RESEARCH REPORT - USER'S MANUAL

FOR

COMPUTER PROGRAM AT81Y003 SHABERTH

MAY 1981

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

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SKF REPORT NO. AT81D040

SUBMITTED TO:

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KING OF PRUSSIA, PA

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16. Abstract

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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FOREWORD

The SHABERTH computer program was originally developed by Kellstrom [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:

1. Modification of the cage module to calculate cage pocket and cage land forces in 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.

2. 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 Λ (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].

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.

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 shaft. 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 Mapping and Heat Dissipation Analyses - based on the methods of Harris [15] Fernlund [16] and Andreason [17].

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.

3. 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^o 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^oC.

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:

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

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:

$$\frac{\Delta t_i}{t_i} \leq \text{EP2 for all nodes } i \qquad (2.2)$$

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, $q_{oi} + q_{gi} = 0$ 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 $(q_{oi} + q_{gi}) = (EQ)i$.

The second convergence criterion is satisfied if:

$$\left[\sum_{i=1}^{n} \frac{(EQ)_i^2}{n} \right]^{\frac{1}{2}} \leq 100 \times \text{EP2} \qquad (2.3)$$

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.

$$\rho i \; Cp_i \; V_i \; \frac{dt_i}{dT_i} = q_i \qquad (2.4)$$

where ρ = 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

$$t_i(T_s) = t_{oi} \qquad i = 1, 2, \ldots, n \tag{2.5}$$

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

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

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]

$$\frac{dt_{i,k+1}}{dt_{i,k}} \geq 0 \qquad i = 1, 2, \ldots, n \tag{2.7}$$

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.

Differentiating equation (2.6) for node i, one obtains

$$\frac{dt_{i,k+1}}{dt_{i,k}} = 1 + \frac{\Delta T_i}{\rho_i C_{pi} V_i} \cdot \frac{dq_{i,k}}{dt_{i,k}} \qquad i = 1,2,\ldots n \qquad (2.8)$$

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

$$\frac{dq_{i,k}}{dt_{i,k}} = \frac{q_i(t_i + \Delta t_i) - q_i(t_i)}{\Delta t_i} \qquad (2.9)$$

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.

1. 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 node 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 $q_{ci,j}$ 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

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

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 $q_{vi,j}$ transferred between nodes i and j can be calculated from the equation, (2.11)

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

where α_v = 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

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

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 $q_{wi,j}$ transferred by forced convection can be obtained from the following equation.

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

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,

$$N_u = aR_e{}^b P_r{}^c \tag{2.13}$$

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

N_u = Nusselt number = $\alpha_w L/\lambda$
L = characteristic length
λ = conductivity of the fluid
R_e = Reynold's number = $UL\rho/\eta$
U = characteristic speed
ρ = density of the fluid
η = dynamic viscosity of the fluid
P_r = Prandtl number = $\eta C_p/\lambda$
C_p = 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_w for turbulent flow and heating of petroleum oils is given by

$$\alpha_w = k_9 \cdot \eta(t)^{k_{10}} \qquad (2.14)$$

where k_9 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,

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

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 5.76×10^{-8} watts/m^2-$(^oK)^4$ 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.

$$q_{Ri,j} = \frac{\varepsilon\sigma A_i\left[(t_i+273)^4 - (t_j+273)^4\right]}{1 + (1-\varepsilon)\ (A_i/A_e)} \qquad (2.16)$$

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.

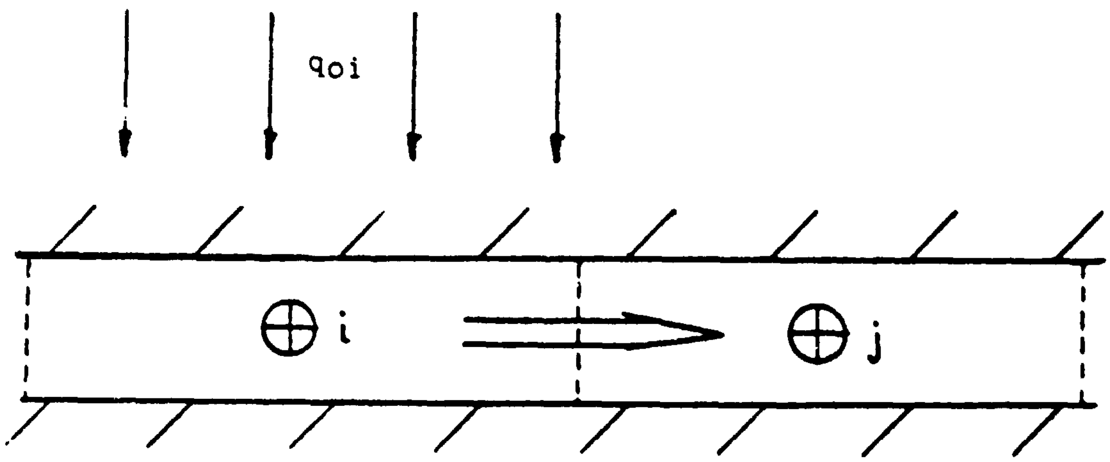


FIGURE 2.1 CONVECTIVE HEAT TRANSFER

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,

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

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.

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

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

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

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

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

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

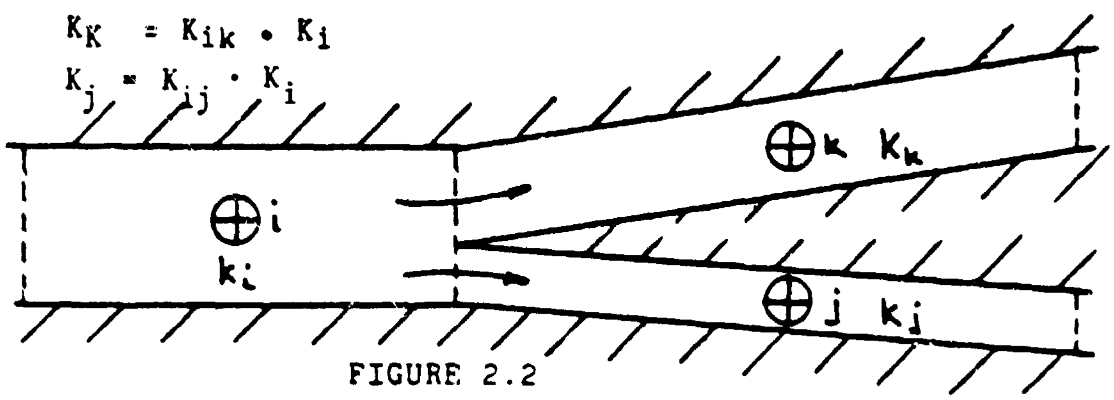


FIGURE 2.2
DIVIDED FLUID FLOW FROM NODE i

2.1.3.7 Total Heat Transferred

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

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

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_W and q_R. Therefore the equations to be solved for a steady state solution are non-linear. The sub-program 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 implicitly by

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

where

Δt is temperature difference
q is heat flow

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

$$\Omega_1 = \frac{h}{\lambda A} \qquad (2.23)$$

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,

$$\Omega_2 = \frac{1}{\lambda \ell_{re,i}} \left(\frac{\pi\psi}{2b_i V}\right)^{\frac{1}{2}} \qquad (2.24)$$

where 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 = $\lambda/(\rho . C_p)$

ρ = density

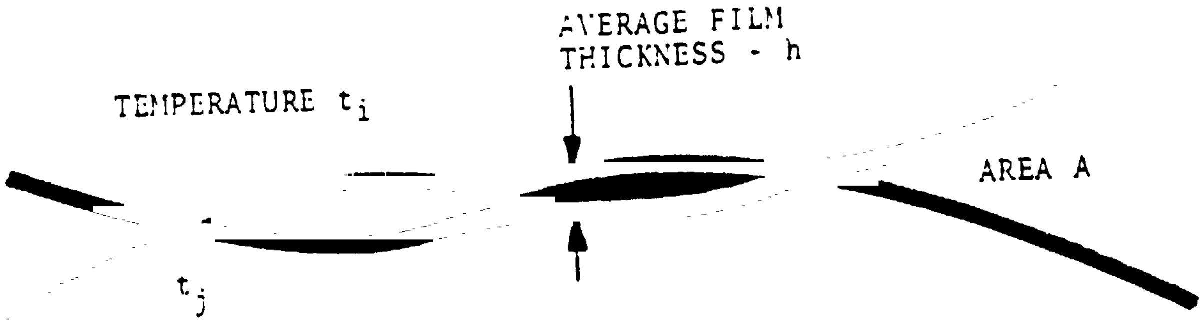
C_p = specific heat

b = half the contact width

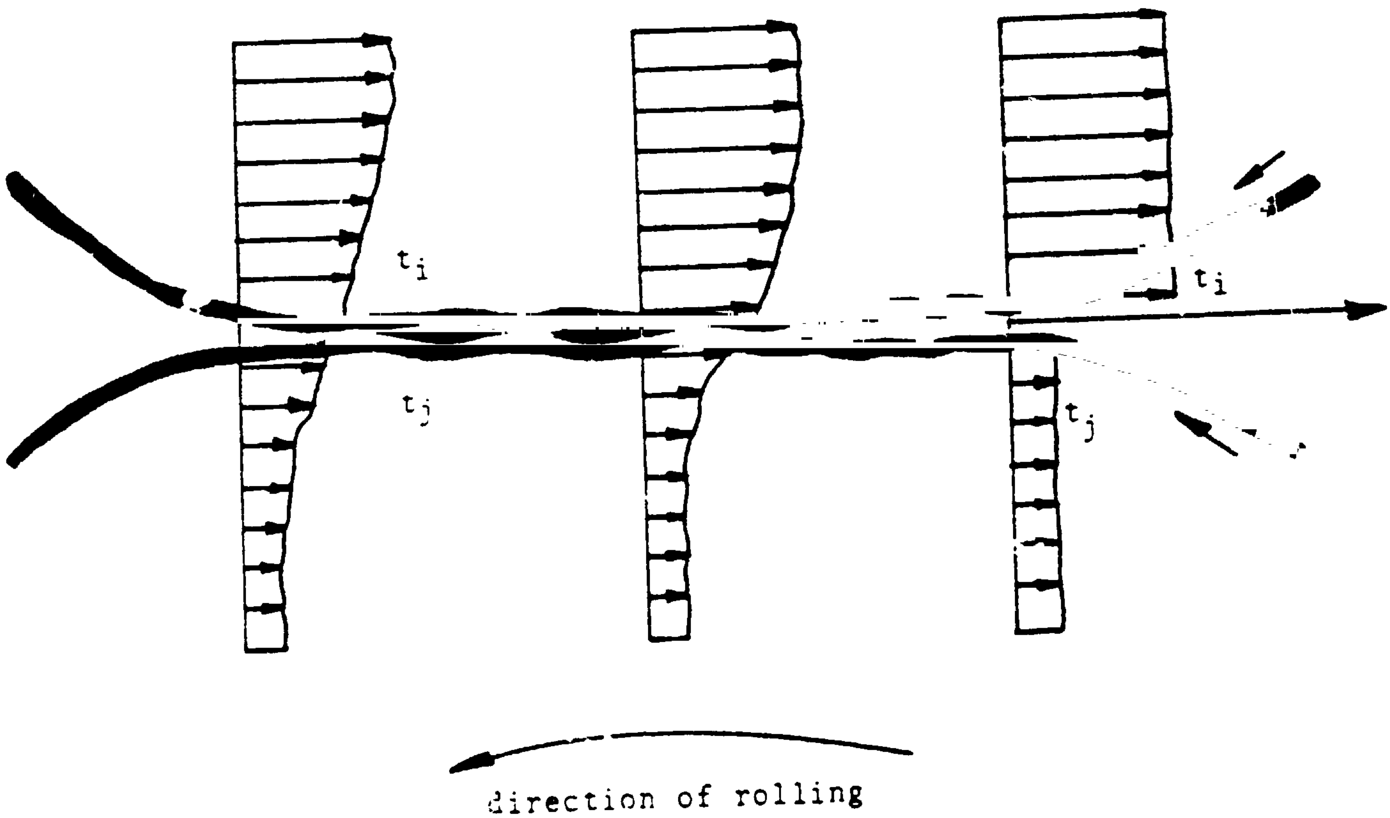
V = rolling speed

The resultant resistance is

$$\Omega_{res} = \Omega_1 + \Omega_2 \qquad (2.25)$$



(a) Schematic Concentrated Contact



(b) Temperature Distribution at Rolling, Concentrated Contact Surfaces

FIGURE 2.3

CONTACT GEOMETRY AND TEMPERATURES

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

$$\frac{1}{\Omega_{eqv}} = \sum_{i=1}^{n} \frac{1}{\Omega_{res,i}} \tag{2.26}$$

2.2 Bearing Dimensional Change Analysis

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

$$\Delta DCL = f\{(Fits)_m, t_i, \Omega_m, (Q_r)_m\}, \tag{2.27}$$

$m = 1,2$ for inner and outer rings respectively

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

where: ΔDCL is the change in bearing diametral clearance
Fits are the cold mounted shaft and housing fits.
t_i are the component temperatures
Ω_m refers to the ring rotational speeds
Q_r^m 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:

$$\frac{|(\Delta DCL)_N - (\Delta DCL)_{N-1}|}{D} \leq \text{EPSFIT for all bearings} \quad (2.28)$$

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:

$$(\vec{FM_b})_i - (\vec{FM_s})_i = 0 \text{ for all bearings, } i \quad (2.29)$$

where: $\vec{FM_b}$ denotes a vector of bearing loads and moments resulting from rolling element/race forces and moments.

$$\vec{FM}_{bi} = \begin{bmatrix} F_{bxi} \\ F_{byi} \\ F_{bzi} \\ \hline M_{byi} \\ M_{bzi} \end{bmatrix} \begin{matrix} \text{Forces} \\ \\ \text{Moments} \end{matrix} \tag{2.30}$$

If the bearing solution considers friction, $\vec{FM}_b$ 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_b is comprised only of rolling element/race normal contact forces.

$\vec{FM}_{si}$ denotes a similar vector of loads, exerted on the inner ring by the shaft.

$$\vec{FM}_{si} = \begin{bmatrix} F_{sxi} \\ F_{syi} \\ F_{szi} \\ \hline M_{syi} \\ M_{szi} \end{bmatrix} \begin{matrix} \text{Forces} \\ \\ \text{Moments} \end{matrix} \tag{2.31}$$

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

$$\vec{FM}_b = \vec{FM}_b\ (\vec{\Delta}_b) \tag{2.32}$$

The deflection $\vec{\Delta}_b$ of a bearing is described by two radial deflections δ_y and δ_z two angular deflections θ_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

$$\frac{\delta\,(\vec{\Delta})_{ij}}{(\vec{\Delta})_{ij}} \le \begin{array}{l} \text{EPS1 (frictionless)} \\ \text{EPS2 (friction)} \end{array} \qquad (2.33)$$

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, $(\vec{\Delta})_{ij} = 0$, Eq (2.34) is used in place of (2.33).

$$\frac{\delta(\vec{\Delta})_{ij}}{(0.001 \times \text{NBRG})} \le \begin{array}{l} \text{EPS1 (frictionless)} \\ \text{ESP2 (friction)} \end{array} \qquad (2.34)$$

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 2×10^{-6} radians and all linear deflections are less than 5×10^{-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:

1) 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

$$\sum_m \left[\int_{-a_m}^{a_m} (\vec{Q}_m + \vec{f}_m) ds + \vec{F}_m \right] + \vec{F} = 0 \qquad (2.35)$$

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 contacts are possible.

where: $\vec{Q}_m$ is the vector normal load per unit length of the contact. See Ref. [1].

$\vec{f}_m$ is the vector of friction force per unit length of the contact. See Ref. [10].

$\vec{F}$ is the vector of inertia and drag forces. See Ref. [1]

s 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].

$\vec{F}_m$ 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:

$$\sum_m \left[\int_{-a_m}^{a_m} \vec{r}_m \times (\vec{Q}_m + \vec{f}_m) ds \right] + \vec{r}_m \times \vec{F}_m + \vec{M}_I = 0 \qquad (2.36)$$

$\vec{Q}_m, \vec{f}_m, \vec{F}_m, a_m$, and s are defined above, $\vec{M}_I$ is a vector of inertia moments. For the definition of M_I refer to Ref. [1].

$\vec{r}_m$ is a vector from the rolling element center to the point of contact.

In the frictionless elastic solution $\vec{F}_m$ and $\vec{f}_m = 0$. Additionally, the only rolling element inertia term considered in the frictionless solution is centrifugal force, plus the gyroscopic 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

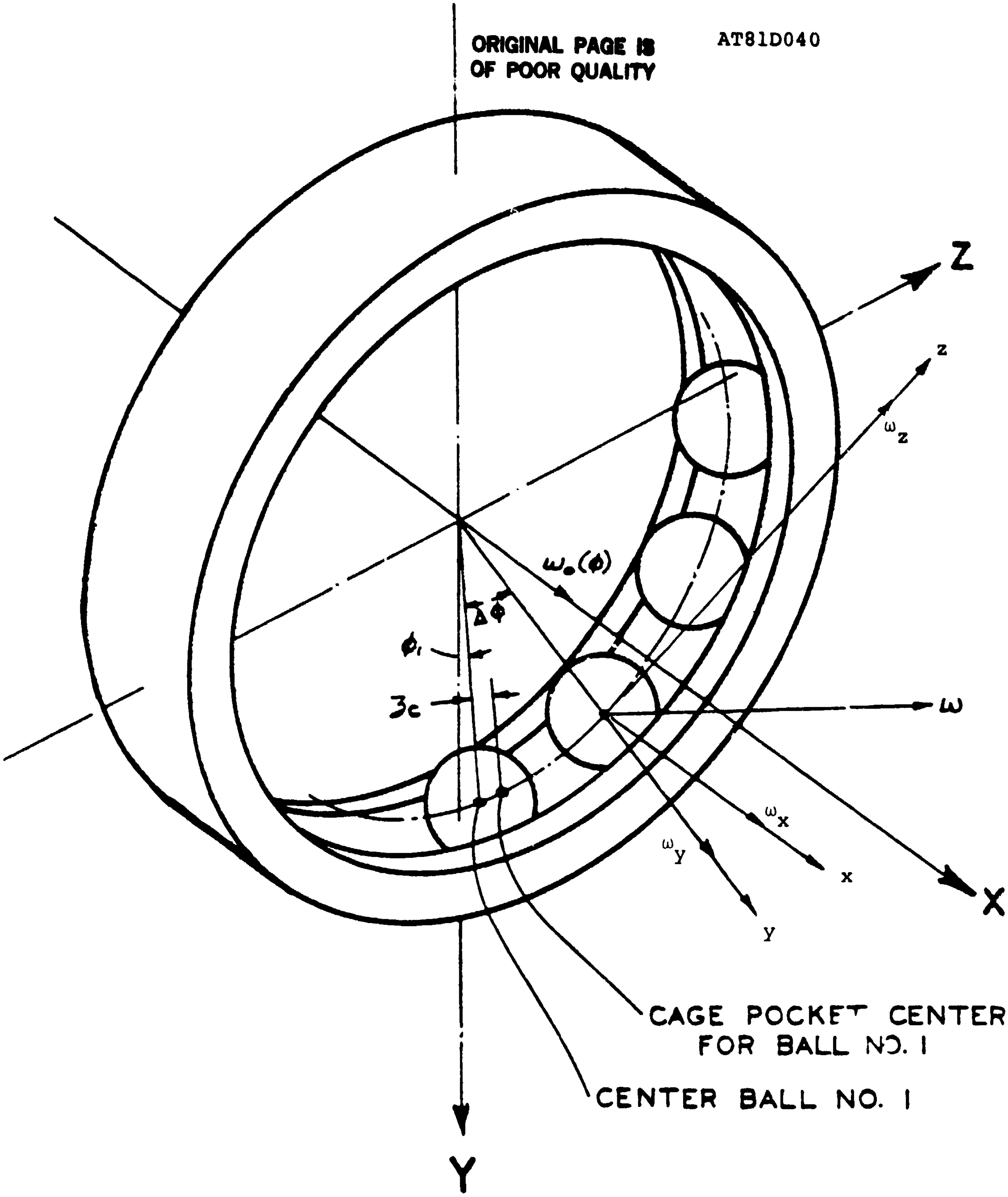


FIGURE 2.4

BEARING INERTIAL (XYZ) AND ROLLING ELEMENT (xyz), COORDINATE SYSTEMS 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 xl, yl, ωx, ωy, ωz, and ωo.

where x_1 is the rolling element axial position relative to the outer race groove curvature center.
y_1 is 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 $\Theta z = \arctan(\omega y/\omega x)$.

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_c. 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 Θ_c 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.

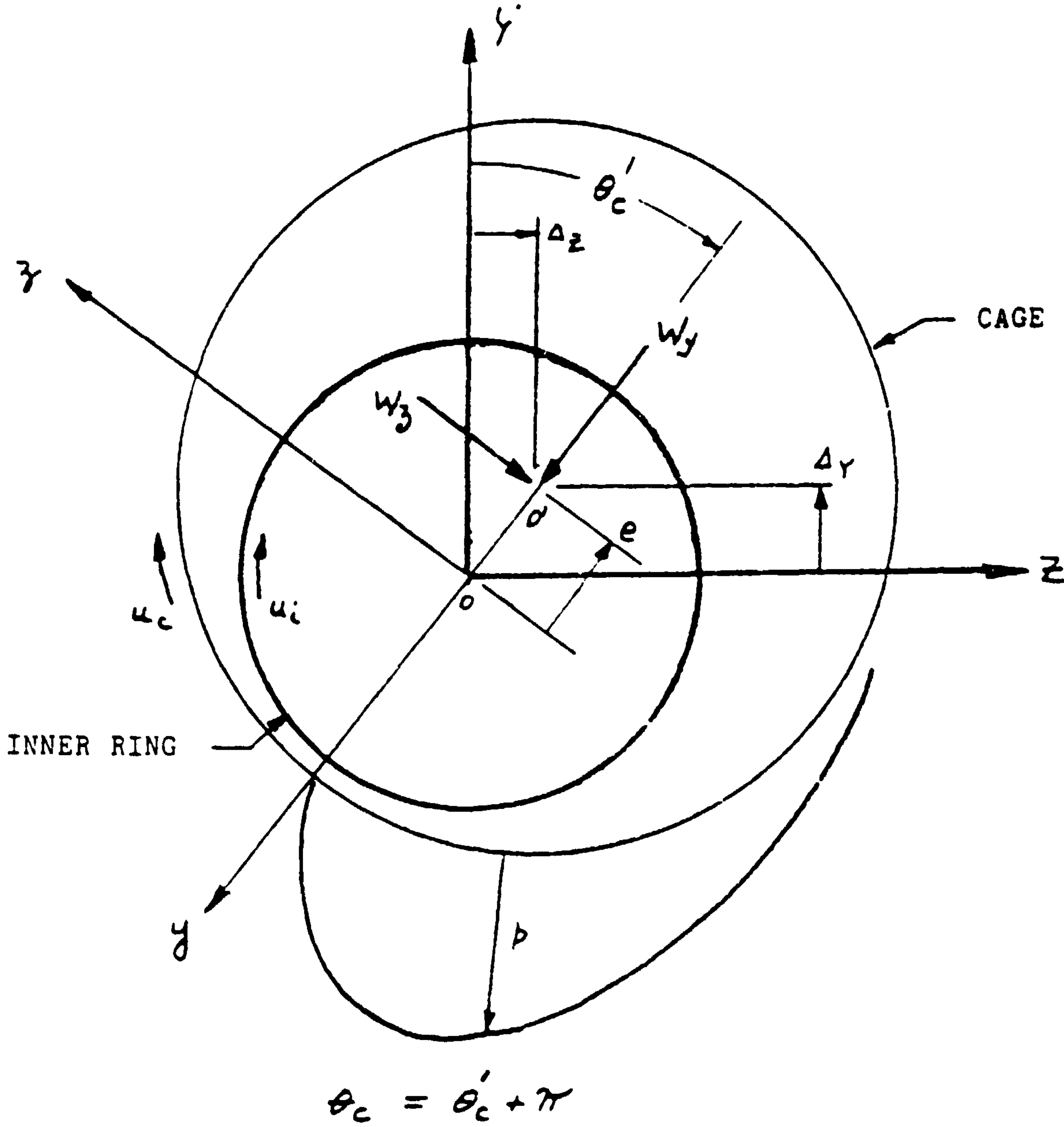


FIGURE 2.5 INNER RING-CAGE LAND CONTACT GEOMETRY

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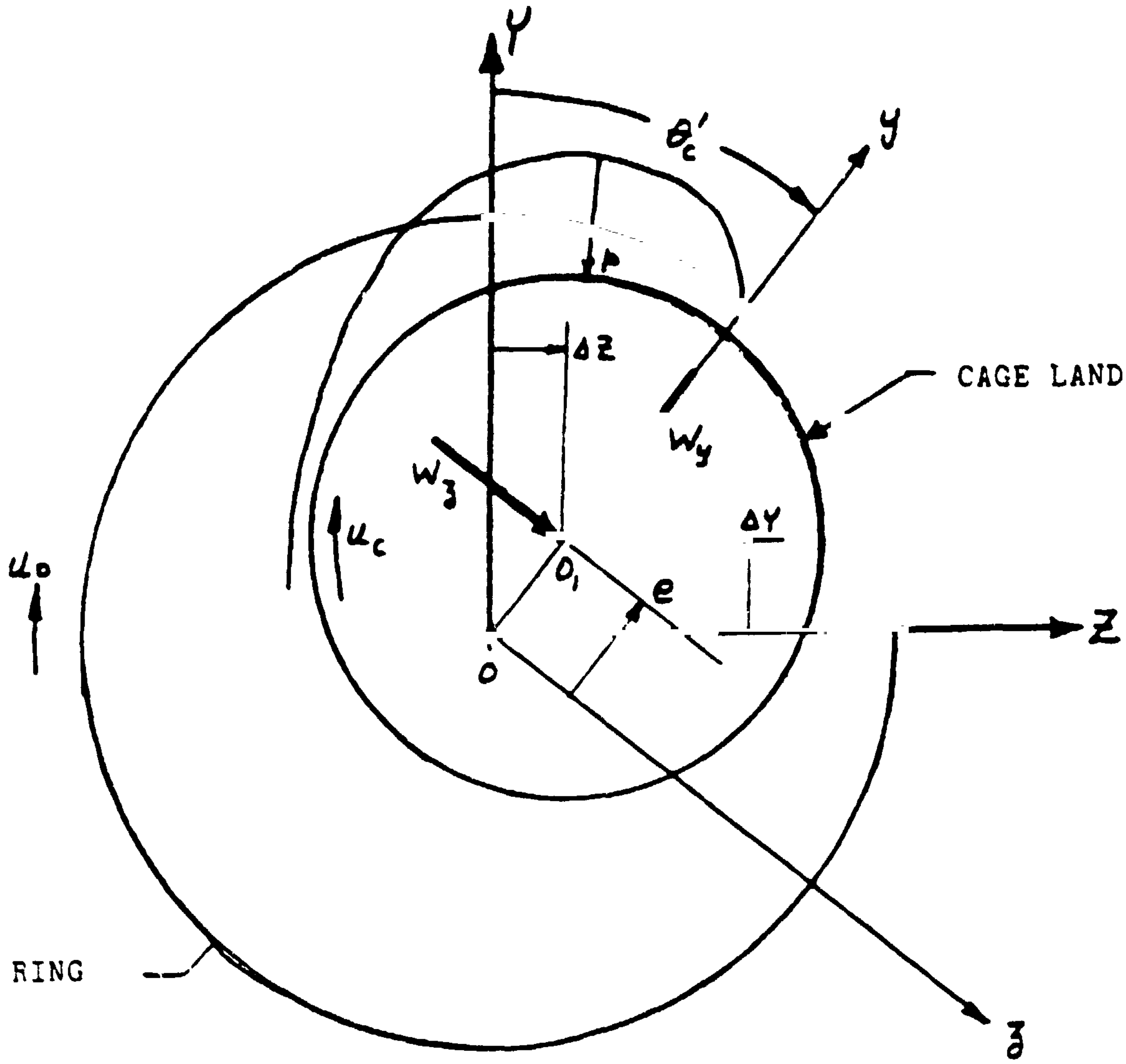


$\theta_c = \theta_c'$

FIGURE 2.6 OUTER RING-CAGE LAND CONTACT GEOMETRY

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:

$$\left|\frac{\Delta X_{(K)i}}{X_{(K-1)_i}}\right|_{i=1\ldots n} < \begin{cases} \text{EPS1} & \text{frictionless} \\ \text{EPS2} & \text{friction} \end{cases} \qquad (2.37)$$

or Where K and K-1 refer to iteration numbers

$$\left[\frac{\sum_{i=1}^{n} EQ_i^2}{n}\right]^{\frac{1}{2}} < \quad 100^* \quad \begin{cases} \text{EPS1} & \text{frictionless} \\ \text{EPS2} & \text{friction} \end{cases} \qquad (2.38)$$

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 10^5 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/mm^2)
Density - (gm/cm^3)
Kinematic Viscosity - (cs)
Temperature - (degrees centigrade) (^oC)
Coefficient of Thermal Expansion - ($^oC^{-1}$)
Thermal Conductivity - (Watts/m/^oC)

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:

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

ΔX_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:

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

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.

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 0 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:

0 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.

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 $\Lambda \leq 0.6$ to 3.0 for $\Lambda \geq 10.0$.

Item 6: Columns 71-78, "Orientation angle of the first rolling element". ($\phi 1$) (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. $\phi_1 = 0$ places a rolling element on the Y axis and is the choice customarily made. ϕ_1 can be designated 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. <u>To take advantage of bearing symmetry and the computer time savings which result, ϕ_1 must be specified as zero or left blank.</u>

<u>Item: 7</u> 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 <u>Card Type B2 - Bearing Geometry and Outer Ring Speed</u>

3.3.2.1. <u>Ball Bearing Geometry</u>

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 <u>is not</u> the manufacturer's designated contact angle, α, but an auxilliary angle, α_o, the calculation for which shall be demonstrated.

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

$$\alpha = \cos^{-1}\left[\frac{2A - Pd}{2A}\right] \tag{3.1}$$

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

where: r_o 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.

$$S_\alpha = A \sin\alpha \tag{3.3}$$

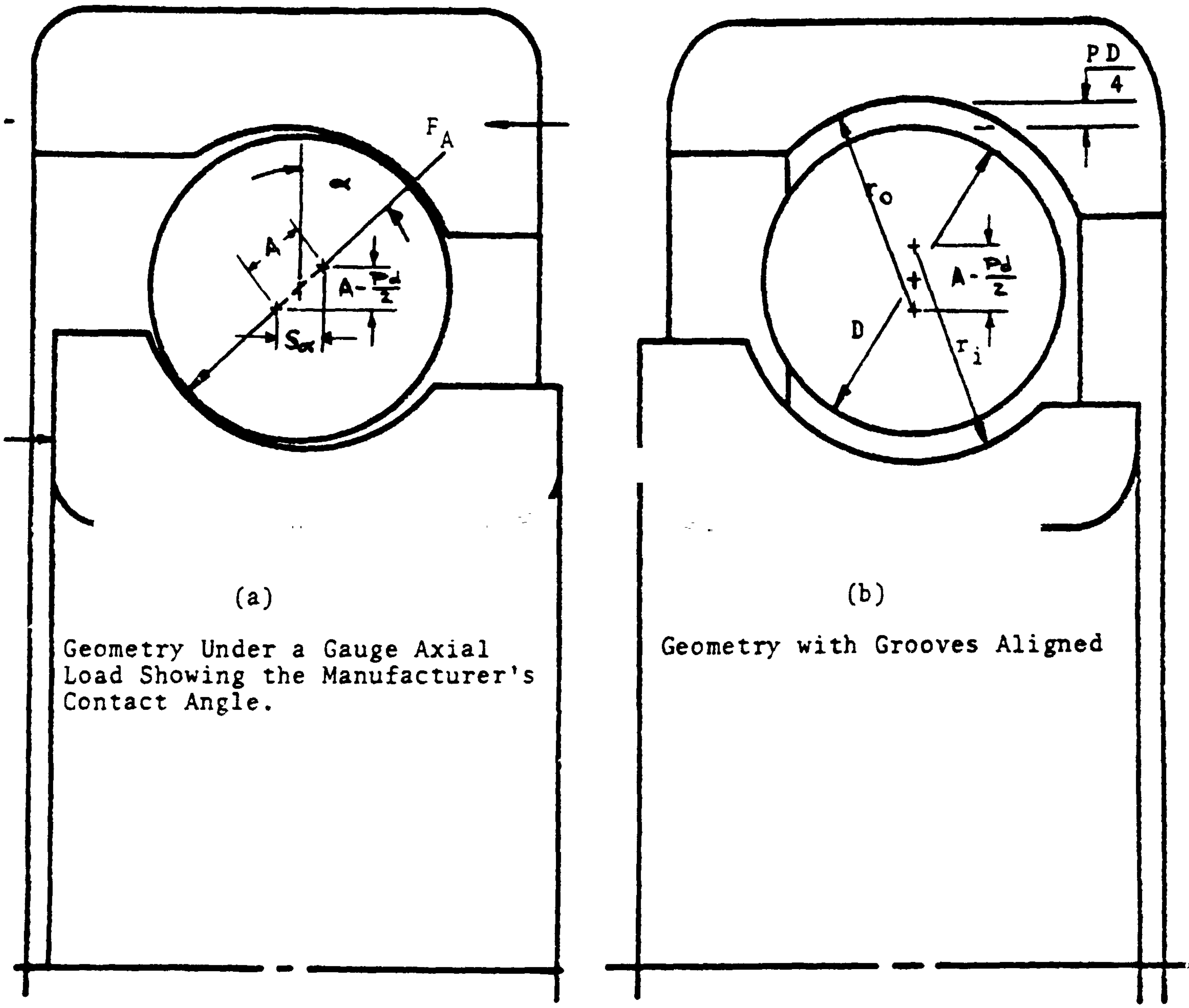
When angular contact ball bearings are mounted with some diametral play, the grooves are offset an amount $S\alpha_o$ such that $S_{\alpha o} < S_\alpha$. The diametral play which obtains at this condition is Sd. This diametral play is usually known by the engineer or designer and is usually required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

$$\alpha_o = \tan^{-1}\left[\frac{S_{\alpha o}}{A - \frac{Pd}{2}}\right] \tag{3.4}$$

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

If $S\alpha_o$ is unknown, the following equation may be solved for α_o.

FIGURE 3.1 ANGULAR CONTACT BALL BEARING GEOMETRY



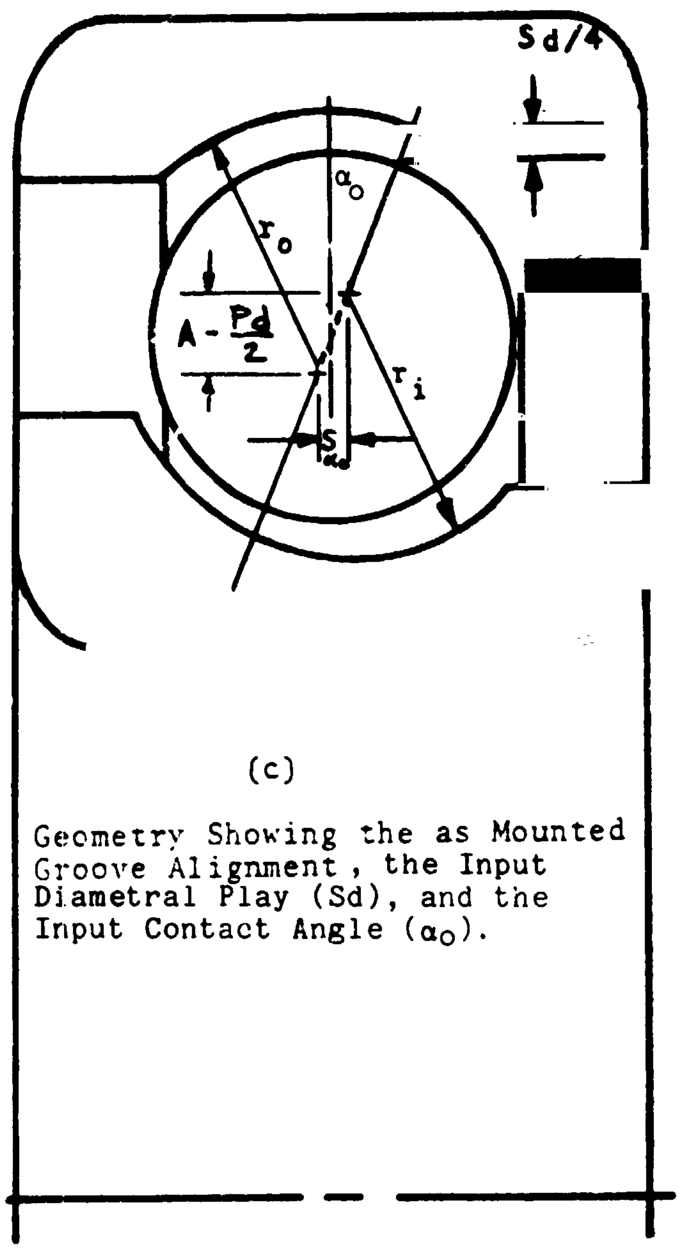
(a)

Geometry Under a Gauge Axial Load Showing the Manufacturer's Contact Angle.

(b)

Geometry with Grooves Aligned

FIGURE 3.1 ANGULAR CONTACT BALL BEARING GEOMETRY (CONTINUED)



(c)

Geometry Showing the as Mounted Groove Alignment, the Input Diametral Play (Sd), and the Input Contact Angle (α_o).

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

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_o and diametral play Sd must be determined and input. Typically the values of D, r_o, 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:

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

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

$$Sd = Sd' + (2r_i-D)(1-\cos \alpha_s) - 2A(1-\cos \alpha_o) \tag{3.7}$$

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.

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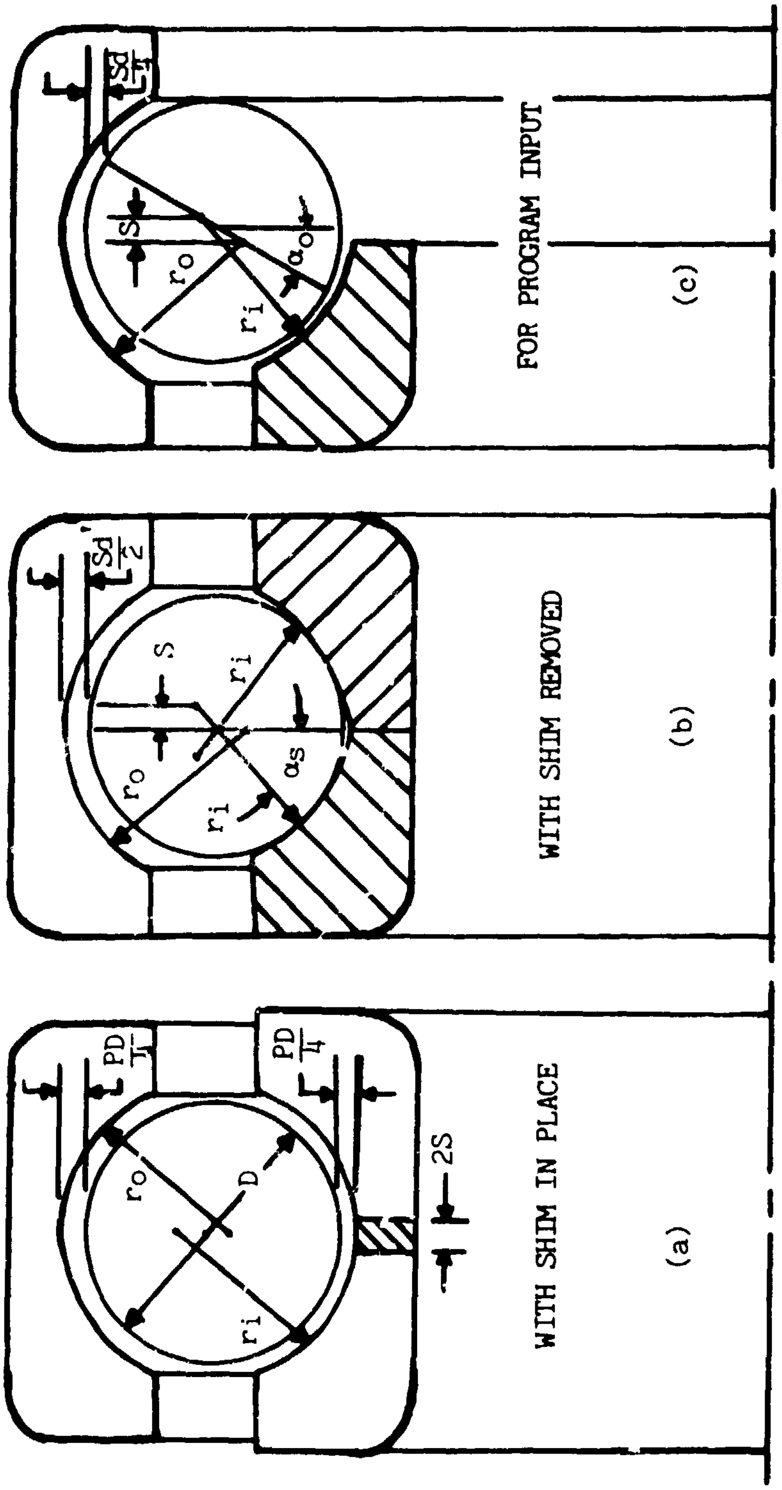


FIGURE 3.2 SPLIT INNER RING BALL BEARING GEOMETRY

3.3.2.2 Tapered Roller Geometry

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 cup angle (α_o) 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.

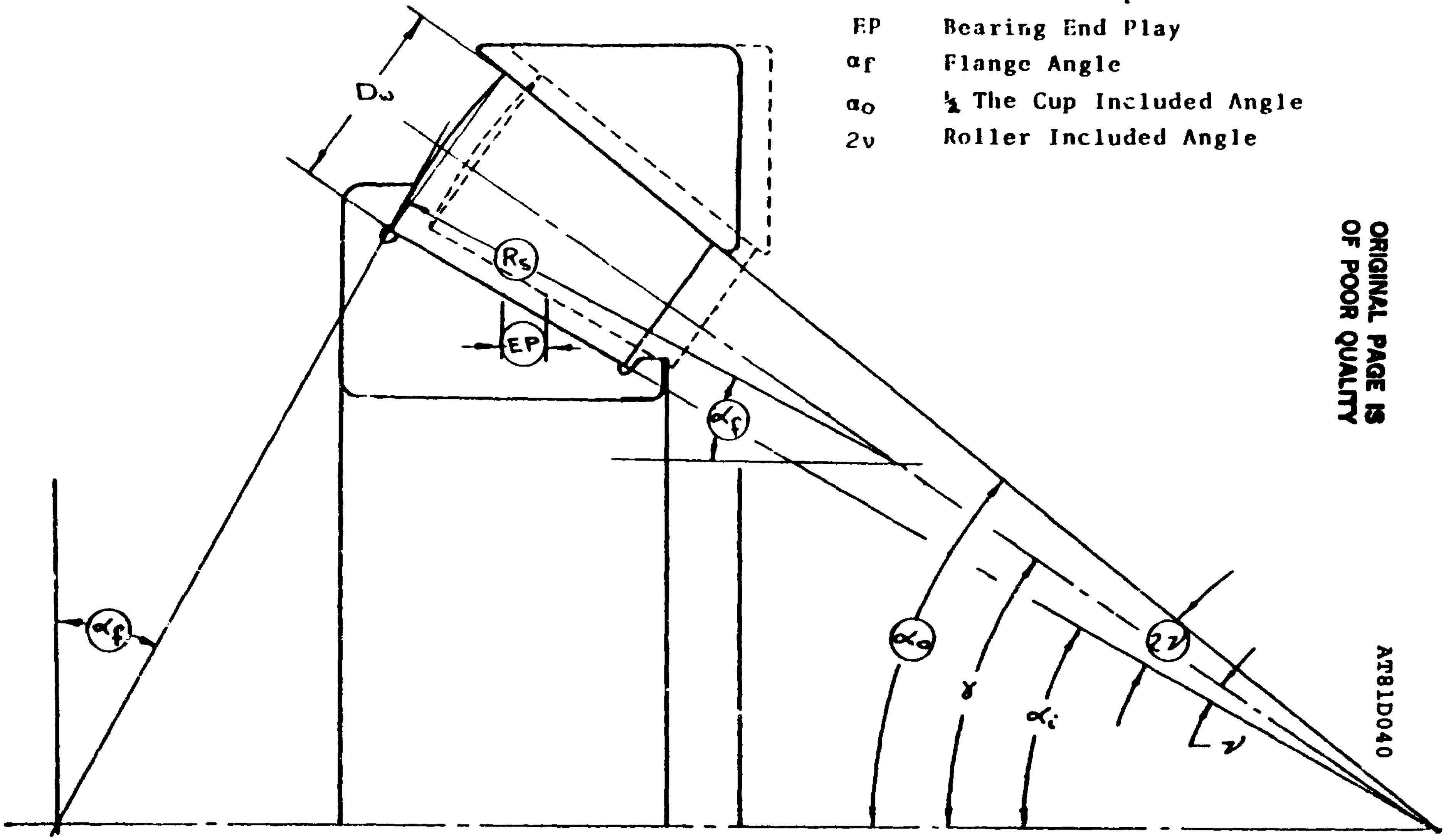
3.3.2.3 Cylindrical Roller Bearing Geometry

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

FIGURE 3.3 TAPERED ROLLER BEARING GEOMETRY

Note: Circled nomenclature represent input variable.

RS	Roller End Sphere Radius
EP	Bearing End Play
α_f	Flange Angle
α_o	½ The Cup Included Angle
2ν	Roller Included Angle



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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_w in Fig. 3.4. All bearing geometrical relationships are calculated based on D_w.

$$D_w = D + 2r_{eo} \sin \nu \tag{3.8}$$

where: D is the measured large end diameter. r_{eo} 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 r_o is the corner radius at the roller large end

$$r_{eo} = \frac{r_o (1 + \sin \nu)}{\cos \nu} \tag{3.9}$$

ν 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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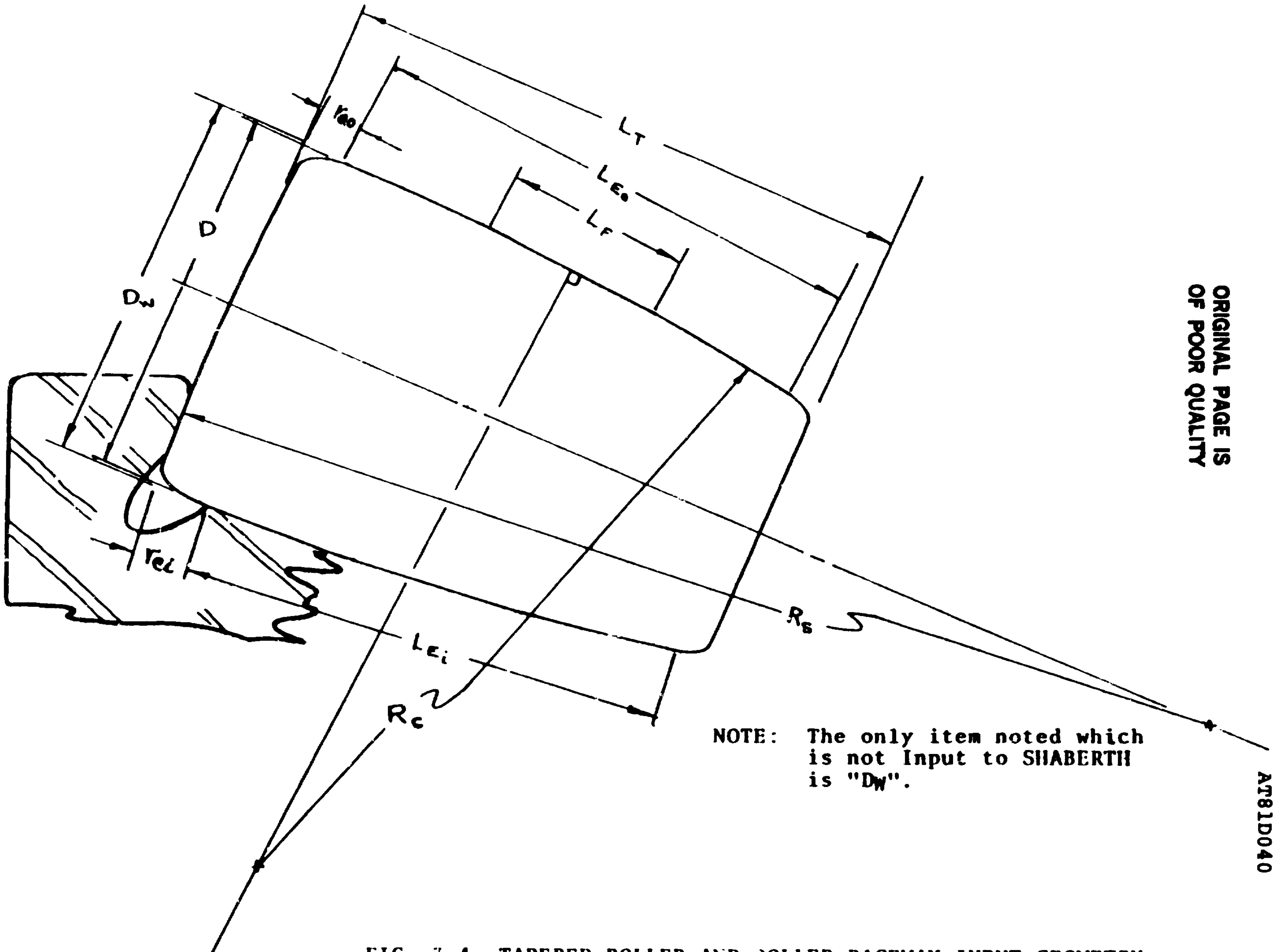


NOTE: The only item noted which is not Input to SHABERTH is "D_W".

FIG. 3.4 TAPERED ROLLER AND ROLLER-RACEWAY INPUT GEOMETRY

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_c. 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.

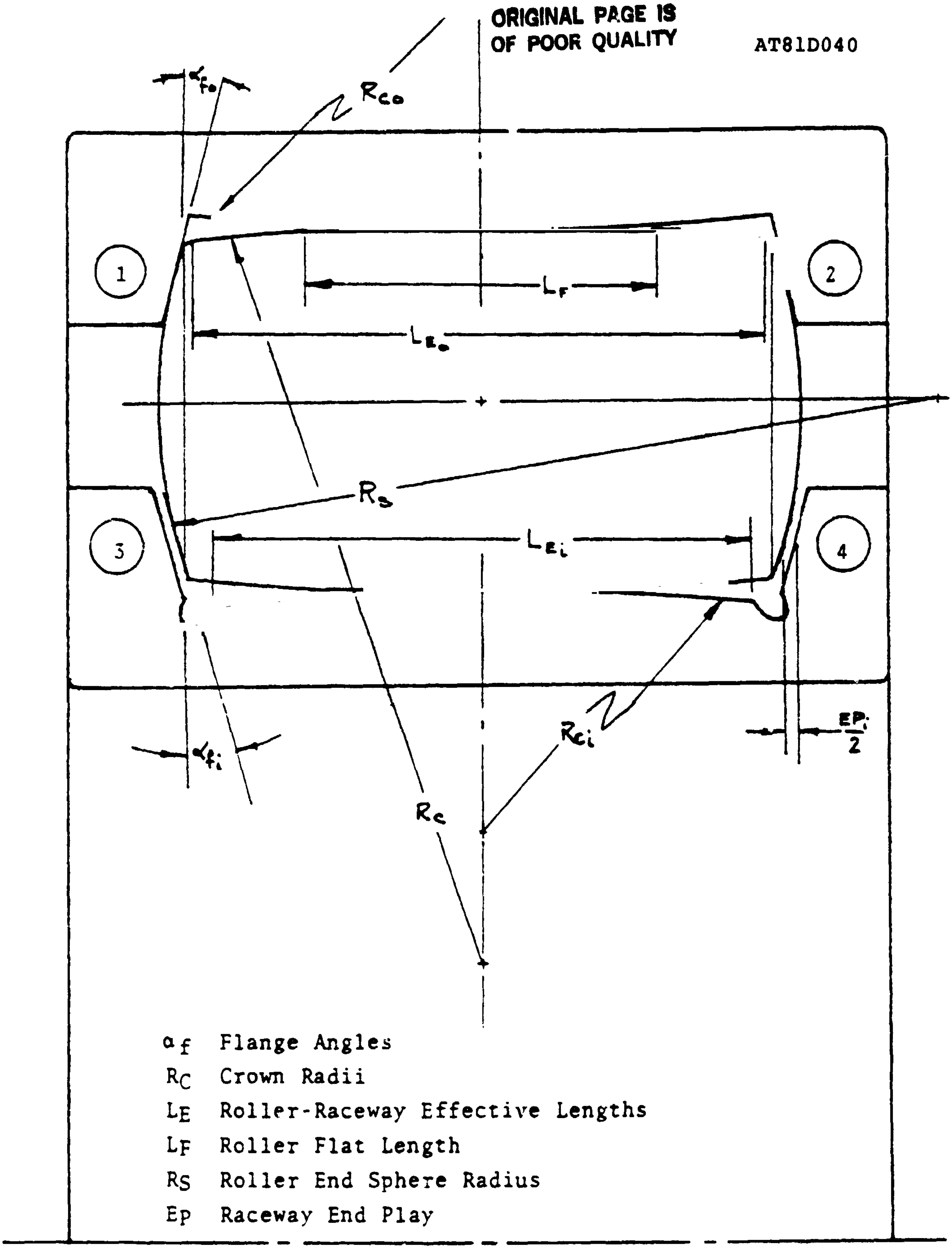
3.3.4.2 Tapered Roller Bearing Contact Geometry

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:

$$L_e^* = L/\cos\nu \qquad (3.10)$$

However, since the roller has corners at the large and small ends, the actual effective length is less than L_e^*. 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.



α_f Flange Angles
R_C Crown Radii
L_E Roller-Raceway Effective Lengths
L_F Roller Flat Length
R_S Roller End Sphere Radius
E_P Raceway End Play

FIG. 3.5 CYLINDRICAL ROLLER BEARING GEOMETRY

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. x 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 Card Type B4B Roller Flange Geometry for Cylindrical Roller Bearings

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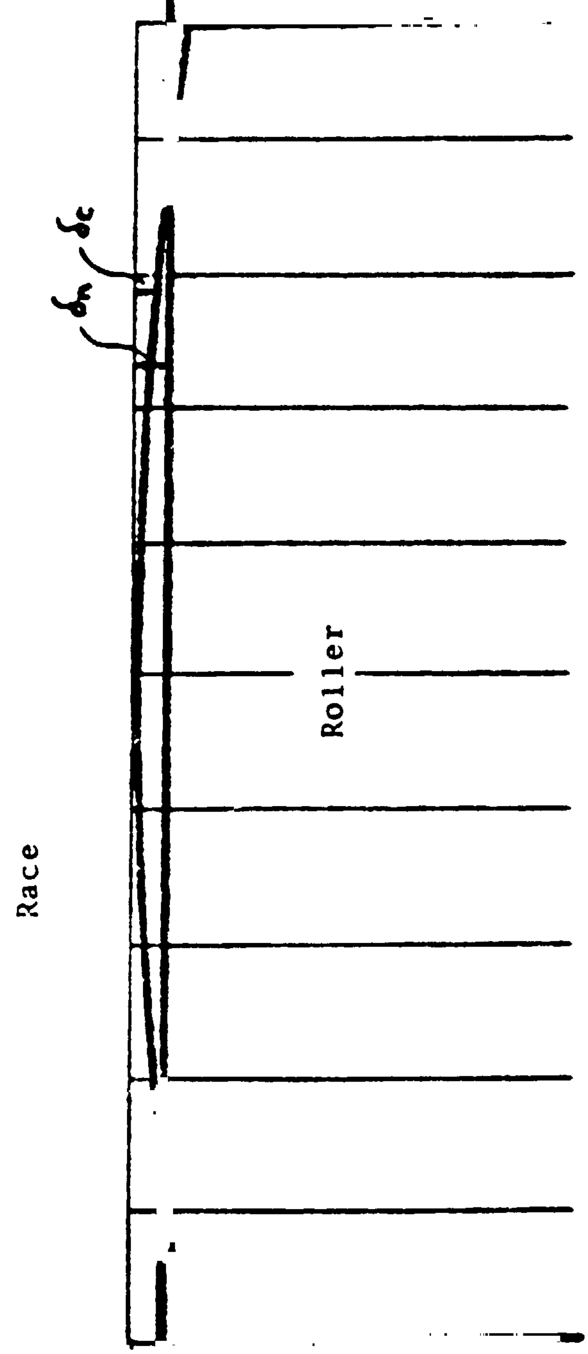

FIGURE 3.6 ROLLER-RACE LAMINATION SHOWING RELATIVE APPROACH (δn) AND CROWN DROP (δc)

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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 outer 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.

3.3.6 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 micro-geometry 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 race-way and rolling element surfaces are 1 to 2 degrees. This card is omitted if the solution level is NPASS = 0.

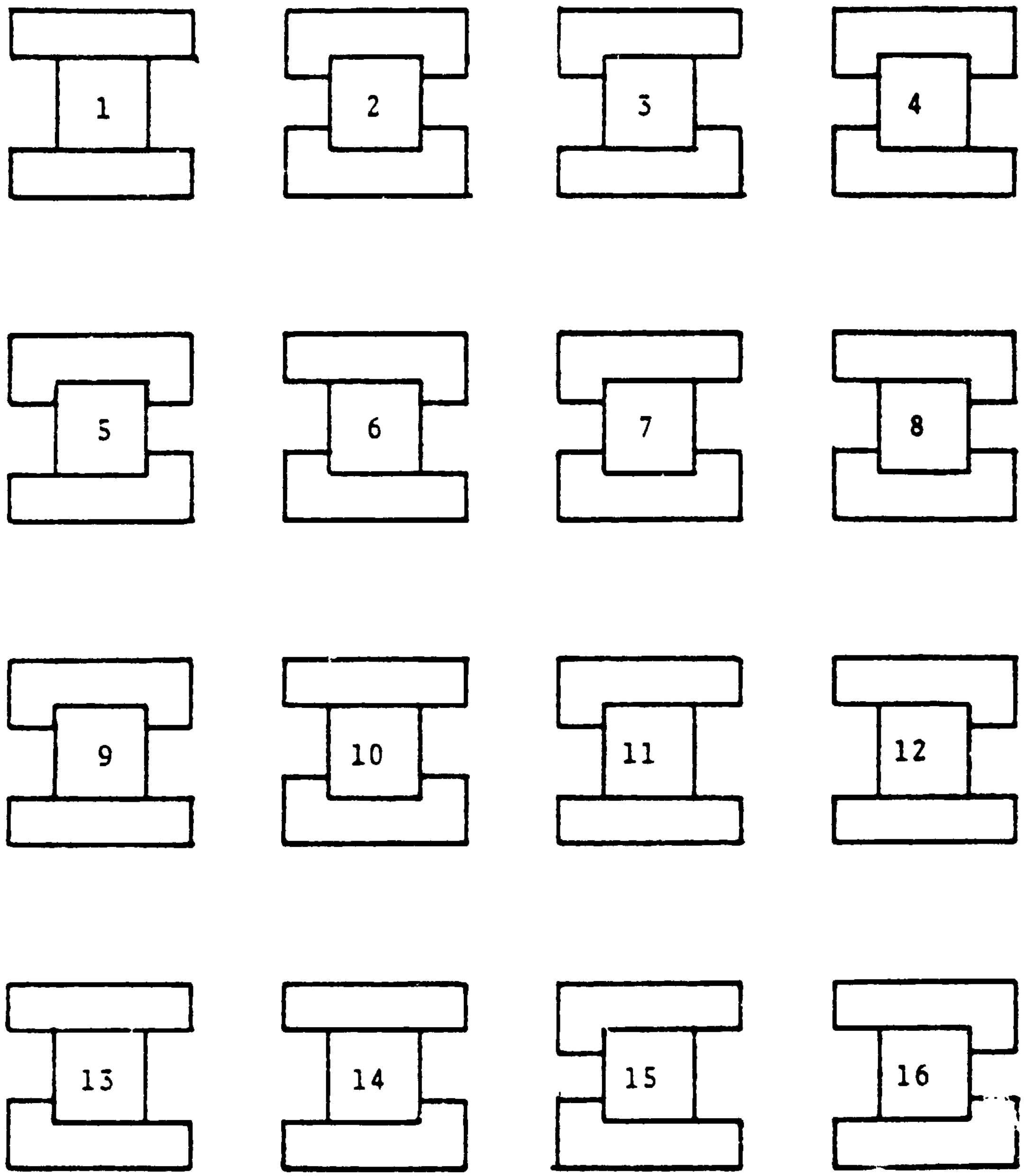

FIG. 3.7 CYLINDRICAL ROLLER BEARING FLANGE INVERSIONS

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 B11 - 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/mm^2 ($29.6 x 10^6$ 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/cm^3 (0.282 lb/in^3) 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 $12.24 x 10^{-6}$ $1/^{o}C$ ($6.8 X 10^{-6}$ $1/°F$) is used.

3.3.15 Card Type B15 - Lubrication and Friction Data

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

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. $0 \leq XCAV \leq 100$.

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.

Item 7 is 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.

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 executed (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 T1

Card type T1 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 T1 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 TYPE	KINEMATIC VISCOSITY (CS)		WALTHER EQUATION CONSTANTS		DENSITY @ 60°F ρ GM/CM3	THERMAL CONDUCTIVITY KF WATT/M-°C	THERMAL COEFFICIENT OF EXPANSION G 1/°C	TEMPERATURE VISCOSITY COEFFICIENT β 1/°C
		100°F 37.8°C	210°F 100°C	A	B				
1	Mineral Oil	64	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	7.09×10^{-4}	0.0238
3	C-Ether	25.4	4.13	11.452	4.113	1.201	0.119	7.47×10^{-4}	0.0302
4	MIL-L-23699	28.0	5.1	10.207	3.655	1.010	0.152	7.45×10^{-4}	0.0290

TABLE 1

LUBRICANT PROPERTIES OF FOUR OILS USED

AT81D040

as Item 3 of Card Type T1 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.

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 output 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). IT1 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 $|t_{(N)i} - t_{(N-1)i}| \leq$ 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, T_f . The transient solution will generate a history of the system performance which will encompass a total elapsed time of

$$(T_f - T_s) \text{ seconds}$$

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

$$t_{k+1} = t_k + \frac{q_k}{\rho C_p V} \Delta T \qquad (3.11)$$

$$\Delta T = \text{STEPIN}$$

The user may specify STEPIN. If left 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 (STEPIN) even if the user inadvertently inputs a shorter interval. Computational time savings result if BTIME is greater than STEPIN, however, accuracy might be lost.

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 numbers 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 $\lambda = 46.7$ 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 which in turn is a function of the Reynolds number Re, the Prandtl number P_r as follows, (cf. [22])

$$\alpha = Nu\ \lambda\ oil/L$$

$$N_u = aR_e^b P_r^c$$

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) α 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, L_1 and L_2 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 loading to be applied to the shaft. The loads applied to each bearing are a product of the shaft-bearing solution. There is no need for 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^2.

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 Life Multipliers

4.2.2.2.1 Lubrication - This life multiplier is a function of h/σ at each concentrated contact. Its value ranges from 0.21 for $h/\sigma \leq 0.6$ to 3.0 at $h/\sigma \geq 10$. 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.

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

ORIGINAL PAGE IS
OF POOR QUALITY

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 (ω_y/ω_x) and Arctan (ω_z/ω_x) 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 Z 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 Q_{asp}/Q_{tot}.

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].

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 centrifugal 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 inner 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 "NASA" 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 print 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 GO 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 $0 \leq FA \leq 1$. 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 <u>From Subroutine SOLVXX</u>

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 necessarily 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 input bearing operating diametral clearance 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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APPENDIX A

SKF COMPUTER PROGRAM AT81Y003 SHABERTH
HIERARCHICAL 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 embodied 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 subordinate 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[illegible] the program to simulate bearings with up to 30 rolling element[illegible]

S H A B E R T H FLOW CHART (N A S A VERSION)

```
ALWAYS
 | SKF                                      Read and set bearing and bearing solution
 |   | FLAGS                                control data
 |   | TYPE
 |   -------------------------------------
 |   | PROPST                               Set material and lubricant properties and
 |   | LUBDOP                               calculate constants for temperature dependency
 |   | LUBCON                               calculations
 |   -------------------------------------
 |   | DATOT                                Write bearing input and hardcoded, preset data
 |   |   | TITLE
 |   |   | FITDAT
 |   |   | ROLDAT
 |   -------------------------------------
 |   | CNVRT                                Calculate bearing related constants
 |   | CONS
 |   |   | CONBR1
 |   |   | BCON
 |   |   |   | ABDEL
 |   |   | TCON
 |   |   |   | ABDEL
 |   |   |   | SLICES
 |   |   | CRCON
 |   |   |   | ABDEL
 |   |   |   | SLICES
 |   |   | CONBR2
 |   | SPRING
 |   -------------------------------------
 | TEMPIN                                   Read and write thermal and thermal solution
 |   | INDUM                                control data and calculate heat transfer
 |   | RWHTC                                coefficients
 |   | RWG
 |   | TMAP
 | SHAFT                                    Read and write shaft geometry, loading and
 |   | ZERO                                 bearing position data
 |   | ARRANG
 |   |   | ORDERR
 |   | OUTINP
 |   -------------------------------------
 |   | AXLBOJ                               Calculate shaft deflection constants
 |   |   | JMVIKT
 |   |   | SNITMT
 |   |   | NUMLOS
 |   |   |   | DUBSIM
 |   |   |   |   | MEIE
 |   |   |   |   | MEIL
 |   |   | SIMQ
 |   | REACT
 |   | INDEL
 |   |   | AXLBOJ
 |   -------------------------------------
 |   | PARI                                 Calculate shaft influence coefficients
 |   |   | PAR
 |   |   |   | AXLBOJ
 |   -------------------------------------
 | GUESS                                    Guess values of rolling element and cage
 |   | GRBRG                                variables, positions and speeds
 |   |   | GROLL
 |   | GBBRG
 |   |   | GBALL
 |   | GTBRG
 |   |   | GSTPOS
 |   |   | GSTSPD
 |   | GUESCG
 |   | VARRDC
 -------------------------------------------
 | ALLT                                     Begin the solution of the steady state transient
 -------------------------------------------  thermal and temperature dependent shaft-bearing
     | SHABC                                analyses
     |                                      Calculate Shaft-Bearing Equilibrium, in steps,
     |                                      for the desired NPASS solution level
     |   ---------------------------------
     |   | FIT                              Calculate bearing diametral clearance
     |   |   | INTFIT
     |   |   |   | SIMEQ
     |   ---------------------------------
     |   | SONR1    PERFORM THESE SONR1 CALCULATIONS    Establish iteration scheme to satisfy shaft
     |   |   | BEARC    FOR ALL NPASS VALUES.          and bearing inner ring equilibrium
     |   |   ---------------------------------
     |   |   |   |
     |   |   |   |
```

#1 THERMAL EQUILIBRIUM TIME TRANSIENT CALCULATIONS
#2 BEARING CLEARANCE
#3 SHAFT BEARING EQUILIBRIUM
#4 ROLLING ELEMENT EQUILIBRIUM

```
BEAR                                            Establish iteration scheme to satisfy rolling
  PREPAR                                        element equilibrium. Solve rolling element
    INITX                                       equilibrium, bearing by bearing, one element
      UNLOAD                                    at a time
      UNLODT
    MAXMIN
  SOLVIS
    INSOLV
    EQCHEK
      DAMPCO
      ERCHEK
    EQS = BRGGEQ (WITHOUT FRICTION)             Evaluate ball-raceway contact loads, ball
      EQBALL (BALL BEARING)                     centrifugal force and ball equilibrium equations
        BGCTRL
        BALLIN
        BALLEQ
      EQTAPR (TAPERED ROLLING BEARING)          Evaluate tapered roller-raceway and flange
        BGCTRL                                  contact loads, centrifugal force and gyromoment
        TAPIN                                   and roller equilibrium equations
        TAPEQ
          TPNORM
          FLNORM
            PUSH
      EQCYL (CYLINDRICAL ROLLER BEARING)        Evaluate cylindrical roller-raceway and flange
        BGCTRL                                  contact loads, centrifugal force and roller
        ROLLIN                                  equilibrium equations
          FLNDEF
          EQ
          TPNORM
          FLNORM
            PUSH
            PULL
    PARDER                                      Calculate partial derivatives of rolling element
      EQS = BRGGEQ                              equilibrium equations with respect to rolling
    SIMQ                                        element positions and calculate new positions
    EQCHEK
      EQS = BRGGEQ
      ERWRIT
  SUMF                                          Sum rolling element-inner ring forces and moments
    SUMK
    SUMFL
  LIFE                                          Calculate bearing fatigue life
BFILL                                           Add partial derivatives of bearing inner ring
SHAPA                                           force with respect to inner ring displacement
  FILLB                                         to the shaft equilibrium equation and obtain
SIMEQ                                           new inner ring positions
BEAR                                            After shaft equilibrium is satisfied, calculate
  PRESS                                         bearing reaction forces at the bearing equilibrium
                                                positions and estimate a new set of rolling element
                                                speeds
VISC 2                                          Evaluate temperature dependent lubricant
  ALPHAO                                        properties and constants.
DRAGNO
STCCN
EVALUT (IF NPASS = 1) SKIP IF NPASS = 2 OR 3    Evaluate bearing performance with estimated
  PREPAR                                        rolling element speeds.
  BRGGEQ (WITH FRICTION)
    EQBALL                                      Evaluate ball-raceway contact loads.
      BGCTRL
      BALLIN
      BALLEQ
        FMIX/NASA                               Evaluate ball-raceway EHD film thickness,
          TINT                                  inlet and concentrated contact friction
          FILM                                  forces and heat generation rate.
          THERF1
          STARFC
          HDFRI1
            FBAR
          FRINT/NASA
            ALLEN
            FRICTN
      CAGESP                                    Evaluate cage speed, ball-cage normal and
      CAGEEQ                                    friction forces and heat rates plus cage
        CGLAND                                  rail-ring land normal and friction forces
          CGDRY                                 plus heat generation rates.
          CGWET
        CGNE
          CGBALL
          CGROLL
        SUMCGL
        SUMRE
```

```
BRGAX                                   Evaluate ball, position dependent, accelerations.
   BRAX
LIFE                                    Calculate bearing fatigue life
   FLMFAC
HUTRAC                                  Calculate ball-raceway heat transfer coefficients
   HUT
EQTAPR                                  Evaluate roller-raceway and roller-flange contact
   BGCTRL                               loads
   TAPIN
   TAPEQ
      TPNORM
      FLNORM
         PUSH
         PULL
      FMIXR/NASA                        Evaluate roller raceway EHD film thickness inlet
         SPEEDS                         and concentrated contact friction forces and
         FILM                           heat generation rate
         THERF1
         STARFC
         HRFRIC
         SPEEDS
         ALLEN
         FRICTN
      FLMIX/NASA                        Evaluate roller-flange EHD film thickness inlet
         FILM                           and concentrated contact friction forces and
         THERF1                         heat generation rate
         STARFC
         HDFRI1
            FBAR
         ALLNPT
         FRICTN
   CAGSPD                               Evaluate tapered roller bearing cage speed, cage
   CAGEEQ                               forces, roller accelerations, bearing fatigue
   BRGAX                                life, roller-raceway and roller-flange heat
   LIFE                                 transfer coefficients
   HUTRAC
   HUTFLN
EQCYL                                   Evaluate cylindrical roller-raceway and flange
   BGCTRL                               normal and friction forces and heat generation
   ROLLIN                               rates plus EHD film thicknesses
      FLNDEF
   ROLLEQ
      TPNORM
      FLNORM
         PUSH
      FMIXR/NASA
      FLMIX/NASA
   CAGSPD                               Evaluate cylindrical roller bearing cage speed,
   CAGEEQ                               cage forces, roller acceleration, bearing fatigue
   BRGAX                                life, plus roller-raceway and roller-flange heat
   LIFE                                 transfer coefficients
   HUTRAC
   HUTFLN (IF NPASS = 1, RETURN TO ALLT.)

BEAR  (IF NPASS = 2) SKIP IF NPASS = 3  Solve the rolling element and cage equilibrium
   PREPAR                               equations with friction included, using the
#4 SOLV13                               inner ring positions established by the
      EQS = BRGGEQ  (WITH FRICTION)     frictionless solution

SONR1  (IF NPASS = 3)                   Solve shaft, rolling element and cage equilibrium
   BEARC                                equations with friction included
#3    BEAR
         PREPAR
      #4 SOLV13
            EQS = BRGGEQ  (WITH FRICTION)
         SUMF
            SUMK
            SUMFL

DELIV3/SKF
   TITLE
   RITE
   RITE1
   REOUT3                               Write shaft-bearing output
      RITE2
   REFLOT
      RITE4
      RITE2
   FLNGOT
```

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Routine	Description
FILLGT SOLV13 | EQS = NET (FOR STEADY STATE THERMAL SOLUTION) | | 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 — 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. LUBRICATION AND FRICTION EFFECTS ARE CONSIDERED USING RACEWAY CONTROL (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 RING PLUS ROLLING ELEMENT AND CAGE EQUILIBRIUM ARE DETERMINED CONSIDERING ALL ELASTIC AND FRICTION FORCES.

APPENDIX B

HEAT TRANSFER COMPUTATION NOTES

APPENDIX B

HEAT TRANSFER INFORMATION

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 $q_{ci,j}$(W) that is conducted from node i to node j may be expressed by,

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

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°C).

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

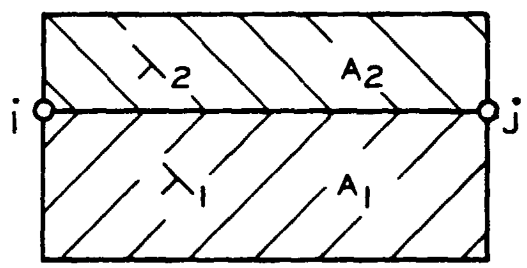

Fig. B-1

$$\lambda_{ij} = \frac{\lambda_1 A_1 + \lambda_2 A_2}{A_{ij}}$$

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

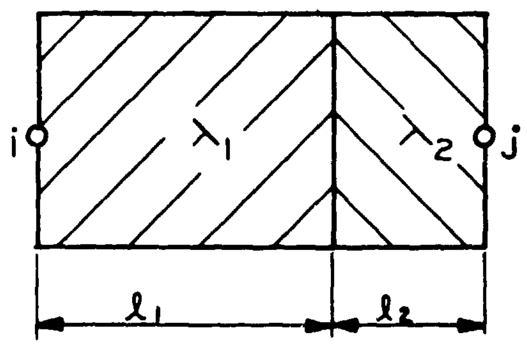

Fig. B-2

$$\lambda_{ij} = \frac{\ell_{ij}}{\ell_1/\lambda_1 + \ell_2/\lambda_2}$$

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

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 <u>free convection</u> may be expressed by

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

where

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

in which

$$\alpha_{ij} = \begin{cases} 2.5 \cdot 10^{-2}\ \text{W/m}^2\text{-}(\text{degC})^{1.25} & \text{for hot surfaces facing upward and cold surfaces facing downward} \\ 1.4 \cdot 10^{-2}\ \text{W/m}^2\text{-}(\text{degC})^{1.25} & \text{for hot surfaces facing downward and cold surfaces facing upward} \\ 1.8 \cdot 10^{-2}\ \text{W/m}^2\text{-}(\text{degC})^{1.25} & \text{for vertical surfaces} \end{cases}$$

For other special conditions, α_{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

$$N_u = \frac{\alpha L}{\lambda}$$

the Reynolds number

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

and the Prandtl number

$$P_r = \frac{\rho \nu C_p}{\lambda}$$

where

L 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)

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 ($W/M^{o}C$)

ν is the fluid kinematic viscosity (M^2/sec)

ρ is the fluid density (kg/m^3)

c_p is the fluid specific heat ($J/kg^{o}C$)

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: $R_e < 2300$

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

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

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

where the Reynolds number R_e is developed by the shaft rotation.

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

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,

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

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

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

$\dot{V}_{ij}$ 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 ^oC.

T_j = temperature of node j in ^oC.

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

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

where σ, the Stefan-Boltzmann constant, is

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

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

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

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 directior $A_{i,j}$ 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,

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

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

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))

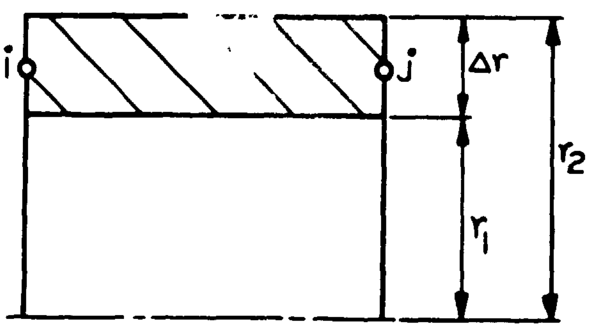

Fig. B-3

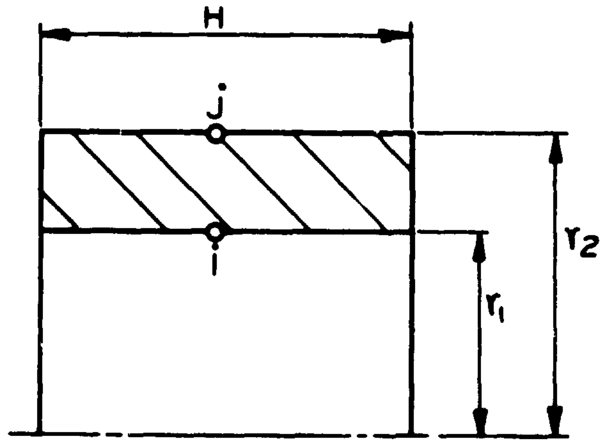

Fig. B-4(a)

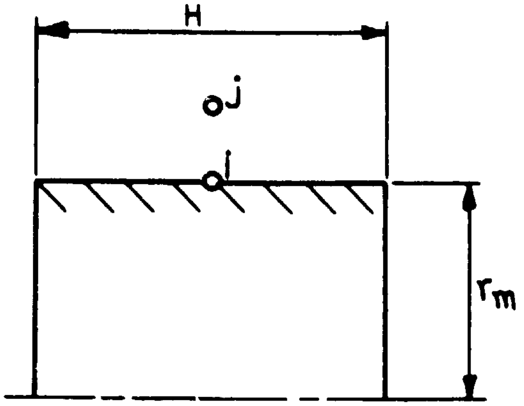

Fig. B -4(b)

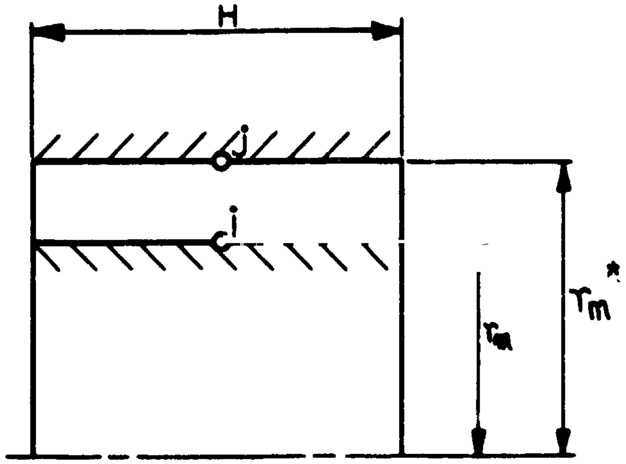

Fig. B -4(c)

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.

APPENDIX C

SKF COMPUTER PROGRAM SHABERTH
INPUT DATA FORMS

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HOW TO READ AND USE THE FOLLOWING FORMS

PROGRAM NAMES OF INPUT ITEMS, FOR EXAMPLE: V1 (columns 31–40), DIA (columns 41–50), DD(5) (columns 51–60), IZ (columns 61–65)

Card columns 1–80.

	Columns 1–16	Columns 17–19	Columns 20–30	Columns 38–41	Columns 44–50	Columns 51–60	Columns 64–65	Columns 66–80
FORMAT, IF BLANK, WRITE		I . I	ALPHANUMERIC IN THE FIELD, OR:	I I I OR:	I . I I E + I	RIGHT JUSTIFIED	I I	MEANS INTEGER RIGHT JUSTIFIED
YOU CAN WRITE YOUR DATA ON THIS LINE, FOR EXAMPLE:				3 6	5 . 6 3 E + 2	3 . 6 1 ' E + 1 [illegible]	1 6	

DESCRIPTION OF ITEM — DESCRIPTION OF ITEM — DESCRIPTION OF ITEM FOR EXAMPLE: BALL DIAMETER — EX. NO. OF BALLS

ALL VALUES ARE TO BE GIVEN IN THE UNITS OF THE SI-SYSTEM, THE INTERNATIONAL SYSTEM. EXCEPTIONS ARE LENGTHS WHICH ARE EXPRESSED IN MILLIMETERS INSTEAD OF METERS, AND PRESSURES WHICH ARE GIVEN IN NEWTONS/MM^2 INSTEAD OF NEWTONS/M^2. VOLUMES IN MM^3.

Title Card
Card Type TT1

Title(28) 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
2	0	A	4																																																																										

Title to be printed on each page.

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Columns	Field name	Format	Description
1–10	GOV(1)	F10.0	Shaft Speed (Revolutions per minute)
11–15	NBRG / IFLAG(1)	F5.0	Number of bearings
16–20	NPRINT / IFLAG(2)	F5.0	Print flag. If not blank debug output will be obtained. If=1 main loop debug only. If=2 full debug output.
21–25	IFIT / IFLAG(3)	F5.0	Fit calc. flag. Blank means no fit calculation & no iteration. Positive no. means fit iteration MAX=5. Negative no. means fit iteration MAX=given number.
26–30	ITMAIN / IFLAG(4)	F5.0	Main loop iteration flag. Positive no. or blank means pre-set max.=15 is used. Negative no. means iteration limit=absolute value of given number.
31–40	GOV(2) / EPSFIT	F10.0	Fit loop accuracy. It is defined as change of diametral clearance divided by rolling element diameter.
41–50	GOV(3) / EPS1	F10.0	Main loop accuracy for frictionless solution. When the latest change of bearing deflection is less than (EPS1 or EPS2 x Bearing Deflection), or the bearing angular and linear deflections are less than 2.10^{6} & 5.10^{-8} respectively, then iteration is stopped. If left blank, pre-set accuracy of 0.001 is used.
51–60	GOV(4) / EPS2	F10.0	Main loop accuracy for EHD or Coulumb solution. If left blank, pre-set accuracy of 0.0001
78	JUSTBR		Single Bearing Flag ***
79	IMT		Material Property Flag **
80	NPASS		Bearing Solution Level Flag*

...T CARD 2
ARD TYPE TT2

* NPASS, IFLAG(5), BEARING SOLUTION LEVEL FLAG

0 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) ASSUMPTION TO ESTIMATE ROLLING ELEMENT AND CAGE SPEEDS

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.

** IMT IF ONE SET OR MORE OF THE BEARING RING OR ROLLING ELEMENT MATERIALS IS OTHER THAN STEEL, SET IMT TO 1 AND INCLUDE CARD TYPES B11 THROUGH B14 FOR ALL BEARINGS.

*** JUSTBR IF A SINGLE BEARING IS TO BE ANALYZED, SET JUSTBR TO 1 AND OMIT ALL SHAFT DATA CARDS. USE LOADING DATA CARD INSTEAD.

Bearing Data, One Set per Bearing. All applicable cards B1 through B16 for Bearing 1, followed by B1 through B16 for Bearing 2, etc.

CARD TYPE B1

Columns	1–10	11–30	31–50	51–60	61–70	71–79	80
Variable	BD(1)			BD(43)	B^(44)	BD(2)	NC
Format	A1	5A4	5A4	F10.0	F10.0	F8.0	
Group		Ring Material Designation	Ring Material Designation	Ring Life Factors	Ring Life Factors		
Field	Bearing Type BALL CYLINDRICAL TAPERED	Inner Ring	Outer Ring	Outer	Inner		Crown Drop Flag
Description	Type B, C or T in column 1	Alphanumeric identification of inner ring material	Alphanumeric identification of outer ring material	Life Multiplier for outer ring material (See Sect. 3.3.1)	Life Multiplier for inner ring material (see sect. 3.3.1)	Orientation Angle in Degrees of Rolling Elements. 0.0 or Blank means first element is on the Y-axis.	

*NC Normally zero or blank, If set to 1, non uniform roller-raceway profile geometry is input on card types B5 and B6.

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BEARING DATA
CARD TYPE B2, One Card

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable	BD(5)	BD(6)	BD(7)	BD(8)	BD(20)	BD(3)		
Format	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0		
Description	Pitch Dia. (mm) Unit for Tapered Roller Bearing	No. of Rolling Elements	Diametral Clearance (mm)	Contact Angle (Deg)	Flange Angle (Tapered Only) See Fig.3.3 (Deg)	Outer Ring Speed (RPM)		
Tapered			Axial Play for Tapered, Single Bearing Only (mm)	½ Cup Incl. Angle for Tapered. (Deg.)				

BEARING D TA CARD TYPE B3 One Card

BD (4)

Columns	1–10
Format	F10.0

BALL BEARING

Columns 1–10
Ball Dia. (mm)

BD (4) BD (10) BD (11) BD (12) BD (20) BD (21)

Columns	1–10	11–20	21–30	31–40	41–50	51–60
Field	BD (4)	BD (10)	BD (11)	BD (12)	BD (20)	BD (21)
Format	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0

ROLLER BEARING

Roller Diameter (mm) Diameter of Roller Large End for Tapered (mm)	Roller Length End to End (mm)	Roller End Sphere Radius (mm)	Roller Included Angle (Deg)	Roller Crown Radius (mm)	Roller Flat Length (mm)

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BEARING DATA
CARD TYPE B4A - ONE CARD

Columns	1–10	11–20	21–80
Field	BD (21)	BD (22)	
Format	F10.0	F10.0	

BALL BEARING

BD (21)	BD (22)
Outer Raceway Curvature $f_o = Ro/_D$	Inner Raceway Curvature $f_i = Ri/_D$

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Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Field	BD (23)	BD (24)	BD (25)	BD (26)	BD (27)	BD (28)	BD (22)	
Format	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	

ROLLER BEARING

BD (23)	BD (24)	BD (25)	BD (26)	BD (27)	BD (28)	BD (22)
Roller Raceway Effective Length Outer (mm)	Roller Raceway Effective Length Inner (mm)	Raceway Crown Radius Outer (mm)	Raceway Crown Radius Inner (mm)	Roller Corner or Raceway Undercut Relief Outer (mm)	Roller Corner or Raceway Undercut Relief Inner (mm)	No. of Slices

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ARING DATA
BD TYPE B4B - Include Only for Cylindrical Bearings

Field	Columns	Format	Description
BD (31)	1–10	F10.0	Flange Angle Outer Ring (Deg)
BD (32)	11–20	F10.0	Flange Angle Inner Ring (Deg)
BD (33)	21–30	F10.0	Outer Ring End Play (mm)
BD (34)	31–40	F10.0	Inner Ring End Play (mm)
BD (35)	41–50	F10.0	Flange Inversion Flag.
	51–80		

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Omit Cards B5 and B6 if crown drop flag (column 80 on card B1) is 0 or blank.

Bearing Data

Card Type B5, as many as needed, maximum of 3, for the outer race

RLAM(1,5)	(2,5)	(3,5)	(4,5)	(5,5)	(6,5)	(7,5)	(8,5)
Columns 1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Crown Drop at Laminum 1 (mm)	Crown Drop at Laminum 2	... and so on, use more than one card if No. of laminae is greater than 8.					

Card Type B6, as many as needed maximum of 3, for the inner race.

RLAM(1,4)	(2,4)	(3,4)	(4,4)	(5,4)	(6,4)	(7,4)	(8,4)
Columns 1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Crown Drop at Laminum 1 (mm)	Crown Drop at Laminum 2	...and so on					

Bearing Data
Card Type B7A, one card

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

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable	BD(13)	BD(14)	BD(15)	BD(16)	BD(17)	BD(18)		
Format	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0		
Description	CLA Roughness Surface (microns)			Asperity Slope (Degrees)				
	Outer	Inner	Rolling Element	Outer	Inner	Rolling Element		

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Omit if NPASS (column 80, card TT2) is 0 or blank.

BEARING DATA
CARD TYPE B7B - <u>INCLUDE ONLY FOR ROLLER BEARINGS</u>

Variable	Columns	Format	Description
BD(225)	1–10	F10.0	Surface Roughness Outer Ring Flange (Microns)
BD(226)	11–20	F10.0	Surface Roughness Inner Ring Flange (Microns)
BD (227)	21–30	F10.0	Surface Roughness Rolling Element End (Microns)
BD (228)	31–40	F10.0	Asperity Slope Outer Ring Flange (Deg)
BD (229)	41–50	F10.0	Asperity Slope Inner Ring Flange (Deg)
BD (230)	51–60	F10.0	Asperity Slope Roller End (Deg)
	61–80		

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BEARING DATA
CARD TYPE B8 ONE CARD PER BEARING
OMIT IF NPASS, TITLE CARD 2, IS ZERO OR BLANK

Variable	Columns	Description
BD(36)	1–10	CAGE TYPE: -1. FOR OUTER-RING LAND RIDING; +1. FOR INNER-RING LAND RIDING; 0. FOR ROLLING ELEMENT RIDING
BD(37)	11–20	RAIL-LAND DIAMETER (mm)
BD(38)	21–30	SINGLE RAIL WIDTH (mm)
BD(39)	31–40	RAIL-LAND DIAMETRAL CLEARANCE (mm)
BD(40)	41–50	ROLLING ELEMENT CAGE POCKET DIAMETRAL CLEARANCE (mm)
BD(41)	51–60	CAGE WEIGHT (kg)
MCG	70	DEGREES OF FREEDOM OF CAGE (1 OR 3)

Card example (columns 1–5): -1.0 .0

Bearing Data

Omit Cards B9 and B10 if operating clearances are not to be calculated (ITFIT, columns 21-25 of card TT2, is 0 or blank)

Card Type B9, one card

Columns	1-10	11-20	21-30	31-40	41-50	51-60
Variable	BD(51)	BD(52)	BD(53)	BD(54)	BD(55)	BD(56)
Format	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Description	Shaft Fit Positive if Interference (mm)	Housing Fit, Positive if Interference (mm)	Shaft Effective Length (mm)	Bearing Inner Ring Width (mm)	Bearing Outer Ring Width (mm)	Housing Effective Width (mm)

Fits are diametral.

Card Type B10, one card

Columns	1-10	11-20	21-30	31-40	41-50	51-60
Variable	BD(57)	BD(58)	BD(60)	BD(61)	BD(62)	BD(64)
Format	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Description	Shaft Inner Diameter (mm)	Bearing Bore Diameter (mm)	Bearing Inner Ring Mean Outer Diameter (mm)	Bearing Outer Ring Mean Inner Diameter (mm)	Bearing Outer Diameter (mm)	Housing Outer Diameter (mm)

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Bearing Data

Include card types B11 through B14 only if operating clearances are to be calculated (ITFIT, columns 21-25 of card TT2, is nonzero) or if IMT (column 79 of card TT2 is 1).

Card Type B11, one card

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable	BD(66)	BD(67)	BD(68)	BD(69)	BD(70)			
Format	F10.0	F10.0	F10.0	F10.0	F10.0			
Modulus of Elasticity N/mm^2	Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

Card Type B12, one card

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable	BD(71)	BD(72)	BD(73)	BD(74)	BD(75)			
Format	F10.0	F10.0	F10.0	F10.0	F10.0			
Poisson's Ratio	Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

Card Type B13, one card

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable	BD(76)	BD(77)	BD(78)	BD(79)	BD(80)			
Format	F10.0	F10.0	F10.0	F10.0	F10.0			
Density gm/cm^3	Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

Card Type B14, one card

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable	BD(81)	BD(82)	BD(83)	BD(84)	BD(85)			
Format	F10.0	F10.0	F10.0	F10.0	F10.0			
Coefficient Thermal Expansion 1/°C	Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

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BEARING DATA
CARD TYPE B15, one card, Omit if NPASS, Title Card 2, is Zero or Blank

BD (102)	BD (103)	BD (104)	BD (106)	BD (107)	BD (108)	BD (109)	BD (110)
1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Lubricant Replenishment Layer Thickness (mm) — Outer Raceway	Lubricant Replenishment Layer Thickness (mm) — Inner Raceway	Percent Lubricant in Bearing Cavity	Raceway-Rolling Element Asperity Friction Coefficient	Cage Friction Coefficient	Outer Race Flange, Repl. Layer Thickness. (mm)	Inner Race Flange Repl. Layer Thickness. (mm)	Flange-Roller End Friction Coefficient

BEARING DATA
CARD TYPE B16   ONE CARD PER BEARING
LUBRICANT PROPERTIES
OMIT IF
- NPASS, TITLE CARD 2, IS ZERO OR BLANK
- ITEM 3, CARD TYPE B15, IS ZERO OR BLANK WHICH IMPLIES DRY FRICTION

BD(93)		BD(94)	BD(95)	BD(96)	BD(97)	BD(98)		
Card columns 1–2	3–20 (A2)	21–30	31–40	41–50	51–60	61–70	71–75	76–80
LUBRICANT NCODE F2.0	LUBRICANT DESIGNATION	VIS1 KINEMATIC VISCOSITY (cs) AT 100°F	VIS2 KINEMATIC VISCOSITY (cs) AT 210°F	RHO60 DENSITY gm/cu cm AT 15.5°C	G COEFFICIENT OF THERMAL EXPANSION 1/°C	THERMAL CONDUCTIVITY (w/m°C)	AKN EHD HIGH-CONTACT-STRESS FACTOR	FRIC ALLEN FRICTION COEF-FICIENT

IF NCODE IS ≤ 0 READ THIS DATA

INPUT THESE VALUES ONLY WHEN EXECUTING NASA VERSION (SEE APPENDIX F)

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Thermal Data
Card Type T1
One Controlling Card. If no temp. calculation is desired, then leave the card blank.

Field	M	N	TEMP	IPUNCH	IUB	IT1	EP1	IT2	EP2	START	STOP	STEP IN	TTIME	DTIME
Columns	1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–45	46–50	51–55	56–60	61–65	66–70
Format	I5	I5	F5.0	I5	I5	I5	F5.0	I5	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0
Group	GENERAL	GENERAL	GENERAL	GENERAL	GENERAL	GENERAL	STEADY STATE ONLY	STEADY STATE ONLY	STEADY STATE ONLY	TRANSIENT ONLY	TRANSIENT ONLY	TRANSIENT ONLY	TRANSIENT ONLY	TRANSIENT ONLY
Description	Highest Node Number	Highest Unknown Node Number. Same as number of unknown Nodes	Common Initial Temperature (°C) Selected Temperatures can be given different initial values, using card T3.	Punch Flag usually zero. If = 0 Final Temperatures will be punched according to the format of Card T3. Then, they can be read in as initial temperatures in a later run.	Output Flag, usually Zero. If = 1 Bearing output less the rolling element data and a temperature map will be printed every time the shaft bearing Program has been called. If = 2, same as 1 but rolling element output will also be printed.	Maximum No. of calls of the Shaft Bearing Program For steady state analysis the max. no. of thermal iterations to be performed. For transient analysis, a value of N* given by eqn. below should be input.	Absolute accuracy of temperatures (°C)	Iteration Limit, usually left blank If ≠ 0 the intermediate iteration limit. If blank. pre-set limit is used.	Accuracy usually left blank. If ≠ 0 it is the intermediate rel. accuracy	Starting Time (SEC)	Stopping Time (SEC)	Calculation time step. If left blank, the program will calculate a suitable step (SEC)	Time Interval between printed temp. maps. The interval will always be at least equal to the calc. time step. (SEC)	Time interval between calls of shaft-bearing program. Always at least equal to the calc. time step. (SEC)

$$^{*}N = \text{IFIX}\left(\frac{\text{STOP-START}}{\text{DTIME}}\right) + 1$$

Where IFIX indicates that digits to the right of the decimal point are truncated.

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Thermal Data - Bearing Temperatures (^{o}C)

T2, One Card/Brg Use only if no temperature calc. is desired, and then give no more thermal data.

Columns	1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–45	46–50	51–55	56–60	61–65	66–70	71–75	76–80
Format	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0			
Field	Outer Race	Inner Race	Bulk Oil	*Flange #1	*Flange #2	*Flange #3	*Flange #4	Cage	Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

*See Fig. 3.5 for the Flange numbering scheme. Note that a tapered roller bearing Flange is Flange #1.

Ball Bearings are not considered to have Flanges.

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Thermal Data - Individual Initial Temperatures (^{o}C)
T3, As many cards as needed, followed by a blank card

Card columns	1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–45	46–50	51–55	56–60	61–65	66–70	71–75	76–80
Format	I5 (x at col. 5)	F5.0	I5 (x at col. 15)	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5 (x at col. 75)	F5.0
Field	Node Number	Initial Temperature	Etc.												Node Number	Initial Temp.

Thermal Data - Bearing Node Numbers

T4. Include one T4 card for each bearing

Field	Card columns	Mark	Format
Outer Race	1–5	x	I5
Inner Race	6–10	x	I5
Bulk Oil	11–15	x	I5
Flange #1	16–20	x	I5
Flange #2	21–25	x	I f
Flange #3	26–30	x	I5
Flange #4	31–35	x	I5
Cage	36–40	x	I5
Shaft	41–45	x	I5
Inner Ring	46–50		I5
Rolling Element	51–55		I5
Outer Ring	56–60		I5
Housing	61–65		I5
	66–70		
	71–80		

See Fig. 3.5 for the flange numbering scheme. Note that a tapered roller bearing inner ring Flange is Flange #1.

Ball Bearings are not considered to have Flanges.

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Thermal Data - Nodes Where Bearing Frictional Heat is Generated

TS. Include one TS card for each bearing

Columns	1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–45	46–50	51–60	61–70	71–80
Format	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5			
Field	Outer Race: Half the heat is generated at this node.	Outer Race: Half the heat is generated at this node.	Inner Race		Drag		Cage-Roll Element		Cage - Land				

Outer Race note: The above two node numbers can be the same number

TSA. Include a TSA card immediately after the TS card <u>for roller bearings only</u>.

Columns	1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–50	51–60	61–70	71–80
Format	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5	X / I5				
Field	Flange #1		Flange #2		Flange #3		Flange #4					

See Fig. 3.5 for the flange numbering scheme. Note that a tapered roller bearing inner ring flange is Flange #3.

Ball Bearings are not considered to have flanges.

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Thermal Data - Nodes with a constant power source

T6, as many cards as neede , <u>followed by a blank card</u>

Card columns	1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–45	46–50	51–55	56–60	61–65	66–70	71–75	76–80
	x x (cols 4–5)		x x (cols 14–15)		x x (cols 24–25)		x x (cols 34–35)		x x (cols 44–45)		x x (cols 54–55)		x x (cols 64–65)		x x (cols 74–75)	
Format	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0
Field	Node Number	Power (Watts)	Node Number	Power (Watts)	...									...	Node Number	Power

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DATA CARD T7
HEAT TRANSFER COEFFICIENTS   Input one or two cards per coefficient. Use as many T7 cards as required followed by a blank card.

Columns 1–10	11–20	21–30	31–40	41–50	51–60	61–70
I10	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0

INDEX FOR LATER IDENTIFICATION OF HEAT TRANSFER COEFF.	MAGNITUDE OF HEAT TRANSFER COEFFICIENT	EXPONENT USED WITH TEMP. DIFFERENCE	
1-10 (Conduction)	k(w/m-°c)	—	($Q = KA\ \Delta T/L$)
11-20 (Free Conv.)	α_v(w/m²-°c[a])	a(default = 1.25)	($Q = \alpha_v A\ \Delta T^a$)
21-30 (Forced Conv.*)	α_w(w/m²-°c)	—	($Q = \alpha_w A\ \Delta T$)
31-40 (Radiation)	ε	—	
41-50 (Mass Transport)	$\rho c_p \dot{v}$(w/°c)	—	ρ = Density (KG/M³), Cp = Specific Heat (W-SEC/KG-°C), $\dot{v}$ = Flow Rate (M³/SEC)

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*The forced convection coefficient can be calculated internally by the program using one of three options. For options 1 and 3, α_w is calculated by $\alpha_w = \lambda\ Nu/L$ where $Nu = K Re^A Pr^B$ (Re = Reynold's No. = $UL\rho/\eta$, Pr = Prandtl No. = $\eta C_p/\lambda$). For option 2, $\alpha_w = c\eta^D$. The viscosity is a function of temperature for options 2 and 3. To activate this feature, input data for card T7 as shown below and follow immediately with card T7A.

HEAT TRANS. INDEX							
21-30	K	A	B	L (M)	U (M/SEC)	λ (W/m-°c)	(OPTION 1)
21-30	C	D	—	—	—	—	(OPTION 2)
21-30	K	A	B	L (M)	U (M/SEC)	λ (W/m-°c)	(OPTION 3)

DATA CARD T7A   Input only if it is desired to calculate forced convection coefficient internally by the program.

Columns 1–10	11–20	21–30	31–40	41–50	51–60	61–70
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0

η(N-sec/M²)	ρ(KG/M³)	C_p(W-sec/KG-°c)	—	—	—	—	(OPTION 1)
—	—	—	T_L (°c)	η @ T_L (N-sec/M²)	T_H (°c)	η @ T_H (N-sec/M²)	(OPTION 2)
—	ρ(KG/M³)	C_p(W-sec/KG-°c)	T_L (°c)	η @ T_L (N-sec/M²)	T_H (°c)	η @ T_H (N-sec/M²)	(OPTION 3)

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THERMAL DATA – HEAT FLOW PATHS
10, AS MANY CARDS AS NEEDED, MAX. 500, FOLLOWED BY A BLANK CARD
INDEX

Columns	1–10	11–15	16–20	21–30	31–40	41–50	51–80
Format	I10	I5	I5	F10.0	F10.0	F10.0	

INDEX (INDAB = |INDEX|)	NODE I	NODE J				Description
1 ≤ INDAB ≤ 10	NODE I	NODE J	L_1 (mm)	L_2 (mm)	L_3 (mm)	CONDUCTION BETWEEN I AND J. AREA = $2\pi L_1 L_2$. IF INDEX<0 AREA = $L_1 L_2$. DISTANCE I–J = L_3.
11 ≤ INDAB ≤ 20	NODE I	NODE J	L_1	L_2	BLANK	NATURAL CONVECTION BETWEEN I AND J. AREA = $2\pi L_1 L_2$. IF INDEX<0 AREA = $L_1 L_2$.
21 ≤ INDAB ≤ 30	NODE I	NODE J	L_1	L_2	BLANK	FORCED CONVECTION BETWEEN I AND J. AREA AS ABOVE. [illegible]
31 ≤ INDAB ≤ 40	NODE I	NODE J	L_1	L_2	(L_3)	RADIATION BETWEEN I AND J. AREA AS ABOVE. FOR DESCRIPTION OF L_3, SEE USER'S MANUAL.
41 ≤ INDEX ≤ 50	NODE I	NODE J	INDEX OF FLUID FLOW AT NODE J, 41 ≤ INDEX ≤ 50	BLANK	BLANK	FLUID FLOW FROM NODE I TO NODE J. FIRST Index is index of total fluid flow into node i. Second index represents fluid flow ~~from node i to node j~~
INDEX = 51	NODE I	NODE J	BEARING NUMBER NB 1 ≤ NB ≤ 5	CONTACT FLAG 1, INNER RACE CONTACT 2, OUTER RACE CONTACT 3, Flange Contact#1 4, Flange Contact#2 5 Flange Contact#3 6, Flange Contact#4	FACTOR, USUALLY = 1. IF I OR J IS A NODE IN THE OIL BETWEEN THE CONTACTING SURFACES, THE FACTOR IS 0.5.	CONDUCTION THROUGH A BEARING

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Thermal Data - Node heat capacities, include only for transient calculations.

T9, One Card/Node

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Format	x x I10	F10.0	F10.0	F10.0	F10.0	F10.0		
Content	Node Number	$L_1 \cdot L_2 \cdot L_3$ L_1 (mm)	L_2 (mm)	L_3 (mm)	Density (KG/M^3)	Specific Heat WS/KG°C		

Volume at node $= L_1 \cdot L_2 \cdot L_3$ (mm)^3. If it is given without a sign, rotational symmetry is assumed and the volume is multiplied by 2π. If it is negative the volume is not changed.

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LOADING DATA - APPLIED LOADS IF THERE IS NO SHAFT (JUSTBR=1 ON CARD TT2). USE THIS CARD AND OMIT ALL SHAFT DATA.

CARD TYPE LL, ONE CARD FOR SINGLE BEARING ONLY <u>OMIT THIS CARD WHEN USING SHAFT DATA.</u>

Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–80
F10.0	F10.0	F10.0	F10.0	F10.0	
CONCENTRATED FORCE F_Y (NEWTONS)	CONCENTRATED MOMENT M_Z (N.-mm.)	CONCENTRATED FORCE F_Z (NEWTONS)	CONCENTRATED MOMENT M_Y (N.-mm.)	CONCENTRATED AXIAL LOAD F_X (NEWTONS)	

SHAFT DATA - SHAFT GEOMETRY
CARD TYPE S1, AS MANY CARDS AS NEEDED
OMIT ALL SHAFT DATA CARDS IF THERE ARE NO BEARINGS OR IF THERE IS NO SHAFT (JUSTBR=1 ON TITLE CARD TT2)

Columns 1–5	Columns 6–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
1		F 1 0 . 0						
TYPE OF SHAFT CARD THIS CARD TYPE IS 1	BLANK	X-COORDINATE OF SHAFT SECTION (mm)	SHAFT INNER DIA. IMMEDIATELY BEFORE THE SECTION (mm)	SHAFT INNER DIA. IMMEDIATELY AFTER THE SECTION (mm)	SHAFT OUTER DIA. IMMEDIATELY BEFORE THE SECTION (mm)	SHAFT OUTER DIA. IMMEDIATELY AFTER THE SECTION (mm)	SHAFT MODULUS OF ELASTICITY IF LEFT BLANK, STEEL IS ASSUMED $2.04 \times 10^5 N/mm^2$	

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SHAFT DATA   BEARING POSITION AND SUPPORT, X-Y PLANE.
CARD TYPE S2, AS MANY CARDS AS THE NUMBER OF BEARINGS, ARRANGED IN ORDER OF INCREASING X-COORDINATE.

Columns 1–5	Columns 6–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60
2		F10.0				
TYPE OF SHAFT CARD THIS CARD TYPE IS 2	BLANK	X-COORDINATE OF BEARING POSITION IS DEFINED AS THE LOCATION OF POINT A, WHICH IS FOR BALL BEARING, THE INNER RING GROOVE CURVATURE CENTER; AND FOR ROLLER BEARING, THE MIDDLE OF THE INNER RING CONTACT AREA. (mm)	RADIAL MOUNTING ERROR OF THE OUTER RING IN THE Y-DIRECTION (mm)	RADIAL SPRING CHARACTERISTICS OF THE HOUSING IN THE Y-DIRECTION (mm/N) NOT CURRENTLY CONSIDERED IN ANALYSIS	ANGULAR MOUNTING ERROR OF THE OUTER RING IN THE Z-DIRECTION (RADIANS)	ANGULAR SPRING CHARACTERISTICS OF THE HOUSING IN THE Z-DIRECTION (RADIANS/N.mm) NOT CURRENTLY CONSIDERED IN ANALYSIS

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SHAFT DATA   SHAFT LOADING, X-Y PLANE FORCES, Z-MOMENTS
CARD TYPE S3, AS MANY CARDS AS NEEDED, FOLLOWED BY A BLANK CARD

Columns 1–5	Columns 6–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
							F10.0	F10.0
3		F10.0	F10.0	F10.0	F10.0	F10.0		
TYPE OF SHAFT CARD. THIS CARD IS TYPE 3	BLANK	X-COORDINATE OF SHAFT SECTION (mm)	CONCENTRATED FORCE F_Y (N)	CONCENTRATED MOMENT M_Z (N-mm)	DISTRIBUTED LOAD. LOAD INTENSITY IMMEDIATELY BEFORE THE SECTION (N/mm)	DISTRIBUTED LOAD. LOAD INTENSITY IMMEDIATELY AFTER THE SECTION (N/mm)	CONCENTRATED AXIAL LOAD F_X (N)	Y-COORDINATE FOR THE LINE OF ACTION OF THE AXIAL LOAD. (mm)

Shaft Data - Bearing position and support, X-Z plane

S2, as many cards as needed. Card needed only if any data is different from X-Y plane   Be sure to include bearing number in column 10.

Card columns	Entry	Description
1–5	2	Type of shaft card This card type is 2
6–10	I5 (x in column 10)	Bearing number
11–20		Blank
21–30	F10.0	Radial Mounting error of the outer ring in the Z-direction (mm)
31–40	F10.0	Radial spring* characteristics of the housing in the Z-direction (mm/N.)
41–50	F10.0	Angular mounting error of the outer ring in the Y-direction. (radians)
51–60	F10.0	Angular spring* characteristics of the housing in the Y-direction (radians/Nmm.)
61–70		
71–80		

* Not currently considered in analysis

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Shaft Data - Shaft Loading, X-Z plane forces, Y-moments.

S3, as many cards as needed, followed by a blank card. This card must be included if a bearing analysis is to be performed.

Columns 1–5	6–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
x								
3		F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Type of shaft card. This card is type 3	Blank	X-coordinate of shaft section. (mm)	Concentrated force Fz. (N)	Concentrated moment My. N-mm)	Distributed load. Load intensity immediately before the section, (N/mm)	Distributed load. Load intensity immediately after the section, (N/mm)	Concentrated axial load. (N)	Z-coordinate for the line of action of the axial load (mm)

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APPENDIX D

SKF 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 second set of examples display program output for a system in which an input pinion gear load is supported by a flanged cylindrical roller 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.
- L_{10} 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].

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SKF

2 TAPERED BEARINGS

LEVEL 1

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THIS DATA SET CONTAINS 2 BEARINGS

BEARING NO. (1) - TAPERED ROLLER BEARING

BEARING NO. (2) - TAPERED ROLLER BEARING

SOLUTION LEVEL = 1

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UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	16	.000	.000	.000	-20.000	36000.	0.
2	10	.000	.000	.000	10.800	36000.	0.

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	INNER RING LAND RIDING	.230000	1.9050	39.0500	.328	.100000
2	INNER RING LAND RIDING	.100000	.9650	26.1900	2.120	.100000

S T E E L D A T A

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	STEEL INNER RING	5.000	STEEL OUTER RING	5.000
2	STEEL INNER RING	5.000	STEEL OUTER RING	5.000

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R O L L I N G   E L E M E N T   D A T A

BEARING NUMBER (1)   TYPE - TAPERED ROLLER BEARING

ROLLER DIAMETER	ROLLER LENGTH	ROLLER PROFILE FLAT LENGTH	ROLLER CROWN RADIUS	ROLLER END RADIUS	ROLLER INCLUDED ANGLE	NUMBER OF AXIAL LAMINAE
7.9400	16.3200	.0000	4500.0000	87.3000	4.9500	10

EFFECTIVE LENGTH	OUTER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	EFFECTIVE LENGTH	INNER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	INNER RING FLANGE ANGLE	BEARING AXIAL PLAY
15.0000	.0000	.0000	15.6000	.0000	.0000	15.5500	-.0034

BEARING NUMBER (2)   TYPE - TAPERED ROLLER BEARING

ROLLER DIAMETER	ROLLER LENGTH	ROLLER PROFILE FLAT LENGTH	ROLLER CROWN RADIUS	ROLLER END RADIUS	ROLLER INCLUDED ANGLE	NUMBER OF AXIAL LAMINAE
5.7150	9.2200	.0000	12700.0000	95.3870	3.4330	10

EFFECTIVE LENGTH	OUTER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	EFFECTIVE LENGTH	INNER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	INNER RING FLANGE ANGLE	BEARING AXIAL PLAY
7.1170	.0000	.0000	7.1170	.0000	.0000	7.7968	-.0083

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S U R F A C E D A T A

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	.13	.13	.08	2.000	2.000	2.000
2	.13	.13	.08	2.000	2.000	2.000

BEARING NUMBER	CLA ROUGHNESS OUTER FL.	CLA ROUGHNESS INNER FL.	CLA ROUGHNESS ROLL. ELM. END	RMS ASPERITY SLOPE OUTER FL.	RMS ASPERITY SLOPE INNER FL.	RMS ASPERITY SLOPE ROLL. ELM. END
1	.13	.13	.08	2.000	2.000	2.000
2	.13	.13	.08	2.000	2.000	2.000

L U B R I C A N T D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	SANTOTRAC 50	33.60	5.61	.8890	4.10-04	.100
2	SANTOTRAC 50	33.60	5.61	.8890	4.10-04	.100

L 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 NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	2.00	.3000-03	.2000-03	.10
2	2.00	.3000-03	.2000-03	.10

BEARING NUMBER	CAGE FRICTION COEFFICIENT	FILM REPLENISHMENT LAYER THICKNESS	FILM REPLENISHMENT LAYER THICKNESS	FLG-R.E. FRICTION COEFFICIENT
1	.10	.4000-03	.4000-03	.10
2	.10	.4000-03	.4000-03	.10

UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

GIVEN TEMPERATURES

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BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	100.00	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00
2	100.00	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00

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SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD. PLANE X - Y.

7 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 2 BEARINGS. MODULUS OF ELASTICITY = 2.041+05

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT
1	.0	16.7	16.7	33.3	33.3				
2	19.5	16.7	16.7	33.3	33.3				
3	33.4	16.7	16.7	33.3	52.4				
4	43.5	16.7	16.7	46.9	46.9	650.4	-50190.0		
5	59.7	16.7	3.3	41.4	17.5				
6	74.5	9.2	9.2	17.5	17.5				
7	108.0	9.2	9.2	17.5	17.5				

	BEARING SEAT POS.ERR	BEARING SEAT DEFL/FOR	BEARING SEAT ANG.ERR	BEARING SEAT DEFL/MOM
1				
2	.000	0.00	.0000	0.00
3				
4				
5				
6	.000	0.00	.0000	0.00
7				

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SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X - Z.

4 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 2 BEARINGS. MODULUS OF ELASTICITY = 2.041+05

THRUST LOAD = -2.147+03

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT
1	.0	16.7	16.7	33.3	33.3				
2	19.5	16.7	16.7	33.3	33.3				
3	39.1	16.7	16.7	33.3	52.4				
4	49.5	16.7	16.7	46.9	46.9	-2705.2			
5	59.7	16.7	9.5	41.4	17.5				
6	79.5	9.2	9.2	17.5	17.5				
7	108.0	9.2	9.2	17.5	17.5				

	BEARING SEAT POS.ERR	DEFL/FOR	ANG.ERR	DEFL/MOM
1				
2	.000	0.00	.0000	0.00
3				
4				
5				
6	.000	0.00	.0000	0.00
7				

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B E A R I N G S Y S T E M O U T P U T METRIC UNITS

LINEAR (MM) AND ANGULAR (RADIANS) DEFLECTIONS — REACTION FORCES (N) AND MOMENTS (MM-N)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	3.373-03	7.256-03	-5.537-03	-2.030-05	-1.844-04	-2.689+03	1.463+03	-1.341+03	1.079+04	1.148+04
2	3.373-03	-2.381-03	-2.946-03	-1.276-04	-1.291-04	541.	-812.	-1.366+03	-4.712+03	2.466+03

	FATIGUE LIFE (HOURS)			H/SIGMA		LUBE-LIFE FACTOR		MATERIAL FACTOR	
BRG.	O. RACE	I. RACE	BEARING	O. RACE	I. RACE	O. RACE	I. RACE	O. RACE	I. RACE
1	5.088+04	5.936+04	2.958+04	1.97	1.79	2.09	1.95	5.00	5.00
2	2.698+04	4.014+03	3.637+03	1.32	1.14	1.27	.810	5.00	5.00

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	100.00	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00
2	100.00	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00

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B E A R I N G S Y S T E M O U T P U T METRIC UNITS

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

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	1.066+03	0.000	414.	172.	1.154+03	1.155-07	18.6	2.923+03	49.
2	297.	0.000	112.	34.7	45.7	1.863-08	2.00	402.	107.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W/DEG.C)	
1	.375	.342	.993	.980	.819	.819	.420	.232	1.87	.891
2	.252	.218	.998	.996	.875	.872	.443	.274	1.77	1.55

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B E A R I N G S Y S T E M O U T P U T METRIC UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1	OUTER	100.000	.8544	5.488	4.688	.1313-01
	INNER	100.000	.8544	5.488	4.688	.1313-01
	BULK	100.000	.8544	5.488	4.688	.1313-01
	FLANGE	100.000	.8544	5.488	4.688	.1313-01

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 2	OUTER	100.000	.8544	5.488	4.688	.1313-01
	INNER	100.000	.8544	5.488	4.688	.1313-01
	BULK	100.000	.8544	5.488	4.688	.1313-01
	FLANGE	100.000	.8544	5.488	4.688	.1313-01

C A G E D A T A METRIC UNITS (CAGE HAS ONE DEGREE OF FREEDOM)

CAGE RAIL - RING LAND DATA CAGE SPEED DATA

BRG.	TORQUE (MM-N)	HEAT RATE (WATTS)	SEP.FORCE (NEWTONS)	ECCENTRICITY RATIO	EPICYCLIC SPEED (RAD/SEC)	EPICYCLIC SPEED (RPM)	CALCULATED SPEED (RAD/SEC)	CALCULATED SPEED (RPM)	CALC/EPIC RATIO	CAGE/SHAFT RATIO
1	8.68	18.6	.591	.777	1.627+03	1.553+04	1.627+03	1.553+04	1.00	.431
2	.893	2.00	.494	.983	1.531+03	1.462+04	1.531+03	1.462+04	1.00	.406

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R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)					SPEED VECTOR ANGLES (DEGREES)	
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-12873.152	1.373	.000	12873.152	1626.701	179.99	180.00
22.50	-12873.152	1.192	.000	12873.152	1626.701	179.99	180.00
45.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
67.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
90.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
112.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
135.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
157.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
180.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
202.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
225.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
247.50	-12873.152	.452	.000	12873.152	1626.701	180.00	180.00
270.00	-12873.152	.673	.000	12873.152	1626.701	180.00	180.00
292.50	-12873.152	.882	.000	12873.152	1626.701	180.00	180.00
315.00	-12873.152	1.259	.000	12873.152	1626.701	179.99	180.00
337.50	-12873.152	1.320	.000	12873.152	1626.701	179.99	180.00

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R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-.000	673.013	530.251	913.591	818.346	.0417	.0610	-20.00	-15.05
22.50	-.000	391.789	49.904	757.919	421.755	.0374	.0441	-20.00	-15.05
45.00	-.000	118.884	.000	479.254	.000	.0236	.0000	-20.00	-15.05
67.50	-.000	115.048	.000	473.308	.000	.0233	.0000	-20.00	-15.05
90.00	-.000	111.228	.000	467.301	.000	.0230	.0000	-20.00	-15.05
112.50	-.000	107.419	.000	461.231	.000	.0227	.0000	-20.00	-15.05
135.00	-.000	103.628	.000	455.097	.000	.0224	.0000	-20.00	-15.05
157.50	-.000	99.852	.000	448.897	.000	.0221	.0000	-20.00	-15.05
180.00	.000	96.093	.000	442.627	.000	.0218	.0000	-20.00	-15.05
202.50	.000	92.352	.000	436.285	.000	.0215	.0000	-20.00	-15.05
225.00	.000	88.629	.000	429.868	.000	.0212	.0000	-20.00	-15.05
247.50	.000	355.093	13.326	730.303	275.270	.0365	.0272	-20.00	-15.05
270.00	.000	612.464	269.930	883.344	752.198	.0409	.0551	-20.00	-15.05
292.50	.000	872.448	529.116	998.219	959.391	.0437	.0647	-20.00	-15.05
315.00	.000	987.295	643.600	1044.360	1031.355	.0444	.0664	-20.00	-15.05
337.50	.000	910.181	566.723	1015.927	987.680	.0439	.0653	-20.00	-15.05

HERTZ STRESS ACROSS THE OUTER RACEWAY-ROLLER PROFILE

.0000	.3719+05	.9796+05	.1267+06	.1413+06	.1448+06	.1373+06	.1166+06	.7392+05	.0000
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

ACROSS THE INNER RACEWAY-ROLLER PROFILE

.0000	.0000	.8332+05	.1203+06	.1370+06	.1331+06	.1265+06	.9355+05	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

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R O L L E R - F L A N G E O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH ANGLE (DEG.)	CONTACT LOAD (N)	HERTZ STRESS (N/MM**2)	MAJ-SEMI AXIS (MM)	MIN-SEMI AXIS (MM)	ROLLING VELOCITY (M/S)	SLIDING VELOCITY (M/S)	EHD FILM THICKNESS (MICRONS)	CONTACT HEAT RATE (WATTS)	HYDRO-DYN HEAT RATE (WATTS)
.00	149.913	349.074	.472	.435	45.940	14.4187	.4574	9.17	2.78
22.50	125.644	329.118	.445	.410	45.940	14.4187	.4641	7.34	2.75
45.00	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
67.50	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
90.00	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
112.50	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
135.00	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
157.50	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
180.00	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
202.50	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
225.00	120.686	324.730	.439	.405	45.940	14.4187	.4657	6.97	2.74
247.50	122.466	326.320	.441	.407	45.940	14.4187	.4651	7.11	2.74
270.00	144.680	344.964	.466	.430	45.940	14.4187	.4588	8.77	2.77
292.50	167.120	361.949	.489	.451	45.940	14.4187	.4555	10.51	2.79
315.00	177.033	368.968	.498	.460	45.940	14.4187	.4512	11.30	2.80
337.50	170.379	364.287	.492	.454	45.940	14.4187	.4526	10.77	2.80

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

FLANGE NO.	EHD FILM THICKNESS (MICRONS)	FILM REDUCTION FACTORS STARVATION	FILM REDUCTION FACTORS THERMAL	H/SIGMA	EFFECTIVE TRACTION COEFFICIENT	QASP/QTOT	HEAT GENERATION RATES (WATTS) CONTACT	HEAT GENERATION RATES (WATTS) INLET	TOT FLNG-ROL CONDUCTIVITY (W/DEG C)
3	.457	.520	.823	2.397	.004	.044	9.17	2.78	1.25

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MAB - L411700 STRADDLE

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)					SPEED VECTOR ANGLES (DEGREES)	
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-9575.741	.000	.000	9576.741	1531.482	180.00	180.00
36.00	-9576.741	.000	.000	9576.741	1531.482	180.00	180.00
72.00	-9575.741	.000	.000	9576.741	1531.482	180.00	180.00
108.00	-9576.741	.000	.000	9576.741	1531.482	180.00	180.00
144.00	-9576.741	-1.229	.000	9576.741	1531.482	-179.99	180.00
180.00	-9576.741	-1.113	.000	9576.741	1531.482	-179.99	180.00
216.00	-9576.741	-.728	.000	9576.741	1531.482	-180.00	180.00
252.00	-9576.741	-.213	.000	9576.741	1531.482	-180.00	180.00
288.00	-9576.741	.235	.000	9576.741	1531.482	180.00	180.00
324.00	-9576.741	.426	.000	9576.741	1531.482	180.00	180.00

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MBB - LM11700 STRADDLE

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNEP	OUTER	INNER
.00	.001	46.774	.000	344.656	.000	.1684	.0000	10.80	7.37
36.00	.000	46.658	.000	344.374	.000	.1684	.0000	10.80	7.37
72.00	.000	46.541	.000	344.092	.000	.1683	.0000	10.80	7.37
108.00	.000	46.425	.000	343.809	.000	.1683	.0000	10.80	7.37
144.00	.000	172.709	116.660	602.404	589.984	.1937	.2621	10.80	7.37
180.00	-.000	437.694	381.269	882.851	973.613	.1916	.2424	10.80	7.37
216.00	-.000	630.152	573.447	1028.530	1174.928	.1859	.2320	10.80	7.37
252.00	-.000	658.679	601.948	1039.684	1209.848	.1822	.2273	10.80	7.37
288.00	-.000	509.939	453.410	918.575	1072.586	.1795	.2276	10.80	7.37
324.00	-.001	253.949	197.752	671.142	757.055	.1802	.2390	10.80	7.37

HERTZ STRESS ACROSS THE OUTER RACEWAY-ROLLER PROFILE

.1211+06	.1260+06	.1297+06	.1320+06	.1332+06	.1332+06	.1320+06	.1295+06	.1256+06	.1202+06
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

ACROSS THE INNER RACEWAY-ROLLER PROFILE

.1513+06	.1542+06	.1556+06	.1555+06	.1539+06	.1508+06	.1460+06	.1393+06	.1305+06	.1131+06
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

MRR - LM11700 STRADDLE

R O L L E R - F L A N G E O U T P U T FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH ANGLE (DEG.)	CONTACT LOAD (N)	HERTZ STRESS (N/MM**2)	MAJ-SEMI AXIS (MM)	MIN-SEMI AXIS (MM)	ROLLING VELOCITY (M/S)	SLIDING VELOCITY (M/S)	EHD FILM THICKNESS (MICRO M)	CONTACT HEAT RATE (WATTS)	HYDRO-DYN HEAT RATE (WATTS)
.00	10.573	133.425	.206	.183	24.746	8.4400	.4557	.94	.94
36.00	10.573	133.425	.206	.183	24.746	8.4400	.4557	.94	.94
72.00	10.573	133.425	.206	.183	24.746	8.4400	.4557	.94	.94
108.00	10.573	133.425	.206	.183	24.746	8.4400	.4557	.94	.94
144.00	17.594	158.108	.244	.217	24.746	8.4400	.4388	1.69	.96
180.00	33.453	195.876	.303	.269	24.746	8.4400	.4184	3.49	.99
216.00	44.973	216.183	.334	.297	24.746	8.4400	.4093	4.83	1.01
252.00	46.680	218.883	.338	.301	24.746	8.4400	.4082	5.03	1.01
288.00	37.773	203.968	.315	.280	24.746	8.4400	.4147	3.99	1.00
324.00	22.447	171.484	.265	.236	24.746	8.4400	.4310	2.23	.97

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

FLANGE NO.	EHD FILM THICKNESS (MICRONS)	FILM REDUCTION FACTORS STARVATION	FILM REDUCTION FACTORS THERMAL	H/SIGMA	EFFECTIVE TRACTION COEFFICIENT	QASP/QTOT	HEAT GENERATION RATES (WATTS) CONTACT	HEAT GENERATION RATES (WATTS) INLET	TOT FLNG-ROL CONDUCTIVITY (W/DEG C)
3	.456	.614	.878	2.388	.011	.128	.94	.94	.30

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NASA

2 TAPERED BEARINGS

LEVEL 1

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     M38 - LM11700 STRADDLE


          THIS DATA SET CONTAINS  2  BEARINGS
           BEARING NO. (1) -   TAPERED ROLLER BEARING
           BEARING NO. (2) -   TAPERED ROLLER BEARING



          SOLUTION LEVEL = 1
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MRB - LM11700 STRADDLE

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	16	.000	.000	.000	-20.000	36000.	0.
2	10	.000	.000	.000	10.000	36000.	0.

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	INNER RING LAND RIDING	.230000	1.9050	39.0500	.328	.100000
2	INNER RING LAND RIDING	.100000	.9650	26.1900	2.120	.100000

S T E E L D A T A

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	STEEL INNER RING	5.000	STEEL OUTER RING	5.000
2	STEEL INNER RING	5.000	STEEL OUTER RING	5.000

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R O L L I N G E L E M E N T D A T A

BEARING NUMBER (1) TYPE - TAPERED ROLLER BEARING

ROLLER DIAMETER	ROLLER LENGTH	ROLLER PROFILE FLAT LENGTH	ROLLER CROWN RADIUS	ROLLER END RADIUS	ROLLER INCLUDED ANGLE	NUMBER OF AXIAL LAMINAE
7.9400	16.3200	.0000	4500.0000	87.3000	4.9500	10

EFFECTIVE LENGTH	OUTER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	EFFECTIVE LENGTH	INNER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	INNER RING FLANGE ANGLE	BEARING AXIAL PLAY
15.0000	.0000	.0000	15.0000	.0000	.0000	15.5500	-.0034

BEARING NUMBER (2) TYPE - TAPERED ROLLER BEARING

ROLLER DIAMETER	ROLLER LENGTH	ROLLER PROFILE FLAT LENGTH	ROLLER CROWN RADIUS	ROLLER END RADIUS	ROLLER INCLUDED ANGLE	NUMBER OF AXIAL LAMINAE
5.7150	9.2200	.0000	12700.0000	95.3870	3.4330	10

EFFECTIVE LENGTH	OUTER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	EFFECTIVE LENGTH	INNER RACEWAY CROWN RADIUS	LARGE END CORNER RELIEF	INNER RING FLANGE ANGLE	BEARING AXIAL PLAY
7.1170	.0000	.0000	7.1170	.0000	.0000	7.7960	-.0083

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S U R F A C E   D A T A

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	.13	.13	.08	2.000	2.000	2.000
2	.13	.13	.08	2.000	2.000	2.000

BEARING NUMBER	CLA ROUGHNESS OUTER FL.	CLA ROUGHNESS INNER FL.	CLA ROUGHNESS ROLL. ELM. END	RMS ASPERITY SLOPE OUTER FL.	RMS ASPERITY SLOPE INNER FL.	RMS ASPERITY SLOPE ROLL. ELM. END
1	.13	.13	.08	2.000	2.000	2.000
2	.13	.13	.08	2.000	2.000	2.000

L U B R I C A N T   D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	SANTOTRAC 50	33.60	5.61	.8890	4.10-04	.100
2	SANTOTRAC 50	33.60	5.61	.8890	4.10-04	.100

L 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 NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	2.00	.3000-03	.2000-03	.10
2	2.00	.3000-03	.2000-03	.10

BEARING NUMBER	CAGE FRICTION COEFFICIENT	FILM REPLENISHMENT LAYER THICKNESS		FLG-R.E. FRICTION COEFFICIENT
1	.10	.4000-03	.4000-03	.10
2	.10	.4000-03	.4000-03	.10

UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

GIVEN TEMPERATURES

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BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	100.00	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00
2	100.00	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00

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NRR - LMII70D STRADDLE

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X - Y.

7 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 2 BEARINGS, MODULUS OF ELASTICITY = 2.041+05

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT
1	.0	16.7	16.7	33.3	33.3				
2	19.5	16.7	16.7	33.3	33.3				
3	39.4	16.7	16.7	33.3	52.4				
4	47.5	16.7	16.7	46.3	46.?	650.4	-50190.0		
5	59.7	16.7	3.3	41.4	17.5				
6	74.5	9.2	9.2	17.5	17.5				
7	108.0	9.2	9.2	17.5	17.5				

	BEARING SEAT POS.ERR	DEFL/FOR	ANG.ERR	DEFL/MOM
1				
2	.000	0.00	.0000	0.00
3				
4				
5				
6	.000	0.00	.0000	0.00
7				

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MBR - LM11700 STRADDLE

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X - Z.

4 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 2 BEARINGS, MODULUS OF ELASTICITY = 2.041+05

THRUST LOAD = -2.147+03

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT		BEARING SEAT POS.ERR	BEARING SEAT DEFL/FOR	BEARING SEAT ANG.ERR	BEARING SEAT DEFL/MOM
1	.0	16.7	16.7	33.3	33.3					1				
2	19.5	16.7	16.7	33.3	33.3					2	.000	0.00	.0000	0.00
3	39.4	16.7	16.7	33.3	52.4					3				
4	49.5	16.7	16.7	46.9	46.9	-2705.2				4				
5	59.7	16.7	9.3	41.4	17.5					5				
6	74.5	9.2	9.2	17.5	17.5					6	.000	0.00	.0000	0.00
7	108.0	9.2	9.2	17.5	17.5					7				

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MOB - L111700 STRADDLE

B E A R I N G S Y S T E M O U T P U T METRIC UNITS

LINEAR (M) AND ANGULAR (RADIANS) DEFLECTIONS — REACTION FORCES (N) AND MOMENTS (MM-N)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	5.373-05	7.256-03	-5.537-03	-2.030-05	-1.844-04	-2.689+03	1.463+03	-1.341+03	1.079+04	1.148+04
2	3.373-03	-2.381-03	-2.946-03	-1.276-04	-1.291-04	541.	-812.	-1.366+03	-4.712+03	2.[illegible]+03

	FATIGUE LIFE (HOURS)			H/SIGMA		LUBE-LIFE FACTOR		MATERIAL FACTOR	
BRG.	O. RACE	I. RACE	BEARING	O. RACE	I. RACE	O. RACE	I. RACE	O. RACE	I. RACE
1	5.[illegible]1+04	6.371+04	3.447+04	[illegible].59	2.30	2.42	2.29	5.00	5.00
2	4.142+04	9.028+03	7.048+03	1.79	1.49	1.95	1.62	5.00	5.00

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	100.20	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00
2	100.00	100.00	100.00	100.00	.00	.00	.00	100.00	100.00	100.00	100.00	100.00	100.00

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438 - LM11700 STRADDLE

B E A R I N G S Y S T E M O U T P U T   METRIC UNITS

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

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	1.020+03	0.000	383.	1.136+03	1.154+03	1.155-07	18.6	3.711+03	984.
2	133.	0.000	106.	28.1	45.7	1.863-08	2.00	381.	101.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W/DEG.C)	
1	.445	.439	.393	.990	.819	.819	.420	.232	1.17	.503
2	.341	.384	.998	.996	.875	.872	.443	.274	.746	.569

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B E A R I N G S Y S T OUTPUT METRIC UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1	OUTER	100.000	.8544	5.488	4.688	.1313-01
	INNER	100.000	.8544	5.488	4.688	.1313-01
	BULK	100.000	.8544	5.488	4.688	.1313-01
	FLANGE	[illegible]00.000	.8544	5.488	4.688	.1313-01

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 2	OUTER	[illegible]	.8544	5.488	4.688	.1313-01
	INNER	[illegible]	.8544	5.488	4.688	.1313-01
	BULK	[illegible]	.8544	5.488	4.688	.1313-01
	FLANGE	[illegible]	.8544	5.488	4.688	.1313-01

C A G E D A T A METRIC UNITS

CAGE RAIL - RING LAND DATA

CAGE HAS ONE DEGREE OF FREEDOM)

CAGE SPEED DATA

BRG.	TORQUE (MM-N)	HEAT RATE (WATTS)	SEP. FORCE (NEWTONS)	ECCENTRICITY RATIO	EPICYCLIC SPEED (RAD/SEC)	EPICYCLIC SPEED (RPM)	CALCULATED SPEED (RAD/SEC)	CALCULATED SPEED (RPM)	CALC/EPIC RATIO	CAGE/SHAFT RATIO
1	8.68	8.5	.58[illegible]	.777	1.627+03	1.553+04	1.627+03	1.553+04	1.00	.431
2	.893	[illegible]00	.499	.989	1.531+03	1.462+04	1.531+03	1.462+04	1.00	.406

D:30

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MOD - LM11700 STRADDLE

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)					SPEED VECTOR ANGLES (DEGREES)	
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-12873.152	1.373	.000	12873.152	1626.701	179.99	180.00
22.50	-12873.152	1.192	.000	12873.152	1626.701	179.99	180.00
45.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
67.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
90.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
112.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
135.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
157.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
180.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
202.50	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
225.00	-12873.152	.000	.000	12873.152	1626.701	180.00	180.00
247.50	-12873.152	.352	.000	12873.152	1626.701	180.00	180.00
270.00	-12873.152	.673	.000	12873.152	1626.701	180.00	180.00
292.50	-12873.152	.992	.000	12873.152	1626.701	180.00	180.00
315.00	-12873.152	1.259	.000	12873.152	1626.701	179.99	180.00
337.50	-12873.152	1.320	.000	12873.152	1626.701	179.99	180.00

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498 - LM11700 STRADDLE

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-.000	673.013	330.251	913.591	818.346	.0000	.0000	-20.00	-15.05
22.50	-.000	391.789	43.904	757.919	421.755	.0000	.0000	-20.00	-15.05
45.00	-.000	119.884	.000	479.254	.000	.0000	.0000	-20.00	-15.05
67.50	-.000	115.048	.000	473.308	.000	.0000	.0000	-20.00	-15.05
90.00	-.000	111.226	.000	467.301	.000	.0000	.0000	-20.00	-15.05
112.50	-.000	107.419	.000	461.231	.000	.0000	.0000	-20.00	-15.05
135.00	-.000	103.629	.000	455.097	.000	.0000	.0000	-20.00	-15.05
157.50	-.000	99.352	.000	448.897	.000	.0000	.0000	-20.00	-15.05
180.00	.000	96.093	.000	442.627	.000	.0000	.0000	-20.00	-15.05
202.50	.000	92.352	.000	436.285	.000	.0000	.0000	-20.00	-15.05
225.00	.000	88.629	.000	429.868	.000	.0000	.0000	-20.00	-15.05
247.50	.000	355.093	13.326	730.303	275.270	.0000	.0000	-20.00	-15.05
270.00	.000	612.464	269.930	883.344	752.198	.0000	.0000	-20.00	-15.05
292.50	.000	872.448	529.116	998.219	959.391	.0000	.0000	-20.00	-15.05
315.00	.006	987.295	643.609	1044.360	1031.355	.0000	.0000	-20.00	-15.05
337.50	.000	910.181	566.723	1015.927	987.680	.0000	.0000	-20.00	-15.05

HERTZ STRESS ACROSS THE OUTER RACEWAY-ROLLER PROFILE

.0000	.3719+05	.9796+05	.1267+06	.1413+06	.1448+06	.1373+06	.1166+06	.7332+05	.0000
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

ACROSS THE INNER RACEWAY-ROLLER PROFILE

.0000	.0000	.8332+05	.1203+06	.1370+06	.1391+06	.1265+06	.9355+05	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

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4RB - L411700 STRADDLE

R O L L E R - F L A N G E O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH ANGLE (DEG.)	CONTACT LOAD (N)	HERTZ STRESS (N/MM**2)	MAJ-SEMI AXIS (MM)	MIN-SEMI AXIS (MM)	ROLLING VELOCITY (M/S)	SLIDING VELOCITY (M/S)	EHD FILM THICKNESS (MICRO M)	CONTACT HEAT RATE (WATTS)	HYDRO-DYN HEAT RATE (WATTS)
.00	149.913	349.074	.472	.435	45.940	14.4187	.7650	79.58	5.52
22.50	125.644	329.118	.445	.410	45.940	14.4187	.7715	58.00	5.30
45.00	120.686	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
67.50	120.686	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
90.00	120.686	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
112.50	120.685	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
135.00	120.686	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
157.50	120.686	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
180.00	120.686	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
202.50	120.636	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
225.00	120.686	324.730	.439	.405	45.940	14.4187	.7730	53.77	5.24
247.50	122.466	326.320	.441	.407	45.940	14.4187	.7725	55.27	5.26
270.00	144.680	344.764	.466	.430	45.940	14.4187	.7665	75.14	5.48
292.50	157.120	351.349	.489	.451	45.940	14.4187	.7610	95.20	5.67
315.00	177.033	368.968	.499	.460	45.940	14.4187	.7590	104.94	5.75
337.50	170.379	364.267	.492	.454	45.940	14.4187	.7603	98.34	5.69

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

FLANGE NO.	EHD FILM THICKNESS (MICRONS)	FILM REDUCTION FACTORS STARVATION	FILM REDUCTION FACTORS THERMAL	H/SIGMA	EFFECTIVE TRACTION COEFFICIENT	QASP/QTOT	HEAT GENERATION RATES (WATTS) CONTACT	HEAT GENERATION RATES (WATTS) INLET	TOT FLNG-ROL CONDUCTIVITY (W/DEG C)
3	.765	.521	.829	4.009	.037	.000	79.58	5.52	.69

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M28 - LM11700 STRADDLE

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R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)					SPEED VECTOR ANGLES (DEGREES)	
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-9576.741	.000	.000	9576.741	1531.482	180.00	180.00
36.00	-9576.741	.000	.000	9576.741	1531.482	180.00	180.00
72.00	-9576.741	.000	.000	9576.741	1531.482	180.00	180.00
108.00	-9576.741	.000	.000	9576.741	1531.482	180.00	180.00
144.00	-9576.741	-1.229	.000	9576.741	1531.482	-179.99	180.00
180.00	-9576.741	-1.113	.000	9576.741	1531.482	-179.99	180.00
216.00	-9576.741	-.728	.000	9576.741	1531.482	-180.00	180.00
252.00	-9576.741	-.213	.000	9576.741	1531.482	-180.00	180.00
288.00	-9576.741	.235	.000	9576.741	1531.482	180.00	180.00
324.00	-9576.741	.426	.000	9576.741	1531.482	180.00	180.00

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MHH - LM11700 STRADDLE

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	.001	46.774	.000	344.656	.000	.0000	.0000	10.80	7.37
36.00	.000	46.658	.000	344.374	.000	.0000	.0000	10.80	7.37
72.00	.000	46.541	.000	344.092	.000	.0000	.0000	10.80	7.37
108.00	.000	46.425	.000	343.809	.000	.0000	.0000	10.80	7.37
144.00	.000	172.719	116.660	602.404	589.984	.0000	.0000	10.80	7.37
180.00	-.000	437.694	391.269	882.851	973.613	.0000	.0000	10.80	7.37
216.00	-.000	630.152	573.447	1028.530	1174.928	.0000	.0000	10.80	7.37
252.00	-.000	658.679	601.948	1039.684	1209.848	.0000	.0000	10.80	7.37
288.00	-.000	509.939	453.410	918.575	1072.586	.0000	.0000	10.80	7.37
324.00	-.001	253.949	197.752	671.142	757.055	.0000	.0000	10.80	7.37

HERTZ STRESS ACROSS THE OUTER RACEWAY-ROLLER PROFILE

.1211+06	.1250+06	.1297+06	.1320+06	.1332+06	.1332+06	.1320+06	.1295+06	.1256+06	.1202+06
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

ACROSS THE INNER RACEWAY-ROLLER PROFILE

.1513+06	.1542+06	.1556+06	.1555+06	.1539+06	.1508+06	.1460+06	.1393+06	.1305+06	.1191+06
.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

MBB - LM11703 STRADDLE

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R O L L E R - F L A N G E O U T P U T FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH ANGLE (DEG.)	CONTACT LOAD (N)	HERTZ STRESS (N/MM**2)	MAJ-SEMI AXIS (MM)	MIN-SEMI AXIS (MM)	ROLLING VELOCITY (M/S)	SLIDING VELOCITY (M/S)	EHD FILM THICKNESS (MICRO M)	CONTACT HEAT RATE (WATTS)	HYDRO-DYN HEAT RATE (WATTS)
.00	10.573	133.425	.206	.183	24.746	8.4400	.7769	.51	1.08
36.00	10.573	133.425	.206	.183	24.746	8.4400	.7769	.51	1.08
72.00	10.573	133.425	.206	.183	24.746	8.4400	.7769	.51	1.08
108.00	10.573	133.425	.206	.183	24.746	8.4400	.7769	.51	1.08
144.00	17.534	158.108	.244	.217	24.746	8.4400	.7562	.94	1.19
180.00	33.453	195.876	.303	.269	24.746	8.4400	.7322	2.16	1.36
216.00	44.373	216.183	.334	.297	24.746	8.4400	.7220	3.27	1.45
252.00	46.680	218.883	.338	.301	24.746	8.4400	.7207	3.45	1.46
288.00	37.773	203.968	.315	.280	24.746	8.4400	.7279	2.55	1.39
324.00	22.447	171.984	.265	.236	24.746	8.4400	.7468	1.27	1.25

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

FLANGE NO.	EHD FILM THICKNESS (MICRONS)	FILM REDUCTION FACTORS STARVATION	FILM REDUCTION FACTORS THERMAL	H/SIGMA	EFFECTIVE TRACTION COEFFICIENT	QASP/QTOT	HEAT GENERATION RATES (WATTS) CONTACT	HEAT GENERATION RATES (WATTS) INLET	TOT FLNG-ROL CONDUCTIVITY (W/DEG C)
3	.777	.619	.879	4.072	.006	.000	.51	1.08	.15

SKF

1 CYLINDRICAL
2 BALL BEARINGS

LEVEL 1

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

THIS DATA SET CONTAINS 3 BEARINGS

BEARING NO. (1) - CYLINDRICAL ROLLER BEARING , FLANGE IS TYPE NO. 1.

BEARING NO. (2) - BALL BEARING

BEARING NO. (3) - BALL BEARING

SOLUTION LEVEL = 1

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	10	.000	32.100	.025	.000	35963.	0.
2	11	.000	38.500	-.006	25.000	35963.	0.
3	11	.000	38.500	-.006	25.000	35963.	0.

D:39

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	INNER RING LAND RIDING	.150000	2.0000	24.0000	.150	.013000
2	INNER RING LAND RIDING	.150000	2.0000	24.0000	.150	.013000
3	INNER RING LAND RIDING	.150000	2.0000	24.0000	.150	.013000

S T E E L D A T A

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	M-50 STEEL	5.000	M-50 STEEL	5.000
2	M-50 STEEL	5.000	M-50 STEEL	5.000
3	M-50 STEEL	5.000	M-50 STEEL	5.000

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R O L L I N G E L E M E N T D A T A

BEARING NUMBER (1) TYPE - CYLINDRICAL ROLLER BEARING

ROLLER DIAMETER	ROLLER LENGTH	ROLLER PROFILE FLAT LENGTH	ROLLER CROWN RADIUS	ROLLER END RADIUS	ROLLER INCLUDED ANGLE	NUMBER OF AXIAL LAMINAE
7.0000	7.0000	4.5000	670.0000	10000.0600	.0000	11

	OUTER RACEWAY				INNER RACEWAY		
EFFECTIVE LENGTH	CROWN RADIUS	AXIAL PLAY	FLANGE ANGLE	EFFECTIVE LENGTH	CROWN RADIUS	AXIAL PLAY	FLANGE ANGLE
6.3000	.0000	-.0005	.0000	6.3000	.0000	-.0005	.0000

BEARING NUMBER (2) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
9.5250	.518	.516

BEARING NUMBER (3) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
9.5250	.518	.516

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

S U R F A C E D A T A

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	.13	.13	.08	2.000	2.000	2.000
2	.10	.10	.01	2.000	2.000	2.000
3	.10	.10	.01	2.000	2.000	2.000

BEARING NUMBER	CLA ROUGHNESS OUTER FL.	CLA ROUGHNESS INNER FL.	CLA ROUGHNESS ROLL. ELM. END	RMS ASPERITY SLOPE OUTER FL.	RMS ASPERITY SLOPE INNER FL.	RMS ASPERITY SLOPE ROLL. ELM. END
1	.13	.13	.08	2.000	2.000	2.000

L U B R I C A N T D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	SANTOTRAC	33.60	5.61	.8890	4.10-04	.100
2	SANTOTRAC	33.60	5.61	.8890	4.10-04	.100
3	SANTOTRAC	33.60	5.61	.8890	4.10-04	.100

L 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 NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	1.00	.3000-03	.1500-03	.10
2	1.00	.3000-03	.1500-03	.10
3	1.00	.3000-03	.1500-03	.10

BEARING NUMBER	CAGE FRICTION COEFFICIENT	FILM REPLENISHMENT LAYER THICKNESS		FLG-R.E. FRICTION COEFFICIENT
1	.08	.3000-03	.1500-03	.08
2	.08	.0000	.0000	.00
3	.08	.0000	.0000	.00

UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

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GIVEN TEMPERATURES

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00
2	150.00	150.00	150.00	.00	.00	.00	.00	150.00	150.00	150.00	150.00	150.00	150.00
3	150.00	150.00	150.00	.00	.00	.00	.00	150.00	150.00	150.00	150.00	150.00	150.00

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X - Y.

11 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 3 BEARINGS. MODULUS OF ELASTICITY = 2.041+05

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT		BEARING SEAT POS.ERR	BEARING SEAT DEFL/FOR	BEARING SEAT ANG.ERR	BEARING SEAT DEFL/MOM
1	.0	.0	8.0	.0	17.0					1				
2	7.0	8.0	8.0	17.0	17.0					2	.000	0.00	.0000	0.00
3	14.0	8.0	8.0	17.0	22.0					3				
4	19.0	8.0	13.0	31.0	37.0					4				
5	27.3	13.0	13.0	42.0	42.0	650.4	50193.0			5				
6	35.3	13.0	13.0	47.0	47.0					6				
7	45.0	13.0	13.0	25.0	25.0					7				
8	51.4	13.0	13.0	25.0	25.0					8	.000	0.00	.0000	0.00
9	66.2	13.0	13.0	25.0	25.0					9	.000	0.00	.0000	0.00
10	75.8	13.0	13.0	25.0	25.0					10				
11	90.8	13.0	13.0	25.0	25.0					11				
12	101.9	13.0	13.0	25.0	22.0					12				

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X - Z.

11 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 3 BEARINGS, MODULUS OF ELASTICITY = 2.041+05

THRUST LOAD = 2.147+03

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT
1	.0	.0	8.0	.0	17.0				
2	7.0	3.0	8.0	17.0	17.0				
3	14.0	8.0	8.0	17.0	22.0				
4	17.0	8.0	13.0	31.0	37.0				
5	27.3	13.0	13.0	42.8	42.0	2705.0			
6	36.3	13.0	13.0	47.0	47.0				
7	45.0	13.0	13.0	25.0	25.0				
8	51.4	13.0	13.0	25.0	25.0				
9	66.2	13.0	13.0	25.0	25.0				
10	75.8	13.0	13.0	25.0	25.0				
11	90.8	13.0	13.0	25.0	25.0				
12	101.8	13.0	13.0	25.0	22.0				

	BEARING SEAT POS.ERR	DEFL/FOR	ANG.ERR	DEFL/MOM
1				
2	.000	0.00	.0000	0.00
3				
4				
5				
6				
7				
8	.000	0.00	.0000	0.00
9	.000	0.00	.0000	0.00
10				
11				
12				

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

B E A R I N G S Y S T E M O U T P U T METRIC UNITS

LINEAR (MM) AND ANGULAR (RADIANS) DEFLECTIONS / REACTION FORCES (N) AND MOMENTS (MM-N)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	-1.436-03	-1.821-02	1.939-02	2.847-04	4.052-04	0.000	-707.	1.331+03	224.	165.
2	-1.436-03	8.791-03	7.737-03	2.759-04	4.084-04	1.048+03	406.	843.	8.405+03	-3.1[illegible]1+03
3	-1.436-03	1.477-02	3.677-03	2.700-04	4.124-04	1.839+03	951.	530.	5.346+03	-7.980+03

FATIGUE LIFE (HOURS) / H/SIGMA / LUBE-LIFE FACTOR / MATERIAL FACTOR

BRG.	O. RACE	I. RACE	BEARING	H/SIGMA O. RACE	H/SIGMA I. RACE	LUBE-LIFE O. RACE	LUBE-LIFE I. RACE	MATERIAL O. RACE	MATERIAL I. RACE
1	8.300+03	1.106+03	1.013+03	.793	.832	.280	.230	5.00	5.00
2	4.252+04	8.133+03	7.121+03	1.38	1.14	1.35	.820	5.00	5.00
3	3.244+04	5.493+03	4.886+03	1.34	1.13	1.32	.790	5:00	5.00

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00
2	150.00	150.00	150.00	.00	.00	.00	.00	150.00	150.00	150.00	150.00	150.00	150.00
3	150.00	150.00	150.00	.00	.00	.03	.00	150.00	150.00	150.00	150.00	150.00	150.00

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

B E A R I N G   S Y S T E M   O U T P U T   METRIC UNITS

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

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	126.	0.000	35.9	0.000	27.0	2.235-08	3.60	193.	51.1
2	29.3	0.000	51.7	0.000	45.8	2.671+03	2.79	2.801+03	744.
3	23.7	0.000	53.9	0.000	46.2	3.057+03	2.98	3.189+03	847.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W/DEG.C)	
1	.149	.124	.999	.938	.943	.943	.538	.287	2.88	1.68
2	.170	.144	.992	.938	.912	.914	.314	.203	1.90	1.40
3	.153	.142	.992	.939	.913	.915	.315	.203	1.96	1.48

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B E A R I N G S Y S T E M O U T P U T   METRIC UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1	OUTER	150.000	.8339	2.519	2.101	.1033-01
	INNER	150.000	.8339	2.519	2.101	.1033-01
	BULK	150.000	.8339	2.519	2.101	.1033-01

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 2	OUTER	150.000	.8339	2.519	2.101	.1033-01
	INNER	150.000	.8339	2.519	2.101	.1033-01
	BULK	150.000	.8339	2.519	2.101	.1033-01

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 3	OUTER	150.000	.8339	2.519	2.101	.1033-01
	INNER	150.000	.8339	2.519	2.101	.1033-01
	BULK	150.000	.8339	2.519	2.101	.1033-01

C A G E D A T A   METRIC UNITS

	CAGE RAIL - RING LAND DATA				CAGE SPEED DATA					
BRG.	TORQUE (MM-N)	HEAT RATE (WATTS)	SEP.FORCE (NEWTONS)	ECCENTRICITY RATIO	EPICYCLIC SPEED (RAD/SEC)	EPICYCLIC SPEED (RPM)	CALCULATED SPEED (RAD/SEC)	CALCULATED SPEED (RPM)	CALC/EPIC RATIO	CAGE/SHAFT RATIO
1	1.57	3.60	.116	.460	1.472+03	1.406+04	1.472+03	1.406+04	1.00	.391

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2	1.35	2.99	1.534-02	1.000-01	1.555+03	1.485+04	1.555+03	1.485+04	1.00	.413
3	1.35	2.98	1.535-02	1.000-01	1.559+03	1.489+04	1.559+03	1.489+04	1.00	.414

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R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)					SPEED VECTOR ANGLES (DEGREES)	
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
36.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
72.00	-8224.357	.280	.000	8224.357	1472.391	180.00	180.00
108.00	-8224.357	1.308	.000	8224.357	1472.391	179.99	180.00
144.00	-8224.357	1.825	.000	8224.357	1472.391	179.99	180.00
180.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
216.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
252.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
288.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
324.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	.001	73.186	.000	355.733	.000	.6797	.0000	.00	.00
36.00	.001	73.186	.000	355.733	.000	.6797	.0000	.00	.00
72.00	.000	247.653	174.556	653.946	693.411	.5309	.6235	.00	.00
108.00	.000	916.232	843.070	1282.130	1448.422	.4204	.4577	.00	.00
144.00	.000	691.326	618.064	1082.740	1293.313	.4363	.4777	.00	.00
180.00	-.000	73.237	.000	355.870	.000	.6796	.0000	.00	.00
216.00	-.000	73.188	.000	355.733	.000	.6797	.0000	.00	.00
252.00	-.000	73.186	.000	355.733	.000	.6797	.0000	.00	.00
288.00	-.001	73.158	.000	355.733	.000	.6797	.0000	.00	.00
324.00	-.001	73.188	.000	355.733	.000	.6797	.0000	.00	.00

HERTZ STRESS ACROSS THE OUTER RACEWAY-ROLLER PROFILE

.1271+06	.1734+06	.1744+06	.1726+06	.1708+06	.1690+06	.1671+06	.1653+06	.1634+06	.1587+06
.1023+06	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

ACROSS THE INNER RACEWAY-ROLLER PROFILE

.1489+06	.2089+06	.2101+06	.2078+06	.2054+06	.2031+06	.2007+06	.1983+06	.1959+06	.1897+06
.1156+06	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-7246.961	1767.463	-26.334	7459.428	1489.919	166.29	-179.79
32.73	-7091.933	1990.580	-29.350	7366.056	1474.454	164.32	-179.76
65.45	-7050.622	2113.999	-31.186	7360.790	1475.199	163.31	-179.75
98.18	-7093.371	2151.788	-31.974	7412.633	1495.932	163.12	-179.74
130.91	-7231.971	2033.274	-30.807	7570.204	1515.145	164.42	-179.76
163.64	-7674.830	1758.030	-27.626	7873.714	1571.409	167.10	-179.79
196.36	-8063.223	1456.297	-23.760	8193.713	1631.504	169.76	-179.83
229.09	-8291.956	1214.783	-20.220	8370.597	1664.526	171.66	-179.86
261.82	-8246.810	1103.651	-18.252	8320.352	1653.819	172.38	-179.87
294.55	-7378.242	1176.609	-18.969	8064.559	1603.715	171.61	-179.86
327.27	-7585.624	1434.716	-22.063	7720.142	1537.779	169.29	-179.83

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-134.272	354.060	216.104	1274.538	1309.186	.1528	.2261	17.07	28.74
32.73	-58.074	516.752	373.603	1445.724	1579.636	.1559	.2193	19.51	27.04
65.45	25.837	583.553	447.139	1505.513	1668.251	.1561	.2168	20.76	27.56
98.18	103.734	496.635	360.857	1426.714	1553.193	.1544	.2182	20.99	29.53
130.91	160.685	347.178	208.651	1262.558	1293.960	.1480	.2237	19.39	33.21
163.64	172.827	248.328	112.115	1132.855	1051.965	.1345	.2333	16.07	37.85
196.36	123.963	208.047	69.688	1067.526	977.773	.1209	.2447	12.76	41.25
229.09	26.279	191.451	51.333	1038.349	810.798	.1138	.2542	10.40	42.53
261.82	-83.104	188.273	47.339	1032.573	789.197	.1154	.2565	9.50	41.84
294.55	-160.605	199.703	59.234	1053.058	850.428	.1255	.2497	10.46	37.73
327.27	-177.269	241.129	101.988	1121.350	1019.286	.1402	.2379	13.35	33.07

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NASA ALTERNATE INPUT PINION DESIGN 2A, EOL LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 3 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-7068.245	1949.471	-28.603	7332.212	1467.239	164.58	-179.77
32.73	-7037.496	2042.769	-29.982	7328.039	1467.729	163.81	-179.76
65.45	-7079.845	2054.387	-30.820	7380.368	1478.602	163.59	-179.75
98.18	-7289.767	1982.039	-29.956	7554.474	1511.355	164.79	-179.76
130.91	-7713.513	1735.243	-27.375	7906.333	1577.582	167.32	-179.80
163.64	-8139.949	1487.715	-24.512	8273.838	1647.611	169.64	-179.83
196.36	-8378.461	1290.623	-21.762	8477.310	1666.193	171.24	-179.85
229.09	-8556.432	1188.552	-19.899	8420.757	1674.262	171.89	-179.86
261.82	-8027.641	1259.783	-20.031	8122.837	1615.710	171.22	-179.86
294.55	-7576.418	1472.068	-22.637	7718.135	1537.735	169.00	-179.83
327.27	-7237.737	1776.567	-26.378	7424.037	1483.087	166.14	-179.79

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NASA ALTERNATE INPUT PINION DESIGN 2A: 60% LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 3 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-66.677	579.549	441.629	1502.062	1661.378	.1566	.2182	19.19	25.55
32.73	28.857	647.219	509.923	1558.386	1742.938	.1567	.2162	20.14	25.72
65.45	118.458	526.847	390.357	1455.077	1594.415	.1551	.2180	20.41	28.08
98.18	185.228	342.818	206.828	1260.885	1290.163	.1482	.2242	18.93	32.52
130.91	208.247	243.229	106.779	1124.596	1035.085	.1328	.2348	15.79	38.31
163.64	144.011	205.824	67.996	1063.710	890.445	.1175	.2451	12.91	42.55
196.36	30.978	191.369	52.058	1038.201	814.595	.1067	.2540	10.92	44.12
229.09	-95.992	189.082	48.821	1033.984	797.358	.1135	.2558	10.12	42.84
261.82	-186.103	200.508	60.505	1354.458	856.466	.1238	.2488	10.94	38.98
294.55	-204.819	244.724	105.714	1126.895	1031.552	.1408	.2369	13.78	33.25
327.27	-154.181	381.220	242.973	1306.320	1361.339	.1538	.2248	17.26	27.74

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NASA

1 CYLINDRICAL
2 BALL BEARINGS

LEVEL 1

C-3

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     THIS DATA SET CONTAINS  3  BEARINGS
      BEARING NO. (1) -   CYLINDRICAL ROLLER BEARING , FLANGE IS TYPE NO.  1.
      BEARING NO. (2) -   BALL BEARING
      BEARING NO. (3) -   BALL BEARING


     SOLUTION LEVEL = 1
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UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	10	.000	32.100	.025	.000	35963.	0.
2	11	.000	38.500	-.006	25.000	35963.	0.
3	11	.000	38.500	-.006	25.000	35963.	0.

D:57

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	INNER RING LAND RIDING	.150000	2.0000	24.0000	.150	.013000
2	INNER RING LAND RIDING	.150000	2.0000	24.0000	.150	.013000
3	INNER RING LAND RIDING	.150000	2.0000	24.0000	.150	.013000

S T E E L D A T A

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	M-50 STEEL	5.000	M-50 STEEL	5.000
2	M-50 STEEL	5.000	M-50 STEEL	5.000
3	M-50 STEEL	5.000	M-50 STEEL	5.000

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R O L L I N G E L E M E N T D A T A

BEARING NUMBER (1) TYPE - CYLINDRICAL ROLLER BEARING

ROLLER DIAMETER	ROLLER LENGTH	ROLLER PROFILE FLAT LENGTH	ROLLER CROWN RADIUS	ROLLER END RADIUS	ROLLER INCLUDED ANGLE	NUMBER OF AXIAL LAMINAE
7.0000	7.0000	4.5000	670.0000	10000.0000	.0000	11

OUTER RACEWAY				INNER RACEWAY			
EFFECTIVE LENGTH	CROWN RADIUS	AXIAL PLAY	FLANGE ANGLE	EFFECTIVE LENGTH	CROWN RADIUS	AXIAL PLAY	FLANGE ANGLE
6.3000	.0000	-.0005	.0000	6.3000	.0000	-.0005	.0000

BEARING NUMBER (2) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
9.5250	.518	.516

BEARING NUMBER (3) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
9.5250	.518	.516

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S U R F A C E D A T A

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	.13	.13	.08	2.000	2.000	2.000
2	.10	.10	.01	2.000	2.000	2.000
3	.10	.10	.01	2.000	2.000	2.000

BEARING NUMBER	CLA ROUGHNESS OUTER FL.	CLA ROUGHNESS INNER FL.	CLA ROUGHNESS ROLL. ELM. END	RMS ASPERITY SLOPE OUTER FL.	RMS ASPERITY SLOPE INNER FL.	RMS ASPERITY SLOPE ROLL. ELM. END
1	.13	.13	.08	2.000	2.000	2.000

L U B R I C A N T D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	SANTOTRAC	33.60	5.61	.8890	4.10-04	.100
2	SANTOTRAC	33.60	5.61	.8890	4.10-04	.100
3	SANTOTRAC	33.60	5.61	.8890	4.10-04	.100

L 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 NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	1.00	.3000-03	.1500-03	.10
2	1.00	.3000-03	.1500-03	.10
3	1.00	.3000-03	.1500-03	.10

BEARING NUMBER	CAGE FRICTION COEFFICIENT	FILM REPLENISHMENT LAYER THICKNESS		FLG-R.E. FRICTION COEFFICIENT
1	.08	.3000-03	.1500-03	.08
2	.08	.0000	.0000	.00
3	.08	.0000	.0000	.00

UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

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GIVEN TEMPERATURES

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00
2	150.00	150.00	150.00	.00	.00	.00	.00	150.00	150.00	150.00	150.00	150.00	150.00
3	150.00	150.00	150.00	.00	.00	.00	.00	150.00	150.00	150.00	150.00	150.00	150.00

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X - Y.

11 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 3 BEARINGS. MODULUS OF ELASTICITY = 2.041+05

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT		BEARING SEAT POS.ERR	DEFL/FOR	ANG.ERR	DEFL/MOM
1	.0	.0	8.0	.0	17.0					1				
2	7.0	8.0	8.0	17.0	17.0					2	.000	0.00	.0000	0.00
3	14.0	8.0	8.0	17.0	22.0					3				
4	19.0	8.0	13.0	31.0	37.0					4				
5	27.3	13.0	13.0	42.0	42.0	650.4	50133.0			5				
6	36.3	13.0	13.0	47.0	47.0					6				
7	45.0	13.0	13.0	25.0	25.0					7				
8	51.4	13.0	13.0	25.0	25.0					8	.000	0.00	.0000	0.00
9	66.2	13.0	13.0	25.0	25.0					9	.000	0.00	.0000	0.00
10	75.8	13.0	13.0	25.0	25.0					10				
11	90.8	13.0	13.0	25.0	25.0					11				
12	101.8	13.0	13.0	25.0	22.0					12				

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X - Z.

11 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 3 BEARINGS. MODULUS OF ELASTICITY = 2.041+05

THRUST LOAD = 2.147+03

	POSI-	INNER DIAM.		OUTER DIAM.		POINT	POINT	LOAD INTENSITY			BEARING SEAT			
	TION	LEFT	RIGHT	LEFT	RIGHT	FORCE	MOMENT	LEFT	RIGHT		POS.ERR	DEFL/FOR	ANG.ERR	DEFL/MOM
1	.0	.0	8.0	.0	17.0					1				
2	7.0	8.0	8.0	17.0	17.0					2	.000	0.00	.0000	0.00
3	14.3	8.0	8.0	17.0	22.0					3				
4	19.3	8.0	13.0	31.0	37.0					4				
5	27.3	13.0	13.0	42.0	42.0	2705.0				5				
6	36.3	13.0	13.0	47.0	47.0					6				
7	45.0	13.0	13.0	25.0	25.0					7				
8	51.4	13.0	13.0	25.0	25.0					8	.000	0.00	.0000	0.00
9	66.2	13.0	13.0	25.0	25.0					9	.000	0.00	.0000	0.00
10	75.8	13.0	13.0	25.0	25.0					10				
11	90.8	13.0	13.0	25.0	25.0					11				
12	101.8	13.0	13.0	25.0	22.0					12				

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

B E A R I N G S Y S T E M O U T P U T METRIC UNITS

LINEAR (MM) AND ANGULAR (RADIANS) DEFLECTIONS — REACTION FORCES (N) AND MOMENTS (MM-N)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	-1.436-03	-1.021-02	1.939-02	2.047-04	4.052-04	0.000	-707.	1.331+03	224.	165.
2	-1.436-03	3.731-03	7.737-03	2.759-04	4.084-04	1.048+03	406.	843.	8.405+03	-3.121+03
3	-1.436-03	1.477-02	3.677-03	2.700-04	4.124-04	1.099+03	951.	530.	5.346+03	-7.988+03

BRG.	FATIGUE LIFE (HOURS) O. RACE	I. RACE	BEARING	H/SIGMA O. RACE	I. RACE	LUBE-LIFE FACTOR O. RACE	I. RACE	MATERIAL FACTOR O. RACE	I. RACE
1	3.705+04	2.632+03	2.572+03	1.31	1.02	1.25	.560	5.00	5.00
2	7.087+04	2.212+04	1.778+04	2.23	2.18	2.25	2.23	5.00	5.00
3	5.456+04	1.516+04	1.248+04	2.18	2.10	2.22	2.18	5.00	5.00

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00
2	150.00	150.00	150.00	.00	.00	.00	.00	150.00	150.00	150.00	150.00	150.00	150.00
3	150.00	150.00	150.00	.00	.00	.00	.00	150.00	150.00	150.00	150.00	150.00	150.00

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

B E A R I N G   S Y S T E M   O U T P U T   METRIC UNITS

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

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	129.	0.000	37.7	0.000	27.0	2.235-08	3.69	197.	52.4
2	32.2	0.000	136.	0.000	45.8	2.671+03	2.99	2.888+03	767.
3	33.9	0.000	142.	0.000	46.2	3.057+03	2.98	3.282+03	872.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W/DEG.C)	
1	.251	.135	.333	.338	.943	.943	.558	.287	.853	.473
2	.221	.274	.332	.348	.912	.914	.314	.203	.737	.413
3	.274	.264	.332	.388	.913	.915	.315	.203	.765	.442

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

B E A R I N G   S Y S T E M   O U T P U T   METRIC UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1	OUTER	150.000	.8339	2.519	2.101	.1033-01
	INNER	150.000	.8339	2.519	2.101	.1033-01
	BULK	150.000	.8339	2.519	2.101	.1033-01

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 2	OUTER	150.000	.8339	2.519	2.101	.1033-01
	INNER	150.000	.8339	2.519	2.101	.1033-01
	BULK	150.000	.8339	2.519	2.101	.1033-01

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 3	OUTER	150.000	.8339	2.519	2.101	.1033-01
	INNER	150.000	.8339	2.519	2.101	.1033-01
	BULK	150.000	.8339	2.519	2.101	.1033-01

C A G E   D A T A   METRIC UNITS

	CAGE RAIL - RING LAND DATA				CAGE SPEED DATA					
BRG.	TORQUE (MM-N)	HEAT RATE (WATTS)	SEP.FORCE (NEWTONS)	ECCENTRICITY RATIO	EPICYCLIC SPEED (RAD/SEC)	EPICYCLIC SPEED (RPM)	CALCULATED SPEED (RAD/SEC)	CALCULATED SPEED (RPM)	CALC/EPIC RATIO	CAGE/SHAFT RATIO
1	1.37	3.60	.116	.460	1.472+03	1.406+04	1.472+03	1.406+04	1.00	.391

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2	1.35	2.99	1.534-02	1.000-01	1.555+03	1.485+04	1.555+03	1.485+04	1.00	.413
3	1.35	2.98	1.535-02	1.000-01	1.559+03	1.489+04	1.559+03	1.489+04	1.00	.414

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R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
36.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
72.00	-8224.357	.280	.000	8224.357	1472.391	180.00	180.00
108.00	-8224.357	1.308	.000	8224.357	1472.391	179.99	180.00
144.00	-8224.357	1.825	.000	8224.357	1472.391	179.99	180.00
180.00	-8224.357	.000	.000	8224.357	1472.391	1[illegible].00	180.00
216.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
252.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
288.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00
324.00	-8224.357	.000	.000	8224.357	1472.391	180.00	180.00

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	.001	73.186	.000	355.733	.000	.0000	.0000	.00	.00
36.00	.001	73.186	.000	355.733	.000	.0000	.0000	.00	.00
72.00	.000	247.653	174.556	653.946	693.411	.0000	.0000	.00	.00
108.00	.000	916.232	843.070	1202.130	1448.422	.0000	.0000	.00	.00
144.00	.000	691.326	618.064	1082.740	1293.313	.0000	.0000	.00	.00
180.00	-.000	73.237	.000	355.870	.000	.0000	.0000	.00	.00
216.00	-.000	73.186	.000	355.733	.000	.0000	.0000	.00	.00
252.00	-.000	73.186	.000	355.733	.000	.0000	.0000	.00	.00
288.00	-.001	73.186	.000	355.733	.000	.0000	.0000	.00	.00
324.00	-.001	73.186	.000	355.733	.000	.0000	.0000	.00	.00

HERTZ STRESS ACROSS THE OUTER RACEWAY-ROLLER PROFILE

.1271+06	.1734+06	.1744+06	.1726+06	.1708+06	.1690+06	.1671+06	.1653+06	.1634+06	.1587+06
.1023+06	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

ACROSS THE INNER RACEWAY-ROLLER PROFILE

.1449+06	.2089+06	.2101+06	.2078+06	.2054+06	.2031+06	.2007+06	.1983+06	.1959+06	.1897+06
.1156+06	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-7246.961	1767.463	-26.334	7459.428	1489.919	166.29	-179.79
32.73	-7091.933	1990.580	-29.350	7366.056	1474.454	164.32	-179.76
65.45	-7050.622	2113.999	-31.186	7360.790	1475.199	163.31	-179.75
98.18	-7093.371	2151.788	-31.974	7412.633	1485.332	163.12	-179.74
130.91	-7231.971	2033.274	-30.807	7570.204	1515.145	164.42	-179.76
163.64	-7674.890	1758.030	-27.626	7873.714	1571.409	167.10	-179.79
196.36	-8063.223	1456.297	-23.760	8193.713	1631.504	169.76	-179.83
229.09	-8291.956	1214.783	-20.220	8370.597	1664.526	171.66	-179.86
261.82	-8246.810	1103.651	-18.252	8320.352	1653.819	172.38	-179.87
294.55	-7978.242	1176.609	-18.969	8064.559	1603.715	171.61	-179.86
327.27	-7585.624	1434.716	-22.063	7720.142	1537.779	169.29	-179.83

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-134.272	354.066	216.104	1274.538	1309.186	.0000	.0000	17.07	26.74
32.73	-58.076	516.752	379.603	1445.724	1579.636	.0000	.0000	19.51	27.04
65.45	25.837	583.553	447.139	1505.513	1668.251	.0000	.0000	20.76	27.56
98.18	103.734	496.635	360.857	1426.714	1553.193	.0000	.0000	20.99	29.53
130.91	160.687	344.176	209.651	1262.558	1293.960	.0000	.0000	19.39	33.21
163.64	172.827	248.628	112.115	1132.855	1051.965	.0000	.0000	16.07	37.86
196.36	123.963	208.047	69.688	1067.526	897.773	.0000	.0000	12.76	41.25
229.09	26.279	191.451	51.333	1038.349	810.798	.0000	.0000	10.40	42.33
261.82	-83.104	188.273	47.339	1032.573	789.197	.0000	.0000	9.50	41.04
294.55	-160.605	199.703	59.234	1053.058	850.428	.0000	.0000	10.46	37.73
327.27	-177.259	241.129	101.986	1121.350	1019.286	.0000	.0000	13.35	33.07

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60% LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 3 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)					SPEED VECTOR ANGLES (DEGREES)	
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.03	-7068.245	1949.471	-28.603	7332.212	1467.239	164.58	-179.77
32.73	-7037.496	2042.769	-29.982	7328.039	1467.729	163.81	-179.76
65.45	-7079.845	2084.387	-30.820	7380.368	1478.602	163.59	-179.75
98.18	-7289.767	1982.039	-29.956	7554.474	1511.355	164.79	-179.76
130.91	-7715.513	1735.243	-27.375	7906.333	1577.582	167.32	-179.80
163.64	-8138.949	1487.719	-24.512	8273.838	1647.611	169.64	-179.83
196.36	-8378.461	1290.623	-21.762	8477.310	1686.193	171.24	-179.85
229.09	-8536.432	1188.552	-19.899	8420.757	1674.262	171.89	-179.86
261.82	-8027.641	1239.783	-20.031	8122.837	1615.710	171.22	-179.86
294.55	-7576.418	1472.068	-22.637	7718.135	1537.735	169.00	-179.83
327.27	-7237.737	1778.567	-26.378	7424.037	1483.087	166.14	-179.79

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NASA ALTERNATE INPUT PINION DESIGN 2A, 60(LOAD

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 3 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-66.677	579.549	441.629	1502.062	1661.370	.0000	.0000	19.19	25.55
32.73	28.657	547.219	509.923	1558.386	1742.938	.0000	.0000	20.14	25.92
65.45	118.450	526.847	370.357	1455.077	1594.415	.0000	.0000	20.41	28.08
98.18	135.228	392.810	206.820	1260.885	1290.163	.0000	.0000	18.93	32.52
130.91	200.247	243.229	106.779	1124.596	1035.005	.0000	.0000	15.79	38.31
163.64	144.011	205.824	67.996	1063.710	890.445	.0000	.0000	12.91	42.55
196.36	50.978	191.369	52.058	1038.201	814.595	.0000	.0000	10.92	44.12
229.09	-95.392	183.002	48.821	1033.904	737.350	.0000	.0000	10.12	42.84
261.82	-186.193	200.530	60.505	1054.458	856.466	.0000	.0000	10.94	38.98
294.55	-204.619	244.724	105.714	1126.895	1031.552	.0000	.0000	13.70	33.25
327.27	-154.181	381.220	242.973	1306.320	1361.339	.0000	.0000	17.26	27.74

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SKF

1 BALL BEARING

LEVEL 2

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

TURBINE COMPRESSOR BEARING

THIS DATA SET CONTAINS 1 BEARING

BEARING NO. (1) - BALL BEARING

SOLUTION LEVEL = 2

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

TURBINE COMPRESSOR BEARING

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	8	.000	57.500	.054	14.791	36936.	0.

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	BALL RIDING	.476300	1.0668	57.5000	.889000	.400000

S T E E L D A T A

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	M-50 STEEL	1.000	M-50 STEEL	1.000

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TURBINE COMPRESSOR BEARING

R O L L I N G E L E M E N T D A T A

BEARING NUMBER (1) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
13.9937	.520	.520

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TURBINE COMPRESSOR BEARING

S U R F A C E   D A T A

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	.12	.12	.12	2.000	2.000	2.000

L U B R I C A N T   D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152

L 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 NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	2.00	.6350-03	.3810-03	.10

UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

GIVEN TEMPERATURES

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00

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TURBINE COMPRESSOR BEARING

LOADING IN THE X - Y PLANE

CONCENTRATED FORCE, FY	CONCENTRATED MOMENT ABOUT Z
111.2 NEWTONS	2500.0 NEWTON-MM.

LOADING IN THE X - Z PLANE

CONCENTRATED FORCE, FZ	CONCENTRATED MOMENT ABOUT Y
.0 NEWTONS	.0 NEWTON-MM.

THRUST LOAD FX = 1912.6 NEWTONS

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TURBINE COMPRESSOR BEARING

B E A R I N G   S Y S T E M   O U T P U T   METRIC UNITS

LINEAR (MM) AND ANGULAR (RADIANS) DEFLECTIONS — REACTION FORCES (N) AND MOMENTS (MM-N)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	1.562-02	1.455-02	1.253-07	2.753-11	8.260-04	2.169+03	103.	-25.5	12.6	3.508+03

	FATIGUE LIFE (HOURS)			H/SIGMA		LUBE-LIFE FACTOR		MATERIAL FACTOR	
BRG.	O. RACE	I. RACE	BEARING	O. RACE	I. RACE	O. RACE	I. RACE	O. RACE	I. RACE
1	420.	617.	267.	.738	.663	.290	.230	1.00	1.00

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00

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TURBINE COMPRESSOR BEARING

B E A R I N G S Y S T E M O U T P U T METRIC UNITS

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

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	247.	0.000	503.	0.000	350.	95.7	0.000	1.303+03	337.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W.DEG.C)	
1	.166	.141	.977	.998	.903	.925	.552	.416	8.14	4.09

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TURBINE COMPRESSOR BEARING

B E A R I N G S Y S T E M O U T P U T METRIC UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1	OUTER	143.000	.8580	1.709	1.466	.7726-02
	INNER	143.000	.8580	1.709	1.466	.7726-02
	BULK	143.000	.8580	1.709	1.466	.7726-02

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TURBINE COMPRESSOR BEARING

ROLLING ELEMENT OUTPUT FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-9626.030	15.423	-4.286	9626.045	1694.434	179.90	-179.97
45.00	-9199.249	10.137	-3.388	9199.255	1715.117	179.94	-179.98
90.00	-9246.387	141.074	-48.219	9247.589	1726.205	179.13	-179.70
135.00	-9266.331	333.254	-114.227	9273.575	1721.699	177.94	-179.29
180.00	-9218.897	444.918	-153.731	9231.091	1709.080	177.21	-179.04
225.00	-9138.061	420.540	-145.667	9148.065	1695.735	177.36	-179.10
270.00	-9050.170	284.583	-95.564	9055.150	1699.511	178.20	-179.40
315.00	-8993.559	125.203	-40.732	8994.523	1693.006	179.20	-179.74

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TURBINE COMPRESSOR BEARING

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	15.754	1245.761	453.103	1580.492	1404.171	.4040	.5406	11.23	31.85
45.00	15.176	1251.044	451.118	1582.300	1391.951	.3992	.5437	11.44	33.41
90.00	4.774	1242.541	442.427	1578.787	1382.954	.390[illegible]	.5463	12.19	36.82
135.00	-7.664	1237.621	454.525	1576.620	1395.549	.3968	.5443	13.31	39.98
180.00	-14.896	1228.152	462.978	1572.589	1404.044	.3995	.5422	13.89	41.03
225.00	-13.915	1209.177	450.792	1564.448	1391.616	.4030	.5434	13.39	39.79
270.00	-5.729	1199.730	437.563	1559.930	1377.973	.4360	.5450	12.33	36.67
315.00	6.396	1217.523	448.556	1568.039	1389.311	.4069	.5427	11.55	33.29

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NASA

1 BALL BEARING

LEVEL 2

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TURBINE COMPRESSOR BEARING

THIS DATA SET CONTAINS 1 BEARING

BEARING NO. (1) - BALL BEARING

SOLUTION LEVEL = 2

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TURBINE COMPRESSOR BEARING

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	8	.000	57.500	.054	14.791	36936.	0.

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	BALL RIDING	.476300	1.0668	57.5000	.889000	.400000

S T E E L D A T A

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	M-50 STEEL	1.000	M-50 STEEL	1.000

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TURBINE COMPRESSOR BEARING

R O L L I N G   E L E M E N T   D A T A

BEARING NUMBER (1)   TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
13.4937	.520	.520

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TURBINE COMPRESSOR BEARING

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S U R F A C E D A T A

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	.12	.12	.12	2.000	2.000	2.000

L U B R I C A N T D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152

L 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 NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	2.00	.6350-03	.3810-03	.10

UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

GIVEN TEMPERATURES

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00

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TURBINE COMPRESSOR BEARING

LOADING IN THE X - Y PLANE

CONCENTRATED FORCE, FY	CONCENTRATED MOMENT ABOUT Z
111.2 NEWTONS	2500.0 NEWTON-MM.

LOADING IN THE X - Z PLANE

CONCENTRATED FORCE, FZ	CONCENTRATED MOMENT ABOUT Y
.0 NEWTONS	.0 NEWTON-MM.

THRUST LOAD FX = 1912.6 NEWTONS

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TURBINE COMPRESSOR BEARING

B E A R I N G S Y S T E M O U T P U T METRIC UNITS

LINEAR (MM) AND ANGULAR (RADIANS) DEFLECTIONS — REACTION FORCES (N) AND MOMENTS (MM-N)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	8.652-02	1.455-02	5.263-09	2.953-10	8.260-04	1.610+03	125.	-333.	-7.185+03	2.041+03

	FATIGUE LIFE (HOURS)			H/SIGMA		LUBE-LIFE FACTOR		MATERIAL FACTOR	
BRG.	O. RACE	I. RACE	BEARING	O. RACE	I. RACE	O. RACE	I. RACE	O. RACE	I. RACE
1	425.	1.031+03	316.	.587	.566	.210	.210	1.00	1.00

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)

BRG	O.RACE	I.RACE	BULK OIL	FLNG.1	FLNG.2	FLNG.3	FLNG.4	CAGE	SHAFT	I.RING	ROLL.EL.	O.RING	HSG.
1	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00	149.00

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TURBINE COMPRESSOR BEARING

B E A R I N G S Y S T E M O U T P U T METRIC UNITS

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

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	984.	0.000	404.	0.000	361.	270.	0.000	2.020+03	522.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W/DEG.C)	
1	.137	.167	.998	.996	.910	.848	.556	.404	4.71	1.90

TURBINE COMPRESSOR BEARING

BEARING SYSTEM OUTPUT METRIC UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1	OUTER	143.000	.9580	1.709	1.466	.7726-02
	INNER	143.000	.9580	1.709	1.466	.7726-02
	BULK	143.000	.9580	1.709	1.466	.7726-02

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TURBINE COMPRESSOR BEARING

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)					SPEED VECTOR ANGLES (DEGREES)	
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-8703.802	150.356	-106.768	8705.755	1635.644	179.01	-179.30
45.00	-8513.593	172.407	-122.255	8516.216	1685.069	178.84	-179.18
90.00	-8719.650	172.647	-122.941	8722.226	1738.970	178.87	-179.19
135.00	-9075.521	267.270	-151.745	9080.524	1767.725	178.31	-179.04
180.00	-9132.952	204.652	-145.916	9136.390	1784.398	178.72	-179.08
225.00	-9154.411	290.237	-159.556	9160.400	1768.488	178.18	-179.00
270.00	-8772.096	170.047	-133.631	8775.662	1716.705	178.83	-179.13
315.00	-8918.397	277.389	-220.197	8926.031	1696.616	178.22	-178.59

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TURBINE COMPRESSOR BEARING

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	8.121	1198.327	459.855	1559.755	1490.880	.0006	.0000	11.46	31.66
45.00	47.117	909.587	237.935	1422.807	1124.732	.0000	.0008	13.88	30.93
90.00	54.582	994.234	111.041	1465.640	872.346	.0000	.0000	14.82	34.21
135.00	36.774	1090.115	386.897	1511.316	1322.491	.0000	.0000	13.95	39.21
180.00	5.076	1146.110	358.240	1536.762	1288.998	.0000	.0000	14.51	40.42
225.00	-36.855	1063.401	371.260	1498.869	1304.428	.0000	.0000	14.03	39.06
270.00	-38.088	1176.265	389.593	1550.123	1325.555	.0000	.0000	12.60	36.41
315.00	-27.346	1143.675	443.192	1535.673	1383.752	.0003	.0000	11.81	33.82

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APPENDIX E

Calculation of Cage Pocket and Cage Land Forces

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NOMENCLATURE

APPENDIX E

$A = 16.9706\eta_o U_y \cdot \frac{R^{1/2}}{3 + 2k}$

$B = 16.9706\eta_o U_x R_x^{1/2}$

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

C_ℓ = cage land radial clearance (in.)

$C_o = \eta_o |U_y| (R_x R)^{1/2}\ K1$

C_p = cage pocket radial clearance (in.)

$k = \frac{R}{R_x}$

$K1 = \left\{ \frac{1}{(3 + 2k)^2} + \frac{1}{k(3 + \frac{2}{k})^2} \cdot \left(\frac{U_x}{U_y}\right)^2 \right\}^{1/2}$

ℓ = roller total length (in.)

R = ball or roller radius (in.)

R_ℓ = cage land radius (in.)

$R_x = \left(\frac{1}{R} + \frac{1}{R + C_p}\right)^{-1}$ (in.)

$U = \left(U_x^2 + U_y^2\right)^{\frac{1}{2}}$ = entrainment velocity (in./sec.)

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NOMENCLATURE continued......
APPENDIX E

U_ℓ = entrainment velocity of 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

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

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

V = $\left(V_x^2 + V_y^2\right)^{\frac{1}{2}}$ = sliding velocity (in./sec.)

V_ℓ = 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_y = $-R\omega_x$ (in./sec.)

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

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

η_o = absolute ambient viscosity (lb.-sec./in.^{2})

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.1. 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 ω_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

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

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.

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

$$K = \frac{67}{(C_p - .33R\eta_\circ \ell U)} \approx \frac{67}{C_p} \qquad \text{(roller bearings)}$$

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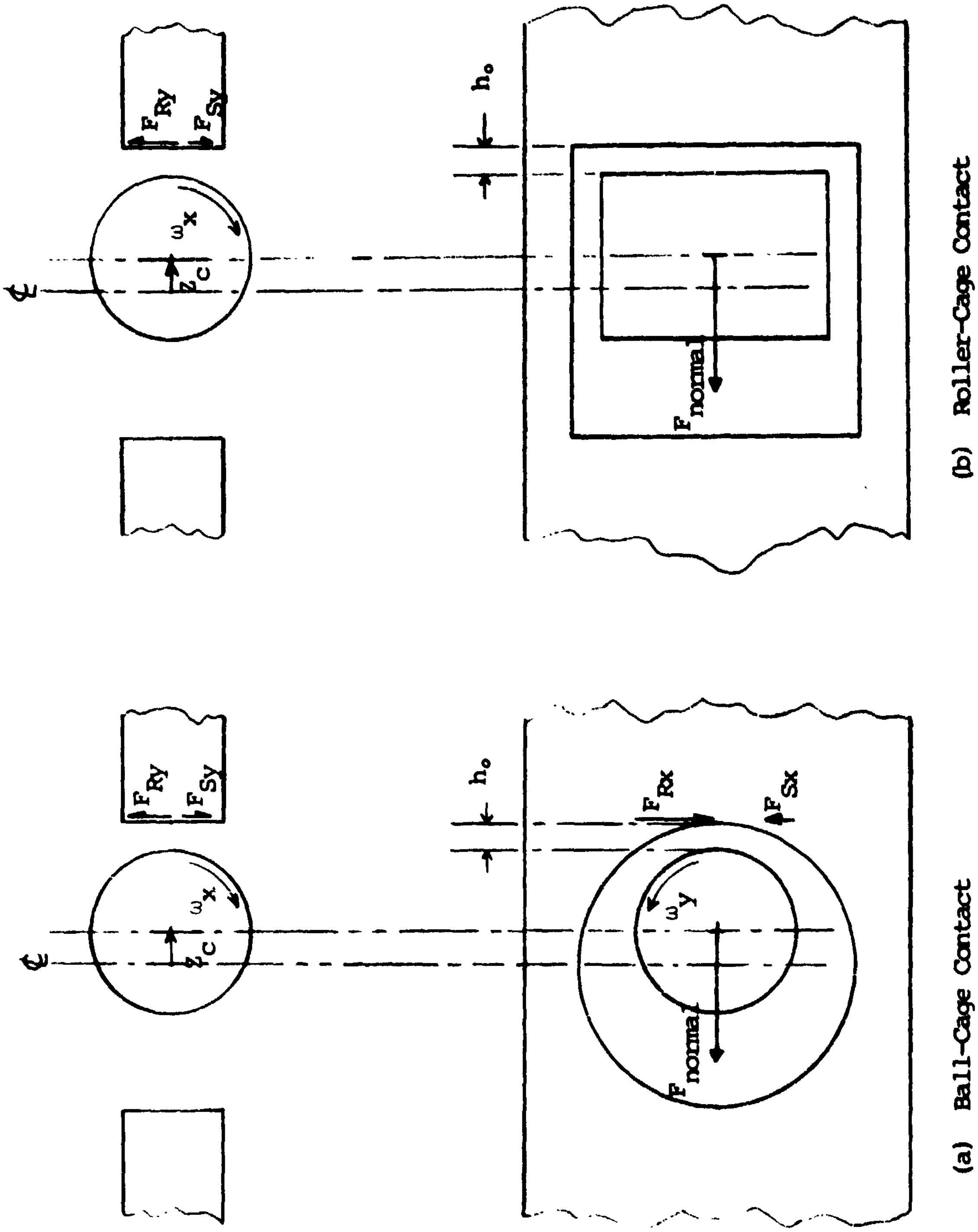

Figure E.1: Configuration of Contacts

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 $\overline{F}_R$ and $\overline{F}_S$ [25].

Forces:

Rolling Components

$$F_{Ry} = \tfrac{1}{2} C_\circ \overline{F}_R \cos\gamma \tag{E.3}$$

$$F_{Rx} = \tfrac{1}{2} C_\circ \overline{F}_R (\sin\gamma) \sqrt{k} \tag{E.4}$$

Sliding Components

$$F_{Sy} = \overline{F}_S \eta_\circ V_X (R_X R_Y)^{\frac{1}{2}} \tag{E.5}$$

$$F_{Sx} = \overline{F}_S \eta_\circ V_y (R_x R_y)^{\frac{1}{2}} \tag{E.6}$$

Moments:

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

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

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

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

or

$$\overline{F}_R = \frac{F_{normal} \, R}{C_o R_x \cos\gamma} \qquad \text{(E.9)}$$

F_{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

$$\overline{F}_S = .26\ \overline{F}_R + 10.90 \qquad \text{(E.10)}$$

Roller/Cage Contacts

The analytic description of lubricated, roller-to-cage pocket contact, Figure E.1 (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

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

where F_{norm} is computed from (E.1). μ_h is a hydrodynamic friction coefficient

---

* $\overline{F}_R$ is shown vs. dimensionless meniscus distance (ρ_1) in Figure E.2; $\overline{F}_S$ is shown vs. ρ_1 in Fig. E.3. The functions are closely approximated by:

$$\overline{F}_R = 34.74\ (\ell n \rho_1) - 27.60 \qquad \text{(E.11)}$$

$$\overline{F}_S = 8.82\ (\ell n \rho_1) + 3.89 \qquad \text{(E.12)}$$

for hydrodynamic contact. We get (E.10) by solving E.11) for $\ell n \rho_1$ and inserting this expression into (E.12).

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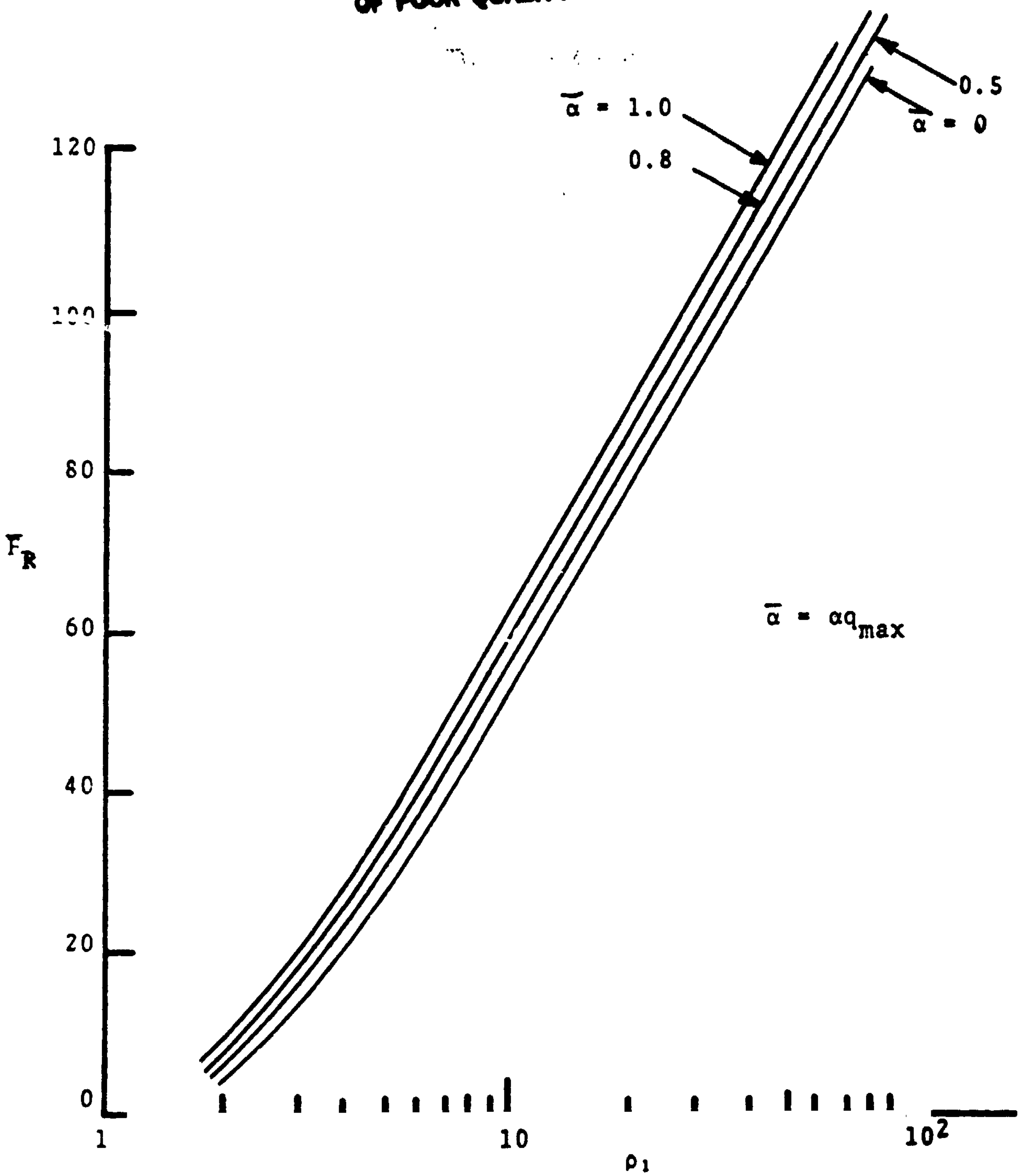

Figure E.2 Variation of $\overline{F_R}$ with the Dimensionless Meniscus Distance ρ_1 (taken from reference [25])

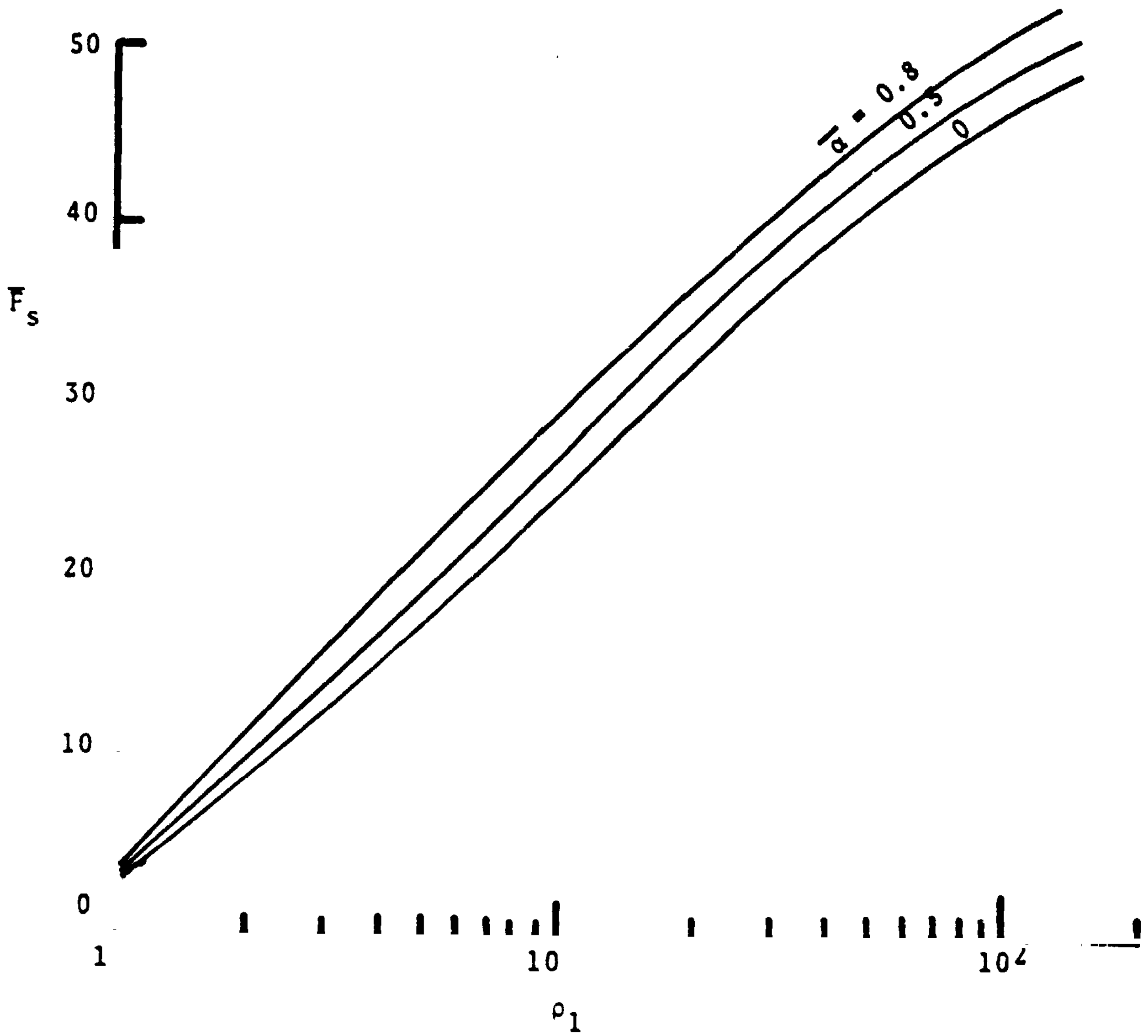

Figure E.3 Variation of $\overline{F}_s$ With ρ_1 (taken from reference [25])

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$$\mu_h = \frac{1}{2\sqrt{\left|\frac{F_{norm}}{2\eta_o U_y \ell}\right|}} \tag{E.14}$$

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

$$M_{Fx} = R\mu_h \left|F_{norm}\right| \tag{E.15}$$

<u>Cage Land Normal Forces and Friction Moment</u>

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 θ_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]

$$F_y' = \frac{\pm\eta_o U_\ell L^3}{C_\ell^2} \cdot \frac{\varepsilon^2}{(1-\varepsilon^2)^2} \tag{E.16}$$

$$F_z' = \frac{\mp\eta_o U_\ell L^3}{C_\ell^2} \frac{\pi\varepsilon}{4(1-\varepsilon^2)^{\frac{3}{2}}} \tag{E.17}$$

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* These figures were repeated from section 2.4.1. for convenience.

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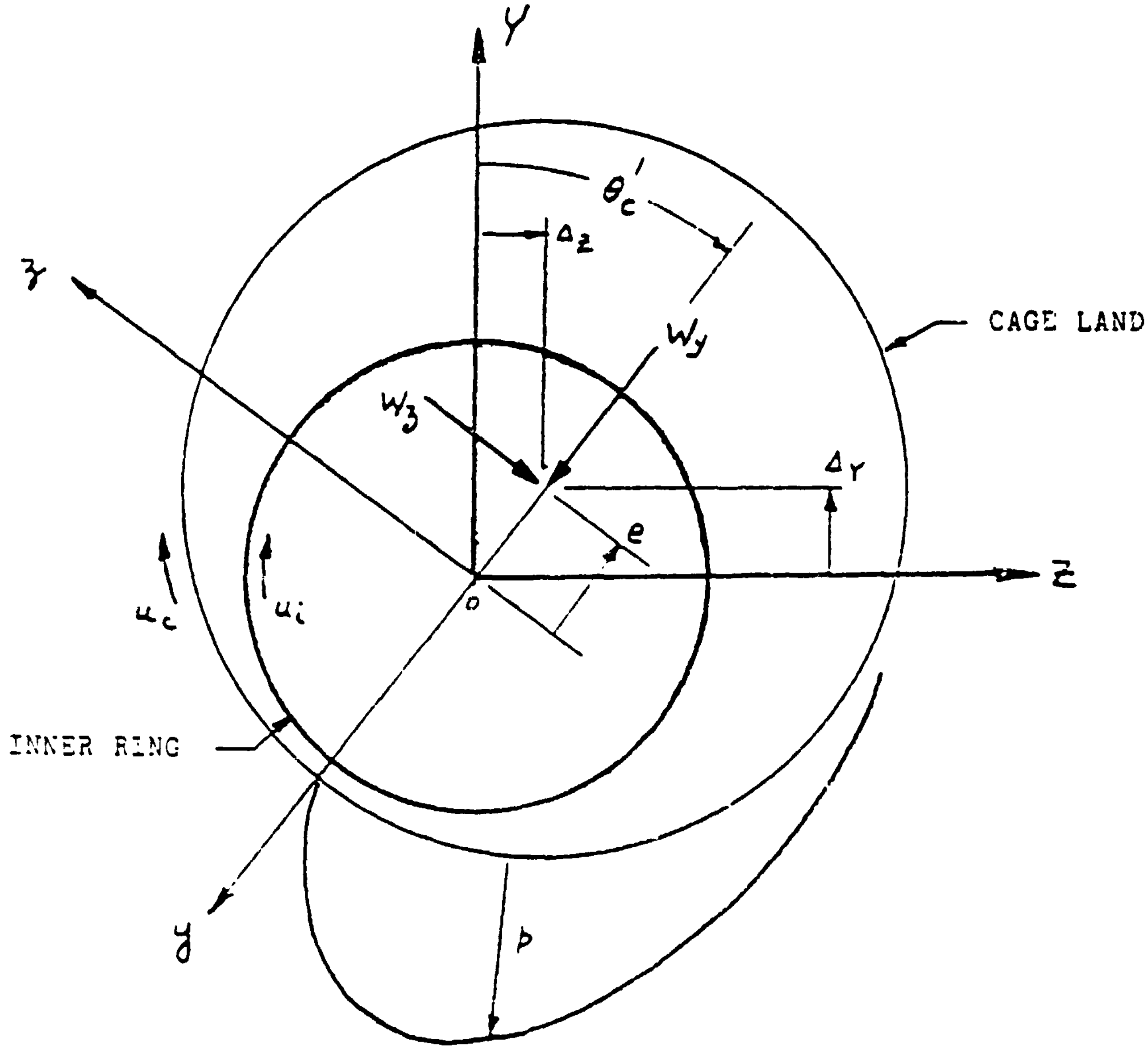
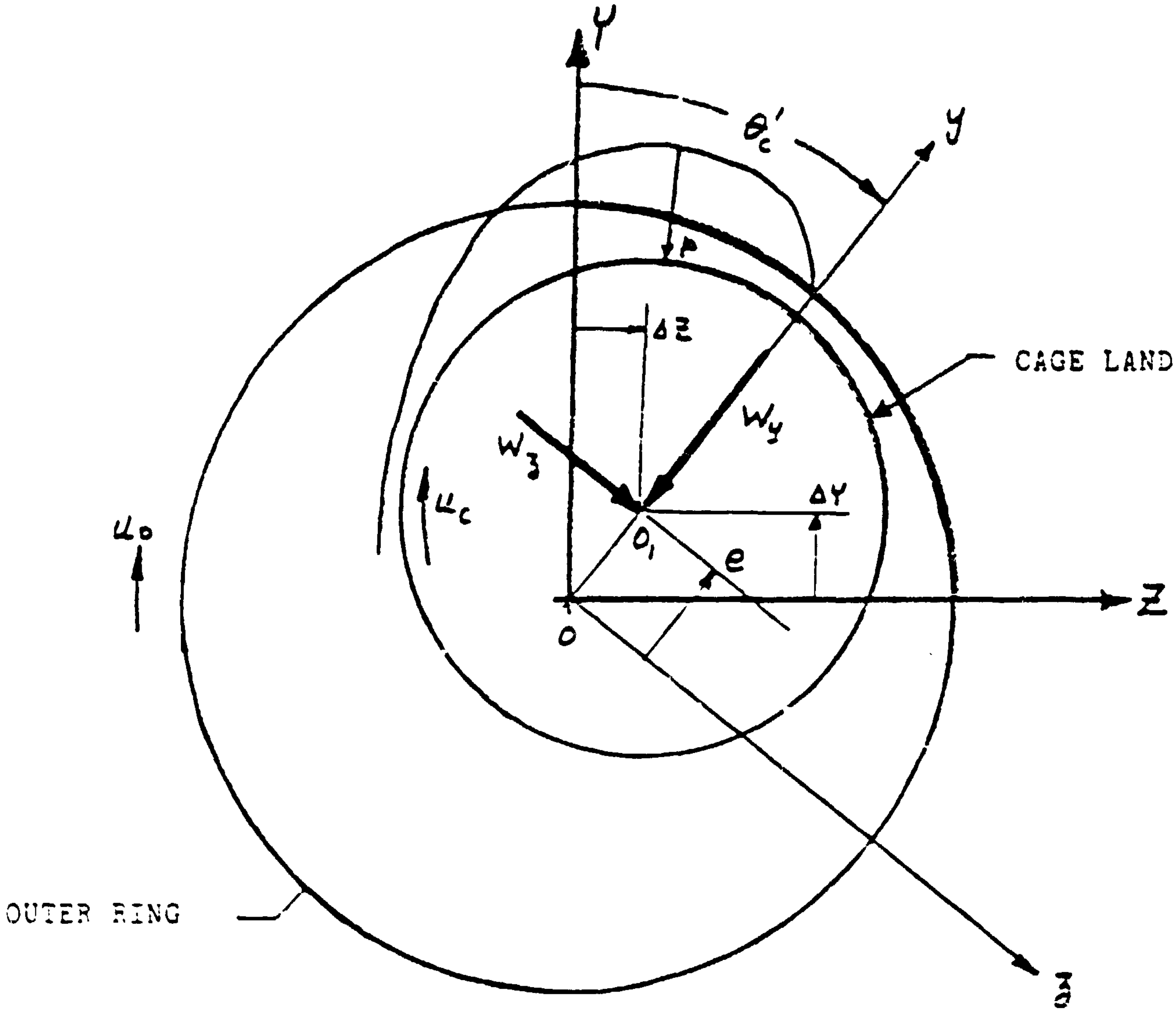


$$\theta_c = \theta_c' + \pi$$

FIGURE E.4 INNER RING-CAGE LAND CONTACT GEOMETRY



$$\theta_c = \theta_c'$$

FIGURE E.5 OUTER RING-CAGE LAND CONTACT GEOMETRY

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

$$\varepsilon = \frac{e}{C} \qquad \text{(E.18)}$$

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

$$M_c' = \pm \frac{\eta_o V_\ell R_\ell^2 L}{C_\ell} \frac{2\pi}{(1-\varepsilon^2)^{1/2}} \qquad \text{(E.19)}$$

The angle θ_c is used to transform forces from the local xyz frame to the cage reference frame:

$$\begin{Bmatrix} M_{cX} \\ F_{cY} \\ F_{cZ} \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\theta_c & -\sin\theta_c \\ 0 & \sin\theta_c & \cos\theta_c \end{bmatrix} \begin{Bmatrix} M_c' \\ F_y' \\ F_z' \end{Bmatrix} \qquad \text{(E.20)}$$

APPENDIX F

SKF AND NASA VERSIONS OF 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 F.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:

1) 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:

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

$$Q = Q_{EHD} + Q_{ASP} \tag{2}$$

where

F is the traction force

Q_{EHD} is the normal load carried by the EHD film

μ_{EHD} is the friction coefficient which develops from lubricant shear

Q_{ASP} is the normal load carried by the asperities

μ_{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 calculating 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

$$\tau = \eta \frac{v}{h} \qquad (3)$$

where

τ is the shear stress

η 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:

$$\eta = \eta_o e^{\alpha \cdot s} \qquad (4)$$

where

η_o is the dynamic viscosity at atmospheric pressure

α 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

$$\tau = \eta \frac{v}{h} \quad \text{if } \eta\frac{v}{h} \leq \tau_c$$

$$\tau = \bar{f}s \quad \text{if } \eta \frac{v}{h} > \bar{f}s \text{ and } \eta\frac{v}{h} > \tau_c$$

where

τ_c is the critical shear stress for which a value of 0.0069 N/mm^2 (1000 psi) is normally used.

$\bar{f}$ is called the lubricant friction coefficient and has been determined for specific lubricants. Typical values of $\bar{f}$ lie in the range $0.05 \leq \bar{f} \leq 0.08$.

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:

1) 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