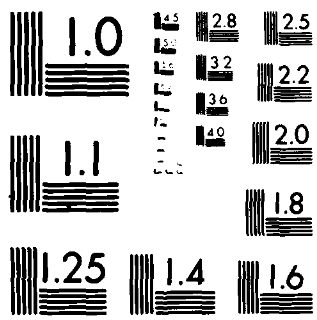
AD-A108 265 ARMY ARMAMENT RESEARCH AND DEVELOPMENT COMMAND DOVER--ETC F/G 16/4
MESH EFFICIENCY OF THE INTEGRATED SAFE/ARM DEVICE OF THE PERSHI--ETC(U)
NOV 81 F R TEPPER
UNCLASSIFIED ARLCD-TR-81020 SBI-AD-E400 711 NL

END
1-82


1.0
1.1
1.25
1.4
1.6
1.8
2.0
2.2
2.5
2.8
3.2
3.6
4.0

(4) LEVEL III

AD

AD-E400 711

AD A108265

TECHNICAL REPORT ARLCD-TR-81020

MESH EFFICIENCY OF THE INTEGRATED SAFE/ARM DEVICE OF THE PERSHING II MISSILE SYSTEM

FREDERICK R. TEPPER

NOVEMBER 1981

US ARMY ARMAMENT RESEARCH AND DEVELOPMENT COMMAND
LARGE CALIBER
WEAPON SYSTEMS LABORATORY
DOVER, NEW JERSEY

81 12 08 294

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER Technical Report ARLCD-TR-81020	2. GOVT ACCESSION NO. AD-A108 265	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) MESH EFFICIENCY OF THE INTEGRATED SAFE/ARM DEVICE OF THE PERSHING II MISSILE SYSTEM		5. TYPE OF REPORT & PERIOD COVERED 6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Frederick R. Tepper		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS ARRADCOM, LCWSL Nuclear and Fuze Division (DRDAR-LCN-C) Dover, NJ 07801		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS ARRADCOM, TSD STINFO Division (DRDAR-TSS) Dover, NJ 07801		12. REPORT DATE November 1981 13. NUMBER OF PAGES 96
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Pershing II
Integrated safe/arm device
Clock gears
Involute gears
Propulsion gear assembly
Exoatmospheric drive assembly
Valve drive assembly
Mesh efficiency

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report describes the results of an analytical study of the minimum point efficiencies of the gear meshes of the integrated safe/arm device (ISAD) of the Pershing II missile system. It expands earlier work on gear mesh efficiency by considering the possibility of involute meshes which have a non-unity contact ratio. Conclusions concerning the mesh efficiency as a function of the coefficient of friction, the presence of undercutting of the involute pinions, and the contact ratio of the involute meshes are presented. The analyses on which these results are based are given in detail in the report.

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

CONTENTS

TABLES

FIGURES

INTRODUCTION

An investigation was conducted of the minimum point efficiencies of the gear and pinion meshes of the Pershing II Integrated Safe/Arm Device (ISAD) (fig. 1) to identify potential problem areas. This work was an application as well as an extension of the analytical tools developed during an earlier study of fuze-related gear trains conducted in cooperation with personnel of City College of New York (ref 1).

In the earlier study, torque transfer efficiency expressions were derived for single pass involute and clock gear-type meshes which operate in non-spin environments. In this present study, the computer program CLOCK2--given in reference 1 for calculating the efficiency of clock-type gears--was utilized to determine the minimum point efficiencies of the ISAD timer gear meshes. Where necessary for computational purposes, the actual clock gear tooth was modified to accommodate a rounded tip (app A).

In reference 1, the involute meshes were limited to those having a unity contact ratio. Since the involute gears in the ISAD have contact ratios which are substantially greater than one, the efficiency expressions derived in reference 1 were modified to consider two pairs of involute teeth in contact simultaneously in a given mesh (app B). Appropriate logic was used to ascertain how many pairs of teeth are in contact for a specific position of the mesh. Further, a technique based on work described in reference 2 was used to determine where the involute action is initiated. The resulting computer simulation was then applied to calculate the minimum point efficiencies of the ISAD power gear meshes.

For clarity, the minimum point efficiencies, obtained as a function of the coefficient of friction, are presented in graphical form for both the clock and involute meshes. Also, since the output of the computer programs for the involute meshes gives other relevant information such as the contact ratio, the print-out (not including the point-by-point efficiencies) for each of the involute meshes is contained herein. The associated computer programs for an involute mesh having a pinion for the driver and for a mesh having the gear as a driver are listed in appendixes C and D, respectively.

CLOCK GEAR-TYPE MESHES

Determination of Torque Transfer Efficiency

A typical clock gear-type mesh is shown in figure 2. Computer program CLOCK2 (ref 1) is used to determine the point efficiency of such a mesh. The parameters required to calculate the point efficiency include the number of teeth on the gear and pinion, the pitch radii of the gear and pinion, the distances from the pivots of the gear and pinion to the respective centers of curvature of the circular arc portions of their teeth (a_G and a_P), the radii of curvature of the circular arc portions of the gear and pinion profiles (ρ_G and ρ_P), the tooth

thicknesses at the pitch circle of the gear and pinion, the pivot radii of the gear and pinion, the distance between the pivots of the gear and pinion (b), and the coefficient of friction between the gear and pinion teeth and at the gear and pinion pivots.

ISAD Clock Gear Mesh Parameters

The above quantities for the four different clock tooth-type meshes of the ISAD are listed in tables 1 through 4. The analysis on which CLOCK2 is based considers a constant curvature tooth as shown by the solid profile line in figure 2. The rounded tooth tip of the actual clock gears, as shown by the dotted line in figure 2, led to numerical difficulties when the given outside radius was used to obtain the distance a_G from the gear pivot to the tooth center of curvature. This problem was avoided by modification of the tooth to have a constant curvature. The necessary derivations to achieve this are described in appendix A.

Efficiency Results

Computer program CLOCK2 was applied to each of the clock meshes. The coefficient of friction was permitted to vary from 0 to 0.3 in steps of 0.05. The point efficiencies for each mesh for a given coefficient of friction were calculated and the minimum value determined and retained. These minimum values for each mesh were then printed out in graphical form as shown in figures 3 through 6.

INVOLUTE GEAR-TYPE MESHES

Determination of Torque Transfer Efficiency

The initial work on efficiency of involute gear-type meshes (ref 1) was confined to unity contact ratio meshes; therefore, that work could not be used directly in this study because the meshes of the ISAD have contact ratios which significantly exceed unity. Changes were required in the kinematics. Also, since some of the pinions were undercut, a determination was needed of the radius at which involute action begins. Further, because the center distance for each mesh was greater than the sum of the standard gear and pinion pitch radii, it was necessary to compute the actual mesh pressure angle. In addition, the derivation of efficiency expressions for dual point contact was needed. Finally, the existing computer program for calculating involute mesh efficiency had to be revised to include these considerations.

Kinematics

When only one pair of teeth is in contact at a given time, the computations proceed according to the method established in reference 1. A technique to be used when two pairs of teeth are in simultaneous contact is given below. It identifies the beginning and end of dual contact, provides other information relevant to the determination of gear mesh efficiency for dual contact, and, in addition, furnishes the contact ratio for the mesh.

To begin, certain nomenclature must be defined. In figure 7, which corresponds to the case where the pinion drives the gear, points T and T' are the points of tangency to the base circles of radius R_b and r_b and the distance d = TT'. Initial contact is made either at point Z where the line of action intersects the gear addendum circle of radius R_o, or at a point Z' where the involute curve begins. The radius corresponding to the start of the involute profile, called the inner form radius, is given by r_f*.

Contact begins at the point which is farthest from point T', or alternately, at the point which is nearest to the pitch point P. Final contact occurs at point W, where the line of action intersects the pinion addendum circle of radius r_o. The positions of the instantaneous contact points C_1 and C_2 with respect to point T'; that is, lengths a_1 and a_2, are expressed with the help of the instantaneous angles α_1 and α_2, which have their origin at the line O_1T'. The actual pressure angle is given by θ'. A method for calculating θ' is described later in this section.

The case where the gear is the driver is shown in figure 8. The nomenclature is parallel to that of figure 7.

Procedure for Determining Kinematics of Dual Contact When the Pinion Drives

(1) Find whether PZ < PZ'. The smaller quantity governs.

$$PZ' = PT' - Z'T' \tag{1}$$

Since

$$PT' = r_b \tan\theta' \tag{2}$$

and

$$Z'T' = \sqrt{r_f^2 - r_b^2}\ , \tag{3}$$

$$PZ' = r_b \tan\theta' - \sqrt{r_f^2 - r_b^2} \tag{4}$$

---

* When the involute actually begins at the base circle, $r_f = r_b$. If part of the involute profile has been removed by the cutting tool (undercutting) or if the cutting tool has been withdrawn to a position where cutting begins above the base circle (to avoid undercutting), then $r_f > r_b$.

Also $$PZ = ZT - PT \tag{5}$$

Since $$ZT = \sqrt{R_o^2 - R_b^2} \tag{6}$$

and $$PT = R_b \tan \theta' \,, \tag{7}$$

$$PZ = \sqrt{R_o^2 - R_b^2} - R_b \tan \theta' \tag{8}$$

(2) Determine initial rotation angle (α_{in}) of contact point C_1.

If PZ < PZ'

$$\alpha_{in} = \frac{ZT'}{r_b} \tag{9}$$

From figure 7

$$ZT' = (R_b + r_b) \tan \theta' - \sqrt{R_o^2 - R_b^2} \tag{10}$$

Thus

$$\alpha_{in} = \frac{(R_b + r_b) \tan \theta' - \sqrt{R_o^2 - R_b^2}}{r_b} \tag{11}$$

If PZ' < PZ

$$\alpha_{in} = \frac{Z'T'}{r_b} \tag{12}$$

Using equation 3

$$\alpha_{in} = \frac{\sqrt{r_f^2 - r_b^2}}{r_b} \tag{13}$$

(3) Compute rotation angle α_2 of contact point C_2 given rotation angle α_1 of contact point C_1.

$$\alpha_2 = \alpha_1 + \frac{p_b}{r_b} \tag{14}$$

where p_b is the base pitch, that is, the distance measured on the base circle, from a point on one tooth to a corresponding point on an adjacent tooth.

(4) Calculate distances a_1 and a_2.

$$a_1 = r_b \, \alpha_1 \tag{15}$$

$$a_2 = a_1 + p_b \tag{16}$$

(5) Find angular position corresponding to end of dual contact.

If PZ < PZ', the total length of the line of action is given by WZ. Thus, since the contact point C_2 is a distance p_b from C_1, it is initially a distance WZ − p_b from the final contact position. This corresponds to an angular displacement $\Delta\alpha_1$, from the initial contact position given by

$$\Delta\alpha_1 = \frac{WZ - p_b}{r_b} \tag{17}$$

With

$$WZ = \sqrt{R_o^2 - R_b^2} + \sqrt{r_o^2 - r_b^2} - (R_b + r_b)\tan\theta' \tag{18}$$

equation 17 becomes

$$\Delta\alpha_1 = \frac{\sqrt{R_o^2 - R_b^2} + \sqrt{r_o^2 - r_b^2} - (R_b + r_b)\tan\theta' - p_b}{r_b} \tag{19}$$

The angular position of C_1 corresponding to the end of dual contact is therefore given by

$$\alpha_{fin} = \alpha_{in} + \Delta\alpha_1 \tag{20}$$

If PZ' < PZ, for similar reasons

$$\Delta\alpha_1 = \frac{WZ' - p_b}{r_b} \tag{21}$$

Since

$$WZ' = \sqrt{r_o^2 - r_b^2} - \sqrt{r_f^2 - r_b^2} \tag{22}$$

$$\Delta\alpha_1 = \frac{\sqrt{r_o^2 - r_b^2} - \sqrt{r_f^2 - r_b^2} - p_b}{r_b} \tag{23}$$

and equation 20 again holds.

(6) Stop computations when

$$\alpha_1 = \alpha_{in} + \frac{p_b}{r_b} \tag{24}$$

that is, the pinion has rotated through an angle corresponding to one base pitch. After this, the computations begin repeating.

(7) Determine contact ratio

If $PZ < PZ'$, the contact ratio CR is given by

$$CR = \frac{\sqrt{R_o^2 - R_b^2} + \sqrt{r_o^2 - r_b^2} - (R_b + r_b) \tan \theta'}{p_b} \tag{25}$$

If $PZ' < PZ$

$$CR = \frac{\sqrt{r_o^2 - r_b^2} - \sqrt{r_f^2 - r_b^2}}{p_b} \tag{26}$$

Procedure for Determining Kinematics of Dual Contact When the Gear Drives

(1) Find whether $PZ < PZ'$. The smaller quantity governs. Equations 4 and 8 are still valid.

(2) Determine the initial rotation angle (α_{in}) of point C_1.

$$\alpha_{in} = \frac{WT}{r_b} \tag{27}$$

where

$$WT = (R_b + r_b) \tan \theta' - \sqrt{r_o^2 - r_b^2} \tag{28}$$

(3) Compute rotation angle α_2 of contact point C_2 for rotation angle α_1 of contact point C_1.

$$\alpha_2 = \alpha_1 + \frac{p_b}{R_b} \tag{29}$$

(4) Calculate distances a_1 and a_2

$$a_1 = R_b \, \alpha_1 \tag{30}$$

$$a_2 = a_1 + p_b \tag{31}$$

(5) Find angular position corresponding to end of dual contact.

If $PZ < PZ'$

$$\Delta\alpha_1 = \frac{WZ - p_b}{R_b} \tag{32}$$

where WZ is given by equation 18. Equation 20 again holds.

If PZ' < PZ

$$\Delta\alpha_1 = \frac{WZ' - p_b}{R_b} \tag{33}$$

where WZ' is given by equation 22. Equation 20 is then applied to find the angle corresponding to the end of dual contact.

(6) Stop computations when

$$\alpha_1 = \alpha_{ir} + \frac{p_b}{R_b} \tag{34}$$

(7) Determine contact ratio. Equations 25 and 26 are valid for this case.

Determination of Radius at Which Involute Action Begins and Presence of Undercutting

The inner form radius r_f was determined by a technique developed in reference 2. The method is based on finding the point of intersection of the involute curve and the trochoid curve, which is the curve generated by the cutting tool (rack or hob). This involves solving the following equations (ref 2, equation 5).

$$\tan^{-1}\left\{\left[\left(\frac{r_f\cos\theta}{r_b - b\cos\theta}\right)^2 - 1\right] - \left(1 - \frac{b\cos\theta}{r_b}\right)\left[\left(\frac{r_f\cos\theta}{r_b - b\cos\theta}\right)^2 - 1\right]^{1/2}\right.$$
$$\left. - \theta + \left(1 - \frac{b\cos\theta}{r_b}\right)\tan\theta - \left[\left(\frac{r_f}{r_b}\right)^2 - 1\right]^{1/2} + \tan^{-1}\left\{\left[\left(\frac{r_f}{r_b}\right)^2 - 1\right]^{1/2}\right\}\right\} = 0 \tag{35}$$

where θ is the cutting tooth pressure angle and b is the distance from the sharp corner of the rack tooth (or hob) to the pitch line of the rack. For a rack or hob with a standard addendum this can be expressed as

$$b = \frac{1.2}{P_d} + 0.002 - \text{hob tip radius} \times (1 - \sin\theta) \tag{36}$$

where P_d is the diametral pitch of the mesh. As noted in reference 2, equation 35 cannot be readily solved for r_f in closed form. However, values of r_f can easily be found by the trial-and-error method.

Figure 9, which is a reproduction of figure 6 of reference 2, presents solutions to equation 35 for pressure angles of 10°, 14 1/2°, 20°, 22 1/2°, 25°, 27°, and 30°. The solutions to the left of the minimum value of r_f/r_b for a given pressure angle correspond to cases where part of the involute surface is

destroyed by the cutting operation while those to the right represent cases where the involute is not completely generated (due to withdrawal of the hob). This provides an opportunity for determining whether a pinion tooth is undercut:

$$\text{If } (1 - b/r_p) < r_{as}\text{, tooth is undercut} \tag{37}$$

$$\text{If } (1 - b/r_p) > r_{as}\text{, tooth is not undercut} \tag{38}$$

In these inequalities, r_{as} is the value of $(1 - b/r_p)$ where r_f/r_b is a minimum, and r_p is the pitch radius of the pinion tooth.

Determination of Actual Pressure Angle

Standard pitch circles are the ones which would come into existence, when the gear and pinion are meshed, if none of the standard dimensions are changed. The center distances for the ISAD meshes are slightly larger than the sum of the gear and pinion pitch radii, indicating that the center distances have been extended. This will change the pressure angle as well as the pitch radii. These nonstandard values can be determined by first considering that the base circles remain the same whether the tooth dimensions are changed or not. Thus

$$R_b = R_p \cos\theta = R'_p \cos\theta' \tag{39}$$

$$r_b = r_p \cos\theta = r'_p \cos\theta' \tag{40}$$

where R_p, r_p, and θ are the standard gear pitch radius, pinion pitch radius, and standard pressure angle, respectively, while R'_p, and r'_p, and θ' are the corresponding nonstandard dimensions.

Solving the above equations for R'_p and r'_p

$$R'_p = \frac{R_p \cos\theta}{\cos\theta'} \tag{41}$$

$$r'_p = \frac{r_p \cos\theta}{\cos\theta'} \tag{42}$$

Adding these equations, and noting that the actual center distance c_d can be expressed as

$$c_d = R'_p + r'_p, \tag{43}$$

one obtains

$$c_d = \frac{(R_p + r_p) \cos\theta}{\cos\theta'} \tag{44}$$

from which

$$\theta' = \cos^{-1}\left[\frac{(R_p + r_p)\cos\theta}{c_d}\right] \tag{45}$$

Point Efficiency Expressions for Dual Contact

According to the derivation given in appendix B, the point efficiency ε_p of a pinion-driven involute gear mesh having two pairs of teeth in contact simultaneously is given by

$$\varepsilon_p = \frac{\dfrac{2-\mu[s_1(d - a_1)+s_2(d-a_2)]}{R_b} - \dfrac{\mu\rho_N}{R_b\,(1 + \mu^2)}\sqrt{4 + \mu^2\,[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}}{\dfrac{2-\mu(s_1a_1+s_2a_2)}{r_b} + \dfrac{\mu\rho_n}{r_b\,(1 + \mu^2)}\sqrt{4+\mu^2[4+(s_1+s_2)^2]+\mu^4(s_1+s_2)^2}} \tag{46}$$

while that of a gear-driven mesh is given by

$$\varepsilon_p = \frac{\dfrac{2-\mu[s_1(d-a_1)+s_2(d-a_2)]}{r_b} - \dfrac{\mu\rho_n}{r_b(1+\mu^2)}\sqrt{4+\mu^2[4+(s_1+s_2)^2]+\mu^4(s_1+s_2)^2}}{\dfrac{2-\mu(s_1a_1+s_2a_2)}{R_b} + \dfrac{\mu\rho_N}{R_b(1+\mu^2)}\sqrt{4+\mu^2[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}} \tag{47}$$

In the above equations, ρ_N and ρ_n are the gear and pinion pivot radii, respectively; μ is the coefficient of friction between the gear and pinion teeth as well as at the gear and pinion pivots; and s_1 and s_2 are signum functions which take the values -1, 0, or +1 depending on whether contact points C_1 and C_2 are located before, at, or after the pitch point. These are expressed mathematically by equation B3. In addition, the distances a_1 and a_2 (figs. 7 and 8) can be obtained from equations 15 and 16 if the pinion is the driver or from equations 30 and 31 if the gear is the driver, while the distance d (figs. 7 and 8) is given by

$$d = (R_b + r_b)\tan\theta' \tag{48}$$

Computer Programs

Since the efficiency expressions and logic controls depend on whether the gear or the pinion is the driver, two computer programs ISAD1 and ISAD2 were written. ISAD1 corresponds to the case where the pinion is the driver and ISAD2

corresponds to the case where the gear is the driver. Both programs and a sample output are given in appendixes C, D, and E. Because of the basic similarity of the two programs, they will be discussed together, with differences noted as required.

Input Parameters. The following parameters represent the input data of the program:

PSUBD = P_d, the diametral pitch

NP = N_p, the number of pinion teeth. (This is used to determine the base pitch p_b.)

CAPRP = R_p, the pitch radius of the gear

RP = r_p, the pitch radius of the pinion

CAPRO = R_o, the outside radius of the gear

RO = r_o, the outside radius of the pinion

THETAD = θ, the rack pressure angle (in degrees)

RHOCAPN = ρ_N, the gear pivot radius

RHON = ρ_n, the pinion pivot radius

CD = c_d, the actual center distance

HOBTIPR, the hob tip radius

K, range divisor, that is, the number of points at which the efficiency computations are performed for a rotation of the driving element corresponding to one base pitch. (The computations repeat after this.)

RASFACT = r_{as}, the value of $(1-b/r_p)$ corresponding to the minimum value of r_f/r_b. It can be obtained from figure 9 for a given value of θ.

Computations. The first set of computations relate to the determination of the base circle radii of the gear and pinion (R_b = CAPRB, r_b = RB) as well as of the base pitch (p_b = PB) using standard gear equations.

The main program next calls on the subroutine INNERF to find the inner form radius (r_f = RF) of the pinion. INNERF initially computes the cutter addendum (b = B), as given by equation 36. It then uses the pinion base circle radius (r_b = RB) as the initial "guess" for the inner form radius. This value is substituted into equation 35, which is rewritten in the form:

$$\text{TEST} = \tan^{-1}\{[(\frac{r_f \cos\theta}{r_b - b\cos\theta})^2 - 1]^{1/2}\} - (1 - \frac{b\cos\theta}{r_b})[(\frac{r_f \cos\theta}{r_b - b\cos\theta})^2 - 1]^{1/2}$$

$$- \theta + (1 - \frac{b\cos\theta}{r_b})\tan\theta - [(\frac{r_f}{r_b})^2 - 1]^{1/2} + \tan^{-1}\{[(\frac{r_f}{r_b})^2 - 1]^{1/2}\} \qquad (49)$$

If TEST $\neq 0$, the value of RF is incremented by 0.000001 and the result is substituted in equation 49. This iterative process is continued until a value of RF is obtained for which either TEST $< 10^{-8}$ or the new magnitude of TEST is opposite in sign to that of the previous value of TEST (indicating a root has been located).

At this point, control is returned to the main program, retaining the last value of RF, which will be used as the inner form radius. This value of RF, together with RASFACT, is used in equations 37 and 38 to ascertain whether the pinion tooth is undercut. Based on the results, the program prints out either THE PINION IS UNDERCUT or THE PINION IS NOT UNDERCUT.

The actual gear and pinion pitch radii (R'_p = CAPRP, r'_p = RP)* and pressure angle (θ' = THETA)* are computed next according to equations 41, 42, and 45.

The procedure for determining the kinematic quantities relative to dual contact is then applied. If the pinion is the driver, equations 11, 13, 19, 24, and 25 are used to calculate the initial pinion angle (α_{in} = ALIN), the pinion angle (α_{fin} = ALFIN) corresponding to the end of dual contact, and the contact ratio (CR). If the gear is the driver, equations 18, 22, 25 through 28, 32, and 33 are used to find the initial gear angle (α_{in} = ALIN), the gear angle (α_{fin} = ALFIN) corresponding to the end of dual contact, and the contact ratio (CR). The distance (d = D) between the points of tangency to the base circle is calculated according to equation 48.

The efficiency computations begin with the first contact point at its earliest possible location (corresponding to α_{in}) and the second contact point one base pitch forward along the line of action. The angular increment $\Delta\alpha$ of the driving element is expressed as

$$\text{DELALPH} = \Delta\alpha = (p_b/r_b)/K \qquad (49)$$

if the pinion is the driver, and as

$$\text{DELALPH} = \Delta\alpha = (p_b/R_b)/K \qquad (50)$$

if the gear is the driver.

---

* The computer programs are written in such a manner that the use of identical nomenclature for the standard and actual pitch radii and pressure angles does not cause errors.

Once the current value of α_1 = ALPHA1 is established by adding DELALPH to the previous value of ALPHA1 (ALPHA1 = ALIN for the first round of computations), the distance a_1 = A1 is computed by using either equation 15 or equation 30 depending on which element is the driver. The signum function s_1 is then determined using equation B3.

The program next decides whether the mesh is in the single or dual contact mode by comparing the magnitude of ALPHA1 to that of ALFIN. If ALPHA1 is less than ALFIN, dual contact exists and the distance a_2 is computed according to equation 16 when the pinion is the driver, or according to equation 31 when the gear is the driver. The signum function s_2 is then found by applying equation 47. Finally, the point efficiency ε_p = POINTEF is obtained from equation 46 if the pinion is the driver, or from equation 48 if the gear is the driver.

If ALPHA1 is greater than ALFIN, only single-point contact exists. If the pinion is the driving element, equation A-26 of appendix A of reference 1 can then be used to determine POINTEF. If the gear is the driving element, a modified form of this equation (where the gear and pinion parameters are interchanged) can be used to determine the point mesh efficiency. The point efficiency computations terminate after the value of ALPHA1 has been incremented by DELALPH K times.

While the cycle efficiency (CYCLEFF) is not needed in this study, it is computed by the computer programs for informational purposes. It is based on equation C10 of appendix C of reference 1 with the total angular range for the calculations now given by p_b/r_b when the pinion is the driver and p_b/R_b when the gear is the driver. Thus equation C10 of reference 1, when adapted to the present case, becomes

$$\text{CYCLEFF} = \frac{\Delta\alpha \; \Sigma\varepsilon_p}{p_b/r_b} \text{ for the pinion driving} \tag{51}$$

$$\text{CYCLEFF} = \frac{\Delta\alpha \; \Sigma\varepsilon_p}{p_b/R_b} \text{ for the gear driving} \tag{52}$$

where $\Sigma\varepsilon_p$ is the sum of the point efficiencies for the K computation points.

The above set of computations is initially performed with μ = MU = 0. After the computations are completed, MU is incremented by 0.025. This continues until MU = 0.3.

<u>Output of Program</u>. The input parameters PSUBD, NP, CAPRP, RP, CAPRO, RO, THETAD, RHOCAPN, RHON, CD, HOBTIPR, K, and RASFACT are reproduced. The program then prints computed gear and pinion parameters CAPRB, RB, PB, and RF. The output next indicates whether the pinion is undercut. This is followed by the actual gear parameters CAPRP, RP, and THETAD. In addition, the contact ratio CR, the initial gear or pinion angle ALIN, and the angle corresponding to the end of the dual contact ALFIN are provided. POINTEF as a function of the driver angle ALPHA1D, the signum functions s_1 and s_2, and the cycle efficiency CYCLEFF are also listed.

ISAD Involute Gear Mesh Parameters

Tables 5 through 13 present the parameters necessary to use computer programs ISAD1 and ISAD2. These tables also give the operating center distance for the mesh and the tip radius of the cutting tool.

Efficiency Results

Computer programs ISAD1 and ISAD2 were applied to the involute meshes of the ISAD mechanism. Again, the coefficient of friction was permitted to vary from 0 to 0.3 in steps of 0.05. A sample output using the drive arm pinion gear and main shaft mesh of the exoatmospheric drive assembly as an example is shown in appendix E. (The general point efficiencies of all of the meshes are not given in this report since only the minimum efficiencies are needed.) The minimum value of the point efficiency was retained for each value of the coefficient of friction. These were then used to generate the minimum efficiency graphs for each mesh (figs. 10 through 18).

Additionally, data pertaining to the involute meshes, such as the pinion inner form radius, the contact ratio, and a statement indicating the existence of undercutting, are shown in tables 14 through 22 for each of the involute meshes studied.

CONCLUSIONS AND DISCUSSION

It can be seen from figures 2 through 6 and 8 through 17 that the lowest minimum efficiency for the clock meshes is approximately 0.8 while that for the involute meshes is 0.7. Of course, these occur for the highest coefficient of friction, $\mu = 0.3$. At $\mu = 0.1$, which is a more reasonable value of the coefficient of friction, the efficiencies for both types of meshes are the same, approximately 0.9. Because the mesh efficiencies are well above zero, the gears should transmit torque as designed, and binding of the meshes should not be regarded as a potential problem area.

Involute meshes in tables 14 through 22 show that most of the pinions are undercut. The difference between the pinion inner form radius and its base radius indicates the amount of undercutting. The most significant undercutting occurs for the output pinion and valve gear of the valve drive assembly. In this case, 0.1774 in. - 0.1762 in. = 0.0014 in. (4.506 mm - 4.476 mm = 0.030 mm) of the pinion tooth is removed. Since the length of the dedendum is $1.2/P_d = 1.2/32 = 0.0375$ in. (0.9525 mm), 3.7% of the dedendum is cut off. While this is a small amount, it is recommended that a stress analysis study be conducted for each of the undercut gear meshes.

Tables 14 through 22 also show that the contact ratio for each of the involute meshes is greater than one. This means that at least one pair of teeth will always be in contact, providing smooth motion and eliminating the possibility of impulsive loading of the gear teeth.

REFERENCES

1. G. G. Lowen and F. R. Tepper, "Fuze Gear Train Analysis," Technical Report ARLCD-TR-79030, ARRADCOM, Dover, NJ, December 1979.

2. R. A. Shaffer, "An Analysis of the Undercutting Problem in Involute Spur Gearing," Technical Report R-1606, Frankford Arsenal, Philadelphia, PA May 1962.

Table 1. Propulsion timer gear and pinion

Properties	Gear[a] (driver)	Pinion[b]
Diametral pitch	100	100
Number of teeth	72	18
Pitch radius (in.)	0.360	0.090
(mm)	9.144	2.286
Distance from pivot to center of curvature (in.)	0.3577[c]	0.09012[d]
(mm)	9.086	2.2890
Radius of curvature of circular arc portion of tooth profile (in.)	0.0223	0.008
(mm)	0.5664	0.203
Tooth thickness (in.)	0.014	0.012
(mm)	0.356	0.305
Pivot radius (in.)	0.06237	0.0155
(mm)	1.5842	0.3937

---

[a] Drawing 2406-344

[b] Drawing 1060-34-24

[c] Based on computations in appendix A.

[d] Based on computations in reference 1, appendix D.

Table 2. Gear no. 1 and pinion no. 1

Properties	Gear[a] (driver)	Pinion[b]
Diametral pitch	134.9	134.9
Number of teeth	55	8
Pitch radius (in.)	0.20385	0.02965
(mm)	5.1778	0.75311
Distance from pivot to center of curvature (in.)	0.20392[c]	0.02974[d]
(mm)	5.1796	0.75540
Radius of curvature of circular arc portion of tooth profile (in.)	0.011	0.007
(mm)	0.279	0.179
Tooth thickness (in.)	0.011	0.0093
(mm)	0.279	0.2362
Pivot radius (in.)	0.0155	0.01025
(mm)	0.3937	0.26035

[a] Drawing 1060-17-1

[b] Drawing 1060-34-15

[c] Based on computations in appendix A.

[d] Based on computations in reference 1, appendix D.

Table 3. Fifty-tooth gear and escape wheel pinion

Properties	Gear[a] (driver)	Pinion[b]
Diametral pitch	134.9	134.9
Number of teeth	50	7
Pitch radius (in.)	0.18535	0.02595
(mm)	4.7099	0.65913
Distance from pivot to center of curvature (in.)	0.18543[c]	0.0245[d]
(mm)	4.70992	0.6223
Radius of curvature of circular arc portion of tooth profile (in.)	0.011	0.007
(mm)	0.279	0.178
Tooth thickness (in.)	0.011	0.009
(mm)	0.279	0.229
Pivot radius (in.)	0.01025	0.0085
(mm)	0.26035	0.2154

[a] Drawing 1060-17-10

[b] Drawing 1060-34-14

[c] Based on computations in appendix A.

[d] Based on computations in reference 1, appendix D.

Table 4. Exoatmospheric timer gear and pinion

Properties	Gear[a] (driver)	Pinion[b]
Diametral pitch	100	100
Number of teeth	112	18
Pitch radius (in.)	0.560	0.090
(mm)	14.224	2.286
Distance from pivot to center of curvature (in.)	0.55791[c]	0.09012[d]
(mm)	14.171	2.2890
Radius of curvature of circular arc portion of tooth profile (in.)	0.0223	0.008
(mm)	0.5664	0.203
Tooth thickness (in.)	0.014	0.012
(mm)	0.356	0.305
Pivot radius (in.)	0.078	0.0155
(mm)	1.981	0.3937

[a] Drawing 2406-366

[b] Drawing 1060-34-24

[c] Based on computations in appendix A.

[d] Based on computations in reference 1, appendix D.

Table 5. Drive plate and valve lock cam mesh of propulsion drive assembly

Properties	Gear[a] (driver)	Pinion[b]
Diametral pitch	64	64
Number of teeth	80	64
Pitch radius (in.)	0.625	0.500
(mm)	15.875	12.700
Outside radius (in.)	0.64063	0.51563
(mm)	16.672	13.097
Pressure angle (deg)	20	20
Pivot radius (in.)	0.062	0.062
(mm)	1.575	1.575

The operating center distance is 1.128 in. (28.651 mm).

The hob-tip radius is 0.

[a] Drawing 2406-342

[b] Drawing 2406-361

Table 6. Drive arm pinion gear and main shaft of exoatmospheric drive assembly

Properties	Gear[a] (driver)	Pinion[b]
Diametral pitch	80	80
Number of teeth	72	16
Pitch radius (in.)	0.450	0.100
(mm)	11.430	2.540
Outside radius (in.)	0.4625	0.1125
(mm)	11.748	2.858
Pressure angle (deg)	20	20
Pivot radius (in.)	0.125	0.09
(mm)	3.175	2.286

The operating center distance is 0.551 in. (13.995 mm).

The hob-tip radius is 0.

---

[a] Drawing 2406-376

[b] Drawing 2406-364

Table 7. Differential gear and motor pinion of valve drive assembly

Properties	Gear[a]	Pinion[b] (driver)
Diametral pitch	80	80
Number of teeth	40	14
Pitch radius (in.)	0.25	0.0875
(mm)	6.35	2.223
Outside radius (in.)	0.2626	0.100
(mm)	6.670	2.540
Pressure angle (deg)	20	20
Pivot radius (in.)	0.062	0.062
(mm)	1.575	1.575

The operating center distance is 0.338 in. (8.585 mm).

The hob-tip radius is 0.

[a] Drawing 2406-316

[b] Drawing 2406-314

Table 8. Differential gear and differential pinion of valve drive assembly

Properties	Gear[a]	Pinion[b] (driver)
Diametral pitch	80	80
Number of teeth	40	15
Pitch radius (in.)	0.25	0.09375
(mm)	6.35	2.3813
Outside radius (in.)	0.2625	0.10625
(mm)	6.678	2.6988
Pressure angle (deg)	20	20
Pivot radius (in.)	0.062	0.020
(mm)	1.575	1.575

The operating center distance is 0.3453 in. (8.771 mm).

The hob-tip radius is 0.

[a] Drawing 2377-336

[b] Drawing 2406-316

Table 9. Differential pinion mesh of valve drive assembly

Properties	Driving* pinion	Driven* pinion
Diametral pitch	80	80
Number of teeth	15	15
Pitch radius (in.)	0.09375	0.09375
(mm)	2.3813	2.3813
Outside radius (in.)	0.10625	0.10625
(mm)	2.6988	2.6988
Pressure angle (deg)	20	20
Pivot radius (in.)	0.020	0.020
(mm)	1.575	1.575

The operating center distance is 0.1875 in. (4.763 mm).

The hob-tip radius is 0.

* Drawing 2377-336

Table 10. Idler gear and differential output pinion of valve drive assembly

Properties	Idler gear[a]	Pinion[b] (driver)
Diametral pitch	64	64
Number of teeth	24	14
Pitch radius (in.)	0.1875	0.10938
(mm)	4.763	2.7783
Outside radius (in.)	0.20312	0.125
(mm)	5.1593	3.175
Pressure angle (deg)	20	20
Pivot radius (in.)	0.062	0.062
(mm)	1.575	1.575

The operating center distance is 0.3005 in. (7.633 mm).

The hob-tip radius is 0.

---

[a] Drawing 2406-477

[b] Drawing 2406-314

Table 11. Valve drive gear and idler pinion of valve drive assembly

Properties	Gear[a]	Pinion[b] (driver)
Diametral pitch	64	64
Number of teeth	50	24
Pitch radius (in.)	0.39063	0.1875
(mm)	9.9220	4.763
Outside radius (in.)	0.40625	0.20313
(mm)	10.319	5.1595
Pressure angle (deg)	20	20
Pivot radius (in.)	0.094	0.062
(mm)	2.388	1.575

The operating center distance is 0.5818 in. (14.778 mm).

The hob-tip radius is 0.

[a] Drawing 2406-313

[b] Drawing 2406-477

Table 12. Output gear and valve drive pinion of valve drive assembly

Properties	Gear[a]	Pinion[b] (driver)
Diametral pitch	48	48
Number of teeth	36	12
Pitch radius (in.)	0.375	0.125
(mm)	9.525	3.175
Outside radius (in.)	0.39583	0.14583
(mm)	10.054	3.7041
Pressure angle (deg)	20	20
Pivot radius (in.)	0.125	0.094
(mm)	3.175	2.388

The operating center distance is 0.5046 in. (13.731 mm).

The hob-tip radius is 0.

---

[a] Drawing 2406-318

[b] Drawing 2406-315

Table 13. Valve gear and output pinion of valve drive assembly

Properties	Gear[a] (driver)	Pinion[b]
Diametral pitch	32	32
Number of teeth	36	12
Pitch radius (in.)	0.5625	0.1875
(mm)	14.288	4.763
Outside radius (in.)	0.59375	0.21875
(mm)	15.081	5.5563
Pressure angle (deg)	20	20
Pivot radius (in.)	0.125	0.125
(mm)	3.175	3.175

The operating center distance is 0.751 in. (19.075 mm).

The hob-tip radius is 0.

---

[a] Drawing 2406-412

[b] Drawing 2406-317

Table 14. Data for drive plate and valve lock cam mesh of propulsion drive assembly

```
DIAMETRAL PITCH (PSUBD) =  64.0
PINION NUMBER OF TEETH (NP) = 64.
STANDARD GEAR PITCH RADIUS (CAPRP) = .62500   STANDARD PINION PITCH RADIUS (RP) = .50000
GEAR OUTSIDE RADIUS (CAPRO) = .64063   PINION OUTSIDE RADIUS (RO) = .51563
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .062   PINION PIVOT RADIUS (RHON) = .064
OPERATING CENTER DISTANCE (CD) = 1.128
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .5873
BASE RADIUS OF PINION (RB) = .4698
BASE PITCH = .0461
PINION INNER FORM RADIUS (RF) = .4826
THE PINION IS NOT UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .62667
ACTUAL PINION PITCH RADIUS (RP) = .50133
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 20.41

CONTACT RATIO (CR) =1.62
INITIAL GEAR ANGLE (ALIN) = 17.663
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 20.462
```

Figure 15. Minimum efficiency of idler and different 1 output pinion of valve drive assembly vs coefficient of friction

```
DIAMETRAL PITCH (PSUBD) =  80.0

PINION NUMBER OF TEETH (NP) = 16.

STANDARD GEAR PITCH RADIUS (CAPRP) = .45000   STANDARD PINION PITCH RADIUS (RP) = .10000

GEAR OUTSIDE RADIUS (CAPRO) = .46250   PINION OUTSIDE RADIUS (RO) = .11250

PRESSURE ANGLE IN DEGREES (THETAD) = 20.00

GEAR PIVOT RADIUS (RHOCAPN) = .125   PINION PIVOT RADIUS (RHON) = .090

OPERATING CENTER DISTANCE (CD) =  .551

GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000

RANGE DIVISOR (K) =  25

SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883


BASE RADIUS OF GEAR (CAPRB) = .4229

BASE RADIUS OF PINION (RB) = .0940

BASE PITCH = .0369

PINION INNER FORM RADIUS (RF) = .0942

THE PINION IS UNDERCUT

ACTUAL GEAR PITCH RADIUS (CAPRP) = .45082

ACTUAL PINION PITCH RADIUS (RP) = .10018

ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 20.28


CONTACT RATIO (CR) =1.49

INITIAL GEAR ANGLE (ALIN) =  4.132

ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 13.948
```

Table 16. Data for motor pinion and differential gear mesh of valve drive assembly

```
DIAMETRAL PITCH (PSUBD) =  80.0
PINION NUMBER OF TEETH (NP) = 14.
STANDARD GEAR PITCH RADIUS (CAPRP) = .25000   STANDARD PINION PITCH RADIUS (RP) = .08750
GEAR OUTSIDE RADIUS (CAPRO) = .26250   PINION OUTSIDE RADIUS (RO) = .10000
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .062   PINION PIVOT RADIUS (RHON) = .062
OPERATING CENTER DISTANCE (CD) =  .338
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .2349
BASE RADIUS OF PINION (RB) = .0822
BASE PITCH = .0369
PINION INNER FORM RADIUS (RF) = .0826
THE PINION IS UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .25037
ACTUAL PINION PITCH RADIUS (RP) = .08763
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 20.23

CONTACT RATIO (CR) =1.32
INITIAL GEAR ANGLE (ALIN) =  5.743
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 14.955
```

Table 17. Data for differential pinion mesh of valve drive assembly

```
DIAMETRAL PITCH (PSUBD) =  80.0
PINION NUMBER OF TEETH (NP) = 15.
STANDARD GEAR PITCH RADIUS (CAPRP) = .09375   STANDARD PINION PITCH RADIUS (RP) = .09375
GEAR OUTSIDE RADIUS (CAPRO) = .10625   PINION OUTSIDE RADIUS (RO) = .10625
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .020   PINION PIVOT RADIUS (RHON) = .020
OPERATING CENTER DISTANCE (CD) =  .188
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .0881
BASE RADIUS OF PINION (RB) = .0881
BASE PITCH = .0369
PINION INNER FORM RADIUS (RF) = .0884
THE PINION IS UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .09375
ACTUAL PINION PITCH RADIUS (RP) = .09375
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 20.00

CONTACT RATIO (CR) =1.41
INITIAL GEAR ANGLE (ALIN) =  3.077
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 12.812
```

Table 18. Data for differential gear and differential pinion mesh of valve drive assembly

```
DIAMETRAL PITCH (PSUBD) =  80.0
PINION NUMBER OF TEETH (NP) = 15.
STANDARD GEAR PITCH RADIUS (CAPRP) = .25000   STANDARD PINION PITCH RADIUS (RP) = .09375
GEAR OUTSIDE RADIUS (CAPRO) = .26250   PINION OUTSIDE RADIUS (RO) = .10625
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .062   PINION PIVOT RADIUS (RHON) = .020
OPERATING CENTER DISTANCE (CD) =  .345
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .2349
BASE RADIUS OF PINION (RB) = .0881
BASE PITCH = .0369
PINION INNER FORM RADIUS (RF) = .0884
THE PINION IS UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .25115
ACTUAL PINION PITCH RADIUS (RP) = .09418
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 20.71

CONTACT RATIO (CR) =1.41
INITIAL GEAR ANGLE (ALIN) =  4.896
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 14.461
```

Table 19. Data for idler and differential output pinion mesh of valve drive assembly

```
DIAMETRAL PITCH (PSUBD) =  64.0
PINION NUMBER OF TEETH (NP) = 14.
STANDARD GEAR PITCH RADIUS (CAPRP) = .18750   STANDARD PINION PITCH RADIUS (RP) = .10938
GEAR OUTSIDE RADIUS (CAPRO) = .20313   PINION OUTSIDE RADIUS (RO) = .12500
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .062   PINION PIVOT RADIUS (RHON) = .062
OPERATING CENTER DISTANCE (CD) =  .301
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .1762
BASE RADIUS OF PINION (RB) = .1028
BASE PITCH = .0461
PINION INNER FORM RADIUS (RF) = .1032
THE PINION IS UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .18979
ACTUAL PINION PITCH RADIUS (RP) = .11071
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 21.82

CONTACT RATIO (CR) =1.31
INITIAL GEAR ANGLE (ALIN) =  5.919
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 13.946
```

Table 20. Data for valve drive gear and idler mesh of valve drive assembly

```
DIAMETRAL PITCH (PSUBD) =  64.0
PINION NUMBER OF TEETH (NP) = 24.
STANDARD GEAR PITCH RADIUS (CAPRP) = .39063   STANDARD PINION PITCH RADIUS (RP) = .18750
GEAR OUTSIDE RADIUS (CAPRO) = .40625   PINION OUTSIDE RADIUS (RO) = .20313
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .094   PINION PIVOT RADIUS (RHON) = .062
OPERATING CENTER DISTANCE (CD) =  .582
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .3671
BASE RADIUS OF PINION (RB) = .1762
BASE PITCH = .0461
PINION INNER FORM RADIUS (RF) = .1762
THE PINION IS NOT UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .39311
ACTUAL PINION PITCH RADIUS (RP) = .18869
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 20.97

CONTACT RATIO (CR) =1.45
INITIAL GEAR ANGLE (ALIN) = 11.108
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 17.868
```

Table 21. Data for output gear and valve drive pinion mesh of valve drive assembly

```
DIAMETRAL PITCH (PSUBD) =  48.0
PINION NUMBER OF TEETH (NP) = 12.
STANDARD GEAR PITCH RADIUS (CAPRP) = .37500   STANDARD PINION PITCH RADIUS (RP) = .12500
GEAR OUTSIDE RADIUS (CAPRO) = .39583   PINION OUTSIDE RADIUS (RO) = .14583
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .125   PINION PIVOT RADIUS (RHON) = .094
OPERATING CENTER DISTANCE (CD) =  .505
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .3524
BASE RADIUS OF PINION (RB) = .1175
BASE PITCH = .0615
PINION INNER FORM RADIUS (RF) = .1184
THE PINION IS UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .37845
ACTUAL PINION PITCH RADIUS (RP) = .12615
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 21.39

CONTACT RATIO (CR) =1.17
INITIAL GEAR ANGLE (ALIN) =  7.086
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 12.461
```

Table 22. Data for output gear and valve drive pinion mesh of valve drive assembly

```
DIAMETRAL PITCH (PSUBD) =  32.0
PINION NUMBER OF TEETH (NP) = 12.
STANDARD GEAR PITCH RADIUS (CAPRP) = .56250   STANDARD PINION PITCH RADIUS (RP) = .18750
GEAR OUTSIDE RADIUS (CAPRO) = .59375   PINION OUTSIDE RADIUS (RO) = .21875
PRESSURE ANGLE IN DEGREES (THETAD) = 20.00
GEAR PIVOT RADIUS (RHOCAPN) = .125   PINION PIVOT RADIUS (RHON) = .125
OPERATING CENTER DISTANCE (CD) =  .751
GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000
RANGE DIVISOR (K) =  25
SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883

BASE RADIUS OF GEAR (CAPRB) = .5286
BASE RADIUS OF PINION (RB) = .1762
BASE PITCH = .0923
PINION INNER FORM RADIUS (RF) = .1774
THE PINION IS UNDERCUT
ACTUAL GEAR PITCH RADIUS (CAPRP) = .56325
ACTUAL PINION PITCH RADIUS (RP) = .18775
ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 20.21

CONTACT RATIO (CR) =1.18
INITIAL GEAR ANGLE (ALIN) = 14.068
ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) - 15.866
```

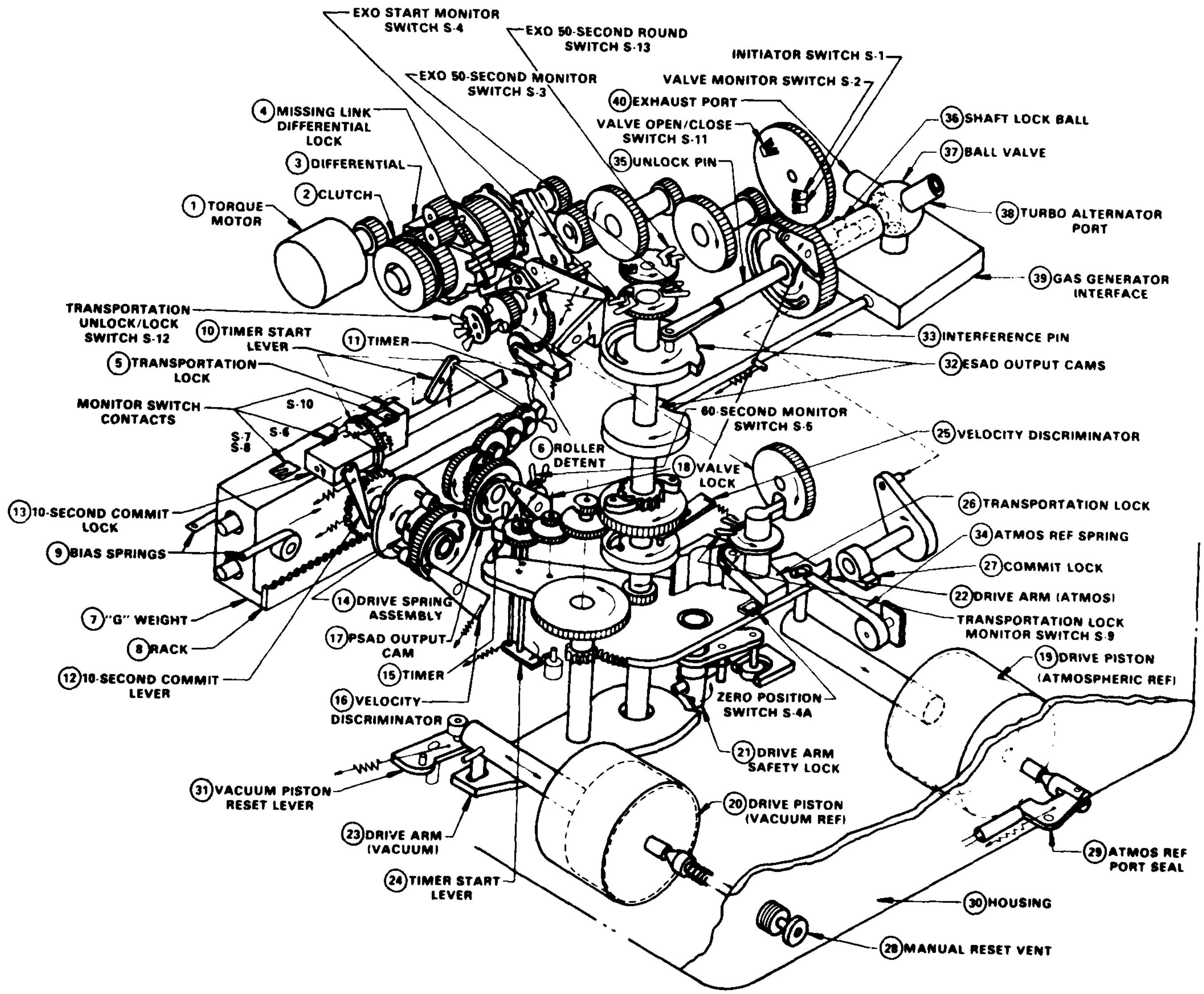


Figure 1. Pershing II integrated safe/arm device (functional schematic)

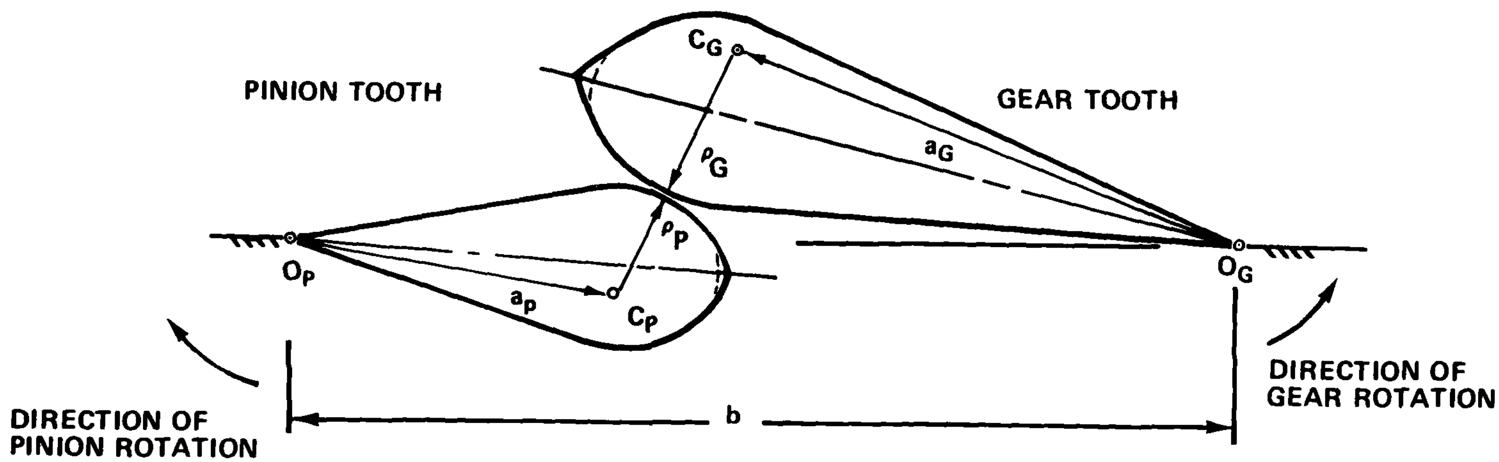


Figure 2. Typical clock gear type mesh

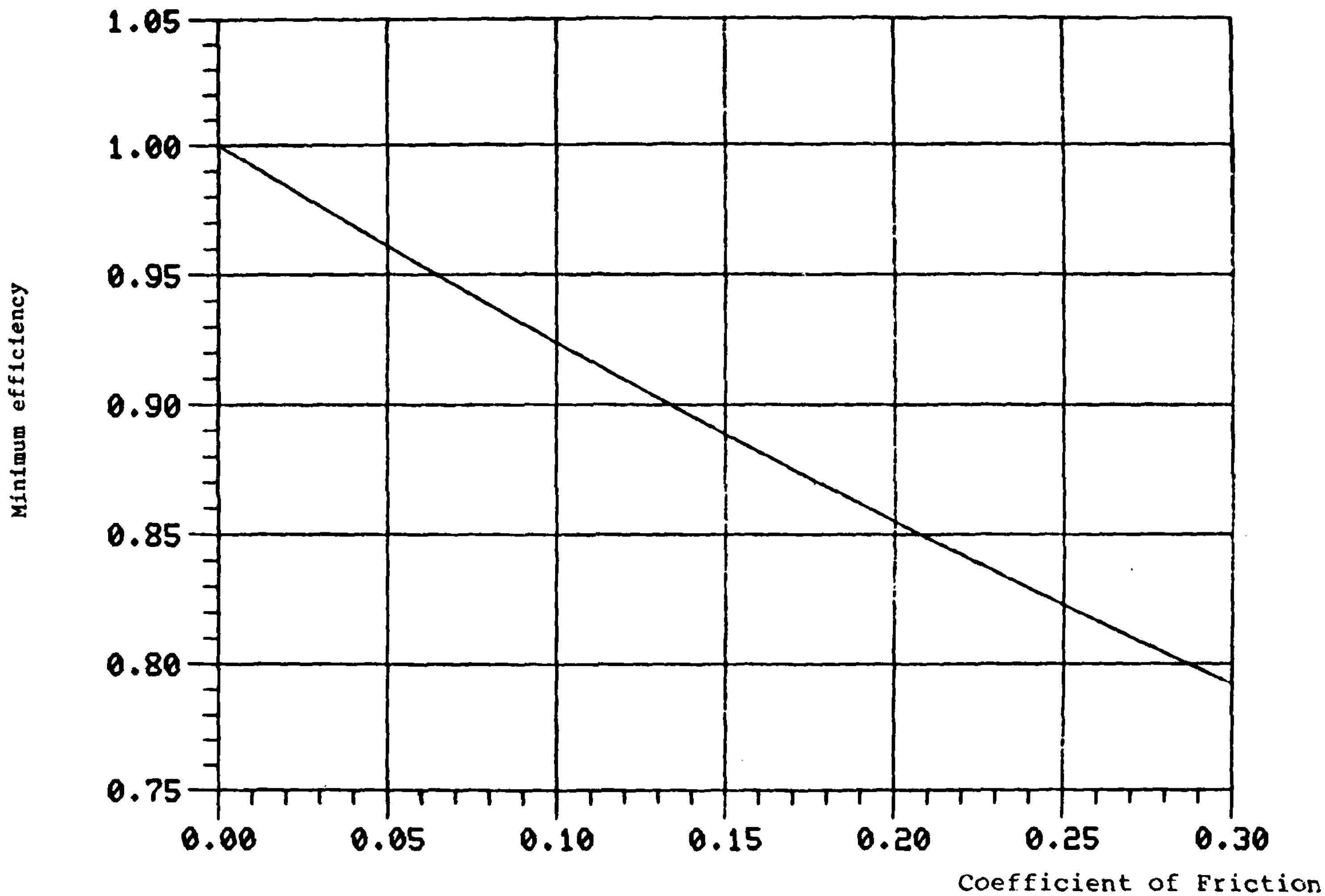


Figure 3. Minimum efficiency of propulsion gear and pinion vs coefficient of friction

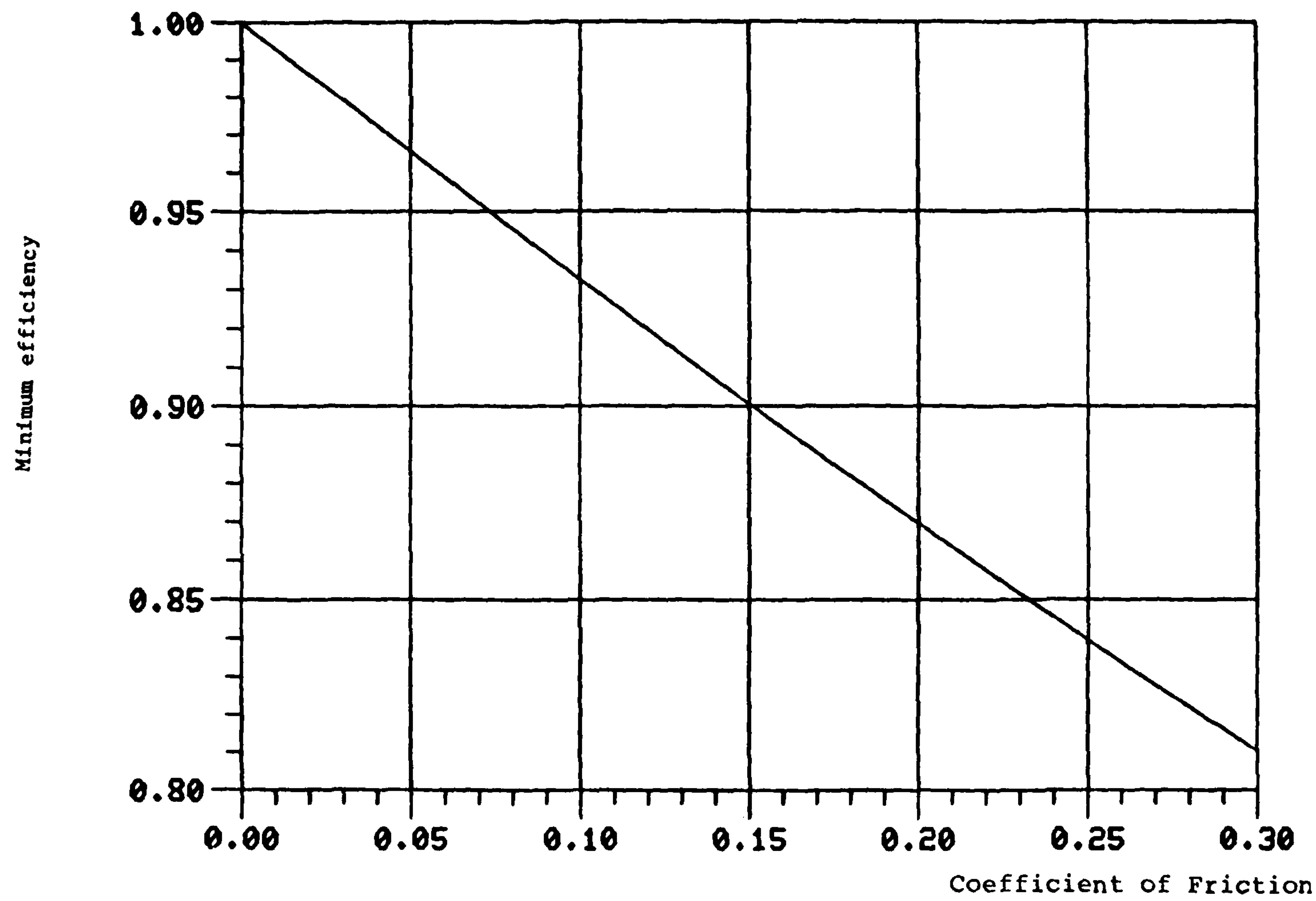


Figure 4. Minimum efficiency of gear no. 1 and pinion no. 1 vs coefficient of friction

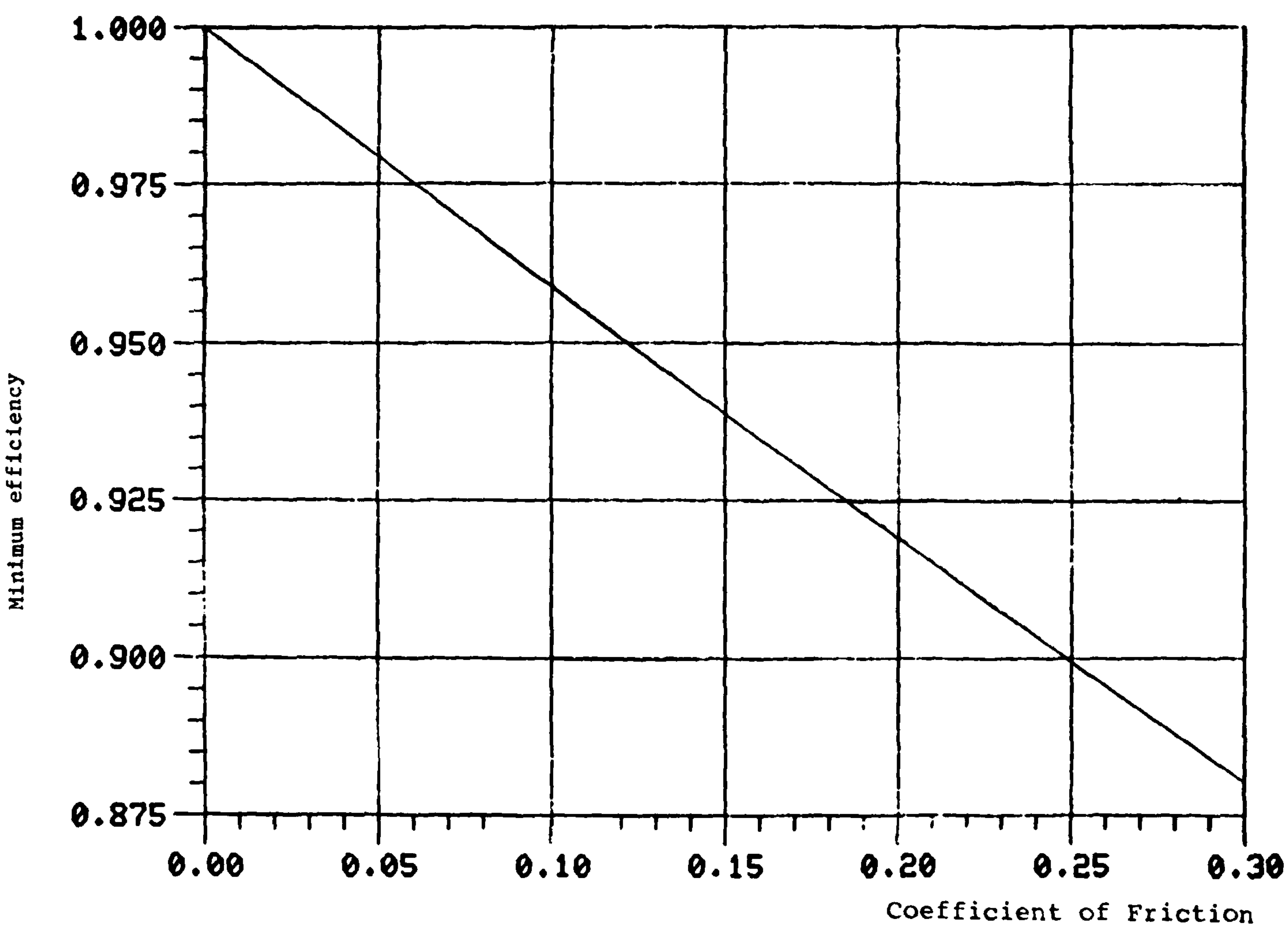


Figure 5. Minimum efficiency of 50-tooth gear and escape wheel pinion vs coefficient of friction

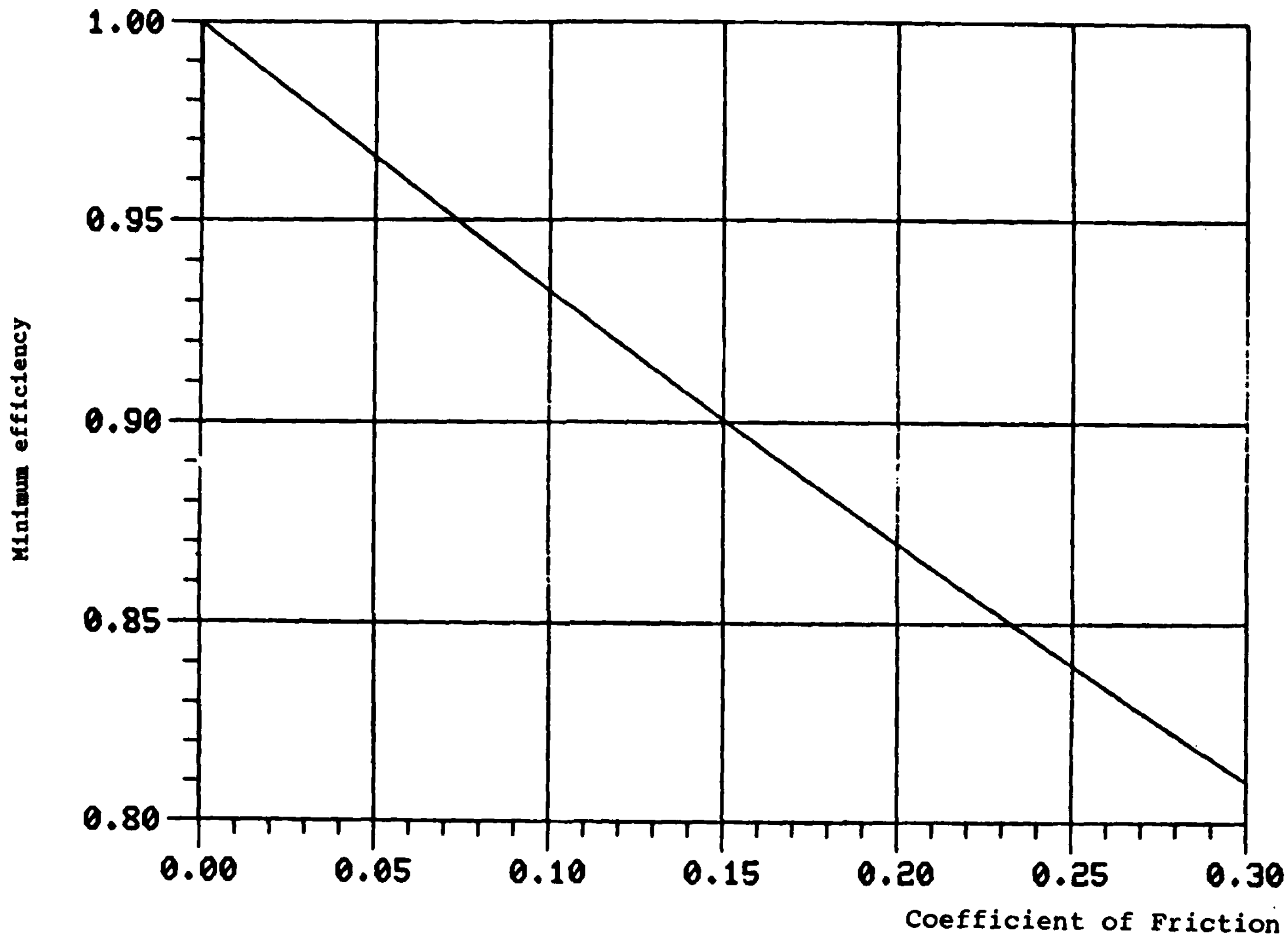


Figure 6. Minimum efficiency of exoatmospheric timer gear and pinion vs coefficient of friction

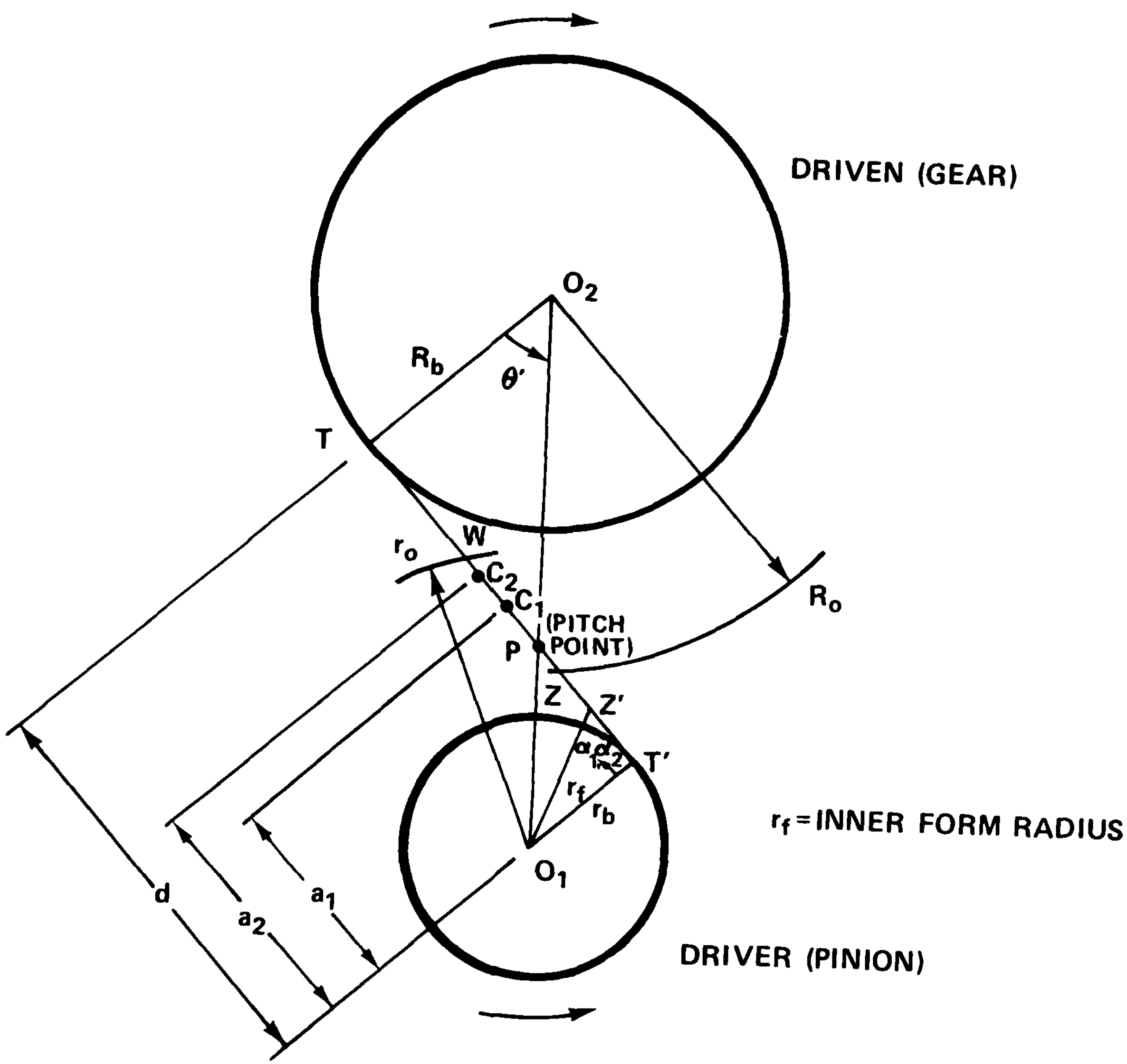


Figure 7. Nomenclature for involute mesh with pinion as driver

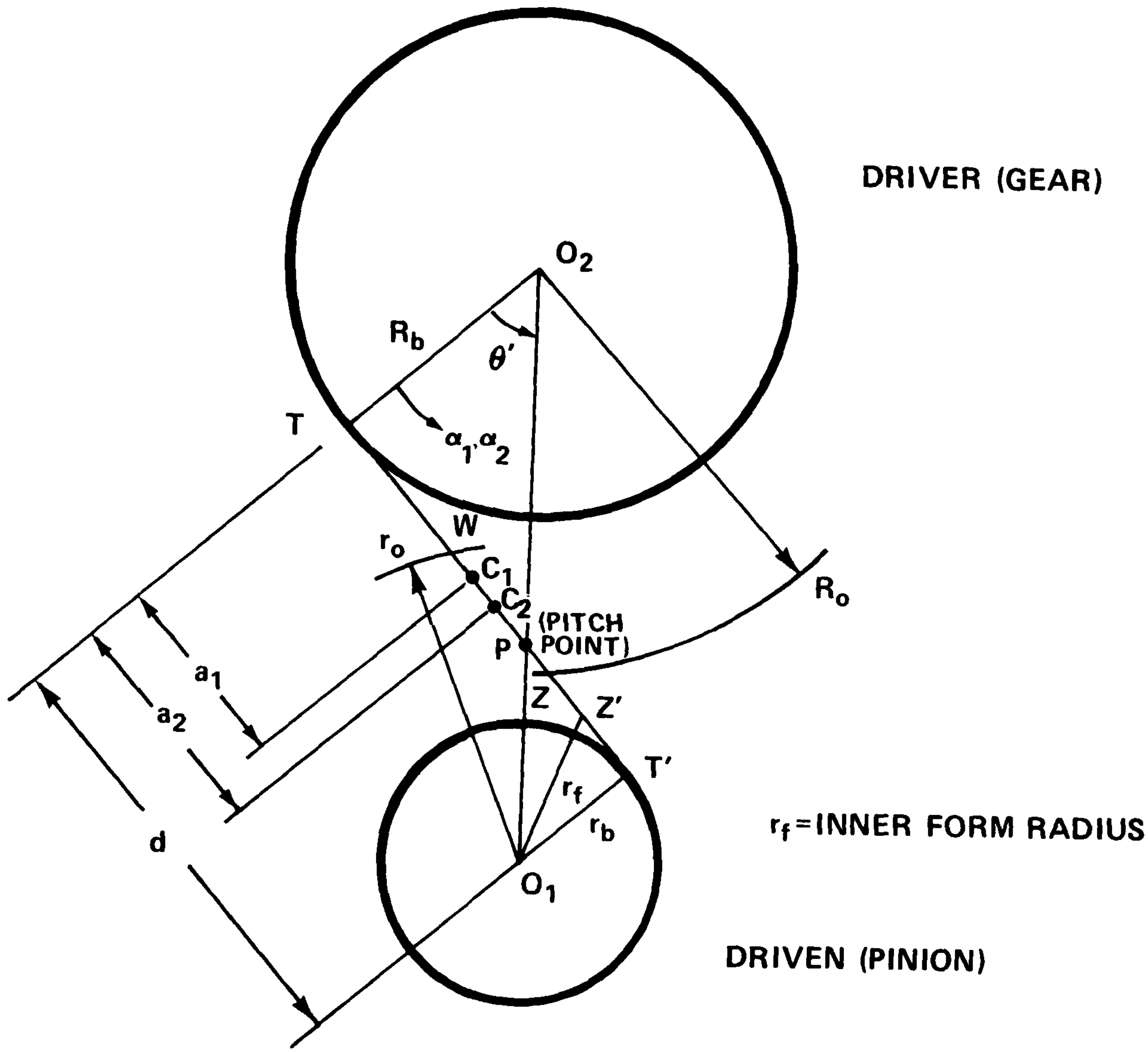


Figure 8. Nomenclature for involute mesh with gear as driver

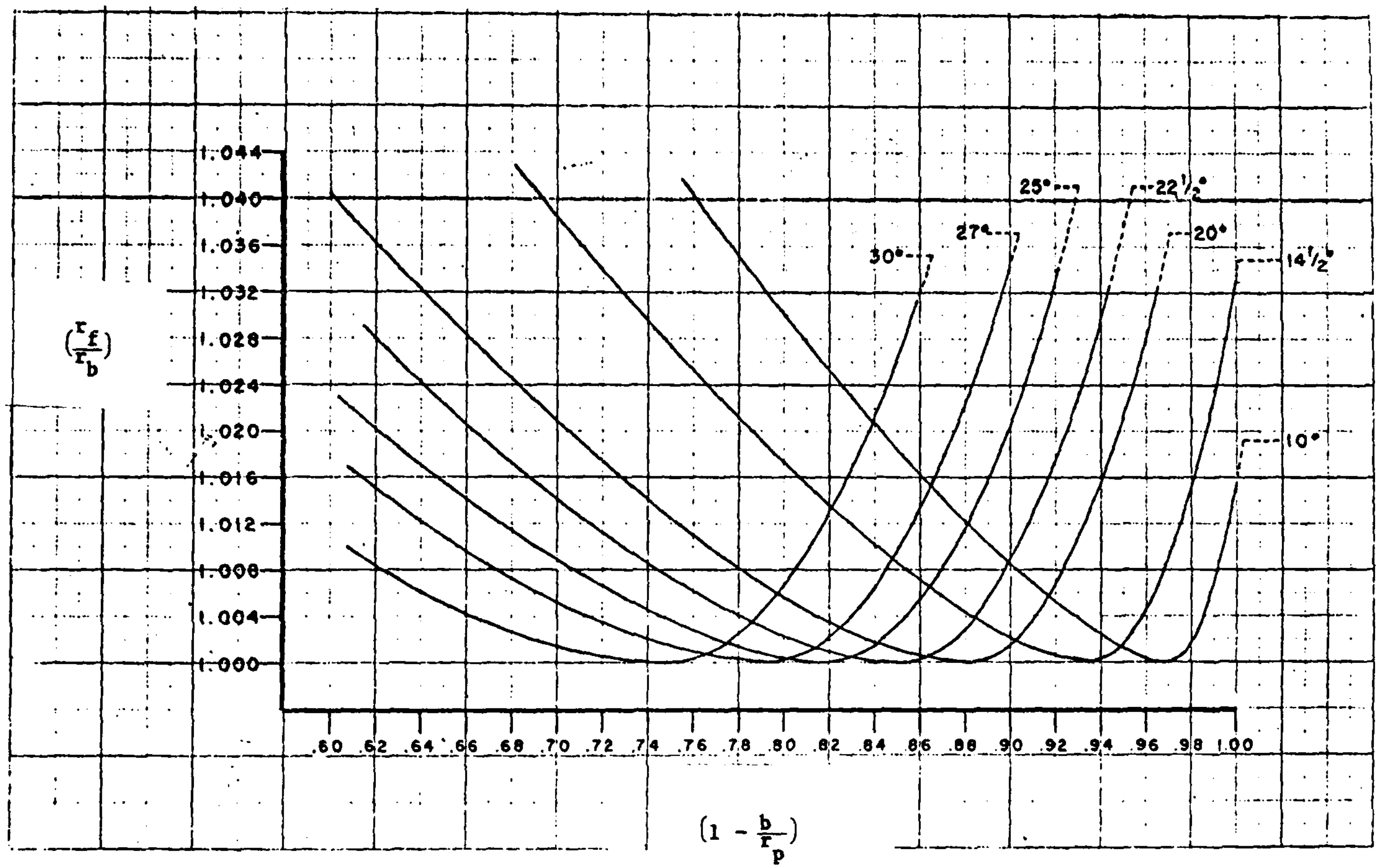


Figure 9. True involute form radius of an undercut gear

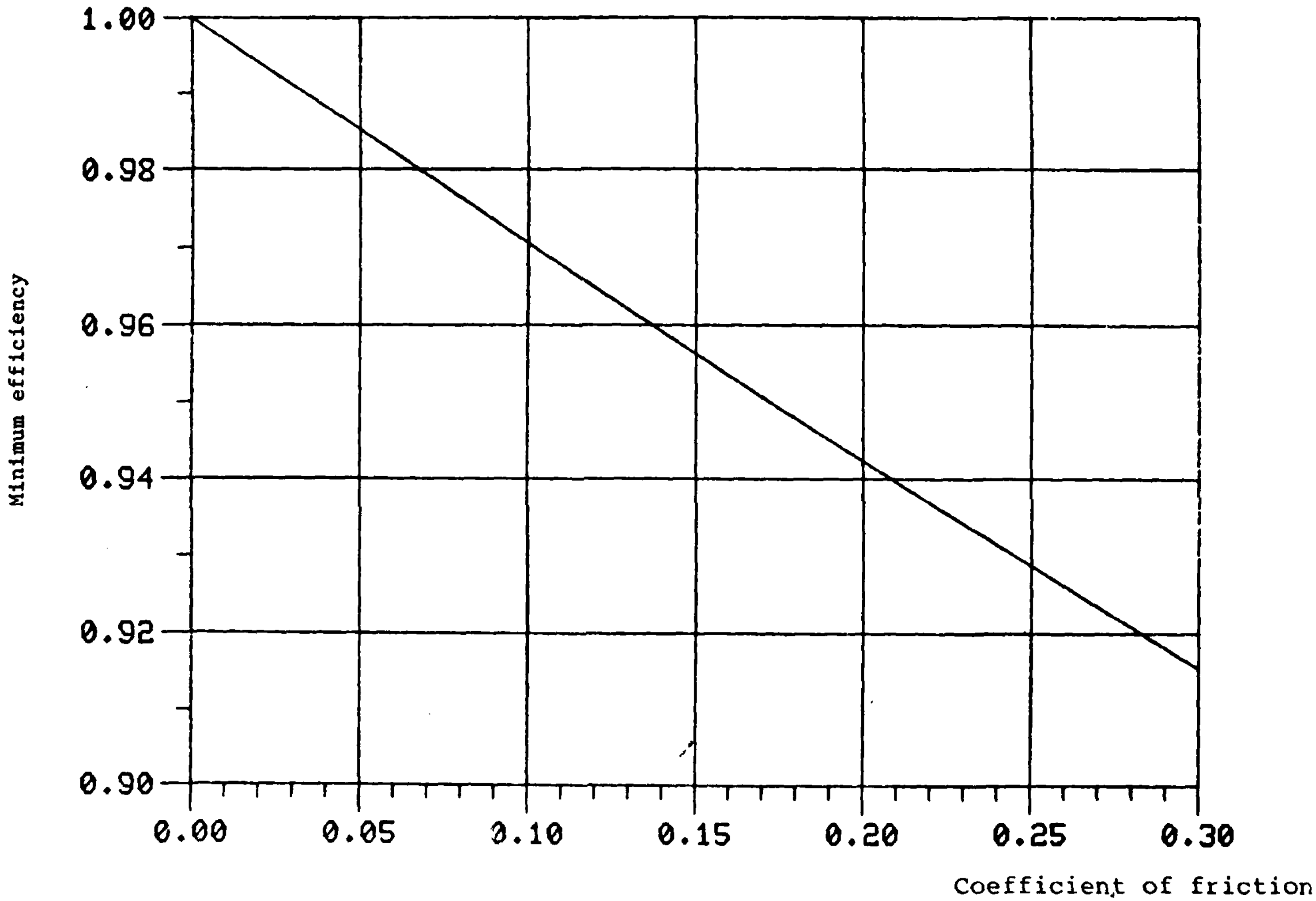


Figure 10. Minimum efficiency of drive plate and valve lock cam mesh of propulsion drive assembly vs coefficient of friction

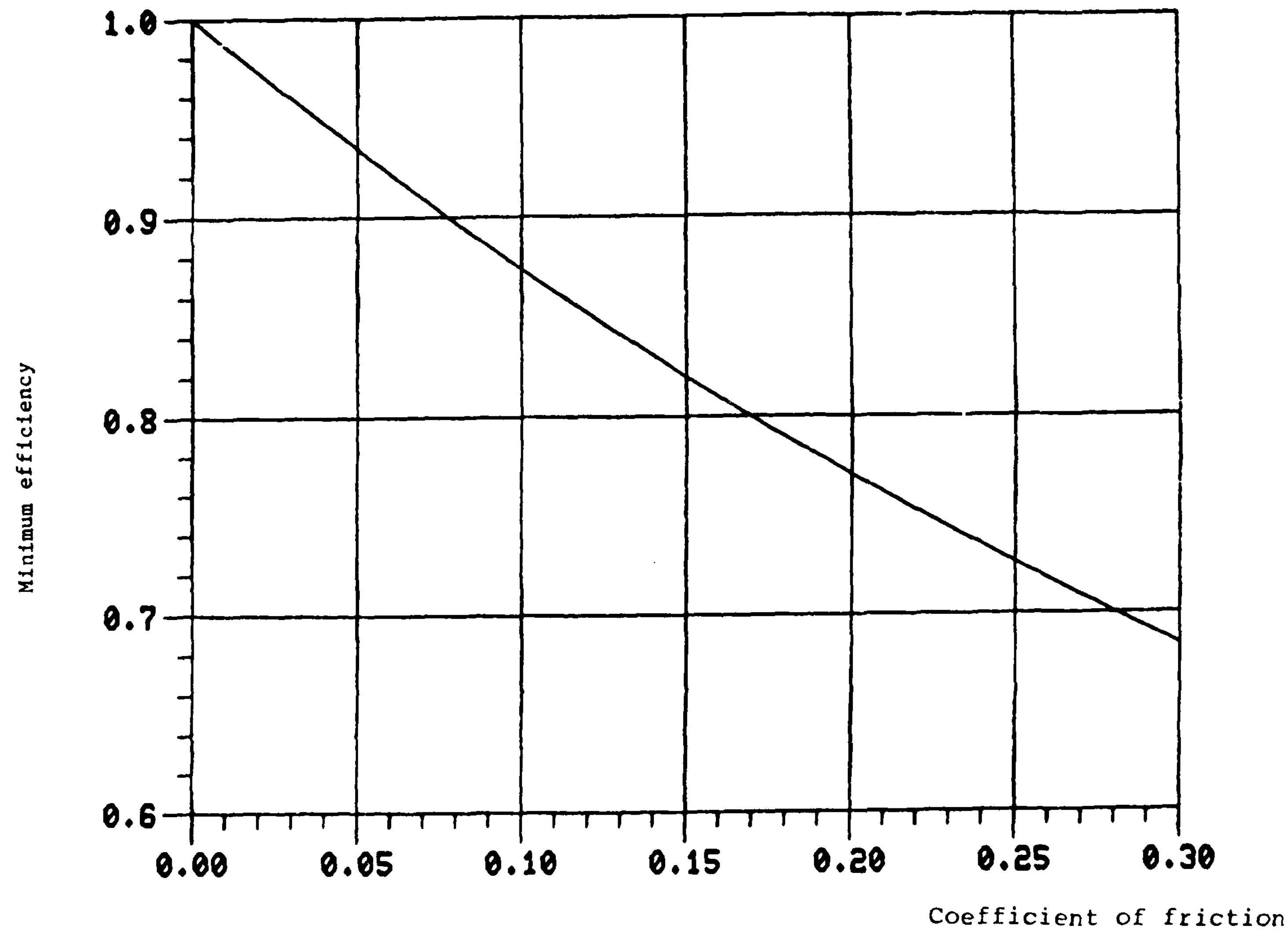


Figure 11. Minimum efficiency of drive arm pinion gear and main shaft gear mesh of exoatmospheric drive assembly vs coefficient of friction

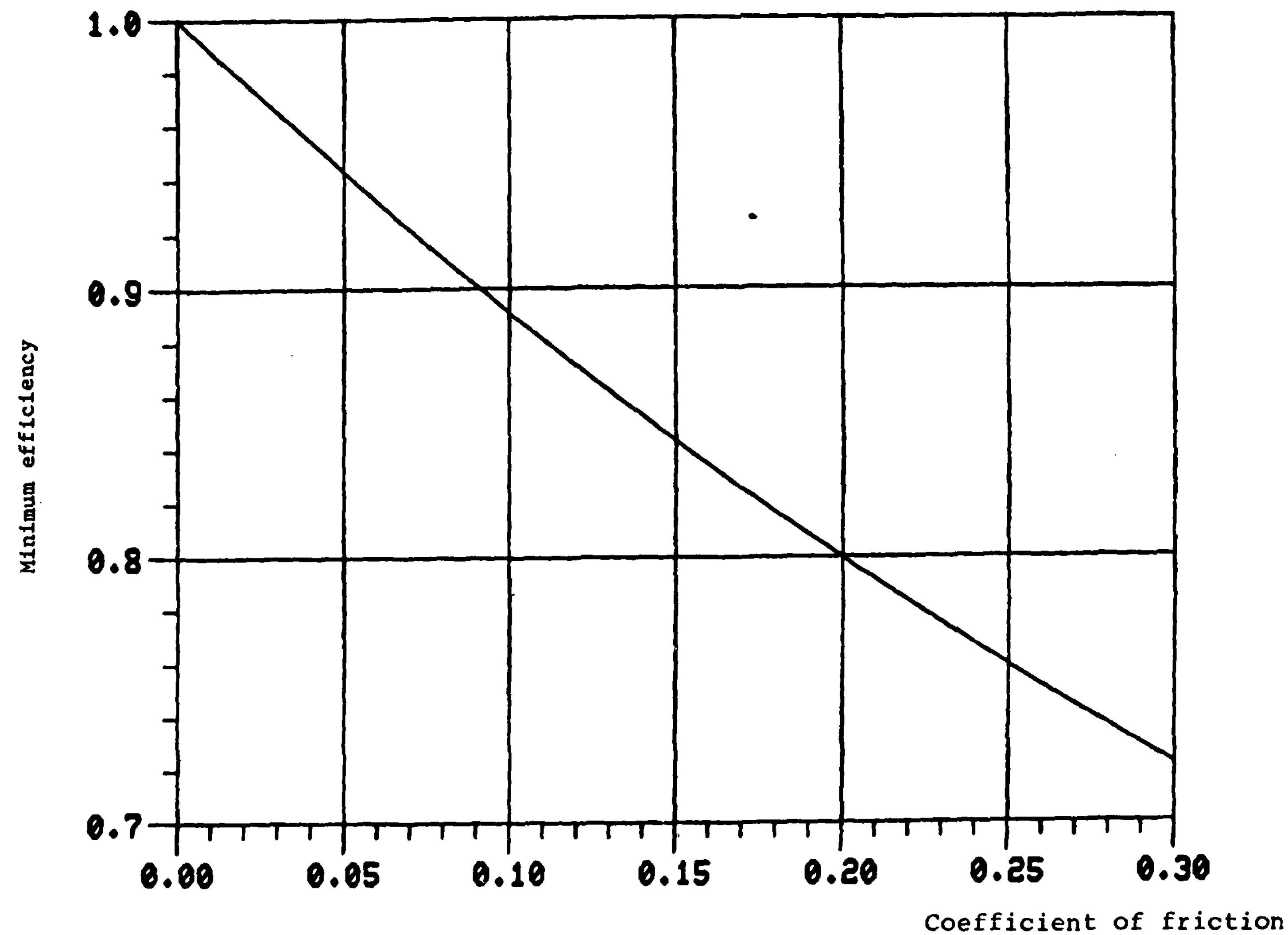


Figure 12. Minimum efficiency of motor pinion and differential gear mesh of valve drive assembly vs coefficient of friction

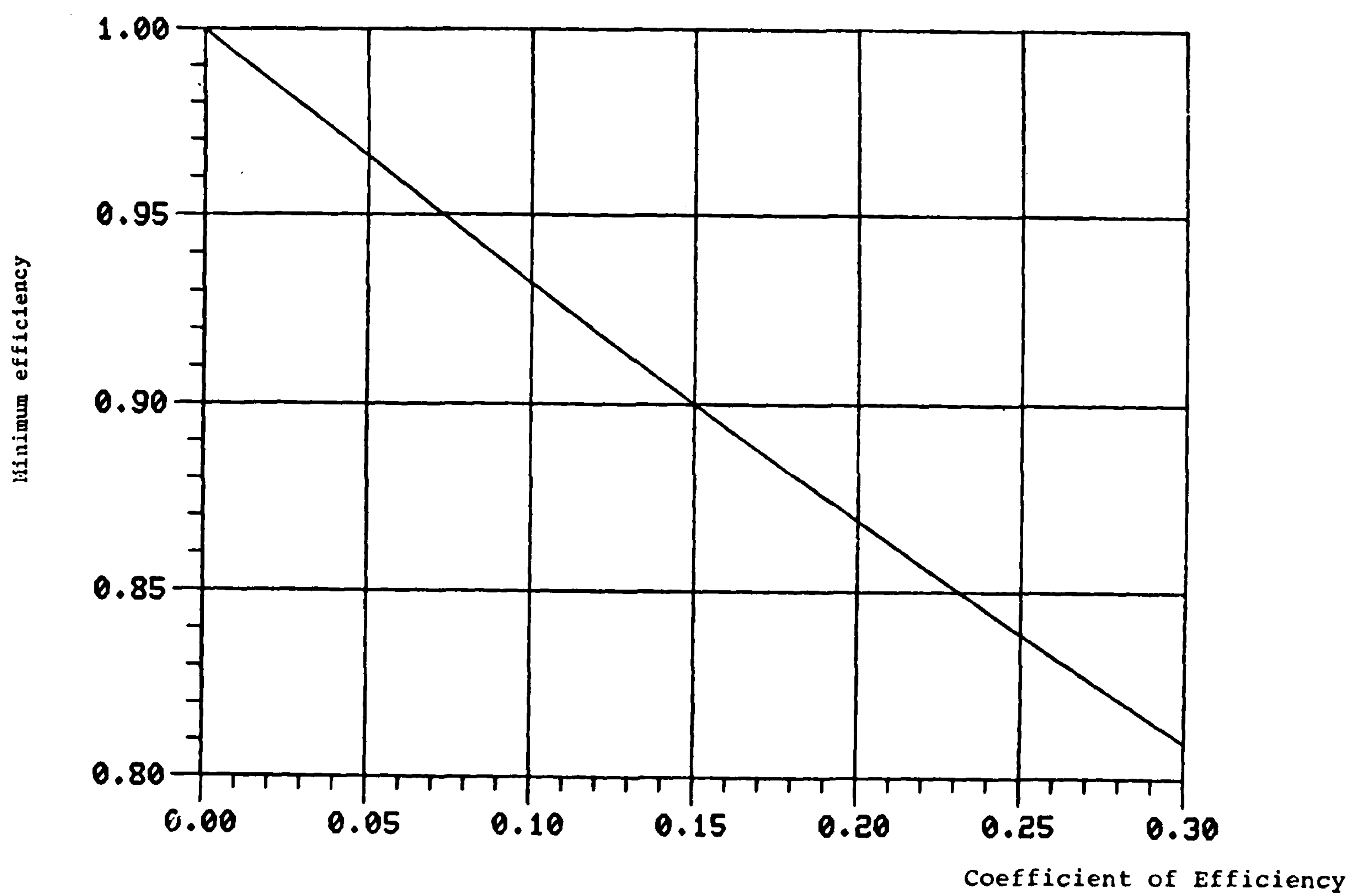


Figure 13. Minimum efficiency of differential gear and pinion mesh of valve drive assembly vs coefficient of friction

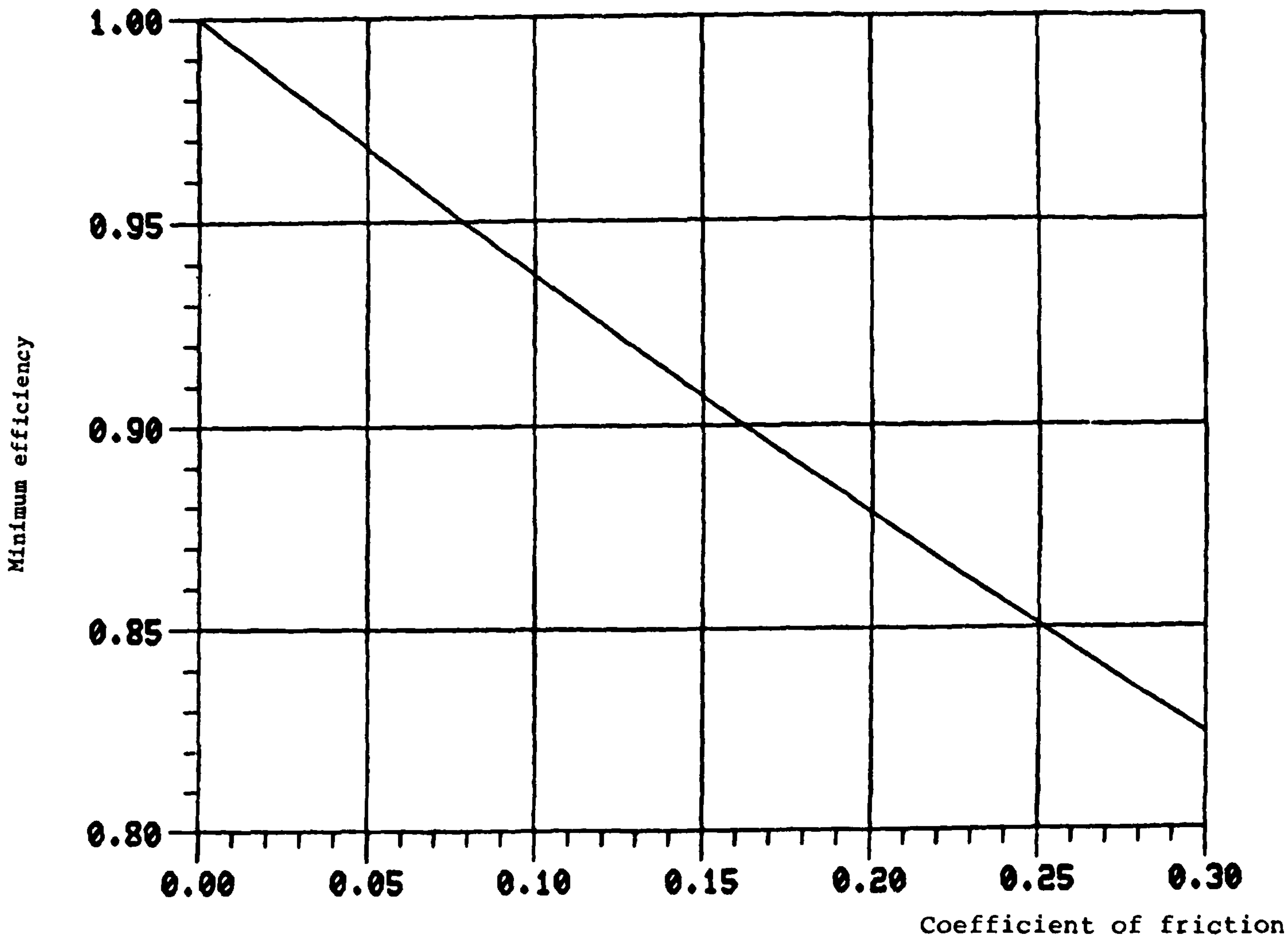


Figure 14. Minimum efficiency of differential pinion and pinion mesh of valve drive assembly vs coefficient of friction

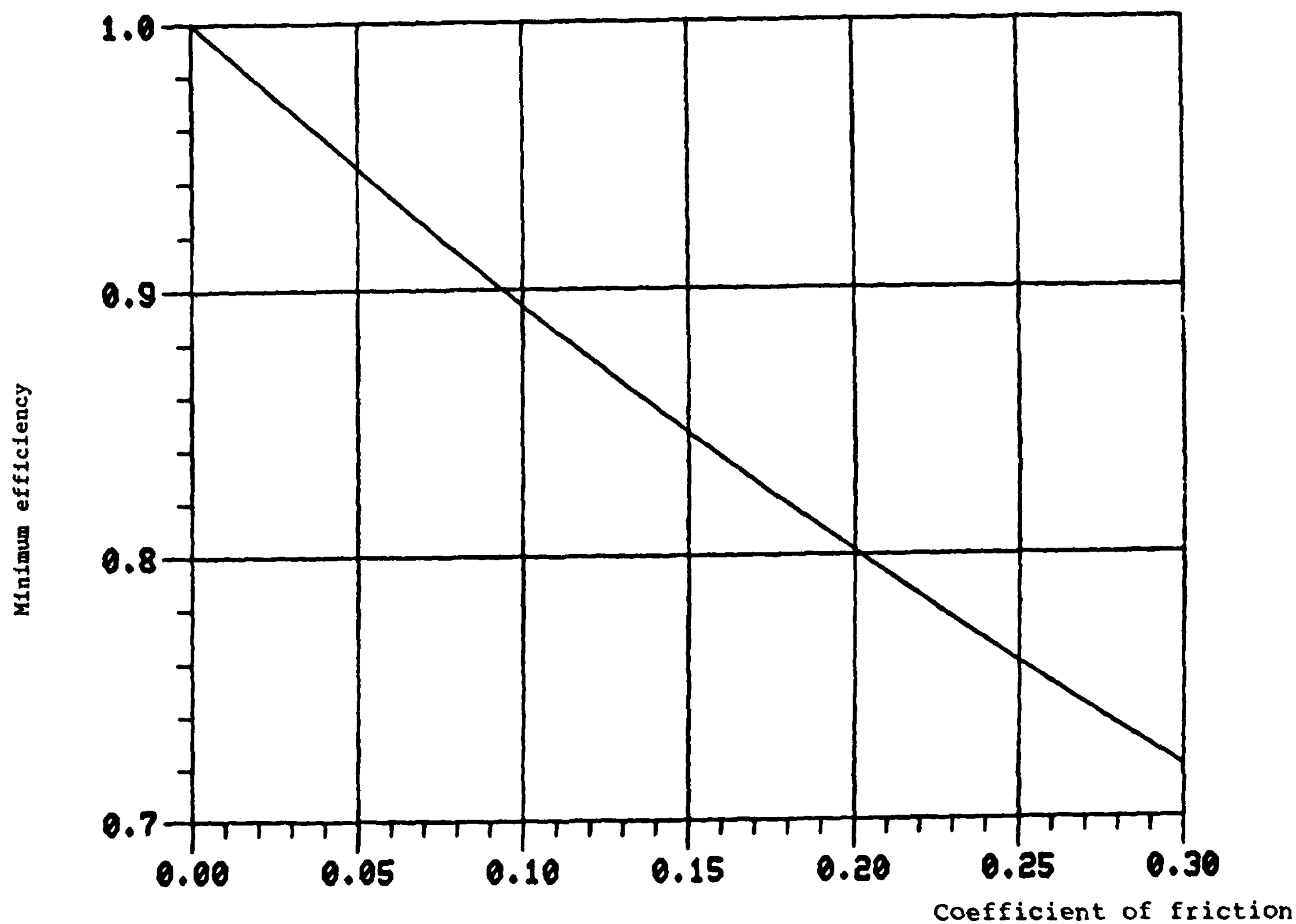


Table 15. Data for drive arm pinion gear and main shaft mesh of exoatmospheric drive assembly

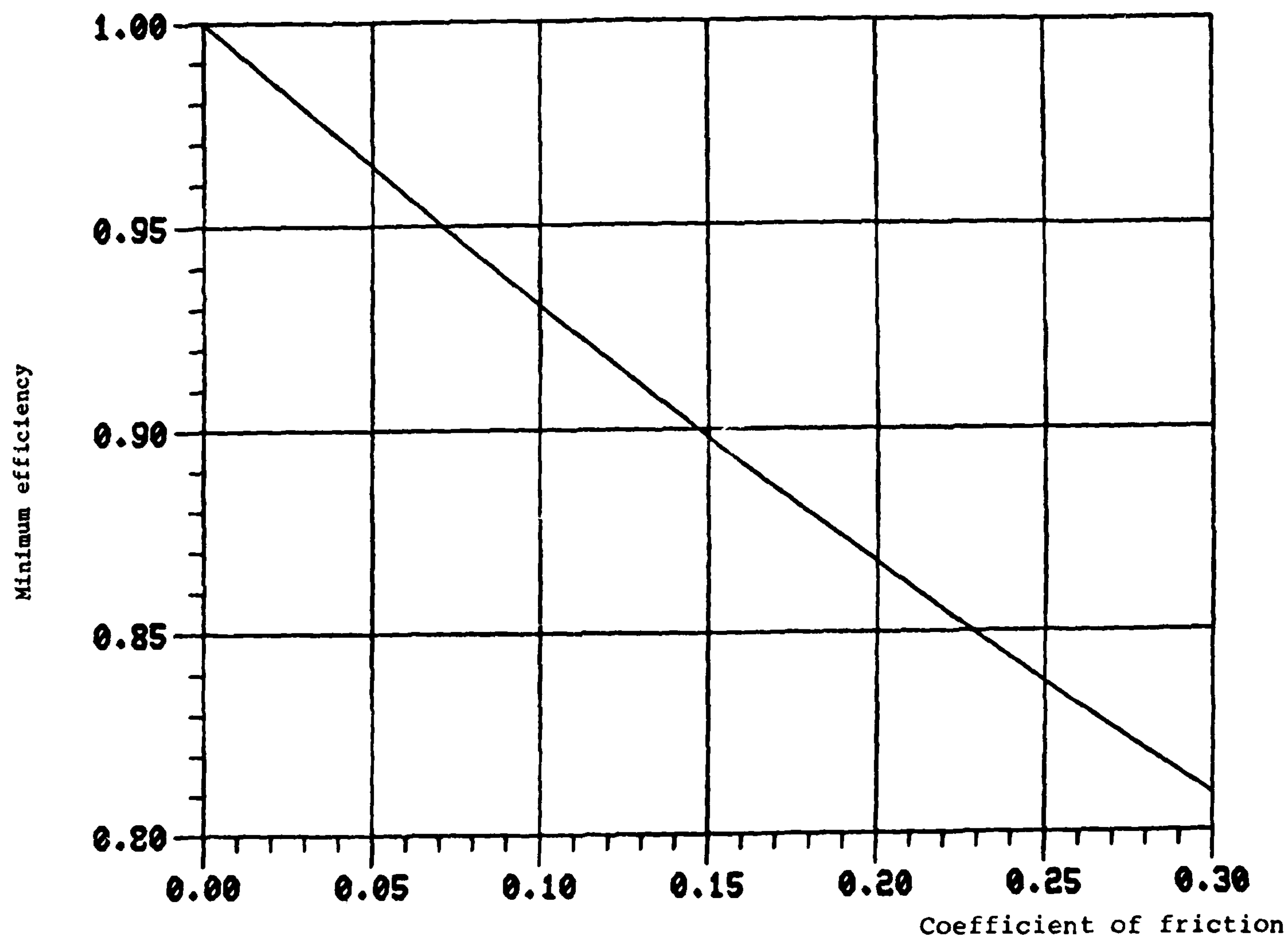


Figure 16. Minimum efficiency of valve drive gear and idler mesh of valve drive assembly vs coefficient of friction

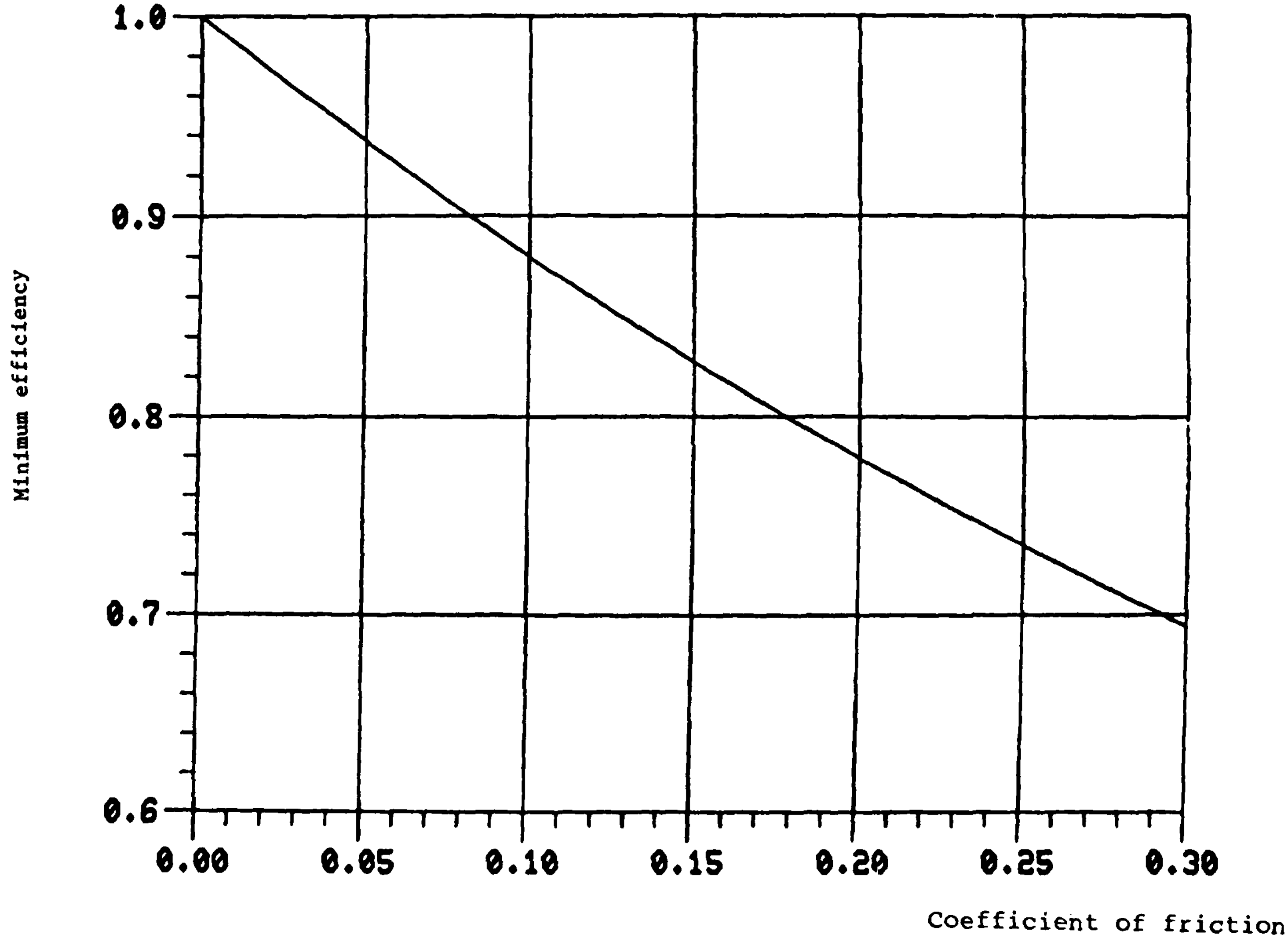


Figure 17. Minimum efficiency of output gear and valve drive pinion mesh of valve drive assembly vs coefficient of friction

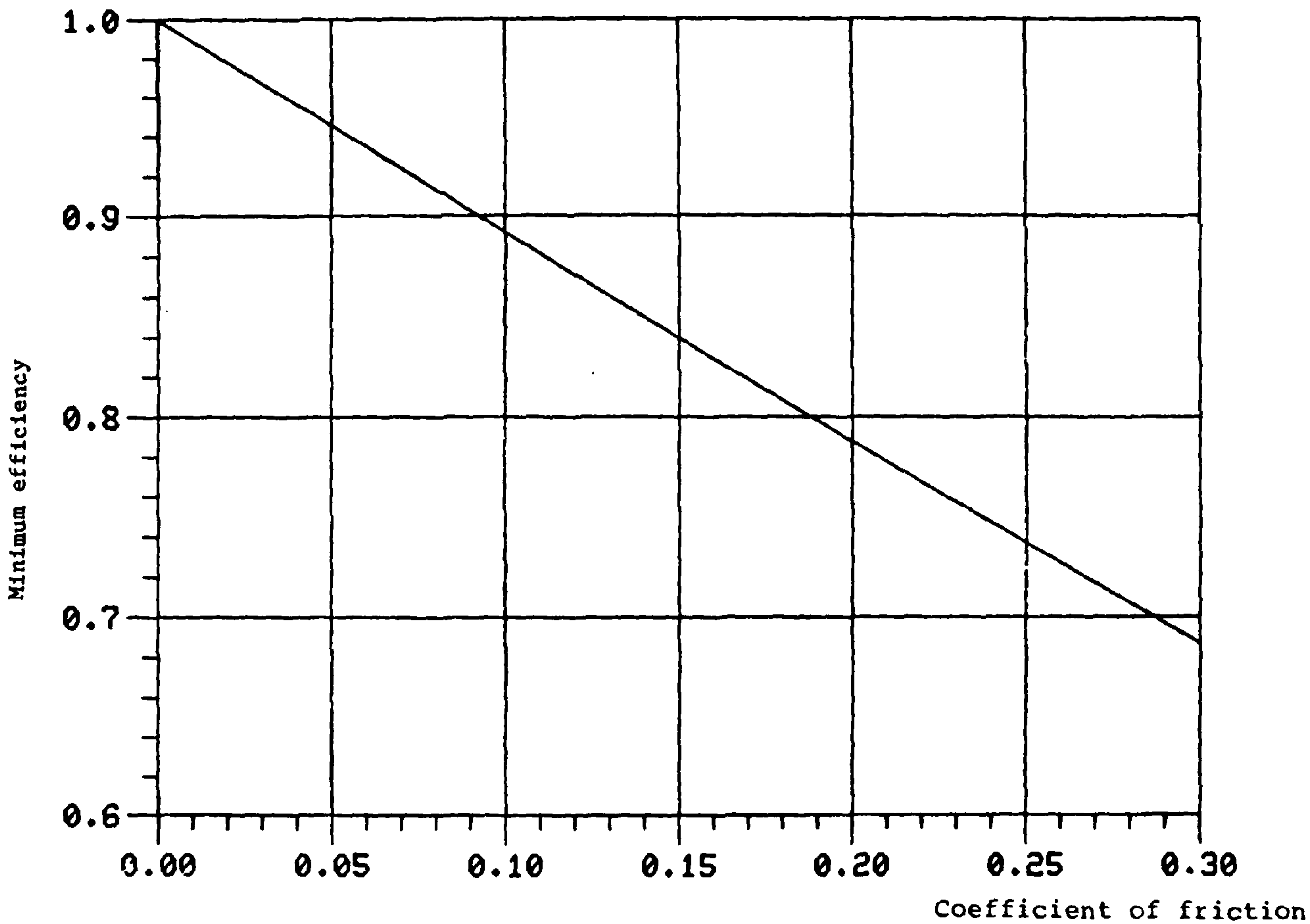


Figure 18. Minimum efficiency of valve gear and output pinion mesh of valve drive assembly vs coefficient of friction

APPENDIX A

MODIFICATION OF CLOCK GEAR TOOTH FOR EFFICIENCY COMPUTATIONS

The solid tooth profile shown in figure A-1 represents the actual clock tooth shape as given in the engineering drawings. The computer program for determining the efficiency of a clock tooth gear mesh, as given in reference 2, does not take the rounded tip into account, but instead considers a constant curvature tooth as represented by the dotted line in figure A-1.

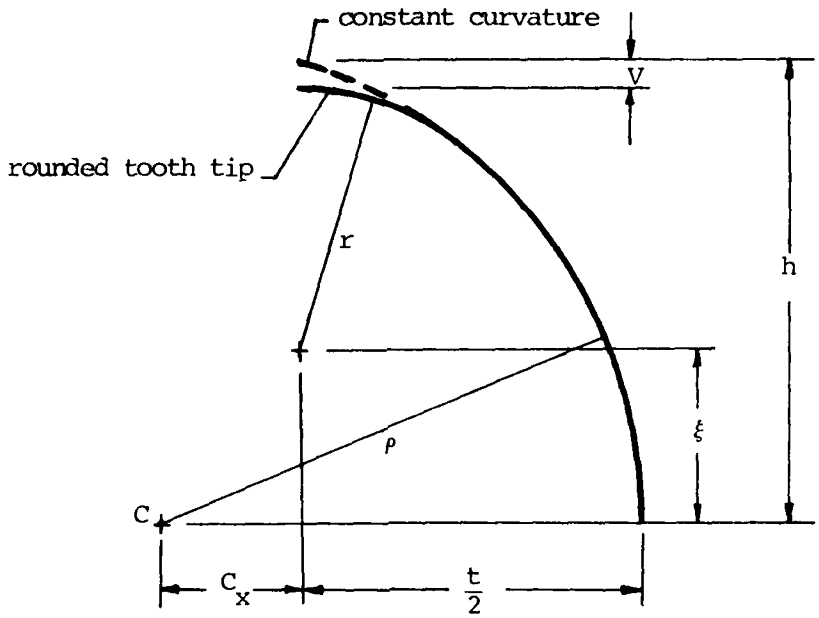


Figure A-1. Modification of clock tooth profile

To apply the computer program, the outside radius of the gear will be modified to include the distance V, which is the difference in the outside radius between the constant curvature tooth and the rounded tip tooth.

From the engineering drawing, the tooth radius ρ, the tip radius r and the tooth thickness t is given. Thus

$$C_x = \rho - \frac{t}{2} \tag{A1}$$

$$h = \sqrt{\rho^2 - C_x^2} \tag{A2}$$

In order to determine V, consider that the slopes of the circle representing the rounded tip and of the circle representing the flank of the tooth at the point of intersection must be identical. Using the center of curvature of the tooth flank as the origin of the coordinate system, the circle representing the tooth flank is given by

$$x^2 + y^2 = \rho^2 \tag{A3}$$

while the circle representing the rounded tip is given by

$$(x - C_x)^2 + (y - \xi)^2 = r^2 \tag{A4}$$

where

$$\xi = h - r - v \tag{A5}$$

To find the slopes of each of the circles, differentiate equations A3 and A4 implicitly and solve for y':

$$y' = \frac{-x}{y} \tag{A6}$$

$$y' = -\left(\frac{x - C_x}{y - \xi}\right) \tag{A7}$$

Equating these two expressions and using equation A3, one obtains for x and y:

$$x = \frac{\rho C_x}{\sqrt{C_x^2 + \xi^2}} \tag{A8}$$

$$y = \frac{\rho \xi}{\sqrt{C_x^2 + \xi^2}} \tag{A9}$$

Now substituting these equations into equation A1 and collecting like terms

$$\sqrt{C_x^2 + \xi^2} = \frac{\xi^2 + C_x^2 + \rho^2 - r^2}{2\rho} \tag{A10}$$

Squaring both sides of this equation and regrouping terms leads to

$$\xi^4 + 2\,(C_x^2 - \rho^2 - r^2)\,\xi^2 + [(C_x^2 + \rho^2 - r^2)^2 - 4\rho C_x^2] = 0 \tag{A11}$$

Using the quadratic formula to solve for ξ and then collecting like terms

$$\xi^2 = -c_x^2 + (\rho \pm r)^2 \tag{A12}$$

From figure A1 it can be seen that

$$c_x^2 + \xi^2 < \rho^2 \tag{A13}$$

Therefore, the negative sign holds in equation A12.

Solving for ξ

$$\xi = \sqrt{(\rho - r)^2 - c_x^2} \tag{A14}$$

Then from equation A5

$$v = h - r - \sqrt{(\rho - r)^2 - c_x^2} \tag{A15}$$

The outside radius r_{om} of the modified tooth may now be written

$$r_{om} = r_o + v \tag{A16}$$

where r_o is the outside radius of the actual tooth.

Since the computer program requires the distance from the gear pivot to the center of curvature of the circular arc of the tooth, this may now be given as

$$a = \sqrt{(r_{om} - h)^2 + c_x^2} \tag{A17}$$

APPENDIX B

DERIVATION OF EFFICIENCY EXPRESSIONS FOR SINGLE INVOLUTE GEAR MESH

WITH CONTACT RATIO GREATER THAN ONE

Free body diagrams of the gear and pinion of a single involute mesh where the pinion is driven by a counterclockwise input movement M_{In} are given in figure B-1. In the position shown, there are two pairs of teeth in contact located at point C_1 and C_2. In the analysis it will be assumed that the pairs of teeth in contact share the transmitted load equally.

a. Unit Vectors

The unit vector from point O_n to point T' is given by

$$\overline{n}_{\theta} = \sin\theta\ \overline{i} + \cos\theta\ \overline{j} \tag{B1}$$

while the unit vector from point L to point L' is given by

$$\overline{n}_{\theta T} = -\cos\theta\ \overline{i} + \sin\theta\ \overline{j} \tag{B2}$$

where θ represents the actual pressure angle.

b. Nomenclature and Signum Convention

F_{xN}, F_{yN}	=	x and y components of normal force acting on gear pivot
μF_{xN}, μF_{yN}	=	friction force components acting on gear pivot
F_{xn}, F_{yn}	=	x and y components of normal force acting on pinion pivot
μF_{xn}, μF_{yn}	=	friction force components acting on pinion pivot
μ	=	coefficient of friction
F_C	=	normal force acting between gear and pinion at both points of contact
μF_C	=	tooth contact friction force
ρ_N, ρ_n	=	gear and pinion pivot radii
R_b, r_b	=	gear and pinion base circle radii
d	=	length of the line of action between base circle tangent points T and T'
a_1, a_2	=	distances of the contact points C_1 and C_2 from point T' along the line of action

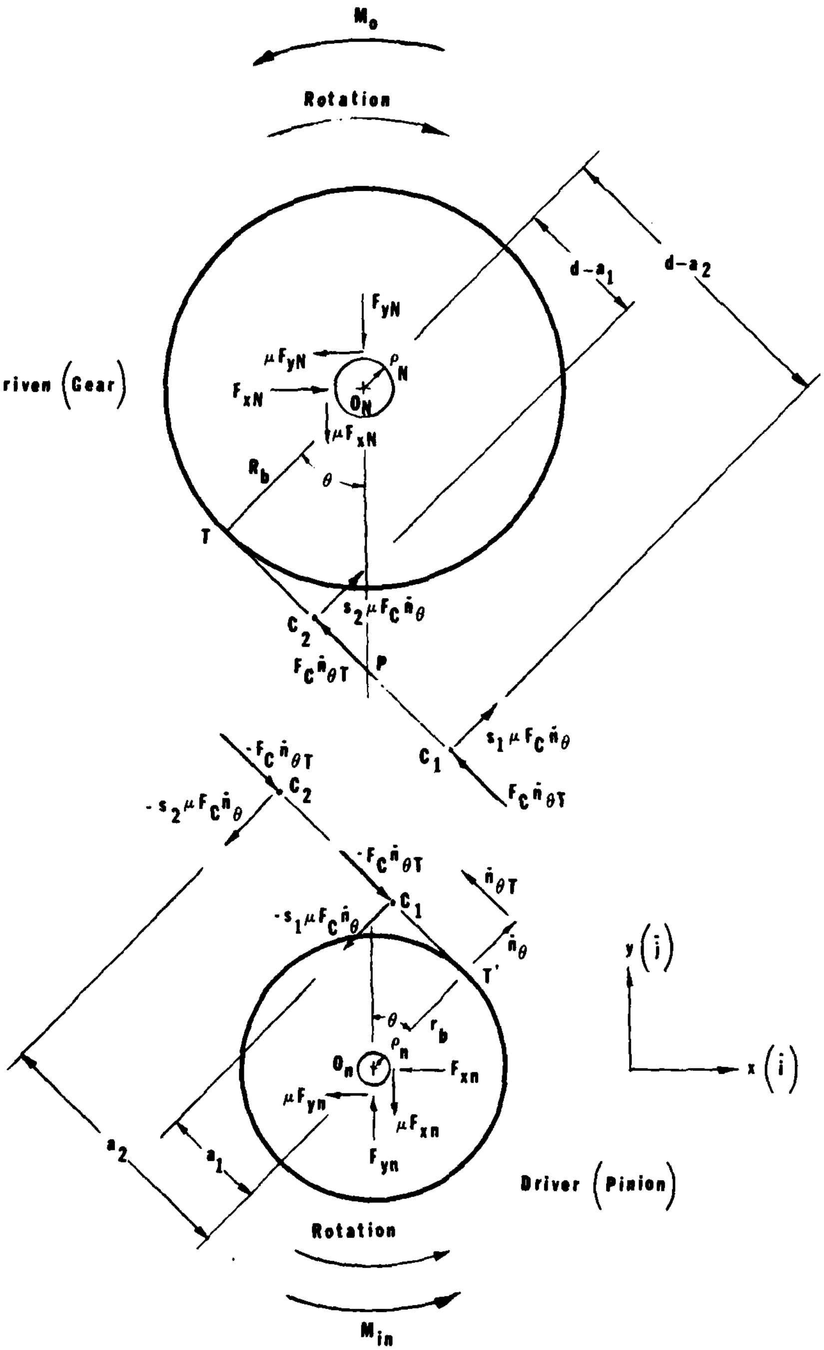


Figure B-1. Free body diagrams of gear and pinion

$$
\begin{aligned}
s_1 &= +1 \text{ for } a_1 < T'P \text{ or } \alpha_1 < \tan\theta \\
s_2 &= +1 \text{ for } a_2 < T'P \text{ or } \alpha_2 < \tan\theta \\
s_1 &= -1 \text{ for } a_1 > T'P \text{ or } \alpha_1 > \tan\theta \\
s_2 &= -1 \text{ for } a_2 > T'P \text{ or } \alpha_2 > \tan\theta \\
s_1 &= 0 \text{ for } a_1 = T'P \text{ or } \alpha_1 = \tan\theta \\
s_2 &= 0 \text{ for } a_2 = T'P \text{ or } \alpha_2 = \tan\theta
\end{aligned}
\tag{B3}
$$

c. Pinion Equilibrium Equations

From the free body diagram of the pinion, force equilibrium can be expressed by

$$-2F_C\bar{n}_{\theta T} - s_1\mu F_C\bar{n}_\theta - s_2\mu F_C\bar{n}_\theta - F_{xn}\bar{i} - \mu F_{xn}\bar{j} + F_{yn}\bar{j} - \mu F_{yn}\bar{i} = 0 \tag{B4}$$

Moment equilibrium can be expressed as

$$
\begin{aligned}
&M_{in}\,\bar{k} - \rho_n\mu \; F_x^2 + F_{yn}^2 \; \bar{k} + r_b\bar{n}_\theta \times (-)\; 2F_C\bar{n}_{\theta T} \\
&+ (r_b\bar{n}_\theta + a_1\bar{n}_{\theta T}) \times (-)\; s_1\mu F_C\bar{n}_\theta + (r_b\bar{n}_\theta + a_2\bar{n}_{\theta T}) \times (-)\; s_2\mu F_C\bar{n}_\theta = 0
\end{aligned}
\tag{B5}
$$

where M_{in} is the input moment.

Substituting equations B1 and B2 into equation B4 and expressing the result in scalar form, one obtains

$$2F_C\cos\theta - \mu(s_1 + s_2)\, F_C\sin\theta - F_{xn} - \mu F_{yn} = 0 \tag{B6}$$

$$-2F_C\sin\theta - \mu(s_1 + s_2)\, F_C\cos\theta + F_{yn} - \mu F_{xn} = 0 \tag{B7}$$

Similarly, for equation B5

$$M_{in} - \rho_n\mu \;\; F_{xn}^2 + F_{yn}^2 - 2r_bF_C + \mu(s_1a_1 + s_2a_2)\, F_C = 0 \tag{B8}$$

Now solve equations B6 and B7 simultaneously to find F_{xn} and F_{yn}. Solving equation B6 for F_{xn}:

$$F_{xn} = 2F_C \cos\theta - \mu(s_1 + s_2) F_C \sin\theta - \mu F_{yn} \tag{B9}$$

Substitute this equation into equation B7, and solve for F_{yn}:

$$F_{yn} = F_C \left\{\frac{[2 - \mu^2 (s_1 + s_2)] \sin\theta + \mu [(s_1 + s_2) + 2] \cos\theta}{1 + \mu^2}\right\} \tag{B10}$$

Now substituting equation B10 into equation B6 and solving for F_{xn}:

$$F_{xn} = F_C \left\{\frac{[2 - \mu^2(s_1 + s_2)] \cos\theta - \mu [2 + (s_1 + s_2)] \sin\theta}{1 + \mu^2}\right\} \tag{B11}$$

Using equations B10 and B11 in equation B8 leads to

$$M_{in} - \frac{\mu\rho_n F_C}{1 + \mu^2} \sqrt{[2 - \mu^2(s_1 + s_2)]^2 + \mu^2 [(s_1 + s_2) + 2]} \tag{B12}$$

$$- 2 r_b F_C + \mu(s_1 a_1 + s_2 a_2) F_C = 0$$

Expanding the term under the square root sign, and solving the resulting equation for F_C, one obtains

$$F_C = \frac{M_{in}}{2 r_b - \mu(s_1 a_1 + s_2 a_2) + \dfrac{\mu\rho_N}{1 + \mu^2} \sqrt{4 + \mu^2 [4 + (s_1 + s_2)^2] + \mu^4 (s_1 + s_2)^2}} \tag{B13}$$

d. Gear Equilibrium Equations

Force equilibrium of the gear is given by

$$2F_C \bar{n}_{\theta T} + (s_1 + s_2)\mu F_C \bar{n}_\theta + F_{xN}\bar{i} - \mu F_{xN}\bar{j} - F_{yN}\bar{j} - \mu F_{yN}\bar{i} = 0 \tag{B14}$$

while moment equilibrium is given by

$$M_o\bar{k} + \rho_N\mu \quad F_{xN}^2 + F_{yN}^2 \; \bar{k} + [-R_b\bar{n}_\theta - (d - a_1)\bar{n}_{\theta T}] \times (F_C\bar{n}_{\theta T} + \mu s_1 F_C\bar{n}_\theta)$$

$$+ [R_b\bar{n}_\theta - (d - a_2)\bar{n}_{\theta T}] \times (F_C\bar{n}_{\theta T} + \mu s_2 F_C\bar{n}_\theta) = 0 \qquad \text{(B15)}$$

where M_o is the equilibrant moment.

Proceeding in a manner similar to that of the preceding section, the contact force becomes

$$F_C = \frac{M_o}{2R_b - \mu[s_1(d-a_1) + s_2(d-a_2)] - \dfrac{\mu\rho_N}{1 + \mu^2}\sqrt{4+\mu^2[4 + (s_1+s_2)^2]+\mu^4(s_1+s_2)^2}} \qquad \text{(B16)}$$

e. Moment Input-Output Relationship

The equilibrant moment, M_o, may be expressed as a function of the input moment, M_{in}, after equations B13 and B16 have been set equal to each other. Thus

$$M_o = M_{in}\frac{2R_b - \mu[s_1(d-a_1) + s_2(d-a_2)] - \dfrac{\mu\rho_N}{1 + \mu^2}\sqrt{4+\mu^2[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}}{2r_b - \mu(s_1a_1+s_2a_2) + \dfrac{\mu\rho_N}{1 + \mu^2}\sqrt{4+\mu^2[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}} \qquad \text{(B17)}$$

The input-output relationship may also be expressed as

$$M_o = M_{in}\frac{R_b}{r_b}\varepsilon_p \qquad \text{(B18)}$$

where

$$\varepsilon_p = \frac{2 - \dfrac{\mu[s_1(d-a_1) + s_2(d-a_2)]}{R_b} - \dfrac{\mu\rho_N}{R_b(1 - \mu^2)}\sqrt{4+\mu^2[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}}{2 - \dfrac{\mu(s_1a_1+s_2a_2)}{r_b} + \dfrac{\mu\rho_n}{r_b(1 + \mu^2)}\sqrt{4+\mu^2[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}} \qquad \text{(B19)}$$

which represents the point efficiency of a single step-up involute mesh having two pairs of teeth in contact simultaneously with the pinion being the driver. If the gear is the driving element, the efficiency expression may be obtained directly from equation B19 by interchanging the gear and pinion parameters. Thus, when the gear is the driver

$$\varepsilon_p = \frac{2 - \dfrac{\mu[s_1(d-a_1) + s_2(d-a_2)]}{r_b} - \dfrac{\mu\rho_n}{r_b(1+\mu^2)}\sqrt{4+\mu^2[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}}{2 - \dfrac{\mu(s_1a_1+s_2a_2)}{r_b} + \dfrac{\mu\rho_N}{R_b(1+\mu^2)}\sqrt{4+\mu^2[4+(s_1+s_2)^2] + \mu^4(s_1+s_2)^2}} \qquad (B20)$$

APPENDIX C

COMPUTER PROGRAM ISAD1

```
      PROGRAM ISAD1(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
C
C POINT AND CYCLE EFFICIENCIES FOR SINGLE PASS INVOLUTE GEAR MESH
C (PINION DRIVES GEAR)
C
      REAL MU,MTOT,NP
      PI=3.14159
      Z=PI/180.
C
C   READ DATA
C
    1 READ(5,2)PSUBD,NP,CAPRP,RP,CAPRO,RO,THETAD,RHOCAPN,RHON,CD,HOBTIPR
     1,K,RASFACT
    2 FORMAT(8F10.4/3F10.4,6X,I4,F10.4)
      IF(EOF(5).NE.0)STOP
C
C   WRITE DATA
C
      WRITE(6,3)PSUBD,NP,CAPRP,RP,CAPRO,RO,THETAD,RHOCAPN,RHON,CD,
     1HOBTIPR,K,RASFACT
    3 FORMAT(*1*///6X,*DIAMETRAL PITCH (PSUBD) =*,F6.1/*0*,5X,*PINION NU
     1MBER OF TEETH (NP) =*,F4.0/*0*,5X,*STANDARD GEAR PITCH RADIUS (CAP
     2RP) =*,F7.5,3X,*STANDARD PINION PITCH RADIUS (RP) =*,F7.5/*0*,5X,
     3*GEAR OUTSIDE RADIUS (CAPRO) =*,F7.5,3X,*PINION OUTSIDE RADIUS (RO
     4) =*,F7.5/*0*,5X,*PRESSURE ANGLE IN DEGREES (THETAD) =*,F6.2/*0*,
     55X,*GEAR PIVOT RADIUS (RHOCAPN) =*,F5.3,3X,*PINION PIVOT RADIUS (R
     6HON) =*,F5.3/*0*,5X,*OPERATING CENTER DISTANCE (CD) =*,F6.3/*0*,5X
     7,*GEAR CUTTER TIP RADIUS (HOBTIPR) =*,F7.5/*0*,5X,*RANGE DIVISOR (
     8K) =*,I4/*0*,5X,*SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASF
     9ACT) =*,F5.3///)
C
C   CONVERT THETA TO RADIANS
C
      THETA=THETAD*Z
C
C   DETERMINE BASE RADIUS OF GEAR AND PINION AND BASE PITCH
C
      CAPRB=CAPRP*COS(THETA)
      RB=RP*COS(THETA)
      PB=2.*PI*RB/NP
      WRITE(6,4)CAPRB,RB,PB
    4 FORMAT(6X,*BASE RADIUS OF GEAR (CAPRB) =*,F6.4/*0*,5X,*BASE RADIUS
     1 OF PINION (RB) =*,F6.4/*0*,5X,*BASE PITCH =*,F6.4)
C
C   DETERMINE PINION INNER FORM RADIUS
C
      CALL INNERF(NP,PSUBD,THETA,HOBTIPR,RB,B,RF)
      WRITE(6,5)RF
    5 FORMAT(*0*,5X,*PINION INNER FORM RADIUS (RF) =*,F6.4)
C
C   DETERMINE IF PINION IS UNDERCUT
C
      U=1.-B/RP
      IF(U.GT.RASFACT)GO TO 7
      WRITE(6,6)
    6 FORMAT(*0*,5X,*THE PINION IS UNDERCUT*)
      GO TO 9
```

```
      7 WRITE(6,8)
      8 FORMAT(*0*,5X,*THE PINION IS NOT UNDERCUT*)
C
C    DETERMINE ACTUAL PITCH RADII AND PRESSURE ANGLE
C
      9 C=CAPRP+RP
        CAPRP=CAPRP*CD/C
        RP=RP*CD/C
        THETA=ACOS((C*COS(THETA)/CD))
        THETAD=THETA/Z
        WRITE(6,10)CAPRP,RP,THETAD
     10 FORMAT(*0*,5X,*ACTUAL GEAR PITCH RADIUS (CAPRP) =*,F7.5/*0*,5X,
       1*ACTUAL PINION PITCH RADIUS (RP) =*,F7.5/*0*,5X,*ACTUAL PRESSURE A
       2NGLE IN DEGREES (THETAD) =*,F6.2//)
C
C    DETERMINE IF PZP (PZPRIME) IS GREATER THAN PZ
C
        PZP=RB*TAN(THETA)-SQRT(RF*RF-RB*RB)
        PZ=SQRT(CAPRO*CAPRO-CAPRB*CAPRB)-CAPRB*TAN(THETA)
        IF(PZP.GT.PZ)GO TO 11
C
C    CALCULATE CONTACT RATIO, INITIAL PINION ANGLE AND ANGLE
C    CORRESPONDING TO END OF DUAL CONTACT
C
        CR=(SQRT(RO*RO-RB*RB)-SQRT(RF*RF-RB*RB))/PB
        ALIN=SQRT(RF*RF-RB*RB)/RB
        DELAL=(SQRT(RO*RO-RB*RB)-SQRT(RF*RF*RB*RB)-PB)/RB
        GO TO 12
     11 CR=(SQRT(CAPRO*CAPRO-CAPRB*CAPRB)+SQRT(RO*RO-RB*RB)-(CAPRB+RB)*
       1TAN(THETA))/PB
        ALIN=((CAPRB+RB)*TAN(THETA)-SQRT(CAPRO*CAPRO-CAPRB*CAPRB))/RB
        DELAL=(SQRT(CAPRO*CAPRO-CAPRB*CAPRB)+SQRT(RO*RO-RB*RB)-(CAPRB+RB)
       1*TAN(THETA)-PB)/RB
     12 ALFIN=ALIN+DELAL
C
C    CONVERT AL1IN AND AL1FIN TO DEGREES
C
        ALIND=ALIN/Z
        ALFIND=ALFIN/Z
C
C    PRINT CONTACT RATIO AND INITIAL AND FINAL PINION ANGLES
C
        WRITE(6,13)CR,ALIND,ALFIND
     13 FORMAT(*0*,5X,*CONTACT RATIO (CR) =*,F4.2/*0*,5X,*INITIAL GEAR ANG
       1LE (ALIN) =*,F7.3/*0*,5X,*ANGLE CORRESPONDING TO END OF DUAL CONTA
       2CT (ALFIN) =*,F7.3///)
C
C    EFFICIENCY COMPUTATIONS
C
        DO 21 J = 1,13
        MU = .000 + (J-1)*.025
        WRITE(6,14)MU
     14 FORMAT(5X,*COEFFICIENT (MU) =*,F5.3/)
        DELALPH=(PB/RB)/K
        MTOT=0.
        D=(CAPRB+RB)*TAN(THETA)
        DO 19 I=1,K
```

```
          ALPHA1=ALIN+(I-1)*DELALPH
          ALPHA1D=ALPHA1/Z
          A1=RB*ALPHA1
          IF(ALPHA1.LT.TAN(THETA))S1=1.
          IF(ALPHA1.EQ.TAN(THETA))S1=0.
          IF(ALPHA1.GT.TAN(THETA))S1=-1.
          IF(ALPHA1.GT.ALFIN)GO TO 16
          ALPHA2=ALPHA1+PB/RB
          A2=A1+PB
          IF(ALPHA2.LT.TAN(THETA))S2=1.
          IF(ALPHA2.EQ.TAN(THETA))S2=0.
          IF(ALPHA2.GT.TAN(THETA))S2=-1.
          AA=2.-MU/CAPRB*(S1*(D-A1)+S2*(D-A2))-MU*RHOCAPN/(CAPRB*(1.+MU*MU))
         1*SQRT(4.+MU*MU*(4.+(S1+S2)*(S1+S2))+MU*MU*MU*MU*(S1+S2)*(S1+S2))
          BB=2.-MU/RB*(S1*A1+S2*A2)+MU*RHON/(RB*(1.+MU*MU))*SQRT(4.
         1+MU*MU*(4.+(S1+S2)*(S1+S2))+MU*MU*MU*MU*(S1+S2)*(S1+S2))
          POINTEF=AA/BB
C
C   PRINT PINION ANGLE (ALPHA1D - DEGREES), SIGN FOR FRICTION FORCE AT
C   FIRST CONTACT POINT (S1), SIGN FOR FRICTION FORCE AT SECOND CONTACT
C   POINT (S2) AND POINT EFFICIENCY (POINTEF)
C
          WRITE(6,15)ALPHA1D,S1,S2,POINTEF
       15 FORMAT(5X,*ALPHA1D =*,F6.2,3X,*S1 =*,F4.1,3X,*S2 =*,F4.1,3X,*POINT
         1EF =*,F6.4)
          GO TO 18
       16 AA=1.-MU*(RHOCAPN+S1*(D-A1))/CAPRB
          BB=1.+MU*(RHON-S1*A1)/RB
          POINTEF=AA/BB
C
C   PRINT PINION ANGLE (ALPHA1D - DEGREES), SIGN FOR FRICTION FORCE AT
C   CONTACT POINT (S1) AND POINT EFFICIENCY (POINTEF)
C
          WRITE(6,17)ALPHA1D,S1,POINTEF
       17 FORMAT(5X,*ALPHA1D =*,F6.2,3X,*S1 =*,F4.1,14X,*POINTEF =*,F6.4)
       18 MTOT=MTOT+POINTEF
       19 CONTINUE
          CYCLEFF=DELALPH*MTOT/(PB/RB)
C
C   PRINT CYCLE EFFICIENCY (CYCLEFF)
C
          WRITE(6,20)CYCLEFF
       20 FORMAT(//5X,*CYCLEFF =*,F6.4/////)
       21 CONTINUE
          GO TO 1
          END
```

```
      SUBROUTINE INNERF(NP,PSUBD,THETA,HOBTIPR,RB,B,RF)
C
C THIS SUBROUTINE COMPUTES THE INNER FORM RADIUS OF A PINION
C
C
C   COMPUTE CUTTER ADDENDUM
C
      B=1.2/PSUBD+.002-HOBTIPR*(1.-SIN(THETA))
C
C   STARTING VALUE FOR ITERATION IS BASE CIRCLE RADIUS
C
      RBSTART=RB
C
C   BEGIN ITERATION TO FIND INNER FORM RADIUS USING EQUATION FROM
C   R. A. SHAFFER - AN ANALYSIS OF THE UNDERCUT PROBLEM IN INVOLUTE
C   SPUR GEARING
C
      N=1
      DO 2 I=1,10000
      RF=RBSTART+(I-1)*.000001
      IF(RF*COS(THETA)/(RB-B*COS(THETA)).GE.1.)GO TO 3
      N=I+1
      GO TO 2
    3 C1=SQRT((RF*COS(THETA)/(RB-B*COS(THETA)))**2-1.)
      C2=1.-B*COS(THETA)/RB
      C3=SQRT((RF/RB)**2-1.)
      TEST=ATAN(C1)-C2*C1-THETA+C2*TAN(THETA)-C3+ATAN(C3)
      IF(ABS(TEST).LT.1.E-8)GO TO 1
      IF(I.EQ.N)SIGN1=TEST/ABS(TEST)
      SIGN=TEST/ABS(TEST)
      IF(SIGN1.EQ.SIGN)GO TO 2
    1 RETURN
    2 CONTINUE
      RF=RB
      RETURN
      END
```

APPENDIX D

COMPUTER PROGRAM ISAD2

```
      PROGRAM ISAD2(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
C
C POINT AND CYCLE EFFICIENCIES FOR SINGLE PASS INVOLUTE GEAR MESH
C (GEAR DRIVES PINION)
C
      REAL MU,MTOT,NP
      PI=3.14159
      Z=PI/180.
C
C   READ DATA
C
    1 READ(5,2)PSUBD,NP,CAPRP,RP,CAPRO,RO,THETAD,RHOCAPN,RHON,CD,HOBTIPR
     1,K,RASFACT
    2 FORMAT(8F10.4/3F10.4,6X,I4,F10.4)
      IF(EOF(5).NE.0)STOP
C
C   WRITE DATA
C
      WRITE(6,3)PSUBD,NP,CAPRP,RP,CAPRO,RO,THETAD,RHOCAPN,RHON,CD,
     1HOBTIPR,K,RASFACT
    3 FORMAT(*1*///6X,*DIAMETRAL PITCH (PSUBD) =*,F6.1/*0*,5X,*PINION NU
     1MBER OF TEETH (NP) =*,F4.0/*0*,5X,*STANDARD GEAR PITCH RADIUS (CAP
     2RP) =*,F7.5,3X,*STANDARD PINION PITCH RADIUS (RP) =*,F7.5/*0*,5X,
     3*GEAR OUTSIDE RADIUS (CAPRO) =*,F7.5,3X,*PINION OUTSIDE RADIUS (RO
     4) =*,F7.5/*0*,5X,*PRESSURE ANGLE IN DEGREES (THETAD) =*,F6.2/*0*,
     55X,*GEAR PIVOT RADIUS (RHOCAPN) =*,F5.3,3X,*PINION PIVOT RADIUS (R
     6HON) =*,F5.3/*0*,5X,*OPERATING CENTER DISTANCE (CD) =*,F6.3/*0*,5X
     7,*GEAR CUTTER TIP RADIUS (HOBTIPR) =*,F7.5/*0*,5X,*RANGE DIVISOR (
     8K) =*,I4/*0*,5X,*SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASF
     9ACT) =*,F5.3///)
C
C   CONVERT THETA TO RADIANS
C
      THETA=THETAD*Z
C
C   DETERMINE BASE RADIUS OF GEAR AND PINION AND BASE PITCH
C
      CAPRB=CAPRP*COS(THETA)
      RB=RP*COS(THETA)
      PB=2.*PI*RB/NP
      WRITE(6,4)CAPRB,RB,PB
    4 FORMAT(6X,*BASE RADIUS OF GEAR (CAPRB) =*,F6.4/*0*,5X,*BASE RADIUS
     1 OF PINION (RB) =*,F6.4/*0*,5X,*BASE PITCH =*,F6.4)
C
C   DETERMINE PINION INNER FORM RADIUS
C
      CALL INNERF(NP,PSUBD,THETA,HOBTIPR,RB,B,RF)
      WRITE(6,5)RF
    5 FORMAT(*0*,5X,*PINION INNER FORM RADIUS (RF) =*,F6.4)
C
C   DETERMINE IF PINION IS UNDERCUT
C
      U=1.-B/RP
      IF(U.GT.RASFACT)GO TO 7
      WRITE(6,6)
    6 FORMAT(*0*,5X,*THE PINION IS UNDERCUT*)
      GO TO 9
```

```
    7 WRITE(6,8)
    8 FORMAT(*0*,5X,*THE PINION IS NOT UNDERCUT*)
C
C   DETERMINE ACTUAL PITCH RADII AND PRESSURE ANGLE
C
    9 C=CAPRP+RP
      CAPRP=CAPRP*CD/C
      RP=RP*CD/C
      THETA=ACOS((C*COS(THETA)/CD))
      THETAD=THETA/Z
      WRITE(6,10)CAPRP,RP,THETAD
   10 FORMAT(*0*,5X,*ACTUAL GEAR PITCH RADIUS (CAPRP) =*,F7.5/*0*,5X,
     1*ACTUAL PINION PITCH RADIUS (RP) =*,F7.5/*0*,5X,*ACTUAL PRESSURE A
     2NGLE IN DEGREES (THETAD) =*,F6.2//)
C
C   DETERMINE INITIAL GEAR ANGLE
C
      ALIN=((CAPRB+RB)*TAN(THETA)-SQRT(RO*RO-RB*RB))/CAPRB
C
C   DETERMINE IF PZP (PZPRIME) IS GREATER THAN PZ
C
      PZP=RB*TAN(THETA)-SQRT(RF*RF-RB*RB)
      PZ=SQRT(CAPRO*CAPRO-CAPRB*CAPRB)-CAPRB*TAN(THETA)
      IF(PZP.GT.PZ)GO TO 11
C
C   CALCULATE CONTACT RATIO AND ANGLE CORRESPONDING TO END
C   END OF DUAL CONTACT
C
      CR=(SQRT(RO*RO-RB*RB)-SQRT(RF*RF-RB*RB))/PB
      DELAL=(SQRT(RO*RO-RB*RB)-SQRT(RF*RF-RB*RB)-PB)/CAPRB
      GO TO 12
   11 CR=(SQRT(CAPRO*CAPRO-CAPRB*CAPRB)+SQRT(RO*RO-RB*RB)-(CAPRB+RB)*
     1TAN(THETA))/PB
      DELAL=(SQRT(CAPRO*CAPRO-CAPRB*CAPRB)+SQRT(RO*RO-RB*RB)-(CAPRB+RB)
     1*TAN(THETA)-PB)/CAPRB
   12 ALFIN=ALIN+DELAL
C
C   CONVERT AL1IN AND AL1FIN TO DEGREES
C
      ALIND=ALIN/Z
      ALFIND=ALFIN/Z
C
C   PRINT CONTACT RATIO AND INITIAL AND FINAL GEAR ANGLES
C
      WRITE(6,13)CR,ALIND,ALFIND
   13 FORMAT(*0*,5X,*CONTACT RATIO (CR) =*,F4.2/*0*,5X,*INITIAL GEAR ANG
     1LE (ALIN) =*,F7.3/*0*,5X,*ANGLE CORRESPONDING TO END OF DUAL CONTA
     2CT (ALFIN) =*,F7.3///)
C
C   EFFICIENCY COMPUTATIONS
C
      DO 21 J = 1,13
      MU = .000 + (J-1)*.025
      WRITE(6,14)MU
   14 FORMAT(5X,*COEFFICIENT (MU) =*,F5.3/)
      DELALPH=(PB/CAPRB)/K
      MTOT=0.
```

```
      D=(CAPRB+RB)*TAN(THETA)
      DO 19 I=1,K
      ALPHA1=ALIN+(I-1)*DELALPH
      ALPHA1D=ALPHA1/Z
      A1=CAPRB*ALPHA1
      IF(ALPHA1.LT.TAN(THETA))S1=1.
      IF(ALPHA1.EQ.TAN(THETA))S1=0.
      IF(ALPHA1.GT.TAN(THETA))S1=-1.
      IF(ALPHA1.GT.ALFIN)GO TO 16
      ALPHA2=ALPHA1+PB/CAPRB
      A2=A1+PB
      IF(ALPHA2.LT.TAN(THETA))S2=1.
      IF(ALPHA2.EQ.TAN(THETA))S2=0.
      IF(ALPHA2.GT.TAN(THETA))S2=-1.
      AA=2.-MU/RB*(S1*(D-A1)+S2*(D-A2))-MU*RHON/(RB*(1.+MU*MU))
     1*SQRT(4.+MU*MU*(4.+(S1+S2)*(S1+S2))+MU*MU*MU*MU*(S1+S2)*(S1+S2))
      BB=2.-MU/CAPRB*(S1*A1+S2*A2)+MU*RHOCAPN/(CAPRB*(1.+MU*MU))*SQRT(4.
     1+MU*MU*(4.+(S1+S2)*(S1+S2))+MU*MU*MU*MU*(S1+S2)*(S1+S2))
      POINTEF=AA/BB
C
C    PRINT PINION ANGLE (ALPHA1D - DEGREES), SIGN FOR FRICTION FORCE AT
C    FIRST CONTACT POINT (S1), SIGN FOR FRICTION FORCE AT SECOND CONTACT
C    POINT (S2) AND POINT EFFICIENCY (POINTEF)
C
      WRITE(6,15)ALPHA1D,S1,S2,POINTEF
   15 FORMAT(5X,*ALPHA1D =*,F6.2,3X,*S1 =*,F4.1,3X,*S2 =*,F4.1,3X,*POINT
     1EF =*,F6.4)
      GO TO 18
   16 AA=1.-MU*(RHON+S1*(D-A1))/RB
      BB=1.+MU*(RHOCAPN-S1*A1)/CAPRB
      POINTEF=AA/BB
C
C    PRINT PINION ANGLE (ALPHA1D - DEGREES), SIGN FOR FRICTION FORCE AT
C    CONTACT POINT (S1) AND POINT EFFICIENCY (POINTEF)
C
      WRITE(6,17)ALPHA1D,S1,POINTEF
   17 FORMAT(5X,*ALPHA1D =*,F6.2,3X,*S1 =*,F4.1,14X,*POINTEF =*,F6.4)
   18 MTOT=MTOT+POINTEF
   19 CONTINUE
      CYCLEFF=DELALPH*MTOT/(PB/CAPRB)
C
C    PRINT CYCLE EFFICIENCY (CYCLEFF)
C
      WRITE(6,20)CYCLEFF
   20 FORMAT(//5X,*CYCLEFF =*,F6.4/////)
   21 CONTINUE
      GO TO 1
      END
```

```
      SUBROUTINE INNERF(NP,PSUBD,THETA,HOBTIPR,RB,B,RF)
C
C THIS SUBROUTINE COMPUTES THE INNER FORM RADIUS OF A PINION
C
C
C   COMPUTE CUTTER ADDENDUM
C
      B=1.2/PSUBD+.002-HOBTIPR*(1.-SIN(THETA))
C
C   STARTING VALUE FOR ITERATION IS BASE CIRCLE RADIUS
C
      RBSTART=RB
C
C   BEGIN ITERATION TO FIND INNER FORM RADIUS USING EQUATION FROM
C   R. A. SHAFFER - AN ANALYSIS OF THE UNDERCUT PROBLEM IN INVOLUTE
C   SPUR GEARING
C
      N=1
      DO 2 I=1,10000
      RF=RBSTART+(I-1)*.000001
      IF(RF*COS(THETA)/(RB-B*COS(THETA)).GE.1.)GO TO 3
      N=I+1
      GO TO 2
    3 C1=SQRT((RF*COS(THETA)/(RB-B*COS(THETA)))**2-1.)
      C2=1.-B*COS(THETA)/RB
      C3=SQRT((RF/RB)**2-1.)
      TEST=ATAN(C1)-C2*C1-THETA+C2*TAN(THETA)-C3+ATAN(C3)
      IF(ABS(TEST).LT.1.E-8)GO TO 1
      IF(I.EQ.N)SIGN1=TEST/ABS(TEST)
      SIGN=TEST/ABS(TEST)
      IF(SIGN1.EQ.SIGN)GO TO 2
    1 RETURN
    2 CONTINUE
      RF=RB
      RETURN
      END
```

APPENDIX E

SAMPLE OUTPUT: DRIVE ARM PINION GEAR AND MAIN SHAFT MESH

OF EXOATMOSPHERIC DRIVE ASSEMBLY

```
DIAMETRAL PITCH (PSUBD) =  64.0

PINION NUMBER OF TEETH (NP) = 14.

STANDARD GEAR PITCH RADIUS (CAPRP) = .18750   STANDARD PINION PITCH RADIUS (RP) = .10938

GEAR OUTSIDE RADIUS (CAPRO) = .20313   PINION OUTSIDE RADIUS (RO) = .12500

PRESSURE ANGLE IN DEGREES (THETAD) = 20.00

GEAR PIVOT RADIUS (RHOCAPN) = .062   PINION PIVOT RADIUS (RHON) = .062

OPERATING CENTER DISTANCE (CD) =  .301

GEAR CUTTER TIP RADIUS (HOBTIPR) =0.00000

RANGE DIVISOR (K) =  25

SHAFFER FACTOR FOR DETERMINING UNDERCUTTING (RASFACT) = .883


BASE RADIUS OF GEAR (CAPRB) = .1762

BASE RADIUS OF PINION (RB) = .1028

BASE PITCH = .0461

PINION INNER FORM RADIUS (RF) = .1032

THE PINION IS UNDERCUT

ACTUAL GEAR PITCH RADIUS (CAPRP) = .18979

ACTUAL PINION PITCH RADIUS (RP) = .11071

ACTUAL PRESSURE ANGLE IN DEGREES (THETAD) = 21.82


CONTACT RATIO (CR) =1.31

INITIAL GEAR ANGLE (ALIN) =  5.919

ANGLE CORRESPONDING TO END OF DUAL CONTACT (ALFIN) = 13.946


COEFFICIENT (MU) =0.000

ALPHA1D =  5.92   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D =  6.95   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D =  7.98   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D =  9.01   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D = 10.03   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D = 11.06   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D = 12.09   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D = 13.12   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
```

```
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF =1.0000
ALPHA1D = 14.03   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 14.93   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 15.83   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 16.73   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 17.63   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 18.53   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 19.43   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 20.33   S1 = 1.0              POINTEF =1.0000
ALPHA1D = 21.23   S1 =-1.0              POINTEF =1.0000
ALPHA1D = 22.13   S1 =-1.0              POINTEF =1.0000
ALPHA1D = 23.03   S1 =-1.0              POINTEF =1.0000
ALPHA1D = 23.93   S1 =-1.0              POINTEF =1.0000
ALPHA1D = 24.83   S1 =-1.0              POINTEF =1.0000
ALPHA1D = 25.73   S1 =-1.0              POINTEF =1.0000

CYCLEFF =1.0000


COEFFICIENT (MU) = .025

ALPHA1D =  4.13   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D =  5.03   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D =  5.93   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D =  6.83   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D =  7.73   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D =  8.63   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D =  9.53   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D = 10.43   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF = .9637
ALPHA1D = 14.03   S1 = 1.0              POINTEF = .9655
ALPHA1D = 14.93   S1 = 1.0              POINTEF = .9659
ALPHA1D = 15.83   S1 = 1.0              POINTEF = .9664
ALPHA1D = 16.73   S1 = 1.0              POINTEF = .9669
ALPHA1D = 17.63   S1 = 1.0              POINTEF = .9673
ALPHA1D = 18.53   S1 = 1.0              POINTEF = .9678
ALPHA1D = 19.43   S1 = 1.0              POINTEF = .9682
ALPHA1D = 20.33   S1 = 1.0              POINTEF = .9687
ALPHA1D = 21.23   S1 =-1.0              POINTEF = .9696
ALPHA1D = 22.13   S1 =-1.0              POINTEF = .9692
ALPHA1D = 23.03   S1 =-1.0              POINTEF = .9687
ALPHA1D = 23.93   S1 =-1.0              POINTEF = .9683
ALPHA1D = 24.83   S1 =-1.0              POINTEF = .9678
ALPHA1D = 25.73   S1 =-1.0              POINTEF = .9674

CYCLEFF = .9660


COEFFICIENT (MU) = .050
```

```
ALPHA1D =  4.13    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D =  5.03    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D =  5.93    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D =  6.83    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D =  7.73    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D =  8.63    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D =  9.53    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D = 10.43    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D = 11.33    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D = 12.23    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D = 13.13    S1 = 1.0    S2 =-1.0    POINTEF = .9295
ALPHA1D = 14.03    S1 = 1.0                POINTEF = .9321
ALPHA1D = 14.93    S1 = 1.0                POINTEF = .9330
ALPHA1D = 15.83    S1 = 1.0                POINTEF = .9339
ALPHA1D = 16.73    S1 = 1.0                POINTEF = .9348
ALPHA1D = 17.63    S1 = 1.0                POINTEF = .9357
ALPHA1D = 18.53    S1 = 1.0                POINTEF = .9365
ALPHA1D = 19.43    S1 = 1.0                POINTEF = .9374
ALPHA1D = 20.33    S1 = 1.0                POINTEF = .9383
ALPHA1D = 21.23    S1 =-1.0                POINTEF = .9412
ALPHA1D = 22.13    S1 =-1.0                POINTEF = .9403
ALPHA1D = 23.03    S1 =-1.0                POINTEF = .9395
ALPHA1D = 23.93    S1 =-1.0                POINTEF = .9386
ALPHA1D = 24.83    S1 =-1.0                POINTEF = .9378
ALPHA1D = 25.73    S1 =-1.0                POINTEF = .9369

CYCLEFF = .9336


COEFFICIENT (MU) = .075

ALPHA1D =  4.13    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D =  5.03    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D =  5.93    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D =  6.83    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D =  7.73    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D =  8.63    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D =  9.53    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D = 10.43    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D = 11.33    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D = 12.23    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D = 13.13    S1 = 1.0    S2 =-1.0    POINTEF = .8972
ALPHA1D = 14.03    S1 = 1.0                POINTEF = .8999
ALPHA1D = 14.93    S1 = 1.0                POINTEF = .9012
ALPHA1D = 15.83    S1 = 1.0                POINTEF = .9025
ALPHA1D = 16.73    S1 = 1.0                POINTEF = .9037
ALPHA1D = 17.63    S1 = 1.0                POINTEF = .9050
ALPHA1D = 18.53    S1 = 1.0                POINTEF = .9062
ALPHA1D = 19.43    S1 = 1.0                POINTEF = .9075
ALPHA1D = 20.33    S1 = 1.0                POINTEF = .9088
ALPHA1D = 21.23    S1 =-1.0                POINTEF = .9144
ALPHA1D = 22.13    S1 =-1.0                POINTEF = .9132
ALPHA1D = 23.03    S1 =-1.0                POINTEF = .9120
ALPHA1D = 23.93    S1 =-1.0                POINTEF = .9108
ALPHA1D = 24.83    S1 =-1.0                POINTEF = .9096
ALPHA1D = 25.73    S1 =-1.0                POINTEF = .9084
```

CYCLEFF = .9029

COEFFICIENT (MU) = .100

ALPHA1D = 4.13	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 5.03	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 5.93	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 6.83	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 7.73	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 8.63	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 9.53	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 10.43	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 11.33	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 12.23	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 13.13	S1 = 1.0	S2 =-1.0	POINTEF = .8666
ALPHA1D = 14.03	S1 = 1.0		POINTEF = .8688
ALPHA1D = 14.93	S1 = 1.0		POINTEF = .8704
ALPHA1D = 15.83	S1 = 1.0		POINTEF = .8720
ALPHA1D = 16.73	S1 = 1.0		POINTEF = .8736
ALPHA1D = 17.63	S1 = 1.0		POINTEF = .8752
ALPHA1D = 18.53	S1 = 1.0		POINTEF = .8769
ALPHA1D = 19.43	S1 = 1.0		POINTEF = .8785
ALPHA1D = 20.33	S1 = 1.0		POINTEF = .8801
ALPHA1D = 21.23	S1 =-1.0		POINTEF = .8893
ALPHA1D = 22.13	S1 =-1.0		POINTEF = .8877
ALPHA1D = 23.03	S1 =-1.0		POINTEF = .8862
ALPHA1D = 23.93	S1 =-1.0		POINTEF = .8847
ALPHA1D = 24.83	S1 =-1.0		POINTEF = .8831
ALPHA1D = 25.73	S1 =-1.0		POINTEF = .8816

CYCLEFF = .8736

COEFFICIENT (MU) = .125

ALPHA1D = 4.13	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 5.03	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 5.93	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 6.83	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 7.73	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 8.63	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 9.53	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 10.43	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 11.33	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 12.23	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 13.13	S1 = 1.0	S2 =-1.0	POINTEF = .8378
ALPHA1D = 14.03	S1 = 1.0		POINTEF = .8387
ALPHA1D = 14.93	S1 = 1.0		POINTEF = .8406
ALPHA1D = 15.83	S1 = 1.0		POINTEF = .8425
ALPHA1D = 16.73	S1 = 1.0		POINTEF = .8445
ALPHA1D = 17.63	S1 = 1.0		POINTEF = .8464

ALPHA1D = 18.53	S1 = 1.0		POINTEF = .8483
ALPHA1D = 19.43	S1 = 1.0		POINTEF = .8503
ALPHA1D = 20.33	S1 = 1.0		POINTEF = .8522
ALPHA1D = 21.23	S1 =-1.0		POINTEF = .8655
ALPHA1D = 22.13	S1 =-1.0		POINTEF = .8637
ALPHA1D = 23.03	S1 =-1.0		POINTEF = .8619
ALPHA1D = 23.93	S1 =-1.0		POINTEF = .8601
ALPHA1D = 24.83	S1 =-1.0		POINTEF = .8583
ALPHA1D = 25.73	S1 =-1.0		POINTEF = .8565

CYCLEFF = .8458

COEFFICIENT (MU) = .150

ALPHA1D = 4.13	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 5.03	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 5.93	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 6.83	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 7.73	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 8.63	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 9.53	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 10.43	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 11.33	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 12.23	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 13.13	S1 = 1.0	S2 =-1.0	POINTEF = .8106
ALPHA1D = 14.03	S1 = 1.0		POINTEF = .8095
ALPHA1D = 14.93	S1 = 1.0		POINTEF = .8117
ALPHA1D = 15.83	S1 = 1.0		POINTEF = .8139
ALPHA1D = 16.73	S1 = 1.0		POINTEF = .8162
ALPHA1D = 17.63	S1 = 1.0		POINTEF = .8184
ALPHA1D = 18.53	S1 = 1.0		POINTEF = .8206
ALPHA1D = 19.43	S1 = 1.0		POINTEF = .8229
ALPHA1D = 20.33	S1 = 1.0		POINTEF = .8251
ALPHA1D = 21.23	S1 =-1.0		POINTEF = .8431
ALPHA1D = 22.13	S1 =-1.0		POINTEF = .8410
ALPHA1D = 23.03	S1 =-1.0		POINTEF = .8389
ALPHA1D = 23.93	S1 =-1.0		POINTEF = .8369
ALPHA1D = 24.83	S1 =-1.0		POINTEF = .8348
ALPHA1D = 25.73	S1 =-1.0		POINTEF = .8327

CYCLEFF = .8193

COEFFICIENT (MU) = .175

ALPHA1D = 4.13	S1 = 1.0	S2 =-1.0	POINTEF = .7849
ALPHA1D = 5.03	S1 = 1.0	S2 =-1,0	POINTEF = .7849
ALPHA1D = 5.93	S1 = 1.0	S2 =-1.0	POINTEF = .7849
ALPHA1D = 6.83	S1 = 1.0	S2 =-1.0	POINTEF = .7849
ALPHA1D = 7.73	S1 = 1.0	S2 =-1.0	POINTEF = .7849
ALPHA1D = 8.63	S1 = 1.0	S2 =-1.0	POINTEF = .7849
ALPHA1D = 9.53	S1 = 1.0	S2 =-1.0	POINTEF = .7849

```
ALPHA1D = 10.43   S1 = 1.0   S2 =-1.0   POINTEF = .7849
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF = .7849
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF = .7849
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF = .7849
ALPHA1D = 14.03   S1 = 1.0              POINTEF = .7813
ALPHA1D = 14.93   S1 = 1.0              POINTEF = .7838
ALPHA1D = 15.83   S1 = 1.0              POINTEF = .7862
ALPHA1D = 16.73   S1 = 1.0              POINTEF = .7887
ALPHA1D = 17.63   S1 = 1.0              POINTEF = .7912
ALPHA1D = 18.53   S1 = 1.0              POINTEF = .7937
ALPHA1D = 19.43   S1 = 1.0              POINTEF = .7962
ALPHA1D = 20.33   S1 = 1.0              POINTEF = .7988
ALPHA1D = 21.23   S1 =-1.0              POINTEF = .8219
ALPHA1D = 22.13   S1 =-1.0              POINTEF = .8196
ALPHA1D = 23.03   S1 =-1.0              POINTEF = .8173
ALPHA1D = 23.93   S1 =-1.0              POINTEF = .8150
ALPHA1D = 24.83   S1 =-1.0              POINTEF = .8127
ALPHA1D = 25.73   S1 =-1.0              POINTEF = .8104

CYCLEFF = .7940


COEFFICIENT (MU) = .200

ALPHA1D =  4.13   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D =  5.03   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D =  5.93   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D =  6.83   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D =  7.73   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D =  8.63   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D =  9.53   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D = 10.43   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF = .7606
ALPHA1D = 14.03   S1 = 1.0              POINTEF = .7540
ALPHA1D = 14.93   S1 = 1.0              POINTEF = .7567
ALPHA1D = 15.83   S1 = 1.0              POINTEF = .7594
ALPHA1D = 16.73   S1 = 1.0              POINTEF = .7621
ALPHA1D = 17.63   S1 = 1.0              POINTEF = .7648
ALPHA1D = 18.53   S1 = 1.0              POINTEF = .7676
ALPHA1D = 19.43   S1 = 1.0              POINTEF = .7703
ALPHA1D = 20.33   S1 = 1.0              POINTEF = .7731
ALPHA1D = 21.23   S1 =-1.0              POINTEF = .8018
ALPHA1D = 22.13   S1 =-1.0              POINTEF = .7993
ALPHA1D = 23.03   S1 =-1.0              POINTEF = .7967
ALPHA1D = 23.93   S1 =-1.0              POINTEF = .7942
ALPHA1D = 24.83   S1 =-1.0              POINTEF = .7917
ALPHA1D = 25.73   S1 =-1.0              POINTEF = .7892

CYCLEFF = .7699
```

```
COEFFICIENT (MU) = .225

ALPHA1D =  4.13   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D =  5.03   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D =  5.93   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D =  6.83   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D =  7.73   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D =  8.63   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D =  9.53   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D = 10.43   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF = .7377
ALPHA1D = 14.03   S1 = 1.0              POINTEF = .7275
ALPHA1D = 14.93   S1 = 1.0              POINTEF = .7304
ALPHA1D = 15.83   S1 = 1.0              POINTEF = .7333
ALPHA1D = 16.73   S1 = 1.0              POINTEF = .7362
ALPHA1D = 17.63   S1 = 1.0              POINTEF = .7392
ALPHA1D = 18.53   S1 = 1.0              POINTEF = .7422
ALPHA1D = 19.43   S1 = 1.0              POINTEF = .7451
ALPHA1D = 20.33   S1 = 1.0              POINTEF = .7482
ALPHA1D = 21.23   S1 =-1.0              POINTEF = .7827
ALPHA1D = 22.13   S1 =-1.0              POINTEF = .7800
ALPHA1D = 23.03   S1 =-1.0              POINTEF = .7773
ALPHA1D = 23.93   S1 =-1.0              POINTEF = .7746
ALPHA1D = 24.83   S1 =-1.0              POINTEF = .7719
ALPHA1D = 25.73   S1 =-1.0              POINTEF = .7692

CYCLEFF = .7469


COEFFICIENT (MU) = .250

ALPHA1D =  4.13   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D =  5.03   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D =  5.93   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D =  6.83   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D =  7.73   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D =  8.63   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D =  9.53   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D = 10.43   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF = .7160
ALPHA1D = 14.03   S1 = 1.0              POINTEF = .7018
ALPHA1D = 14.93   S1 = 1.0              POINTEF = .7049
ALPHA1D = 15.83   S1 = 1.0              POINTEF = .7080
ALPHA1D = 16.73   S1 = 1.0              POINTEF = .7111
ALPHA1D = 17.63   S1 = 1.0              POINTEF = .7143
ALPHA1D = 18.53   S1 = 1.0              POINTEF = .7174
ALPHA1D = 19.43   S1 = 1.0              POINTEF = .7206
ALPHA1D = 20.33   S1 = 1.0              POINTEF = .7238
ALPHA1D = 21.23   S1 =-1.0              POINTEF = .7646
ALPHA1D = 22.13   S1 =-1.0              POINTEF = .7617
ALPHA1D = 23.03   S1 =-1.0              POINTEF = .7588
ALPHA1D = 23.93   S1 =-1.0              POINTEF = .7559
ALPHA1D = 24.83   S1 =-1.0              POINTEF = .7531
```

```
ALPHA1D = 25.73   S1 =-1.0              POINTEF = .7503

CYCLEFF = .7249


COEFFICIENT (MU) = .275

ALPHA1D =  4.13   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D =  5.03   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D =  5.93   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D =  6.83   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D =  7.73   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D =  8.63   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D =  9.53   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D = 10.43   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF = .6955
ALPHA1D = 14.03   S1 = 1.0              POINTEF = .6768
ALPHA1D = 14.93   S1 = 1.0              POINTEF = .6801
ALPHA1D = 15.83   S1 = 1.0              POINTEF = .6834
ALPHA1D = 16.73   S1 = 1.0              POINTEF = .6867
ALPHA1D = 17.63   S1 = 1.0              POINTEF = .6900
ALPHA1D = 18.53   S1 = 1.0              POINTEF = .6934
ALPHA1D = 19.43   S1 = 1.0              POINTEF = .6967
ALPHA1D = 20.33   S1 = 1.0              POINTEF = .7002
ALPHA1D = 21.23   S1 =-1.0              POINTEF = .7474
ALPHA1D = 22.13   S1 =-1.0              POINTEF = .7443
ALPHA1D = 23.03   S1 =-1.0              POINTEF = .7413
ALPHA1D = 23.93   S1 =-1.0              POINTEF = .7382
ALPHA1D = 24.83   S1 =-1.0              POINTEF = .7352
ALPHA1D = 25.73   S1 =-1.0              POINTEF = .7323

CYCLEFF = .7039


COEFFICIENT (MU) = .300

ALPHA1D =  4.13   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D =  5.03   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D =  5.93   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D =  6.83   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D =  7.73   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D =  8.63   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D =  9.53   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D = 10.43   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D = 11.33   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D = 12.23   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D = 13.13   S1 = 1.0   S2 =-1.0   POINTEF = .6761
ALPHA1D = 14.03   S1 = 1.0              POINTEF = .6526
ALPHA1D = 14.93   S1 = 1.0              POINTEF = .6560
ALPHA1D = 15.83   S1 = 1.0              POINTEF = .6595
ALPHA1D = 16.73   S1 = 1.0              POINTEF = .6629
```

ALPHA1D = 17.63	S1 = 1.0	POINTEF = .6664
ALPHA1D = 18.53	S1 = 1.0	POINTEF = .6699
ALPHA1D = 19.43	S1 = 1.0	POINTEF = .6735
ALPHA1D = 20.33	S1 = 1.0	POINTEF = .6771
ALPHA1D = 21.23	S1 =-1.0	POINTEF = .7309
ALPHA1D = 22.13	S1 =-1.0	POINTEF = .7277
ALPHA1D = 23.03	S1 =-1.0	POINTEF = .7245
ALPHA1D = 23.93	S1 =-1.0	POINTEF = .7214
ALPHA1D = 24.83	S1 =-1.0	POINTEF = .7183
ALPHA1D = 25.73	S1 =-1.0	POINTEF = .7151

CYCLEFF = .6837

DISTRIBUTION

Commander
U.S. Army Armament Research
and Development Command
ATTN: DRDAR-TSS (5)
DRDAR-GCL
DRDAR-LCN, H. Grundler
G. Demitrack
R. Drummond
L. Horowitz
A. Nash
R. Brennan
W. Dunn
A. Giovannoli
C. Janow (3)
L. McConnell
F. Tepper (10)
L. Wisse
Dover, NJ 07801

Administrator
Defense Technical Information Center
ATTN: Accessions Division (12)
Cameron Station
Alexandria, VA 22314

Director
U.S. Army Materiel Systems
Analysis Activity
ATTN: DRXSY-MP
Aberdeen Proving Ground, MD 21005

Commander/Director
Chemical Systems Laboratory
U.S. Army Armament Research
and Development Command
ATTN: DRDAR-CLJ-L
DRDAR-CLB-PA
APG, Edgewood Area, MD 21010

Director
Ballistics Research Laboratory
U.S. Army Armament Research
and Development Command
ATTN: DRDAR-TSB-S
Aberdeen Proving Ground, MD 21005

Chief
Benet Weapons Laboratory, LCWSL
U.S. Army Armament Research
  and Development Command
ATTN: DRDAR-LCB-TL
Watervliet, NY 12189

Commander
U.S. Army Armament Materiel
  Readiness Command
ATTN: DRSAR-LEP-L
Rock Island, IL 61299

Director
U.S. Army TRADOC Systems
  Analysis Activity
ATTN: ATAA-SL
White Sands Missile Range, NM 88002