NASA
Technical Memorandum 81426

AVRADCOM
Technical Report 80-C-2

(NASA-TM-81426) EFFECT OF GEOMETRY AND OPERATING CONDITIONS ON SPUR GEAR SYSTEM POWER LOSS (NASA) 31 p HC A03/MF A01 CSCL 20I

N80-18406

Unclas
G3/37 47356

EFFECT OF GEOMETRY AND OPERATING CONDITIONS ON SPUR GEAR SYSTEM POWER LOSS

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Prepared for the
Third International Power Transmission and Gearing Conference
sponsored by the American Society of Mechanical Engineers
San Francisco, California, August 18-22, 1980

1. Report No. NASA TM 81426 AVRADCOM TR-80-C-2	2 Government Accession No	3 Recipient's Catalog No
4 Title and Subtitle EFFECT OF GEOMETRY AND OPERATING CONDITIONS ON SPUR GEAR SYSTEM POWER LOSS		5 Report Date
		6 Performing Organization Code
7. Author(s) Neil E. Anderson and Stuart H. Loewenthal		8 Performing Organization Report No E-350
		10 Work Unit No
9 Performing Organization Name and Address NASA Lewis Research Center and AVRADCOM Research and Technology Laboratories Cleveland, Ohio 44135		11 Contract or Grant No
		13 Type of Report and Period Covered Technical Memorandum
12 Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546 and U.S. Army Aviation Research and Development Command, St. Louis, Missouri 63166		14 Sponsoring Agency Code
15 Supplementary Notes		

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17. Key Words (Suggested by Author(s))	18 Distribution Statement
Spur-gear efficiency Efficiency Power loss Spur gear	Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif (of this page)	21. No of Pages	22. Price*
Unclassified	Unclassified		

* For sale by the National Technical Information Service, Springfield, Virginia 22161

E-350

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ABSTRACT

The results of an analysis of the effects of spur gear size, pitch, width and ratio on total mesh power loss for a wide range of speeds, torques and oil viscosities are presented. The analysis uses simple algebraic expressions to determine gear sliding, rolling and windage losses and also incorporates an approximate ball bearing power loss expression. The analysis shows good agreement with published data. Large diameter and fine-pitched gears had higher peak efficiencies but lower part-load efficiency. Gear efficiencies were generally greater than 98 percent except at very low torque levels. Tare (no-load) losses are generally a significant percentage of the full-load loss except at low speeds.

INTRODUCTION

With today's emphasis on minimizing energy consumption of rotating machinery, methods to accurately predict drive train power losses have taken on renewed importance. A significant source of power loss in many drive systems is due to the gearing. Many methods have been proposed to calculate gear power loss [1-5]. Most of these methods utilize a friction coefficient to calculate the gear power loss. Few consider the losses

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*Member ASME.

associated with forming an elastohydrodynamic film (rolling traction), gear windage or those associated with the support bearings. These later power loss terms contribute significantly to the power loss occurring under part-load operation. Consideration of these speed-dependent loss terms becomes important in determining the cumulative power consumption of machines that spend much of their operating lives at less than full-power levels.

Furthermore, most of these earlier methods do not conveniently account for the effects of gear mesh geometry, such as diametral pitch, tooth number, width, ratio, and operating conditions on gear power loss. An exception to this is the spur gear efficiency analysis of [5]. In this investigation instantaneous values of sliding and rolling power loss were integrated over the path of contact and averaged. The effect of gear geometry is incorporated into this analysis.

In [6] this approach was extended to include windage and support bearing loss terms and improved expressions for sliding and rolling traction loss components. This method showed good agreement with power loss data generated on a back-to-back spur gear test rig reported in [7]. The work of [6] concluded that the rolling traction, support bearings and, to a lesser extent, windage power losses comprise a significant portion of the total mesh loss.

The predictive technique of [6] makes an excellent tool for studying the effects of gear geometry and operating conditions on spur gear power loss and efficiency. Accordingly, the objectives of the present study are to use the method of [6] to investigate the effects of spur gear size, diametral pitch, ratio, width, lubricant viscosity, pitch line velocity and pinion torque on full- and part-load gear performance.

SYMBOLS

C_s	support bearing basic static capacity, N (lbf)
C_1 to C_8	constants of proportionality
D	pitch circle diameter, m (in.)
D_m	bearing pitch diameter, m (in.)
F_{ST}	static equivalent bearing load, N (lbf)
$\mathcal{F}$	face width of tooth, m (in.)
f	coefficient of friction
f_o	ball bearing lubrication factor
h	central film thickness, m (in.)
K	gear capacity factor
l_1 to l_6	path of contact distances, m (in.)
M	bearing friction torque, N-m (in-lbf)
M_L	load-dependent part of bearing friction torque, N-m (in-lbf)
M_V	viscous part of bearing friction torque, N-m (in-lbf)
m_g	gear ratio, D_2/D_1
n	rotational speed, rpm
P_{BRG}	total power loss in support bearings, kW (hp)
P_R	power loss due to rolling traction, kW (hp)
P_S	power loss due to tooth sliding, kW (hp)
P_W	power loss due to windage, kW (hp)
R	pitch circle radius, m (in.)
V_S	sliding velocity, $V_1 - V_2$, m/sec (in/sec)
V_T	rolling velocity, $V_1 + V_2$, m/sec (in/sec)
w	gear contact normal load, N (lbf)
$\dot{X}P$	distance along path of contact from initial contact to pitch point, m (in.)

x	distance along path of contact, m (in.)
μ	lubricant absolute viscosity, cp (reyns)
ν	lubricant kinematic viscosity, cs (ft^2/sec)
φ_t	thermal reduction factor

Subscripts:

R	rolling
S	sliding
1	refers to pinion
2	refers to gear

Superscript:

$(\hat{\ })$	simplified

ANALYSIS

Part- and full-load gear efficiency for a wide range of operating conditions and gear geometries were analyzed by the method of [6]. This method is applicable to a jet lubricated spur gear set in which the gears do not contact oil in the sump thus eliminating churning loss. In addition to the gear mesh losses, gear windage and support bearing losses were also considered.

The gear mesh losses, which account for a major portion of the system loss, consist of a sliding component and a rolling traction or pumping component. Sliding losses arise from the friction forces developed as two gear teeth move across each other. Rolling losses are caused by the hydrodynamic pressure forces acting on the gear teeth as an elastohydrodynamic, EHD, film is formed. These pressure forces, generated by the relative motion of the gear teeth, retard the motion of the two rotating gears and thus absorb power.

In [6] the mesh losses were calculated by numerically integrating the sliding and rolling losses over the path of contact. A simple tooth loading diagram shown in Fig. 1 was assumed. The effect of tooth load sharing was included. The frictional sliding loss was based on disc machine data generated by Benedict and Kelley [8]. This friction coefficient expression is considered to be applicable to gear sliding loss calculations in the EHD lubrication regime where some asperity contact occurs for λ ratios less than 2 (λ = ratio of minimum EHD film thickness to composite surface roughness). In [6] rolling losses were based on disc machine data generated by Crook in [9]. Crook found that the rolling loss was simply a constant value multiplied by the EHD central film thickness. Gear tooth film thickness was calculated by the method of Hamrock [10] and adjusted for thermal effects using Cheng's thermal reduction factor [11]. At high pitch line velocities the isothermal equations such as Hamrock's will predict an abnormally high film thickness since inlet shear heating of the lubricant is not considered. Cheng's thermal reduction factor will account for the inlet shear heating and reduce the film thickness accordingly. Inlet starvation effects at high speeds were not considered, however.

In Fig. 2 a typical distribution of instantaneous sliding and rolling power loss across the path of contact is shown. A simpson's rule integration was used to obtain an average power loss over the path of contact. The relatively simple shape of the instantaneous loss curves suggested that a simplified expression might be utilized to approximate this integration without the use of numerical methods. In [6] such an expression was developed and the results were found to be within 1 percent of the numerical integrated value for a large number of cases. The simplified expression for rolling and sliding losses are repeated here:

$$\hat{P}_S = \frac{[P_S(l_1) + P_S(l_2)]l_3 + \dfrac{[P_S(l_4)](l_5)}{2}}{l_6} \tag{1}$$

$$\hat{P}_R = \frac{[P_R(l_1) + P_R(l_2)]l_3 + [P_R(X_P)]l_5}{l_6} \tag{2}$$

where

$$P_S = C_1 V_S f w \tag{3}$$

$$P_R = C_2 V_T h \varphi_t \mathscr{F} \tag{4}$$

$C_1 = 1\times10^{-3}$ (SI units) $C_1 = 1.515\times10^{-4}$ (English units)

$C_2 = 8.96\times10^{4}$ (SI units) $C_2 = 1.970$ (English units)

In addition to the mesh losses, an expression for gear windage loss was also developed in [6] from experimental data on turbine disc windage losses. To account for the oily atmosphere within the gearbox the density and viscosity of the gearbox atmosphere were corrected to reflect a 34.25 part air to 1 part oil combination as in [12]. Constant values for air density and viscosity at 339 K (150° F) and oil specific gravity of 0.9 were assumed. The expression for pinion and gear windage were found to be:

$$P_{W,1} = C_3 \left(1 + 2.3 \frac{\mathscr{F}}{R_1}\right) n_1^{2.8} R_1^{4.6} (0.028\ \mu + C_4)^{0.2} \tag{5}$$

$$P_{W,2} = C_3 \left(1 + 2.3 \frac{\mathscr{F}}{R_2}\right) \left(\frac{n_1}{m_g}\right)^{2.8} R_2^{4.6} (0.028\ \mu + C_4)^{0.2} \tag{6}$$

$C_3 = 2.82\times10^{-7}$ (SI units) 4.05×10^{-13} (English units)

$C_4 = 0.019$ (SI units) 2.86×10^{-9} (English units)

Support bearing loss was also included in [6]. An approximate method described by Harris in [13] was used. A straddle mounted deep groove ball bearing arrangement was assumed for comparison purposes. The deep groove ball bearing losses are a function of the bearing pitch diameter, static capacity, lubricant viscosity, shaft speed and bearing load. These equations are:

$$P_{BRG} = C_5(M_1 n_1 + M_2 n_2) \tag{7}$$

$$C_5 = 2.10\times10^{-4} \text{ (SI units) } C_5 = 3.18\times10^{-5} \text{ (English units)}$$

M is a torque loss consisting of a load-dependent and a viscous term. For a deep groove ball bearing:

$$M_L = 0.0009 \frac{F_{ST}^{1.55}}{C_s^{0.55}} D_m \tag{8}$$

$$M_V = \begin{cases} C_6 f_o (\nu n)^{2/3} D_m^3 & \text{for } (\nu n) > 2000 \\ C_7 f_o D_m^3 & \text{for } (\nu n) \leq 2000 \end{cases} \tag{9}$$

$$C_6 = 9.79\times10^{-2} \text{ (SI units) } C_6 = 2.91\times10^{-2} \text{ (English units)}$$

$$C_7 = 24.1 \text{ (SI units) } C_7 = 3.49\times10^{-3} \text{ (English units)}$$

COMPARISON WITH DATA

To test the accuracy of this power loss method, calculated power loss values were compared with the data of [7]. These data were generated on a back-to-back test stand for a gearset described in Table 1. Power loss was measured as a function of speed, torque, oil flow rate, oil jet location, gear width, and lubricant viscosity. The results of this comparison are shown in Fig. 3 where gear power loss (including bearing loss) is plotted as a function of torque.

The theory of [6] shows good agreement with the data especially for the most inefficient method of gearset lubrication (11.4 l/min (3 gpm) oil flow rate with the oil jet directed into the gear mesh). At all three test speeds shown in Fig. 3 the theory of [6] follows the trends shown by the data.

Figure 3 also presents the power loss prediction of [5]. This method [5] predicts a power loss which is greater than either that measured or predicted by the theory of [6]. The reason for this difference is primarily due to the less accurate expression chosen for predicting the friction coefficient.

DISCUSSION OF RESULTS

Power loss calculations utilizing the method of [6] were performed for a wide range of gear geometry and operating variables. These variables included diametral pitch, pitch diameter, lubricant viscosity, gear ratio, pitch line velocity, pinion torque, and pinion width/diameter ratio. The results are presented in Figs. 4 through 11.

Effect of Speed, Size, and Pitch

In Fig. 4 the gear power loss (excluding bearing loss) of gearsets which have pinion pitch diameters of 10 cm (4.0 in.) and 41 cm (16.0 in.), respectively, are shown as a function of pinion torque for three operating surface speeds. Pitch line velocity is used to compare the performance of the two gearsets since there is a great difference in their size. The torque levels shown may be far greater than the capacity of any 10 cm pinion but the power loss is calculated for comparison purposes nonetheless.

For this figure, the lubricant viscosity was held at 30 cp, the pinion width/diameter ratio was 0.5 and the gear ratio was 1.0.

For a given gearset at constant surface speed the windage loss is constant. The variation in power loss with torque shown in Fig. 4 is due to changes in the mesh rolling and sliding loss. Rolling power loss will decrease slightly with increasing torque (or load) since the film thickness theoretically decreases with load to the -0.07 power.

Sliding loss increases nearly in direct proportion to load. At very low torque levels the gear power loss is essentially the tare (no-load) loss of the gearset at that speed. Since the sliding loss is near zero at low torque levels, the loss is made up of just windage and rolling losses. The film thickness and, thus, the rolling loss is not a strong function of load. As a result, the power loss remains constant for a wide range of low torque values.

At high torque values the power loss curves form nearly a 45 degree angle with the torque axis (slope = 1). This indicates that the power loss is directly proportional to the torque being transmitted. This is due to the stronger influence of sliding loss on the total loss at higher torque levels. At intermediate torque levels there is a cross-over between decreasing rolling and windage loss, and increasing sliding loss as torque is increased. Figure 4 shows that this cross-over occurs at lower torque levels for the smaller gears.

The data of Fig. 4 can be analyzed in terms of gearset efficiency. In Fig. 5 the power loss data of Fig. 4 has been replotted and combined with data for gears of 4 to 32 diametral pitch. Data for a 20-cm (8.0-in.) diameter pinion has also been included. The efficiency of all three size gears reach 99 percent at relatively low torque levels. Efficiency is generally greater than 98 percent at a pinion torque level that is only 5 percent of the torque transmitted at peak efficiency.

Pitch diameter has a strong effect on gearset efficiency at low torque levels but less of an effect as torque is increased. Peak efficiencies for all gearsets ranged from approximately 99.3 to 99.9 percent. It should be kept in mind that these small differences in efficiency can cause substantial changes in the power loss and consequently in cooling considerations. At high torque levels the larger gear is more efficient. This is because the teeth of a large diameter gear are subjected to lower loads than the teeth on a smaller gear at the same torque level. This reduces the coefficient of friction and the sliding loss.

Also shown in Fig. 5 is the effect of diametral pitch. At low torques the coarse-pitched gears are more efficient. At higher torques the trend is reversed and the fine-pitched gears become more efficient. The reason is that at high torques, the sliding loss becomes a dominant factor in the gear loss. Since fine-pitch gearing reduces the length of the path of contact and the magnitude of the sliding velocity, the sliding power loss is also reduced. This results in a more efficient gearset under these high torque conditions.

To normalize the data of Fig. 5 in terms of gear capacity, the efficiency data for the 10- and 41-cm gears have been replotted in Fig. 6 versus a gear capacity factor K as described in [1]:

$$K = \frac{C_8 w(m_g + 1)}{\mathcal{F} D_1 m_g} \tag{10}$$

where

$$C_8 = 1.45\times10^{-4} \text{ (SI units) } C_8 = 1.0 \text{ (English units)}$$

The K-factor provides an estimate of the required gear face width and diameter for a given torque level that the gearset must carry.

The K-factors for helical and spur gears tabularized in [1] generally range from a value of about 100 for low hardness-generated-steel-gears to about 1000 for aircraft quality, case hardened and ground, high-speed gearing. A nominal K-rating for a general-purpose industrial drive, with 300 BHN steel gears, carrying a uniform load at a pitch line velocity of 15 m/sec (3000 fpm) or less would typically range from 275 to 375. It is apparent from Fig. 6 that gear sets with pinions ranging in size from 10 to 41 cm and diametral pitch from 4 to 32 generally reach their peak efficiency at K-factors from approximatesly 150 to 400. Above these ratings the efficiency remains relatively constant or in the case of coarse-pitched gears, falls off slightly.

In comparing Figs. 5 and 6, it is apparent that the effect of gear size on gear efficiency is less at an equivalent K-factor than at an equivalent torque level. However, it is still true that large gears have a slight efficiency advantage, particularly for coarse-pitched gears, at an equivalent percent of rated capacity.

Effect of Pitch Line Velocity and Pitch

In Fig. 7 the effect of pitch line velocity on efficiency is shown for a 20-cm (8.0-in.) pitch diameter pinion mesh with a gear ratio of 1.0, a pinion width/diameter ratio of 0.5 for diametral pitch values of 4 to 32, and a lubricant having a viscosity of 30 cp. Pitch line velocities from 1.27 m/sec (250 ft/min) to 40.6 m/sec (8000 ft/min) do not alter the effect that diametral pitch has on efficiency shown in Fig. 5. The coarse-pitched gears have better part-load efficiency but lower peak efficiencies.

Pitch line velocity has a stronger effect on part-load gear efficiency than it does on peak efficiency due to its strong influence on rolling traction and windage losses. Figure 7 shows that an increase in speed generally benefits peak efficiency levels, particularly for coarse-pitched gears. This efficiency improvement with speed is caused by a trade-off of rolling and sliding losses. At high speeds the rolling loss increases due to the increase in film thickness. This increase in rolling loss, however, is more than offset by a reduction in the coefficient of friction and consequently sliding loss. This results in an overall higher efficiency for the higher speed gearset.

Effect of Lubricant Viscosity and Size

Figure 8 shows the effect of lubricant viscosity on efficiency of gearsets with 10 and 41 cm pitch diameter pinions. The gear ratio is held constant at 1.0 and the pinion width/diameter ratio is 0.5. This comparison is made at a pitch line velocity of 20.3 m/sec (4000 ft/min) for an oil whose viscosity ranges from 2 to 500 cp. This variation in viscosity represents a temperature range of 289 K (60^{o} F) to 489 K (420^{o} F) for an SAE 30 oil. For a MIL-L-23699 oil, this viscosity range represents a temperature range of 267 K (20^{o} F) to 436 K (325^{o} F).

The viscosity variation has a great effect on efficiency for either gearset at low torque conditions. Efficiency is less sensitive to oil viscosity at higher torque levels. The 10-cm diameter pinion has a cross-over torque level at which the gearset efficiency becomes greater with the more viscous oil. This is due to the nature of the friction coefficient.

The Benedict and Kelley coefficient of friction model [8] predicts that for steel rollers or gears under heavy loads, a more viscous oil will cause a greater separation of the two surfaces and less asperity contact, resulting in less friction. A less viscous oil may allow greater asperity contact which increases the coefficient of friction and, thus, the power loss. This cross-over effect is not shown for the larger gear in Fig. 8 because the peak efficiency of the 41-cm (16-in.) gear has not yet been reached at the maximum torque of 11 300 N-m (100 000 in-lbf).

Effect of Face Width and Size

The effect of pinion face width/diameter ratio (F/D) is shown in Fig. 9 for the 10- and 41-cm diameter pinion gearset at a pitch line velocity of 20.3 m/sec (4000 ft/min), diametral pitch = 8, ratio = 1.0, and lubricant viscosity of 30 cp. As with pitch diameter and pitch line velocity, F/D ratio has a great effect on efficiency at low-torque levels but a lesser effect at high-torque values. The 0.5 and 1.0 F/D curves converge to within 0.4 percentage points at high torque levels for both the 10- and 41-cm diameter pinions. Rolling loss is directly proportional to gear width while sliding loss will decrease for wider gears since the unit loading is decreased. At low torques the increased rolling loss of the wider gear produces a lower gearset efficiency. At high torques the sliding loss becomes more significant and offsets the increased rolling loss produced by the wider gear.

The data of Fig. 9 is replotted against the K-factor in Fig. 10. When plotted in this manner the F/D ratio does not affect gearset efficiency. The band of values shown in Fig. 9 reduces to a single line for both the 10- and 41-cm diameter pinion gearsets. This is due to the normalizing effect the K-factor has on gearset efficiency.

Effect of Ratio

In Fig. 11 the effect of ratio is shown for the 10- and 41-cm diameter pinions at a pitch line velocity of 20.3 m/sec (4000 ft/min), diametral pitch = 8, F/D = 0.5, and lubricant viscosity of 30 cp. Since the pinion diameter is held constant, an increase in reduction ratio means an increase in gear diameter. Also, since the pinion F/D ratio is held constant at 0.5, the width of the gearset is held constant. The effect of changing ratio from 1 to 8 has an effect similar to that of changing the F/D ratio from 0.5 to 1.0 as shown in Fig. 9. As might be anticipated, the gearset with the larger reduction ratio (or larger gear diameter) is less efficient at low torque levels. However, the 10-cm pinion at a ratio of 8 becomes more efficient that the same pinion at a ratio of 1 at torques above 339 N-m (3000 in-lbf). This effect is caused by the higher rolling velocities generated by the larger gear diameter which tend to decrease the friction coefficient and, thus, the sliding power loss. A similar effect would be anticpated for the 41-cm pinion at higher torque levels.

Effect of Support Bearing Loss

The efficiency data presented in Figs. 4 through 11 represent only the power loss due to gear sliding, rolling, and windage. Gear system efficiency must include the support bearing losses as well. For comparison purposes a deep-groove ball bearing system was selected for the 10- and 41-cm pinion diameter gearsets. The bearings selected are described in Table 2. The effect of adding the rolling-element bearing loss to the gear loss is shown in Fig. 12 for a pitch line velocity of 20.3 m/sec (4000 ft/min); ratio of 1.0, F/D of 0.5, and a lubricant viscosity of 30 cp. A significant loss in efficiency results when the bearing loss is included.

Breakdown of Gear System Losses

To gain further insight into the effects of the various component losses on the system efficiency, a breakdown of the losses relative to the full-load loss (loss at peak efficiency) for each speed is shown in Fig. 13 for the 10-cm diameter pinion gearset and in Fig. 14 for the 41-cm diameter pinion set. In each case the diametral pitch was 8, the gear ratio was 1.0, the F/D ratio was 0.5, and the lubricant viscosity was 30 cp. It should be noted that the pinion torque scales are different in Figs. 13 and 14. The maximum torque levels correspond approximately to the torque levels at peak efficiency of each system.

The trends shown in Figs. 13 and 14 are similar. At low pitch line velocity (1.27 m/sec (250 ft/min)), the sliding loss dominates at all conditions except for very low loads as this loss approaches zero. At higher speeds, the rolling and bearing losses become significant. At the maximum speed and torque condition, the sliding loss drops to approximately 37 percent of the total losses for the 10-cm diameter pinion and to 25 percent for the 41-cm diameter pinion set. The bearing losses account for nearly 50 percent of the system loss for either gearset at this operating condition.

Windage reaches a maximum of 8 percent of the system loss at maximum speed for the 10-cm pinion and 18 percent for the 41-cm pinion. Thus, windage also becomes important for the larger gearset.

Also shown in Figs. 13 and 14 are the no-load or tare loss at each pitch line velocity. At 1.27 m/sec (250 ft/min) the tare loss is less than 10 percent of the full-load loss. At 40.6 m/sec (8000 ft/min) the tare loss increases to approximately 60 percent of the full-load loss. This is due to the fact that most of the losses at high pitch line velocity are speed-dependent and remain as a tare loss when the load is removed from the gearset.

SUMMARY

Spur gear efficiency was calculated using the method reported in [6] for a wide range of gear geometries and operating conditions. This method algebraically accounts for gear sliding, rolling, and windage loss components and also incorporates an approximate ball bearing power loss expression to estimate the loss of a ball bearing support system. A theoretical breakdown of the total spur gear system loss into individual components was performed to show their respective contributions to the total system loss. The range of gear geometry and operating variables included the following:

- Pinion pitch diameters from 10 cm (4.0 in.) to 41 cm (16.0 in.)
- Diametral pitch values from 4 to 32
- Pinion width/diameter ratios from 0.5 to 1.0
- Gear reduction ratios from 1 to 8
- Pinion torques from 0.113 N-m (1.0 in-lbf) to 11 300 N-m (100 000 in-lbf)
- Pitch line velocities from 1.27 m/sec (250 ft/min) to 40.6 m/sec (8000 ft/min)
- Lubricant viscosities from 2 to 500 cp

The torque and speed limits associated with the above gear geometry variables were not defined in this study. However, some of the results were presented in terms of gear capacity by introcuing a K-factor as an independent variable. The following results were obtained:

1. Gear efficiency was mildly dependent on the amount of torque being transmitted above some minimum torque value. At torques greater than 5 percent of the torque transmitted at maximum efficiency, gear efficiency (bearing losses excluded) ranged from 98.6 to 99.9 percent.

2. Under high loads, fine-pitched gears were generally more efficient than coarse-pitched gears. This advantage diminished, however, with increases in gear size or pitch line velocity.

3. Peak gear efficiency generally improved with an increase in pitch line velocity while part-load efficiency diminished due to increased tare power losses. Caorser-pitched gears enhanced this effect.

4. Large gears generally had higher peak efficiencies than small gears. This efficiency advantage was more marked for coarse-pitched gears.

5. Gear ratio and pinion width/diameter ratio had relatively minor effects on gear efficiency with higher ratio, wider gears showing slightly higher peak efficiencies but lower part-load efficiencies. An increase in lubricant viscosity showed a similar but slightly stronger effect. Pinion width/diameter ratio had no effect on efficiency when the K-factor was held constant.

6. Support ball bearing losses can be a significant part of spur gear system power loss. At pitch line velocities greater than 20 m/sec bearing losses accounted for more than 35 percent of the full-load system loss.

7. Tare (no-load) losses of a gearset are significant except at low speeds. At pitch line velocities greater than 20 m/sec the power loss of an unloaded gearset was more than 35 percent of the full-load loss.

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TABLE 1. - GEAR GEOMETRY AND OPERATING PARAMETERS

FOR TEST GEARS OF [7]

	Pinion	Gear
Pitch diameter, cm (in.)	15.2 (6)	25.4 (10)
Number of teeth	48	80
Diametral pitch	8	
Pressure angle (deg)	20	
Width, cm (in.)	4.0 (1.56)	---------
Lubricant	Mineral oil with anti-oxidant additive	
Viscosity at oil jet temperature = 333 K (140° F)	60 cs	

TABLE 2. - BALL BEARING DATA USED IN FIGS. 13 THROUGH 14

	Pinion pitch diameter	
	10 cm (4.0 in.)	41 cm (16.0 in.)
Bore	40 mm (1.58 in.)	150 mm (5.91 in.)
Outside diameter	90 mm (3.54 in.)	320 mm (12.59 in.)
Width	23 mm (0.91 in.)	65 mm (2.56 in.)
Pitch diameter	65 mm (2.56 in.)	235 mm (9.25 in.)
Static capacity	22 330 N (5020 lbf)	240 100 N (56 000 lbf)

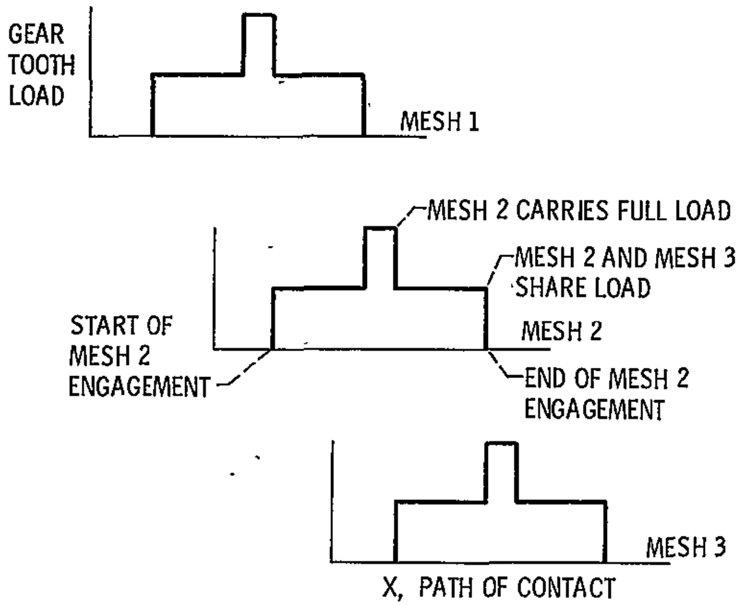

Figure 1. - Tooth load sharing diagram.

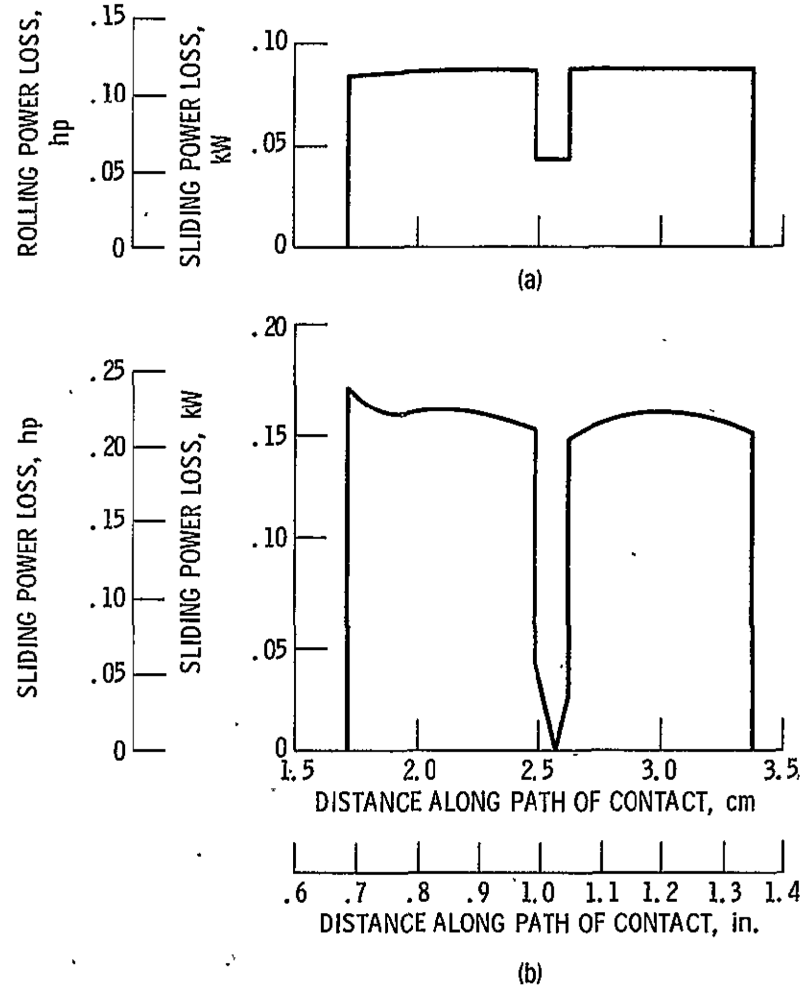

Figure 2. - Typical distribution of rolling and sliding power loss along the path of contact.

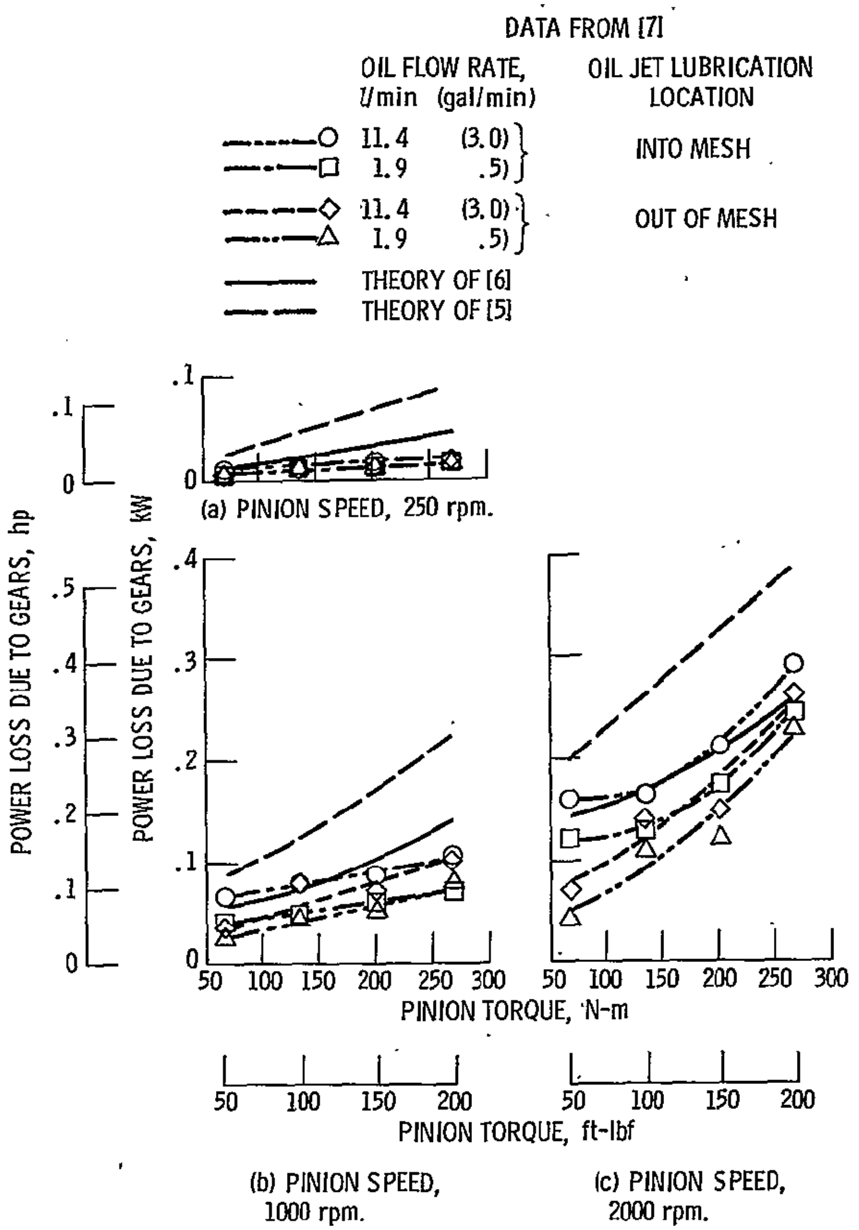

Figure 3. - Comparison of predicted gear power loss with data of [7]. (See Table I for gear geometry and operating data.)

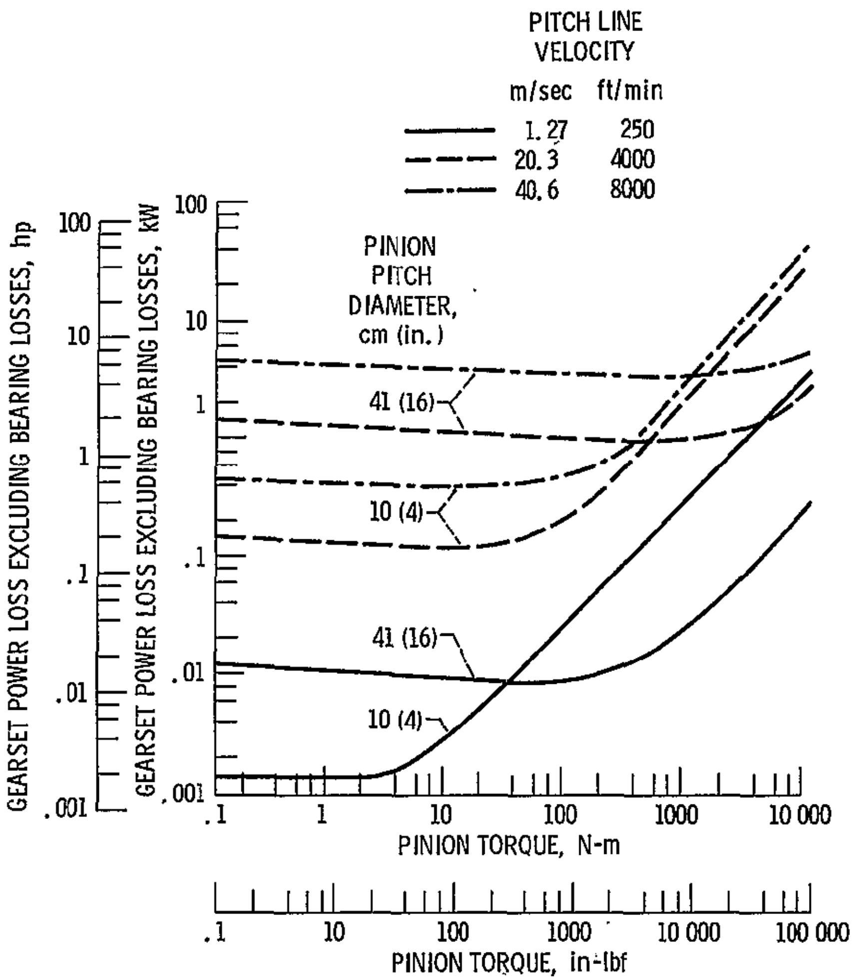

Figure 4. - Gearset power loss as a function of pinion torque for three speeds. Diametral pitch, 8; ratio, 1.0; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

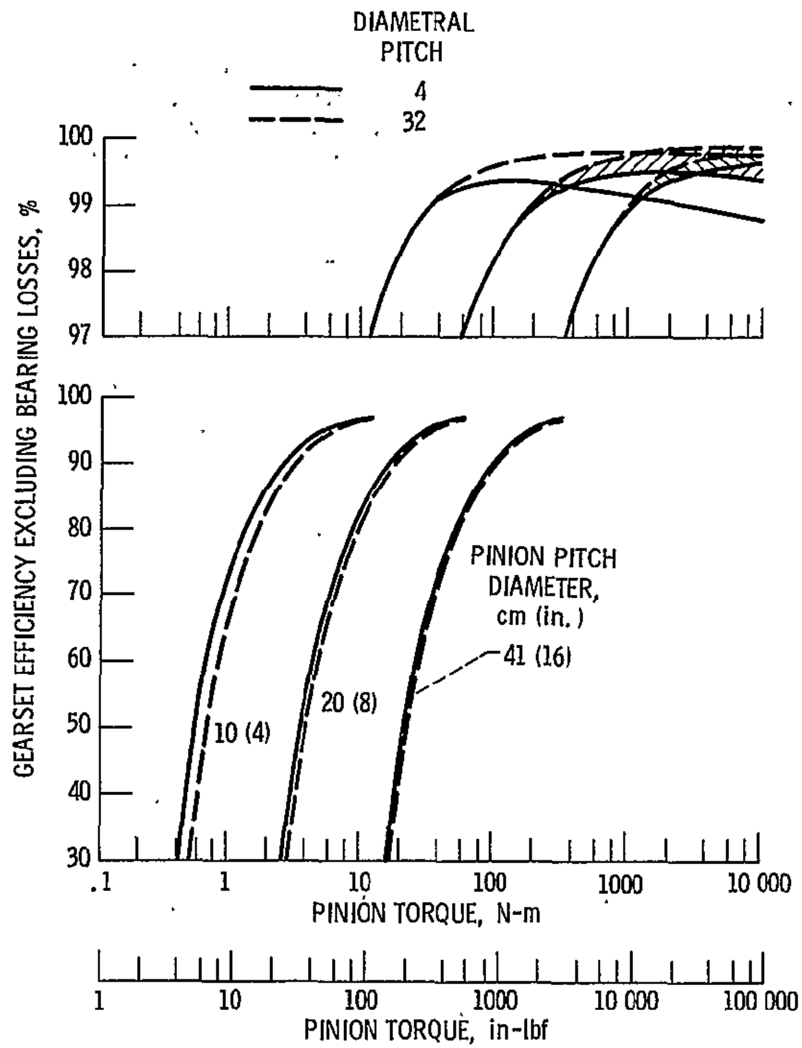

Figure 5. - Effect of diametral pitch and pitch diameter on gearset efficiency. Pitch line velocity, 20.3 m/sec (4000 ft/min); ratio, 1.0; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

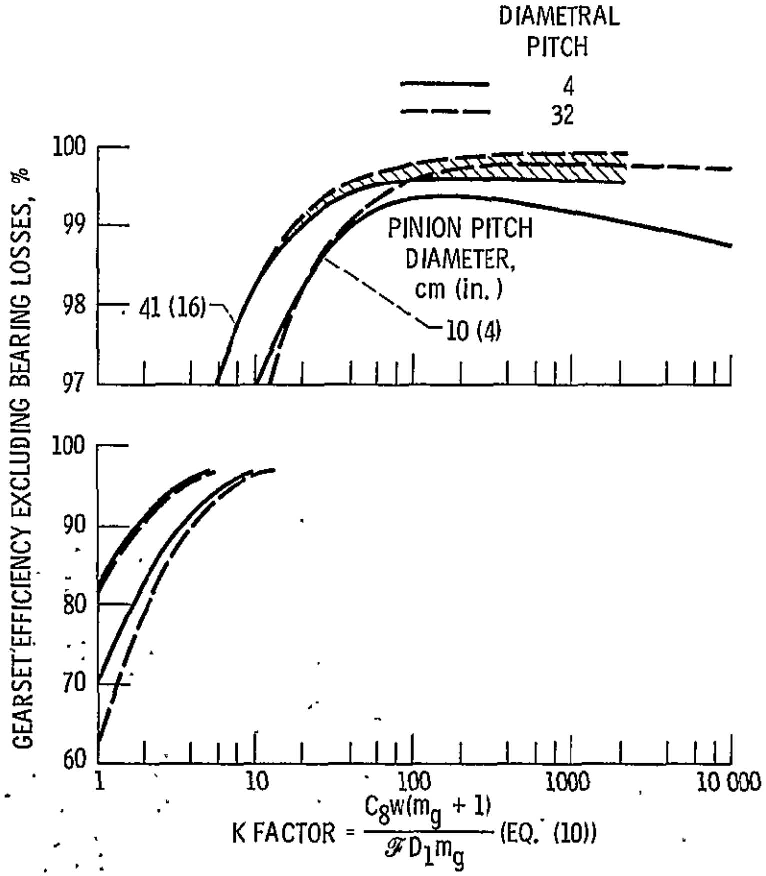

Figure 6. - Effect of diametral pitch and pitch diameter on gearset efficiency as a function of K factor. Pitch line velocity, 20.3 m/sec (4000 ft/min); ratio, 1.0; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

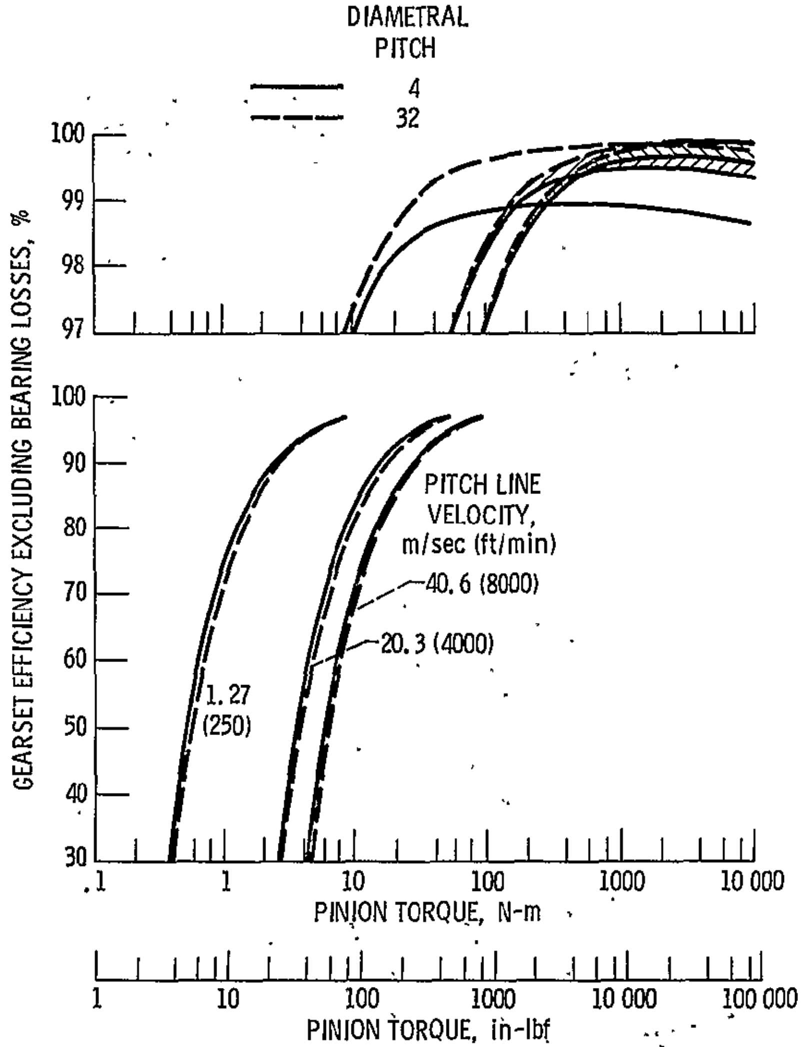

Figure 7. - Effect of diametral pitch and pitch line velocity on gearset efficiency. Pinion-pitch diameter, 20 cm (8.0 in.); ratio, 1.0; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

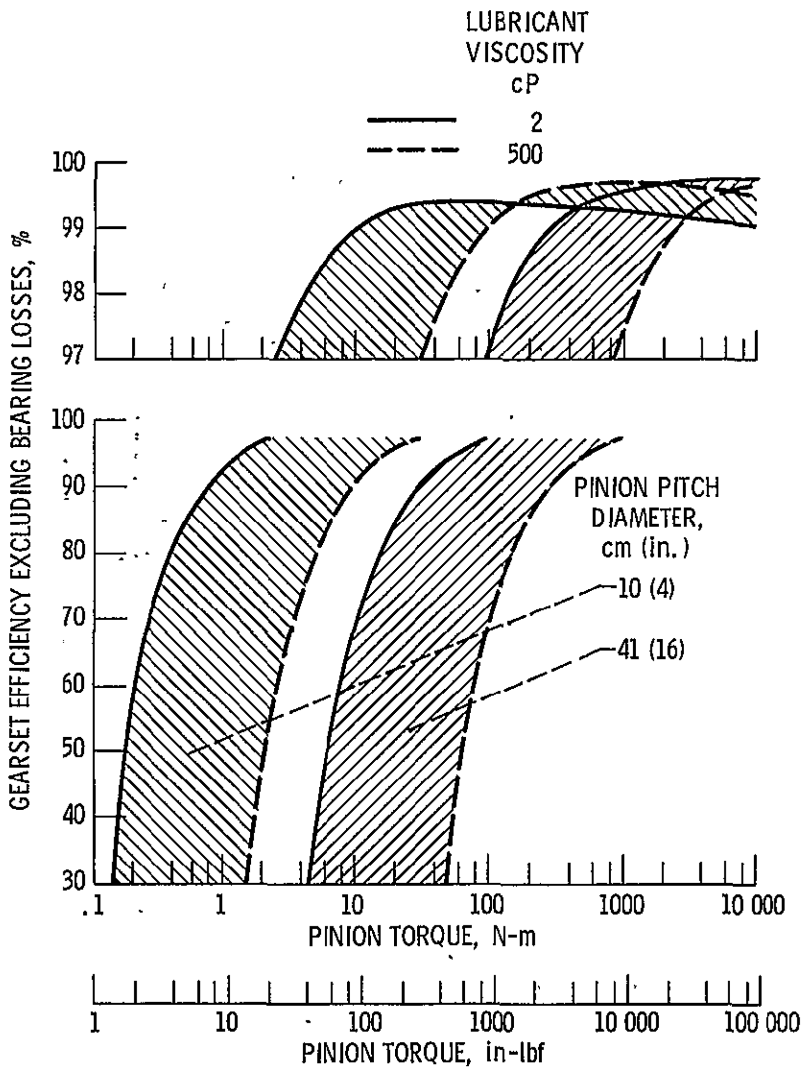

Figure 8. - Effect of lubricant viscosity on gearset efficiency. Pitch line velocity, 20.3 m/sec (4000 ft/min); diametral pitch, 8; ratio, 1.0; pinion width/diameter ratio, 0.5.

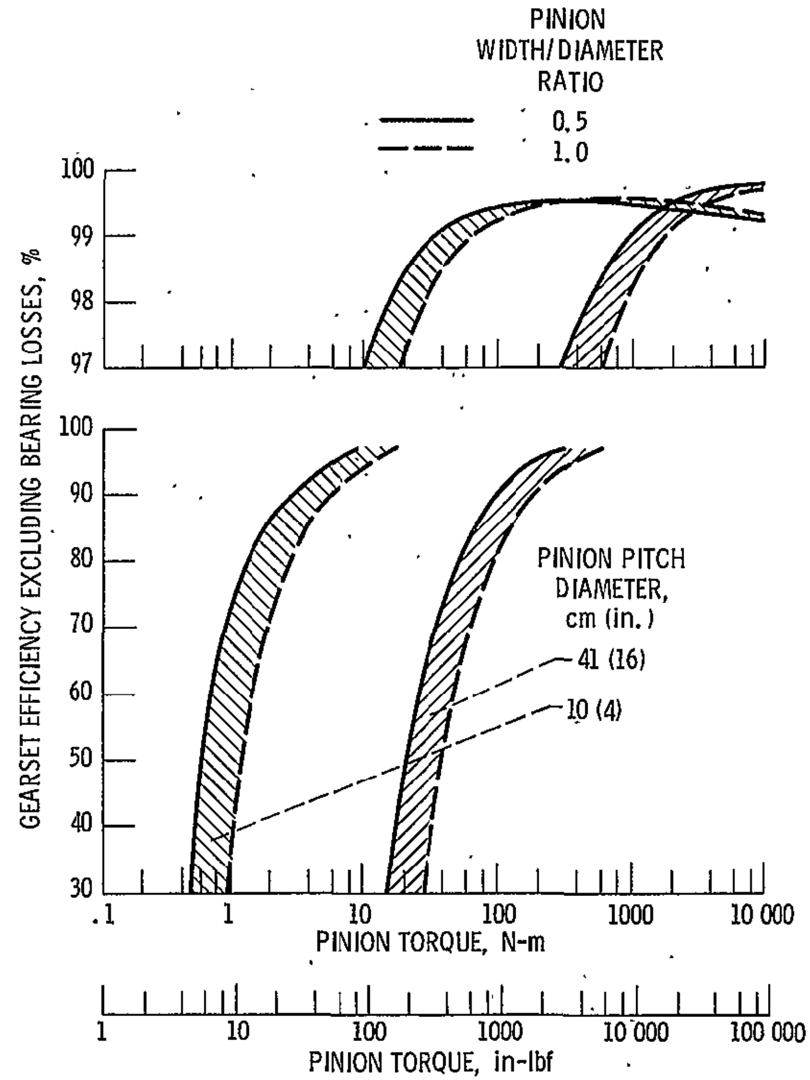

Figure 9. - Effect of pinion width/diameter ratio on gearset efficiency. Pitch line velocity, 20.3 m/sec (4000 ft/min); diametral pitch, 8; ratio, 1.0; lubricant viscosity, 30 cP.

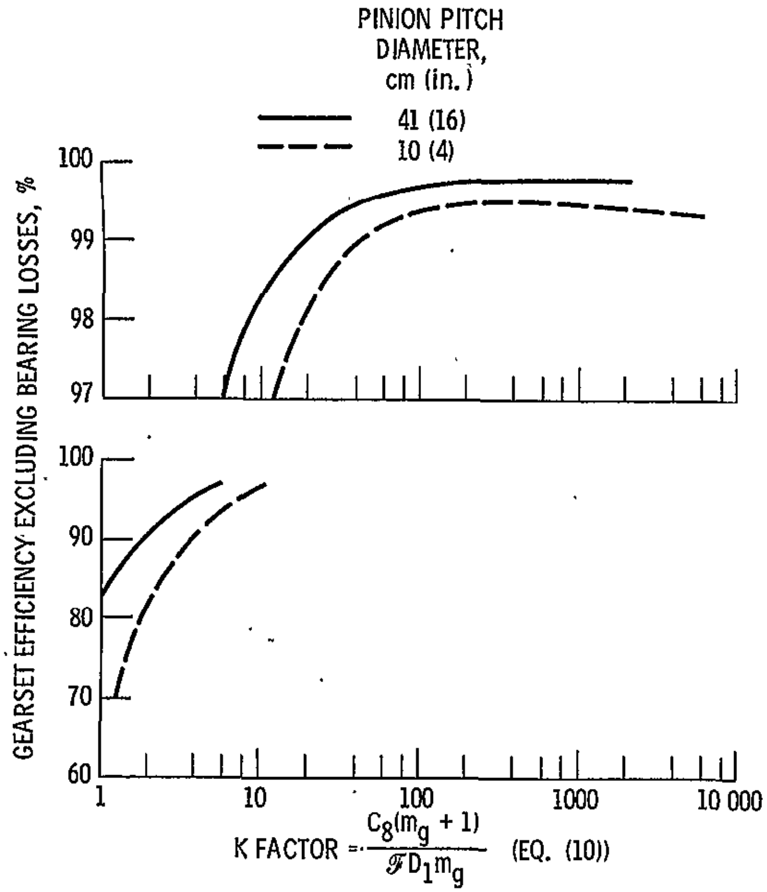

Figure 10. - Effect of pinion width/diameter ratio on gearset efficiency as a function of K factor. Pitch line velocity, 20.3 m/sec (4000 ft/min); diametral pitch; 8; ratio, 1.0; lubricant viscosity, 30 cP.

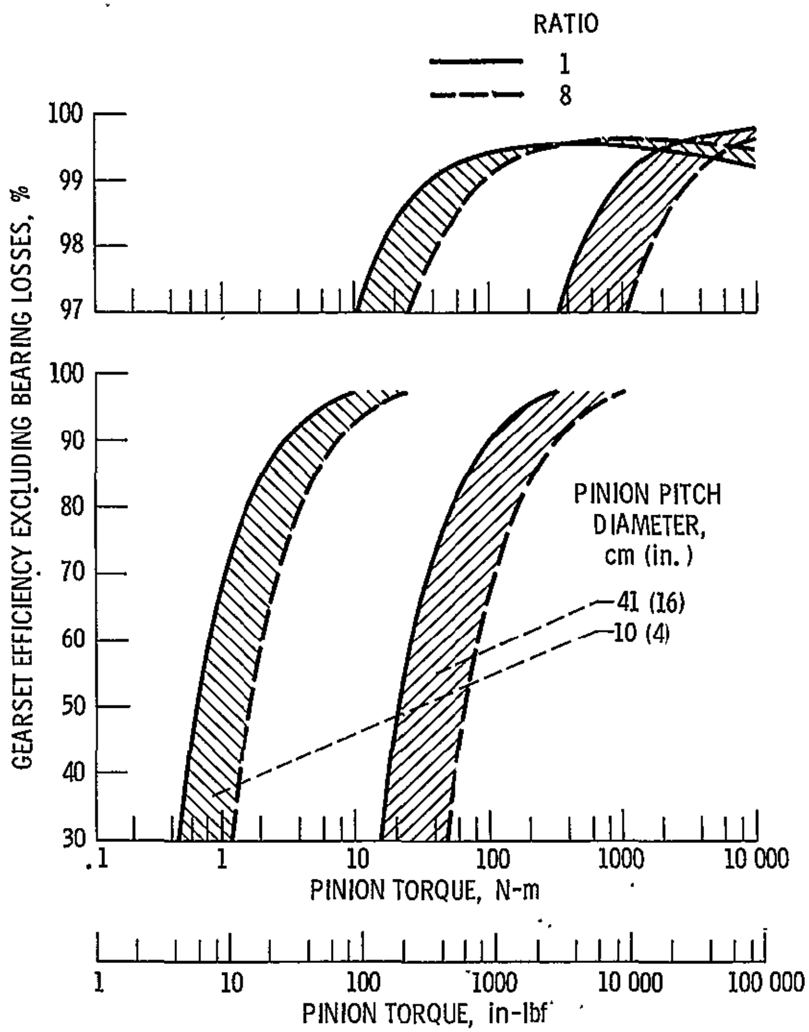

Figure 11. - Effect of ratio on gearset efficiency. Pitch line velocity, 20.3 m/sec (4000 ft/min); diametral pitch, 8; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

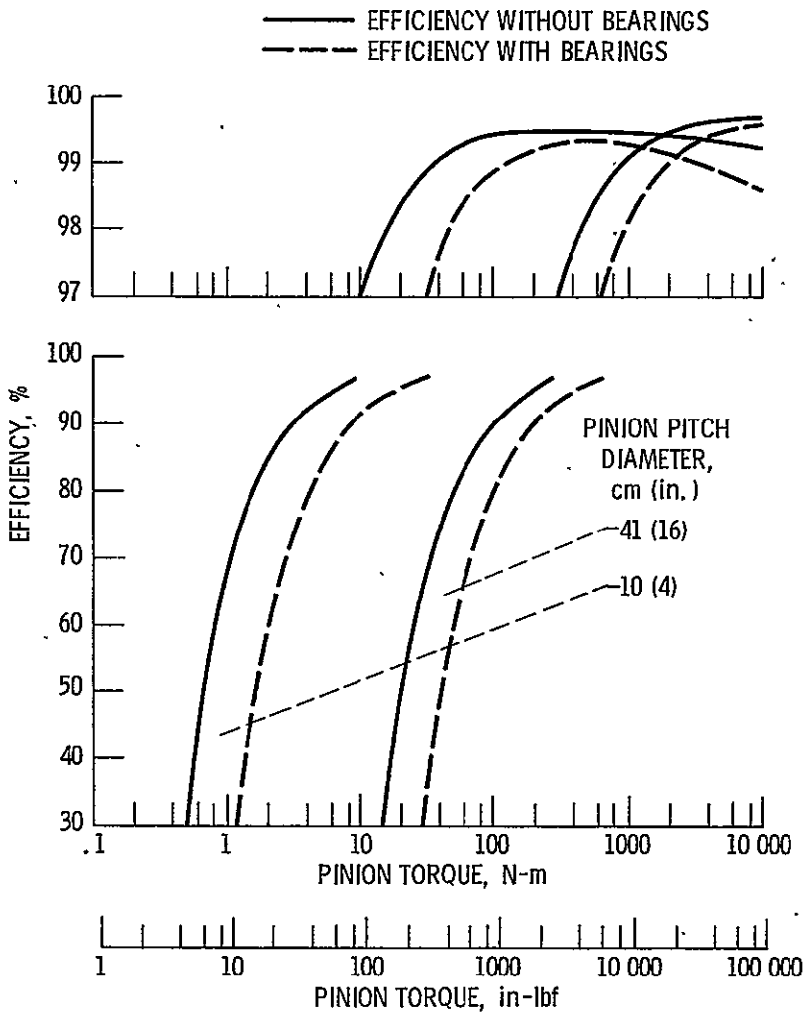

Figure 12. - Effect of support bearings on gear efficiency. Pitch line velocity, 20.3 m/sec (4000 ft/min); diametral pitch, 8; ratio, 1.0; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

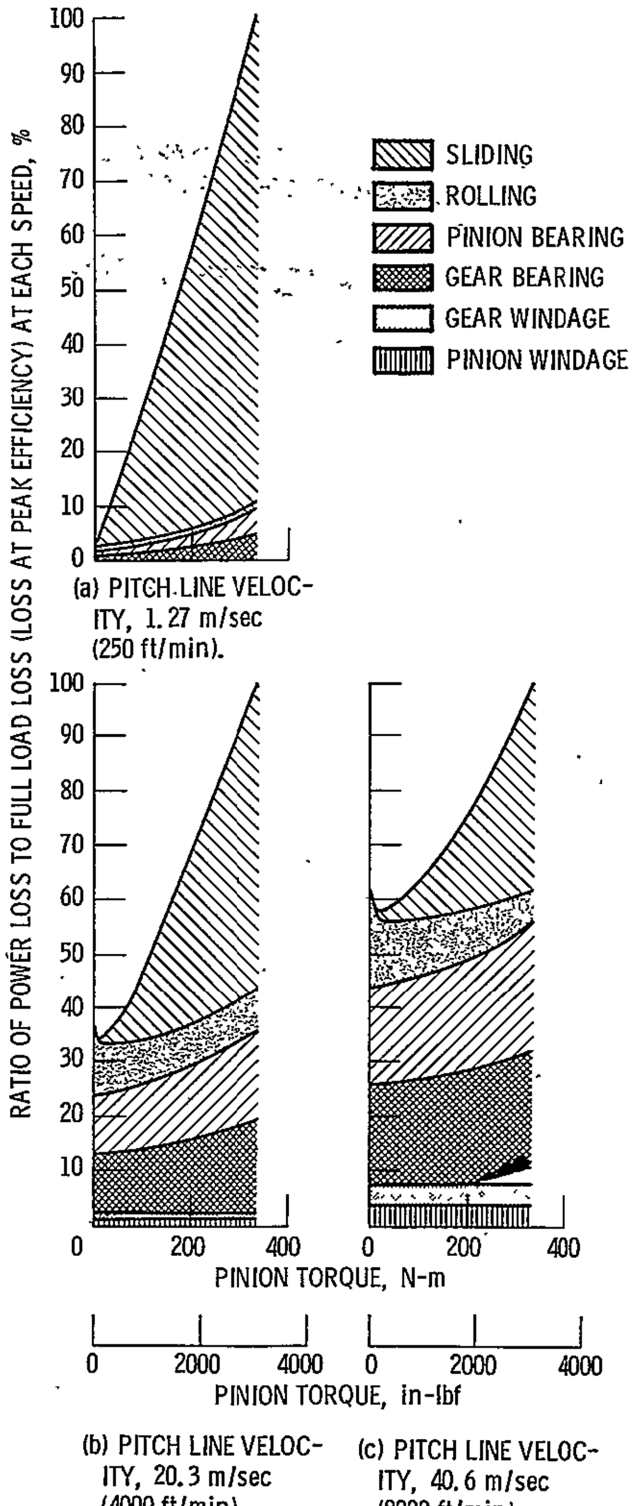

Figure 13. - Percentage breakdown of the sources of gearbox power loss as a function of input torque for three speeds. Pitch diameter, 10 cm (4.0 in.); diametral pitch, 8; ratio, 1.0; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

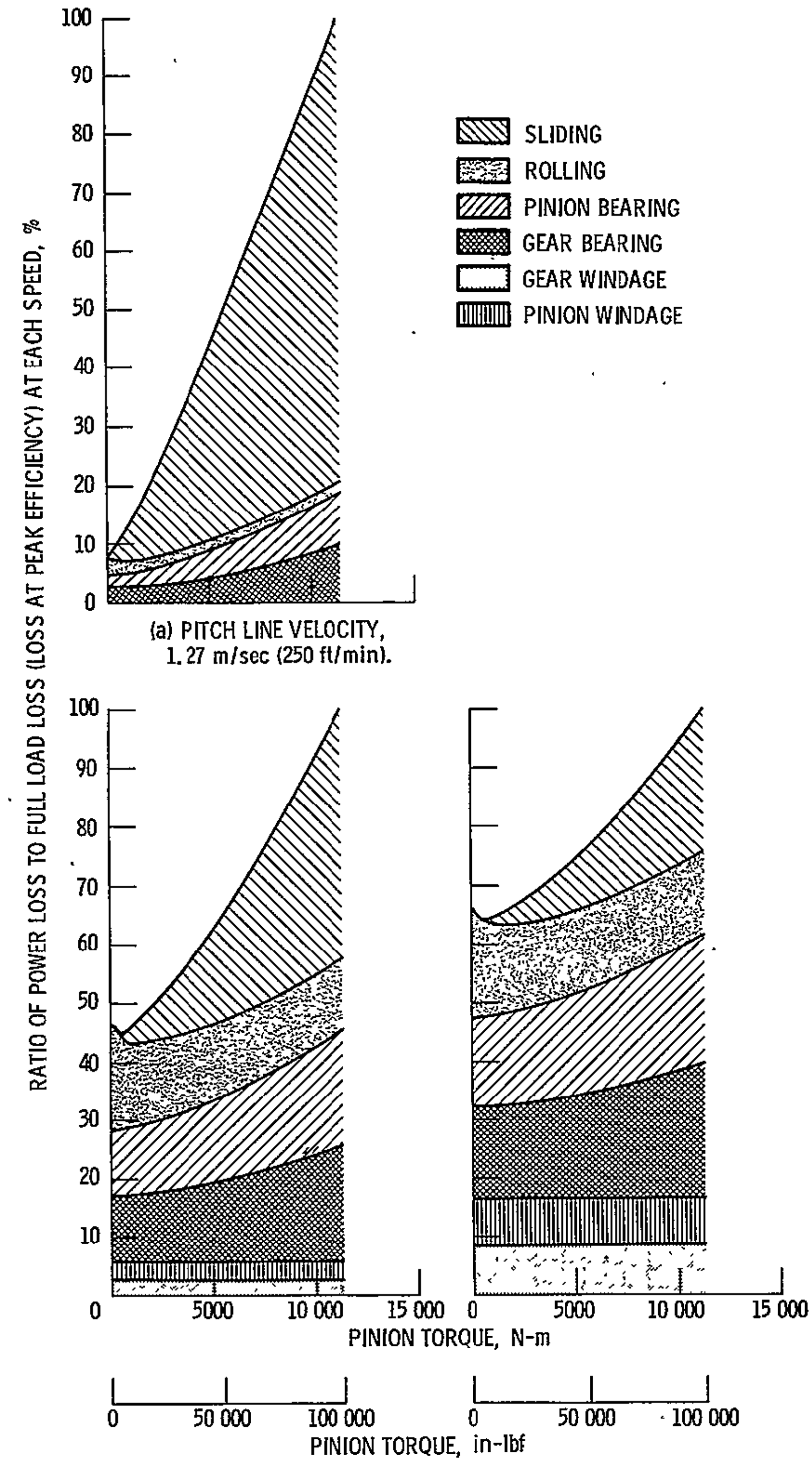

Figure 14. - Percentage breakdown of the sources of gearbox power loss as a function of input torque for three speeds. Pitch diameter, 41 cm (16.0 in.); diametral pitch, 8; ratio, 1.0; pinion width/diameter ratio, 0.5; lubricant viscosity, 30 cP.

1. Report No. NASA TM 81426 AVRADCOM TR-80-C-2	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle EFFECT OF GEOMETRY AND OPERATING CONDITIONS ON SPUR GEAR SYSTEM POWER LOSS		5. Report Date
		6. Performing Organization Code
7. Author(s) Neil E. Anderson and Stuart H. Loewenthal		8. Performing Organization Report No. E-350
		10. Work Unit No.
9. Performing Organization Name and Address NASA Lewis Research Center and AVRADCOM Research and Technology Laboratories Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Memorandum
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546 and U.S. Army Aviation Research and Development Command, St. Louis, Missouri 63166		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

The results of an analysis of the effects of spur gear size, pitch, width and ratio on total mesh power loss for a wide range of speed, torques and oil viscosities are presented. The analysis uses simple algebraic expressions to determine gear sliding, rolling and windage losses and also incorporates an approximate ball bearing power loss expression. The analysis shows good agreement with published data. Large diameter and fine-pitched gears had higher peak efficiencies but lower part-load efficiencies. Gear efficiencies were generally greater than 98 percent except at very low torque levels. Tare (no-load) losses are generally a significant percentage of the full load loss except at low speeds.

17. Key Words (Suggested by Author(s)) Spur-gear efficiency; Efficiency; Power loss; Spur gear	18. Distribution Statement Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages	22. Price*

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Space Administration

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