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RESEARCH REPORT - USER'S MANUAL

FOR

COMPUTER PROGRAM AT81Y003 SHABERTH

MAY 1981

Ste dy State and Transient Thermal Analysis of a Shaft Bearing System Including Ball, Cylindrical and Tapered Roller Bearings

CONTRIBUTORS:

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SUBMITTED TO:

NATIONAL AERONAUTICS & SPACE ADMINISTRATION
LEWIS CENTER
21000 BROOKPARK ROAD
CLEVELAND, OH 44135
UNDER CONTRACT NO. NAS3-22690



SUBMITTED BY:

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The material presented in this manual is structured to guide the user in the practical and correct implementation of the SHABERTH computer program. SHABERTH is capable of simulating the thermomechanical performance of a load support system consisting of a flexible shaft supported by up to 5 rolling element bearings. Any combination of ball, cylindrical, and tapered roller bearings can be used to support the shaft. In this version of SHABERTH, the user can select either SKF or NASA models in calculating lubricant film thickness and traction forces. Also, the formulation of the cage pocket/rolling element interaction model was revised to improve solution numerical convergence characteristics.				
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TABLE OF CONTENTS

				PAGE
TABL	E OF	CONTENTS		i
LIST	OF T	ABLES	• • • • • • • • • • • • • • • • • • • •	vii
LIST	OF F	GURES	• • • • • • • • • • • • • • • • • • • •	viii
FORE	WORD.	• • • • • • •	• • • • • • • • • • • • • • • • • • • •	1
1.0	INTR	ODUCTION	• • • • • • • • • • • • • • • • • • • •	3
2.0	PROB	LEM FORM	ULATION AND SOLUTION	4
	2.1	Tempera	ture Calculations	5
		2.1.1 2.1.2 2.1.3	Steady State Temperature Map Transient Temperatures Calculation of Heat Transfer Rate	5 6 8
			2.1.3.1 Generated Heat	8 9 9 10 11 12 13
		2.1.4	Conduction Through a Bearing	14
			2.1.4.1 Thermal Resistance	14
	2.2	Bearing	Dimensional Change Analysis	17
	2.3	Bearing	Inner Ring Equilibrium	18
	2.4	Bearing	Quasi-Dynamic Solution	20
		2.4.1	Cage Degrees of Freedom	23
3.0	PROG	RAM INPUT	۲	28
	3.1	Types of	Input Data	28
	3.2	Data Set	I - Title Cards	29
		3.2.1 3.2.2	Title Card 1	29 29

TABLE OF CONTENTS (continued)

			PAGE
3.3	Data Se	et II - Bearing Data	32
	3.3.1	Card Type 1 - Bearing Type and Material Designations	33 34
		3.3.2.1 Ball Bearing Geometry 3.3.2.2 Tapered Roller Geometry 3.3.2.3 Cylindrical Roller Bearing Geometry	34 40 40
	3.3.3	Card Type B3 - Rolling Element Geometry	42
		3.3.3.1 Ball Geometry	42 42 44
	3.3.4	Card Type B4 - Rolling Element Ring Geometry	44
		3.3.4.1 Ball Bearing	44
		Contact Geometry	44 46
		3.3.4.3.1 Card Type B4A - Roller Raceway Geometry 3.3.4.3.2 Card Type B4B - Roller Flange Geometry for Cylin-	46
	3.3.5	drical Roller Bearings Roller-Raceway Non-Uniform Profile Definitions	46
		3.3.5.1 Card Type B5 - Outer Raceway Roller Contact	48
	3.3.6	Card Type B6 - Inner Raceway Roller Contact	48
	3.3.7	Ring - Rolling Element Surface Data	48

TABLE OF CONTENTS (continued)

			PAGE
		3.3.7.1 Card Type B7A - Raceway Rolling Element Surface	40
		Data	48
		Roller End Surface Data	50
	3.3.8	Card Type B8 - Cage Data	50
	3.3.9	Card Type B9 - Shaft and Housing Fit Dimensions	50
	3.3.10	Card Type B10 - Shaft & Housing Fit Dimensions	51
	3.3.11	Card Type Bll - Elastic-Moduli	51
	3.3.12	Card Type Bl2 - Poisson's Ratio	51
	3.3.13	Card Type B13 - Density	51 .
	3.3.14	Card Type Bl4 - Coefficient of Thermal Expansion	51
	3.3.15	Card Type B15 - Lubrication and Friction Data	51
	3.3.16	Card Type Bl6	52
3.4	Data Se	t III - Thermal Model Data	53
	3.4.1	Card Type T1	53
	3.4.2	Card Type T2	58
	3.4.3	Card Type T3	58
	3.4.4	Card Type T4	58
	3.4.5	Card Type T5 & T5A	58
	3.4.6	Card Type T6	59
	3.4.7	Card Type T7	59
	3.4.8	Card Type T8	60
	3.4.9	Card Type T9	61

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TABLE OF CONTENTS (continued)

1

-

]

-

				PAGE
	3.5	Data Se	t IV - Loading Data	61
	3.6	Data Se	t V - Shaft Input Data	61
		3.6.1	Card Type S1	62
		3.6.2	Card Type S2	63
		3.6.3	Card Type S3	63
4.0	COMP	UTER PRO	GRAM OUTPUT	64
	4.1	Introdu	ction	64
	4.2	Bearing	Output	64
			4.2.1.1 Linear and Angular Deflections	64
			4.2.1.2 Reaction Forces and Moments	65
		4.2.2	Fatigue Life Data	65
			4.2.2.1 h/sigma	65 66 66
		4.2.3	Temperatures Relevant to Bearing Performance	66
		4.2.4	Frictional Heat Generation Rate and Bearing Friction Torque	66
			4.2.4.1 Frictional Heat Generation Rate	66 67
		4.2.5	EHD Film and Heat Transfer Data	67
			4.2.5.1 EHD Film Thickness	67 67 67 67

TABLE OF CONTENTS (continued)

			PAGE
	4.2.6	Fit and Dimensional Change Data	68
		4.2.6.1 Fit Pressures	68
		Ring)4.2.6.3 Clearances	68 68
	4.2.7	Lubricant Temperatures and Physical Properties	69
	4.2.8	Cage Data	69
		4.2.8.1 Cage-Land Interface	69 69
4.3	Rolling	Element Output	69
	4.3.1	Rolling Element Kinematics	69
		4.3.1.1 Rolling Element Speeds 4.3.1.2 Speed Vector Angles	70 70
	4.3.2	Rolling Element Raceway Loading	70
		4.3.2.1 Normal Forces	70 70 71 72
	4.3.3	Roller End-Flange Contact Data	72
		4.3.3.1 Normal Force	72 72 72 72 73 73
4.4	Thermal	Data	73
4.5	Shaft Da	ata	73
4.6	Program	Error Messages	73

TABLE OF CONTENTS (continued)

		PAGE
4.	6.1 From Subroutine ALLT	73
4.	6.2 From Subroutine SHABE	73
4.	6.3 From Subroutine SOLVXX	74
4.	.6.4 From Subroutine INTFIT	75
5.0 GUIDES	TO PROGRAM USE	76
6.0 LIST OF	REFERENCES	78
APPENDIX A -	SKF Computer Program AT81Y003 SHABERTH Hierarchical Flow Chart	A:1
APPENDIX B -	- Heat Transfer Computation Notes	B:1
	SKF Computer Program SHABERTH Input Data Forms	C:1
APPENDIX D -	SKF Computer Program SHABERTH Sample Output	D:1
APPI DI. E -	Calculation of Cage Pocket and Cage Land Forces	E:1
APPENDIX F -	SKF and NASA Versions of Film Thickness and	F:1

i

1/

LIST OF TABLES

NO.	TITLE	PAGE
1	Lubricant Properties of Four Oils	54

LIST OF FIGURES

ij

1

NO.	TITLE	PAGE
2.1	Convective Heat Transfer	12
2.2	Divided Fluid Flow From Node i	13
2.3	Contact Geometry and Temperatures	16
2.4	Bearing Inertial (XYZ) and Rolling Element (xyz) Coordinate Systems	22
2.5	Inner Ring-Cage Land Contact Geometry	24
2.6	Outer Ring-Cage Land Contact Geometry	25
3.1	Angular Contact Ball Bearing Geometry	36
3.2	Split Inner Ring Ball Bearing Geometry	39
3.3	Tapered Roller Bearing Geometry	41
3.4	Tapered Roller and Roller Raceway Geometry	43
3.5	Cylindrical Roller Bearing Geometry	45
3.6	Roller-Raceway Lamination Showing Relative Approach (δu) and Crown Drop (δc)	47
3.7	Cylindrical Roller Bearing Flange Inversions	49

1185C -1 - PR100

FOREWORD

The SHABERTH computer program was originally developed by Kell-strom [1] under U.S. Army Contract DAAD05-73-C-0011 sponsored by the Ballistic Research Laboratory (BRL), Aberdeen Proving Ground, Maryland, to simulate the thermo-mechanical performance of load support systems consisting of a shaft supported by up to 5 rolling-element bearings. The program has since undergone extensive development to add new capabilities and improve its execution performance.

This user's manual describes the use of a version of the SHABERTH computer code developed under NASA-Lewis Research Center Contract NAS3-22690 with Mr. H. Coe as Technical Monitor. The revisions made to the program include:

- Modification of the cage module to calculate cage pocket and cage land forces i ball, cylindrical and tapered roller bearings, using the models originally developed for NASA Computer Program CYBEAN [2]. An option has been provided to allow the specification of single or multiple degrees of freedom cage simulation by input data.
- Addition of an option which permits the program to analyze a single ball or roller bearing without the specification of shaft geometry.
- 3. Combination of two versions of code within the program: The SKF version and the NASA version. The differences between the two versions reside in the calculation of the elastohydrodynamic (EHD) film thickness and traction forces which develop between rolling element-raceway and rolling element-cage concentrated contacts. The original film thickness models (Archard-Cowking [3] and Dowson-Higginson [4]) and the Tallian traction model [5] are used in the SKF version, while the NASA version uses the Loewenthal model [6] to calculate film thickness and the Allen model [7] to determine traction forces.

Additionally, a new subroutine, FLMFAC (replacing LRHS), is used to determine the lubricant life factor as a function of A (film thickness/surface roughness). The values of the lube life factor produced by FLMFAC adhere closely to the curve recommended by the ASME [8, 9].

AT81D040

A new section has been added, Appendix E, detailing the calculation of cage pocket and cage land forces in ball and roller bearings. Another new section, Appendix F, describes the differences between the SKF and NASA methods of calculating film thickness and traction forces, and explains the differences in executing each version of the code.

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AT81D040

I. INTRODUCTION

SHABERTH performs a thermo-mechanical simulation of a load support system consisting of a flexible shaft supported by up to five (5) rolling element bearings. The shaft can be hollow or solid and of arbitrary geometry. Any combination of ball, cylindrical, or tapered roller bearings can be used to support the haft. The cylindrical roller bearing analysis permits thrust load to be carried by inner and outer ring flanges. The applied loading can consist of point or distributed moments, point or distributed forces, and shaft misalignments.

Concentrated contact EHD traction models [5,10] are included in the program. Hydrodynamic rolling and shear forces in the inlet zone of the lubricated contacts are accounted for [10]. The effects of surface roughness [5], heating of the lubricant in the contact inlet [11], and lubricant starvation [12] are considered. Bearing operating clearance is determined as a function of shaft and housing fits, component temperatures, and rotational speed[13,14]. A cage model simulates contact between the cage and rolling element as well as the cage and the piloting land.

A lumped mass thermal model allows calculation of steady state or time transient system termperatures considering free and forced convection, conduction, radiation, and mass transport heat transfer [15,16,17]. A maximum of one hundred (100) temperature nodes can be used to describe the thermal system.

The SHABERTH program consists of the following major subprograms:

- 1. Bearing Analysis These programs are largely based upon the methods of Harris [18,19].
- 2. Three Dimensional Shaft Deflection Analysis developed by Norlander and Friedrichson.
- 3. Bearing Dimensional Change Analysis based on the methods of Timoshenko[13] and adapted to the shaft-bearing-housing system by Crecelius [14].
- 4. Generalized Steady State and Transient Temperature Napping and Heat Dissipation Analyses based on the methods of Harris [15] Fernlund [16] and madreason [17].

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2. PROBLEM FORMULATION AND SOLUTION

The purpose of the program is to provide a tool with which the shaft bearing system performance characteristics can be determined as functions of system temperatures. These system temperatures may be a function of steady state operation or a function of time variant conditions brought on by a change in the system steady state condition. Such a change would be the termination of lubricant supply to the bearings and other lubricated mechanical elements.

The program is structured with four nested, calculation schemes as follows:

- 1. Thermal, steady state or transient temperature calculations which predict system temperatures at a given operating state.
- 2. Bearing dimensional equilibrium which uses the bearing temperatures predicted by the temperature mapping subprograms and the rolling element raceway load distribution, predicted by the bearing subprograms, to calculate bearing diametral clearance at a given operating state.
- 5. Shaft-bearing system load equilibrium which calculates bearing inner ring positions relative to the respective outer rings such that the external loading applied to the shaft is equilibrated by the rolling element loads which develop at each bearing inner ring at a given state.
- 4. Bearing rolling element and cage load equilibrium which calculates the rolling element and cage equilibrium positions and rotational speeds based upon the relative inner-outer ring positions, inertia effects and friction conditions, which if lubricated, are temperature dependent.

The above program structure allows complete mathematical simulation of the real physical system. The program has been coded to allow various levels of program execution which prove useful and economical in bearing design studies.

These levels of execution are explained fully in Sections 3, 4, and 5.

The structure of the program and the nesting of the solution loops noted above can be seen clearly in the Program Flow Chart which is discussed in Appendix A.

The sections below present the systems of field equations which are solved in each of the nested calculation schemes. A more detailed discussion is contained in [1,10 and 20].

2.1 Temperature Calculations

Subsequent to each calculation of bearing generated heat rates, either the steady state or transient temperature mapping solution scheme may be executed. This set of sequential calculations is terminated as follows:

- 1. For the steady state case; when each system temperature is within EPI °Centrigrade of its previously predicted value, (EPI is specified by the user). If it is zero or left blank, a default value of 1° Centigrade is used. This criterion implies that the steady state equilibrium condition has been reached.
- 2. The transient calculation terminates when the user specified time up is reached or when one of the system temperatures exceeds 600° C.

2.1.1 Steady State Temperature Map

The mechanical structure to be analyzed is thought of as divided into a number of elements or nodes, each represented by a temperature. The net heat flow to node i from the surrounding nodes j, plus the heat generated at node i, must numerically equal zero. This is true for each node i, i going from 1 to n, n being the number of unknown temperatures.

After each calculation of bearing generated heat, which results from a solution of the shaft-bearing system portion of the program, a set of system temperatures is determined which satisfy the system of equations:

 $q_i = q_{oi} + q_{gi} = 0$ for all temperature nodes i (2.1)

where q_{oi} is the heat flow from all neighboring nodes to node i

q_{gi} is the heat generated at node i. These values may be input or calculated by the shaft bearing program as bearing frictional heat

This scheme is solved with a modified Newton-Raphson method which successfully terminates when either of two conditions are met:

$$\frac{\Delta^{t_i}}{t_i} \le EP2$$
 for all nodes i (2.2)

where: Δt represents the Newton-Raphson correction to the temperature t at a given iteration such that, $t_{N+1} = t_N + \Delta t$ and N + 1, and N, refer to the next and current iteration respectively.

EP2 is a user specified constant. If EP2 is left blank or set to zero (0) a default value of 0.001 is used.

A second convergence criterion dependent upon EP2 is also used. In the system of equations, $q_{0i} + q_{gi} = 0$ for all nodes i, absolute convergence would be obtained if the right hand side (EQ) in fact reduced to zero (0). Usually a small residue remains at each node, such that $(q_{0i} + q_{gi}) = (EQ)i$.

The second convergence criterion is satisfied if:

$$\left[\begin{array}{c} n \frac{(EQ)^{\frac{2}{1}}}{2} \le 100 \text{ x EP2} \end{array}\right] (2.3)$$

where n = number of equations in thermal solution = number of unknown temperatures

2.1.2 Transient Temperatures

In the transient case the net heat q_i transferred to a node i heats the element. It is thus necessary for heat balance at node i that the following equations are satisfied.

$$\rho_i \quad Cp_i \quad V_i \quad \frac{dt_i}{dT_i} = q_i \qquad (2.4)$$

AT81D040

where p = density

Cp = specific heat

V= volume of the element

t= temperature

T= time

The temperatures, t_{oi} , at the time of initiation $T = T_s$ are assumed to be known, that is

$$t_i(T_s) = t_{oi}$$
 $i = 1, 2, ..., n$ (2.5)

The problem of calculating the transient temperature distribution in a bearing arrangement thus becomes a problem of solving a system of non-linear differential equations of the first order with certain initial values given. The equations are non-linear since they contain terms of radiation and free convection, which are non-linear with temperature as will be shown later. The simplest and most economical way of solving these equations is to calculate the rate of temperature increase at the time $T = T_k$ from equation 2.4 and then calculate the temperatures at time $T_k + \Delta T$ from

$$t_{k+1} = t_k + \frac{dt_k}{dT} \Delta T = t_k + \frac{q_k}{\rho C \rho V} \Delta T \qquad (2.6)$$

If the time step ΔT used as program input is chosen too large, the temperatures will oscillate, and if it is chosen too small the calculation will be costly. It is therefore desirable to choose the largest possible time step that does not give an oscillating solution. The program optionally calculates such a time step. The step is obtained from the condition, [16]

$$\frac{dt_{i,k+1}}{dt_{i,k}} \ge 0 i = 1, 2, ..., n (2.7)$$

If this derivative were negative, the implication would be that the local temperature at node i has a negative effect on its future value. This would be tantamount to asserting that the hotter a region is now, the colder it will be after an equal time interval. An oscillating solution would result.

AT81D040

Differentiating equation (2.6) for node i, one obtains

$$\frac{dt_{i,k+1}}{dt_{i,k}} = 1 + \frac{\Delta T_{i}}{\rho_{i} C_{pi} V_{i}} \cdot \frac{dq_{i,k}}{dt_{i,k}} \qquad i = 1,2,...n \qquad (2.8)$$

The derivative $dq_{i,k}/dt_{i,k}$ is calculated numerically

$$\frac{dq_{i,k}}{dt_{i,k}} = \frac{q_{i}(t_{i} + \Delta t_{i}) - q_{i}(t_{i})}{\Delta t_{i}}$$
(2.9)

For each node the value of ΔT_i giving a value of zero to the right hand side of Eqn. (2.8) is calculated. The smallest non-zero value of ΔT obtained in this manner is chosen as the time step.

2.1.3. Calculation of Heat Transfer Rate

The transfer of heat within a medium or between two media can occur by conduction, convection, ratiation and fluid flow.

All these types of heat transfer occur in a bearing application as the following examples show.

- Heat is transferred by conduction between inner ring and shaft and between outer ring and housing.
- 2. Heat is transferred by convection between the surface of the housing and the surrounding air.
- 3. Heat is transferred by radiation between the shaft and the housing.
- 4. When the bearing is lubricated and cooled by circulating oil, heat is transferred by fluid flow.

Therefore, in calculating the net heat flow to a node all the above mentioned modes of heat transfer will be considered.

2.1.3.1 Generated Heat

There may be a heat source at rode i giving rise to a heat flow to be added to the heat flowing from the neighboring nodes.

In the case that the heat source is a bearing, it may either be considered to produce known amounts of power, in which case constant numbers are entered as input to the program, or the shaft-bearing program may be used to calculate the bearing generated heat as a function of bearing temperatures.

2.1.3.2 Conduction

The heat flow q_{ci} , which is transferred by conduction from node i to node j, is proportional to the difference in temperature $(t_i - t_j)$ and the cross sectional area A and is inversely proportional to the distance ℓ between the two points, thus

$$q_{ci,j} = \frac{\lambda A}{\delta} (t_i - t_j)$$
 (2.10)

where λ = the thermal conductivity of the medium.

2.1.3.3 Free Convection

Between a solid medium such as a metallic body and a liquid or gas, heat transfer is by free or forced convection. Heat transfer by free convection is caused by the setting in motion of the liquid or gas as a result of a change in density arising from a temperature differential in the medium. With free convection between a solid medium and air, the heat energy $q_{\rm vi}$; transferred between nodes i and j can be calculated from the equation, (2.11)

$$q_{vi,j} = \alpha_v A (t_i - t_j)^{d}. SIGN(t_i - t_j)$$
 (2.11)

where q = the film coefficient of heat transfer by free convection

A = the surface area of contact between the media

d = is an exponent, usually = 1.25, but any value can be specified as input to the program

SIGN =
$$\begin{cases} 1 & \text{if } t_i \ge t_j \\ -1 & \text{if } t_i < t_j \end{cases}$$

AT81D040

The last factor is included to give the expression $q_{vi,j}$ a correct sign.

The value of α v can be calculated for various cases, see Jacob and Hawkins, [21]

2.1.3.4 Forced Convection

Heat transfer by forced convection takes place when liquid or gas moves around a solid body, for example, when the liquid is forced to flow by means of a pump or when the solid body is moved through the liquid or gas. The heat flow qui transferred by forced convection can be obtained from the following equation.

$$q_{wi,j} = \alpha_w A(t_i - t_j)$$
 (2.12)

where α_w is the film coefficient of heat transfer during forced convection. This value is dependent on the actual shape, the surface condition of the body, the difference in speed, as well as the properties of the liquid or gas.

In most cases, it is possible to calculate the coefficient of forced convection from a general relationship of the form,

$$N_{u} = aR_{e}^{b}P_{r}^{c}$$
 (2.13)

where a, b, and c are constants obtained from handbooks such as [22]. Re and Pr are dimensionless numbers defined by

 N_{11} = Nusselt number = $\alpha_w L/\lambda$

L = characteristic length

 λ = conductivity of the fluid

R_e = Reynold's number = ULp/n

U = characteristic speed

ρ = density of the fluid

n = dynamic viscosity of the fluid

 P_{r} = Prandtl number = $\eta C_{p}/\lambda$

C_n = specific heat

The user can input a constant value for the convection coefficient. Alternatively, he can let the program calculate the coefficient using one of the three option described below. For options 2 and 3, the coefficient is allowed to vary with system temperatures.

Constant viscosity

1. Values of the parameters of equation (2.13) are given as input and a constant value of α_w is calculated by the program.

Temperature dependent viscosity

2. The coefficient α_{ij} for turbulent flow and heating of petroleum oils is given by

$$\alpha_{\rm W} = k_9 \cdot \eta(t)^{\rm k} 10$$
 (2.14)

where k_0 and k_{10} are given as input together with viscosity at two different temperatures.

3. Values of the parameters of equation (2.13) are given as input. Viscosity is given at two different temperatures.

2.1.3.5 Radiation

If two flat parallel, similar surfaces are placed close together and have the same surface area A, the heat energy transferred by radiation between nodes i and j representing those bodies, will be,

$$q_{Ri,j} = \varepsilon \sigma A (t_i + 273)^4 - (t_j + 273)^4$$
 (2.15)

where ϵ is the surface emissivity. The value of the coefficient ϵ is an input variable and varies between 1 for a completely black surface and 0 for an absolutely clean surface. In addition σ is Stefan-Boltzmann's radiation constant which has the value 5.76 x 10-8 watts/m²-(°K)⁴ and t_i and t_j are the temperatures at points i and j.

Heat transfer by radiation under other conditions can also be calculated, [21]. The following equation, for instance applies between two concentric cylindrical surfaces.

$$q_{Ri,j} = \frac{\varepsilon \sigma^{A_{i}} \left[(t_{i} + 273)^{4} - (t_{j} + 273)^{4} \right]}{1 + (1 - \varepsilon) (A_{i}/A_{e})}$$
 (2.16)

where A_i is the area of the inner cylindrical surface A_e is the area of the outer cylindrical surface

2.1.3.6 Fluid Flow

Between nodes established in fluids, heat is transferred by transport of the fluid itself and the heat it contains.

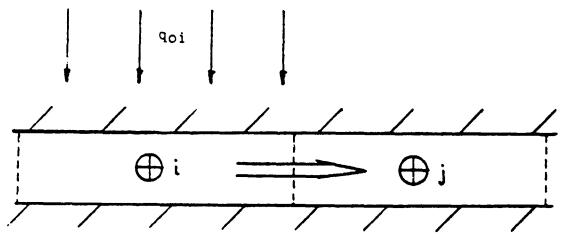


FIGURE 2.1 CONVECTIVE HEAT TRANSFER

Figure 2.1 shows nodes i and j at the midpoints of consecutive segments established in a stream of flowing fluid.

The heat flow $q_{ui,j}$ through the boundary between nodes i and j can be calculated as the sum of the heat flow q_{fi} through the middle of the element i, and half the heat flow q_{oi} transferred to node i by other means, such as convection.

The heat carried by mass flow is,

$$q_{fi} = \rho_i C_{p_i} V_i t_i = K_i t_i$$
 (2.17)

where V_i = the volume flow rate through node i

The heat input to node i is the sum of the heat generated at node i (if any) and the sum over all other nodes of the heat transferred to node i by conduction, radiation, free and forced convection.

$$q_{oi} = q_{G,i} + \sum_{j=1}^{m} (q_{ci,j} + q_{vi,j} + q_{wi,j} + q_{Ri,j})$$
 (2.18)

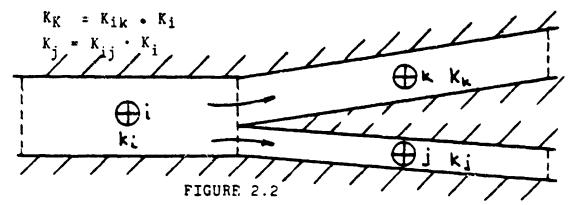
The heat flow between the nodes of Fig. 2.1 is then

$$q_{ui,j} = q_{fi} + q_{oi}/2$$
 (2.19)

If the flow from node i is dividing between nodes j ξ k, (Fig. 2.2) then the heat flow is calculated from

$$q_{ui,j} = K_{ij} (q_{fi} + q_{oi}/2)$$
 (2.20)

where K_{ij} = the proportion of the flow at i going to node j, 0 $< K_{ij} \le 1$. K_{ij} is specified at input.



DIVIDED FLUID FLOW FROM NODE i

2.1.3.7 Total Heat Transferred

The net heat flow rate to node i can be expressed as,

$$q_i = q_{G,i} + \sum_{j=1}^{i} (q_{Ci,j} + q_{ui,j} + q_{vi,j} + q_{wi,j} + q_{Ri,j})$$
 (2.21)

The summation should include all nodes j, both with unknown temperatures as well as boundary nodes, at which the temperature is known, so long as they have a direct heat exchange with node i.

This expression is a non-linear function of temperatures because of the terms $q_{\rm w}$ and $q_{\rm p}$. Therefore the equations to be solved for a steady state solution are non-linear. The subprogram SOLVXX for solving non-linear simultaneous equations is used for this purpose.

2.1.4 Conduction Through a Bearing

As described in Section 2.1.3.2 the conduction between two nodes is governed by the thermal conductivity parameter λ of the medium through which conduction takes place. The value of λ is specified at input.

An exception is when one of the nodes represents a bearing ring and the other a set of rolling elements. In this case the conduction is separately calculated using the principles described below. Note that separate calculations are performed for the rolling element raceway contacts and the rolling element-flange contacts. The methods for both calculations are identical and are performed within the program.

2.1.4.1. Thermal Resistance

It is assumed that the rolling speeds of the rolling elements are so high that the bulk temperature of the rolling elements is the same at both the inner and outer races, except in a volume close to the surface. The resistance to heat flow can then be calculated as the sum of the resistance across the surface and the resistance of the material close to the surface.

The resistance Ω is defined amplicitly by

$$\Delta t = \Omega \cdot q \qquad (2.22)$$

where

At is temperature difference q is heat flow

The resistance due to conduction through the EHD film is calculated as

$$\Omega_1 = \frac{h}{\sum_{i=1}^{N}}$$
 (2.23)

where h is taken to be the calculated plateau film thickness A is the Hertzian contact area at the specific rolling element-ring contact under consideration.

λ is the conductivity of the oil.

The geometry is shown in Figure 2.3(a).

So far, a constant temperature difference between the surfaces has been assumed. But during the time period of contact, the difference will decrease because of the finite thermal diffusivity of the material near the surface, Fig. 2.3 (b).

To points at a distance from the surface, this phenomenon will have the same effect as an additional resistance Ω_2 acting in series with Ω_1 .

This resistance was estimated in [23] as,

$$\Omega_2 = \frac{1}{\lambda \hat{z}_{re,i}} \left(\frac{\pi \psi}{2b_i V} \right)^{\frac{1}{2}} \tag{2.24}$$

where \$\mathcal{l}_{re}\$ = contact length, or in the case of an elliptical contact area, 0.8 times the major axis

 λ = heat conductivity

 ψ = thermal diffusivity = $\lambda/(\rho.C_D)$

ρ = density

7

C_p = specific heat

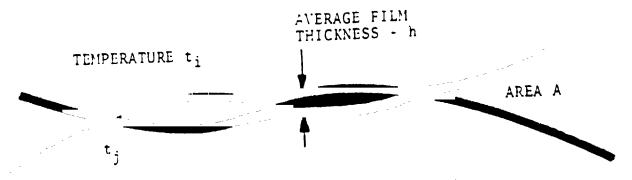
b = half the contact width

V = rolling speed

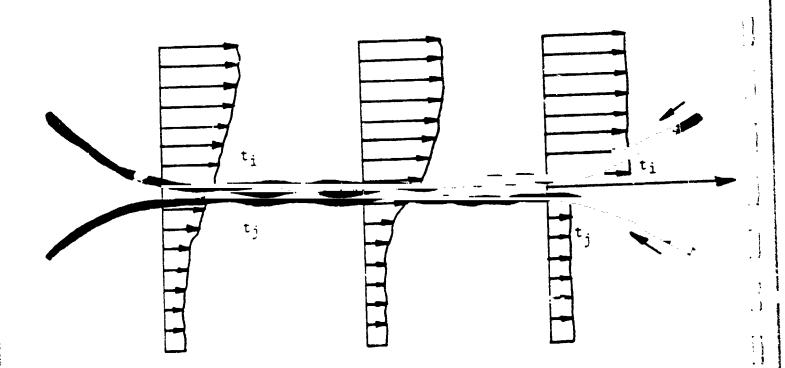
The resultant resistance is

$$\Omega_{\text{res}} = \Omega_1 + \Omega_2 \tag{2.25}$$

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(a) Schematic Concentrated Contact



direction of rolling

(b) Temperature Distribution at Rolling, Concentrated Contact Surfaces

FIGURE 2.3

CONTACT GEOMETRY AND TEMPERATURES

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There is one such resistance at each rolling element. They all act in parallel. The equivalent resistance Ω equivalent

$$\frac{1}{\pi_{\text{eqv}}} = \sum_{i=1}^{n} \frac{1}{\pi_{\text{res},i}}$$
 (2.26)

2.2 Bearing Dimensional Change Analysis

1

The program calculates the changes ir bearing diametral clearances according to the analysis described in [14], and expressed in generalized equation form as,

ΔDCL = f {(Fits)_m, t_{i,Ωm}, (Q_r)_m}, m = 1,2 for inner and outer rings respectively

i = 1,2,3,4,5 for shaft, inner ring, outer ring, housing and rolling element respectively

ADCL is the change in bearing diametral clearance Firs are the cold mounted shaft and housing fits. t; are the component temperatures Ω_m^\pm refers to the ring rotational speeds Q_T^\pm refers to the radial component of the minimum

rolling element-race normal force

A bearing clearance change criterion is satisfied when the change in bearing diametral clearance remains within a narrow, user specified range, for two successive iterations as follows:

$$\frac{|(\Delta DCL)_{N} - (\Delta DCL)_{N-1}|}{D} = \frac{|(\Delta DCL)_{N-1}|}{|(\Delta DCL)_{N-1}|} = \frac{|(\Delta DCL)_{N-1}|}{|(\Delta DCL)_{N$$

where: N denotes the most recent iteration and N-1 denotes the previous iteration, D denotes the ball or roller diameter and EPSFIT is a user specified value, (default value = .0001D)

It should be noted that although ring rotational speeds, and initial, i.e. cold, shaft and housing fits are considered in the clearance change analysis, these two factors are fixed at input and remain constant through the entire solution. Although component temperatures may change as a consequence of the thermal solution, temperatures remain constant through a complete set of clearance change iterations. As a result, only the change in bearing load distribution affects the change in bearing clearance within a set of clearance change iterations.

2.3 Bearing Inner Ring Equilibrium

The bearing inner ring equilibrium solution is obtained by solving the system:

$$(FM_b)_i - (FM_s)_i = 0$$
 for all bearings, i (2.29)

FMb denotes a vector of bearing loads and moments resulting from rolling element/race forces and moments.

AT81D040

$$FM_{bi} = \begin{bmatrix} F_{bxi} \\ F_{byi} \\ F_{bzi} \\ M_{byi} \\ M_{bzi} \end{bmatrix}$$
 Forces (2.30)

If the bearing solution considers friction, FM, is comprised of the ball race friction forces as well as the normal forces.

If the bearing solution is, at the user's option, frictionless, FM, is comprised only of rolling element/race normal contact forces.

FM; denotes a similar vector of loads, exerted on the inner ring by the shaft.

The variables in this system of equations are the bearing inner ring deflections $\overline{\Delta}_{i}$ and the shaft displacements $\overline{\Delta}_{i}$ at all bearing locations. The bearing loads may be expressed as a function of the inner ring deflections.

$$\dot{F}M_b = \dot{F}M_b (\Delta_b^+)$$
 (2.32)

:

The deflection Δ_{b} of a bearing is described by two radial deflections δ_{y} and δ_{z} two angular deflections δ_{y} and θ_{z} and an axial deflection δ_{x} . The axial deflection is assumed to be the same for all bearings and the shaft.

The solution scheme is ended when

$$\frac{\delta (\Delta)_{ij}}{(\Delta)_{ij}} \leq \frac{\text{EPS1 (frictionless)}}{\text{EPS2 (friction)}}$$
(2.33)

i = 1,... (Number of bearings)

j = 1,5 - for the 3 linear and two
angular deflections at
each bearing

If for some i or j, (Δ) ij = 0, Eq (2.34) is used in place of (2.33).

$$\frac{\delta(\Delta)_{ij}}{(0.001 \times NBRG)} \leq EPS1(frictionless)$$
(2.34)

NBRG denotes the number of bearings in the system.

EPS1 or EPS2 is used depending on whether the bearing solutions are fricitonless or include friction, respectively. If the bearing deflections are extremely small, computer-generated numerical inaccuracies may prevent convergence according to the above criteria although a perfectly good solution has been obtained. To overcome this problem, the iteration is terminated if all angular deflections are less than 2 x 10-6 radians and all linear deflections are less than 5 x 10-8 inches. Any one of the above criteria imply that inner ring equilibrium is satisfied.

2.4 Bearing Quasi-Dynamic Solution

The bearing quasi-dynamic solution is obtained through a two step process:

- Elastic Solution considering rolling element centrifugal force, plus the gyroscopic moment for a tapered roller.
- 2) Elastic and Quasi-dynamic Solution*

*Quasi-dynamic equilibrium is used to connote that the true dynamic equilibrium terms containing first derivatives of the ball rotational speed vectors and the second derivatives of rolling element position vectors with respect to time are replaced by numerical expressions which are position rather than time dependent.

The equations which define rolling element quasi-dynamic force equilibrium takes the form

m = 1-3 refers to the outer inner and cage rolling element contacts respectively m = 1-4 for tapered rollers with one roller end-flange contact. m = 1-7 for cylindrical rollers where four roller and flange (2.35) contacts are possible.

where: Q_m is the vector normal load per unit length of the contact. See Ref. [1].

 f_m is the vector of friction force per unit length of the contact. See Ref. [10].

F is the vector of inertia and drag forces. See Ref.[1] is a coordinate along the contact perpendicular to the direction of rolling (usually the major axis)

a, is half the contact length. See Ref. [1].

is the vector sum of the hydrodynamic forces acting on the rolling element at the m-th contact. For ball-raceway contact see Ref. [10]. For the roller-raceway contact, see Ref. [21].

Rolling element moment equilibrium is defined by:

$$\sum_{m} \left[\int_{m}^{a_{m+1}} r_{m} \times (Q_{m} + f_{m}) ds \right] + r_{m} \times F_{m} + M_{I} = 0$$
 (2.36)

 Q_m , f_m , F_m , f_m , and s are defined above, M_T is a vector M_T , M_T , of inertia moments. For the definition of M_T

refer to Ref. [1].

rm is a vector from the rolling element center to the point of contact.

In the frictionless elastic solution F_m and $f_m=0$. Additionally, the only rolling element inertia term considered in the frictionless solution is centrifugal force, plus the gyro-

scopic moment for tapered rollers. As a consequence only the axial and radial force equilibrium equations are solved for each ball. For each roller the radial and axial force equilibrium and the tilting moment about the z axis of Fig. 2.4 is solved. A dummy equation for axial force equilibrium is included in the

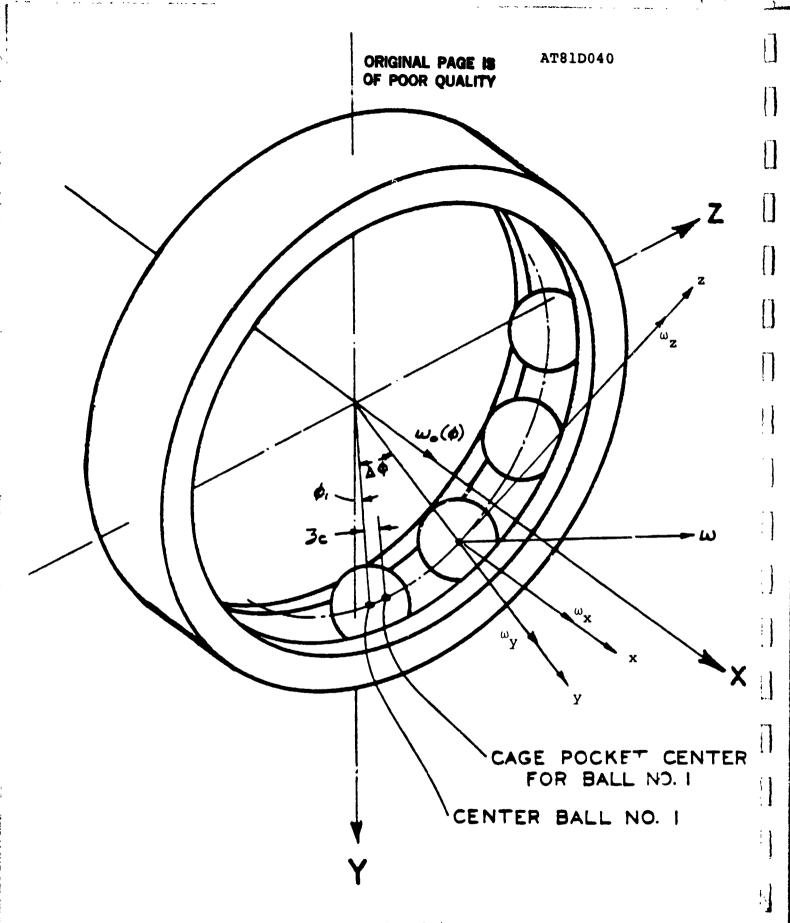


FIGURE 2.4

BEARING INERTIAL (XYI) AND ROLLING ELEMENT (xyz), COORDINATE SYSTEMS

AT81D040

solution matrix which keeps the roller centered with respect to the outer race if the cylindrical roller bearing takes no thrust load.

The friction solution determines ball quasi-dynamic equilibrium for six degrees of freedom. The rolling-element variables in this solution are x1, y1, ω x, ω y, ω z, and ω o.

where x_lis the rolling element axial position relative to the outer race groove curvature center.

y_lis the rolling element radial position relative to the outer race groove curvature center.

ωx, ωy, ωz are orthogonal rolling element rotational speeds relative to the cage speed, about the x, y, and z axes and Wo is the rolling element orbital speed.

The variables x_1 and y_1 are the ball unknowns in the frictionless solution. The variables in the roller frictionless solution are x_1 , y_1 , and 0z = arctan $(\omega y/\omega x)$

Details of the cage analysis are contained in Appendix E. Either one or three cage equilibrium equations are considered, depending upon the number of degrees of freedom given to the cage.

The cage equations and cage-rolling element interactions are not considered when the friction forces are omitted from the rolling element equilibrium equations.

2.4.1. Cage Degrees of Freedom

The program has been modified to allow the user to specify the number of degrees of freedom (DOF) of the cage as either 1 or 3. The single degree of freedom corresponds to a smaller angular rotation about the bearing axis, measured with respect to rolling element 1. The angular displacement is converted to a linear dimension by multiplying it by the bearing pitch diameter and is noted in Fig. 2.4 as z. When a single degree of freedom is input, the sum of moments acting on the cage about the bearing X axis is required to be zero. This moment equation considers the cage-rolling element normal and friction forces as well as the torque generated at the cage-ring surface.

If the user assigns three degrees of freedom to the cage, it is permitted to move to an eccentric position with respect to the land on which it is piloted. The additional degrees of freedom are the cage center of mass radial displacement, e, and the angular displacement 0 of the center of mass, with respect to the bearing Y axis. (See Figures 2.5 and 2.6.) The radial friction forces as well as the pressure build-up between the cage and its piloting surface are considered in the equilibrium equations. The effect of the cage mass is neglected.

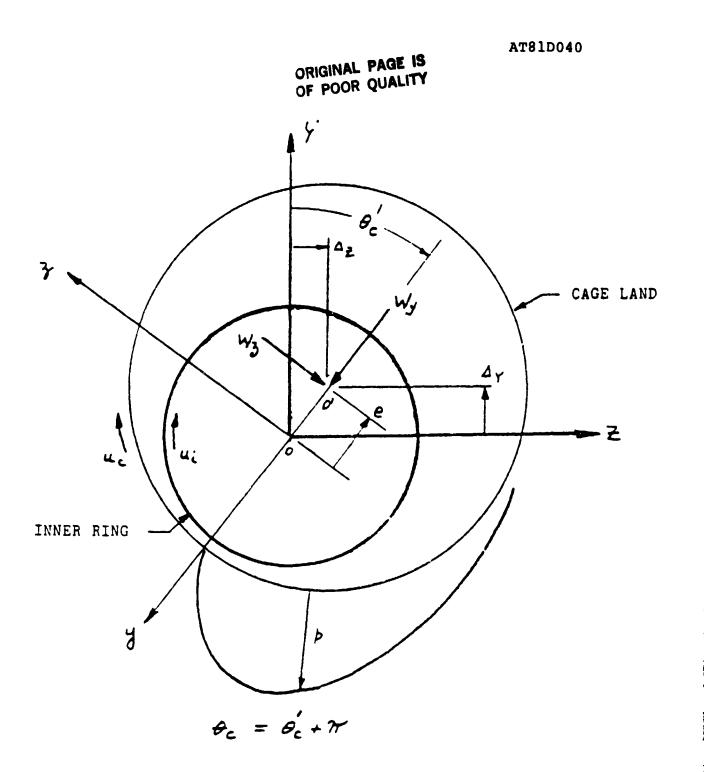


FIGURE 2.5 INNER RING-CAGE LAND CONTACT GEOMETRY

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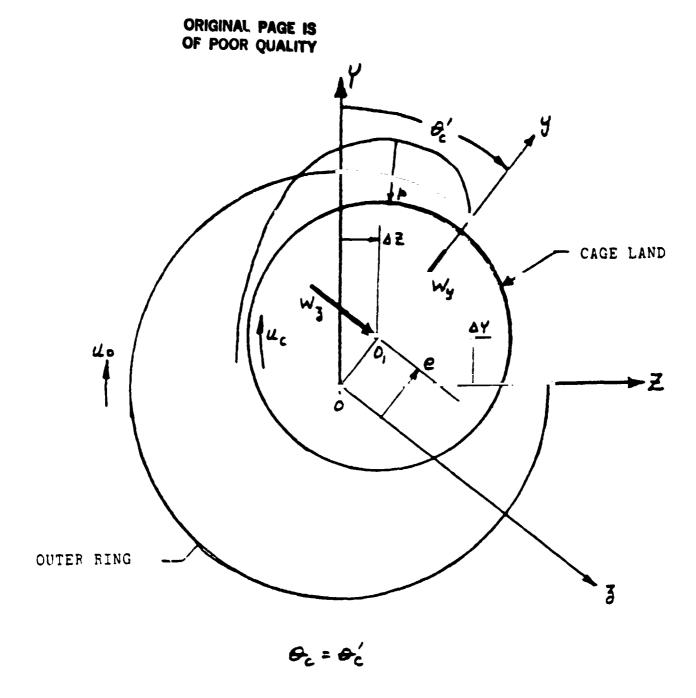


FIGURE 2.6 OUTER RING-CAGE LAND CONTACT GEOMETRY

AT81D040

Occasionally, the program will determine that an input value of three degrees of freedom is impractical, in which case it will override the input and allow the cage to have only one DOF. This occurs if the cage will tend to rotate concentrically with respect to the ring on which it is riding. Such a condition is determined as a function of the rolling element orbital speed variation and prevails with most roller bearings and with ball bearings subjected only to axial loading. In both cases, orbital speed variation is often inconsequential. Also, when the cage rides on the rolling elements, it is allowed only one degree of freedom.

The ball bearing friction solution is thus obtained by solving 6Z+(1 or 3) equations where Z is the number of rolling elements. The ball bearing frictionless solution is obtained by solving 1, (Z/2) (Z/2+1) or Z sets of 2 equations, depending upon the number of rolling elements in the bearing and the degree of load symmetry which prevails. The various symmetry conditions are explained below.

The roller bearing friction solution contains 4Z+(1 or 3) equations and the frictionless solution contains Z/2, Z/2+1 or Z sets of three equations again depending upon the number or rolling elements and whether or not load symmetry exists.

The various load symmetry conditions are as follows. Axial symmetry is utilized if the load is axial only, then only one set of two or three equations is solved for the frictionless case and six rolling element and one cage equilibrium equations are solved when friction is included. All rolling elements are assumed to behave identically.

Radial load symmetry is utilized if the non-axial shaft loading is comprised of only radial components parallel to the Y axis and moment components parallel to the Z axis and the position of the first rolling element is utilized. When this symmetry exists, only half the rolling elements need be considered if the number of rolling elements is even and one half plus one need be considered if the number is odd. Because of inertia terms, radial load symmetry can only be utilized in the frictionless calculations.

If load symmetry is not present, then Z sets of two (ball bearing) or Z sets of three (roller bearing) equations must be solved to obtain the frictionless solution.

As with the steady state temperature mapping scheme, the Newton-Raphson scheme in subprogram SOLV13 is used to solve the sets of equations for each bearing. The iteration scheme terminates when either:

AT81D040

$$\begin{cases} \frac{\Delta X_{(K)i}}{X_{(K-1)i}} & < \\ i=1...n \end{cases}$$
 EPS1 frictionless (2.37)

or Where K and K-1 refer to iteration numbers

Experience has shown that the second criterion is usually responsible for terminating the solution. However, when rolling element loads are extremely large, on the order of 10⁵ Newtons, it becomes difficult to reduce the equation residues to less than 10 Newtons. In those instances, the first criterion usually terminates the iteration scheme.

3. PROGRAM INPUT

3.1 Types of Input Data

A complete set of input data comprises data of four distinct categories. Within these categories, cards which convey specific kinds of information are referred to as card types. Depending on the complexity of the problem, the input data set may contain none, one or several cards of a given type. The categories are listed below.

I. Title Cards

A title card plus a second card which provides the program control information for the shaft-bearing solution.

- II. Bearing Data Cards
 A set of up to sixteeen (16) card types. Each set describes one bearing in the assembly. All bearings must be so described. The card sets must be input sequentially in order of increasing distance from a selected end of the shaft.
- III. Thermal Data Cards
 A set of up to nine (9) card types to describe the thermal model of the assembly.
 - IV. Loading Data Card
 One card describing the loading on a single bearing.
 This card type is used only when Shaft Data Cards are omitted.
 - V. Shaft Data Cards
 A set of three (3) card types to describe the shaft
 geometry, bearing locations on the shaft and shaft loading. Used only when Loading Data Card is omitted.

If the program is being used to predict the performance of a bearing assembly, cards from sets I, II, III, and V must be included in the runstream. If the program is being used to thermally model a mechanical system wherein no bearing calculations need be performed, the cards from sets II, IV and V are omitted.

The review of required input information which follows is broken into the five sets of data categories given above, with special emphasis on program control data.

The input data instructions are given in Appendix C, and are for the most part, self-explanatory. They are laid out in the format of an eighty column data card. A description of the variables is given in the input instruction forms.

The units used for input data are as follows:

Linear Dimensions - (mm)

Angles - (degrees)

Surface Roughness (microns)

Bearing Angular Mounting Errors - (radians)

Rotational Speeds - (RPM)

Force - (Newtons) (N)

Moments - (N-mm)

Pressure, Elastic Modulus - (N/mm²)

Density - (gm/cm³)

Kinematic Viscosity - (cs)

Temperature - (degrees centigrade) (°C)

Coefficient of Thermal Expansion - (°C-¹)

Thermal Conductivity - (Watts/m/°C)

3.2 Data Set I - Title Cards

3.2.1 Title Card 1

This card should contain the computer run title and any information which might prove useful for future identification. The full eighty (80) columns are available for this purpose. The title will appear at the top of each page of Program output.

3.2.2 Title Card 2

This card provides the control information for the shaft bearing solution.

Item 1: Shaft Speed in rpm, GOV (1). All bearings have the same shaft and inner ring speed.

Item 2: Number of Bearings on the shaft (NBRG), a minimum of zero is permitted if no bearing solution is being sought. A maximum of five is permitted. Note that a bearing is defined as a single row of rolling elements. Thus a double row bearing is treated as two separate, single row bearings.

Item 3: Print Flag (NPRINT), NPRINT equal to zero is normal and will result in no intermediate or debug output. With a value of one, a low level intermediate print is obtained at the end of each shaft bearing iteration. The values of the inner ring displacements (DEL), equation residues, bearing inner ring residual loads, and bearing partial derivatives are printed for each iteration. This level is recommended in cases where proper convergence of the bearing solution does not occur.

At the end of each bearing iteration, wherein the rolling element and cage equilibrium equations are solved, an error parameter is printed which has the value:

Error Parameter = $\Delta X_N / X_{N-1}$

 $\Delta \textbf{X}_N$ is the change in the variable X specified at iteration N.

 X_{N-1} is the value of the variable specified at the previous iteration.

The Error Parameter is calculated for each of the bearing variables, but only the largest one is printed.

Additionally, at the end of each Clearance Change iteration, the clearance change error parameter is printed. This error is defined:

Error Parameter = $\frac{DCL_{N} - DCL_{N-1}}{Rolling Element Diameter}$

where DCL_N and DCL_{N-1} denote the clearance changes calculated at the current and previous iterations respectively.

If NPRINT is set at 2 all of the above information is printed. Additionally the variable values and residue values are printed for each iteration of the rolling element and cage equilibrium solution. This level is not generally recommended because of the large volume of output produced.

Item 4: ITFIT controls the number of iterations allowed to satisfy the bearing clearance change iteration scheme. If ITFIT is set to zero (0), or left blank, the clearance change portion of the program is not executed. If a positive integer is input, the clearance change scheme is utilized with a maximum iteration limit of five (5). If a negative integer is input, the scheme is used with a maximum iteration limit equal to the absolute value of the negative integer.

Item 5: ITMAIN limits the number of iterations attempted during the solution of the shaft and bearing inner ring equilibrium problems, i.e. establishing the equilibrium of bearing reactions and applied shaft loads. If ITMAIN is left blank, set to zero, or to a positive integer, then (15) iterations are permitted. If ITMAIN is set to a negative integer the number of iterations is limited to the absolute value of that integer.

AT81D040

Item 6: GOV(2) or EPSFIT is the convergence criterion for the diametral clearance change portion of the analysis. As mentioned under item 3 above, this error parameter is defined by Eq. 2.28.

The iteration scheme is terminated when the error parameter is less than the input value of EPSFIT. If EPSFIT is left blank or is set to zero (0), the program default value of 0.0001 times the rolling element diameter is used.

Items 7 & 8: Main loop accuracy for frictionless elastic (EPS1) and friction solution (EPS2). These accuracy values control the accuracy of the shaft bearing deflection solution as well as the quasi-dynamic solution of the component dynamics. If EPS1 and EPS2 are left blank or set to zero (0), default values of 0.001 and 0.0001 respectively are used.

Item 9: JUSTBR, column 78, is a flag indicating whether or not a single ball or roller bearing is to be analyzed. If a value of 1 is input, data are needed for only one bearing, and loads are input on Loading Data card L1 following the Thermal Data cards. Also, Shaft Data cards are omitted. If JUSTBR is O or blank, up to 5 bearings are analyzed with Shaft Data, and the Loading Data card is omitted.

Item 10: IMT, if set to 1, the material properties for both bearing rings and the rolling elements are to be input on card types B 11 through B 14. If IMT is zero or blank, the rings and rolling elements are assumed to be 52100 bearing steel. Card types B 11 through B 14 are required if the change in bearing diametral clearance is to be calculated or if a system component has properties different from steel.

Item 11: NPASS controls the level of the bearing solution:

- O Elastic Contact Forces are calculated. No lubrication or friction effects are considered.
- 1 Elastic Contact Forces are calculated. Lubrication and friction effects are considered using raceway control (ball bearing) or epicyclic (roller bearing) assumptions to estimate rolling element and cage speeds.

AT81D040

: 1

- 2 Inner Equilibrium is satisfied considering only the Elastic Contact Forces. Using the inner ring positions thus obtained, rolling element and cage equilibrium are determined considering friction.
- 3 Complete Solution. The inner ring, rolling element and cage equilibrium is determined considering all elastic and friction forces.

3.3 Data Set II - Bearing Data

Most of the input instructions are self-explanatory. Where certain items are deemed to require more explanation than given in the input data format instructions they are treated on an individual basis by card type and item number.

Most of the bearing input data is read into a two dimensional array named "BD," which has the dimensions (1830, 5). For each of the five bearings permitted on a shaft, a total of 1830 pieces of data may be stored. Denoting BD(I,J), I represents a specific piece of bearing data, J represents the bearing number. The bearing input data of Data Set II occupies the first 106 locations of the 1830 allotted. On the input data format sheets the designation BD(I) where I=1...106, denotes the location within the BD array where each piece of input data is stored.

\

3.3.1 Card Type 1 - Bearing Type and Material Designations

Item 1: Bearing type, columns 1-10 must be specified, left justified, i.e., "B", "C" or "T" in column 1. This format must be followed since the Program recognition of bearing type, (ball, cylindrical or tapered roller bearing), is derived from reading the "B", "C", or "T" in the first column of this card.

Items 2 & 3: Columns 11-30 and 31-50, "Steel designations", inner and outer rings respectively. The alphameric-literal description of the steel types such as "M-50" or "AISI 52100" is input.

Items 4 & 5: Columns 51-60 and 61-70, the numbers input for items 4 and 5 are used to account for improved materials and multiply the raceway fatigue lives as determined by Lundberg-Palmgren methods. Typical life factor values for modern steels are in the neighborhood of 2.0 to 3.0. If the ASME Publication Life Adjustment Factors for Ball and Roller Bearings, is referenced by the user, the Material Factor D and the Material Process Factor E should be used multiplicatively as inputs for items 4 and 5. The program computes a lubricant life factor based on the value of Λ (EHD plateau film thickness/composite RMS surface roughness). The calculated lube life factor ranges from 0.21 for $\Lambda < 0.6$ to 3.0 for $\Lambda > 10.0$.

Item 6: Columns 71-78, "Orientation angle of the first rolling element". (ϕ 1) (degrees). Refer to Fig. (2.4). The quasi-dynamic rolling element bearing problem has an infinite number of solutions which fall within a narrow envelope having a periodic shape. The solution obtained is a function of the rolling element positions relative to the bearing system coordinate axes. ϕ_1 = 0 places a rolling element on the Y axis and is the choice customarily made. ϕ_1 can be desig-

nated as any value $0<\phi_1<360/Z$ where Z is the number of rolling elements. For each different value assigned to ϕ_1 a different ent, although similar, bearing solution will be obtained. To take advantage of bearing symmetry and the computer time savings which result, ϕ_1 must be specified as zero or left blank.

Item: 7 Column 80, a signal, termed the crown drop flag, which specifies for a cylindrical or tapered roller bearing, whether the roller-race crown drops will be calculated, or read directly. If item 7 is blank or zero, the crown drops are calculated based on the roller-race crown radius, and effective flat length input information. If the crown drop flag is other than zero or blank the non-uniform separation of the roller and raceway must be specified at the center of each slice into which the roller-raceway effective contact length is divided. The slice widths are identical. The number of slices is input as item 7 card type B4. The non-uniform roller-raceway separation is input on card types B5 and B6.

3.3.2 Card Type B2 - Bearing Geometry and Outer Ring Speed

3.3.2.1. Ball Bearing Geometry

Items 1, 2 and 6 are self explanatory. Item 5 pertains only to a tapered roller bearing, as discussed later. Items 3 and 4 require explanation however.

Through the proper specification of the diametral clearance and contact angle, the Program can properly handle deep groove, split inner, and angular contact ball bearings.

The deep groove ball bearing requires the specification of the contact angle corresponding to either the operating diametral clearance Pd or the off the shelf diametral clearance, if the dimensional change analysis is utilized.

The angular contact bearing is fully described through specification of the contact angle which obtains under a gauge, axial load. However, this method of input does not accurately define the system if there is more than one angular contact supporting the shaft and at least one of those bearings has its grooves offset in the direction opposite to the other bearings and if the shaft is capable of axial and/or radial play. In other words, if, what are known as angular contact ball bearings, are mounted such that some diametral shaft play is permitted, an auxilliary angle as well as the diametral play must be specified at input. The angle input is not the

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manufacturer's designated contact angle, α , but an auxilliary angle, α_0 , the calculation for which shall be demonstrated.

Refer to Figure 3.1. The manufacturer's contact angle is calculated as follows:

$$\alpha = Cos \left[\frac{2A - Pd}{2A} \right]$$
 (3.1)

$$A = r_0 + r_i - D \tag{3.2}$$

where: r_0 and r_i are the outer and inner raceway groove radii respectively

D is the ball diameter

Under a gauge axial load, α is obtained at both inner and outer raceways for each ball. Under this condition, the outer and inner raceways are axially offset an amount S_α .

$$S_{\alpha} = A \sin \alpha$$
 (3.3)

When angular contact ball bearings are mounted with some diametral play, the grooves are offset an amount $S\alpha_0$ such that $S_{\alpha 0} < S_{\alpha}$. The diametral play which obtains at this condition is Sd. This diametral play is usually known by the engineer or designer and is usually required to allow some forgiveness when thermal gradients are encountered. Assuming that the user has the values for α , r_0 , r_1 , D and $S_{\alpha 0}$ then:

$$\alpha_0 = \tan^{-1} \left[\frac{3\alpha_0}{A - \frac{3\alpha}{2}} \right]$$
 (3.4)

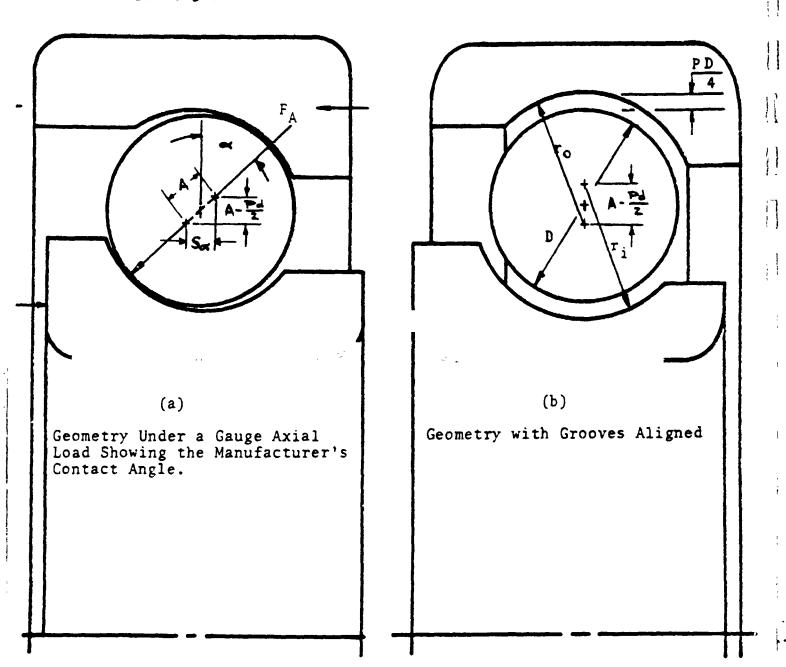
where: Pd and A may be calculated from Eqs. (3.1) and (3.2).

If S_{α_0} is unknown, the following equation may be solved for $_{\alpha_0}.$

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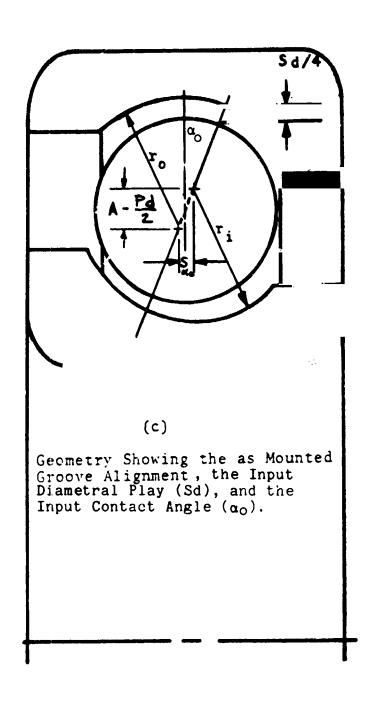
FIGURE 3.1 ANGULAR CONTACT BALL BEARING GEOMETRY



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FIGURE 3.1 ANGULAR CONTACT BALL BEARING GEOMETRY (CONTINUED)



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$$\alpha_0 = \cos^{-1} \left[\frac{2A - Pd + Sd}{2A} \right] \tag{3.5}$$

In order that the Program properly handle split inner ring ball bearings an auxiliary angle and diametral play must be input. Referring to Figure 3.2, the auxiliary angle α_0 and diametral play Sd must be determined and input. Typically the values of D, r_0 , r_i , α_S and Sd' (assembled bearing diametral play) are known.

The unloaded half of the inner ring must be removed from consideration and the ball moved such that its center lies on the line connecting the origins of r_i and r_o and positioned such that the auxiliary clearance Sd/4 exists at both the inner and outer raceways. The auxiliary angle is given by:

$$\alpha_0 = \tan^{-1} \left[\frac{(r_i - D/2)\sin \alpha_s}{r_0 - D/2 - Sd^{1/2} + (r_i - D/2)\cos \alpha_s} \right]$$
 (3.6)

The input bearing diametral play, Sd, can then be calculated as follows:

$$Sd = Sd' + (2r_i - D)(1 - \cos \alpha_S) - 2A(1 - \cos \alpha_O)$$
 (3.7)

The angle associated with each ball bearing must be specified with the correct sign. A positive contact angle allows the bearing to accept a positively directed axial load transmitted by the shaft.

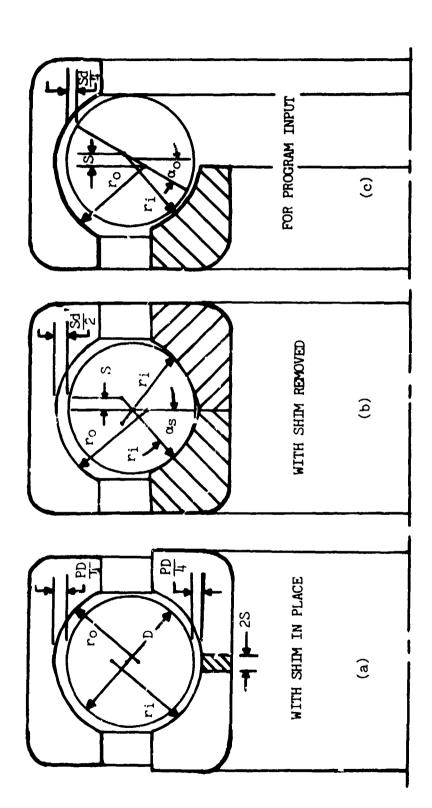


FIGURE 3.2 SPLIT INNER RING BALL BEARING GEOMETRY

39

3.3.2.2 <u>Tapered Roller Geometry</u>

Items 2 and 6 are self explanatory.

Item 1: Bearing Pitch Diameter

The tapered roller bearing pitch diameter may be calculated through specification of the roller measured large end diameter, the roller large end corner radius, one half the cup included angle, the roller included angle and the roller total length. This calculation is performed within the program. The user therefore need not specify the tapered roller bearing pitch diameter.

Item 3: Bearing Axial Play

For a tapered roller bearing, the bearing axial play rather than diametral clearance must be specified. (See Fig. 3.3) Note that this end play pertains to only the bearing in question. For two identical tapered roller bearings on a shaft, one half the total shaft axial play should be specified for each bearing. For two tapered roller bearings of dissimilar size, the total shaft axial play should be apportioned according to the bearing size such that the sum of the axial plays specified for the two bearings equals the total for the shaft.

Item 4: Bearing Contact Angle

For the tapered roller bearing, one half the included cun angle (a) is input as the contact angle. (See Fig. 3.3)

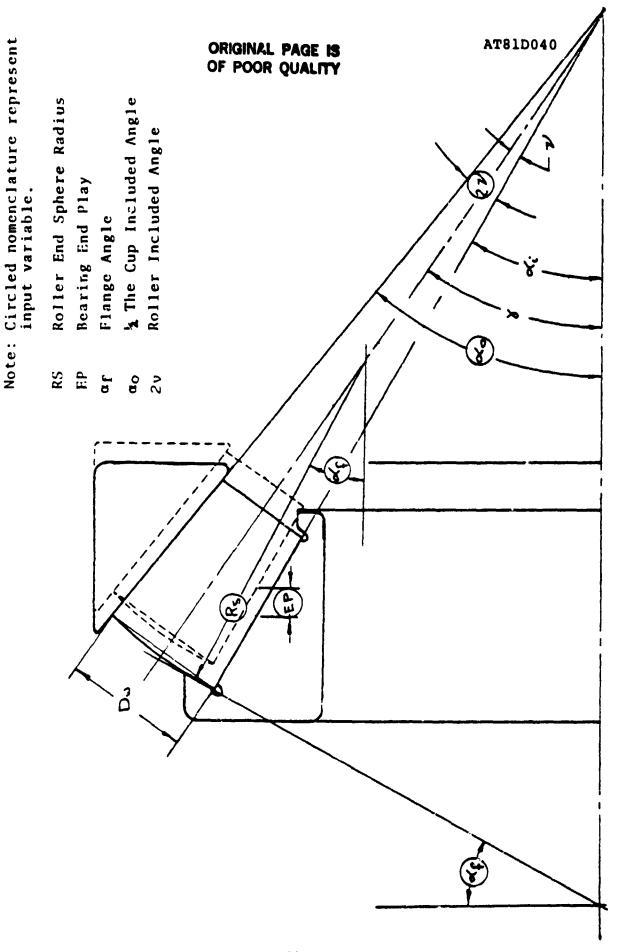
This angle must be specified with the correct sign. A positive angle allows the bearing to accept a positively directed axial load transmitted by the shaft and vice versa for a negative angle.

Item 5: Tapered Roller Bearing Flange Angle

The flange angle is shown by α_f in Fig. 3.3. The flange angle must always be positive.

3.3.2.3 Cylindrical Roller Bearing Geometry

Items 1, 2, 3 and 6 are self explanatory. Both items 4 and 5 should be left blank.



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3.3.3 Card Type B3 - Rolling Element Geometry

3.3.3.1 Ball Geometry

The geometry of a ball is fully defined by its diameter.

3.3.3.2 Tapered Roller Geometry

Item 1 - Roller, Measured Large End Diameter

For a tapered roller, the input diameter required is the largest measureable diameter. (See Fig. 3.4.) Typically this measurement should be taken where the large end corner radius becomes tangent to the roller surface profile. Within the program a "working" large end diameter is calculated. This diameter is shown as $D_{\rm w}$ in Fig. 3.4. All bearing geometrical relationships are calculated based on $D_{\rm w}$.

(3.8)

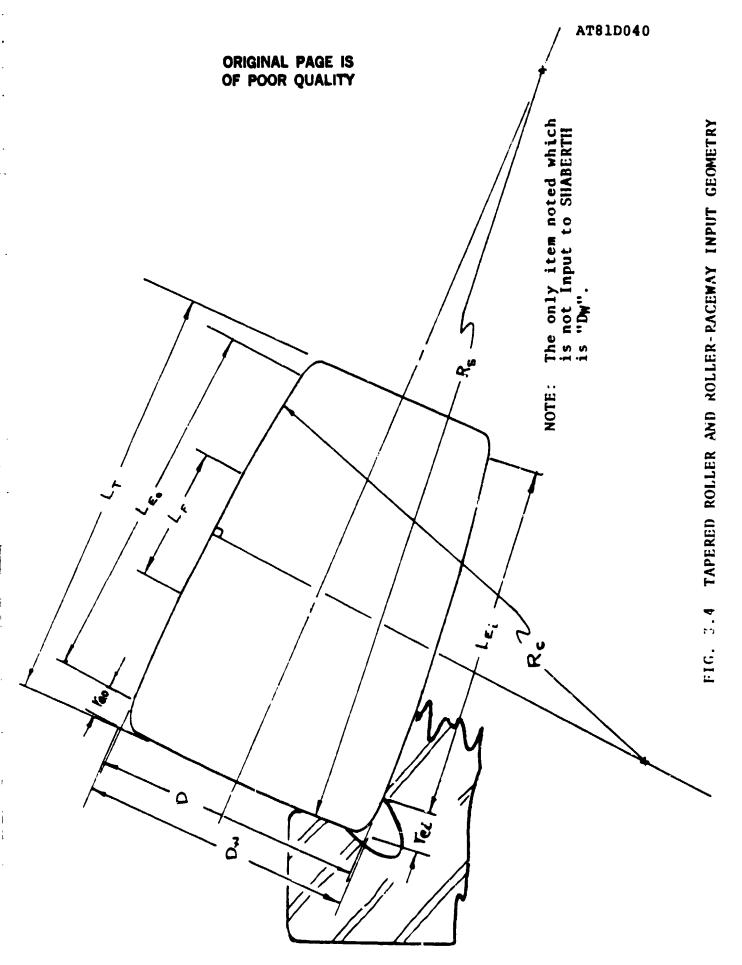
11

where: D is the measured large end diameter, reo is the distance from the roller end to the beginning of the roller effective length at the outer raceway surface, measured parallel to the roller surface. If ro is the corner radius at the roller large end

$$\Gamma_{\bullet O} = \frac{\Gamma_{\bullet} (1 + \sin \nu)}{\cos \nu}$$
 (3.9)

v is one half the roller included angle.

Item 2 through 6 are shown in Fig. 3.4 and are self explanatory. Note that the program can handle a nonzero roller flat length, Item 6. Most tapered rollers are, however, fully crowned.



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3.3.3.3 Cylindrical Roller Geometry

Typically, cylindrical rollers are partially crowned as shown in Fig. 3.5. The center of the roller is flat. Toward the ends the roller profile is formed by a crown radius, R. There are usually rounded corners at the roller ends. These corners reduce the load carrying surface of the roller such that if there are no raceway undercuts the roller raceway effective length equals the roller total length less the two corner radii (see Section 3.3.4.2). Note that a partially crowned roller is specified through input of a non-zero flat length. If the flat length is zero, the roller is fully crowned with its profile defined by the roller crown radius.

The roller end sphere radius, R_s, is defined in Fig. 3.5. For a cylindrical roller, the roller included angle is zero.

3.3.4 Card Type B-4 - Rolling Element-Ring Geometry

3.3.4.1 Ball Bearing

Items 1 and 2 refer to the outer and inner raceway curvatures respectively where curvature is defined as the cross groove radius divided by the ball diameter. Typical values range from 0.515 to 0.57.

3.3.4.2 Tapered Roller Bearing Contact Geometry

Items 1 and 2 - Roller Raceway Effective Length

The roller-raceway load bearing surface is measured parallel to the roller surface such that if there were no relief at the roller ends the effective contact length would be:

$$L_{\Omega}^{\star} = L/\cos v \tag{3.10}$$

However, since the roller has corners at the large and small ends, the actual effective length is less than La. The consideration of raceway undercuts at the inner raceway flanges may result in a raceway effective length less than the roller effective length in which case the shorter of the two should be input.

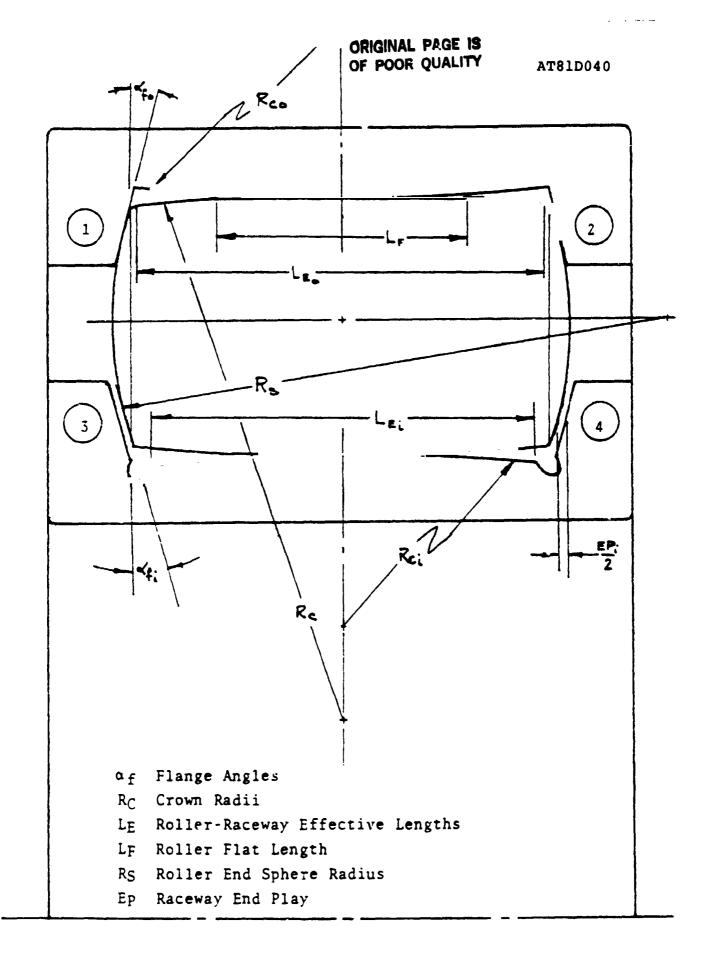


FIG. 3.5 CYLINDRICAL ROLLER BEARING GEOMETRY

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Items 3 and 4 - Raceway Crown Radius

The present analysis permits both the roller and raceway to be crowned. If the raceway is crowned, it must be fully crowned with no flat specified. If the raceways are flat the input crown radius may be left blank, in which case a default value of 1. \times 10⁺ 10 inches is used.

Note that the unloaded roller-raceway separation along the roller profile, (δc) Fig. 3.6, calculated at the center of each roller raceway slice is comprised of the sum of the roller and raceway crown drops.

Items 5 and 6 - Roller Large End Corner Relief

These data specify the distance from the roller large end to the point on the roller surface where the roller effective length begins. For the outer raceway contact this distance may be calculated using Eq. 3.9. For the inner raceway use Eq.3.9 or the width of the inner raceway undercut at the large end. (Fig. 3.4)

Item 7 - The number of slices into which the roller raceway contacts are divided.

A maximum value of twenty (20) is permitted. A default value of eleven (11) is used if a blank or zero is read.

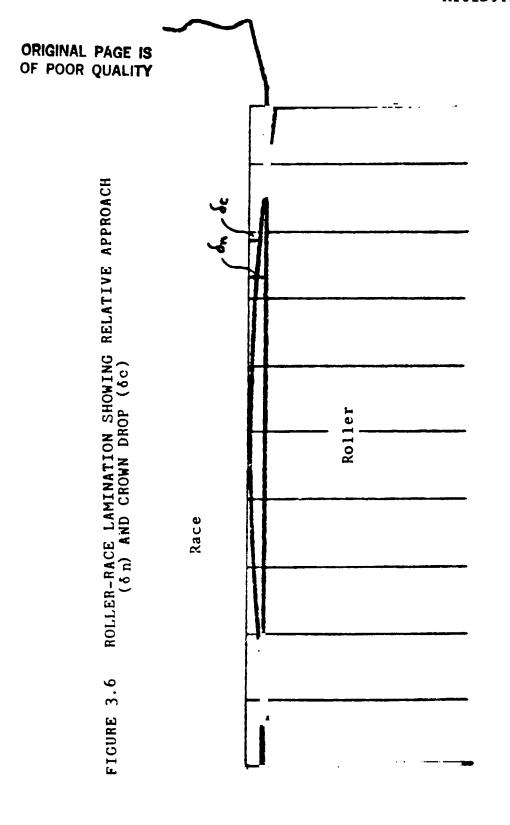
3.3.4.3 Cylindrical Roller Contact Geometry

3.3.4.3.1 Card Type B4A Roller Raceway Geometry

Items 1 through 4 and 7 have the same definitions as they had for the tapered roller bearing.

Items 5 and 6 - Roller End Corner Relief is not required input for the cylindrical roller bearing since the roller raceway effective length is assumed to be centered along the roller. This was not the case for the tapered roller bearing.

3.3.4.3.2 <u>Card Type B4B Roller Flange Geometry for</u> Cylindrical Roller Bearings



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Items 1 - 4 - The flange angle and end play definitions can be seen in Fig. 3.5. Note that when a ring has a single flange such as Inversion No. 3. Fig. 3.7, the end play is the distance between the roller and the flange when the roller is centered on the raceway.

Item 5 - The Flange Inversion Index - See Fig. 3.7
The number which corresponds to the particular Flange inversion being examined must be input. Note that since the inversions greater than eight (8) cannot carry axial loading, the bearing carries load only on the raceways, and thus the Program resets the inversion index to one if the input value is greater than eight. The inversion index must be input as a real number, (with a decimal point).

3.3.5 Roller-Raceway Non-Uniform Profile Definition

3.3.5.1 Card Type B5 - Outer Raceway Roller Contact

These cards are used to input the separation between the cute: raceway & roller at the center of each slice along the roller profile with the high points of the roller and race in contact, i.e., with all clearance between roller and raceway removed. These cards must be omitted if item 7 of the Bearing Data Title card is zero or blank. These data are used only when the roller-raceway profile geometry cannot be defined by card types B3 and B4A.

5.3.6 Card Type B6 - Inner Raceway Roller Contact

Same as Card Type B5 for the inner raceway-roller contact.

3.3.7 Ring-Rolling Element Surface Data

3.3.7.1 Card Type B7A - Raceway - Rolling Element Surface Data

Items 1 through 6 define the statistical surface microgeometry parameters of the rollers and raceways. Items 1 through 3 require the input of center line average CLA surface roughness. Within the program CLA values are converted to RMS by multiplying by 1.25.

Items 4 through 6 are RMS values of the slopes measured in degrees, of the surface asperities as measured in a traverse across the groove for rings, longitudinally for rollers and in any arbitrary direction for balls. Typical values for raceway and rolling element surfaces are 1 to 2 degrees. This card is omitted if the solution level is NPASS = 0.

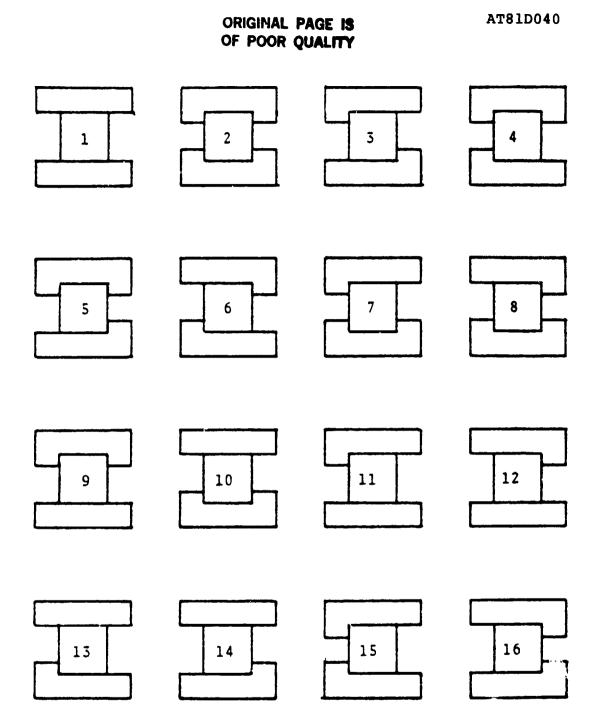


FIG. 3.7 CYLINDRICAL ROLLER BEARING
FLANGE INVERSIONS

3.3.7.2 Card Type B7B - Flange-Roller End Surface Data

The data are identical to the data in Card Type B7A but refer to the flange and roller end surfaces rather than the raceway and roller rolling surfaces.

Note that both card types B7A and B7B are omitted if the solution level is NPASS = 0.

3.3.8 Card Type B8 - Cage Data

This card is omitted if the solution level is NPASS =0. These data are self-explanatory. Note that the cage weight is an input item. It is included mainly for future consideration of cage stability predictions. A nonzero value should be input to avoid divide checks.

The number of degrees of freedom of the cage, MCG, is also an input item (either 1 or 3). When NPASS=0, cage DOF is defaulted to 1. See Section 2.4.1, Cage Degrees of Freedom.

3.3.9 Card Type B9 - Shaft and Housing Fit Dimensions

These cards are to be included only if the change in bearing diametral clearance with operating conditions is to be calculated, i.e. if item 4 ITFIT on the Bearing Title Card is non-zero. On Card Type B9, tight interference fits bear a positive sign and loose fits, a negative sign.

Item 3 and 6 on Card No. 9 are termed the shaft and housing effective widths respectively. The value specified for these effective widths may be as great as twice the ring width.

Use of an effective width is an attempt to account for the greater radial rigidity of a shaft being longer than the ring that is pressed on to it, owing to the fact that the shaft deflects over a distance that extends beyond the ring width. In the program the calculated internal pressure on the ring due to its interference fit with the shaft, is distributed over the shaft effective width and this (lower) pressure is used in computing the shaft deflection. Using double the actual width as the effective width is customary.

3.3.10 Card Type B10 - Shaft&Housing Fit Dimensions

These items are self explanatory.

Note: Bearing System Components Material Properties. Card types B11 through B14.

These card types define the material properties of the shaft, inner ring, rolling element, outer ring and housing, data items 1 through 5 respectively. This set of cards is to be included if either the bearing clearance change analysis is used, i.e. item 4, ITFIT, Bearing Title Card 2 is non zero, or if the bearing rings or rolling elements are not steel, i.e. item 10, IMT, Bearing Title Card 2 is equal to 1. If any item on card types B11 through B14 is left blank, the program inserts the appropriate value of the steel property.

3.3.11 Card Type Bll - Elastic-Moduli

This card defines the elastic modulus for the shaft, inner ring, rolling element, outer ring, and housing respectively. A default value of 204083 N/mm² (29.6x10⁰ PSI) is used.

3.3.12 Card Type B12 - Poisson's Ratio

This card defines the Poisson's ratio for the shaft, inner ring, rolling element, outer ring, and housing respectively. A default value of 0.30 is used.

3.3.13 Card Type B13 - Density

This card defines the density for the shaft, inner ring, rolling element, outer ring, and housing respectively. A default value of 7.806 g/cm³ (0.282 lb/in³) is used.

3.3.14 Card Type B14 - Coefficient of Thermal Expansion

This card defines the coefficient of thermal expansion for the shaft, inner ring, rolling element, outer ring, and housing respectively. A default value of 12.24×10^{-6} $1/^{\circ}$ C (6.8 X 10^{-6} $1/^{\circ}$ F) is used.

3.3.15 Card Type B15 - Lubrication and Friction Data

This card is omitted if the solution level is NPASS = 0.

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Items 1 and 2

Items 1 and 2 are the amounts by which the combined thickness of the lubricant film on the rolling track and rolling element is increased during the time interval between the passage of successive rolling elements, from whatever replenishment mechanisms are operative. Item 1 applies to the outer and Item 2 to the inner race-rolling contacts respectively. If Item 1 is zero or blank the mode of friction is assumed to be dry.

At the present time the magnitude of the inner and outer raceway replenishment layers has not been correlated to lubricant flow rate, lubricant application methods and bearing size and speed factors. At this point then, the user is forced to establish proper values for the replenishment layer thickness. As a rough guide the following suggestions are made.

- 1) To avoid starvation, the replenishment layer thicknesses should be one to two times the EHD film thickness which develops in the rolling element raceway contacts.
- 2) Because of centrifugal force effects, intuition suggests that the outer raceway replenishment layer should be several times thicker than that prescribed at the inner raceway.

Item 3, XCAV, describes the percentage of the bearing cavity, estimated by the user to be occupied by the lubricant. $0 \le XCAV \le 100$.

As with the replenishment layer thicknesses, the amount of free lubricant should be able to be correlated with lubricant flow rate, lubricant application methods and bearing size and speed factors. At this time such correlations do not exist. XCAV values less than five percent are recommended.

Item 4 is the coefficient of coulomb friction applicable for the contact of asperities. If Item 1 and 2 are zero, then Item 4 serves as the coulomb friction coefficient which prevails in all contacts.

Items 5 and 6 are the lubricant replenishment layer thicknesses for the outer and inner ring flanges respectively. These items should be left blank for ball bearings it the coefficient of coulomb friction applicable for the asperity interactions at the roller end-flange contacts. This value should also be left blank for ball bearings.

3.3.16 Card Type B16

This card is omitted if NPASS title card 2 is zero or blank or if Item 1 card B15 is zero or blank which implies dry friction.

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This card specifies the lubricant type. If Item 1, NCODE is 1, 2, 3, or 4, the Program uses preprogrammed lubricant properties as presented in Table 1. and no further information is required.

NCODE	Lubricant
1	A specific mineral oil
2	A MIL-L-7808G
3	Polyphenyl-Ether
4	A MIL-L-23699

NCODE may also be specified as negative (-1 to -4), in which case the traction characteristics of the respective lubricant NCODE noted above are used but the actual properties specified by Items 2 through 9 override the hard coded data. This option is most useful in specifying various mineral oils i.e. NCODE = -1. If items 8 and 9, AKN and FRIC are left unspecified, default values are set at 50.0 and 0.07, respectively. AKN and FRIC are only used when the NASA version of the code is exerted (see Appendix F).

3.4 DATA SET III - THERMAL MODEL DATA

Appendix B has been included to aid the user in data preparation and calculation of heat transfer coefficients required at input.

3.4.1 Card Type Tl

Card type Tl is a control card. If no temperature map is to be calculated, this card is to be included as a blank card followed by a Type T2 card for each bearing on the shaft. Card Type Tl contains control input for both steady state and transient thermal analyses. It is not intended, however, that both analyses be executed with the same run.

- Item 1: The highest node number (M). The temperature nodes must be numbered consecutively from one (1) to the highest node number. The highest node number must not exceed one hundred (100).
- Item 2: Node Number of the Highest Unknown Temperature Node (N). This number should equal the total number of unknown node temperatures. It is required that all nodes with unknown temperatures be assigned the lowest node numbers. The program assumes that all node numbers greater than N (from N+1 to M) represent known boundary temperatures.
- Item 3: Common Initial Temperature (TEMP) C: The temperature solution iteration scheme requires a starting point, i.e., guesses of the equilibrium temperatures. Card Type T3 allows the user to input guesses of individual node temperatures. When a node is not given a specific initial temperature, the temperature specified

OIL NO.	OIL	KINEMATIC VISCOSITY (CS)	ATIC SITY	WALTHER EQUATION CONSTANTS	EQUATION ANTS	DENSITY @ 600F	THERMAL	THERMAL	TEMPERATURE VISCOSITY
		100°F 37.8°C	2100F 1000C	V	æ	p GM/CM3	KF WATT/M- ^o c	OF EXPANSION G 1/OC	COEFFICIENT B 1/oc
1	Mineral Oil	h9	8.0	10.349	3.673	0.8800	0.116	6.34×10-4	0.0347
2	MIL-L-7808G	12.8	3.2	10.215	3.698	0.9526	0.152	7.09x10-4	0.0238
٣	C-Ether	₹. 22.4	4.13	11.452	4.113	1.20]	0.119	7.47×10-4	0.0302
=	MIL-L-23699	28.0	5.1	10.207	3.655	1.010	0.152	7.45x10-4	0.0290

TABLE 1

LUBRICANT PROPERTIES OF FOUR OILS USED

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as Item 3 of Card Type Tl is assigned.

Item 4: Punch Flag (IPUNCH): If the Punch Flag is not zero (0) or blank, the system equilibrium temperatures along with the respective node numbers will be punched according to the format of Card T3. This option is useful if, for instance, the user makes a steady state run with lubrication, and then wishes to use the resultant temperatures as the initiation point for a transient dry friction run in order to assess the consequence of lubricant flow termination.

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Item 5: "Output Flag" (IUB). If the "Output Flag" is not zero the bearing program output and a temperature map will be printed after each call to the shaft bearing solution scheme. This printout will allow the user to observe the flow of the solution and to note the interactive effects of system temperatures and bearing heat generation rates. Two levels of bearing cutput are permitted. If IUB is 1, the rolling element output is not printed. If IUB is 2, full bearing output is obtained.

Item 6: "Maximum Number of Calls to the Shaft Bearing Program" (IT1). If I is the limit on the number of Thermal-Shaft-Bearing iterations, i.e., the external temperature equilibrium calculation. The user must input a non-zero integer such as 5 or 10 in order for the Program to iterate to an equilibrium condition. If IT1 is left blank or set to zero (0) or 1, shaft bearing performance will be based on the initially guessed temperatures of the system. The temperatures printed out will be based on the bearing generated heats. It is unlikely that an acceptable equilibrium condition will be achieved. However, the temperatures which result may provide better intial guesses, for a subsequent run, than those specified by the user.

IT1 also serves as a limit on the transient temperature solution scheme, by limiting the number of times the shaft-bearing-solution scheme is called. Each call to the shaft-bearing scheme will input a new set of bearing heats to the transient temperature scheme until a steady state condition is approached or until the transient solution time up limit is reached.

Item 7: "Absolute Accuracy of Temperatures for the External Thermal Solution" (EPI). In the steady state thermal solution scheme, each calculation of system temperatures occurs after a call to the shaft-bearing scheme which produces bearing generated heats. After the system temperatures have been calculated for each iteration, using the internal temperature solution scheme, each node temperature is checked against the nodal temperature at the previous iteration.

If $\{t_{(N)i} - t_{(N-1)i}\} \le EP1$ for all nodes i then equilibrium has been achieved and the iteration process stops.

 $t_{(N)i}$ = temperature of ith node at Nth thermal iteration.

 $t_{(N-1)i}$ = temperature of ith node at N-1 th thermal iteration.

Item 8: "Iteration Limit for the Internal Thermal Solution" (IT2). After each call to the shaft bearing program, the internal temperature calculation scheme is used to determine the steady state equilibrium temperatures based on the calculated set of bearing heat generation rates. If the program is used to calculate the temperature distribution of a non bearing system it is the internal temperature scheme which is employed. If IT2 is left blank or set to zero, the number of internal iterations is limited to twenty (20).

Item 9: "Accuracy for Internal Thermal Solution" (EP2). The use of EP2 is explained in Section 2.1.1. If EP2 is left blank or set to zero (0), a default value of 0.001 is used.

Item 10: "Starting Time" (START) is a time T_S at which the transient solution begins; usually set to zero (0).

Item 11: "Stopping Time" (STOP) is the time in seconds at which the transient solution terminates, Tf. The transient solution will generate a history of the system performance which will encompass a total elapsed time of

$$(T_f - T_s)$$
 seconds

Item 12: "Calculation Time Step" (STEPIN). The transient internal solution scheme solves the system of equations

$$t_{k+1} = t_k + \frac{q_k}{\rho C_p V} \Delta T$$

$$\Delta T = STEPIN$$
(3.11)

The user may specify STEPIN. If 1-ft blank or set to zero (0), the Program calculates an appropriate value for STEPIN using the procedure described in Section 2.1.2.

Item 13: "Time Interval Between Printed Temperature Maps" (TTIME) seconds. The user must specify the length of time which will elapse between each printing of the temperature map. The interval will always be at least as large as the "calculation timestep" (STEPIN).

Item 14: "Time Interval Between Calls of the Shaft Bearing Portion of the Program" (BTIME). BTIME will always have a value larger than or equal to (STFPIN) even if the user inadvertently inputs a shorter interval. Computational time savings result if BTIME is greater than STEPIN, however, accuracy might be lost.

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3.4.2 Card Type T2

Card Type T2 is required, one card for each bearing if no thermal analysis is being performed. The temperature data is used within the shaft-bearing analysis portion of the program to fix temperature dependent properties of the lubricant in which case the inner race, outer race, lubricant bulk cavity and flange temperatures are used. Note the flange numbering scheme depicted in Fig. 3.5 for a cylindrical roller bearing. For a tapered roller bearing, the inner ring flange is considered flange No. 1. The assembly component temperatures at each bearing location are used in the analysis which calculates the change in bearing diametral clearance from "off the shelf" to operating conditions.

3.4.3 Card Type T3

In the steady state analysis this card is used to input initial guesses of individual nodal temperatures for unknown nodes as well as the constant temperatures for known nodes, such as ambient air and/or an oil sump.

In the transient analysis, Card Type T3 is used to input the nodal temperatures of all nodes at (START) = T_S i.e. at the initiation of the transient solution.

3.4.4 Card Type T4

With this card, node numbers are assigned to the components of each bearing, one card per bearing. With this information the proper system temperatures are carried into each respective bearing analysis. The inner race and inner ring node numb>rs may or may not be the same at the user's discretion. Similarly the outer race and outer ring node numbers may or may not be the same.

3.4.5 Card Type T5 & T5A

Card Type T5 is required, one card per bearing, if a thermal analysis is to be performed. This card designates two nodes to share equally each of the various types of bearing generated heat calculated internally by the program. For cylindrical and tapered roller bearings an additional card, T5A, follows immediately after card T5 specifying the two nodes which will equally share the heat generated at each of the flange contacts. For ball bearings card T5A must be omitted.

3.4.6 Card Type T6

This card specifies the node numbers and the heat generation rate for those nodes where heat is generated at a constant rate such as at rubbing seals or gear contacts.

3.4.7 Card Type T7

This card type is used to input the numerical values of the various heat transfer coefficients which appear in the equations for heat transfer by conductivity, free convection, forced convection, radiation and fluid flow. Up to ten coefficients of each type may be used. Separate values of each type of coefficient are assigned an index number via card T7. In describing heat flow paths (Card Type T8 below) it is necessary only to list the index number by which heat transfers between node pairs.

Indices 1-10 are reserved for the conduction coefficient λ , 11-20 for the free convection parameters, 21-30 for forced convection, 31-40 for emissivity and 41-50 for fluid flow (product of specific heat, density and volume flow rate).

As an example, for heat transfer by conduction with coefficient λ of 53.7 watts/M°C one could prepare a card type T7 with the digit 1 punched in column 10 and the value 53.7 punched in the field corresponding to card columns 11-20. If a conduction coefficient of 46.7 were applicable for certain other nodes in the system one could punch an additional card assigning index No. 2 to the value λ = 46.7 by punching a "2" in card column 10 and 46.7 anywhere within card columns 11-20.

Rather than inputting constant forced convection coefficients, optionally, these coefficients can be calculated by the program in one of three ways. If the calculation option is exercised a pair of cards is used in place of a single card containing a fixed value of α . The contents of the pair of cards depends upon which of the three optional methods are used.

Option 1) α is independent of temperature but is calculated as a function of the Nusselt number Nu whith in turn is a function of the Reynolds number Re, the Prandtl number P_r as follows, (cf. [22])

 $\alpha = Nu \lambda oil/L$

$$N_u = aR^b P^c$$

where λ_{oil} is the lubricant conductivity, L is a characteristic length (with a unit of meters) and a and b, and c are constants.

Option 2) α is a function only of fluid dynamic viscosity and viscosity is temperature dependent.

$$\alpha = c\eta^d$$

Option 3) a is a function of the Nusselt, Reynolds and Prandtl numbers and viscosity is temperature dependent.

3.4.8 Card Type T8

This card defines the heat flow paths between pairs of nodes. Every node must be connected to at least one other node, i.e., two or more independent node systems may not be solved with a single Program execution.

The calculation of heat transfer areas is based on lengths, L₁ and L₂ input using Card Type T8. Additionally, the type of surface for which the area is being calculated is indicated by the sign assigned to the heat transfer coefficient index. If the surface is cylindrical or circular the index should be positive, if the surface is rectangular the index should be input as a negative integer.

In the case of radiation between concentric axially symmetric bodies, L_3 is the radius of the larger body. For radiation between two parallel flat surfaces or for conduction between nodes, L_3 is the distance between them.

Fluid flow heat transfer accounts for the energy which the fluid transports across a node boundary. Along a fluid node at which convection is taking place, the temperature varies. The nodal temperature which is output is the average of the fluid temperature at the output and input boundaries. If the emerging temperature of the fluid is of interest, it is necessary to have a fluid node at the fluid outlet. At this auxiliary node only fluid flow heat transfer occurs and the fluid temperature would be constant throughout the node. Thus the true fluid outlet temperature will be obtained.

Conduction of heat through a bearing is controlled by index 51. The actual heat transfer coefficient which contains a conductivity, area and a path length term is calculated in the bearing portion of the program. The term is based upon conduction through an average outer race and inner race rolling element contact.

3.4.9 Card Type T9

This card inputs data required to calculate the heat capacity of each node in the system. This card type is required only for a transient analysis and must be omitted for a steady state analysis.

3.5 DATA SET IV - LOADING DATA

If a single ball or cylindrical roller bearing is to be analyzed, the user need not model the shaft geometry. In such a case, this loading card may replace all Shaft Input Data (Data Set V).

Data describing a dummy shaft is generated within the program.

Applied loads, acting through the center of the bearing, are input on this card. These loads may have the form of concentrated radial forces (FY, FZ), concentrated moments (MY, MZ) and a concentrated axial load (FX).

If more than one bearing is to be analyzed, this card is omitted, and loads are supplied with the Shaft Input Data.

3.6 DATA SET V - SHAFT INPUT DATA

The shaft-bearing analysis requires all lo. ing to be applied to the shaft. The loads applied to each beari are a product of the shaft-bearing solution. There is no need to the user to solve the statically determinate or indeterminate system for bearing loads.

In the analysis the housing is assumed to be rigid. Provision has been allowed to input data for housing radial and angular spring characteristics. However, this has been done for future consideration of an elastic housing and is therefore currently unavailable.

The shaft input data consists of three card types:

- 1) Shaft Geometry and Elastic Modulus Data
- 2) Bearing Position and Mounting Error Data
- 3) Shaft Load Data

3.6.1 Card Type S1

This card type is used to describe shaft geometry at up to twenty locations along the shaft. The user must place his shaft in a cartesian coordinate system with the end of the shaft at the origin and with the shaft lying along the X-axis.

The shaft is described by specifying two outer and two inner diameters at each axial location along the shaft which define the diameters immediately to the left and the right of the X-coordinate. In this way stepwise or linear variations of the shaft can be handled. A linear variation in shaft diameter is assumed if the diameter to the left of one axial location is different than the diameter to the right of the preceding axial location. Complex shaft geometries may be approximated with a set of linear diameter variations spaced at close intervals.

If an Elastic Modulus is not specified at the designated input location, the modulus of steel is assumed, 204083N/mm.

3.6.2 Card Type S2

This card type locates the bearing inner ring on the shaft in the X-Y and X-Z planes. For a ball bearing, the X coordinate specified locates the inner ring center of curvature. For cylindrical roller bearings the X coordinate locates the center of the inner race roller path.

For tapered roller bearings in the strictest sense the X coordinate locates the point where a line from the roller center of gravity, intersects and is perpendicular to the inner raceway, with all bearing end play removed. It is sufficiently accurate however to allow the X coordinate to locate the center of the inner raceway.

In addition to specifying bearing location, the Type 2 card is also used to specify housing radial and angular mounting errors. As mentioned previously, space has been reserved for inputting housing radial and angular spring characteristics, however, these characteristics are not used in the system analysis currently.

Two sets of Type 2 cards may be required. The first set is always required and defines housing alignment errors in the shaft X-Y plane. The second set defines the housing alignment errors in the shaft X-Z plane. Type 2 cards for the X-Z plane are required only for bearings having alignment errors in the X-Z plane that are different than those specified for the X-Y plane. The second set of Type 2 cards must be placed after the blank card following the Type 3 cards for the X-Y plane (see Appendix C).

The first set of Type 2 cards must contain a card for each bearing. The second set of Type 2 cards must give the appropriate bearing number in column 10.

3.6.3 Card Type S3

Type 3 cards are used to specify shaft loadings at a given X coordinate. Loading may be applied in the X-Y and X-Z planes, thus requiring two distinct sets of Type 3 cards. Applied loads may have the form of concentrated radial forces, concentrated moments, linearly distributed radial forces and concentrated axial loads which may be eccentrically applied. If an axial load is eccentrically applied, the moment which results will be included automatically if the point of application in the radial plane (Y or Z coordinate) is specified in columns 71-80. Alternatively, the moment generated by the axial load can be specified in columns 31-40.

Variations in distributed radial loads are handled at input just as shaft linear diameter variations are handled.

Note that each set of Type 3 cards must be followed by a blank card.

Also note that in order for symmetry conditions to be considered the second Type 3 card must be void of any loading data.

4.0 COMPUTER PROGRAM OUTPUT

4.1 Introduction

The Program Output is intended to provide the engineer or designer with a complete picture of the shaft-bearing system performance.

In addition to the calculated output data, the input data is listed, thus producing a complete record of the computer run.

Sample output of three bearing-shaft systems is included in Appendix D. These studies demonstrate the ability of the NASA version of the program to calculate performance characteristics of tapered and flanged cylindrical roller bearings, and to demonstrate the functioning of the new cage simulation algorithms and the single-bearing analysis capability. The three configurations are:

- 1. A system in which an input pinion is supported by tapered roller bearings in a straddle configuration.
- 2. A system in which an input pinion gear load is supported by a flanged cylindrical roller bearing operating in conjunction with two angular contact ball bearings.
- 3. A single ball bearing system operating under combined radial and thrust loading.

Key output items are discussed briefly below.

4.2 Bearing Output

4.2.1.1 Linear and Angular Deflections

These deflections refer to the bearing inner ring relative to the outer ring and are defined in the inertial coordinate system of Figure 2.4. The bearing deflections are not necessarily equal to the shaft displacements since the bearing outer ring radial or angular mounting errors may be specified as non-zero input.

4.2.1.2 Reaction Forces and Moments

These values reflect bearing reactions to shaft applied loading and outer ring mounting errors.

When the bearing inner ring has achieved an equilibrium position, the summation of all bearing reaction loads should numerically equal the shaft applied loading. When the level of solution indicated by "NPASS" = 2 is employed, as discussed in Section 5, differences between shaft applied and bearing reaction loads will exist but will typically be less than 10%. This difference is a consequence of friction forces contributing to the reaction loads whereas the inner ring equilibrium position has been determined considering elastic contact forces only.

4.2.2 Fatigue Life Data

The L_{10} fatigue life of the outer and inner raceways as well as the bearing are presented. The bearing life represents the statistical combination of the two raceway lives. These lives reflect the combined effects of the lubricant film thickness and material life factors. The lubricant film thickness life factor is described in detail in Section 3.3.1.

4.2.2.1 h/sigma

The ratio h/ σ , also referred to as λ , is printed for the most heavily loaded rolling element. The variable h, represents the EHD plateau film thickness with thermal and starvation effects considered. The variable σ represents the composite root mean square surface roughness of the rolling element and the relevant raceway.

4.2.2.2 <u>Life Multipliers</u>

4.2.2.2.1 <u>Lubrication</u> - This life multiplier is a function of h/ σ at each concentrated contact. Its value ranges from 0.21 for h/ σ < 0.6 to 3.0 at h/ σ > 10. This subject is covered in more detail in Section 3.3.1.

4.2.2.2.2 Material - This output simply reflects the input value, Again, it is covered in Section 3.

4.2.3 Temperatures Relevant to Bearing Performance

These temperatures fully describe the temperature conditions which affect the performance of a given bearing. If one of the temperature mapping options is used, the temperatures printed reflect the results of the particular option. If, neither temperature option was used, the list is simply a repeat of the input data. Note that there are separate temperatures for outer and inner raceways and flanges and ring temperatures. The raceway and flange temperatures are used to determine lubricant properties. The ring temperatures are used in the bearing dimension change analysis. The raceway, flange and ring temperatures may be the same value.

4.2.4 Frictional Heat Generation Rate and Bearing Friction Torque

4.2.4.1 Frictional Heat Generation Rate

The various sources of frictional heat generated within the bearings are listed. The values printed for "OUTER RACE, OUTER RING FLANGES, INNER RACE, INNER RING FLANGES, R.E.DRAG AND R.E. CAGE" represent the sum of the generated heats for all rolling elements. Additionally, the heats printed for the outer and inner raceways and flanges, plus the rolling element cage, reflect the friction developed outside the concentrated contacts, i.e., the HD friction as well as the EHD friction developed within the concentrated contacts. The raceway and flange data also includes any heat generated as a consequence of asperity contacts when the SKF friction model is used. "R. E. DRAG" should be interpreted as the heat resulting from lubricant churning as the rolling elements plow through the air-oil mixture.

4.2.4.2 Torque

The torque value is calculated as a function of the total generated heat and the sum of the inner and outer ring rotational speeds. The intent is to present a realistic value of the torque required to drive the bearing. Under conditions of inner ring rotation the torque value reflects the torque required to drive the inner ring.

4.2.5 EHD Film and Heat Transfer Data

4.2.5.1 EHD Film Thickness

These values refer to the calculated EHD plateau film thickness at both contacts of the most heavily loaded rolling element and include the effects of the thermal and starvation reduction factors.

4.2.5.2 Starvation Reduction Factor

These factors give for the inner and outer ring contacts, the reduction in EHD film thickness ascribable to lubricant film starvation according to the methods of Chiu, [11].

These factors pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

4.2.5.3 Thermal Reduction Factor

These factors are calculated according to the methods of Cheng, [10] and pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

4.2.5.4 Meniscus Distance

These values are calculated according to the methods of Chiu, [11] and pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded

rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

4.2.5.5 Raceway-Rolling Element Conductivity

These data reflect the amount of heat transfer between rolling element and raceway for each degree centigrade difference between the two components. These data reflect the average of all outer and inner contacts respectively. (See Sections 2.1.4 & 2.1.4.1 for a discussion of the calculation procedure.)
4.2.6 Fit and Dimensional Change Data

4.2.6.1 Fit Pressures

These data refer to the pressures built up as a consequence of interference fits between shaft and inner ring and housing and outer ring. Pressures are presented both for the standard cold-static condition (16°C) and at operating conditions.

4.2.6.2 Speed Giving Zero Fit Pressure (Between the shaft and inner ring)

This is a calculated value based upon operating conditions and provides a measure of the adequacy of the initial shaft fit.

4.2.6.3 Clearances

"Original" refers to cold unmounted clearance which is specified at input if the diametral clearance change analysis is executed. "Change" refers to the change in diametral clearance at operating conditions relative to the cold unmounted condition. A minus sign indicates a decrease in clearance. "Operating" refers to the clearance at operating conditions. For all types of ball bearings the decrease in clearance can be combined with the initial diametral clearance, and the free operating contact angle at operating conditions may be calculated. Note that the change in clearance should be compared against the diametral play of the split inner ring ball bearing in order to determine if the possibility for three point contact exists. The Program does not account for three point contacts even though the change in clearance might suggest that three point contact is obtained.

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4.2.7 Lubricant Temperatures and Physical Properties

The lubricant properties, particularly the dynamic viscosity and to a lesser degree the pressure viscosity coefficient, are heavily temperature dependent. These factors enter the EHD film thickness calculation and the HD and EHD friction models. The lubricant is assumed to be at the same temperature as the relevant raceway. As noted elsewhere, these temperatures may be either input directly or calculated by the Program.

The physical properties printed are self explanatory. The units are enumerated.

4.2.8 Cage Data

4.2.8.1 Cage-Land Interface

The cage data indicate the performance parameters at the interface between the cage rail and the ring land on which the cage is guided. The torque, heat rate and separating force require no explanation. The eccentricity ratio defines the degree to which the cage approaches the ring on which it is guided at the point of nearest approach. The radial displacement of the cage relative to the bearing axis is divided by one half the cage-land diametral clearance. An eccentricity ratio of one indicates cage-land contact. A ratio of zero indicates that the cage rotation is concentric with the bearing axis.

Only the cage-land and rolling element pocket forces are considered in determining the cage eccentricity. The cage weight and centrifugal force which result from the eccentricity although available, are not considered in the analysis. The omission of these considerations helps reduce convergence problems.

4.2.8.2 Cage Speed Data

Cage speed data present the comparison between the cage speed calculated based upon the quasidynamic equilibrium considerations and the speeds calculated with raceway control theory for ball bearings and the epicyclic speeds of the roller bearing components.

4.3 Rolling Element Output

4.3.1 Rolling Element Kinematics

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4.3.1.1 Rolling Element Speeds

All of the rolling element speeds tend to vary from position to position when the bearing is subjected to combined loading.

The total rolling element speed is with reference to the cage and represents the vector sum of the three orthogonal components.

4.3.1.2 Speed Vector Angles

The rolling element speed vector angles, Arctan $(\omega_\chi/\omega_\chi)$ and Arctan (ω_Z/ω_χ) are presented in order to show a clearer picture of the predicted ball kinematics. The ball speed vector tends to become parallel with the bearing X axis with increasing shaft speed and decreasing contact friction.

4.3.2 Rolling Element Raceway Loading

4.3.2.1 Normal Forces

The normal forces acting on each rolling element are printed. The rolling element race normal forces are self explanatory. The cage force is calculated only when the friction solution is employed and is always directed along the rolling element 2 axis. If the rolling element orbital speed is positive, a positive cage force indicates that the cage is pushing the rolling element, tending to accelerate it. Cage force is a function of rolling element position within the cage pocket. Its magnitude is derived using hydrodynamic lubrication assumptions, when the distance between the rolling element and cage web is large, and EHD assumptions when the separation is of the order of the EHD film thickness or when rolling element-cage web interference exists (SKF version). The NASA version (see Appendix F) solves for cage forces using hydrodynamic lubrication assumptions only.

4.3.2.2 Hertz Stress

The stress printed represents the maximum normal stress at the center of each ball race contact or at the most heavily loaded slice of the roller raceway contact.

4.3.2.3 Load Ratio Qasp/Qtot-

If the EHD film thickness is small compared to the RMS composite rolling element-race surface roughness, the rolling element-race normal load will be shared by the EHD film and asperity contacts. The load ratio reflects the portion of the total load carried by the asperities.

when the NASA version of the code is executed (see Appendix F), $Q_{asp}/Q_{tot=0}$, since asperities are not accounted for in the Allen Traction model [7].

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4.3.2.4 Contact Angles

A ball bearing, subject to axial loading, misalignment or mounted such that the inner ring is always displaced axially relative to the outer rings, (i.e. a duplex set of angular contact ball bearings) will have non-zero contact angles. At low ball orbital speeds the inner and outer race angles will be substantially the same. At high speeds, ball contribugal force will cause the outer race contact angle to be less than the inner race angle.

4.3.3 Roller End-Flange Contact Data

For a tapered roller bearing a single set of roller end-flange data is printed. For the cylindrical roller bearing, which may have up to four flanges, the program examines the data and prints the results for the most heavily loaded outer and inn r ring flanges. In addition to the data listed below for a cylindrical roller, the semimajor contact axis as well as the concentrated contact and hydrodynamic heat generation rates are printed for the tapered roller.

4.3.3.1 Normal Force

The interference between the roller end and flange is determined from the solution for the relative ring and rolling element positions. Hertz theory is used to calculate the load which results from this interference.

4.3.3.2 Hertz Stress

The Hertz stress printed is the maximum normal stress which occurs in the contact.

4.3.3.3 EHD Film Thickness

The plateau film thickness is calculated using the Archard-Cowking [3] equation for point contacts or the Dowson-Higginson[4] equation for line contacts, when the "SKF" version of the program is executed (see Appendix F). The "NACA" version uses the Loewenthal model [6] to calculate film thickness. Either result is then modified to account for starvation [12] and thermal [11] reduction effects. This modified film thickness is printed.

4.3.3.4 Sliding Velocity

The sliding velocity is defined as the difference between the flange and roller end linear velocities at the center of the contact.

4.3.3.5 Rolling Velocity

Rolling velocity is defined as one half the sum of the roller end and flange linear velocities at the center of the contact.

4.3.3.6 Contact Ellipse Semiminor Axis

To help assess the severity of the roller end flange contact and the possibility for edge loading, the semiminor axis of the contact is printed.

4.4 Thermal Data

As in the case for bearing output, all of the input data is printed. The calculated output data is presented in the form of a temperature map in which a node number and the respective node temperature appear. The appearance of the steady state and transient temperature maps are identical. The transient temperature map also includes the time (T) at which the temperature calculations were made.

4.5 Shaft Data

These data simply reflect the input information.

4.6 Program Error Messages

4.6.1 From Subroutine ALLT

"Steady State Solution with (EP1) degrees accuracy was not obtain after (IT1) Iterations".

This message pertains to the external temperature iteration scheme in which system temperatures and bearing generated heats are being solved for an equilibrium condition.

4.6.2 From Subroutine SHABE

"It was not possible to obtain the change of clearance with an accuracy of (ERFIT) times the rolling element diameter in (ITFIT) iterations".

This message pertains to the bearing diametral clearance change iteration scheme. The solution may be converging in which case the number of iterations (ITFIT) should be increased. This can be checked with an NPRINT = 1 intermediate printout. The intermediate rint may indicate that the solution is oscillating. The most likely cause of oscillation is the alternate prediction of bearing preload with all rolling elements loaded, and then in the next iteration, only a subset of the rolling elements loaded. This problem can usually be overcome by either of two methods.

- 1) In subroutine FIT remove the GC TO 20 statement. This will cause the inner ring load distribution to have no effect on the change in diametral clearance.
- 2) The solution can be damped by redefining the solution damping factor FA, such that it would take on a value 0 €FA €1. FA is presently set to 1 in subroutine SHABE. If this damping technique is used, the number of FIT iterations should be increased as the value of FA is decreased. An upper limit of 10 iterations is recommended.

4.6.3 From Subroutine SOLVXX

1) "SINGULAR SET OF EQUATIONS"

This message might occur when the thermal input data is not input properly.

2) "THE LIMIT FOR NUMBER OF ITERATIONS IS REACHED"

This message might occur either during a steady state temperature solution or bearing solution. Before increasing the number of iterations check the equation residue values. If they are low, the solution may be good enough.

3) "THIS IS THE BEST WE CAN DO. IT MAY BE USEABLE"

This message reflects the fact that the next iteration will result in divergence. The iteration procedure is thus terminated. The equation residue values should be checked, if they are low, the solution may be useable.

As noted above the occurrence of messages 2) and 3), do not necessirily mean that the solution is not good. Generally the messages indicate that the solution has not converged as tightly as the user has requested.

Note: The XX suffix on SOLV specifies the version of SOLV contained in the user's program. As of this writing the current version is SOLV13. The suffix is changed each time improvements are made which require a change in the SOLVXX calling sequence.

4.6.4 From Subroutine INTFIT

"Singular matrix on tight shaft fit"
"Singular matrix on loose shaft fit"

These messages reflect an error in the input data usually as a consequence of inconsistent component diameters, such as the shaft inside diameter being greater than the outside diameter.

5.0 GUIDES TO PROGRAM USE

The Computer Program is a tool. As with any tool the results obtained are at least partially dependent upon the skill of the user. Certainly the economics of the Program usage are highly dependent upon the user's technical need and discriminate use of Program options.

Some general guides for efficient use of the Program are listed below:

- 1. Attempt to use the lowest level of solution possible. For instance if the prime object of a given run is to obtain bearing fatigue lives, execute only the elastic solution (NPASS = 0). If an estimate of bearing frictional heat is required, execute the low level friction evaluation (NPASS = 1). Execute the friction solution (NPASS = 2) only if rolling element and cage kinematics are of interest. Execute the highest level of solution (NPASS = 3) if kinematics are of interest and the bearing reaction loads deviate substantially from the shaft applied loading, i.e. a deviation greater than ten percent.
- 2. Attempt to <u>input bearing operating diametral clearance</u> rather than calculate it. Or, execute the diametral clearance change analysis once for a group of similar runs and use the output from the first run as input to the subsequent runs omitting the clearance change analysis.
- 3. Attempt to input accurate operating temperatures rather than calculate them.
- 4. The more non-linear the problem the more computer time required to solve it. In the bearing friction solution large coefficients of friction seem to increase the degree of nonlinearity. In the thermal solutions, if possible, eliminate nonlinearities by omitting radiation terms and by using constant rather than temperature dependent free and forced convection coefficients.
- 5. In the transient thermal solution, space the calls to the shaft-bearing solution (BTIME) to as large an interval as prudently possible. Be careful however, too long and interval will produce large errors in heat rate predictions.

6. In the steady state thermal analysis attempt to estimate nodal temperatures on a node by node basis.

Nodes which are heat sources should have higher temperatures than the surrounding nodes.

The above suggestions are intended to encourage the use of the Program on a cost effective basis. The intent is not to discourage the use of important program capabilities but to emphasize how the program should be most effectively used.

It is suggested that the user take a simple, axially loaded ball bearing problem and execute the program through the full range of options beginning with a frictionless solution proceeding to the three levels of friction solution with a low (0.01) and high (0.1) friction coefficient. The diametral clearance change analysis and the thermal solutions should also be executed on an experimental basis. This exercise will provide the user with some insight into economics of the Program usage on his computer as well as the results obtained from various levels of solution of the same problem.

It is also suggested that a constant user of the program should study the hierarchical Program flow chart Appendix A, along with the Program listing to gain an appreciation of the program complexity and the flow of the problem solution. The Program is comprised of many small functional subroutines. Knowledge of these small elements may allow the user to more easily piece together the philosophy of the total problem solution.

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AT81D040

APPENDIX A

SKF COMPUTER PROGRAM AT81Y003 SHABERTH
HIEF RCHICAL FLOW CHART

APPENDIX A S K F COMPUTER PROGRAM AT81Y003 FLOW CHART

Flow Chart

The hierarchical flow chart presents the program structure, listing the program elements in the order in which they would be called to solve the shaft-bearing dynamic, as well as steady state and transient temperature distribution problems. The various solution loops are indicated, as well as notes which indicate the functions of various subroutine groupings.

Each line in the flow chart represents a program element, subroutine, function or the main program ALWAYS. The call of one subroutine by another is denoted by indenting the called subroutine relative to the routine doing the calling. As an example, subroutine SKF calls subroutines FLAGS, TYPE, PROPST, LUPROP, LUBCON, DATOT, CNVRT, CONS and SPRING. Subroutine CONS calls CONBR¹, BCON, TCON, CRCON and CONBR2. BCON calls ABDEL. Both TCON and CRCON call ABDEL and SLICES.

The first mention of a subroutine within the flow chart includes the entire list of subordinate program elements. At subsequent calls to that subroutine the list of subordinate elements is omitted. As an example the first call to subroutine AXLBOJ is followed by the subordinate elements, JMVIKT, SNITMT, NUMLOS, DUBSIM, MEIE, MEIL and SIMQ. After the call of AXLBOJ from INDEL, the subordinate elements are not listed but are, nevertheless, employed. The list of subordinate program elements are omitted in repeated calls of subordinate GUESS, BEAR, SOLVXX and DELIV3 as well as AXLBOJ.

As noted earlier, rolling equilibrium is calculated, first without, then if required, with friction forces included. Whether or not friction is considered is highlighted with the words Frictionless or Friction.

If the Program is too large to fit in its entirety on the user's computer, segments of the program may be "overlaid". For this purpose the Program is subdivided into ten (10) modules which can be sequentially "overlaid".

The Program segments SKF, TEMPIN, and GUESS all perform initiation functions and with the exception of GUESS, are called only once per program execution.

The real problem solving portion of the program is endied in segment ALLT. Within this segment the shaft bearing solution is obtained through the call to SHABE, then the steady state or transient temperature distributions are obtained.

This scheme is repeated until the end objective, steady state thermal equilibrium or time up for the transient scheme, is realized.

The nonlinear equation solver SOLV13 is central to the program and deserves special discussion as related to the flow chart. The first call to SOLV13 is from BEAR. Only for this first call are all of the SOLV13 subclainate subroutines listed as noted earlier. These include INSOLV, EQS, PARDER, SIMQ, EQCHEK, ERWRIT and ERCHEK. In the subsequent call to SOLV13 in which the steady state temperatures are being calculated, the above listed subroutines are again called but these calls with the exception of EQS are not listed on the flow chart.

EQS is the name given by SOLV13 to a subroutine which sets up the system of equations to be solved. EQS is brought into SOLV13 through the argument list. When the bearing equations are being solved, subroutine BRGGEQ is brought into SOLV13 and within SOLV13 is referenced by the name EQS. When the heat transfer equations are being solved as a consequence of the call of SOLV13 from ALLT, NET is brought into SOLV13 and is referenced as EQS.

Since storage and execution costs rise dramatically when the program is run at levels higher than NPASS=1, it is recommended that the program be executed at the lowest solution level possible. If a level 2 or 3 shaft-bearing solution is desired for bearings with more than 16 rolling elements, a new executable version must be created by expanding the blank common used in subroutine BEAR. Increasing the length of the C array to 33489 elements enab the program to simulate bearings with up to 30 rolling elements

LHAYS	
SKF FLAGS	Read and set boaring and bearing solution control data
PROPST LUBCON	Set material and lubricant properties and calculate constants for compensature dependency calculations
DATOT TITLE FITDAT ROLDAT	Write bearing input and hardcoded, preset date
CNVRT CONS CONER1 BCON ABDEL TYON ABDEL SLICES CRCON ABDEL SLICES	Calculate bearing related constants
COMBR2 SPRING TEMPIN INDUM RWHTC RWG	Fead and write thermal and thermal solution control data and calculate heat transfer coefficients
TMAP SHAFT ZERO APRANG ORDERR OUTINP	Read and write shaft geometry, londing and bearing position data
AXLBOJ JMVIKT SNITMT NUMLOS UUBSIM MEIE MEIE MEIL SIMQ REACT INDEL	Calculate shaft deflection constants
PARI PAR AXLBOJ	Calculate shaft influence coefficients
GUESS GRBRG GPOLL GBBRC GBALL GTBRG GSTPOS GSTSPD GUESCC VARRDC	Guess values of rolling element and cago variables, posit ons and speeds
ALI.T	Begin the solution of the steady state transien thereal and temperature dependent shaft-bearing Calculate Shaft-Bearing Equilibrium, in steps, for the desired NPASS solution level
FIT INTFIT SINGS	for the desired NPASS solution level Calculate bearing diametral clearance
SONRI PERFORM THESE SONRI CALCULATIONS BEARC FOR ALL NPASS VALUES.	Establish iteration scheme to satisfy shaft and bearing inner ring equilibrium
BEARC FOR ALL REACS VALUES.	and cearing inner ring equilibrium

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BEAR
                                                                             Establish iteration scheme to satisfy rolling element equilibrium. Solve rolling element equilibrium, bearing by bearing, one element
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                                   DAMPCO
                                   ERCHEK
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11 102 03
                               EQS : BROGEQ (WITHOUT FRICTION)
                                                                             Evaluate ball-raceway contact loads, ball centrifugal force and ball equilibrium equations
                                   EQBALL (BALL BEARING)
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                                   EQTAPR (TAPERED ROLLING BEARING) Evaluate tapered relier-encously and flamen | BUCTHOL contact loads, centrifusal force and successions
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                                                                             contact loads, centrifugal force and gyromoment and roller equilibrium equations
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                                                       CAL ROLLER BEARING; Evaluate cylindrical roller-raceway and flange
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                                                 PULL
                               PARDER
                                                                             Calculate partial Jerivatives of rolling element
 T
                                | EQS = BRCCEQ
                                                                             equilibrium equations with respect to rolling
 I
                               SIMQ
                                                                             element positions and calculate new positions
 M
                               EXCHEK
          ţΙ
                                   EQS = BRCGEO
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                                  ERWRIT
 T
                          SUMF
                                                                             Sum rolling element-inner ring forces and moments
 R
                              SUMK
                               SUMFL
 N
                          LIFE
 S
                                                                              Calculate bearing fatigue life
                                                                              Add partial derivatives of bearing inner ring
                BFILL.
 Ē
                                                                             force with respect to inner ring displacement to the shaft equilibrium equation and obtain
                SHAPA
 NT
                  | FILLS
                                                                             new inner ring positions
l c
                                                                              After shaft equilibrium is satisfied, calculate
            BEAR
ALCU
                                                                              bearing reaction forces at the bearing equilibrium positions and estimate a new set of rolling element speeds
            : ESS
            VISC 2
                                                                             Evaluate temperature dependent lubricant properties and constants.
              I ALPHAO
LATI
            DRAGNO
            STOCK
                    (IF NPASS = 1) SKIP II' NPASS = 2 OR 3
                                                                             Evaluate bearing performance with estimated
 Ō
                PREPAR
                                                                              rolling element speeds.
 Ñ
                BRGGEQ
                         (WITH FRICTION)
 S
                     FORAL I
                                                                            Evaluate ball-raceway contact loads.
                          BGCTRL
                          BALL TN
                          BALLEQ
                                                                             Evaluate ball-raceway EMD film thickness,
                               FMIX / NASA
                                                                             inlet and concentrated contact friction
                                    TINT
                                                                             forces and heat generation rate.
                                    FILM
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                                        FBAR
                                    FRINT/NASA
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                          CAGESP
                                                                             Evaluate cage speed, all-cage normal and
                          CACEED
                                                                             friction forces and heat rates plus cage
                               COLAND
                                                                             rail-ring land normal and friction forces
                                   CCDRY
                                                                             plus heat generation rates.
                                   COVET
                               CGILE
                                    CGBALL
                                    CGROLL
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	••
BRGAX BRAX	Evaluate ball, position dependent, necelerations.
LIFE	Calculate bearing fatigue life
HUTRAC HUT	Calculate ball-raceway heat transfer coefficients
EQTAPR BGCTRL TAPIN	Evaluate roller-raceway and roller-flange contact loads
TAPEQ TPNORM FLNORM PUSH PULL	
FMIXR/NASA SPEEDS FILM THERF1 STARFC HRRRIC SPEEDS ALLEN FRICTN	Evaluate roller raceway EHD film thickness inlet and concentrated contact friction forces and heat generation rate
FLMIX/NALA FILM THERFI STARFC HOFRII FBAR ALLNPT FRICTN	Evaluate roller-flange EHD film thickness inlet and concentrated contact friction forces and heat generation rate
CAGSPD CAGEEQ BRGAX LIFE HUTRAC HUTFLN	Evaluate tapered roller bearing cage speed, cage forces, roller accelerations, bearing fatigue life, roller-raceway and roller-flange heat transfer coefficients
EQCYL BGCTRL ROLLIN FUNDEF ROLLEQ ITPNORM ITPNORM FUSH PUSH FMIXR / NASA FIMIX / NASA	Evaluate cylindrical roller-raceway and flange normal and friction forces and heat generation rates plus EHD film thicknesses
CAGSPD CAGEEQ BRGAX LIFE HUTRAC HUTFLN (IF NPASS = 1, RETURN TO ALLT.)	Evaluate cylindrical roller bearing cage speed, cage forces, roller acceleration, bearing fatigue life, plus roller-raceway and roller-flange heat transfer coefficients
BEAR (IF NPASS = 2) SKIF IF NPASS = 3 PREPAR SOLV13 EQS = BROGEQ (WITH FRICTION)	Solve the rolling element and cage equilibrium equations with friction included, using the inner ring positions established by the frictionless solution
SONR1 (IF NPASS = 3) BEARC #3 BEAR PREPAR #4 SOLV13 SOLV13 SUMF SUMF SUMF	Solve shaft, rolling element and cage equilibrium equations with friction included
ELIV3/SKF TITL5 RITE RITE REOUT3 REFLOT RITE4 RITE2	Write shaft-bearing output
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FILLGT COLV13 1 EQS = NET (FOR STEADY STATE THERMAL SOLUT 1 NETEET DELIV3 TMAP	Solve for the steady state temperature distribution. Write shaft-bearing output and temperature map if desired.
STEPMA NET (FOR THE TRANSIENT THERMAL SOLUTION) NETEET DELIV3 NET NETEET TMAP	Determine an appropriate time step and solve for the transient temperature distribution. Write shaft-bearing output and temperature map if desired.
DELIV3 TMAP	Write output for the final solution
NPASS = 0 SHAFI AND REARING INNER RING E	CUILIBRIUM ARE

- NPASS = 0 SHAFT AND BEARING INNER RING EQUILIBRIUM ARE SATISFIED CONSIDERING ELASTIC CONTACT FORCES. NO LUBRICATION OR FRICTION EFFECTS ARE CONSIDERED.
- NPASS = 1 SHAFT AND BEARING INNER RING EQUILIBRIUM ARE SATISFIED CONSIDERING ELASTIC CONTACT FORCES. LIBRICATION AND FRICTION EFFICIE: AND CAMBIDDERED USING RACEWAY CONTHOL (BALL BEARINGS) OR EPICYCLIC (ROLLER BEARINGS) ASSUMPTIONS TO ESTIMATE ROLLING ELEMENT AND CAGE SPEEDS.
- NPASS = 2 SHAFT AND BEARING INNER RING EQUILIBRIUM ARE SATISFIED CONSIDERING ELASTIC CONTACT FORCES. USING THE INNER RING POSITIONS THUS OBTAINED, ROLLING ELEMENT AND CAGE EQUILIBRIUM ARE DETERMINED CONSIDERING FRICTION.
- NPASS = 3 COMPLETE SOLUTION. SHAFT AND BEARING INNER HING PLUS POLLING ELEMENT AND CAGE EQUILIBRIUM ARE DETERMINED CONSIDERING ALL ELASTIC AND FRICTION FORCES.

AT81D040

APPENDIX B

HEAT TRANSFER COMPUTATION NOTES

APPENDIX B

HEAT TRANSFER INFORMATION

AT81D040

B.1 BACKGROUND

The temperature portion of Program AT81Y003 is designed to produce temperature maps for an axisymmetric mechanical system of any geometrical shape. The mechanical system is first approximated by an equivalent system comprising a number of elements of simple geometries. Each element is then represented by a node point having either a known or an unknown temperature. The environment surrounding the system is also represented by one or more nodes. With the node points properly selected, the heat balance equations can be set up accordingly for the nodes of unknown temperature. These equations become non-linear when there is convection and/or radiation between two or more of the node points considered. The problem is therefore reduced to solving a set of linear and/or non-linear equations for the same number of unknown nodal temperatures. It is obvious that the success of the approach depends largely on the physical subdivision of the system. If the subdivision is too fine, there will be a large number of equations to be solved; on the other hand, if the subdivision is too crude, the results may not be reliable.

In a system consisting of rolling bearings, for the sake of simplicity, the elements considered are usually axially symmetrical, e.g., each of the bearing rings can be taken as an element of uniform temperature. For an element which is not axially symmetrical, its temperature is also assumed to be uniform and its presence is assumed not to distort the uniformity in temperature of a neighboring element which is axially symmetrical. That is, the non-symmetrical element is represented by an equivalent axially symmetrical element with approximately the same surface area and material volume. This kind of approximation may seem to be somewhat unrealistic, but with properly devised equivalent systems, it can be used to solve complicated problems with results satisfying some of the important engineering requirements.

The computer program can solve the heat-balance equations for either the steady state or the transient state conditions and produce temperature maps for the mechanical system when the input data are properly prepared.

B. 2 BASIC EQUATIONS

B.2.1 Heat Conduction

The rate of heat flow qci, j(W) that is conducted from node i to node j may be expressed by,

$$q_{ci,j} = \frac{\lambda_{ij} A_{ij}}{L_{ij}} (t_i - t_j)$$

 t_i and t_j are the temperatures at i and j, respectively, $A_{i,j}$ the area normal to the heat flow, (m^2) , L_{ij} the distance (m) and λ_{ij} the thermal conductivity between i and j, $(W/m^{\circ}C)$.

Assuming that the structure between point i and j is composed of different materials, an equivalent heat conductivity may be callulated as follows:

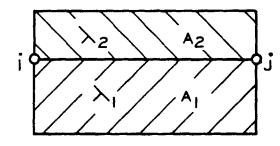


Fig. B-1
$$\lambda_{ij} = \frac{\lambda_1 A_1 + \lambda_2 A_2}{A_{ij}}$$

$$A_{ij} = A_1 + A_2$$

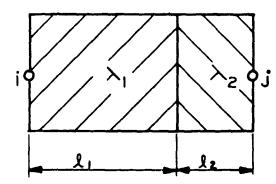


Fig. B-2
$$\lambda_{ij} = \frac{\ell_{ij}}{\ell_1/\lambda_1 + \ell_2/\lambda_2}$$

$$\ell_{ij} = \ell_1 + \ell_2$$

The calculation of the areas will be discussed in Section B.2.5.

B. 2. 2 CONVECTION

The rate of heat flow that is transferred between a solid structure and air by free convection may be expressed by

$$q_{vi,j} = \alpha_{i,j} A_{i,j} |t_i - t_j|^{1.25} \cdot SIGN (t_i - t_j)$$

where

SIGN =
$$\begin{cases} 1, & \text{if } (t_i - t_j) \ge 0 \\ -1, & \text{if } (t_i - t_j) < 0 \end{cases}$$

in which

2.5 · 10⁻² W/m²- (degC)^{1.25} for hot surfaces facing upward and celd surfaces facing downward

a.4 · 10⁻² W/m²- (degC)^{1.25} for hot surfaces facing downward and cold surfaces facing upward

1.8 · 10⁻² W/m²- (degC)^{1.25} for vertical surfaces

For other special conditions, $\alpha_{\mbox{\scriptsize ij}}$ must be estimated by referring to heat transfer literature.

The rate of heat flow that is transferred between a solid structure and a fluid by forced convection may be expressed by

$$q_{wi,j} = \alpha_{i,j} A_{i,j} (t_i - t_j)$$

in which α_{ij} is the heat transfer coefficient.

Now, with $\alpha = \alpha_{ij}$, introduce the Nusselt number

$$N_{\rm u} = \frac{\alpha L}{\lambda}$$

the Reynolds number

$$R_e = \frac{UL}{v}$$

and the Prandtl number

$$P_T = \frac{\rho \vee C_p}{\lambda}$$

where

I is a characteristic length which is equal to the diameter in the case of a cylindrical surface and is equal to the plate length in case of a flat surface (m)

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- U is a characteristic velocity which is equal to the difference between the fluid velocity at some distance from the surface and the surface velocity (m/sec)
- λ is the fluid thermal conductivity (W/M^OC)
- v is the fluid kinematic viscosity (M²/sec)
- ρ is the fluid density (kg/m^3)
- c_p is the fluid specific heat(J/kg°C)

For given values of R_e and P_r the Nusselt number N_u and thus the heat transfer coefficient may be estimated from one of the following expressions:

Laminar flow along a flat plate: R_p < 2300

$$N_u = 0.323 \sqrt{R_e} \cdot \sqrt[3]{P_r}$$

Laminar flow of a liquid in a pipe:

$$N_u = 1.36 \sqrt[3]{R_e \cdot P_r(\frac{D}{L})}$$

where D is the pipe diameter and L the pipe length

Turbulent flow of a liquid in a pipe:

$$N_u = 0.027 \cdot R_e^{0.8} \cdot \sqrt[3]{P_r}$$

Gas flow inside and outside a tube:

$$N_u = 0.3 R_e^{0.57}$$

Liquid flow outside a tube:

$$N_u = 0.6 R_e^{0.5} \cdot P_r^{0.31}$$

Forced free convection from the outer surface of a rotating shaft

$$N_u = 0.11 \left[0.5 R_e^2 \cdot P_1^{0.35} \right]$$

where the Reynolds number $R_{\rm e}$ is developed by the shaft rotation.

$$R_e = \frac{\omega \pi D^2}{V}$$

in which ω is the angular velocity (rev/sec) D is the shaft diameter (m)

The average coefficient of forced convection to the lubricating oil within a rolling contact bearing may be approximated by,

$$\alpha = 0.0986 \left\{ \frac{N}{v} \left[1 + \frac{D \cos(\beta)}{d_m} \right] \right\}^{\frac{1}{2}} \lambda(P_r)^{1/3}$$

using + for outer ring rotation - for inner ring rotation

in which N is the bearing operating speed (rpm) D is the diameter of the rolling elements (mm) d_m is the bearing pitch diameter (mm) B is the bearing contact angle (degrees)

B.2.3 FLUID FLOW

The rate of heat flow that is transferred from fluid node i to fluid node j by fluid flow is

$$q_{fi,j} = \rho \dot{v}_{ij} C_p (t_i - t_j)$$

 $\dot{V}_{i\,j}$ is the volume rate of flow from i to j. It must be observed that the continuity of mass requires the following equation to be satisfied

$$\Sigma \dot{v}_{ij} = 0$$

provided the fluid density is constant. The summation should be extended over all nodes i within the fluid which have heat exchange with node j by fluid flow.

B.2.4 HEAT RADIATION

The rate of heat flow that is radiated to node j from node i is expressed by

$$q_{Ri,j} = \delta_{i,j} \{ (t_i + 273)^4 - (t_j + 273)^4 \}$$

where

 T_i = temperature of node i in ${}^{\circ}C$.

 T_i = temperature of node j in ${}^{o}C$.

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and the value of the coefficient $\delta_{i,j}$ depends on the geometry and the emissivity or the absorptivity of the bodies involved.

For radiation between large, parallel and adjacent surfaces of equal area, $A_{i,j}$ and emissivity, $\epsilon_{i,j}$, $\delta_{i,j}$ is obtained from the equation

$$\delta_{i,j} = \epsilon_{i,j} \quad \sigma A_{i,j}$$

where σ , the Stefan-Boltzmann constant, is

$$\sigma = 5.76 \cdot 10^{-8} \text{ W/m}^2 / (\text{degK})^4$$

For radiation between concentric spheres and coaxial cylinders of equal emissivity, $\epsilon_{i,j}$, $\delta_{i,j}$ is given by the equation

$$\delta_{ij} = \frac{\varepsilon_{i,j} \sigma A_{i,j}}{1 + (1 - \varepsilon_{i,j}) \frac{A_{i,j}}{A^*_{i,j}}}$$

where σ is as above, $A_{i,j}$ is the area of the enclosed body and $A_{i,j}^*$ is the area of the surrounding body, i.e. $A_{i,j} < A_{i,j}^*$.

Expressions for $\delta_{i,j}$ that are valid for more complicated geometries or for different emissivities may be found in the heat transfer literature.

B.2.5 CALCULATION OF AREAS

In the case of heat conduction heat transfer in the axial direction $A_{i,j}$ is given by the equation (Fig. B-3).

$$A_{i,j} = 2\pi r_m \cdot \Delta r$$

Referring to the input instructions, Section 3.4.7, but recalling L must be input in mm not m.

$$L_1 = r_m = \frac{r_1 + r_2}{2}$$

$$L_2 = \Delta r = r_2 - r_1$$

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In the case of heat transfer in the radial direction, $A_{i,j}$ is obtained from the expression

$$A_{i,j} = 2\pi r_m \cdot H; L_1 = r_m; L_2 = H$$

and similarly for the radiation term above

$$A_{i,j}^* = 2\pi r_m^* H$$
 $L_3^* = r_m^*$
 $L_2^* = H$

in which H is the length of the cylindrical surface; where heat is conducted between i and j, r_m is given by the same equation as above (Fig. B-4 (a)); where heat is convected between i and j, r_m is the radius of the cylindrical surface (Fig. B-4(b)); where heat is radiated between i and j, r_m is the radius of the enclosed cylindrical surface and r_m * the radius of the surrounding cylindrical surface (Fig. B-4(c))

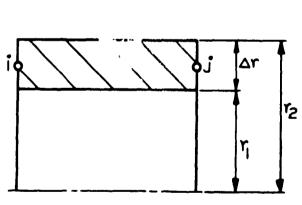


Fig. B-3

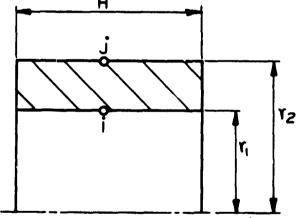
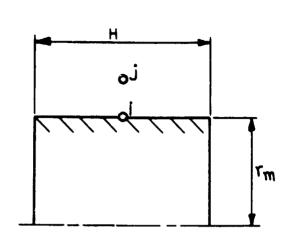


Fig. B-4(a)

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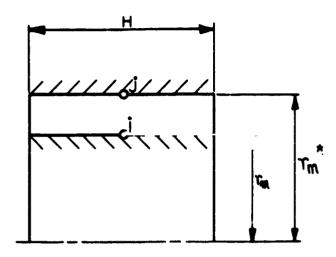


Fig. P -4(b)

Fig. B -4(c)

B.3.1 TRANSIENT ANALYSIS

For the transient analysis all of the data pertaining to the node to node heat transfer coefficients must be provided by the input. Additionally, the volume and the specific heat at each node is required.

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APPENDIX C

SKF COMPUTER PROGRAM SHABERTH INPUT DATA FORMS

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	6.5			Main loop accuracy for frictionless solution.
	9			When the latest change of bearing deflection is less than (EPS1 or EPS2 x Bearing
	1			Deflection), or the bearing angular and
EPSI	40,45,46,47		_	linear deflections are less than
	-		-	2.10 ⁶ § 5.10 ⁻⁸ respectively, then iteration is stopped.
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GOV (3)	36/39 4041 4243		1	If left blank, pre-set accuracy of 0.901 is used.
ن	3		<u>L</u>	useu.
H	8			Fit loop accuracy.
EPSFIT				It is defined as change of diametral clearance
EPS	5			divided by rolling element diameter.
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(2	3			
GOV (2)	5		0	
G.			F 1	
	3		-	Main loop staration flag
<u> </u>	8		G	Main loop iteration flag. Positive no. or blank means pre-set max.=15 is
I FEATO	8			used.
IFLAG(1) IFLAG(2) IFLAG(3) IFLAG(4) NBRG NPRINT (1FIT ITEAL)	c5 26 27 28 29		F S	Negative no. means iteration limit=absolute value of given number.
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(3)	3		0	Fit calc. flag. Blank means no fit calculation & no iteration.
.A6	22 23.4			Positive no. means fit iteration MAX=5.
IFLA6 I TF IT			F S	Negative no, means fit iteration MAX=given number.
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25	.≘]		0	Print flag. If not blank debug output will be obtained.
IFLAG(NPRINT	2]	۲.	If=1 main loop debug only.
<u>x</u>	3		- 114	If=2 full debug output.
	<u>\$</u>			
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* NPASS, IFLAG(S), BEARING SOLUTION LEVEL FLAG

0 LLASTIC CONTACT FORCES ARE CALCULATED, NO LUBRICATION OR FRICTION EFFECTS ARE CONSIDERED

FLASTIF CONTACT F ACES ARE CALCULATED. LUBRICATION AND FRICTION EFFECTS ARE CONSIDERED USING RACEWAY CONTROL (RALL BEARING) OR FPICYCLIC (ROLLER BEARING) ASSUMPTION TO ESTIMATE ROLLING ELEMENT AND CAGE SPEEDS

INNER EQUILIBRIUM IS SATISFIED CONSIDERING ONLY THE ELASTIC CONTACT FORCESIUSING THE INNER RING POSITIONS THUS OBTAINED, ROLLING ELEMENT AND CAGE EQUILIBRIUM ARE DETERMINED CONSIDERING FRICTION COMPLETE SOLUTION. THE INNER RING, ROLLING ELLMENT AND CAGE EQUILIBRIUM IS DETERMINED CONSIDERING ALL ELASTIC AND FRICTION FORCES.

3 2

IF ONE SET OR WORE OF THE BEARTING RING OR ROLLING ELEMENT MATERIALS IS OTHER THAN STEEL, SET IMT TO I AND INCLUDE CARD TYPES BIL THROUGH BIM FOR ALL BEARINGS. HI :

USE LOADING *** JUSTBR IF A DINGLE BEARING IS TO BE ANALYZED, SET JUSTBR TO I AND OMIT ALL SHAFT DATA CARDS. DATA CARD IMSTEAD.

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Beariny Data, One Set per Bearing. All applicable cards Blibsough Bl6 for Bearing 1, fullowed by Blithrough Bl6 for Bearing 2, etc.

CARD TYPE B1

Normally zero or blank, If set to 1, non uniform roller-raceway profile geometry is input on card types BS and B6.

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BD(B)			-		Contact Angle (Deg)	1 1
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g					olling Dissetral Clearance (mm)	1 1
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B ₁₁ (7)					Ho. of Molling Disserted Elements Clearance (mm)	Axial Play for Tapered. Single Rearing Only .mm)
B ₁₁ (7)	1 2 3 4 5 6 7 6 9 6 9 10 11 12 13 14 15 10 17 10 10 20 27 23 23 24 25 12 12 12 12 12 12 12 12 12 12 12 12 12				Ho. of Molling Disserted Elements Clearance (mm)	1 1

BEARING D YA CARD TYPE BS . Une Card

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|F|1|0|.|0| 80 (11) ROLLER BEARING

Roller Flat Roller Crown Radius (mm) Roller Included Angle (Deg) Sphere Radius (mm) Roller End End to End (mm) Roller Length Dismeter of Roller Lorge End for Tepered (mm, Roller Diameter •

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عد اجراجي الاربعيادة عراد المرامع المرامع المرامع المرامع المرامع المرامع للمرامع المرامع المر ORIGINAL PAGE IS OF POOR QUALITY <u>.</u> Sifces F 1 Jo ¥° Roller Carmer or Receway Undercut Relief Inner (am) 80 (23) 80 (23) 80 (24) 80 (25) 80 (25) 80 (26) 80 (26) 80 (26) 80 (26) 80 (26) 80 (27 Roller Corner or Racemay Undercut Relief Onter (sm) 0 1 2 Raceway Crown Radius Inser (sm) 1 2 3 4 5 6 7 6 6 10 11 12 13 14 15 16 17 14 19 20 21 22 23 24 25 26 27 28 29 30 3 32 35 34 35 34 37 38 38 404 0 <u>"</u> Maceway Crown Radius Outer (mm) ō Roller Raceway
Effective Leagth
Irace (mm) <u>-</u> Ö ع Inner Raceusy Curvature fum Ri/D - I Roller Raceway Effective Length Outer (mm) ō ROLLER BEARING 0 1 4 Outer Raceusy Curvature f = Bo/p 0 7 BALL BEARING BEARING DATA CARD TYPE BAA -11.

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Omit Cards B5 and B6 is Reprint	if crown drop flag (Omit Carda B5 and B6 if crown drop flag (column 80 on card Bl)) is 0 or blank.			
Card Type BS, as many RLAM(1, 5)	r as needed, meximum (2,5)	Card Type B5, as many as needed, maximum of 3, for the outer Fe. RLAM(1.5) (3.5)	race (4.5)	(5,5)	(6,5)	(7,5) (8,5)
9	8 8 8 11 19 8 11 11 11 11 11 11 11 11 11 11 11 11 1	58 12 0E 62 62 62 12 92 52 52 52 52 52 52 52 52 52 52 52 52 52		43 44 45 46 49 50 51 5	33 H 35 H 36 H 50 H 60 H 42 H 143 H 60 13 152 13 H 55 15 15 150 H 60 60 60 62 63 H	7 44 65 66 67 66 697 57 17 12 13 14 15 16 17 18 19 19 19 19 19 19 19 19 19 19 19 19 19
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Lrown Drop at Leminum 1 (mm)	Crown Drop at Laminum 2	and so on a grad	more than one card if No. of laminae is greater than	o. of laminae is g	rester them 60.	
Card Type Bo, as meny RLAM(1,4)	y as seeded maximum (2,4)	Card Type B^6 , as many as needed maximum of 3, for the inner race. $(2,4)$	race. (4,4)	(5,4)	(6, 4)	(7,4) (8,4)
1 2 3 4 5 6 7 8 9 10 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1	11 2 12 13 13 13 13 13 13 13 13 13 13 13 13 13	1 2 3 4 5 6 7 8 9 10 11 m 13 m 16 17 18 17 22 21 12 12 12 12 12 12 12 12 12 12 12	2 0 2 4 4 4 4 4 4 4 4 4 4 4 4 4 4 4 4 4	43 44 45 44 73 48 49 59 57 5 F 1 0 . 0	7 2 4 27 33 59 64 4/ 12 4	43 44 45 46 97 48 49 59 57 58 52 57 58 50 64 12 44 47 42 44 57 48 78 77 78 78
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14) BD(15) BD(15) BD(15) BD(15) BD(15) BD(15) BD(16) BD(16) BD(16) BD(16) BD(16) BD(18) BD(18 Rolling Element Asperity Slope(Begrees) Oster Rolling Element Cr.A Roughness Surface (microns) Inner Bearing Data Card Type B7A, one card 211101 0 80 (13) (13) of 5 (5) Onter

Umit if NPASS (column 80, Card T12) is 0 or blank.

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Omit if NPASS (column 80, card TT2) is 8 or blank.

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BD(41)	W 65 72 66 62 85 86 85 80 80 80 80 80 80 80 80 80 80 80 80 80	CAGE WEIGHT (kg)
BD(40)	041,42,43,44,43,40,49,20	ROLLING ELEMANT CAGE POCKET DIAMETRAL CLEARANCE (mm)
BD (39)	7. 8. 5. 5. 5. 5. 5. 5. 5. 5. 5. 5. 5. 5. 5.	RAIL-LAND DIAMETRAL CLEARANCE (MM)
BD(38)	05 65 65 65 65 65 65 65 65 65 65 65 65 65	SINGLE RAIL WIETH (mm)
BD(37)	5 5 6 6 7 6 7	RAIL-LAND DIAMETER (MM)
BD (36)	0. 0. 0. 0. 0. 0. 0. 0. 0. 0. 0. 0. 0. 0	(ALF EN PL1. FOR CUTER-RING LAND RIDING -1. FOR INNER-RING LAND RIDING 0. FOR ROLLING ELETTENT RIDING

PEARING HAIA CARD TEPT BS ONE CARD PLR BEARING OMIT II NPASS, TITLE CARD 2, IS ZERO OR BLANK

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ne(57) 80(BO (58)	B0(60)	(19)08	BD (62)	BD(64)	
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INPUT THESE VALUES ONLY WHEN EXECUTING NASA VERSION (SEE APPENDIX F)

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Thermal Data - Individual Initial Temperatures $\binom{0}{C}$ 73, As many cards as needed, followed by a blank card

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Thermal Data - Bearing Node Numbers

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-	×	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	Balk Flance Flance Flance 63 as
-	×	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	Balk Flance Flance Flance 63 as
-	×	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	Balk Flance Flance Flance 63 as
-	×	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	Flance Flance Flance Sance as
-	ж ж	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	Balk Flance Flance Flance 63 as
-	×	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	laner Baik Flance Flance Flance
-	ж ж	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	laner Baik Flance Flance Flance
-	ж ж	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	laner Baik Flance Flance Flance
2 3 4 5 6 7 6 9 10 11 12 13 14 19 16 17 18 19 20 21 22 24 24 25 26 20 20 31 72 33	ж ж	1   5   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6   1   6	Balk Flance Flance Flance Flance 62 as

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See Fig. 3.5 for the blange numbering streme. Note that a tapered relier bearing inner ring Flange is Flange sl. Bull Bearings are not considered to have Flanges.

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rmal Data - Nodes Where Bearing Frictional Heat is Generaled

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e a 15A card immedia	8 9 m 11 12 13 14 15 16 19	×	5 1   5		
ude a 15A card immedia	7 6 9 60 11 12 13 14 15 16 17	×	5 1   5 1		
clude a 15A card immedia	6 7 8 9 10 11 12 13 14 15 16 17	×	5 1		
Include a TSA card immedia	4 5 6 7 6 9 10 11 12 13 14 15 16 17	×	5 11		-
1. Include a 15A card immedia	4 9 5 7 6 2 11 24 6 8 4 9 5 7	×	1 5 1 1 5	Flange 41 Flange #	
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75A, Include a 15A card immediately after the TS card	1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1	×	5 1   5   1   8		

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The viscosity is a function of temperature ORIGINAL PAGE OF POOR QUALI The forced convection coefficient can be calculated internally by the program using c + of three options. For options 1 and 3,  $\alpha_w$  is calculated by  $\alpha_w = \lambda Nu/L$  where Nu = KReAprB (Re = Reynold's No. = ULp/n, Pr = Prandtl No. =  $\eta C_p/\lambda$ ). For option 2,  $\alpha_w = c_0 D$ . The viscosity is a function of temp 27 74 E4 24 14 04 69 00 49 99 50 0 p = Density (KG/H3), Cp = Specific Beat (W-SEC/KG-OG), v = Flow Rate (H3/SEC) فالاعاجم فعاصه احتاجها لاطاعها لاطاعها لاطاعها بحاصط فعاعه استوعه إدعاءه أداجه إمان فوقعدا محامد احدامواد لاعدا 6 1 0 <u>- 1</u> (Q = KA AT/L) (Q = 6 A AT#) (Q = 6 A AT#) EXPONENT USED WITH TEMP. DIFFERENCE a(default - 1,25) F 10 MAGNITUDE OF HEAT TRANSFER COEF-FICIENT F 1 0 11-20 (Free Conv.) av(w/m-oc) 21-30 (Forced Conv.) av(w/m²-ocª) 21-30 (Forced Conv?) av(w/m²-oc) 31-40 (Radiation) (00/A)4dod Transport) INDEX FOR LATER
IDENTIFICATION OF
HEAT TRANSFER
COEFF.

Input one or two cards per coefficient. Use as many T? cards as required followed by a blank card.

DATA CARD T7
HEAT TRANSFER COEFFICIENTS

) (W/M-0c) (m/=-°c) To activate this feature, input data for card T? as shown below and follow immediately with card T7A. U (M/SBC) U (M/SBC) 3 | 3 **404** for options 2 and 3. HEAT TRANS. INDEX 21-30 21-30 21-30

325

(0PT10M (0PT10M (0PT10M

Input only if it is desired to calculate forced convection coefficient internally by the program.
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#### APPENDIX D

SERF COMPUTER PROGRAM SHABERTH
SAMPLE OUTPUT

#### APPENDIX D

# SKF COMPUTER PROGRAM SHABERTH SAMPLE OUTPUT

The SHABERTH output samples which are displayed on Pages D:3 to D:94 represent SKF and NASA versions of the code for three different shaft-bearing systems. The first set of examples present the results obtained with the SKF and NASA versions for a system in which an input pinion is supported by a preloaded pair of tapered roller bearings in a straddle configuration.

The road set of examples display program output for a system a which an input pinion gear load is supported by a flanged cylindrica. Ther bearing in conjunction with two angular contact ball bearings.

The third set of examples represent execution for a single ball bearing system operating under a combined radial and thrust load.

The differences between the SKF and NASA versions of the code which are described in detail in Appendix F, are reflected in the following output parameters:

- Value of EHD film thickness.
- Ratio of film thickness to surface roughness (H/Sigma) printed for most heavily loaded rolling element.
- Lube-life factor.
- L₁₀ fatigue life.
- Frictional heat generation rates.
- Ratio of load carried by asperities to total contact load  $(Q_{ASP}/Q_{TOT})$  [for NASA version this ratio = 0, see Appendix F].

## SKF

## 2 TAPERED BEARINGS LEVEL 1

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A38 - LA11706 STRADOLE

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*** SHARERTH/BRL ** TECHNOLOGY DIVISION SKF INDISTRIES INC.

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•	aaa - Lalitaa stpaaale	3.6	OF CHITTE SECTIONS 1 LOAD SECTION(S). 2 REARINGS. MODULUS OF ELASTICITY = 2.041+05	-150d	110		~	•	47.5	<b>6</b>	~	
,		SWAFT REPARTOR, REARING LOCATIONS AND SHAFT LOAD.	•									
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		MAR - LM11706	STRADOLE	بعد									
SHAF	SHAFT GEONETRY, BEA	TRY. 8E	ARING LO	CATIONS	A WO SHI	RING LOCATIONS AND SHAFT LOAD,		PLANF X - Z.					
	4 GEONE	GEOMETRIC SEC	CTIONS	1 1040	SECTION	(\$). 2 86	CAR INGS.	HOUGHTOS	TIONS I LOAD SECTION(S). 2 BEARINGS. MOCILLUS OF ELASTICITY = 2.041.05	'Y = 2.0	141.05		
THRU	THRUST LOAD = -2.14		147403										
	POST- 110M	INNER	OLAN.	OUTER LEFT	OUTER DIAM. LEFT RIGHT	FORCE	POINT	LOAD	LOAD INTENSITY LEFT RIGHT	POS-FR	REARINC SEAT Pos-err defl/for aug.err defl/40x	SEAT AUG.ERR	DEFL/109
-	ė	16.7	16.1	33.3	33.3								
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METRIC UNITS

	LIMEAN CRA		AND ANGULAR (RADIANS) DEF! ECTIONS	AUTANS) DEF	ECT TONS	ά.	FACTION FO	RCES (N) A	REACTION FORCES (N) BED ACMENTS (ME-E)	(H-#E)	
€ €	×	70	32	<b>&gt;</b>	29	č	ř	F 2	ž	24	
~ ∿	3.373-83	7.256-03 -2.381-03	3.373-03 7.236-03 -5.337-03 -2.030-05 -1.884-04 -2.689+03 1.463+03 -1.341+03 1.079+14 3.373-03 -2.381-03 -2.946-03 -1.276-04 -1.291-04 5418121.366+03 -4.712+03	-2.030-05	-1.844-04	-2.689+83 541.	1.463+03	-1.341+03	1.079-14	1.148+04	
	FATIGUE LIFE	LIFE (HOURS)	S	H/S16MA		LURE-LIFE FACTOR	FACTOR	HATERIAL FACTOR	FACTOR		
	0. RACE	I. RACE	RACE BEARING	0. RACE	I. RACE	O. RACE	1. RACE	O. RACE	1. RACE		
- ~	5.008.01	5.936+04	2.958+84	1.97	1.79	1.27	1.95	N N 0 0	35 00 00 00		
	TEMPERATURES		RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)	MING PERFC	RMANCE (D)	EGRFES CFW	TIGRADE 3				
2 - 2 3 - 2	7.RACE 100.00 190.00	.RACE' BULK OIL F 180.08 188.68 108.08 188.69	1.PACE BULK OIL FLMG.1 FLMG.2 FLMG.3 FLMG.4 CAGE SHAFT 180.00 180.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100.00 100	.00 .00	FLMG. 3	4.00.	CAGE SHAFT 100-00 100-09 100-00 180-00	c •	1.RING ROLL.FL. D.RING 100.nd 100.00 100.00 100.00 100.00 100.00		HSG. 100.00 100.00

ORI	GINAL	PAGE	13
OF	POOR	QUALI	TY

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COMDUCTIVITY (V/DEG.C)	CONDUCTIV	MENISCUS DIST. (MM)	MENISCUS D		THERMAL FACTOR	FALTOR	STARVATION FALTOR	CRONS	FILM (MICRONS)	. C M D
JUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEVAYS RESPECTI	40 TRNER A	HE OUTER A	DATA FOR T	DUCTIVITY	HEAT CON	ICTCRS AND	RESUCTION FA	ESS+ FILM	END FILM THICKNESS. FILM	EMD 7
	107.	2.923.03	18.6	1.154.03 1.155-07 45.7 1.853-08	1.154.03	172. 34.7	414.	000	1.066.03	- 2
	TCROUF	TOTAL	'I. FLNGS. R.E.DRAG R.FCAGE CAGE-LAND	R.FCAGE	R.E.DRAG F	I. FLNGS.	1. RACE	O. FLN6S.	O. RACE	<b>6</b> 0
			( MM-N)	ION TORQUE	AND FRICT	(UATTS)	ERATION RATE (WATTS) AND FRICTION TORQUE (N-MM)	FRICTIONAL HEAT GENER	FRICTIONA	
					METRIC UNITS	ME TRI	0 U T P U T	S + S + E H	> S	α 
								TRADOLE	MAR - LM117C) STRADOLE	r E
8 R L •••	R + K / E	TECHNOLOGY DIVISION SKF INDUSTRIES INC SHARERIM / BRL	S TMC.	INDUSTRIE	w. ×	JY DIVISIO			K - K	T O

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SHABERTH/RRL *** ** TECHNOLOGY DIVISION S K F INDUSTRIES INC.

HAR - LYII700 STRABOLE

EARTHS SYSTEM OUTPUT METRIC UNITS

LURAICAM TEMPERATURES AND PHYSICAL PROPERTIES

11 Y /N)	ed ed ec ed		1TY //I)	## <b>#</b> # ##	· <del></del>			CAGE/SHAFT Ratio	. 4.1 404.
PRESSURE VISCOSITY COEFFICIENT (MM2/N)	•1313-01 •1313-01 •1313-01		PRESSURE VISCOSITY COEFFICIENT (MM2/4)	•1313-01 •1313-01 •1313-01	1313-01			CALC/EPIC CA	1.00
ວ	44 E E		S	F & &	<b>e</b>			SPEED (RPM)	1.553+04
L TRUFLATIES VISCOSITY OYNAMIC (CP)	4 4 4 4 6 6 6 6 6 6 6 6 6 6 6 6 6 6 6 6	DPERTIES	VISCOSITY Ovnamic (CP)	4 4 4 4 4 4 4 4 4 4 4 4 4 4 4 4 4 4 4	<b>♣•6</b> 88	EDON)		CALCULATED (RAD/SEC)	1.527+03
	5.00 5.00 5.00 5.00 5.00 5.00 5.00 5.00	HYSICAL PR		5 • # # # # # # # # # # # # # # # # # #	5• ♣ & &	SREE OF FRE	DATA	SPEED (RP4)	1.553+04
KINEMATIC (CS)	<u> የ</u>	URES AND PI	KINEMATIC (CS)	ம்ம்	IŲ.	HAS ONE DFGREE OF FREEDOM)	CAGE SPEEN DATA	EPICYCLIC (RAD/SEC)	1.527+03
DF43ITY (GM/C43)	4440 4440 4440 4440	IBRICANT TEMPERATURES AND PHYSICAL PROPERTIES	0E451TY (BH7C43)	ቀ ቀ ቀ ቀ ቁ ቁ ነ ሆነነ ደ ແ ፎ	<b>₹</b>	(CA3E		ECCENTALCITY RATIO	. 983
S	000	L. UBR I		000	000	UMITS	D DATA	SFP.FORCE (NE UTONS)	165.
; EYPERATURES (DEGREFS C+)	100.000 100.000 100.000		TEMPERATURES (DEGREES C+)	100.000	1 110 • 0 00	A YETRIC UNITS	CASE RAIL - RING LAND DATA	HEAT GATE	13.5
LOCATION	JUTES 14NES BULK FLANSE		LICATION	OUTER TANSA HULK	FLASSE	0 A	CASE RAIL	TOROUE H	8.54 .893
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L-111707	
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FOR BEARING NUMBER I METRIC UNITS

A Z I MUTH	•	FNOULAR SPEEDS (RADIANS/SECOND)	(RADIANS	/SECOND)	SPE	SPEED VECTOR ANGLES (DEGREES)	(DEGREES)
V4.1LE (DES.)	ž	>	2 <i>h</i>	TOTAL	ORBITAL	TAN-1 (BY/BX)	TAN-1(82/4X)
	-12973.152	1.173	.000	12073-152	1626.701	179.99	180.00
22.50	-12873-152	1.192	000.	12873-152	1626.701	179.99	180.00
	-17875-152	000.	000.	12873-152	1626.701	180.00	180.00
	-12475-152	.300	.003	12873-152	1626.701	189.00	180.00
	-12475-152	000.	000.	12873-152	1626.701	180.00	180.00
	-12373-152		.000	12873-152	1626.701	180.00	189.00
	-12473-152		.000	12873.152	1626.701	183.00	180.00
	-12473.152		.000	12873-152	1626.701	180.00	180.00
	-12473-152		•000	12873-152	1626.701	180.00	180.00
	-12d/3.152		.000	12873.152	1626.701	180.00	180.00
	-12873-152		.000	12873-152	1626.701	180.00	190.00
	-12373-152		0000	12873.152	1626.701	180.00	180.00
	-12373-152		.000	12873-152	1626.701	180.00	180.00
	-12973-152	. 482	000	12873-152	1626.701	180.00	180.00
315.00	-128/3-152	•	•000	12873-152	1626.701	179.99	140.00
	CHE CRECE	•	6	CHI TECC			***

ORIGINAL PAGE IS OF POOR QUALITY

LLING ELEMENT OUTPUT FOR REARINS NUMBER I METRIC UNITS

*** S H A B E R T H / 9 R L ** TCCHNOLOGY DIVISION S K F INDUSTRIES INC.

MBR - LM11780 STRADDLE

7	AZIMUTH	NORMAL FOR	FORCES (NEWTONS)	(SNC	HZ STRESS	(N/MM··2)	HZ STRESS (N/MM++2) LOAD RATIO GASP/GINT	DASP/9TOT	CONTACT	CONTACT ANGLES (NEG.)	
ž	ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	THAFP	OUTER	INVER	
	.00	000	673.013	539.251	913.591	818.346		1700	-20.00	-15.05	
	22.50	000	391.783	49.384	7:1.319	421.755		. 0 4 4 2	-20.00	-15.05	
	45.00	000	118.884	000.	479.254	.000		.000	-20.00	-15.05	
	47.50	000	115.048	.000	473.30B	000		0000	-20.00	-15.05	
	90.00	000	111.228	000.	467.301	.000		.000	-20.00	-15.05	
	112.50	_	107.419	.000	461.231	000		1000.	-20.00	-15.05	
	135.00	000	103.528	•000	455.097	.000		-000	-20.00	-15.05	
	157.50	_	99.852	0000	448.897	0000		.000	-20.00	-15.05	
	180.00	• 000	96.093	000.	442.627	000		.000	-20.00	-15.05	
	292.54	.000	92.352	000.	436.285	000.		0000	-20.00	-15.05	
-	225.00	000.	88.629	000.	429.968	.000		.000	-20.00	-15.05	
	247.50	.000	355.093	13.326	730.303	275.270		-02 12	-20.00	-15.05	
. •	270.00	000	612.464	269.930	BA3 - 344	752-19R		.0551	-20.00	-15.03	
. •	292.50	000	872.448	529.116	938.219	959.391		10647	-20.00	-15.05	
- 1	315.00	000•	987.299	643.690	1044.340	1031.355		.0664	-20.00	-15.05	
	337.58		910.181	566-723	1015.927	987.680	.0439	.0653	-20.09	-15.05	
HERTZ STRES!	HERIZ STRESS ACROSS THE DUTER		RACE WAY-ROLLER PROFILE	PROFILE							
.0000	.3719+05	0	15 -1267+06		90,	1448+06	.1373+06	.1166.06	.7392+05	0000.	
0000.	0000	0000	000.	•	0000	0000	0000.	.0000	.0000		
	ACROSS THE INVER R	NWER RACEVA	IY-ROLLER F	PROFILE							
0000	0000	.8332+0	332+05 .1203+06		.1370+66 .	•1331+06	•1255•06	-4355+05	0000	0000.	
0000	0000	.0000	000.			0000	0000	0000	.0000	.0060	

** TECHNOLOGY DIVISION

484 - LM11700 STRADOLE

OUTPUT FOR BEARING NUMBER 1

¥ 18	· La.	· ;;	•															
HY DR 0 - D	HEAT GA	CUATT	2.15	2.75	2.14	2.73	2.74	2.74	2.74	2.74	2.74	2.14	2 4	2.74	2.17	2.79	2.40	2.40
CONTACT	HEAT GATE	(UATTS)	9.17	7.34	6.97	6.91	6.31	6.97	16.9	20.9	4.97	6.91	16.9	7.11	8.17	10.51	11.30	10.11
FILM FILM	THICKNESS	(MICD) ")	41.40	.4641	16,14.	1574.	1474.	1534.	.4657	1694.	1634.	46.54	1494.	16.34	22.24	.1555	.4512	46240
St. 1014.	VELOCI1	(S/H)	14.4197	14.4187	14.4187	14.4187	14.4197	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187
ROLL 14G	VEL OC 1 TY	(H/3)	45.940	45.340	45.940	45.940	45.940	45.940	45.340	45.940	45.940	45.940	45.340	45.940	45.340	45.940	45.940	45.940
	AXIS			.410								.405						
MAU-SIMI	AXIS	Î E	.472	.445	.439	.439	.439	.439	.433	.439	.439	.439	.439		.466	.489	664.	.492
HERT2	TRES	/ HH +	349	329	324	324	324	324	324	324	320	324.730	324	326	344	361	368	364
CONTACT	L 0 A D	€	149.913	125.644	120.686	120.686	120.686	120.586	120.686	120.586	120.686	120.686	120.686	122.466	144.680	167.120	177.033	170.379
AZ IMUTH	31944	(056.)	00.	22.50	45.00	67.50	90.00	112.50	135.00	157.50	180.00	202.50	225.00	247.50	270.00	292.50	315.00	337.50

FLANGE-ROLLER LUBRICATION AND FRICTION DATA FOR THE MOST HEAVILY LOADED CONTACT AND CONDUCTIVITY FOR ALL ROLLFAS.

OASO/GIOT HEAT GENERATION RAIFS TOT FLW:-ROL (WATTS) CONDUCTIVITY CONTACT INLET (W/DEG C)	A 1.25
TON RATES	2.7A
HEAT GENERAT (NATTS) Contact	9.17
0459/9707	. 144
EFFECTIVE Traction Coefficient	•00•
H/SIGHA	2.397
ION FACTORS H/SIGHA Thermal	• 823
FILM REDUCTION STARVATION T	.520
EMD FILM THICKNESS (MICRONS)	150.
FLANGE NO.	•

TECHWOLOGY JIVISION

FOR HEARING NUMBER 2 METRIC UNITS

A214UTH	*	ANGULAR SPEEDS (RADIANS/SECOND)	(RADIAHS	/ SEC OND)	las	SPEED VECTOR ANGLES (JEGREES)	()EGREES)
Marc (nfa.)	¥?	<b>*</b>	71	TOTAL	ORTITAL	TAH-1(UY/UX)	144-1682/WX)
.00	-3575.741	000.	0000	95750741	1531.482	180.00	180.00
00.0	-7576.741	000.	.190	9575.741	1531.482	180.00	140.00
2 • n 3	-7575-741	000.	100.	9576.741	1531.482	180.00	180.00
10 1.00	-9576.741	000,	.000	9576.741	1531.482	189.00	180.00
00.1	-3576.741	-1.229	.000	9576.741	1531.482	-179.99	130.00
00.0	-3576-741	-1-113	•000	9576.741	1531.482	-179,49	180.00
5-09	-4576.741	728	.000	9576.741	1531.482	-180.00	190.00
00-3	-9576.741	213	.000	9576.741	1531.482	-180.00	120.00
00.6	-3575.741	. 235	.000	9576.741	1531.482	180.00	180.00
00.	- 3576 - 741	426	000	9576.761	1511,182	60.081	00.081

OLLING ELEMENT OUTPUT FORBEARING NUMBER 2 METRIC UNITS

** TECHNOLOGY DIVISION S K F INDUSTRIES INC.

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ANGLE (DEG.) CAGE OUTER	INNER	OUTCA	INNER	OUTER	INNEP	OUTER	INNER
.00 .001 46.7			000	4471	6		;
				- DC -	0000	10.50	7.57
			000	.1684	J000.	10.80	7.37
800.	140		000	.1683	0000	10.00	7 2 7
			000	1601			
	711					99.04	16.7
			367.784	1937	. 2621	10.60	7.37
900.			973.613	.1916	2000	10.80	7.1.7
•	52 573-447		1174.928	98.6	0010		
					02020		1001
		•	1203.848	-1822	.2273	10.80	7.37
			1072.586	.1795	-227E	10.80	7 - 7
	149 197.752	12 671-142	757.055	.1802	.2390	10.90	7.37
DUTER	RACEUAY-ROLLER PROFILE	,	,				
0000* 0000*	.0000	. 0000	.0000	.1320+06	.1295+86	.1256+06	.1202+06
ACROSS THE INNER RACEMAY-ROLLER PROFILE 1513+06 .1542+06 .1556+06 .1555+06 .0000 .0000	OLLER PROFILE	.1539+06	*1508+06	.1460+06	.1393+06	.1305+06	.1131+06

** SHABERTH / BR L ** ** TECHNOLOGY DIVISION S K F INDUSTRIES INC.

MAR - LMII700 STRADOLE

LER-FLANGE OUTPUT FOR BEAKING NUMBER 2 METRIC UNITS

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	I							4184				
SLIDING	VELOCITY	(H/S)	8.1400	8.4400	A.4.80	8.4400	8.4408	8.4400	8.4400	8.4460	8.4400	
ROLLING	VELOC 1 TY	(W/S)	24.746	24.746	24.746	24.746	24.746	24.746	24.746	24.746	24.746	
IN-SENI	AXIS	(HE)	.183	.183	.183	.183	.217	.269	.297	.301	.280	
MAJ-SINI M	AXIS	(HH)	.206	.206	.206	.206	.244	.303	.334	.338	.315	
HERTZ	STACSS	(N/4H··2)	133.425	133-425	135.425	133.425	158.108	195.876	216.183	218.883	203.968	
CONTACT	LOAD	3	10.573	10.573	10.573	10.571	17.594	33.453	44.973	46.680	37.173	
AZIMUTH	ANGLE	(1)	00.	36.00	72.00	108.60	144.00	180.00	215.00	252.80	284.00	

FLANGE-ROLLER LUBRICATION AND FRICTION DATA FOR THE MOST HFAVILY LOADED CONTACT AND CONDUCTIVITY FOR ALL ROLLERS.

DASP/GTOT HEAT GENERATION RATES TOT FLMG-ROL (UATTS) CONDUCTIVITY T CONTACT INLET (W/DEG C)	40°
ION RATES INLET	•
HEAT GENERAT (VATTS) CONTACT	16
0.45P/0101	101
EFFECTIVE TRACTION COEFFICIENT	
	2.388
FILM REDUCTION FACTORS M/SIGNA Stanvation thermal	878
FILM REDUCT	.614
END FILM THICKNESS (41CRONS)	.456
FLANGE NO.	•

AT81D040

#### NASA

2 TAPERED BEARINGS LEVEL 1

MEARING NO. (2) - TAPERED ROLLER BEARING

BEARING NO. (1) - TAPERED ROLLER BEARING

THIS DATA SET CONTAINS 2 BEARINGS

TECHNOLOGY NIVISION S K F INDUSTRIFS INC. .. S H A B E R T H / H R L ...

*** SHABERTH/9RL

M38 - LM11700 STRADOLE

SOLUTION LEVEL = 1

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UNLESS JIMERWISE STATED. LINEAR DIMEMSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES "N DEGAETS CFNTIGAADE, FORCES IN MEUTONS. Weights in Kilograms, pressures and elastic modin, in meutons per square millimeter, angles and schoes in degrees, surface Roughness in microns, speeds in revolutions per minute, density in grams per curic centifeter, kitematic viscosity in centistokes and theral combuctivity in watts per metgr-degree centigrade.

** S H A B E R 1 H / A R L **

** TECHNOLOGY DIVISION S K F INDUSTRIES INC.

*** S H A H E R T H / G R L H H H B - LM11706 STRADDLE

OUTER RING Speco	• •		VEIGHT	.19696			
INNER RING Speed	35000.		RAIL-LAND CLEARANCE	.328		LIFE FACTOR	5,000
CONTACT	10.900		RAIL-LAND DIAMETER	39.0500			
DJAMETRAL CLEARANCE	0000		RAIL-LAWD UIDTH	1.9050		OUTER RING TYPE	STEEL BUTER RING
P11CH DIAMETER	000		CAGE POCKET CLEARANCE	.230000		CTOR	3.000
AZIMUTH ANGLE ORIENTATION	• • • • • • • • • • • • • • • • • • • •			RIDING		LIFE FACTOR	en a
MUNRER OF Rolling Elements o	9.0	<b>4</b>	CAGE TYPE	INNER RING LAND RI INNER RING LAND RI	4 7 4	INNER RING TYPE	STEEL THREE RING
BEAR ING NUMBER	-~	AGEOATA	bearing Number	••• Ry	STEEL DATA	BR 6 . NO .	- 6

WUMBER OF AXIAL LAMIMAE 10	BEARING AXIAL PLAY 6534		NUMBER OF AXIAL Laminae 10	BEARING Axial PLAY PLAY eess
ROLLER INCLUGED ANGLE 4-9500	IMMER HING Flange Angle 15.5500		ROLLER Included Angle 3.4330	INNER RING FLANGE ANGLE 7.7968
ROLLER END Radfus 87.300	LARGE END COANER RELI		ROLLER END Radius 95.3878	LARGE END CORNER RELIEF
ROLLER Crown Ractus 4500.0000	INMER RACEUAY Croum Radius • 0000		ROLLER Crown Radius 12708-8600	INNER RACEVAY CROUN RADIUS •0000
ROLLER PROFILE Flat LENGTH .0000	) EFFECTIVE 5LIEF LEMGTH 30 15.0000	LLER BEARING	ROLLER PROFILE FLAT LENGTH C	) EFFECTIVE LLIEF LEMGTH 10 7.1170
000	AY LARGE END CORNER RELIEF	E - TAPERFO ROLLER BEARING	•	AY LARSE END CORMER RELIEF .0000
€ 3	OUTER RACERAY CROWN RADIUS	TYP	23	OUTER RACERAY CROWN RADIUS
R72LER D1 A4FTER 7.9400	EFFECTIVE LE46TH 15.0009	BEARING NUMBER (2)	40LLER 51 AMETER 5.7156	EFFECTIVE LE4GTM 7-1170

** TECMNOLOGY DIVISION S K F INDUSTRIES INC.

TYPE - TAPFAED ROLLER BEARING

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*** SHARERTH / BRL ** TECHNOLOGY DIVISION SKF INDUSTRIES INC. ** SHABERTH / BRL ***		
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	BE AR 1MG NUMBER		CLA ROUGHNESS Inner Fl.	OLL. ELM. END	OUTER FL.	RMS ASPERITY INNER FL.	SLOFE Roli. Elm. End
	1 2	21. 21.	.13	α ε. 0 0 . •	2.000	2.000	2.000
1   SAMTOTRAC 50   53.69   5.61   .AM99   4.10-04     2   SAMTOTRAC 50   33.69   5.61   .AM99   4.10-04     1   SAMTOTRAC 50   33.60   5.61   .AM99   4.10-04     2   SAMTOTRAC 50   33.60   5.61   .AM99   4.10-04     4   C A T I O N A N D F R I C T I O N D A T A	ר ח	1 0					
SANTOTRAC 50   33.60   5.61   .8890   4.10-04		DESIGNATION	KINEHATIC (37.7A C)	VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAM. COEFFICIENT	THERMAL CONDUCTIVITY
U B R I C A T I O N A N D F R I C T I O N D A T A  BEARING PERCENT LUBE LATER THICKNESS  "IN CAVITY LATER THICKNESS  "IN CAVITY LATER THICKNESS  "OUTER 1 INNER  1 2.80100.03  "EATER THICKNESS  COEFFICIENT LATER THICKNESS  COEFFICIENT4000-03  "4000-03  "4000-03  "4000-03  "4000-03  "4000-03  "4000-03  "4000-03	- 2		33.68 33.60	5.61	.8890	4.10-04	
PERCENT LURE   LATER THICKNESS   IN CAVITY   LATER THICKNESS   CAVITY   LATER THICKNESS   CAVITY   LATER THICKNESS   CAVITY   LATER THICKNESS   CAVITY   CAVITY   LATER THICKNESS   CAVITY   CAV	>	* * * * * * * * * * * * * * * * * * * *	+ 0 H	4			
2.00 .2000-03 2.00 .3000-03 .2000-03 CASE FILM REPLEMISHMENT FRICTION LAYER THICKNESS COEFFICIENT .4000-03 .10 .4000-03	BEARING Minder	PERCENT LUBE IN CAVITY	FILM LA? 'ROLL.E	REPLEMISMMENT ER THICKMESS LM. + RACEUAY ) R	ASP FRI COEFF	FRITY ICTION ICIENT	
CAGE FILM REPLENISHMENT FRICTION LAYER THICKNESS COEFFICIENT .4000-03 .4000-03 .4000-03	-~	5.00	.3080-03			0.6	
NO-0004" NO-0004"	REARINT NUMBER	CASE FRICTION COEFFICIENT	F11.8	REPLENISHMENT YER THICKNESS	1303	3-R-E • 10110N	
	<b> №</b>	.1.	. 4000-03			C.	

GIVEN FEMPERATURES

HSG. 100.00
0.81NG 100.00 100.00
POLL.EL. 100.00 100.00
1.81NG 1190.90
3HAFT 100.00 100.00
CAGE 100.00 100.00
FLNG.4 .00
FLN7.3
FLN5.2 .09
FLM:.1 103.09 107.00
30LK 91L 1 130.03
1.84CE 104.00 130.33
9. 20CE 193.03 130.00
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Treyoningy by I noustries inc.

SHAFF RECHFFOR BEARING LOCATIONS AND SHAFF LOAD.

Gramfreic sections i Load Sectionis). 2 afailmes, madulus of Elasticity = 2.041+05

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EAT Ang.Fqr	0060.	. 6000
HEARING SEAT POS.FRR DEFLIFOR ANG.FRR DEFLIMON	6.0.7	Ü••0
FOS.FR9	60-3 006-	000.
	- ~ n •	* FU 00 ~
LOAD INTENSITY LEFT RIGHT		
LOAD 1 LEFT		
POTUT	6.60.00	
FORCE	650.4	
PINA.	33.3 33.4 52.4 46.9	17.5 17.5
OUTER LEFT	20 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0	17.5
7144.	15.7	9.2
INNC4 LFFT	15.7	16.7
051- TION	13.0	59.7 74.5 108.0

** TEC" NOLOGY DIVISION S K F INDUSTRIES INC. HAR - LMII100 STRADOLE

4 GEOMETRIC SECTIONS 1 LOAP SECTION(S). 2 BEARINGS. MODILUS OF ELASTICITY = 2.041+05 SHAFT GEOMETRY. BEARING LOCATIONS AND SHAFT LOAD, PLANF X = 2.

	DEFL/30A		6.0				00.0	•
	EAT Ang.err		9090				9000	
	REARING SEAT DEFL/FOR ANG-ERR DEFL/MOM		0.00	! •			00.0	
	POS+ERR		000.				000	
			~	<b>(*7</b>	•	S)	v	~
	LOAF INTENSITY ( T RIGHT							
	POINT							
	FOINT				-2105.2			
	BIAM.	33.3	33.3	52.4	46.9	17.5	17.3	17.5
	OUTER LEFT	33.3	33.3	33.3	46.9	11.1	17.5	17.5
107403	DIAM. RICHT	16.7	16.7	16.7	16.7	9.4	9.2	9.2
-2-1	INNEA LEFT	16.7	16.7	16.7	16.1	15.1	4.2	9.5
RUST LOAD = -2.147	P051-	•	19.3	39.4	49.5	29.1	74.5	108.0

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-	115.A.3	LFN543 (44) A	.'N1 GA1	ULAR CRAC	AND INCULAR (RADIANS) DEFLECTIONS	LECTIONS	~	FICTION	ORCES (N)	RESCTION FORCES (N) AND MOMENTS (MM-N)	(MH-N)
3	J	0 7		20	¥9°	29	×	<b>F</b>	F.2	E	M 2
	75-03	7.256	2- £0	•537-r3 -	.2.030-05 .1.276-04	\$.375-03 7.256-03 -6.537-03 -2.930-05 -1.844-04 \$.375-03 -2.381-03 -2.946-03 -1.276-04 -1.291-04	-2.689+03 541.		1.463+03 -1.341+03 -8121.366+03	-1.341+03 1.079+04 -1.356+03 -4./12+03	1-148-04
	leuf L	FITTOUR LIFF (HOURS)	เรชกษ		HZSICHA		LUBE-LIFE FACTOR	FACTOR	MATERIAL FACTOR	FACTOR	
	O. RACF	I. RACF		HEARING	O. RACE	I. RACE	0. RACE	1. RACE		I. RACE	
	5.531+09 4.192+04	6.371+J4 4.028+03		2.447+04 7.048+03	1.59	2.30	2 • 42 1 • 95	2.29	5.00	90°5	
7	praar	Tryporatures a	FLEVINI	T TO YEAR	IV3 PERFO	RMANCE COE	FELEVANT TO YEARING PERFORMANCE (DEGREES CENTISRADE)	ISRADE)			
	HRG 7.44CF 1.100.26 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1.200.25 1	1.3ACF H 1.7.00 1.00.03	190.03	1918 OIL FLV5.1	FLNG.2	FLNG.3 FLNG.4		C4GE SHAFT 100.00 100.00		1.RING ROLL.EL. 100.00 100.00	0.RIN3

S K F INDUSTRIES INC.

** TECHNOLOGY BIVISION

METRIC UNITS 104116 W S L S L S

INDUSTRIES INC.

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TECHNOLDGY DIVISION

138 - LAILTON STRANDLE

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FRICTIONAL HEAT GENERATION RATE (WATTS) AND FRICTION TORNUE (N-MM)

FHD FILM THICK ISSS. FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEMAYS RESPECTIVELY MENISCUS DIST. (MM) CONDUCTIVITY (W/DEG.C) 1.17 984. 101. TORGUE 3.711+03 1. FLNGS. R.E.DRAG R.E.-CAGE CAGE-LAND TOTAL .420 13.6 2.00 1.136+03 1.154+03 1.155-07 28.1 45.7 1.863-08 .819 .872 THERMAL FACTOR STARVATION FICTOR . 3130 . 3136 1. FLMG: I. RACE 1.020+05 0.000 133. 0.006 FILM (MICRONS) THU. D. RACE j D:29

# E Q T H / B R L ***	PRESSUR	•1313-01 •1313-01 •1313-01 •1313-01	PRESSURE VISCOSITY COEFFICIENT (MM2/1)) •1313-01 •1313-01 •1313-01	CALC/EPIC CARE/SHAFT RATIO RATIO 1.00 1.00
INDUSTRIES INC. ** S H A	ICAL PROPERTIES  VISCOSITY  DYNIMIC (CP)  4-68H	PROPERTIES	0YNAMIC (CP) 4.689 4.688 4.688	EEDOM)  CALCULATED SPEED (RAD/SEC) (RPM)  1.527+02 1.553+0A  1.531+03 1.462+nq
TOGY DIVISION SKF	LURRICANT TEMPERATURES AND PHYSICAL PROPERTIES (GM/C43)  **INEMATIC (CS)************************************	THERATURES AND PHYSICI	5.488 5.489 5.489 5.488	CA HAS ONE DEGREE OF FREEDOM)  CASE SPEED DATA  10 (RAD/SEC) (RPM) (RAD/SEC)  77 1-527-03 1-553-04 (RAD/SEC)  19 1-511-05 1-562-04 1-5531
57440 h.c.	TE 42ER 1TURES (DEGRETS C.) 108.000 103.000 -AC.000	62		HING LAWE BITA  HINT GATE STATTS) THE JTOWNS  FASS
MA9 - L411700 6 E & R I N 7 C Y	LOCATION  GRS. 1 JUNER  TUNER  SULK  FLANSE	LOCATION 9RG. 2 DUTER FYNER	∢ ,	1 50

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S K F INDUSTRIES INC.

TICHARLOCK CAISION

FOR BEARING NUMBER 1 METRIC UNITS

4 Z I MIJ H	Z	ANGULAR SPEEDS (RAGIANS/SECOND)	(RACIANS	/SECOND)	SPI	SPEED VECTOR ANGLES (DEGREES)	(OEGREES)
VI 1. E (UES.)	š	5	71	TOTAL	ORBITAL	TAN-1 (UY/UX)	TAN-1 (WZ/4X)
. n 3	-12873-152	1.173	•000	12873-152	1626.701	179.99	180.00
9.50	-12#73.152	1.192	•000	12873-152	1626.701	179.99	190.00
00.	-17873-152	000.	060*	12873-152	1626.701	180.00	189.00
61.50	-12873-152	.3r0	000	12873-152	1626.701	180.00	00 00 0
00.	-12475-152	000.	000.	12873-152	1626.701	180.00	14.000
01.	-12473-152	0.00	.000	12873-152	1626.701	180.00	18.3.06
135.06	-12473-152	000.	006*	12873-152	1626.701	180.00	180.00
• 50	-12473-152	000.	0000	12873-152	1626.701	186.00	180.00
.00	-12473-152	000.	000	12873-152	1626.701	180.00	180 .00
• 50	-124/3.152	000.	000	12873-152	1626.701	180.00	180.00
.00	-12873-152	.000	000.	12873-152	1626.701	180.00	180.00
00.	-12373.152	. 352	000.	12873-152	1626.701	180.00	180.00
00.	-12373.152	673	.000	12873-152	1626.701	180.00	186,00
.50	-12473-152	286•	000	12873-152	1626.701	180.00	186 30
06.	-128/3.152	1.253	000	12873-152	1626.701	179.99	189.00
5	-12274.142	1.100	000	12474,152	1626.701	170.00	100,001

S K F INDUSTRIES INC. TECHNOLOGY DIVISION 499 - LYIITED STRADOLE

FOR BEARING NUMBER I METRIC UNITS

•																					
CONTACT ANGLES (OFG.)	INNER	10-51-	-15-05	-15.05	-15.05	-15.05	-15.95	-15.15	-15.05	-15.05	-15.05	-15.05	-15.05	-15.05	-15.05	-15.05	-15-05		00000		.0000
CONTACT	OUTER	-20 -110	-20.00	-20.00	-20.00	-20.00	-20.03	-20.00	-20 -00	-20.00	-20.00	-20.00	-20.00	-29.00	-20.00	-20.00	-20.00		27332+05	0000	0000•
UASP/GTOT	IMER	0000	0000	0000	0000	.0000	.0000	•0000	÷0000•	1000	.0000	.0000	0000	.000	0000	0000	0000		.1156+P6	.000	.935 ^c +05
HZ STRESS (N/MM++2) LOAD RATIO GASP/GTOT	OUTER	0000	0000	0000	.0000	0000.	0000	.0000	.000	0000•	.0000	.0000	• 0000	0000	0000	0000	0000		*1373+06	0000	.1265+06
(N/MM++2)	INNER	814.346	421.755	.030	000.	000.	000.	0000	000.	000.	000	000.	275.270	752.19A	959.391	1031,355	947.640		.144R+D6	.0000	.1391+06 .0000
HZ STRESS	OUTER	913.591	757.919	479.254	473.308	467.301	461.231	455.097	448.897	442.627	436.285	427.869	730 - 303	88 G 6 G 6	398.219	1044.360	1015.927		.1413+06 .	-	.1370+05 .
(SNO.	INNER	330.251	41.304	000.	000	•000	.000	.000	000.	000	000.	.000	13.326	259.030	529.116	643.600	566.723	ROLLER PROFILE			40F LLE +06
RCES (NEWTONS)	OUTER	573.013	391.789	119.944	115.048	111.226	107.419	103.629	99.752	96.093	92.352	48.629	355.093	412.464	972.44B	387.295	710.181			.000.	05 .1203 05 .000
WORMAL FORCES	C 4 GE	000	100	000-	000	000	000	000	000	000	000	000	000.	000.	000	30U·	000	OUTER RACE	5 .9734+05	.0000	1'1VER RACE'4AY-R .8532+05 .0000
AZIMUTH	445LE (DEG.)	• 30	22.50	45.00	<b>57.</b> 50	39.00	112.50	135.00	157.50	1 40.00	212.50	225.00	247.50	210-00	292.50	315.00	337.50	HERTZ STRESS ACROSS THE OUTER RACEUAY.	.3719+05	0000	ACROSS THE 1 .3000
	3 D:3	32																HERTZ STRES	.000	.0000	00000

INDUSTRIES INC. SKF TECHNOLOGY DIVISION SHABERTH/BRL 484 - 1411700 ST440JLE

METRIC UNITS

FOR HEARING NUMBER

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HYDRO-DYN	HEAT MATE	(NATTS)	5.52	5.30	5.24	5.54	5.24	5.24	5.24	5.24	5.24	5.24	5.24	5.26	5.48	2.67	5.15	5.69
CONTACT	HEAT RATE	(NATTS)	19.58	58.00	53.17	53,17	53.77	53.77	53.17	53.77	53.77	53.17	53.77	55.27	75.14	95.20	104.94	98.34
END FILM	THICKNESS	(MICRO M)	.7650	.1715	.7750	.7750	.7730	.7756	.7730	.7750	.7730	.7750	.1730	.1725	.7663	.7610	. 7590	.7403
SLIDING	VELOCITY	(W/S)	14.4187	14.4187	14.4187	14.4187	14.4197	14.4187	14.4187	14.4167	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187	14.4187
ROLLING	VELOCITY	(S/H)	45.940	45.340		45.340	45.940	15.340	45.940	45.340	45.940	45.340	45.940	45.940	45.340	45.340	45.940	45.940
HIN-SENI	AXIS	£E)	.435	.410	. 405	405	.405	.465	.405	C04.	.405	.405	.405	.407	.430	.451	.460	.454
		(3.4)	72	.445	.439	.439	6433	(14.	.439	68 40	.433	.453	.439	.441	.466	.489	n64.	.492
217	rress	(:4M++2)	349.074	327.118	324.730	324.730	124.730	324.730	324.730	524.730	324.730	324.730	324.750	326.320	344.764	351.349	36 P . 46R	
CONTACT	(1407)	3	143.913	125.644	120.696	120.696	120.586	120.685	120.0A6	120.686	120.646	120.636	120.486	122.466	144.680	157.120	177.033	170.579
AZIMUTH																		337.50

FLANSF-ROLLER LU9RICATION AND FRICTION DATA FOR THE MOST HEAVILY LOADED CONTACT AND CONDUCTIVITY FOR ALL ROLLERS.

TOT FLMS-AUL CONDUCTIVITY (W/DEG C)	2 .69
TION RATES INLET	5.52
GASP/9TOT HEAT GENERATION RATES (WATTS) CONTACT INLET	79.54
QAS¤/910T	• 000
EFFECTIVE Traction Coefficient	.037
H/S184A	4.039
ON FACTORS H/SISMA THERMAL	.823
FILM REDUCTION	.52.
E4D FILM THICKNESS (MICRONS)	.753
FLANGF NO.	<b>~</b> ∩

S K F INDUSTRIES INC. ** TECHNOLOGY STVISTON

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A214UTH		ANGULAR SPEEDS (RADIANS/SECOND)	(RADIANS/	SECONDI	آه. د	SPEED VECTOR ANGLES	(DEGREES)
CE (OFG.)	×	<b>&gt;</b>	24	TOTAL	ORATTAL	TAN-1 (UY/UX)	TAN-1(42/9X)
00.	-3575.74		000	9575,741	1531.482	180.00	180.00
36.00	-7575.74		•000	9576.741	1531.492	180.00	180.00
72.00	-95750-74		.000	9576.741	1531.482	180.00	180.00
103.00	-9576.14		.000	9516.141	1531.482	190.00	180.00
144.00	-9576-74	•	.000	9576.741	1531.482	-179.99	180.00
190.00	-3576-74	•	•000	9576.741	1531.482	-179.99	180.00
216.09	-9576.74		•000	9576.741	1531-482	-180.00	190.00
252.00	-9776.74		0000	9576.741	1531.482	-180.00	180.00
299.00	-9575.741	1 .235	000.	9576 - 741	1531 - 482	180.00	180.00
324.00	-3576.74		000	9576.741	1531.482	186.00	180.00

S K F INDUSTRIES INC.

** TECHNOLOGY DIVISION

FOR REARING NUMBER 2 METRIC UNITS

.ES (DFG.)	INNER	7.37	7.87	7.57	7.37	7.37	7.57	7.37	7.57	.1292+06 .0000	-1191+06
CONTACT ANGLES (DEG.)	OUTER	10.80	10.00	10.80	10.90	10.80	10.80	10.80	10.80	.1255+06	.1305+06
2ASP/010T	INVER	000	0000		0000	2000	0000	0000	0000	.1295+06	.1393+06
HZ STRESS (N/MM++2) LOAD RATIO JASP/OIDT	OUTER	0000	0000-	0000	.0000	• 0000	0000	0000	0000	.1320+06	•1 460+06
(N/MH++2)	INNER	000	000	589.984	973.613	1174.928	1203.848	1072.586	757.055	.1552+06 .0000	1508+06
HZ STRESS	OUTER	344.656	344.092	602-404	882.451	1029.530	1039.684	918.575	671-142	.1332+06 .1	90+
(SNO	IMNEG	000.	.000	116.660	341.269	573.447	601.948	453.410	197-752	PROFILE .13	11.5
YORMAL FORCES (NEYTONS)	OUTER	46.58	46.541	172-719	437.674	630.152	65A.679	509,339	253.449	4Y-ROLLF4 (6 .1320-	17-ROLLER PROF 16 .1555+05 .0000
YORMAL FOR	CAGE	.001	.000					000-	031	.UTER RACEU .1297+0 .0000	NER RACE JAY 1556+06
AZI MUTH	ATTLE CUEGOD C	.10 35.00	72.00	144.30	190.30	j•30	252.00	244.00	324.00	ACROSS THE 0 .1250+06 .0000	ACROSS THE FINER RACEJAY-ROLI .1542+95 .1556+06 .0000 .0000
iv	A 13L		: 35		e :	21	55	5.4	32	HERTZ STRESS ACROSS THE OUTER RACEUAY-ROLLFY PROFILE •1211+06 •1250+06 •1297+06 •1320+06 •0000 •0000 •0000	4) -1515+05 -0033

. SHABERTH / HRL ... S K F INDUSTRIES INC. ** TECHNOLCGY DIVISION

483 - L411707 STRADDLE

. LER - FLA 4 GEOUTPUT FOR A EARING NU 4 A ER RIC UNITS

HYDRO-DYN HEAT RATE	(HATTS)	1.03	70°I	1.03	1.08	1.19	1.36	10.45	1.4.5	1.33	1-25
CONTACT Heat gate	(NATIS)	15.	.51	.51	100	6.	2.16	3.27	3.45	2.55	1.27
EHD FILM	(MICRO M)	.7769	.1769	.7764	.1159	.75.6.2	.7122	.7520	1621.	.1219	.7468
SL IUING VELOCITY	(S/k)	A.4400	8-1400	9-4400	8.4400	8.4400	8.4400	8.4400	8.4400	8.4400	9-4400
ROLLING S	(S/E)	24.746	24.746	24.746	24.746	24.746	24.746	24.746	24.746	24.746	24.746
MIN-SEMI AKIS	<del>S</del> E	58.T.	.183	. 1 R.3	.183	.217	.263	.291	.301	.280	.236
AJ-S191 AX IS	( <del>T</del>							.534			
HERTZ M	12 HF /N)	133.425	133.425	133.425	133.425	154.108	175.876	216.123	219.883	201.368	171.484
LONTACT HI		10.573	10.573	10.573	10.573	17.534	33.453	44.373	46.680	37.773	22.447
AZIMJTH AVGLF	100,300	00.	36.00	72.00	00-601	144.00	130.33	216.00	252.00	2 tA.00	324.00

FLAWSE-ROLLER LUARICATION AND FRICTION DATA FOR THE MOST HEAVILY LOADED CONTACT AND CONDUCTIVITY FOR ALL POLLERS.

2ASP/0TOT HEAT GENERATION RATES TOT FLNG-ROL (VATTS) CONDUCTIVITY CONTACT IHLET (N/OES C)	.15
TION RATES INLET	1.08
HEAT GENERAL (WATTS) CONTACT	.51
24SP/910T	000
EFFECTIVE Traction Coefficient	•00k
H/S16HA	4.072
ON FACTORS M/SIGMA THERMAL	R79
FILM REDUCTION STARVATION	.614
EHD FILM THICKUESS (MICRONS)	111.
FLAVGE NO.	₽0

### AT81D040

### SKF

- 1 CYLINDRICAL
- 2 BALL BEARINGS

LEVEL 1

S K F INPUSTRICS INC. ** TECHNOLOGY DIVISION MASA ALTERNATE INPUT PINION DESIGN 24, 601 LOAD

THIS DATA SET CONTAINS 3 BEARINGS

HEARING NO. (1) - CYLINORICAL ROLLER BEARING . FLANGE IS TYPE NO. 1.

REARING WO. (2) - BALL BEARING

BEARING NO. (3) - BALL REARING

SOLUTION LEVEL = 1

1

** TECHNOLOGY DIVISION S K F INDUSTRIES INC.

UNIESS OTHERATISE STATED. LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREFS CENTIGRADE. FORCES IN NEWTONS. ALIGHES AND SLOPES IN DEGREES. SURFACE AND SLOPES IN DEGREES. SURFACE ROUGHNESS IN MICAGONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERALL COMBUCTIVITY IN WATTS PER METER-BEGREE CENTIGRADE.
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OUTER RING SPEED	• • •	.013000 .013000 .013000	
INNER RING SPEED	35963. 35963. 35963.	RAIL-LAND CLEARANCE -150 -150	LIFE FACTOR 5.000 5.000
CONTACT	.000 25.000 25.000	RAIL-LAMD DIAMETER 24.0000 24.0000	TYPE LIFE
DIAMETRAL CLEARANCE	. 0025 - 006 - 006	RAIL-LAND VIDTH 2.0000 2.0000 2.0000	OUTER RING M-50 STEEL M-50 STEEL M-50 STEEL
PITCH DIAMETER	32.100 58.500 58.500	CAGE POCKET CLEARANCE .150000 .150000	F & C T O R 5.000 5.000 5.000
AZIMUTH - 4GLE - AGLE		2	LIFE FACTOR 5.000 5.000
TUMBER OF ROLLING	10	CAGE TYPE NER PING LAND NER RING LAND	DATA INHERRING TYPE M-50 STEEL A-50 STEEL M-50 STEEL
HEARING NUMBER	<b>≈</b> € •	GASEDATA BEARINS WINDER  1 1 1 1 1 1 1	S T E E L D MRG-NO. 1 2 3

INNER RACEVAY CURVATURE

OUTER RACFUAY CJRVATURE

AALL DIAMETER

9.5250

BEARING NUMBER (3)

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TTPE - RALL HEARING

TECHNOLOGY DIVISION S K F INDUSTRIES INC.

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ROLLER DIAMETER 7.0000	TOULER Length 7.0000	ROLLEQ PROFILE Flat Length A.5000	ROLLER Crown Radius 670.000	ROLLER END Radius Inddo.ogdd	ROLLER Included Angle	RUMMER OF AKIAL Laminae 11
EFFECTIVE LF4GTM 6.5000	OUTER RACEMAY CROWN RADIUS • 0000	AXTAL FLANGE PLAY ANGLE	FFECTIV Length 00 6-100	INNER RACEVAY E CROUN RADIUS 0	AXIAL PLAY 0005	FLANGE Angle ,9000
SEASTYG NI) 49ER (2)	d.	L BEARING				
MALL DIAMETER	ER OUTER RACEUAY	CEWAY CURVATURE	INMER RACEUAY CURVATURE	RVATURE		
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TY SLOPE RO'L. ELM.	2.000	TY SLOPE • ROLL• ELM• END	2.000		M. THERMAL IT CONDUCTIVITY							
RMS ASPERITY INNER	2.000 2.000 2.000	RMS ASPERITY INNER FL.	2.000		THERMAL EXPAN.	+0-01. +0-01. •		ASPERITY FRICTION COEFFICIENT	000	FLG-R.E. FRICTION COEFFICIENT	605	
OUTER	2.000	OUTER FL.	2.000		DEHSITY AT			ASP FRI COEFF		FL6 7300		
ROLL. EIM.	6000	ROLL. ELM. END	80,		KINEMATIC VISCOSITY 7-78 C) (98-89 C)	5.61 5.61 5.61	<b>4</b> F <b>4</b> G	FILM REPLENISHMENT LAYER THICKNESS LL.ELM. + RACEUAY ) OUTER	.15uJ-03 .1500-03 .1500-63	FILM REPLENISHMEN" LAYER THICKNESS	.1500-03 .0000 .0000	E USED
CLA ROUGHNESS Inner	•13 •10	CLA ROUBHNESS INNER FL.	•13		KINEMATI	33,68 33.60 33.60	1 6 7 1 0	FILM REP LAYER T '(ROLL'ELM' OUTER	.3000-03 .3000-03 .3866-03		. 3000. 3000.	IONAL UNITS AR
OUTER	F 0 0 R	OUTER FL.	.13	ART DATA	DLSIGNATION	SANTOTRAC SANTOTRAC SANTOTRAC	# T O # A N O E F A	PEP-ENT LUBE IN CAVITY	11.00	CAGE FRICTION COEFFICIENT	\$ \$ \$ \$ \$	WISE STATED. INTERNATIONAL UNITS ARE
BEARING NUMBER	<b>~ 0.10</b>	BEARING Number			BEARING NUMBER	N P	Ų	NUMBER NUMBER	C) P	BFAGING Munder	N F	UNLESS OTHERWISE

* SHARERTH / BRL ** TECHNOLOG* DIVISION SKF INDUSTRIES INC.

4858 ALTERNATE INPUT PINION (ESIGN 28, 601 LOAD

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4	AL TERM	ATE IND	NASA ALTERNATE INPUT PINION DE	ON DEST	SIGN 24. 60( LOAD	046 1 940								
i.	T GE JME	TRY, PF	ARING L	OCATION	SHAFT GEOMETRY. PFARING LOCATIONS AND SHAFT LOAD.	FT LOAD.		PLANE X - Y.						
1	3639	TR:C SE	11 GEOMETRIC SECTIONS	1 10	SECTION	S), 3 B	EARINGS.	HODOLUS	AD SECTION(S). 3 BEARINGS. MODULUS OF ELASTICITY = 2.041+05	10117	= 2.04	1+05		
	P081-	INYER Left	SIGHT	OUTE	OUTER DIAM. Left Right	POINT FORCE	POINT	LOAD I LEFT	LOAD INTENSITY Left Right		POS.ERR	BEARING SEAT DEFLIFOR AN	٠.	ANG.ERR DEFL/MOM
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KASA ALTERNATE IMPUT PINION DESIGN 2A+ 601 LOAD
SHAFT GEOMETRY. BEARING LOCATIONS AND SHAFT LOAD. PLANE X - Z.
11 SECTERIC SECTIONS 1 LOAD SECTION(S). 3 BEARINGS. MODULUS OF ELASTICITY = 2.041+05
TOPEN C - CONTRACT

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LOAD INTENSITY LEFT RIGHT												
LOAD 1												
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OUTER :	•	17.0	17.0	31.0	42.3	47.0	25.0	25.0	25.0	25.0	25.0	
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MASA ALTERNATE INDUT PINION DESIGN  BEARING SYSTEN OUT		

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9 R L •••					JCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND IMMER RACFJAYS RESPECTIV	COMBUCTIVITY (W/DES.C)	1 - 6 & 1 - 4 0 1 - 4 3
R T H / 9			TORCIME	51.1	ID IFMER	COMBIJETIN	2.98 1.90 1.96
A H S H A H E			TOTAL	193. 2.801+03 3.189+03	4E OUTER AN	IST. (M4)	.287 .203 .203
		(F1-E)	CAGE-LAND	3.60 2.39 2.98	JATA FOR TH	HENISCUS DIST. (MY)	 ស្រួស ស្ពាស ស្ពាស
INDUSTRIE		TION TOROUE	I. FLNGS. R.E.DRAG R.ECAGE	2.235-08 2.671+03 3.057+03	DUCTIVITY		.943 .914 .915
	METRIC UNITS	ANJ FRICI	R.E.DRAG	27.0 45.8 46.2	HEAT CON	THERMAL FACTOR	.943 .912 .913
A 24. COX LAND	5	TION RATE (UATTS) AND FRICTION TORDUE (N-PM)	I. FLNGS.	0.00.0	FACTORS AND	RVATION FACTOR	866.
16 CHWO ESTGW 24.	_		. I . RACE	35.9 51.1 53.1	REDUCTION	STARVATIO	.999 .172 .472
ALTCAMATE INPUT PINTON DESTON	YSTER	FYICTIONAL HEAT GENEGA	0. FLY6S. 1.	0.000	EHD FILM TWICKWESS, FILM REDU	CRONSI	.124
NATE INPUT P	<b>.</b>	FUICTIONA	0. PACE	126. 29.3 23.7	ILM THICK	FILM (MICAONS)	.149 .170 .153
ALTER	 α		H46.	6 F	EHD F.	493.	- ~ r

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*** SHABERTH/BRL ** TECH'OLOGY DIVISION SKF INDUSTRIES INC. ** SHABERTH/BRL ***

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

HASA ALTERMATE IMPUT PINION DESIGN 24. 601 LOAD

METRIC UNITS

SYSTEM

PRESSURE VISCOSITY COEFFICIENT (MM2/N)	.1033-01 .1033-01 .1933-01		PRESSURE VISCOSIT/ COEFFICIENT (MR2/N)	.1033-01 .1033-01 .1033-01		PRESSURE VISCOSITY COEFFICIENT (MM2/M)	0 - 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1 - 0 1	13 10 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0			CALC/EPIC CAGE/SHAFT RATIO RATIO	1.00
PRECOEF			PRE			COEF					SPEED	1.406.04
DYNAMIC (CP.)	2.101	PERTIES	SITY DYNAMIC (CP)	2.101 2.101 2.101	PERTIES	SITY DYNAMIC (CP)	2.101	2.161			CALCULATED SP (RAD/SEC)	1.472+03 1.
VISCOSITY KINEMATIC (CS) DY	2.519 2.519 2.519	PHYSICAL PRO	VISCOSITY KINEMATIC (CS) DY	2.519 2.519 2.519	PHYSICAL PRO	VISCOSITY Kinepatic (CS) Dy	2.519	2.519		ED DATA	IC SPEED ) (RPM)	3 1.406+04
KINEHAT		JRES AND	KINEHA		URES AND	KINEFA				CAGE SPEED DATA	EPICYCL IC	1.472+03
DENSITY (GA/CA3)	. 8 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5	LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES	DENSITY (GH/CH3)	8. 8. 8. 8. 8. 8. 8. 8. 8. 8. 8. 8. 8. 8	LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES	DENSITY (BA/CH3)	. 8339 9339	.8339			ECCENTRICITY Ratio	. 460
-	000	LUBRI		<u> </u>	LUBRI		9.0	0.	UNITS	DA TA	SEP.FORCE (NEUTONS)	.116
TEMPERATURES (DEGREES C.)	150.000 150.000 150.000		TEMPERATURES (DEGREES C+)	150.000 150.000 150.000		TEMPERATURES (DEGREES C+)	150.000 150.000	150.000	A METRIC UNITS	- RING LAND DATA	HEAT RATE SI	3.60
LOCATION	OUTER INNER BULK		LOCATION	OUTER Inner Bulk		LOCATION	OUTER INNER	BULK	E DAT	CAGE RAIL	TORQUE	1.57
<b>J</b>	145.		-	6RG. 2			88G. 3		ა ▼ ე		88 6.	

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1.485.04
1.555.03
1.465.04
1.555+03
1.000-01
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2.99

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S K F INDUSTRIES INC. ** TECHNOLOGY DIVISION NASA ALTEGNATE INDUT PINION DESIGN 2A+ 601 LOAD

FOR BTARING NUMBER 1 METRIC UNITS

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421M174	<b>Z</b>	ANGULAR SPEEDS (RADIANS/SECOND)	(RADIANS)	SECOND)	SP	SPEED VECTOR ANGLES (DEGREES)	(DEGREES)
WGLC (DES.)	ž	<b>&gt;</b> 3	24	TOTAL	08811.00	TAN-1 (UY /UX)	TAN-1 (UZ/UX)
0:-	-R724.357	.000	.000	8224.357	1472,391	180.00	180.00
36.00	-A224.157	•000	.000	8224.357	1472.391	180.00	180.00
72.00	-R224.357	•280	.000	9224 - 357	1472,391	180.00	180.00
109.00	-8224.357	1.308	0000	H224.357	1472.331	179.99	180.00
144.00	-8224.357	1.825	000	8224.357	1472.391	179.99	190.00
190.00	-8224 . 357	000	0000	8224 . 357	1472.391	190.00	180.00
215.00	-8224.357	000.	000.	8224.357	1472.391	130,00	180.00
252.00	-8224.357	.000	000	A224 . 357	1472.391	180.00	180.00
298.00	-8224 - 357	000	000	9224.357	1472.391	180.00	180.00
324.00	-8004 . KA	000	000	8224.157	1472, 191	20,081	00.00

.. TECHNOLOGY DIVISION S R F INDUSTRIES INC. NASA ALTERNATE IMPUT PINION DESIGN 2A+ 601 LOAD

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.S (DEG	INNER	• •				.1587.06	.1897-86
T AMGLE	_					*	9
CONTACT ANGLES (DEG.)	OUTER	• •		9	0 9 9	.1634+06	.1959+86
GASP/GTOT	INNER	0000	.4577	0000	00000	.1653-86	.1983+06
42 STRESS (N/MM++2) LOAD RATTO GASP/GTOT	OUTER			.6796	.6797 .6797 .6797	.1671.96	.2007+06
(N/4M++2)	INNER	000.			0 = 0		.2031.06
HZ STRESS	OUTER	355.733	555.746 1262.159 1062.740	355-870	355.733 355.733 355.733	.1708+06	.2054+06
OWS	INNER	000.	343.070	000.	0000		
FORCES (NEWTONS)	OUTER	73-186	916.232	73.237	73-186 73-186 73-186	Y - RO	-40
WORMAL FO	CAGE	100.	• • • • • • • • • • • • • • • • • • •	000	• 1000		
AZ1MJTH	INGLE : OEB.)	36.00	000	150.00	252.06 _30.00 324.00	CAOSS THE (-1734-06	ACROSS THE [ HNER .2887+86
A21	ANGLE	36	101	190	252 -38 -38	HERIZ STRESS ACAOSS THE GUTER .1271+06 .1734+06 .1823+86 .0000	AC. .1449+06 .1156+06

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S K F INDUSTRIES INC. *** SHARERTH / 9RL ** TECHNOLOGY DIVISION NASA ALTERNATE INPUT PINION DESIGN 2A: 601 LOAD

FOR BEARING NUMBER 2 METRIC UNITS

	⋖	ANGULAR SPEEDS (RADIANS/SECOND)	(RADIANS)	SECOND	d.	SPEED VECTOR ANGLES (DEGREES)	(DEGREES)
¥		5	715	TOTAL	ORBITAL	TAN-1 (UY /UX)	TAN-1 (UZ/5X)
-1246.	156	1767.463	-26.334	7459.42B	1449.919	166.29	-179.79
-7n91.	9 53	1990.596	-29.350	7366.056	1474.454	164.32	-179.76
-7050.	622	2113,999	-31.186	7360.790	1475-199	165.31	-179.75
-7095	371	2151.783	-31.974	7412.633	1495.932	163-12	-179.74
-72 11.	371	2013.274	-30.807	7570.204	1515.145	164.42	-179.76
-7674.	A70	1758.030	-27.626	7873.714	1571.409	167.10	-179.79
-9063	223	1456.297	-23.750	A193.713	1631.504	169.76	-179.A3
-A291	986	1214.793	-23.220	8370.597	1664.526	171.66	-179.96
-4246.	910	1103.651	-18.252	B320.352	1653,819	172.3A	-179.87
-1378.	245	1176.609	-18.963	8064.559	1603.715	171.61	-179.8K
-7545	,624	1434.716	-22.063	7720-142	1537,779	169.23	-179.83

4 I W .. TECHNOLOGY DIVISION S K F INDUSTRIES INC. MASA ALTERNATE INPUT PINION DESIGN 24, 601 LOAD

FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	NORMAL	FORCES INGUTORS)	OWS	HZ STRESS	(N/MM··2)	(W/MM++2) LOAD RATIO	Q4SP/C101	CONTACT	ANGLES (DEG.)
ANGLE LUEG.)	CABE	OUTER	INNER	OUTER	INNER	OUTER	SHHER	OUTER	N CHICK
90.	-134.272	384.060	216-104	1274.538	1309-186	.1520	.2261	17.07	28.14
32.73	-58.074	516.752	377.603	1445.724	1579.636	.1559	.2193	19.51	27.04
62.45	25.837	583.553	447.139	1505-513	1668.251	.1561	.2168	20.76	27.56
98.18	105.734	496.639	360.857	1426.714	15554193	1544	-2182	20.99	29.53
130.91	160.685	34 . 176	208-651	1262.558	1293.960	.144	.223/	19.39	33.23
163.64	172.827	240.520	112.115	1132.855	1051.965	.1349	.2333	16.07	37.85
196.35	123.963	208.047	69.688	1067.526	(17.773	.1209	.2447	12.76	41.35
229.19	26.279	191,451	51.333	1838.349	610.798	.1138	.2542	10.40	42 0.5
261.82	-83.114	186.273	47.339	1032.573	749.197	.1154	.2565	9.50	41/00
274.55	.160.605	199.703	59.234	1053.058	630.42A	.1255	.2491	10.46	37.73
327.27	-117.269	241.129	101.988	1121.350	1019.286	.1402	.2379	13-35	33.07

NASA ALTEMBATE PAPUT PINIGH DESIGN 24, EDI LOAD

*** > H & & F & T H / A H L ** TECHURLOGY DIVISION S K F INDUSTRIES INC.

FOR BEARING NUMBER 3 METRIC UNITS

(DEBATES)	119 . 17 . 17 . 17 . 17 . 17 . 17 . 17 .	
SPEED VECTOR ANGLES (DEBATES)	TAN-1(MY/WX) 164.58 165.81 165.81 167.79 167.20 171.89 171.82	
SPE	0RBITAL 1467-239 1467-729 1478-602 1511-555 1677-611 16.5-193 1676-262 1615-710	
SECONDI	TOTAL 7332-212 7324-035 7370-358 7554-076 7906-333 8273-838 8477-310 8422-837 7714-135	
CRADIANS/	28.603 -29.982 -30.820 -29.956 -27.375 -24.512 -24.512 -24.512 -24.513 -25.637	
ANGULAR SPEEDS (RADIANS/SECOND)	, y 1949, 471 2049, 471 205, 4, 387 1755, 243 1745, 243 1887, 718 1184, 552 1259, 783 1472, 063	
X	-7064.205 -7054.496 -7079.496 -7779.496 -7715.513 -713.999 -4139.999 -4027.413 -7027.737	
А21МПТН	MHGLE (DEG.)  20.45  52.45  43.18  150.91  150.91  150.91  22.4.00  22.4.50	

*** SHABERTH / BRL ** TECHNOLOGY DIVISION SKF INDUSTRIES INC. ** SHABERTH / BRL ***

FOR BEARING NUMBER S NETRIC UNITS

421mUTH	HORMAL (	FORCES (MENTONS	ONS	HZ STRESS	(B/HH-+2)	LOAU RATIO	QASP/0101	CONTACT	CONTACT ANGLES (D'G.)	
HELE (DEG.)	CAGE	OUTER	INNER	AUTCA	INNER	OUTER	HHER	OUTER	IMPER	
:	-66.677	579.519	441.679	1502.062	1661.370	.1566	.2182	19.19	25.55	
52.73	28.857	647.219	509.923	1558.386	1742.938	.1567	.2162	20-14	25.72	
65.43	114.450	526 . A4.7	390 . 357	1455-077	1594.415	1551	.2160	20-41	24.08	
94.18	185-228	342.810	206.920	1260.885	1230.163	.1482	.22.2	18.43	32.52	
150.91	206.24?	243-229	106.779	1124.596	1035-005	.132A	.2348	15.74	58.31	
163.64	1100421	205-824	61.996	1665-710	890.445	.1175	.2453	12.91	42.55	
196.36	30.978	191.369	52.058	1038.201	814.595	.1067	.2540	10.92	44.12	
2 7.09	-95.992	1119.002	48.A21	1033.904	197.356	.1135	.255A	10.12	42.84	
261.02	-106:103	200.500	60.203	11554.458	836.466	.1230	. 2453	10.94	38.98	
254.55	-204.019	244.724	105.714	1126.895	1,31,552	. 1460	. 2369	13.70	33.25	
327.27	-134 - 181	391.228	242.973	1306.320	1361.339	.1530	.2248	17.26	27.74	

#### NASA

- 1 CYLINDRICAL
- 2 BALL BEARINGS

LEVEL 1

MASA ALTERNATE INPUT PINION DESIGN

INPUSTRIFS INC.

** TECHNOLOGY DIVISION

THIS DATA SET CONTAINS 3 BEARINGS

CYLINDRICAL RULLER BEARING . FLANGE 15 TYPE NO.

REARING NO. (2) - BALL BEAPING

BEARINS NO. (1)

HEARING NO. (3) - BALL REARING

SOLUTION LEVEL = 1

.. TECHNOLOGY DIVISION S K F INDUSTRICS INC.

	E SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREFS CENTIGRADE, FORCES IN NEWTONS, DOULT IN MENTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE OF MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES EGREE CENTIGRADE.
MASS ALLEMANT INVITED HOUSE CAN SEE CONS	UNLESS OTHERAISE STATED. LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREFS CENTIGRADE. FORCES IN NEWTONS. AET-MTS IV KILDGRAMS. PHESSURES AND ELASTIC MODULI IN VEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES. SURFACE ROUGHVESS IN MICAONS, SPEEDS IN REVOLUTIONS PER MINUTE. DENSITY IN 'GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL COMDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

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CEARING MUMBER (5)

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						FION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE DUTER AND IPMER RACEWAYS RESPECTIVELY	COMPUCTIVITY (W/DEG.C)	.473	.413	• • • 5
		TCGFIJE	22.4	747	H72.	ID TPMER RA	COMPUCTIVE	. H5 3	.137	• 755
		TOTAL	197.	2-888+03	3.282+03	1E OUTER AN	IST. (M4)	.287	.203	.203
	(E#-N)	CAGE-LAND	3.57	2.93	2.98	SATA FOR TH	(M) *ISIC SNOSINGM	.558	.314	.315
	FPICTIONAL MEAT GENERATION RATE (JATTS) AND FRICTION TORGUE (N-MM)	R.ECAGE	2.235-0A	2.671+03	3.057+03	OUCTIVITY D		.943	•16•	.915
4FTRIC UNITS	AND FRICT	I. FLWSS. R.E.DPAG R.ECAGE	27.0	45.A	46.2	HEAT CON	THERMAL FACTOR	.943	.912	.913
	TE (JATTS)	I. FLNSS.	000-3	000.0	0.100	FACTORS AND	ATTON FACTOR	. 3 38	870.	• 348
-	EZATIOV RA	I. HACE	1.15	136.	142.	REDIJCTION (	STARVATIO	6£ 6.	335	. 132
æ . L	L MEAT GEN	2. FL'163. I. RA	0.900	00000	0.000	EHÜ FILM THICKYFSS, FILM RED'JC'	CRONSI	-135	*1°	.254
S > S	FPICTIONA	O. WACE	129.	52.2	33.3	ILM THICKY	FILM (MICADUS)	.751	1,70	•274
 œ		e. C.	-	c.	₩	E+10 F	· tat		C!	•

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	PRESSURE VISCOSITY COEFFICIENT (MN2/N)	*1033-01 *1033-01 *1033-01		PRESSURE VISCOSI) ( COEFFICIENT (MN2/N)	.1053-01 .1053-01 .1053-01		PRESSURE VISCOSITY COEFFICIENT (MR2/N)	10-6601. 10-6601. 10-661.			CALC /EPIC CAGE /Spv4P-	.591
ERTIES	ITY DYNAMIC (CP.)	2.101 2.101 2.101	FRTIES	SITY DYNAMIC (CP)	2.101 2.101 2.101	PERTIES	SITY DYNAMIC (CP)	2.101 2.101 2.101			CALCULATED SPEED (RAD/SEC) (RPF)	1.472+05 1.106+04
LUBAICANT TEMPERATURES AND PHYSICAL PROPERTIES	VISCOSITY KINEMATIC (CS) DY	2.519 2.519 2.519	LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES	VISCOSITY KINEHATIC (CS) DY	2.519 2.519 2.519	LUBRICANT TEMPERATURES AMD PHYSICAL PROPERTIES	VISCOSITY KINEMATIC (CS) 07	2.519 2.519 2.519		CAGE SFEED DATA	EPICYCLIC SPEED (RAD/SEC) (RPH)	1.472-03 1.486-04
AICANT TEMPERATUR	DENSITY (64/CMS)	& & & & & & & & & & & & & & & & & & &	IRICANT TEMPERATUR	OEMSITY (GH/CH3)	\$550 E.	SRICANT TEMPERATU	OEWSITY (BH/CH3)	6886. 6886.		J	ECCENTRICITY RATIO (	9
90.1	TEMPERATURES (Degrees C.)	156.000 150.003 156.003	7	TEMPERATIJAES (DEGREES C.)	139.000	ħ1	TEMPERATURES (DEGREES C+)	150.000 150.000	A METRIC UNITS	- RING LAND DATA	MEAT RATE SCP.FORCE (WATTS) (NEUTONS)	3.60
	LOCATION	1 OUTER INVER BULK		LOCATION	2 OUTER INNER BULK		LOCATION	3 OUTER INNER BULK	A G C 0 A T	CAGE RAIL - R	TOROUE (AM-R)	1.37
		8. 0.			60 60			5 · · · · · · · · · · · · · · · · · · ·	U		. B. R. G.	-

*** SHABERTH / BRL ** TECHNOLOGY DIVISION SKF INDUSTRIES INC.

MASA ALTERNATE INPUT PINION DESIGN 2A+ 604 LOAD

METRIC UNITS

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*** SHAMERTH/BRL ** TECHNOLOGY DIVISION SKF INDUSTRIES INC.

NASA ALTERNATE INPUT PINION DESIGN 24. 601 LOAD

	(OEGHEES	TAN-10	180	180	180	1 30	130	180	190	180	180	190
C UNITS	SPEED VECTOR ANGLES (DEGMEES	TAN-1 (UY/UX)	180.00	186.60	130.00	179.99	179.99	157.00	190.00	180.00	170.00	140.00
FOR BEARING WUNBER 1 METRIC UNITS	SPE	ORBITAL	1472.391	1472.391	1472.391	1472.391	1472.391	1472.391	1472.391	1472.391	1472.391	1472.391
REARING NUN	SECONDI	TOTAL	8224.357	8224.357	A224 - 357	A224.357	8224.357	8224.357	8224.357	R224 - 357	9224.357	8224.357
	RADIANS/	20	000	000.	.000	000	.000	.000	.000	000	000	000
0 U T P U T	NGULAR SPEEDS (RADIANS/SECONP)	44	•000	000.	.2A0	1.508	1.825	000.	• • • •	000.	000.	000.
는 보 또 난 나	New	×	-8224.357	-4224.157	-8224.357	-8224.357	-8224-357	-8224.357	-A224.357	-A224.357	-8224.357	-8224.557
R O L 1 1 8 S F	421MUT4	NGLE (0E3.)	•			105.00						
0	~	486							- •	- •	. •	

WASA ALTERVATE INPUT PINION DESIGN 24, 601 LOAD

S K F INDUSTRIES INC.

TFCHNOLOGY DIVISION

FOR BEARING NUMBER 1 METRIC UNITS

7¢ D	AZ1MUTH	NORMAL	NORMAL FORCES (N	(NEWTONS)	HZ ST	RESS (	N/MM++2)	HZ STRESS (N/MM**2) LOAD RATIO GASP/GTOT	9ASP/010T	CONTACT ANGLES (DEG.)	ANGLES	(056.)
1588 2.68	A35LE (DEG.)	CAGE	OUTER	INNER	R OUTER	œ	INNER	OUTER	INNER	OUTER	Z	NHER
	• 00	.001	73.186		.000 355.	733	000	6000		Š		•
•7	36.90	.001	73.186			733	000			•		
~	72.00	.000	247.653			946	70.2		5000	9		00.
10	104.00	000	916.23	2 843-070	070 12001	1 2 0	1440.422		5000	00.		00.
<b>*</b> T	144.00	.000	691,326			7 6 6	77740447		0000	- ·		•0.
-1	00.001	000	73.2		155.870	07.0	000		0000	00.		00
12	216.06	- 000	73.186			7 4 4	-	0000	0000	00.		00.
25	252 • 00	000	73.18			7 6		0000	2000	00•		00•
e c	20.4.00		73.106			133	000.	0000	2000.	00.		•00
: C						133	000.	0000	.000°	00.		•00
20	004.00	100	73.186		000 355.733	733	000	0000•	0000	00.		00.
HERTZ STRESS ACHOSS THE OUTER RACEMAY-ROLLER PROFILE	ACROSS THE	OUTER RA	CEWAY-RCL	LER PROFIL	L.							
.1271+06	.1734+06	.174	.1744+00° .	.1726+06	.1708+06	91.	1630+06	1471404	70483	. 4.5		
*1025*06	0000	0000		• 0000	0000			.0000	90.00°	.0000		0000
<b>Y</b>	ACROSS THE THE RACEMAY-ROLLER PROFILE	HWER RAC	EYAY-ROLL	ER PROFIL	lu.							
•1443+06 •1156+06	•2049+05 •	.210	.2101+06	•2078+06	-	•20	.2031+06	.2007+06	.1983+06	.1959+06		90+168
				0000.	0000	.00	00	• 0000	0000	.0000	•	.0000

INDUSTRIES INC. ** TECHNOLOGY DIVISION S K F NASA ALTERNATE INPUT PINION DESIGN 24 601 LOAD

FOR BEARING NUMBER 2 METRIC UNITS

OUTPUT

HIGHIZV	=	ANGULAR SPEEDS (RADIANS/SECOND)	(RADIANS/	SECOND	SP	SPEED VECTOR ANGLES (DEGREES)	(DEGREES)
MSLE (DEG.)	××	>	42	TOTAL	ORBITAL	TAN-1 (UY /UX)	_
00.	-7246.951		-26.334	7459.428	1489.919	166.29	-179.79
.32 - 7.3	-7n91.933	1990.580	-29.350	7366.056	1474.454	164.32	
65.45	-7050.622		-51.186	7360.790	1475.199	165.31	
99.18	-7093.371	- •	-31.974	7412.633	1495-332	163.12	-179.74
150.31	-7231.371	2033.274	-39.807	7570.204	1515.145	164.42	-179.76
153.54	-7674.430	- •	-27.626	7875.714	1571.409	167.10	-179.79
98.51	-8063.223	•	-23.760	4193.713	1631.504	169.76	-179.83
55.3 • 0 3	-A291.956		-29.220	8370.597	1664.526	171.66	-179.86
151.82	-9246.810		-18.252	8320.352	1653.819	172.38	-179.87
234.55	-1378.242		-18.963	8064.559	1503.715	171.61	-179.8K
327.27	-75A5.624		170.66	7730 143	1517 779	10000	19.02.

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וברשמחרחתו חוא	IGH 2A+ 601 LOA
ונרשמחרחתו חוא	SIGH 24. 601 LOA
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	ITE INPUT PINION DESIGN 24. 601 LOA
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r C	ERNATE INPUT PINION DESIGN 24, 601 LOA
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	NASA ALTERNATE INPUT PINION DESIGN 24, 601 LOA

A214UTH	NORMAL	FORCES (MENTONS)	ONS	HZ STRESS	(N/MM++2) LOAD RATIO	LOAD RATIO	9ASP/6T0T	CONTACT ANGLES (DEG.)	ANGLES	(056.)
NSLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INVEP	OUTER	H	INNER
00.	-134.272		216.104	1274.538	1309.186	0000	0000	17.07		.7.
52.73	-58.076		379.603	1445.724	1579.636	. 1000	0000	19.51		*0
65.45	25. H37		447.139	1505.513	1668.251	.0000	.0000	20.76		.56
38.14	105.734		360.857	1425-714	1553.193	0000	•0000	20.99		.53
150.31	160.647		209.651	1262.558	1295.360	.0000	0000	19.39		.21
153.54	172.427		112.115	1132-455	1051.965	.000	.0000	16.07		96
195.36	123.953		63.588	1067.526	897.173	0000	0000	12.76		.25
22,1.09	26.279	191.451	51.333	1034.349	810.798	0000	.000	10.40		42.33
261.42	-43.104		47.339	1032.573	789.197	0000	0000	9.50		•0•
294.55	-160.605		59.234	1053.058	850.428	0000	• 0000	10.46		.73
127.27	-177.324		300 101	1121.150	1010.284		2000	11,15		.07

FOR BEARING NUMBER 3 METRIC UNITS

INDUSTRIES INC.

** TECHNOLOGY DIVISION

NASA ALTERNITE PUPUL PINTON DESIGN 24, 601 LOAD

ANGU	ANGULAR SPEEDS (RADIANS/SECOND)	(RADIANS/	SECONDI	SP	SPEED VECTOR ANGLES (DEGREES)	(DEGREES)
ÅR.		<b>7</b> P.	TOTAL	ORBITAL	TAN-1 (UY/UX)	TAN-1(NZ/UX)
1349.471	ü	8.603	1332.212	1467.239	164.58	-179.77
2042.769		29.982	7328.039	1467.729	163.81	-179.76
2044.387		30.820	1380.368	1478.602	163.59	-179.75
-1289.767 1782.059 -:	•	-29.956	7554.474	1511.355	164.79	-179.76
1735-243		27.375	7906.333	1577.582	167.32	-179.80
1487.719	Ÿ	14.512	4273.838	1647.611	169.64	-179.83
1290.623	Ċ	1.762	H477.310	1646.193	171.24	-179.85
1148.552	7	9.H99	8420.757	1674.252	171.89	-179.86
1239.783	Ç	0.031	8122.837	1615.710	171.22	-179.86
1472.068	ċı	2.637	7714.135	1537,735	169.00	-179.83
1776.567	``	26.378	7424.037	1483.087	166-14	179.79

S F F INDUSTRIES INC. ** IECHHOLOGY DIVISION 74. 50( LOAD TASA ALTERNATE INPUT PINION DESIGN

H10P17	MORMAL	FORCES (NEWLORS)	UNS)	HZ STRUSS	HE STRESS (W/MM+2) LOAD RATIO	OAN RATIO	QASP/3TOT	CONTACT	ANGLES (IFF.)	
(*930) 31:	CAGE	0'JTER	PHHFA	OUTER	INHER	OUTER	INNEH	OUTER	INNE	
00.	-65.611	519.549	441.629	1502.062	1661.370	0000	0000	19.13	25.55	
\$2.75	28.657	54 / • 219	509.923	1558.386	1/42.938	00000	0000	20.14	25.92	
67-45	118-450	526.441	370.557	1455.077	1594.415	0000	0000	20.41	28.08	
33.13	1 15.22R	542.H10	204.820	1260 - 885	1290 163	0000	0000	18.93	32.52	
.30.91	230.247	243.229	106.179	1124.596	1035.005	0000	0000	15.79	18.51	
5.3.54	144.011	205.824	67.996	1063.710	893.445	0000	0000	12.91	42.55	
15.55	40.97A	191.369	52.058	1038,201	814.595	0000	0000	10.92	44.12	
/2 3•0 3	- 15,792	1, 3.002	48.821	1055.904	737.350	0000	0000	10.12	42.44	
141.72	-186.193	299-510	50.505	1054,458	854.466	0000	0000	10.94	30.00	
194.55	-204-419	244.124	105.714	1126.895	1031.552	0000	0000	1.3.70	33.25	
15.12	-154-141	141.220	242.473	1306.520	1361.539	0000	0000	17.26	27.74	

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1 BALL BEARING

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S K F INDUSTRIES INC.

THIS DATA SET CONTAINS 1 PEARING PEARING 413 + (1) - BALL BEARING

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TECHNOLOGY DIVINION S K F

DEMPERATURES IN DEGREES CEMTIGRADE, FORCES IN NEWTONS, MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES
UPLIES STRIBLES STATED LITER DIREUSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, JEINES HAS LOPES IN DEGREES, SURFACE ROUDHURSS IN ALCORANGE AND SLOPES IN DEGREES, SURFACE ROUDHURSS IN ATCRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAAS PER CUBIC CENTIMETER, KINGARITC VISCOSITY IN CENTISTOKES AND INCAURINT IN SATES PER MILER-DEGREE CRATIGRADE.

REA911.	AUTORA DE ROLLTYA GLEMENTS D	AZIMUTH ANGLE OPTENTATION	PIICH OIAMETER	DIAMFTRAL Clearayce	CONTACT	INNER RING Speed	OUTER RING Speed
-	€0	000•	57.500	• 054	14.791	36936.	<b>a</b>
CASFOAT	<b>▼                                    </b>						
HFART7. ND9-3E-2	CASE TYPE		CASE POCKET CLEARANCE	RAIL-LAVD WIDTH	RAIL-LAND DIAMETER	RAIL-LAUD Clearance	VE I GHT
-	HALL RIDIAS		.476.500	1.066н	57.5000	.889000	00000++
STRELOATA	) A T A						
H4 0.147.	INNER RING TYPE	LIFE FA	FACTOR	OUTER RING TYPE		LIFE FACTOR	
-	N-50 STFEL	1.	1.000	M-50 STEEL		1.000	

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*** SHAPFRIH / 4 RL ** TECHNOLOGY DIVISION SKF INDUSTRIES INC.

OUTER RACEMAY CURVATURE ANLL MANETER

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INNER RACEUAY CURVATURE

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.12 .12 2.000		CLA ROUGHYFSS ROLL. FLM. OUTER 144ER 144ER						

UNLESS OTHERATISE STATED. INTERNATIONAL UNITS ARE USED

SIVEN TE MPRATURES

1.2ACE HILK DIL FLWG.1 FLWG.2 FLWG.3 FLWG.4 CAGE SMAFT 1.RINC FOLL.FL. 0.4ING H3G. 145.nn 143.nn 143.nn 143.nn 143.nn 143.nn 143.nn 149.nn 149.nn 149.nn 143.nn 149.nn 149.nn 7.8 ACE 147.90

INDUSTRIES INC.
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TECHNOLOGY DIVISION
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TURBINE COMPRESSOR BEARING

CONCENTRATED FORCE+ FY LOADING IN THE X - Y PLANE

111.2 REUTONS

CONCENTRATED MOMENT ABOUT 2

2500.0 NEVTON-MM.

LOADING IN THE X - 2 PLANE

CONCENTRATED FORCE+ FZ

.O NE JONS

CONCENTRATED HOMENT ABOUT Y

.0 NEUTON-MM.

1912.6 NEUTONS

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1 1-542-02 1-455-02 1-253-07 2-753-17 9-260-04 2-169+03 10325-5 12-5 3-508+0.  FATIGUE LIFE (MNU4S)  H/5154A  103-1-1FE FACTOR WATERIAL FACTOR  HATERIAL FACTOR  HATERIAL FACTOR  HATERIAL FACTOR  HATERIAL FACTOR  100-1-00  100-1-00  100-1-00	, a .	מע	10		7u	<b>A</b> 5	26	¥	FY	F2 .	¥	ŽĦ.	
FATIOUF LIFE (HOURS)  1. PACE 1. PACE 1. PACE 1. PACE 1. PACE 1. PACE 1. PACE 1. PACE 1. PACE 1.00  1.00	_	1.552-02	2 1.455	5-02 %	,253-07	2-753-11	A-260-04	2-169+03		-25.5	12.5	3.508+03	
7. 4ACE 1. 9ACE 3549149 0. 9ACE 1. 9ACE 0. 8ACE 7. 8ACE 425. 4250 4250 1.00		FATTGUE	LIFE CH	(2500)		WESIS/H		LURE-LIFE	FACTOR	4A TER JAL	FACTOR		
517. 267738 .663 .290 .230 1.00	.:					0. 4ACE	1. GACE	0. RACE	I. RACE				
	_	<b>4</b> 20.	517.		.195	. 7.94	.563	-290	•230	1.00	1.40		
	٠,	שמים שמשנים ו	I-PACE BIJLK DIL	ווני אווי	. FLMG.1	I-PACE BULK DIL FLWG.1 FLWG.2		FLWG.3 FLWG.4 CAGE			I-RING HOLL-EL. D.RING	. O.RING	HSG.

ORIGI	NAL PA	AGE IS JALITY			IVELY		
					M FACTORS AND HEAT CONDIJCTIVITY DATA FOR THE DUTER AND INNER RACEUAYS RESPECTIVELY	CONDUCTIVITY (V. DEG.C)	4.09
z :			TOROJE	337.	NO INNER	COMBINET	3.1.
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· S H A B R F H / 9 R L TUTBIUE COMPRESSOR SEANING		F91C71084	O. BACE	247.	ILM THICKN	FIL4 (41Cq)WS)	.166
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143.000	.3580	1.709	1.465	.1726-02
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AZ IMJTH		ANGULAR SPEEDS (RADIANS/SECOND)	S IRADIAMS	/SEC040)	J S	SPEED VECTOR ANGLES	inebrees)
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45.80	-9144.24		-3-388	9144,255		119.94	-179.98
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135.00	-9246.331		-114.227	9273.575		177.94	-179.29
199.80	-7219.89		-155.731	9231.091		177-21	-179.04
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	IEARING NUM	HZ STRESS	OUTER	1540.492	1582.300	1578.707	1576-629	1572.589	1564.448	1559.930	1568.039
		(5)	INNER	453.103	451-119	442.427	454.525	452.978	450.192	437-563	448.556
	1 9 0 1 9 9 1	FORCES (NEVIOUS)	OUTER	1245.761	1251-044	1242.541	1237.621	1228-152	1209.177	1199.739	1217.523
CARINA		HORMAL	3913	15.754	15.176	4.774	-7.664	-14.836	-13.915	-5.123	6.336
S CO4P475599 75ARIWI		A? I MUTH	4:13E (DEG*)	90.	45.00	10.00	155.00	130.00	223.00	270.00	315.00

IMDUSTRIES INC.

SKF

TECHNOLOGY DIVISION

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1 BALL BEARING

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INDUSTRIES INC.

TECHNOLOGY SIVISION SKF

TURBLUF COMPRESSOR MEARING

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INDUSTRIES INC.

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SKF INDUSTRIES INC. . SHAHFRIH/HRL ...

*** S H 4 % F R T H / 3 9 L ** TFCHHOLO3Y DIVISION

TURBINE COMPRESSOR REATING

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GLE (DEG.)	×	<b>*</b> *	21.	TOTAL	ORBITAL	TAN-1 (UV /-JX)	TAM-1 (WZ/WX)
00.	-47.33.402	150.356	-104.16R	A 705. 755	1635.544	179.01	-179.30
45.30	-4513.593	172.407	-122-255	H516.216	1685.069	178.84	-173.18
10.10	-4713.450	172.547	-122.941	8722.226	1739.970	178-87	-179.13
135.65	- 1.175 . 521	257.270	-151-745	90H0.524	1767.725	178.31	-179.04
1 30.01	-9152.152	204.652	-145.416	9116.390	1744.398	174.72	-179.8R
22,000	-3154.411	293.237	-159.556	9160.400	1764.488	174-18	-179.90
210.00	-1772.046	170.041	-133.631	2115.662	1716.705	178.87	-179.13
\$15.00	-8312.397	277.389	-220-197	4925.031	1695.516	178.22	-178-59

TURBLYE COMPRESSOR BEARING

INDUSTRIES INC.

** TECHNOLOGY DIVISION

FOR MEARING HUMBER 1 METRIC UNITS

3									
CONTACT ANGLES (DE6.)	INNER	31.65	30.93	34.21	39.21	40.42	39.66	36.41	33.02
CONTACT A	OUTER	11.46	13.98	14.82	13.95	14.51	14.03	12.60	11.41
4ASP/CTOT	INNTR	.070;	# D G D .	.0100	. 6000	.0000	.00.	-000	.090
OAD RATIO	OUTER	.0000	0000	0000.	0000.	.0000	0000	. 000	.000
STRESS (W/MM++2) LOAD RATIO	INNER	1490.880	1124.732	A72.546	1322.471	1244.938	1304.428	1325.555	134 5.752
HZ STRUSS	OUTCA	1559.755	1422.307	1465.640	1511.316	1536.762	1498.869	1550.123	1535-673
OHS)	INNER	459.855	257.935	111.041	386.837	35A - 240	371.250	389.593	443.192
HORMAL FORCES INCUTO	OUTER	1194.327	100.587	994.254	1070-115	1146.110	106.3.401	1176.265	1103.679
HORMAL F	CAGE	A.121	47.117	44.582	36-174	5.076	- 14.A55	- 54.039	-27.346
M10F15A	G AVSLF (UEG.)	ē.	15.10	00.07	135.00	149.00	225.60	278.00	u0.515

#### AT81D040

#### APPENDIX E

Calculation of Cage Pocket and Cage Land Forces

AT81D040

#### NOMENCLATURE

#### APPENDIX E

$$A = 16.9706\eta_{0}U_{y} \cdot \frac{R^{\frac{1}{2}}}{3+2k}$$

$$B = 16.9706\eta_{0}U_{x}R^{\frac{1}{2}}$$

$$C = (A^{2} + B^{2})^{\frac{1}{2}}$$

$$C_{g} = \text{cage land radial clearance (in.)}$$

$$C_{o} = \eta_{o}|U_{y}|(R_{x}R)^{\frac{1}{2}}K1$$

$$C_{p} = \text{cage pocket radial clearance (in.)}$$

$$k = \frac{R}{R_{x}}$$

$$K1 = \left\{\frac{1}{(3+2k)^{2}} + \frac{1}{k(3+\frac{2}{k})^{2}} \cdot \left(\frac{U_{x}}{U_{y}}\right)^{2}\right\}^{\frac{1}{2}}$$

$$\ell = \text{roller total length (in.)}$$

$$R = \text{ball or roller radius (in.)}$$

$$R_{g} = \text{cage land radius (in.)}$$

$$R_{g} = \left(\frac{1}{R} + \frac{1}{R} + C_{p}\right)^{-1} \quad \text{(in.)}$$

$$U = \left(U_{x}^{2} + U_{y}^{2}\right)^{\frac{1}{2}} = \text{entrainment velocity (in./sec.)}$$

AT81D040

# NOMENCLATURE continued..... APPENDIX E

U = entrainment velocity of the ring-land interface (in./sec.)

= 
$$R_{\rho}$$
 ( $\omega_{i} + \omega_{c}$ ) for inner ring riding cage

=  $R_{\frac{1}{2}}$  ( $\omega_{\circ}$  +  $\omega_{\text{C}}$ ) for outer ring riding cage

$$U_x = \frac{V_x}{2}$$
 (in./sec.)

$$U_y = \frac{V_y}{2}$$
 (in./sec.)

$$V = \left(V_{x}^{2} + V_{y}^{2}\right)^{\frac{1}{2}} = \text{sliding velocity (in./sec.)}$$

 $V_{\ell}$  = sliding velocity at the ring-land interface (in./sec.)

= 
$$R_{\ell}$$
 ( $\omega_i - \omega_c$ ) for inner ring riding cage

= 
$$R_{\ell}$$
 ( $\omega_{o}$  -  $\omega_{c}$ ) for outer ring riding cage

$$V_{x} = R\omega_{y}$$
 (in./sec.)

$$V_{v} = -R\omega_{x} \text{ (in./sec.)}$$

$$\gamma = \cos^{-1}\left(\frac{A}{C}\right) = -1\left(\frac{B}{C}\right)$$

= angle between the rolling speed vector and the pumping force vector

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#### AT81D040

# NOMENCLATURE continued..... APPENDIX

ω_c = cage orbital speed (rad./sec.)

ω_i = inner ring orbital speed (rad./sec.)

ω_o = outer ring orbital speed (rad./sec.)

ω_x = ball or roller rotational speed, x component (rad./sec.)

ω_y = ball rotational speed, y component (rad./sec.)

#### Calculation of Roller-to-Cage Pocket Forces

A numerically stable means for calculating cage pocket forces in cylindrical roller bearings was developed in [24]. The analysis is generalized here to include cage pocket and cage land forces in ball and tapered roller bearings.

Web geometry is taken as a radially outward cylindrical cavity for ball bearing simulations and a rectangular cavity for roller bearings, Figure E.l. The force exerted by a ball or roller is considered to act at the web midpoint, and all pocket contacts lie on the pitch circle. Interactions between roller ends and cage pocket sides are neglected. The load supported by the web is considered to have two components: one normal to the plane of contact (the surface of the web), and a frictional component in the plane of contact.

The normal force component is calculated as a function of rolling element to cage pocket offset  $Z_C = Z_C(\omega_1, \omega_2, \ldots, \omega_n)$ , where  $\omega_i$  is the orbital speed of the i-th rolling element. Pocket loads are computed as a function of the hydrodynamic lubricant film, h=h( $Z_C$ ) that fills the gap between the cage web and the rolling element surface.

Numerical stability is gained by taking a linear approximation to the force-displacement equation describing the roller-to-pocket normal load component

$$F_{\text{normal}} = KZ$$
 (E.1)

where K is chosen such that equation (E.1) will match the exact solution at  $F_{normal} = 67N$  (15 lb.) for roller bearings, and  $F_{normal} = 11N$  (2.5lb.) for ball bearings.

$$K = \frac{11}{(C_p - .33R\eta_o \ell U)} \approx \frac{11}{C_p}$$
 (ball bearings) (E.2)

$$K = \frac{67}{(C_p - .33Rn_o lU)} \approx \frac{67}{C_p}$$
 (roller bearings)

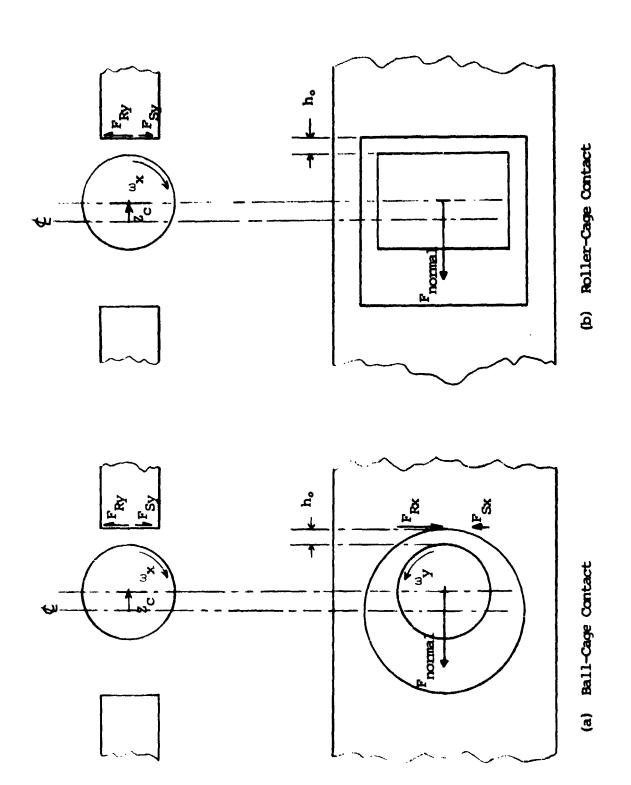


Figure E.1: Configuration of Contacts

#### AT81D040

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# Contact Inlet Region Hydrodynamic Friction Forces Ball/Cage Contacts

In a properly lubricated ball bearing, a constant viscosity fluid film separates the metal surfaces of the ball and cage web. Surface motion causes pressure to build up at the entry to the contact. Ball surface motion entrains the lubricant into a gently narrowing wedge, creating a distributed traction on the ball surface. Area integration of the surface traction yields a three dimensional load vector. Components of the load vector are given in terms of the dimensionless quantities  $\frac{1}{F_R}$  and  $\frac{1}{F_S}$  [25].

#### Forces:

#### Rolling Components

$$F_{Ry} = \frac{1}{2}C_o\overline{F}_R \cos\gamma$$
 (E.3)

$$F_{Rx} = \frac{1}{2} C_{e} \overline{F}_{R} (\sin \gamma) \sqrt{k}$$
 (E.4)

#### Sliding Components

$$\mathbf{F}_{SY} = \overline{\mathbf{F}}_{S} \eta_{\bullet} V_{X} (R_{X} R_{Y})^{\frac{L_{2}}{2}}$$
 (E.5)

$$F_{Sx} = \overline{F}_{S} \eta_{o} V_{Y} (R_{X} R_{Y})^{\frac{1}{2}}$$
 (E.6)

#### Moments:

$$M_{Fx} = -R \cdot (F_{Ry} + F_{Sy}) \qquad (E.7)$$

$$M_{Fy} = R \cdot (F_{Rx} + F_{Sx}) \qquad (E.8)$$

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Values for  $\overline{F}_{\rm R}$  are obtained as follows. From [25], the force directed normal to the plane of contact is

$$F_{normal} = C_o \overline{F}_R \frac{R}{R} \times \cos \gamma$$

$$\overline{F}_{R} = \frac{F_{\text{normal}}}{C_{\circ}R_{\checkmark}\cos\gamma}$$
 (E.9)

 $F_{\text{normal}}$  is known (E.1), and other terms are functions of geometry. Therefore,  $\overline{F}_R$  is known. Functions shown in Figures E.2 and E.3 were curve fit* to give

$$\overline{F}_{S} = .26 \ \overline{F}_{R} + 10.90$$
 (E.10)

#### Roller/Cage Contacts

The analytic description of lubricated, rolles-to-cage pocket contact, Figure E.l (b), is based on the lubrication of a rigid cylinder near a plane [26]. The radially directed friction force is expressed as a function of the normal force

$$F_{fric} = \mu_h F_{norm}$$
 (E.13)

where  $\textbf{F}_{\text{norm}}$  is computed from (E.1).  $\mu_{\boldsymbol{h}}$  is a hydrodynamic friction coefficient

$$\overline{F}_{R} = 34.74 \ (\ln \rho_1) - 27.60 \ (E.11)$$

$$\overline{F}_{S} = 8.82 (\ln \rho_1) + 3.89$$
 (E.12)

for hydrodynamic contact. We get (E.10) by solving E.11) for  $\forall n \geqslant 1$ , and inserting this expression into (E.12).

^{*}  $\overline{F}_R$  is shown vs. dimensionless meniscus distance ( $\rho_1$ ) in Figure E.2:  $\overline{F}_S$  is shown vs.  $\rho_1$  in Fig. E.3. The functions are closely approximated by:

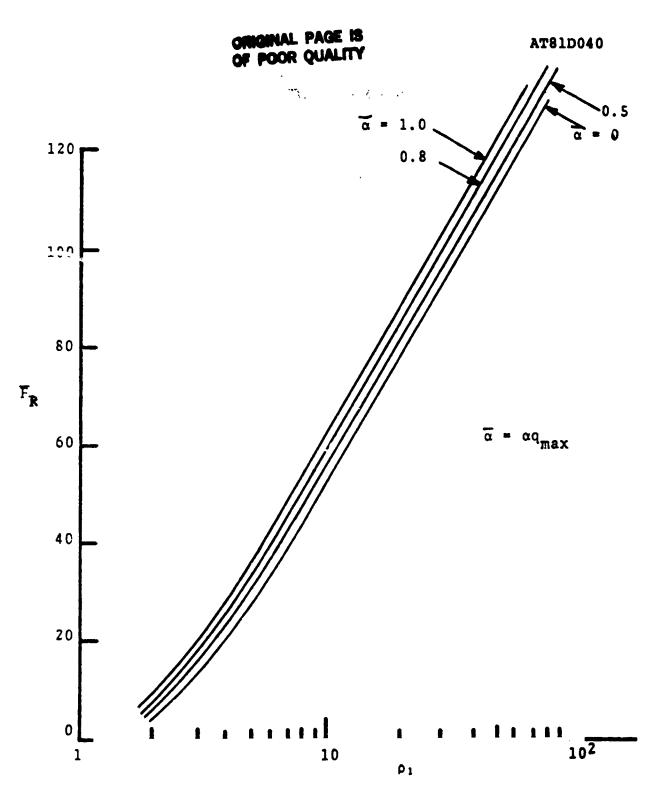


Figure E.2 Variation of  $\overline{F}_R$  with the Dimensionless Meniscus Distance  $\rho_1$  (taken from reference [25])

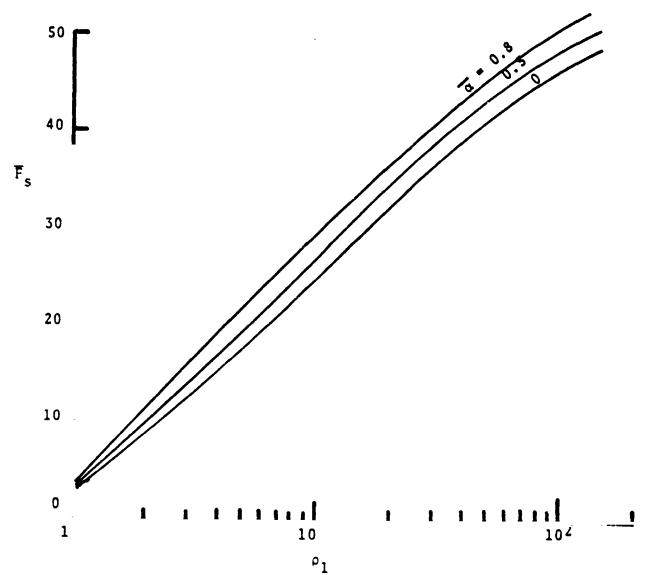


Figure E.3 Variation of  $\overline{F}_s$  With  $\rho_1$  (taken from reference [25])

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$$\mu_{h} = \frac{1}{2\sqrt{\left|\frac{F_{norm}}{2\eta_{o}U_{y}}\ell\right|}}$$
 (E.14)

The moment generated about the roller's x-axis is given by

$$M_{Fx} = R\mu_h |F_{norm}|$$
 (E.15)

#### Cage Land Normal Forces and Friction Moment

Forces which develop between a cage rail and its supporting ring surface are obtained using the hydrodynamic solution for self-acting, short journal bearings [27]. Forces are assumed to act in the plane of rotation.

The resultant of the pressure distribution on the cage can be described by orthogonal force components along and perpendicular to a line passing through the cage center and point of closest approach to the land.

Figure E.4 shows the relevant parameters for an inner ring land riding case, and Figure E.5 illustrates the outer ring land riding case.* The cage undergoes a radial displacement in the bearing XYZ frame, of magnitude e and direction  $\theta_{\rm C}'$ . An xyz frame is attached to the cage, such that the y axis passes through the point of minimum film thickness.

Assuming an isoviscous, Newtonian fluid, the lubricant forces which develop at the guide ring are given as a function of eccentricity [27]

$$F_{y} = \pm \eta_{o} U_{\ell} L^{3} \qquad \varepsilon^{2} \qquad (E.16)$$

$$\frac{C_{0}^{2}}{C_{0}^{2}} \cdot \frac{(1-\varepsilon^{2})^{2}}{(1-\varepsilon^{2})^{2}}$$

$$F_Z' = \frac{\pi_0 U_{\ell} L^3}{C_{\ell}^2} \frac{\pi \epsilon}{4(1-\epsilon^2)^{\frac{3}{2}}}$$
 (E.17)

^{*} These figures were repeated from section 2.4.1. for convenience.

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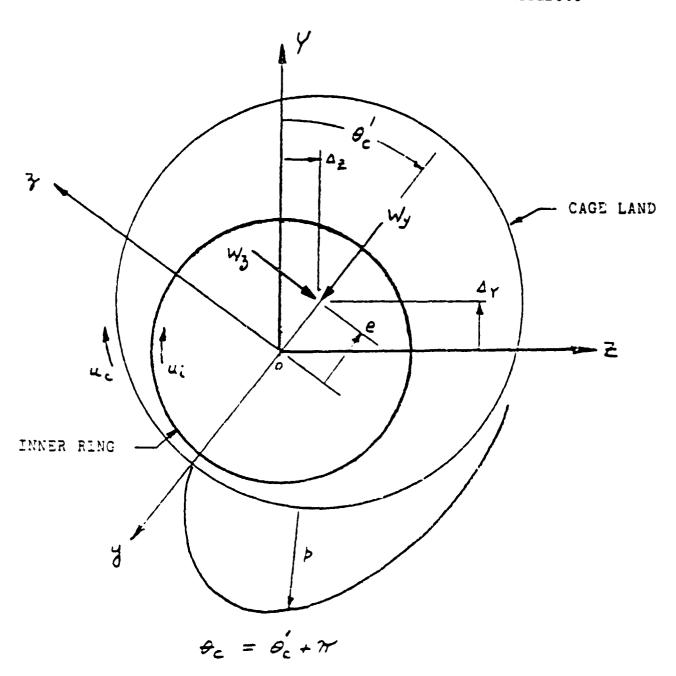


FIGURE E.4 INNER RING-CAGE LAND CONTACT GEOMETRY

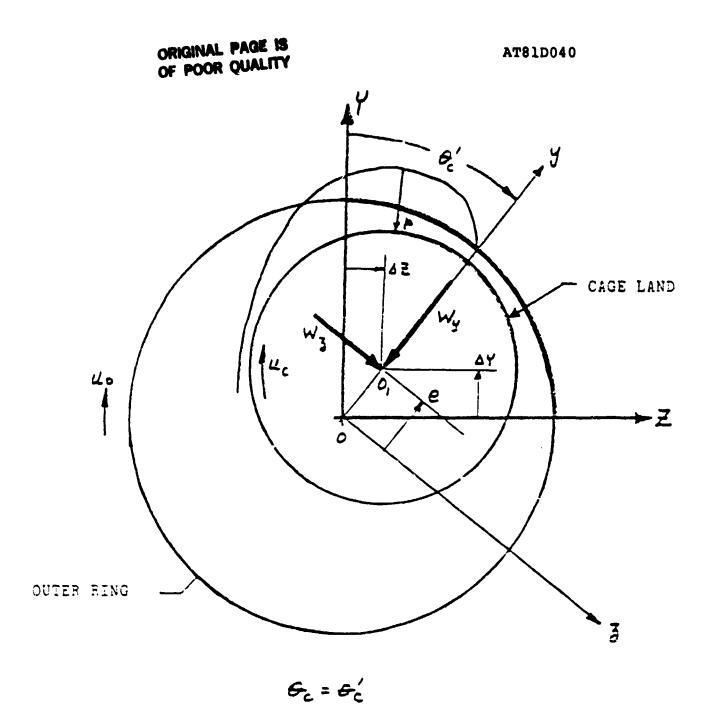


FIGURE E.5 OUTER RING-CAGE LAND CONTACT GEOMETRY

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The upper sign applies to an inner ring and the lower to an outer ring land riding cage. Eccentricity  $\epsilon$  is related to translation e,

$$\varepsilon = \underline{e} \qquad (E.18)$$

For level 1 solutions, the radial force  $F_y$  is set equal to the cage weight. The eccentricity  $\epsilon$  necessary to support that weight is used to determine  $F_z$  and the cage-land friction torque

$$M'_{C} = \pm \frac{\eta V_{\lambda} R_{\ell}^{2} L}{C_{\ell}} = \frac{2 \pi}{(1-\epsilon^{2})^{1/2}}$$
 (E.19)

The angle  $\theta_{_{\hbox{\scriptsize C}}}$  is used to transform forces from the local xyz frame to the cage reference frame:

$$\begin{cases}
 M_{cX} \\
 F_{cY} \\
 F_{cZ}
\end{cases} = 
\begin{bmatrix}
 1 & 0 & 0 \\
 0 & \cos\theta_{c} & -\sin\theta_{c} \\
 0 & \sin\theta_{c} & \cos\theta_{c}
\end{bmatrix} 
\begin{pmatrix}
 M'_{c} \\
 F'_{y} \\
 F'_{z}
\end{pmatrix}$$
(E.20)

# APPENDIX F

SKF AND NASA VERSIONS Or SHABERTH

#### F.1 INTRODUCTION

This appendix describes the SKF and NASA versions of SHABERTH. The primary differences between the two versions encompass the EHD film thickness and the concentrated contact traction force calculations. The relevant mathematical models are discussed in Section 7.2. The differences with respect to program input data are explained in Section F.3.

#### F.2 MATHEMATICAL MODELS

#### F.2.1 EHD Film Thickness

In calculating the elastohydrodynamic film thickness,

SHABERTH/SKF uses the Archard-Cowking equation (3) for point contact

and the Dowson-Higginson equation (4) for line contact. Two film

thickness reduction factors are then multiplicatively applied:

- a thermal factor due to heating in the contact inlet using the formulation of Cheng (11)
- 2) a factor accounting for starvation at the contact developed by Chiu (12).

SHABERTH/NASA uses the film thickness equation developed by Loewenthal et. al. (6). This equation is applicable for both point and line contacts.

### F.2.2 Concentrated Contact Traction

The concentrated contact traction model used in SHABERTH/

SKF accounts for lubricant shear and asperity interaction.

A semi-empirical model developed by Chiu, discussed in (25), is used to calculate an EHD lubricant shear coefficient.

Asperity effects are introduced by determining the portion of the contact load carried by the asperities, using the analysis of Tallian (5), and then calculating the resulting traction as the product of the normal load carried by the asperities times the asperity friction coefficient. In equation form the traction force is:

$$F = Q_{EHD} \mu_{EHD} + Q_{ASP} \cdot \mu_{ASP}$$
 (1)

$$Q = Q_{EHD} + Q_{ASP}$$
 (2)

where

F is the traction force

 $^{
m Q}_{
m EHD}$  is the normal load carried by the EHD film  $^{
m \mu}_{
m EHD}$  is the friction coefficient which develops from lubricant shear

 $^Q ASP$  is the normal load carried by the asperities  $^\mu ASP$  is the asperity friction coefficient Q is the total load

SHABERTH/NASA calculates concentrated contact traction across the EHD film only, using the model developed by Allen, et. al. (7). This model determines the traction force by first calcu-

lating the shear stress and then integrating the shear stress over the respective contact area. For a Newtonian fluid the shear stress is given by the equation

$$\tau = \eta \frac{v}{h} \tag{3}$$

where

 $\tau$  is the shear stress

n is the dynamic viscosity

v is the surface relative sliding velocity

h is the film thickness

The lubricant viscosity is assumed to be an exponential function of pressure of the form:

$$\eta = \eta_0 e^{\alpha \cdot s} \tag{4}$$

where

 $\boldsymbol{\eta}_{o}$  is the dynamic viscosity at atmospheric pressure

 $\boldsymbol{\alpha}$   $% \boldsymbol{\alpha}$  is the pressure viscosity coefficient

s is the normal stress

The Allen formulation requires that the shear stress not exceed a specified fraction of the normal stress such that

$$\tau = \eta \frac{v}{h} \quad \text{if } \eta \frac{v}{h} \le \tau_{c}$$

$$\tau = \overline{f}s \quad \text{if } r \quad \frac{v}{h} \Rightarrow \overline{f}s \quad \text{and } \eta \frac{v}{h} > \tau_{c}$$

where

- $\tau_{\rm c}$  is the critical shear stress for which a value of 0.0069 N/mm  2  (1000 psi) is normally used.
- $\overline{f}$  is called the lubricant friction coefficient and has been determined for specific lubricants. Typical values of  $\overline{f}$  lie in the range  $0.05 < \overline{f} < 0.08$ .

#### F.3 PROGRAM USE

The selection of the desired SHABERTH version has been made possible by the inclusion of two separate Map statements for the Univac 1100 computer:

- The original Map statement @MAP,S ALWAYS/MAP, ALWAYS/ABS enables execution of SHABERTH/SKF
- 2) A new Map statement @MAP,S NASA/MAP, NASA/ABS enables execution of SHABERTH/NASA

The only difference between the two versions with respect to input data is on card type B16. For the NASA version, two additional lubricant data items are specified for NCODE  $\leq$  0:

- 1) AKN Empirical lubricant constant columns 71-75
- 2) FRIC Lubricant Friction Coefficient columns 76-80

Default values of AKN=50.0 and FRIC=0.07 will be used if these spaces are left blank. For NCODE values of 1 to 4, the following values of AKN and FRIC are assigned.

NCODE	AKN	FRIC
1	18.2	0.075
2	18.2	0.045
3	24.9	0.070
4	18.2	0.070