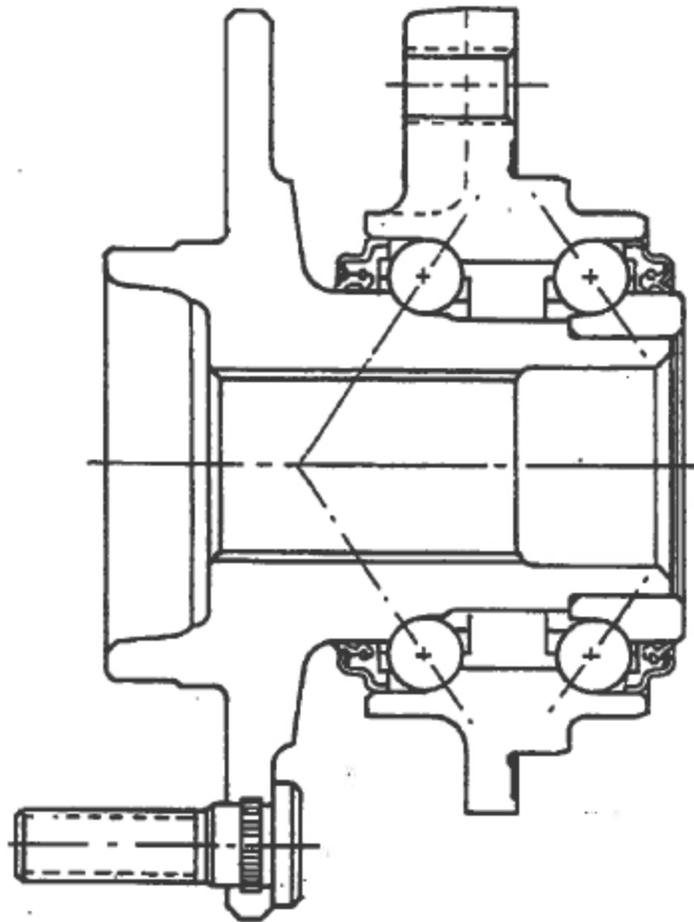
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spindles and into hubs eliminating pieces and simplifying assembly. Also, in the previous design, tapered roller bearings were hand adjusted in the auto assembly plant while, in the new design, ball bearings are precisely adjusted on automated equipment at the bearing plant eliminating human error.



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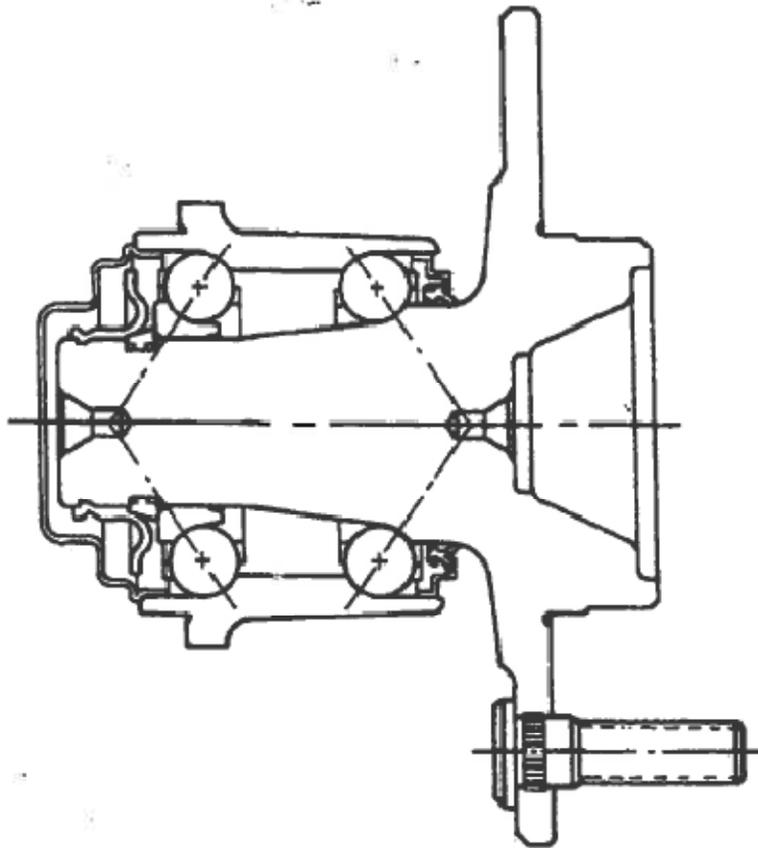
**Figure 7**  
**Integral Spindle**  
**Drive Wheel Bearing**





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**Figure 8**  
**Integral Spindle**  
**Non-Drive Wheel Bearing**

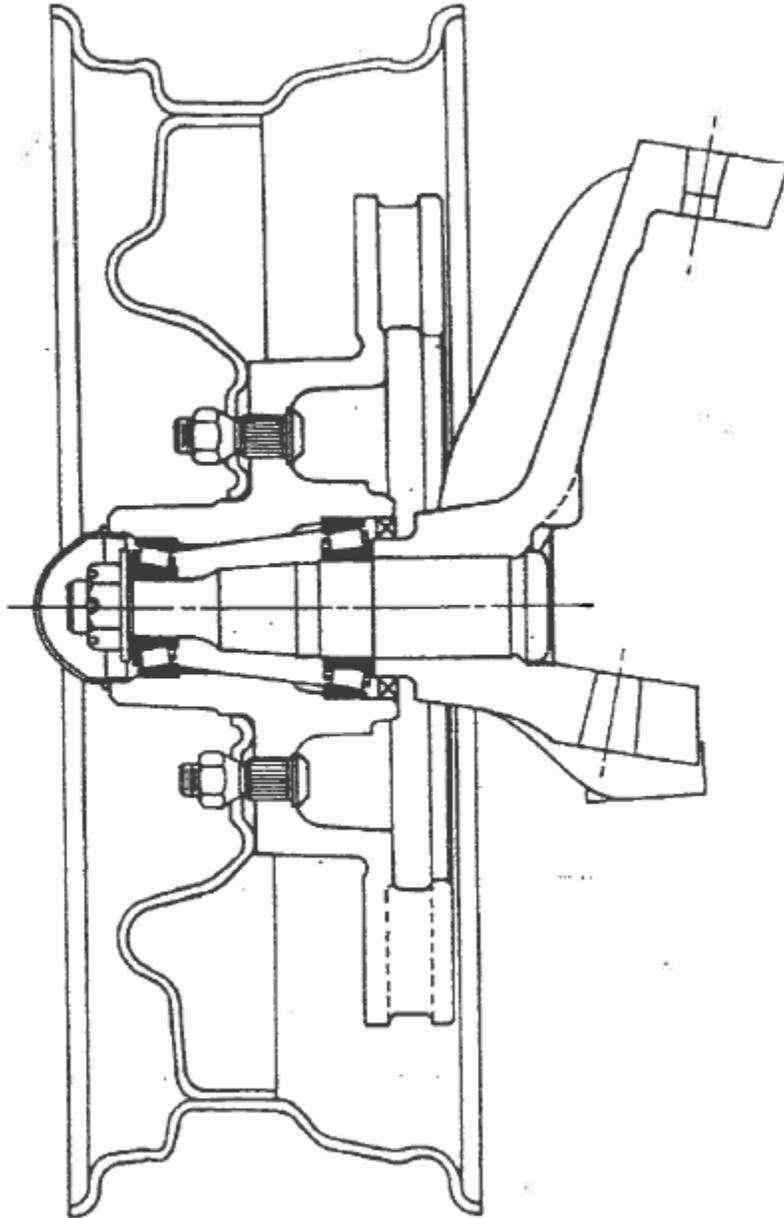




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**Figure 9**

**Wheel Bearing Arrangement**

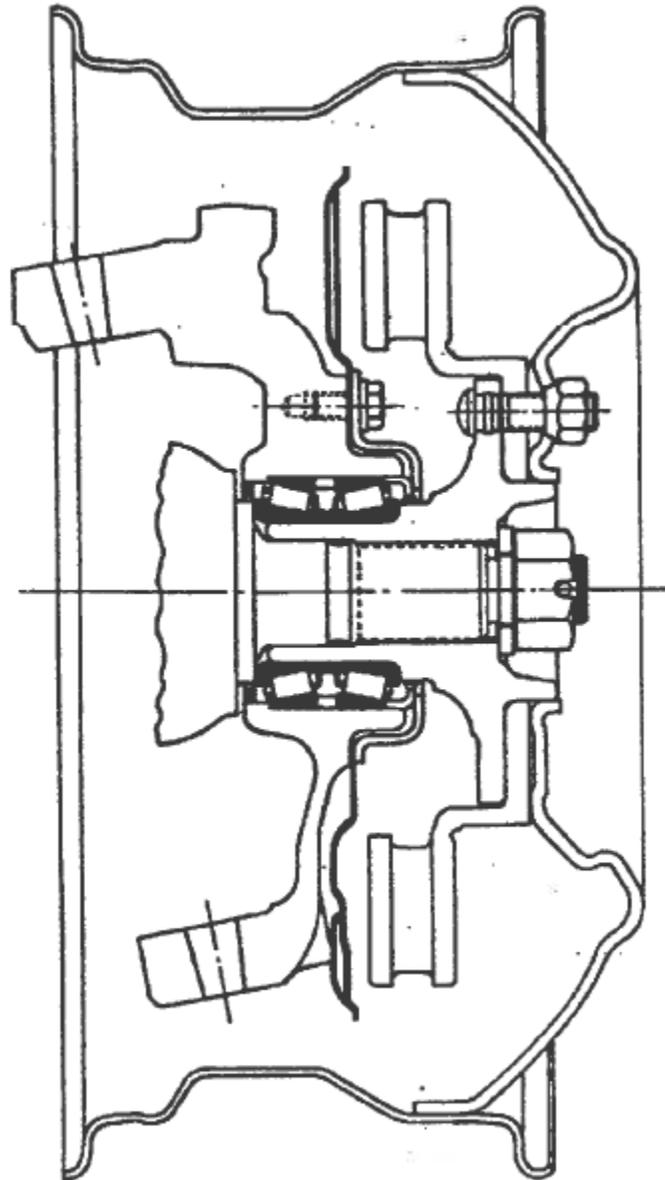




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**Figure 10**

**Wheel Bearing Arrangement**





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One of the biggest reasons for using balls as the rolling element in the new design is for safety considerations. Numerous tests were run comparing balls to tapered rollers in wheel bearings in vehicles. It was found that when ball bearings fail, they make such a loud noise that the driver stops the vehicle and drives no further. When tapered roller bearings fail, the noise is not as loud and the vehicle is continued to be driven. In some instances, if the vehicle is continued to be driven for too long a period of time, catastrophic failure occurs. The resulting heat generation is so great that steel components actually melt; wheels leave the vehicle; and there is a possibility that the vehicle will overturn.

Ordinarily, tapered roller bearings have more load carrying capacity than similar-sized ball bearings. For that reason, an extra number of large diameter balls were incorporated in integral spindle bearings so that the resulting product would match the load carrying capacity of tapered roller bearings. Integral spindle wheel bearings have been designed and produced for most of today's passenger cars and it appears they will be continued to be used for the foreseeable future.

### **Engine Coolant Pump Bearings**

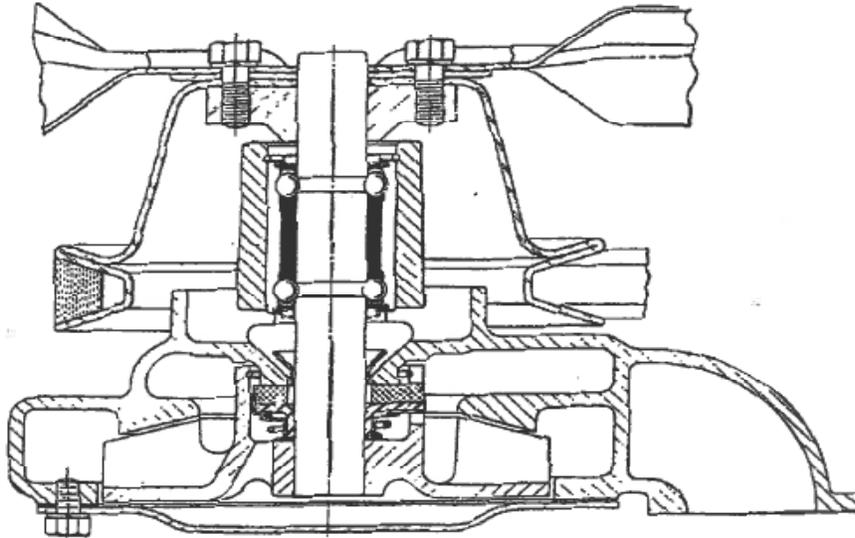
Engine coolant pump bearings with integral shafts were invented by an American bearing company in 1935 and have been in continuous use in automotive vehicles ever since. At the top of Figure 11, is a sketch of an integral shaft bearing in a traditional



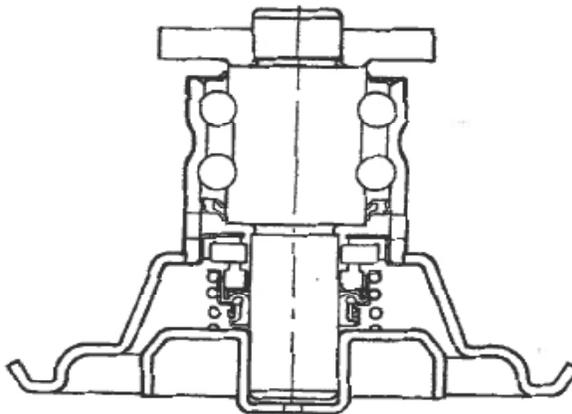
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**Figure 11**

**Automotive Water Pumps**



Traditional Automobile Waterpump  
With Integral Shaft Ball Bearing



Later Version Waterpump  
With Stepped Shaft Bearing



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waterpump. The inner pathways are ground directly onto a hardened steel shaft permitting use of a larger shaft for increased strength and durability. The two ball rows are spread apart for increased stability and resistance to moment loading. A variety of seals can be used at each end to exclude contaminants and contain a large amount of lubricant that is deposited between the spread apart ball rows. In the automotive application as seen at the top of Figure 11, the engine cooling fan is a press fit on the bearing front shaft extension and the waterpump impeller is a press fit on the rear shaft extension.

At the bottom of Figure 11 is a sketch of a later version automotive waterpump with a stamped housing and a stepped shaft bearing. This design features not only ground on shaft pathways but outer pathways that are ground into the housing of the waterpump for additional cost savings. This design has proven to be lighter and less costly than cast metal designs previously used. It is used in new light-weight front drive vehicles with transverse mounted engines.

### **Drive Axle Bearings**

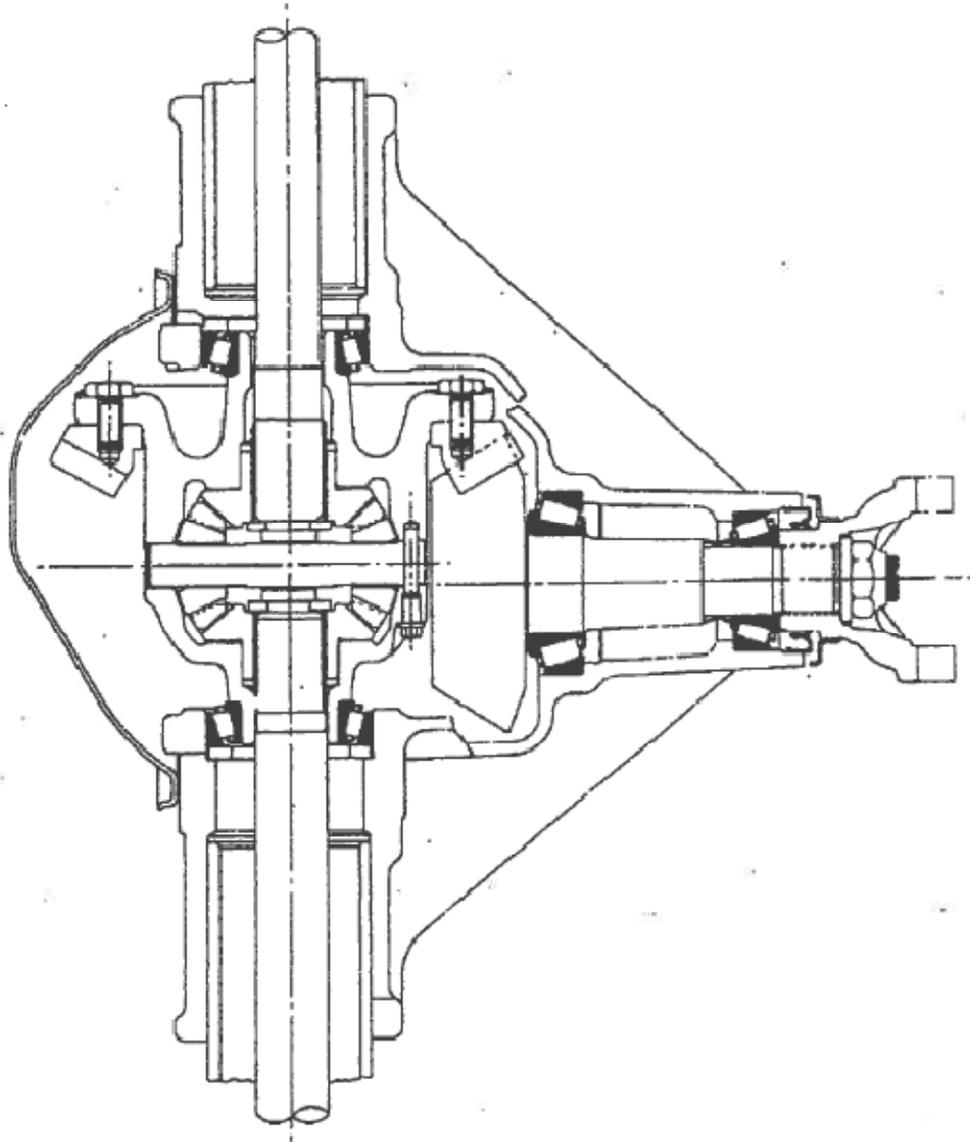
Drive axles are assemblies that transfer engine power from the transmission to the vehicle drive wheels. For a front drive vehicle, the drive axle is combined with the transmission and called the "transaxle". Figure 12 has a sketch of the center section of a drive axle for a rear drive vehicle. The input is on the right and is driven by the drive



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**Figure 12**

**Drive Axle Differential**



**Drive Axle Bearing and Gear Arrangement**



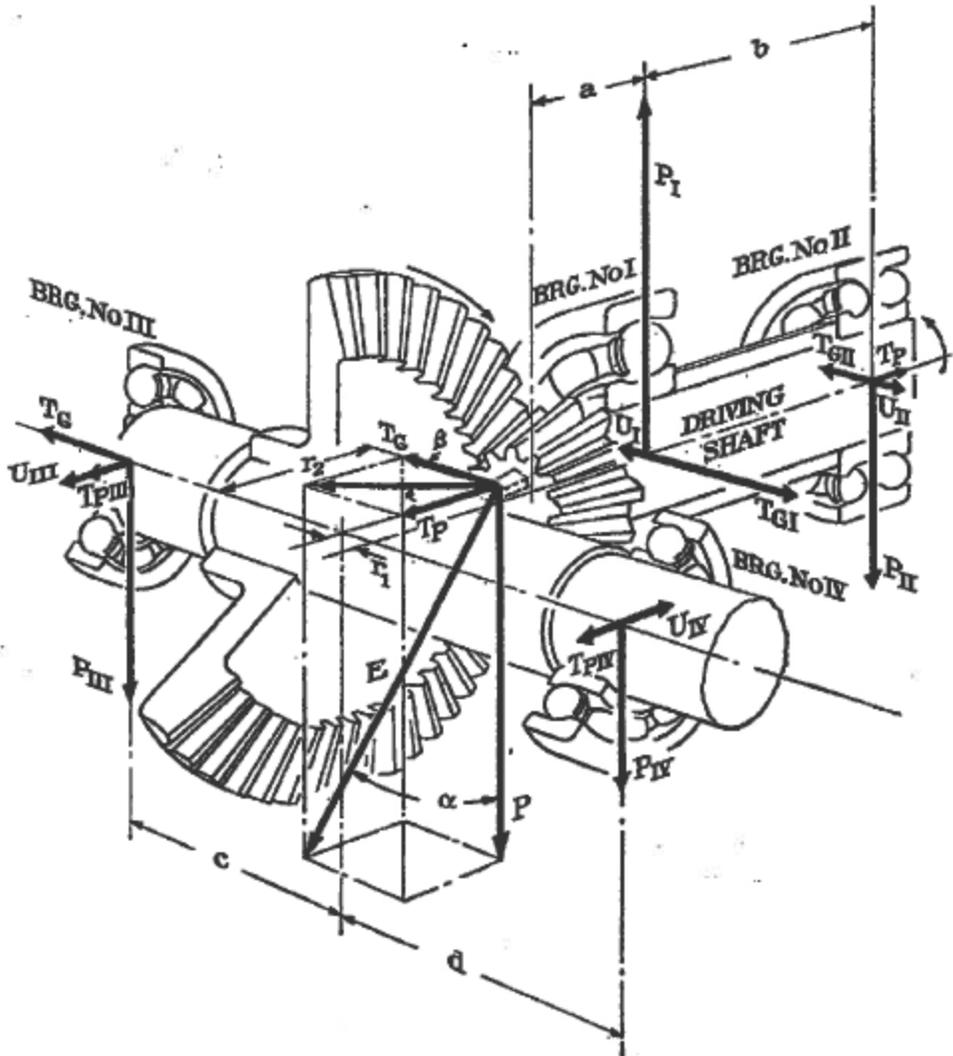
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shaft which runs the length of the vehicle. The input bevel gear pinion drives the larger ring gear which drives the center differential. The differential directs power to the two output shafts even though one may be rotating faster than the other such as when a vehicle is rounding a corner. The differential was patented in 1885 by German engineer Karl Benz and hasn't had any significant changes since. Shown on Figure 13 is a bevel gearset where the axes of the pinion and ring intersect. A hypoid gearset has the pinion axis lowered and is used where it is desirable to lower the drive shaft and increase vehicle interior space.

For a bevel gearset, the torque and tangential force are calculated as was done previously with other gearsets. In the bevel gearset in Figure 13, the tangential force acts down. There is also a thrust force that acts along the pinion axis and another that acts along the ring gear axis. These three components of force are used to calculate the load on each of the four bearings. Calculations are in Appendix III.

**Figure 13**  
**Bevel Gear Bearings**





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**Appendix I**

$$Q = H.P. \times 63025 / N$$

Q is the torque in inch-pounds (in-lbs). H.P. is the input horsepower. N is the driving shaft revolutions per minute (rpm).

$$P = Q / r$$

P is the tangential force of the driving gear in pounds (lbs). r is the pitch radius of the driving gear in inches. The pitch radius equals one-half the number of teeth divided by the diametral pitch. Diametral pitch is a measure of tooth size.

$$S = P \tan \alpha$$

S is the separating force in pounds.  $\alpha$  is the gear tooth pressure angle in degrees.

$$P_1 = Pa / (a + b)$$

$P_1$  is the load on Bearing I due to the tangential force. a and b are bearing I locating distances in inches.

$$S_1 = Sa / (a + b)$$

$S_1$  is the load on Bearing I due to the separating force.

$$L_1 = (P_1^2 + S_1^2)^{1/2}$$

$L_1$  is the total radial load on bearing I. The loads on bearings II, III, and IV are calculated in a similar manner. The equation for bearing operating life is as follows:

$$L_{B10} = 3000 (C/P)^{10/3} (500/S)$$

$L_{B10}$  is the life in hours that 90% of the bearings are expected to operate under the stated load and speed conditions without failure. C is the bearing capacity in pounds which is found in bearing catalogs. P is a factor which accounts for both radial and thrust loads on bearings which is also found in catalogs. S is the bearing speed in rpm.

**Appendix II**

$$P_1 = Q / r_1$$

$P_1$  is the tangential force at the pitch circle of the constant mesh gearset in pounds (lbs). Q is the input torque calculated from the input horsepower and speed in in-lbs.  $r_1$  is the pitch radius of the constant mesh drive pinion in inches.

$$S_1 = P_1 \tan \alpha$$

$S_1$  is the separating force at the constant mesh gears in pounds.  $\alpha$  is the gear tooth pressure angle in degrees.

$$P_2 = P_1 (r_2 / r_3)$$

$P_2$  is the tangential force at the pitch circle of the second speed gearset in pounds.  $r_2$  is the pitch radius of the constant mesh gear in inches.  $r_3$  is the pitch radius of the second speed drive gear.

$$S_2 = P_2 \tan \alpha$$



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$S_2$  is the separating force at the second speed gearset in pounds.

$$P_{1A} = P_1 (b/a)$$

$P_{1A}$  is the force on bearing A due to the tangential force acting on the overhung constant speed pinion.  $b$  and  $a$  are bearing A and B locating dimensions.

$$S_{1A} = S_1 (b/a)$$

$S_{1A}$  is the force on bearing A due to the separating force acting on the overhung constant speed pinion.

$$P_{2A} = P_2 (d/c)(b/a)$$

$P_{2A}$  is the force on bearing A due to the second speed gear mesh tangential force acting on the overhung constant speed pinion.

$$S_{2A} = S_2 (d/c)(b/a)$$

$S_{2A}$  is the force on bearing A due to the second speed gear mesh separating force acting on the overhung constant speed pinion.

$$L_A = [(P_{1A} - P_{2A})^2 + (S_{1A} + S_{2A})^2]^{1/2}$$

$L_A$  is the total radial load acting on bearing A. The calculations for the loads on the remaining bearings are similar. Now that the loads are known, bearing life can be determined.

### Appendix III

$$P = Q/r_1$$

$P$  is the tangential force that acts down in pounds.  $Q$  is the torque calculated from the input horsepower and speed.  $r_1$  is the mean pitch radius of the pinion.

$$T_G = P (\tan \alpha \cos \beta)$$

$T_G$  is a horizontal force that acts along the axis of the gear.  $\alpha$  is the tooth pressure angle in degrees.  $\beta$  is  $1/2$  the pinion pitch cone angle which equals  $\tan^{-1}$  times (the number of teeth in the pinion/ the number of teeth in the gear).

$$T_P = P (\tan \alpha \sin \beta)$$

$T_P$  is a horizontal force that acts along the axis of the pinion.

$$P_I = P (a+b)/b$$

$P_I$  is the force acting on bearing I due to the load  $P$ .  $a$  and  $b$  are bearing I locating distances.

$$T_{GI} = T_G (a+b)/b$$

$T_{GI}$  is the force acting on bearing I due to the load  $T_G$ .

$$U_I = T_P (r_1/b)$$

$U_I$  is the force acting on bearing I due to the load  $T_P$ .

$$L_I = [P_I^2 + (T_{GI} - U_I)^2]^{1/2}$$

$L_I$  is the total (radial) load acting on bearing I.