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**AN ANALYTICAL MEANS FOR DETERMINATION
OF SCORING LIMITED LOAD CAPACITY IN
SLIDING/ROLLING CONTACT**

*POWER SYSTEMS BRANCH
AEROSPACE POWER DIVISION*

DECEMBER 1979

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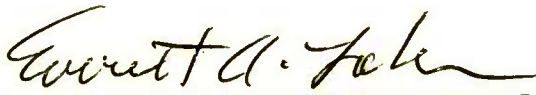
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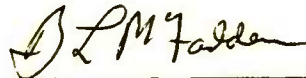
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) Scoring limited load capacity of paired discs in sliding/rolling lubricated contact is addressed. The approach used previously acquired data in a multiple regression analysis. The resulting mathematical expression for load capacity at failure has a correlation coefficient greater than 99%.		

FOREWORD

This report describes an in-house analytical effort conducted by the author. The effort is founded upon background acquired while the author was assigned to the Lubrication Branch (SFL), Fuels and Lubrication Division (SF), Air Force Aero Propulsion Laboratory, Air Force Wright Aeronautical Laboratories, Wright-Patterson Air Force Base, Ohio. The effort was completed during the author's current assignment to the Power Systems Branch (POP), Aerospace Power Division (PO), Air Force Aero Propulsion Laboratory, Air Force Wright Aeronautical Laboratories, Wright-Patterson Air Force Base, Ohio, under Project 3145, "Aerospace Power," Task 314501, "Dynamic Energy Conversion for Aerospace Systems," Work Unit 31450142, "Computer Simulation of Auxiliary Power Systems."

The work reported herein was performed during the period from 1 March 1972 to 27 September 1979 by the author, Everett A. Lake (AFAPL/POP-1), project engineer. The report was released by the author in October 1979 .

I am grateful to Prof G. Shaughnessy, University of Dayton, for his inspirational methods of teaching statistics. His ability to stir and keep the interest of the student is unexcelled.

I am indebted to Mr Howard F. Jones, Chief of the Lubrication Branch of the Air Force Aero Propulsion Laboratory for his help, encouragement, and guidance during the many years of work in this area.

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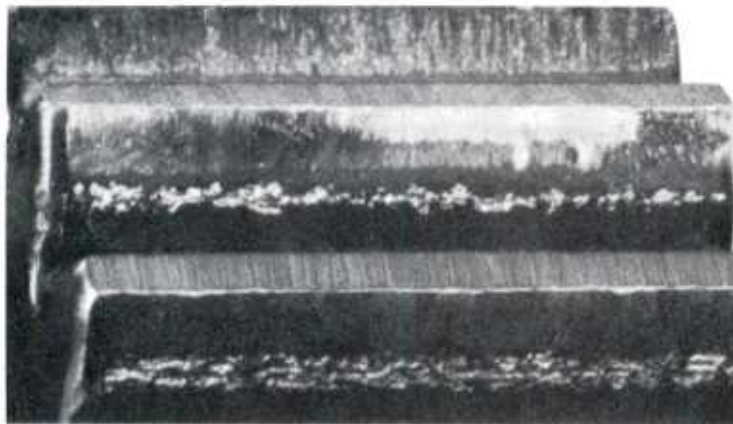
LIST OF SYMBOLS

d	=	Sliding distance, cm
D_1	=	Surface roughness of body 1, μm
D_2	=	Surface roughness of body 2, μm
f	=	Coefficient of friction, nondimensional
G	=	Lubricant flow rate, gal/min
h	=	Oil film thickness, μ inches
h_0	=	Specific film thickness, nondimensional
k	=	Special wear rate, nondimensional
P_m	=	Hardness, kg/m^2
T_c	=	Conjunction temperature, $^{\circ}\text{R}$
T_c'	=	Conjunction temperature, $^{\circ}\text{F}$
T_o'	=	Conjunction (load zone) inlet oil temperature, $^{\circ}\text{R}$
T_s	=	Average disc surface temperature, $^{\circ}\text{R}$
T_s'	=	Average disc surface temperature, $^{\circ}\text{F}$
V	=	Wear volume, cm^3
V_s	=	Sliding velocity, in/sec
V_t	=	Sum velocity, in/sec
W	=	Load, kg
Λ	=	Specific film thickness. The ratio of lubricant film thickness to the composite surface roughness of the two mating surfaces, nondimensional.
ϕ	=	Frictional power loss, Btu/sec
ξ	=	Thickness of lubricant film, m

SECTION I
INTRODUCTION

One failure mode experienced with gears is that of scoring. Evidence of this failure mode is the existence of scratches extending outward along the working surfaces of gear teeth, usually from the pitch line to the tooth tip (see Fig 1). (Occasionally scoring extends below the pitch line to the tooth root.) Continued operation when the surfaces are being scored will lead to significant changes to the tooth working surface contour. A change in the surface contour may be expected, at the very least, to alter noise and vibration levels.

Implicit with the change in surface contour is the generation of wear metal particles. These particles have the potential to further damage the operating surfaces of the scoring gears, other gears in the system, bearings, seals, etc. The final result could be catastrophic failure.



Destructive Scoring: Heavy scoring has taken place above and below the pitch line, leaving the material at the pitch line. As a result, the pitch line pits away as it attempts to redistribute the load. Usually the gear cannot correct itself and ultimately fails.

Figure 1 Scuffed Tooth Surface⁽¹⁾

It is generally conceded scoring can only occur when the lubricating film is no longer of sufficient thickness to prevent metal-to-metal contact of mating surfaces, but scoring does not always occur in the absence of full film lubrication. When conditions are conducive to the formation and maintenance of a boundary lubrication film, operation can continue. In some cases performance may even be improved over the full film condition. This has been noted by Godfrey, Groszek, and others⁽¹⁾⁽²⁾. Unfortunately, the phenomena of boundary lubrication are not well understood. As a result, designers of lubricated surfaces tend to avoid operation in the boundary regime, yet it cannot be avoided entirely for during starting and stopping of some devices (such as gears, cams, bearings, and seals). Operation will almost always occur in the boundary lubricated regime unless lubrication is by hydrostatic means. Of course, temporary overloads often cause operation in the boundary regime too.

SECTION II

GENERAL DISCUSSION

1. Definition of the Objective

The field of boundary lubrication is very complex. It is anticipated that a highly interdisciplinary approach is required to obtain complete understanding of the phenomena. There certainly appears to be considerable difficulty in understanding the details. Understanding seems lacking in areas such as: film formations and destruction; surface texture effects; lubricant/metallurgy interaction effects; and operational environment effects. The level of difficulty in understanding boundary lubrication may be appreciated when Beerbower's depiction of "Regimes and Modes of Wear on Broken-In Machine Parts" is viewed. Beerbower says the dashed lines in this figure (Ref: Fig 2) are "both arbitrary and ill defined"⁽³⁾. In this figure:

Where:

V = Wear volume, cm^3

d = Sliding distance, cm

P_m = Hardness, kg/m^2

W = Load, kg

ξ = Thickness of lubricant film, m

D_1 = Surface roughness of body 1, μm

D_2 = Surface roughness of body 2, μm

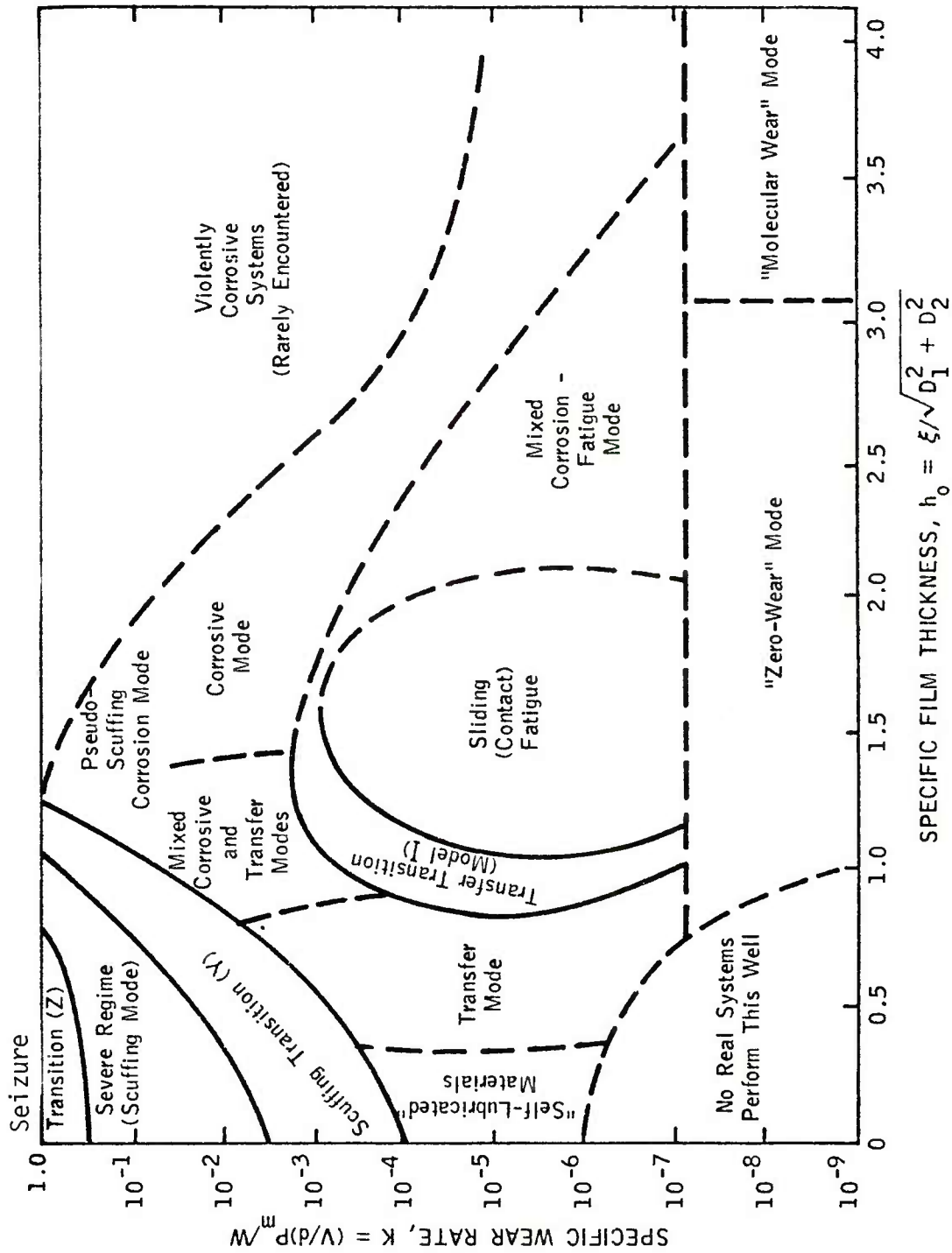


Figure 2 Regimes and Modes of Wear (Excluding Abrasion) on Broken-In Machine Parts (3)

A more recent report, "Lubricant/Metallurgy Interaction Effects on Turbine Engine Lubricant Load Rating"⁽⁴⁾ emphasizes gear tooth failure by scuffing; therefore, a considerable portion of the effort focused upon boundary lubrication. The reported effort included a literature review, analysis, and experiments. The primary experimental tool used was the Air Force Aero Propulsion Laboratory's Disc Tester. This tool, similar to others in the industry, allows simulation of the sliding/rolling contact zone of a lubricated device under conditions reasonably well controlled and/or amenable to measurement.

Typical variables of interest, controlled or measured to varying degrees, included:

a. Sum Velocity. Literally the sum of surface tangential velocities at the line of contact.

b. Sliding Velocity. The difference of the surface's tangential velocities.

c. Lubricant Reactivity. The chemical behavior of the device's surface (material) with the lubricant.

d. Surface Temperature. The average of the mating surfaces.

e. Conjunction Temperature. The temperature in the load or contact zone.

f. Surface Topography. The roughness, waviness, and lay of the surface.

g. Lubricant Supply Temperature. The temperature of the fluid as it is discharged from the nozzle.

h. Heat transfer characteristics of the lubricant and the device.

It was not completely determined how the above listed parameters relate and interact to allow successful operation in the boundary lubrication regime.

It is also interesting to note that, after consideration of the state of the art and their own work, the authors of Lubricant/Metallurgy Interaction Effects on Turbine Engine Load Rating concluded: "The basic mechanisms of scuffing, pitting, and wear under steady operating conditions such as in the sliding/rolling disc system are far from understood."

A review of the reports previously identified and other literature will confirm this conclusion. The conclusion is further reinforced by the various differences of opinion on the meaning of observed test data. As a result, one may well reason that it is better to avoid designing for operation in the boundary lubrication regime.

However, devices which operate in the boundary lubricated regime are many; the lack of a rigorous mathematical relationship notwithstanding. Obviously a sound equation can be expected to improve the ability to successfully design for operation under boundary lubricated conditions. For this effort the ability to better predict scuffing failure load of relatively simple sliding/rolling contact of a disc pair was accepted as a reasonable objective. If this could be achieved, perhaps more complex devices could be considered for further improvement and generalization of the mode.

2. Development of the Model

Predicting disc load at scuffing failure has already been addressed using the tools of thermodynamics, chemistry, physics, etc. in various combinations. The approach selected for model development, in this case, was based simply upon curve fitting of experimental data.

Limited resources precluded the performance of new disc experiments specifically designed to measure and/or control all the suspected parameters of importance. Fortunately the appendix of an Air Force report⁽⁴⁾ contains some excellent experimental data including many of the parameters of interest. Unfortunately some of the parameters reported are calculated from empirical relationships or relationships having unconfirmed accuracy. However, these were considered for inclusion in the model as measured values were not readily available.

The cited data source reports on 280 disc experiments grouped according to the type of lubricant used, method of lubricant distribution, etc. Three of these basic groups were selected for analysis. As a result, the scope was reduced to a more manageable size at the cost of generality.

The number of variables considered independent, yet related, precluded utilizing simple methods of curve fitting. Fortunately a statistical method known as multiple regression, a type of response surface analysis, has been developed⁽⁵⁾.

To use this method, the load capacity, W , was first expressed as:

$$W = F (G, V_t, V_s, f, T_o, T_s, T_c, h, \Lambda, \phi) \quad (1)$$

Where:

- G = Lubricant flow rate, gal/min
- V_t = Sum Velocity, in/sec
- V_s = Sliding Velocity, in/sec
- f = Coefficient of Friction, nondimensional
- T_o = Conjunction (Load Zone) Inlet Oil Temperature, °R
- T_s = Average Disc Surface Temperature, °R
- T_c = Conjunction Temperature, °R
- h = Oil Film Thickness, μ inches
- Λ = Specific Film Thickness. The ratio of lubricant film thickness to the composite surface roughness of the two mating surfaces, nondimensional.
- ϕ = Frictional Power Loss, Btu/sec

This new model was then anticipated to assume the form:

$$W = (G^{\beta_1} V_t^{\beta_2} V_s^{\beta_3} f^{\beta_4} T_o^{\beta_5} T_s^{\beta_6} T_c^{\beta_7} h^{\beta_8} \Lambda^{\beta_9} \phi^{\beta_{10}})C \quad (2)$$

In the above, the β 's and C are constants to be determined.

For the model to be more amenable to further development by the selected response surface analysis, it was transformed to:

$$\begin{aligned} \ln W = & \beta_1 \ln G + \beta_2 \ln V_t + \beta_3 \ln V_s + \beta_4 \ln f + \beta_5 \ln (T_o' + 459.7) \\ & + \beta_6 \ln (T_s' + 459.7) + \beta_7 \ln (T_c' + 459.7) + \beta_8 \ln h \\ & + \beta_9 \ln \Lambda + \beta_{10} \ln \phi + \beta_0 \end{aligned} \quad (3)$$

Where

$$\beta_0 = \ln C$$

and

T_o' , T_s' , and T_c' can now be inserted in °F.

Even with the reduced scope, the task of curve fitting 90 data sets into the above is overwhelming. Fortunately there are computer library routines which allow the task to be accomplished with reasonable ease. One of these routines identified as REGRESS is available in University of Dayton's computer.

Equation 3, along with 90 sets of data, was entered into the computer to solve for each β coefficient. Values for all coefficients except β_9 were found. Without β_9 the F ratio statistic, also determined by REGRESS, was found to have the value of 843.284. At this value the computer associated alpha was zero implying the probability bound had been reached.* The β 's determined are:

$$\begin{aligned} \beta_0 &= 8.41732 \\ \beta_1 &= -0.532615 \\ \beta_2 &= -1.73055 \\ \beta_3 &= 0.052397 \\ \beta_4 &= -0.0631378 \\ \beta_5 &= 2.97684 \\ \beta_6 &= 8.66441 \\ \beta_7 &= -9.997 \\ \beta_8 &= 0.933837 \\ \beta_9 &= 0.00000 \\ \beta_{10} &= 2.07596 \end{aligned}$$

Substituting these β 's into Equation 3 and then taking the analog results in:

$$W = 4524.76 \left[\frac{V_s^{0.0523297} (T_o' + 459.7)^{2.97684} (T_s' + 459.7)^{8.66441}}{G^{0.0532615} V_t^{1.73055} f^{0.0631378} (T_c' + 459.7)^{9.997}} \right. \\ \left. \frac{h^{0.933837} \phi^{2.07596}}{\phantom{G^{0.0532615} V_t^{1.73055} f^{0.0631378} (T_c' + 459.7)^{9.997}}} \right] \quad (4)$$

*This conclusion is due to the limits of the F statistic table stored in the computer. In a physical sense it identifies Λ as having no effect on W at scoring failure. It confirms full film lubrication does not exist when scoring occurs.

The real worth of Eq 4 was assessed by investigating the proportion of variability in W that it could be explained. This was accomplished by calculating the quotient of the regression sum of squares divided by the total sum of squares. For Eq 4, this quotient, the correlation coefficient, was found to be 0.994771. That is, 98.9569 percent of the variation in load, W , can be explained by Equation 4.

It should be noted this statement is valid for mean values of the dependent as well as the independent variables. That is, Equation 4 cannot exactly predict W as all the variables have some distribution (assumed normal) about their mean value.

3. Comparison of the Model to Experimental Data

Test numbers were randomly selected from the 90 data sets used to develop Equation 4. The independent variables from 15 of these sets were used in Equation 4 to compute failure load. These calculated failure loads are compared to experimentally determined values in Table 1. The error values tabulated are based upon:

- Assuming the measured load to be accurate to the nearest 50 lb increment, and
- Adjusting the computed load to the nearest 50 lb increment.

The mean error of these selected data sets is 0.9% with a standard deviation of 8.0%. Based upon the assumption that the variables have normal distributions then, by the use of t statistics, mean error is predicted to lie in the range from -5.3% to 7.0% with 99% confidence.

It was recognized that the worth of Eq 4 could better be judged on its ability to predict failure loads for the full spectrum of data reported in Reference 4. The full data spectrum (280 data sets) encompasses a considerable range expansion of some parameters and other potentially important considerations.

The range of parameters and conditions associated with the basic 90 data sets and the full spectrum of groups are presented in Table 2 for ease of comparison.

TABLE 1
 COMPARISON OF COMPUTED VERSUS EXPERIMENTALLY
 DETERMINED FAILURE LOAD, BASIC GROUPS^(a)

Test Nr (b)	Computer Load (Nearest Pound)	Calculated Load (Nearest 50 Pounds)	Experimental Load (Nearest 50 Pounds)	Error (Percent)
54	1010	1000	950	+5.3
57	513	500	500	0
29	1068	1050	1100	-4.5
92	541	550	500	10.0
97	553	550	600	-8.3
84	857	850	800	6.2
43	2240	2250	2250	0
63	648	650	650	0
71	2992	3000	3300	-9.1
31	1162	1150	1100	4.5
96	653	650	750	13.3
20	3362	3350	3450	-2.9
72	1436	1450	1550	-6.4
94	470	450	500	-10.0
99	1155	1150	1000	15.0

(a) The 90 data sets form the Basic Groups. These sets were used in the development of Equation 4.

(b) These Test Nrs correspond to those used in Reference 4, Appendix D.

TABLE 2
 COMPARISON OF PARAMETER RANGE AND
 CONDITIONS, BASIC GROUPS VERSUS FULL SPECTRUM

Parameter or Condition	Basic Groups	All Groups
Disc Material	AMS 6265	AMS 6263, AMS 6475
Lubricant	MIL-L-7808G	MIL-L-7808G & H
Disc Interacting Frequencies	High Speed Disc 24 to 30 revolutions/contact; Low Speed Disc 9 to 13 revolutions/contact.	Same as Basic Group. Also High Speed Disc 2 to 7 revolutions/contact; Low Speed Disc 1 to 3 revolutions/contact.
Disc Surface Composite Roughness	$6.0 \pm 1.4 \mu$ in AA	$3.5 \pm 0.7 \mu$ in AA and $6.0 \pm 1.4 \mu$ in AA
Oil Supply Method	Conjunction Inlet or Conjunction Exit w/Disc Side	Conjunction Inlet, Conjunction Exit, or Conjunction Exit w/Disc Side
Oil Supply Rate	0.033 to 2.5 gal/min total	0.033 to 100 gal/min total
Oil Supply Jet Temp	190 ⁰ F	190 and 275 ⁰ F
Ambient Pressure (Disc Environment)	14.7	14.7 psia and 22.2 psia
Sum Velocity	600 and 1200 inch/sec	600,1200,2000, and 3000 inch/sec
Sliding Velocity	178, 237, 292, 356, 474, and 583 inch/sec	178, 200, 237, 240, 292, 300, 356, 474, 583 inch/ sec

Test sets from the full data spectrum were randomly chosen. The independent parameters were used in Equation 4 to estimate the load at failure. These computed values are compared to the experimentally determined values in Table 3. The error values tabulated are based upon:

- Assuming the Measured Load to be accurate to the nearest 50 lb increment, and
- Adjusting the Computed Load to the nearest 50 lb increment.

The mean error of this sample of the total population is -2.6% with a standard deviation of 7.9%. Based upon the assumption that the variables have normal distributions then, by the use of t statistics, the mean error is predicted to lie in the range from -8.6% to 3.5% with 99% confidence.

TABLE 3
 COMPARISON OF COMPUTER VERSUS EXPERIMENTALLY
 DETERMINED FAILURE LOAD, ALL GROUPS^(a)

Test Nr (b)	Computed Load (Nearest Pound)	Computed Load (Nearest 50 Pounds)	Experimental Load (Nearest 50 Pounds)	Error (Percent)
150	773	750	750	0
2	2250	2250	2300	-2.2
241	1359	1350	1300	3.8
182	1000	1000	950	5.3
239	520	500	600	-16.7
213	2677	2700	2900	-6.9
94	470	450	500	-10.0
73	3137	3150	3400	-7.4
180	966	950	1000	-5.0
70	2879	2900	3150	-7.9
99	1155	1150	1000	15.0
200	2947	2950	3300	-10.6
186	1791	1800	1800	0
43	2240	2250	2250	0
137	2336	2350	2250	4.4

a. Two hundred eighty data sets; all the available data contained in Ref 4, Appendix D.

b. These test numbers correspond to those used in Ref 4, Appendix D.

SECTION III CONCLUSIONS

Equation 4 provides a reasonably accurate means for the determination of scuffing limited failure load on the AFAPL Disc Tester.

Inspection of Equation 4 predicts the parameters of greatest influence on load capacity, as determined by the AFAPL Disc Tester to be:

T_c = The conjunction temperature

\bar{T}_s = The average disc surface temperature

The next greatest influence is expected from:

T_o = The load zone temperature

ϕ = The frictional power loss

V_t = The sum velocity of the surface

The technique of multiple regression holds considerable promise as a means of developing a more general relationship describing disc and perhaps gear scuffing failure limited loads.

Multiple regression analysis is anticipated to be a useful tool for investigating other complex phenomena. It should be capable of:

- Providing interim empirical relationships;
- Suggesting the most promising approaches to develop more rigorous relationships from first principles; and
- Identifying significant system parameters, with their relative strength of influence. (Monitoring of these system parameters may provide the needed information to support "on condition maintenance" concepts.)

SECTION IV

RECOMMENDATIONS

A Complete multiple regression analysis of data from the AFAPL Disc Tester is warranted. The analysis should result in a more generalized relationship with improved accuracy. This analysis should be followed by a new experimental program with this machine.

Any new experimental program should maximize the parameters measured and/or controlled. The use of calculated parameter values such as the coefficient of friction should be avoided where possible. Again, an improvement in accuracy is anticipated.

Multiple regression analysis should be investigated as a means of establishing relationships from readily measured engine parameters. These relationships (monitoring engine health) may be combined electronically to annunciate not only a need for maintenance but what the problem may be.

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