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Automotive Bearing Applications

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Robert P. Tata, P.E.



Continuing Education and Development, Inc.

P: (877) 322-5800

info@cedengineering.com

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Introduction

There are fundamental principles that the automotive bearing Application Engineer uses to evaluate the forces (loads) that are imposed on bearings and how they affect the operating life of a bearing in a mechanical device. Following are some of the basic principles that are used.

The loads on bearings are either radial or thrust. The sketch at the top of Figure 1 shows that radial loads act perpendicular to the bearing axis of rotation and thrust loads act parallel to the axis of rotation. In some applications there are two radial loads acting 90 degrees apart, as shown on the second sketch of Figure 1. The Pythagorean Theorem is then used to calculate the resultant radial load.

In most applications, there are two bearings supporting a rotating shaft in a piece of mechanical equipment. The third sketch on Figure 1 shows an applied load straddle mounted between two shaft support bearings. The radial loads on bearings I and II are calculated as follows:

$$L_I = \text{Load } b / (a+b) \quad L_{II} = \text{Load } a / (a+b)$$

L is the radial load on bearings I and II in pounds. a and b are bearing locating dimensions in inches shown on the third sketch of Figure 1. It can be seen that, because the applied load is closer to bearing II, it supports the greater portion of the load. The fourth sketch on Figure 1 shows an overhung applied load acting on a shaft supported by two bearings. Overhung loads put a heavy force on the adjacent bearing. The following equations are used to calculate the radial load on bearings III and IV:

$$L_{III} = \text{Load } (d/c) \quad L_{IV} = \text{Load } (c+d)/c$$

It can be seen that the load on adjacent bearing IV is greater than the applied load itself. The loads acting on a bearing (in pounds) and its speed of rotation (revolutions per minute) are used to calculate bearing B10 life. Bearing B10 life predicts how many hours of operation 90% of the bearings will endure. The equation is as follows:

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

L_{B10} is the bearing B10 life. C is the bearing capacity in pounds found in industry catalogs. P is the equivalent radial load which takes into account both radial and thrust loads and is also found in industry catalogs. S is the speed in revolutions per

minute (rpm). Should a bearing operate under a number of different loads and speeds, the following equation is used to calculate B10 life:

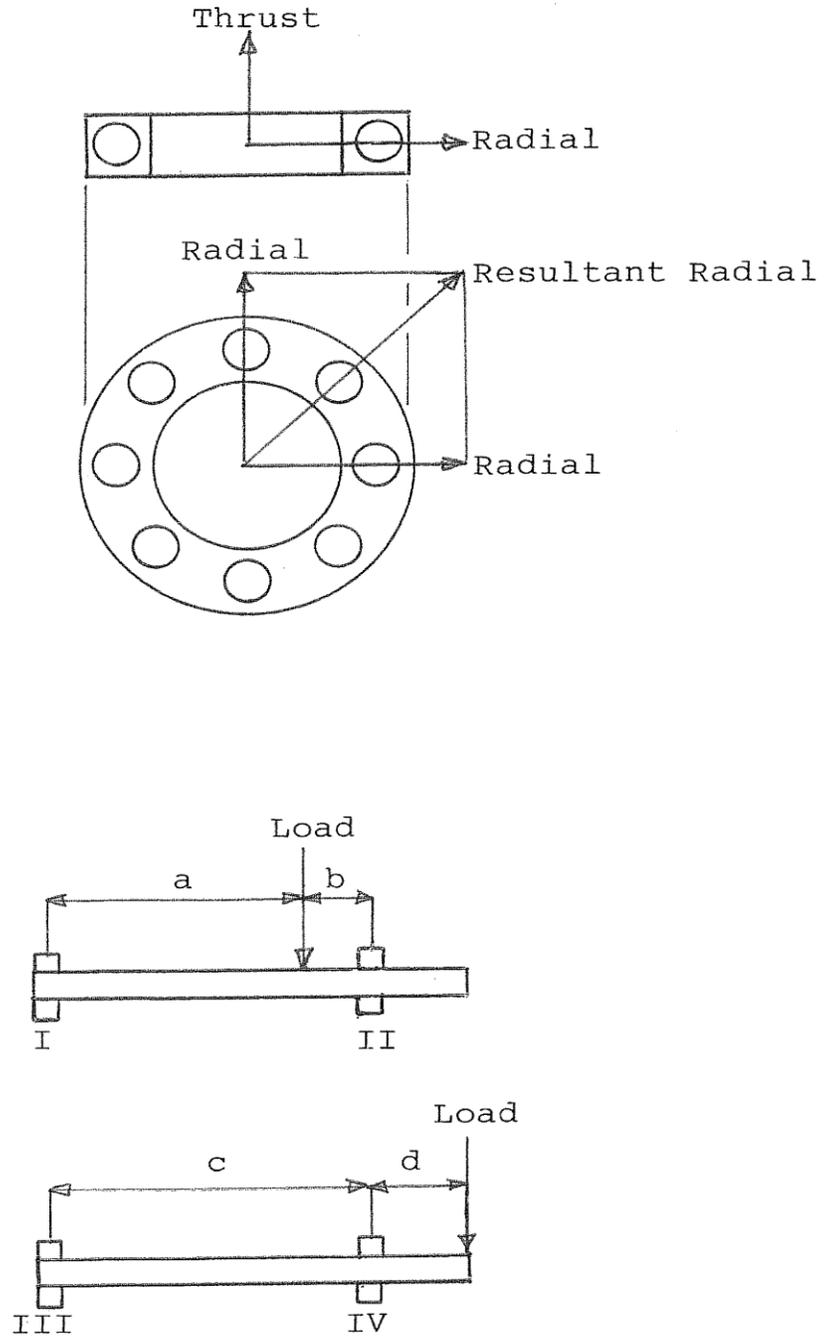
$$L_{B10} = 1/[(t_1 /L_1)+(t_2 /L_2)+(t_3 /L_3)+etc.]$$

L_{B10} is the bearing B10 life in hours. t is the time spent at each different life (L) level. Bearing life calculations are necessary to determine if predicted values meet actual design requirements. The following table gives approximate bearing life levels for other survival rates should the application require something other than B10 life:

<u>% Survival</u>	<u>B - Life</u>	<u>% of B10 Life</u>
99	B-1	21
98	B-2	33
95	B-5	62
90	B-10	100
50	B-50	400
40	B-60	500

The information above will be used in the automotive topics that are discussed later in this course.

Figure 1
Bearing Loads



Automotive Front 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 the traditional rear drive automotive vehicles are used in closely mounted pairs with one bearing referred to as the inner (inboard) bearing and the other, the outer (outboard) bearing. (See Figure 2.) Each bearing pair is installed on a spindle located on each side of the vehicle. There are three things that have to be known before the correct bearings can be chosen for this application:

- 1) Bearing loads during straight ahead driving
- 2) Bearing loads during cornering
- 3) Spindle stress

For straight ahead driving each bearing pair supports a radial load equal to one-half the vehicle front weight. In addition, for angular contact ball bearings (or tapered roller bearings), the smaller outer bearing supports an induced thrust load produced by the larger more heavily loaded inner bearing which, for a typical bearing pair, equals $.454 \times$ (the radial load on the inner bearing minus the radial load on the outer bearing). Using the information above and the information in the introduction, the equations for the load on the inner and outer bearings are as follows:

$$L_I = .5W_F (C/B) \quad L_O = .5W_F (A/B) \quad L_{OT} = .227W_F (C-A)/B$$

L_I is the inner bearing radial load in pounds. L_O is the outer bearing radial load. L_{OT} is the outer bearing thrust load. W_F is the vehicle front weight. C , B , and A are bearing locating dimensions in inches shown on Figure 2. The speed of the bearings in revolutions per minute (rpm) = $168 \times$ (miles per hour/tire radius in inches). Now that loads and speeds are known, bearing life can be calculated for straight ahead driving according to the information given in the introduction and compared to design requirements.

When cornering, there are two factors that produce radial loads on the inner and outer bearings. The first is a radial load on each bearing shown on Figure 3 as G_I and G_O which equals the amount calculated above for straight ahead driving plus a second increment due to a moment load created by centrifugal force CF acting on the front center of gravity CG when the vehicle is in a turn. The second is a radial load on each bearing shown on Figure 3 as S_I and S_O created from a moment load produced by the horizontal skid reaction S at the tire to pavement contact area. The equations are as follows:

$$L_I = G_I + S_I \quad L_O = G_O - S_O \quad L_{OT} = .454(L_I - L_O)$$

L_I is the radial load in pounds on the inner bearing. L_O is the radial load on the outer bearing. L_{OT} is the thrust load on the outer bearing. The equations for G_I , G_O , S_I , and S_O are in Appendix A. Bearing loads during cornering can be very high compared to bearing loads during straight ahead driving.

The spindles or shafts that the front wheel bearings mount on are relatively small, highly stressed tapered components that support the entire front weight of the vehicle. They are manufactured using quality steel and are machined to close precision tolerances. The spindles are considered to be stressed to a maximum when the vehicle is in a turn at such speed that the entire front weight is carried by the outer wheel. The equation for spindle stress is as follows:

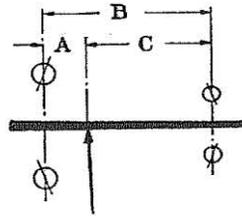
$$S = MC/I = M/.0982d = W_F (.6R_r - A) /.0982d$$

S is the spindle stress in pounds per square inch. M is the maximum bending moment around the center of the inner bearing in inch-pounds. C is the distance in inches from the center of the spindle to the outermost point which is the radius. I is the spindle moment of inertia in inches to the fourth power. d is the diameter of the spindle in inches. W_F is the vehicle front weight in pounds. $W_F (.6R_r)$ is a clockwise moment around the center of the inner bearing produced by the skid reaction $.6W_F$ (shown on Figure 3 as S) times the tire radius R_r . $W_F (A)$ is a counterclockwise moment produced by W_F (shown on Figure 3 as G) times A the distance from the center of the wheel to the center of the inner bearing. The maximum stress must be less than the allowable for the material that is to be used. The spindle outer end must be large enough to prevent excessive bending. A safe outer to inner diameter ratio is 0.63.

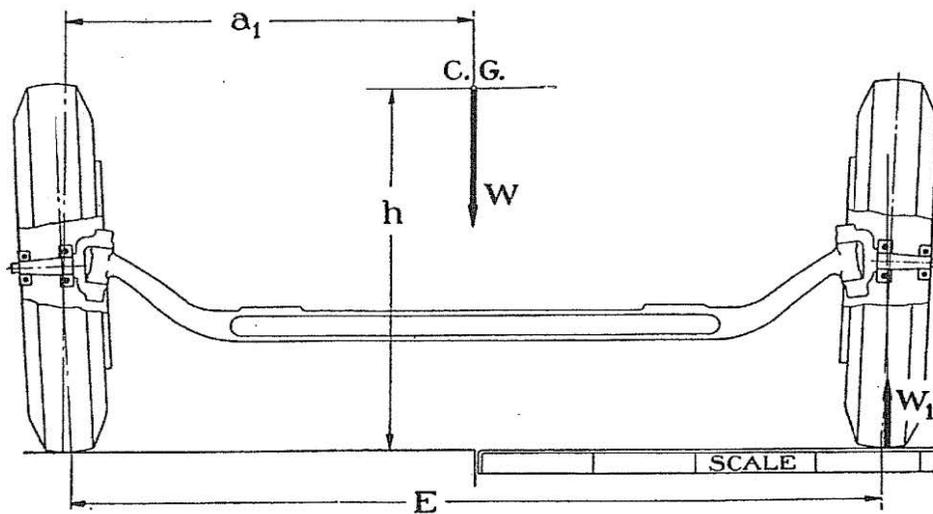
Figure 2

Automotive Front Wheel Bearing Loads

(Straight Ahead Driving)



Inner and Outer Bearing
Representation
With Load Line

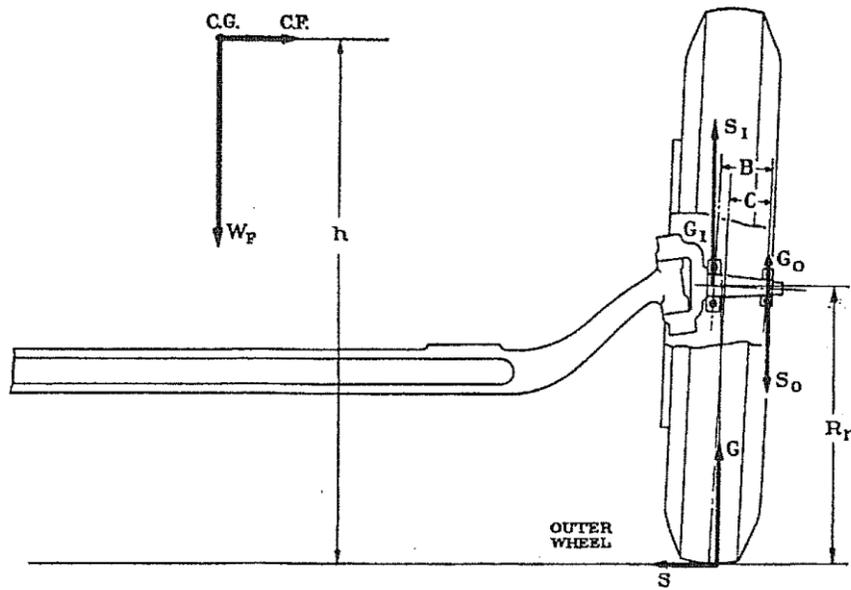


Automotive Front Bearing Arrangement

Figure 3

Automotive Front Wheel Bearing Loads

(Cornering)



Automotive Front Wheel Bearing
Cornering Loads

Automotive Rear Wheel Bearing

Before the bearing for the rear-drive automotive axle shaft can be selected, the diameter of the shaft at the wheel bearing location must be determined. The axle shafts are considered to be stressed to a maximum when a vehicle is traveling around a curve at such a speed that the entire loaded vehicle rear weight W_R is supported by the outer wheel. (See Figure 4.) The moment load acting on the center of the rear wheel bearing D is calculated as follows:

$$M = (W_R B) - (.6W_R R_f)$$

M is the moment load at the center of the rear wheel bearing D in inch-pounds. $(W_R B)$ is a counterclockwise moment produced by the loaded rear end weight W_R times the distance (B) from the center of the wheel to the center of the bearing. $(.6W_R R_f)$ is a clockwise moment load produced by the side skid reaction $(.6W_R)$ times the tire radius R_f . The same equation is used for the bending stress as was used for the front spindle as follows:

$$S = MC/I \quad \text{or} \quad d = 2.168(M/S)$$

S is the axle stress in pounds per square inch. M is the net bending moment as calculated above in inch-pounds. C is the distance from the center of the axle to the outermost point which is the radius. I is the axle moment of inertia in inches to the fourth power. d is the diameter of the axle at the rear wheel bearing D center. The equation can now be used to calculate the diameter of the shaft needed to produce a given stress level which must be less than the allowable for the material that is to be used. Once the diameter of the shaft has been calculated, it is standard practice to increase it to the next standard size bearing inside diameter (bore). The loads on the rear axle wheel bearing are due to the vertical rear end weight and the horizontal tractive force on each wheel. The equations follow:

$$W = .5W_R (B+C)/B \quad F = .05W_R (B+C)/B$$

W is a vertical radial load from the loaded vehicle rear weight in pounds. B and C are bearing locating dimensions in inches shown on Figure 4. F is horizontal radial load from the rear wheel tractive force in pounds which equals $(.1 \times .5W_R)$ or $(.05W_R)$. The total radial load on the rear wheel bearing equals:

$$L = (W^2 + F^2)^{1/2}$$

The bearing speed = $168 \times (\text{miles per hour}) / (\text{tire radius in inches})$. Now that load, speed, and bore are known, the correct bearing can be selected.

A new concept in automotive rear wheel bearings is the "drawn cup" roller bearing. Roller bearings normally have machined, hardened, and ground inner and outer rings. In an automotive rear bearing application, the bearing inner ring is a press fit on the axle and the outer ring is a loose push fit in the tube that surrounds the axle. As a cost saving measure, the drawn cup bearing outer ring is made on a press out of surfaced-hardened sheet steel instead of being machined, hardened, and ground. Figure 5 shows the steps in forming the outer ring and installing the rollers. The drawn cup bearing is a press fit in the tube and having no inner ring. It is run directly on the rear axle which is a hardened and ground part. When introduced into production, the drawn cup bearing has a significant cost savings over standard roller bearings when you apply the cost savings of a pair of bearings over the millions of cars manufactured every year. The drawn cup bearing also replaced a standard design bearing in automotive alternators resulting in further cost savings.

Figure 4

Automotive Rear Axle Shaft and Bearing

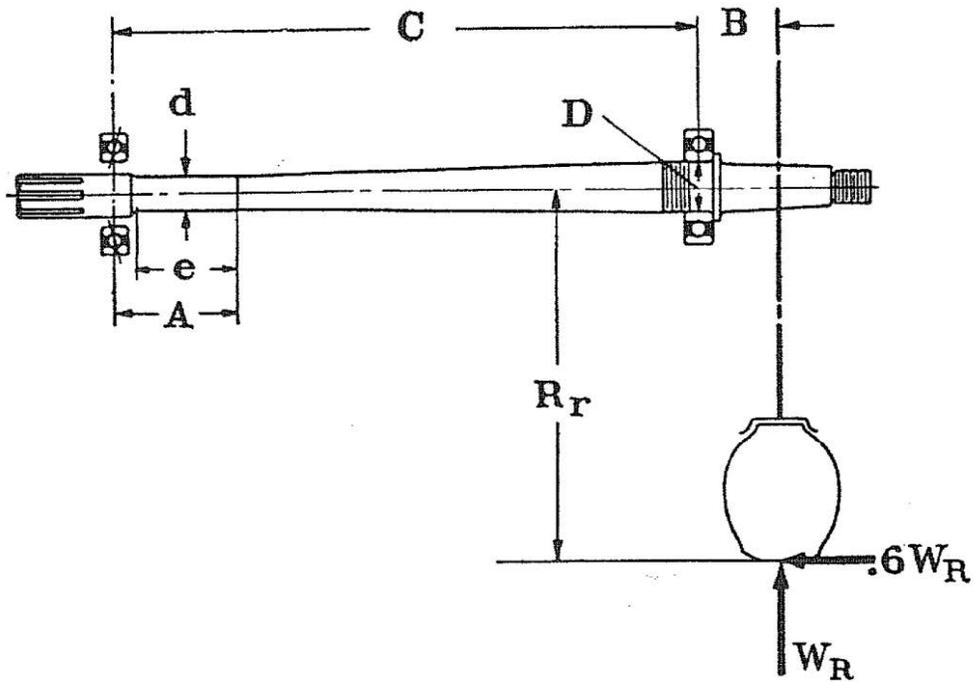
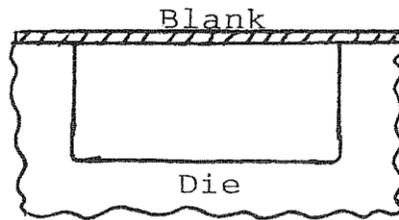
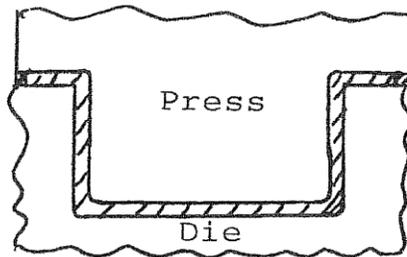


Figure 5

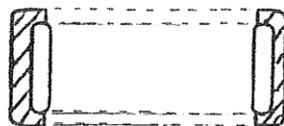
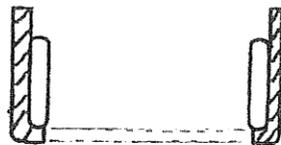
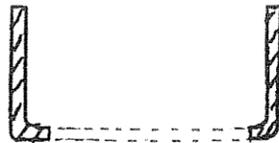
Drawn Cup Roller Bearing



Sheet metal blank
on top of die.



Cup drawn into
die with press.



Integral Spindle Wheel Bearings

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 at the same time that the auto 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 concept of "modular assembly", as it is a one piece unit that bolts to the vehicle and the vehicle wheels bolt to it. The unit is assembled, adjusted, lubricated, sealed, and tested on automated equipment at the bearing manufacturing plant. Prior to that time, wheel bearings and associated parts were individually hand assembled at the automobile manufacturing plants. Figure 6 shows a drawing of an integral spindle drive wheel assembly and Figure 7 shows a 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 sensing devices for anti-lock braking systems (ABS).

Figure 8 shows 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 into a hub, lubricated, sealed, and adjusted by hand. Figure 9 shows 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 into hubs and onto spindles eliminating pieces and simplifying assembly. Also, in the previous design, tapered 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 which is an important consideration for safety related items.

One of the biggest reasons for using balls as the rolling element in the new design is safety. Numerous tests were run comparing balls to tapered rollers in wheel bearings in vehicles. It was found when ball bearings fail, they make such a loud noise that the vehicle is stopped and is not driven further. When tapered roller bearings fail, the noise is not nearly as loud and the vehicle is continued to be driven. In some instances, if the vehicle is continued to be driven for a long time, catastrophic failure occurs. The resulting heat generation is so great that parts 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, in the new integral spindle designs, extra-large balls were incorporated so that the resulting product would match the load carrying capacity of tapered roller bearings. Integral wheel spindle bearings have been designed and produced with load carrying capacities for most automotive products being manufactured today. Past and present performance has shown that the new design will continue to be used for the foreseeable future.

Figure 6

Integral Spindle Drive Wheel Assembly

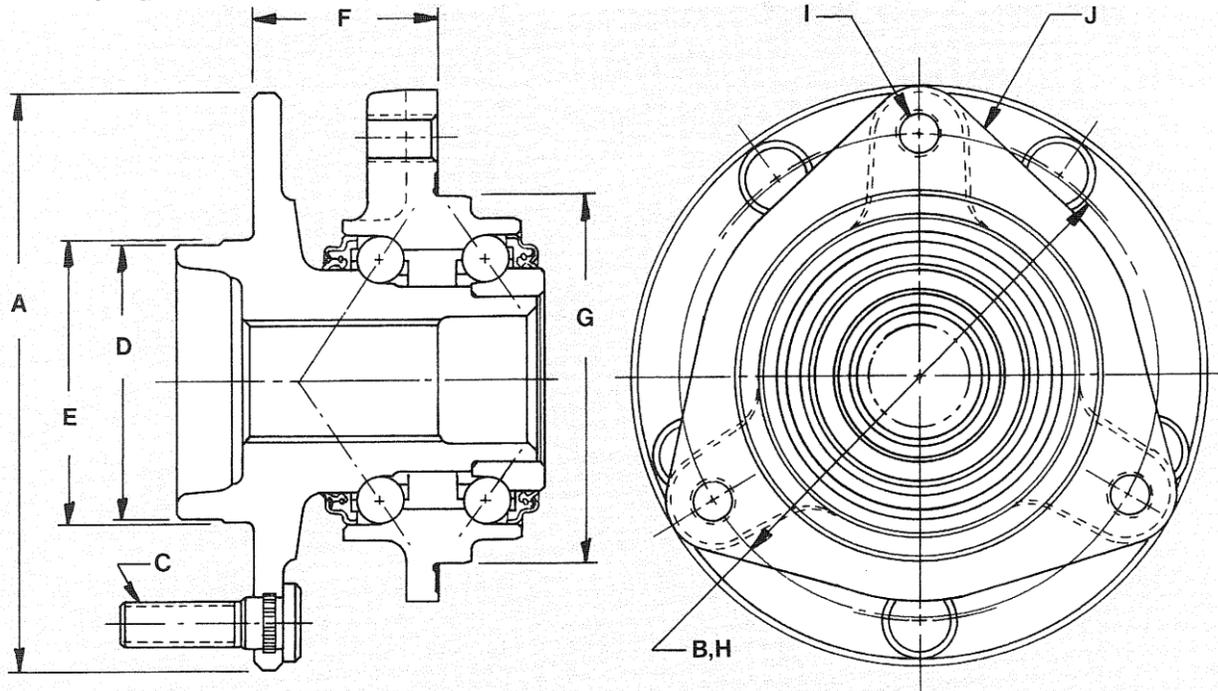


Figure 7

Integral Spindle Non-Drive Wheel Assembly

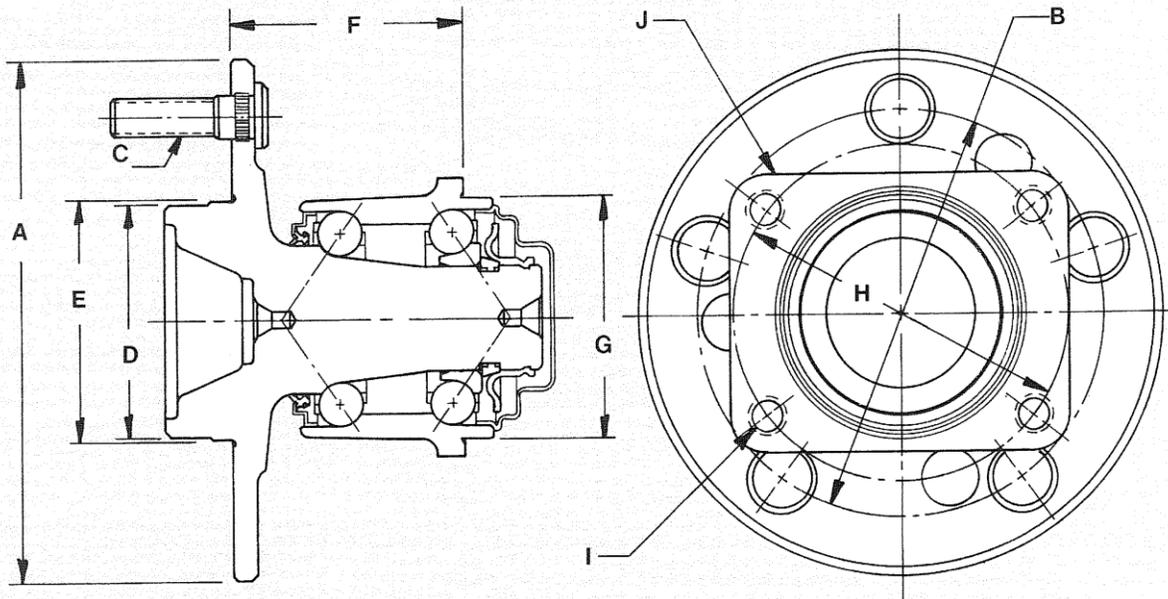
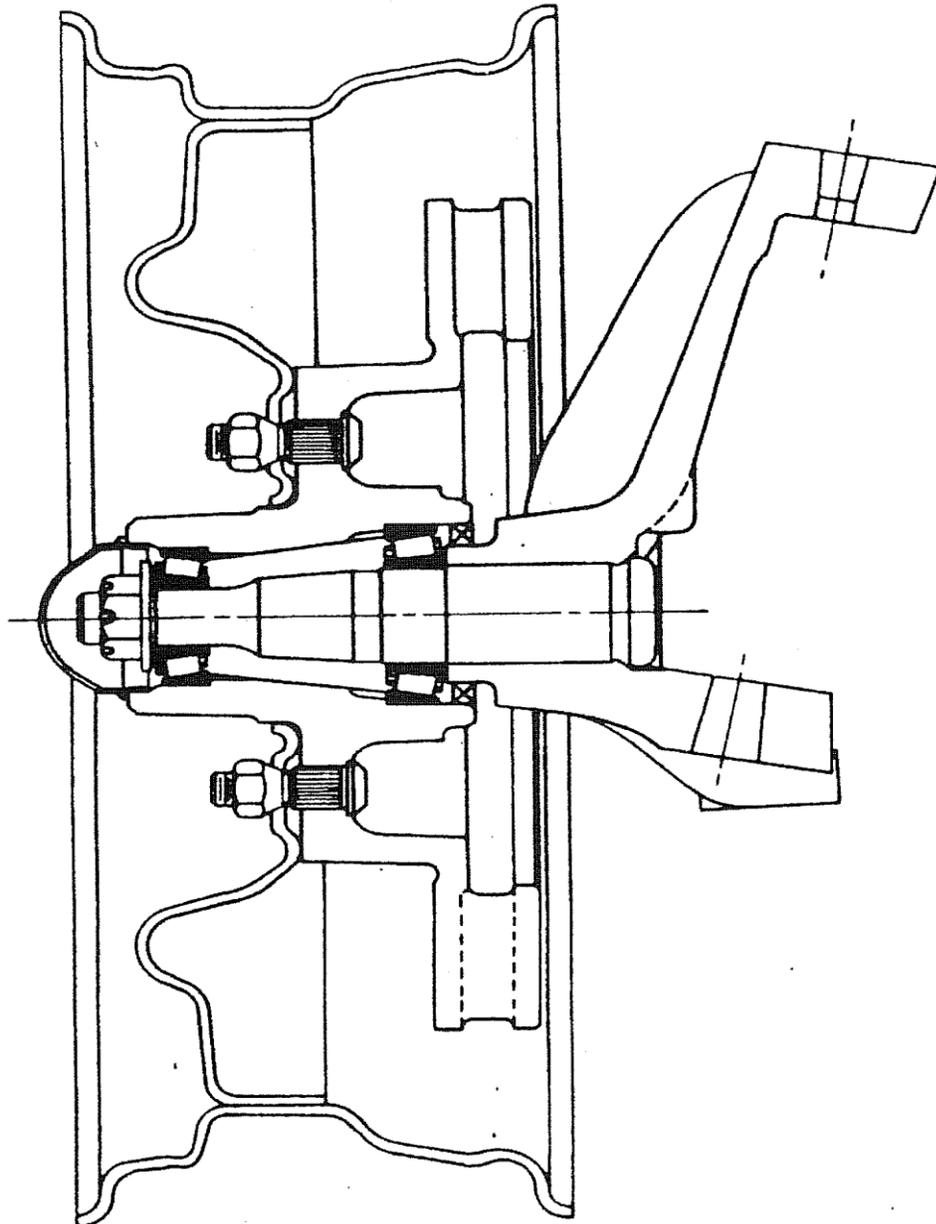


Figure 8

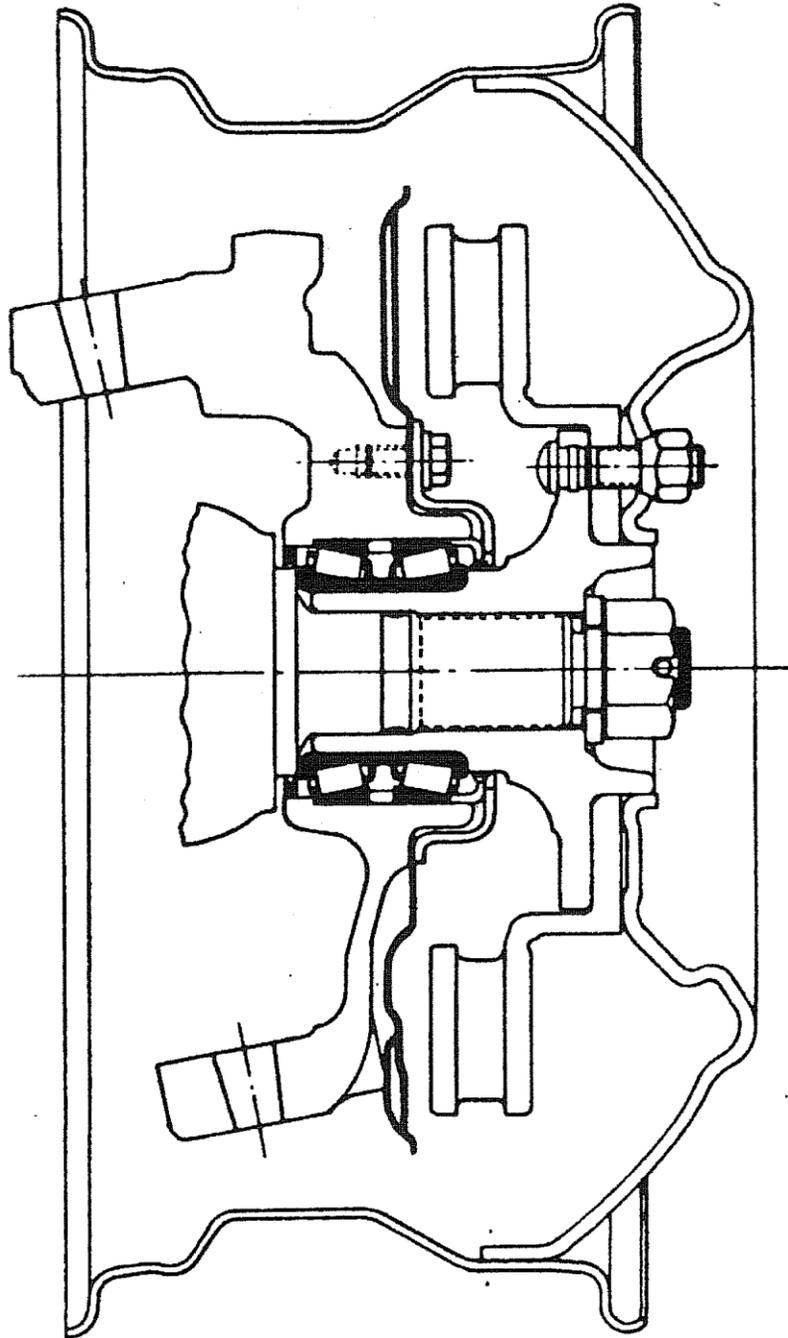
Front Non-Drive Wheel Bearing Assembly



Non-Drive Wheel
Tapered Bearing Arrangement

Figure 9

Front Drive Wheel Bearing Assembly



Drive Wheel Tapered Bearing Arrangement

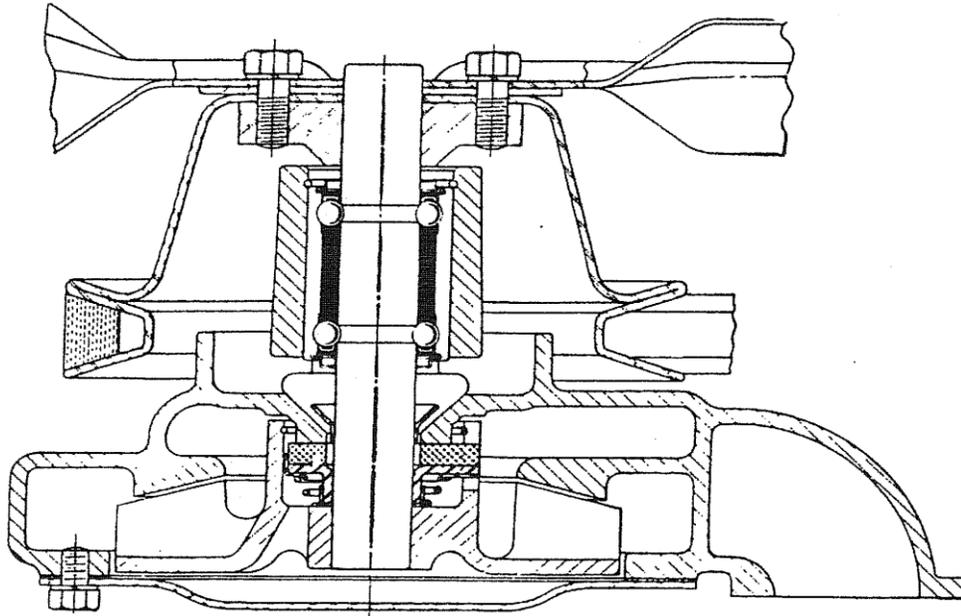
Integral Shaft Waterpump Bearings

The integral shaft waterpump bearing was invented by an American bearing company in 1935 and has been in continuous use in automotive vehicles ever since. At the top of Figure 10 is a sketch of an integral shaft bearing in a traditional waterpump. The inner pathways are ground directly onto a hardened steel shaft permitting the use of a larger shaft for increased strength and durability. The two ball rows are spread apart providing 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 10, 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.

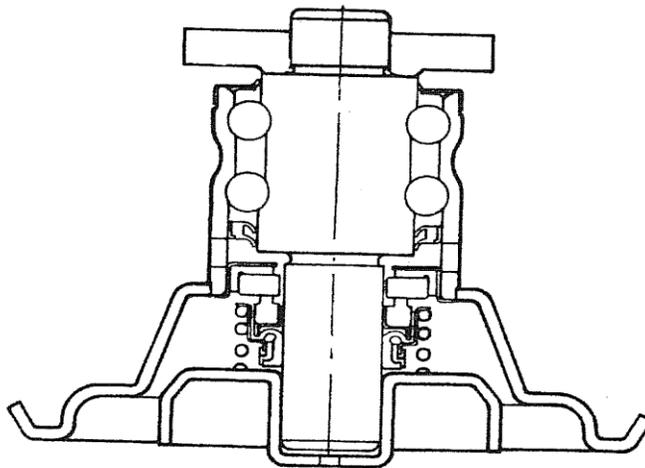
At the bottom of Figure 10 is a sketch of a later version automotive waterpump with a stamped housing and a stepped shaft waterpump bearing. This design features not only ground-on shaft inner pathways but outer pathways that are ground on the housing of the waterpump for additional cost savings. This design has proven to be lighter and less costly than cast designs previously used. It is incorporated on new light-weight front drive vehicles with transverse mounted engines.

Figure 10

Automotive Waterpumps



Traditional Automobile Waterpump
With Integral Shaft Ball Bearing



Later Version Waterpump
With Stepped Shaft Bearing

Automotive Differential Bearings

The automotive differential is located in the front transaxle in front drive automobiles and in the rear axle of rear drive vehicles. The differential is a necessity in the drive line of vehicles so that when cornering, each drive wheel can rotate at a different speed since the outer wheel travels further than the inner wheel. The differential was patented in 1885 by German engineer Karl Benz and hasn't had any significant changes since. Figure 11 has a sketch of a differential which is located in the center section of a drive axle. The smaller pinion gear drives the larger ring gear which drives the shaft on which the two sets of differential bevel gears are mounted. The driven members of the two differential gearsets are connected to the axles that drive the vehicle wheels. This arrangement allows power to be supplied to both wheels even though one wheel may be rotating faster than the other.

One type of gearset used in differential pinion and ring gears is the spiral bevel gear which is shown on Figure 12. It can be seen that the input force component E is divided into three vectors: P is the tangential driving force, T_P is the pinion gear thrust, and T_G is the ring gear thrust. The radial loads on bearing I due to the above three forces are as follows:

$$P_I = P (a/b) \quad T_{GI} = T_G (a/b) \quad U_I = T_P (r_1 /b)$$

P_I , T_{GI} , and U_I shown on Figure 12 are in pounds and are calculated in Appendix A. a and b are bearing locating dimensions in inches shown on Figure 12. r_1 is the mean pinion pitch radius in inches and is calculated in Appendix A. The total radial load on bearing I is as follows:

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

There is also a thrust load on bearing I equal to T_P . The radial loads on bearing II are as follows:

$$P_{II} = P (a+b)/b \quad T_{GII} = T_G (a+b)/b \quad U_{II} = T_P (r_1 /b)$$

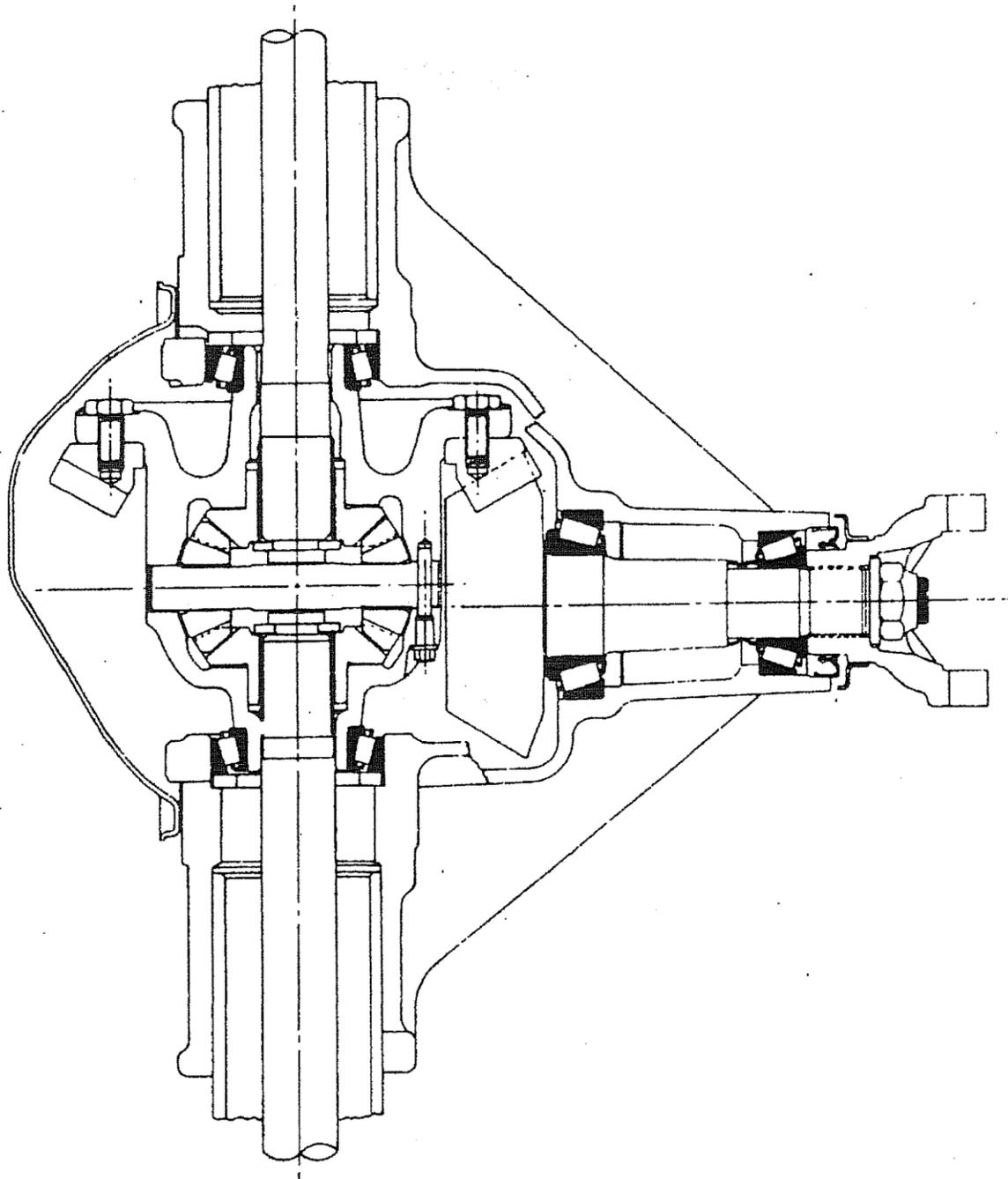
The total radial load on bearing II is as follows:

$$L_{II} = [P_{II}^2 + (T_{GII} - U_{II})^2]^{1/2}$$

The loads on bearings III and IV are calculated in a similar manner except substitute bearing locating dimensions c and d for a and b and mean pinion pitch radius with mean gear pitch radius which is calculated in Appendix A. The rpm of the ring gear = rpm of the pinion x (number of teeth in pinion) / number of teeth in gear). Now that loads and speeds are determined, the bearing life can be calculated.

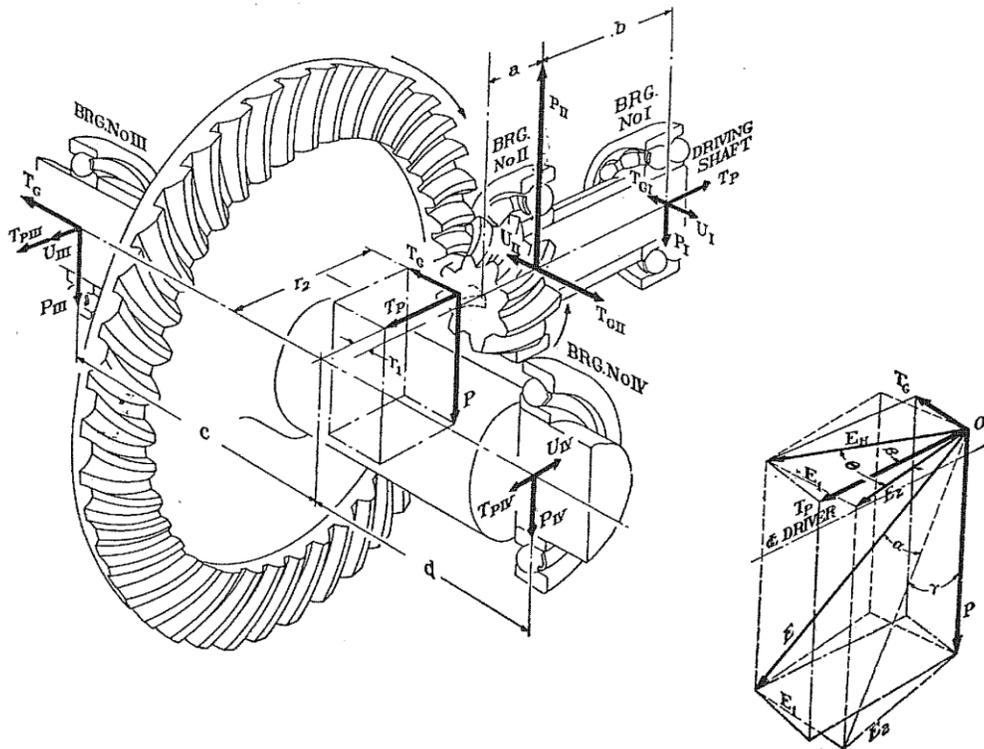
Figure 11

Drive Axle Differential



Drive Axle Bearing and Gear Arrangement

Figure 12
Spiral Bevel Gears



Automotive Transmission Bearings

Planetary gears are very versatile design tools that are used in automotive transmissions. Figure 13 has the sun gear driving the planet carrier with the ring gear (outer) held stationary. 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 used to drive the ring gear with the sun gear stationary, the output is an increase in speed and is used in vehicles as an overdrive gear. When the sun gear is used to drive the ring gear with the planet carrier stationary, the output is in the opposite direction as the input and is used in vehicles for reverse gear. Compound planetary gearsets with one ring gear, two sun gears, and two sets of planets, are used to provide the complete number of gears needed to propel automotive vehicles.

In a simple planetary gearset as shown in Figure 13, all the tangential and separating forces due to the input torque counteract each other resulting in no loads on bearings I, II, IV, and V. However, bearing III will be loaded because of the torque transmitted to the three planet gears through their diametrically opposed meshes. The load on each planet bearing is as follows:

$$L = 2P/3$$

where P is the tangential driving force calculated as follows:

$$P = HP \ 63025/Nr$$

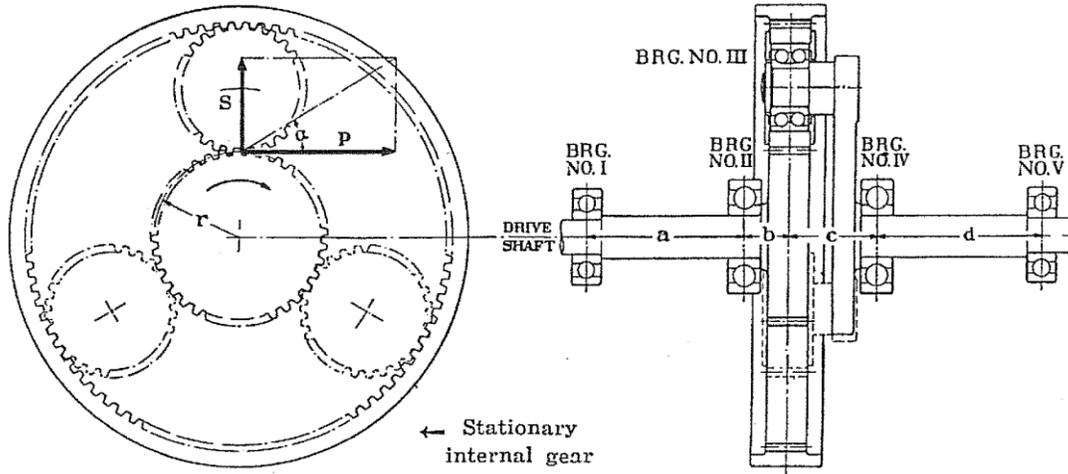
P is the tangential driving force in pounds. HP is the input horsepower. N is the speed in revolutions per minute (rpm) of the driving sun gear. r is the sun gear pitch radius in inches which equals one-half the number of teeth divided by the diametral pitch. The rpm of each planet bearing around its own center is calculated as follows:

$$N = EN/2d [1 - (d/E)^2]$$

E is the diameter of the circle drawn through the center of the three planets in inches. N is the rpm of the sun gear. d is the pitch diameter of the planet gear in inches. The pitch diameter equals the number of teeth divided by the diametral pitch. Now that the load and speed of each planet bearing has been determined, service life can be calculated and compared to design requirements.

Figure 13

Planetary Gears



Appendix A

A) Equations for front wheel bearing loads when a vehicle is turning an average corner with a 140-foot radius at 25 miles per hour are as follows:

$$CF = .5W_F V^2/rg = W_F(25 \times 5280/3600)/(140 \times 32.2) = .298W_F$$

CF is the centrifugal force acting on the vehicle front center of gravity (CG) in pounds. W_F is the vehicle front weight. V is the vehicle velocity in feet per second. r is the turning radius in feet. g is the acceleration due to gravity in feet per second per second. The centrifugal force radial load increment is as follows:

$$F = .298W_F x h/E$$

W_F is the vehicle front weight. h is the distance from the front center of gravity to the ground. E is the distance from front wheel to front wheel. The total front vertical ground reaction G, G inner, and G outer are as follows:

$$G = .5W_F + (.298W_F x h/E) \quad G_I = G(C/B) \quad G_O = G(A/B)$$

The horizontal skid reaction S, S inner, and S outer are as follows:

$$S = G/W \times .298W = .298G \quad S_I = S_O = S(R/B)$$

B) Spiral bevel gear loads:

$$P = Q/r_1 \quad T_P = P(\tan a \sin b / \cos y) + (\tan y \cos b) \quad T_G = P(\tan a \cos b / \cos y) - (\tan y \sin b)$$

Q is the input torque in inch-pounds. r_1 is the mean pinion pitch radius in inches which equals $\frac{1}{2}(\text{pinion pitch diameter} - \text{tooth face} \times \sin b)$. $b = \tan^{-1}(\text{number of teeth in pinion} / \text{number of teeth in gear})$. a is the tooth pressure angle. y is the spiral angle. For gear shaft bearing loads, bearing locating dimensions c and d replace a and b and r_2 replaces r_1 . r_2 equals $r_1 \times (\text{number of teeth in gear} / \text{number of teeth in pinion})$.