Bearing Technology: Analysis, Development and Testing

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PREFACE

During the past decade and a half, industry has witnessed a dramatic upturn in the effort expended by bearing manufacturers to refine and enhance familiar plain and anti-friction bearing configurations to meet the performance criteria of present day applications. Stimulated by the fuel crunch of the early 1970's, the contest to effect higher load carrying capacity, achieve greater speeds, reduce friction, heat generation and noise emissions is gathering momentum.

While cost is a major driving force, various branches of bearing technology, especially those associated with classical tribology, computer aided design and performance analysis are having a significant impact on product and process innovation. Examples include advances in material properties and cleanness, elastohydrodynamic lubrication (EHL) and the quantification of operating environmental factors and internal geometry optimization to maximize life and reliability.

The primary focus of this series of papers relates to product design, development and performance analysis. Within this framework, the authors can be seen to address issues ranging from the most fundamental level, such as the automating of basic life analysis, to some of the latest thinking in material science and design fine tuning. While each may be perceived as dealing with a particular bearing type or application concept, the underlying theory and methodology may generally be extended to other product styles and situations.

The reader will also notice a strong emphasis on testing. Fatigue endurance, stress analysis, metallurgical and a variety of other accepted fields of testing are essential to achieving correlation between theory and "real world" practice in the bearing industry, just as it is in virtually every other field of mechanical design. This is particularly true of areas such as bearing ratings, operating torque, deflection and the many parameters impacting the behavior of bearings in service.

As such, these various topics cover a broad spectrum of current activities and provide valuable insight on the current status and future direction of the bearing industry.

Brian J. Cave
The Timken Company

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Selecting Bearings Without a Catalog

Stephen R. Strong and Glenn R. Fahrni, Jr.
The Timken Co.
Canton, OH

ABSTRACT

This paper describes the capabilities and underlying theory of The Timken Company's electronic tapered roller bearing catalog SELECT-A-NALYSIS$_{TM}$. "SELECT-A-NALYSIS," a computer program created to aid designers in selecting and analyzing tapered roller bearing applications, is available through an international time-sharing network.

The manual bearing selection process is reviewed first to determine its ability to find an optimal solution based on the designer's constraints and objectives. This procedure is then examined and tempered by the expertise of bearing application engineers into a new procedure suitable for computerization and, within the constraints of the input, guaranteed to find an optimal solution (or identify the constraint which prohibits a solution). Finally, principles of information management are discussed as a means for improving the efficiency of the computerized selection procedure.

The discussions contained herein pertain to single-row tapered roller bearings supporting a shaft at two positions. Bearing life calculations are limited to the effects of load and speed to simplify the discussion; however, the bearing selection procedure developed can be extended to account for other environmental effects such as load zone, lubrication and alignment.

TM - Service mark of The Timken Company.

DETERMINING BEARING SELECTION OBJECTIVES AND DESIGN CONSTRAINTS

Bearings are selected to satisfy the performance criteria of specific applications or equipment. These criteria may be based on one or more of the following bearing selection objectives: design fatigue life, speed, power requirements, contact stress, bearing system vibration and/or noise, operating temperature and bearing and/or system rigidity. Of these objectives, design fatigue life (or required life) is usually of prime importance in selecting bearings. The most common bearing life requirement falls under one of the following categories:

1. Selections for economy based on a minimum required life.
2. Selections for maximum life in a given bearing envelope size.
3. Selections for minimum bearing envelope size based on a minimum required life.
4. A combination of the above, i.e. selections for economy based on a given bearing envelope size and a minimum required life.

Of the above, the fourth is by far the most common objective and involves selecting a bearing pair that satisfies a number of constraints simultaneously, thereby restricting the number of sets of potential bearing candidates available.

Based on the above list, the basic design constraints typically considered when selecting tapered roller bearings are:

1. An established minimum and maximum envelope size at each bearing position. This may include cone (inner race) bore, cup (outer race) outside diameter and overall bearing width restrictions.
2. A minimum required calculated life for any or each load condition or a minimum required weighted average life.
3. Interpositional constraints, i.e. dimensional constraints between two bearing positions made necessary by assembly or disassembly requirements.
4. Economy of selection, i.e. the least expensive bearing that satisfies all other constraints.

THE TRADITIONAL BEARING SELECTION PROCESS

The traditional, or manual, approach (also referred to as the catalog method) to bearing selection considers bearing load and speed as well as bearing size requirements. Bearing selection for this approach consists of two steps, as follows:

1. Determination of desired or design bearing life, and
2. Selection of a bearing with sufficient basic dynamic load rating to meet the life requirement within specified envelope size constraints.

DETERMINING DESIGN BEARING LIFE - There are various ways in which a machine or vehicle is designed and introduced to the market. Determination of the design bearing life depends on the designer's goals and, to a larger extent, on past experience. As a result, several methods exist that are used to determine the desired bearing life.

When designing a completely new piece of equipment, for instance, the design life of the new machine should be based on the design life of a similar but successful existing machine if one exists. In conjunction with this, the given bearing envelope size requirement for the new machine must be considered.

When upgrading an existing machine to a new and larger model, the same design life is again usually preferred. Increased power throughput however, results in heavier applied loads. Also, the required bearing envelope size may be slightly restricted.

When redesigning machines having overall marginal performance records a somewhat higher life requirement is justified. Here again, the desired or acceptable bearing envelope size range is often less flexible than that of a newly designed machine, depending on the amount of change the design can tolerate.

Finally, in retrofit situations where a new and more reliable field-fix design is required to repair only a specific but defective portion of an existing machine, an even narrower range of envelope size is needed. Again, life requirements must be comparable to, or greater than those of existing equipment.

BEARING SELECTION PROCEDURE - With the design life established, the traditional bearing selection procedure for each shaft in the application generally is as follows:

1. Determine the applied loads and forces in the shaft system and combine them into a single resultant thrust, F_{ae}, on the beam and into radial resultant reactions, F_r, at both bearing positions A and B, using static beam equations, based on an approximate effective bearing spread.
2. Determine the bearing dynamic equivalent radial loads, P_A and P_B, using equations shown in the bearing manufacturer's catalog and assumed bearing radial/thrust load capacity ratios (K-factors), e.g., K=1.50, to approximate the thrust reactions induced by radial loads. Table 1 shows the various equations for determining the dynamic equivalent radial loads, P, dependent on the bearing mounting and thrust condition for designs using two single-row bearings.
3. Calculate the required dynamic radial load rating, C_{90R}, for the bearing design life from the following equation:

$$C_{90R} = f_A P \left(\frac{L_{10} \times S}{1.5 \times 10^6}\right)^{3/10}$$

where: f_A = application factor
P = dynamic equivalent radial load
L_{10} = design life, in hours
S = bearing speed, in rpm

4. Select bearings having basic dynamic radial load ratings, C_{90}, close to the respective calculated required dynamic radial load ratings, C_{90R}, and within other design constraints, e.g., cone bore, cup outside diameter and width.
5. Using the actual K-factors for the bearings selected and the equations from Table 1, calculate the dynamic equivalent radial load, P, on each bearing.
6. Calculate the L_{10} rating life of each bearing using the following equation:

$$L_{10} = \left(\frac{C_{90}}{f_A P}\right)^{10/3} \times \left(\frac{1.5 \times 10^6}{S}\right)$$

where: C_{90} = basic dynamic radial rating of the bearing

7. Check the calculated life against the design life. If the calculated lives are not adequate, select different bearings with greater basic dynamic load ratings, C_{90}, and/or different K-factors. Then recalculate the dynamic equivalent radial load and L_{10} rating lives.
8. Repeat this process until a suitable pair of bearings is found that satisfies the required design life and meets all other constraints.
9. If economy is important, then select several pairs of bearings in order to find the least expensive pair that meets the life and all other constraints.

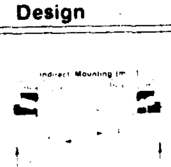
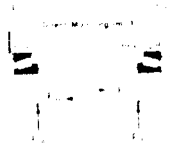
The locations of the effective load centers affect the bearing lives. When statically balancing the forces on a beam supported by tapered roller bearings, the reactions are calculated at the bearing effective load centers. Since the effective centers vary with bearing width and cup contact angle, every selected pair may have its unique spread between supports. For example, if the shaft length between bearing backing shoulders is fixed, the actual bearing reactions may be more

Table 1
DYNAMIC EQUIVALENT RADIAL LOAD EQUATIONS
SINGLE-ROW MOUNTING

To use Table 1 for a single-row mounting, determine if bearings are direct or indirect mounted. Factor "m" is defined as +1 for direct mounted single-row or -1 for indirect mounted single-row bearings. A sign convention is necessary for the external thrust F_{ae} as follows:

a. In case of external thrust applied to the shaft (typical rotating cone application), F_{ae} to the right is positive; to the left is negative.

b. When external thrust is applied to the housing (typical rotating cup application) F_{ae} to the right is negative; to the left is positive.

Design	Thrust Condition	Thrust Load	Dynamic Equivalent Radial Load
	$\frac{0.47 F_{rA}}{K_A} \leq \frac{0.47 F_{rB}}{K_B} - m F_{ae}$	$F_{aA} = \frac{0.47 F_{rB}}{K_B} - m F_{ae}$ $F_{aB} = \frac{0.47 F_{rB}}{K_B}$	$P_A = 0.4 F_{rA} + K_A F_{aA}$ $P_B = F_{rB}$
	$\frac{0.47 F_{rA}}{K_A} > \frac{0.47 F_{rB}}{K_B} - m F_{ae}$	$F_{aA} = \frac{0.47 F_{rA}}{K_A}$ $F_{aB} = \frac{0.47 F_{rA}}{K_A} + m F_{ae}$	$P_A = F_{rA}$ $P_B = 0.4 F_{rB} + K_B F_{aB}$

Note: If $P_A < F_{rA}$, use $P_A = F_{rA}$ or If $P_B < F_{rB}$, use $P_B = F_{rB}$

(or less) than those determined by Step 1 and can, in some cases, disqualify some selections because they no longer meet the design life requirement.

STRENGTHS AND WEAKNESSES: THE MANUAL APPROACH - The manual bearing selection process is a heuristic technique. Bearing selection is reduced to a procedure which can be completed by hand in a few hours with satisfactory results. If the effective center spread remains constant so will the radial reactions. This will reduce the calculations to equivalent radial loads and L_{10} lives for each tapered bearing set. The procedure may be further simplified when selecting radial support bearings where no external thrust is involved. In this case, the equivalent radial loads generally equal the radial reactions and virtually remain unchanged throughout the selection process. Depending on the available time and need for accuracy, a designer could use these relationships to shortcut some of the steps in the detailed selection process discussed previously.

Selecting bearings manually is time-consuming and often frustrating. Searching for the optimal bearing set requires the designer to review pages of information from the manufacturers' catalogs and price schedules. Often, this information is not current or available.

Determining the optimal selection requires analyzing every candidate set, a tedious matter when long hand methods are involved. As a result, the final solution is only the best of all bearing sets considered since few designers have time to consider all possibilities. Moreover, the manual method often lacks sensitivity to design changes and must be reinitiated as the design evolves. A computerized selection procedure addresses many of the traditional drawbacks of the manual method.

AUTOMATING THE PROCEDURE: COMPUTER-AIDED BEARING SELECTION

Despite these shortcomings, seasoned designers (those who understand under what situations the heuristic approach is applicable) often find good bearing selections. They also frequently use bearing performance analysis programs to select bearings by computerizing the catalog equations and testing candidate bearing sets on a trial-and-error basis. The search is initiated by selecting bearings meeting or exceeding the estimated radial rating. The designer may adjust the required radial rating before each trial according to the amount the life constraint was missed on the previous attempt. The human element renders search algorithms based on estimated radial rating unsuitable in automating the bearing selection process.

"SELECT-A-NALYSIS," the bearing selection and analysis program, combines selection logic with an electronic bearing catalog thereby automating the procedure. Whereas the manual method estimates radial ratings as a means of directing the search for bearings, "SELECT-A-NALYSIS" searches for bearings based on envelope constraints. Envelope search methods are not dependent on estimates, making them ideal for computer applications. All applications are analyzed in six successive steps as follows:

1. A coordinate system is established from which all dimensions, bearing positions, load producing elements, and results are identified. This forms a reference frame used in steps two through six.
2. All force vectors and their applied locations are determined and resolved into reactions.
3. All bearings meeting the dimensional envelope requirements are extracted from the database.
4. Bearing sets that meet the interpositional dimensional constraints are identified.
5. All bearing sets that meet required life for each position are selected.
6. Acceptable bearing sets are ranked in order of economy.

Although many of these steps are the same as the manual selection method, the computer algorithm is more systematic, thorough and much faster.

ESTABLISHING A COORDINATE SYSTEM - Computer-based selection algorithms must be sufficiently comprehensive to assure a solution for many different selection situations. The establishment of a coordinate system is the beginning of the automated selection process. Within the coordinate system, every relevant design detail is represented

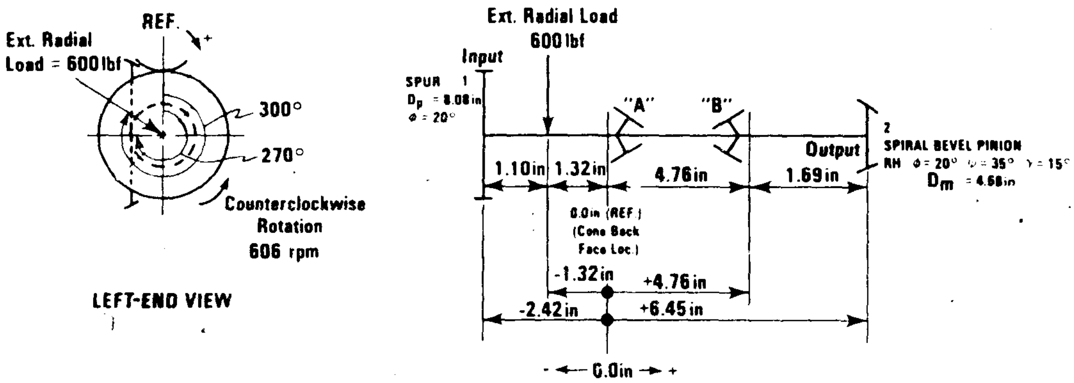


Figure 1: Shaft and bearing sketch and left-end view showing a "SELECT-A-NALYSIS" reference system used in coding the sample problem.

mathematically. This allows the selection process to be translated into a computer code.

"SELECT-A-NALYSIS" users are required to reference their design according to the coordinate system encoded in the program. The user should prepare a sketch of the shaft and bearing system including a left-end view and establish a zero datum point. All bearings and load-imposing machine elements are referenc l according to an axis positive to e right along the shaft. The left end view is used to determine the angularity of gear meshes and applied loads. Figure 1 shows the shaft sketch and left-end view of a sample problem.

The problem description must be provided to "SELECT-A-NALYSIS" in the form of a coded input data file. The coded input data file shown in Figure 2 is the "SELECT-A-NALYSIS" equivalent of the following sample problem description.

SAMPLE PROBLEM DEFINITION - The spiral bevel pinion shaft shown in Figure 1 contains a spur gear overhung to the left. This spur gear is driven and the spiral bevel pinion is a driver. The zero datum point selected is the shaft shoulder of position A and all angles are referenced clockwise from top dead center. The histogram of shaft loading conditions is given as follows:

Cond. No.	Torque lbf-ft	Speed rpm	%time	Radial Load lbf
1	1001	606 ccw	35	0
2	1001	606 cw	35	0
3	700	606 ccw	15	600
4	700	606 cw	15	600

The bearing mounting is indirect, with the cone backface location at both bearing positions defined by the shaft shoulders. The ten least expensive bearing combinations are desired. Bearings are to be selected based on the following dimensional and life constraints:

Constraint	Position A	Position B
BORE	3.125"-3.375"	4.25"-4.5"
O.D.	6.75"-8.0"	6.75"-8.0"
COND. 1	>3000 hours	>3000 hours
WTD LIFE	>5000 hours	>5000 hours

Assembly restrictions require the outside diameter at position A to always be at least .25 inches less than the outside diameter of position B.

```
TITLE SELECTING BEARINGS WITHOUT A CATALOG
TITLE SPIRAL BEVEL PINION SHAFT
SHAFT 1 INPUT
 GEARS
   1 SPUR  -2.42 8.08 20
   2 SPIRALBEVEL 6.45 4.68 20 35 RIGHTHAND RIGHTAPEX 15
 HISTOGRAM TORQUE
   1 1001 -606 35 2 1 -1 0 2 1 270
   2 1001  606 35 2 1 -1 0 2 1 270
   3  700 -606 15 2 1 -1 0 2 1 270
   4  700  606 15 2 1 -1 0 2 1 270
 LOADS
   3 -606 15 1 0 600 300 -1.32
   4  606 15 1 0 600 300 -1.32
 INDIRECT
 SUMMARY 10 BYPRICE WTD
 WTDLIFE 5000
 POSITION CNBF 0.0 A OPP. BEVEL PINION
   TYPE TS
   BORE 3.125 3.375
   OD   6.75  8.0
   LIFE 1 3000
 POSITION CNBF 4.76 B ADJ. BEVEL PINION
   TYPE TS
   BORE 4.25 4.5
   OD   6.75  8.0
   LIFE 1 3000
 INTERPOSITION
   OD (A) + .25 < OD (B)
```

Figure 2: Coded "SELECT-A-NALYSIS" input data file. Details concerning the coding format are included in the "SELECT-A-NALYSIS" Instruction Manual (5)*.

*Numbers in parentheses designate references at the end of the paper.

DETERMINING THE LOCATION AND FORCE VECTORS AND THE BEARING REACTIONS - Gears, pulleys and wheels have established industry standards for calculating force vectors. "SELECT-A-NALYSIS" will accept dimensional data for these common components. In the sample problem, two different gear types are shown. Dimensional data for these gears is entered after the word GEARS in lines 4 through 6 of Figure 2 as follows:

```
line  4   GEARS
line  5     1 SPUR  -2.42 8.08 20
line  6     2 SPIRALBEVEL 6.45 4.68 20 35 RIGHTHAND RIGHTAPEX 15
```

A load array is used to group the location and force vectors contributed by each machine element. The loads produced by every machine element are resolved into a set of force and moment values in each coordinate direction applied to the system at the element's location. Defining the load array which best describes the mechanical component's behavior is the designer's responsibility. Once the forces and moments are defined, they can be coded directly into the "SELECT-A-NALYSIS" input data file.

The load arrays can be generated automatically for each gear mesh according to a histogram of power or torque and speed requirements. This data is entered after the words HISTOGRAM and TORQUE in lines 7 through 11 in Figure 2 as follows:

```
line  7   HISTOGRAM TORQUE
line  8     1 1001 -606 35 2 1 -1 0 2 1 270
line  9     2 1001  606 35 2 1 -1 0 2 1 270
line 10     3  700 -606 15 2 1 -1 0 2 1 270
line 11     4  700  606 15 2 1 -1 0 2 1 270
```

In the sample problem, the 600 lbf external radial load occurring in conditions 3 and 4 is entered following the word LOADS in lines 12 through 14 of Figure 2 as follows:

```
line 12   LOADS
line 13     3 -606 15 1 0 600 300 -1.32
line 14     4  606 15 1 0 600 300 -1.32
```

In the case of multiple operating conditions, each load array must be identified by a condition number. Where standard duty cycles exist, "SELECT-A-NALYSIS" generates load arrays for each condition to aid the designers who benchmark their designs with a standard cycle.

Grouping location and force vectors into load arrays simplifies the task of determining support reactions, especially if the support points are likely to change from bearing set to bearing set. Adjusting for changes in the effective load centers is accomplished using a linear transformation on all vector coordinates as follows:

$$R'_i = R_i - R_a$$

where: R'_i is the translated coordinate of the i th load array
R_i is the coordinate of the i th load array
R_a is the coordinate of the new reference system

The translated zero location, as shown in Figure 3, is the left-most bearing effective center location. As a result, the effective center location of the other bearing (given in the translated system) now is used to find the true effective spread. For each bearing set, the user's reference frame is translated into a new reference frame before the reactions are calculated.

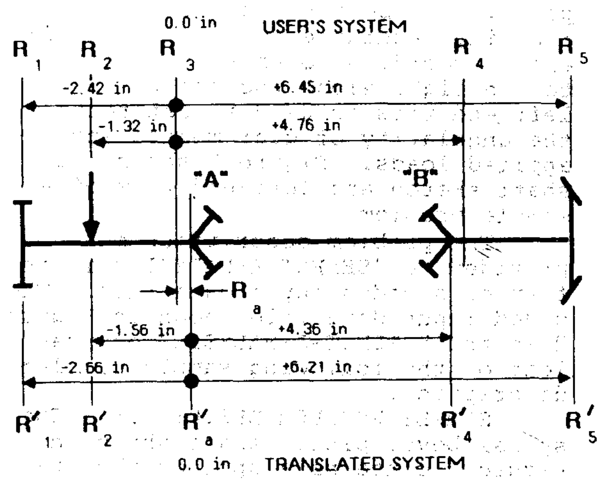


Figure 3: The translated and user-defined reference in the case where the effective center location of position A and B are 0.24 inches and 0.16 inches inside the cone backfaces, respectively.

Bearing reactions for each condition are found using the equations of statics. In three dimensional vector notation, these equations have the following form:

$$\Sigma \vec{r} \times \vec{F} = 0 \quad \Sigma \vec{F} = 0$$

Since moments about the axis of rotation (the i direction shown in Figure 4) do not contribute to bearing reactions, only two equations are applicable. Simplified into the remaining j and k directions, the moment equations are:

$$T_j = (r_k \times F_i) - (r_i \times F_k) + M_j$$

$$T_k = (r_i \times F_j) - (r_j \times F_i) + M_k$$

where: $T_{j,k}$ = moment about R_a caused by load array

$r_{i,j,k}$ = vector coordinates of the load array

$F_{i,j,k}$ = force components of the load array

$M_{j,k}$ = moment components of the load array

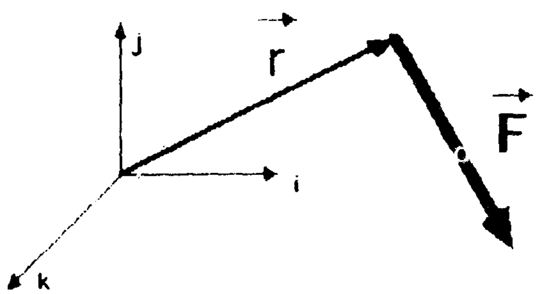


Figure 4: Location and force vectors in 3-D space.

Summation of moments about bearing A is accomplished by accumulating the results of the cross products, T_j and T_k. Dividing the accumulated moments by the effective center spread yields the reactions at bearing B. These reactions are subtracted from the sum of all applied forces in each vector to yield the reactions at bearing A.

SATISFYING ENVELOPE CONSTRAINTS: COLLECTING BEARINGS BY DIMENSIONS - Suitable candidate bearings at each position are defined as those meeting the dimensional envelope constraints. Traditionally, manufacturers' catalogs list bearings sorted by bore. In "SELECT-A-NALYSIS," the manufacturers' catalog has been replaced by a database containing over 26,000 cone and cup part number combinations. The program can retrieve bearings from the database by any one or combination of five characteristics: cone number, cup number, series number, bore and outside diameter. Figure 5 shows a Venn diagram of bearings selected by bore and O.D.

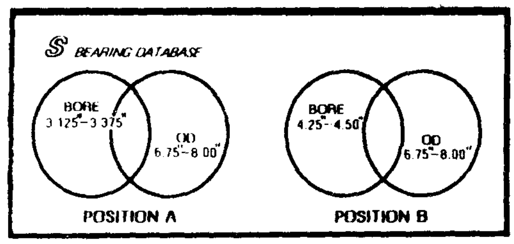


Figure 5: Venn diagram illustrating the subset of bearings suitable at each position based on envelope constraints.

Experience shows these characteristics are the best criteria for retrieving candidate bearings for a variety of applications. By specifying a particular part number the designer may analyze an existing design. However, in a design requiring bearing selections, bore and O.D. range specifications direct "SELECT-A-NALYSIS" to start the search for suitable bearings. At least one of these characteristics must be specified at each bearing position.

In the sample problem, each position is restricted to part numbers within specific bore and outside diameter ranges. All envelope constraints are entered following the word POSITION as shown in figure 2 as follows:

```
line 18    POSITION CNBF 0.0 A OFF. BEVEL PINION
line 19       TYPE TS
line 20       BORE 3.125 3.375
line 21       OD    6.75  8.0

line 23    POSITION CNBF 4.76 B ADJ. BEVEL PINION
line 24       TYPE TS
line 25       BORE 4.25 4.5
line 26       OD    6.75  8.0
```

Other constraints are not as well suited for bearing search characteristics; however, they are considered when creating a list of candidates at each position. "SELECT-A-NALYSIS" includes these additional constraints: overall bearing width, radial rating, K-factor, backing shoulder diameters and filet radii.

SATISFYING INTERPOSITIONAL CONSTRAINTS: FORMING BEARING SETS - "SELECT-A-NALYSIS" finds acceptable bearing sets by combining candidates from each position, which have met the prescribed envelope, and testing them against the interpositional dimensional constraints as in the Venn diagram shown in Figure 6. Interpositional constraints are given in the form of an equation. Assembly restrictions in the sample problem

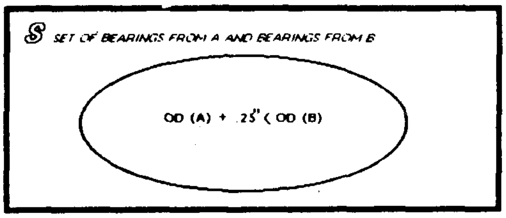


Figure 6: Venn diagram illustrating the subset of bearings suitable at each position based on envelope constraints and also meeting the assembly restrictions.

state that the outside diameter at position A must always be at least .25 inches less than the outside diameter at position B. This relationship is coded as the equation following the word INTERPOSITION in Figure 2 as follows:

```
line 28    INTERPOSITION
line 29      OD (A) + .25 < OD (B)
```

All bearing sets not meeting this constraint are rejected.

SATISFYING LIFE CONSTRAINTS: *FINALIZING THE FEASIBLE SET* - Two types of life constraints may exist for each position: 1) life in any one load condition, i.e. the life based on the load and speed of that loading condition and 2) weighted average life for a load cycle, i.e. the weighted average of the individual condition lives. "SELECT-A-NALYSIS" satisfies these constraints by first calculating the life expected for a given operating condition and comparing it to the life requirement. Bearing sets not meeting this criterion are rejected. Figure 7 shows a Venn diagram of this. During this process, the life depletion rate is accumulated and, as soon as the accumulated life falls below the weighted life requirement, the set is rejected. In this manner, "SELECT-A-NALYSIS" can determine if the weighted life requirement cannot be met before every load condition is reviewed. Lines 17, 22 and 27 of Figure 2 specify the coding format for life requirements as follows:

```
line 17    WTDLIFE 5000
line 22      LIFE 1 3000
line 27      LIFE 1 3000
```

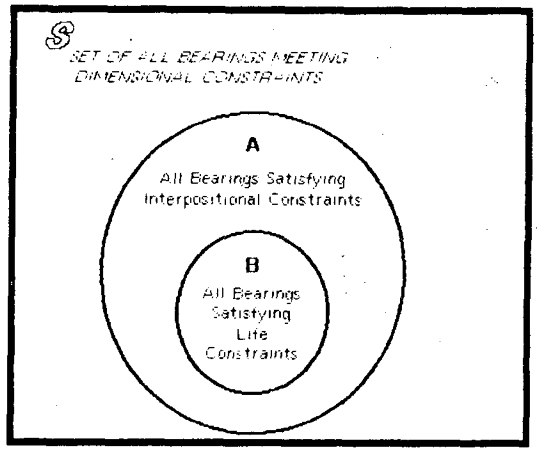


Figure 7: Venn diagram illustrating the subset of bearings suitable at each position based on envelope constraints and also meeting the assembly restrictions and also meeting the life constraints.

RANKING ACCEPTABLE BEARING PAIRS *TO FIND THE OPTIMUM* - Bearing sets meeting all the constraints are ranked by total price. ("SELECT-A-NALYSIS" also allows ranking based by maximum system life.) The optimal solution cannot be determined until every bearing set is constructed and tested. The specification of the optimization criterion appears in line 16 of Figure 2 as follows:

```
line 16    SUMMARY 10 BYPRICE WTD
```

"SELECT-A-NALYSIS" presents the results in a bearing selection summary list shown in Figure 8. From this list of bearing sets, the designer makes his final selection considering any pertinent additional information.

ADVANTAGES OF THE COMPUTERIZED APPROACH - "SELECT-A-NALYSIS" can solve most bearing selection problems in less time and with greater accuracy than the manual method. Since the computer's search is exhaustive and does not drop a bearing set from consideration until it cannot meet a constraint, an optimal selection is assured. However, there is no guarantee that for every problem a solution exists. In this case, "SELECT-A-NALYSIS" can identify the constraints that are prohibiting the solution, thereby allowing the designer to use this information in developing a feasible design. In contrast is the situation wherein the

```
                    BEARING SEARCH SECTION ENTERED
<##><##><##><##><##><##><##><##><##><##><##><##><##><##><##><##><##><##>

 16 PART NUMBERS FOUND FOR POSITION: A OPP. BEVEL PINION       TYPE TS

 69 PART NUMBERS FOUND FOR POSITION: B ADJ. BEVEL PINION       TYPE TS

<##><##><##><##><##><##><##><##><##><##><##><##><##><##><##><##><##><##>

BEARING CATALOG SELECTION SECTION :      1104. BEARING PAIRS REVIEWED.

 BEARING SELECTION SUMMARY LIST
================================
```

PAIR NO.	POS	TIMKEN CO. PART NUMBERS CONE NO.	CUP NO.	CONE BORE INCHES	CUP OD INCHES	BRG. WIDTH INCHES	L-10 WTD LIFE HOURS
1	A	677	674	3.3750	6.7500	1.6250	23863
	B	64450	64708	4.5000	7.0856	1.6250	12891
2	A	9385	9321	3.3125	6.7500	1.9375	125282
	B	64450	64708	4.5000	7.0856	1.6250	10147
3	A	677	674	3.3750	6.7500	1.6250	29388
	B	JHM522649	JHM522610	4.3307	7.0866	1.8504	20031
4	A	677	674	3.3750	6.7500	1.6250	28111
	B	71450	71750	4.5000	7.5000	1.8750	19657
5	A	9385	9321	3.3125	6.7500	1.9375	135571
	B	JHM522649	JHM522610	4.3307	7.0866	1.8504	14891
6	A	9385	9321	3.3125	6.7500	1.9375	130573
	B	71450	71750	4.5000	7.5000	1.8750	14703
7	A	677	674	3.3750	6.7500	1.6250	26942
	B	X32222	Y32222	4.3307	7.8740	2.2047	41821
8	A	9385	9321	3.3125	6.7500	1.9375	123915
	B	X32222	Y32222	4.3307	7.8740	2.2047	31309
9	A	HH221431	HH221410	3.1250	7.5000	2.2500	111154
	B	71450	71788	4.5000	7.8730	1.9488	21211
10	A	90334	90744	3.3465	7.4375	2.0983	211835
	B	71450	71788	4.5000	7.8730	1.9488	14252

\##

Figure 8: "SELECT-A-NALYSIS" bearing selection summary list for the problem coded in Figure 2.

program is loosely constrained and the number of feasible bearing sets may be extremely large. In this case, the bearing search and selection algorithms employed must be efficient.

IMPROVING SEARCH AND SELECTION EFFICIENCY

Up to now, the computer algorithm is regarded as the sequential execution of six independent steps. This method yields a solution but uses the information at hand inefficiently. "SELECT-A-NALYSIS" approaches the bearing selection problem as six interdependent stages, exploiting the interrelationships of the information contained in earlier stages to more effectively complete the objectives of a later stage. Some of the ways "SELECT-A-NALYSIS" combines information from multiple stages include:

1. Ranking the conditions in a duty cycle by severity of load. This reduces the amount of calculations required before the program determines that the selection provides inadequate life.
2. Selecting only one bearing candidate per series at each bearing position in combination with a given series at the opposite position minimizes the number of bearing sets constructed with identical lives.
3. Estimating a conservative minimum radial rating for each bearing position to eliminate the candidate parts that would not meet the life constraints under the best of conditions.
4. Ranking candidate bearings at each position in order of economy and specifying a limit to the number of bearing sets listed in the final selection so as to direct the algorithm toward the most economical solution without checking every feasible combination.

In all cases, these techniques allow the algorithm to determine if a constraint has been violated with the minimum amount of computation.

CONCLUSION: CAPTURING THE KNOWLEDGE OF EXPERTS

Automated bearing selection programs like The Timken Company's "SELECT-A-NALYSIS" contain two essential elements: the bearing selection logic developed by experts and a comprehensive bearing database (an electronic version of the manufacturer's catalog).

The unique "SELECT-A-NALYSIS" input data file format provides for an English-like problem description. This allows designers to describe the application to the computer in the language used daily and not as an endless list of numbers. Thus the tedious manual selection process is reduced to a computer input data file, which states the design problem and selection goals. Designers can leave the task of bearing selection to the program and concentrate on improving their designs.

ACKNOWLEDGEMENTS

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Spherical Plain Bearings for On and Off Road Vehicles

Sieghart Sautter
SKF Kugellagerfabriken GmbH
Howard R. Haden
SKF Industries, Inc.
Bernd Kottwitz
SKF Kugellagerfabriken GmbH

ABSTRACT

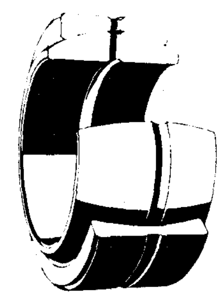
Spherical plain bearings have become increasingly popular for industrial applications throughout the world. This has resulted in development of many types to meet various demands such as high load capacity, longer life, and maintenance-free operation. New experience has been gained in both the laboratory and the field. There are many factors that must be considered by the designer to enable proper selection of bearing type and correct evaluation of expected performance. This is especially true for tough applications such as on and off road vehicles.

SPHERICAL PLAIN BEARINGS are standardized, ready-to-mount units which belong to the family of dry plain bearings. They are - in contrast to hydrodynamic or hydrostatic plain bearings - suitable for heavy loads at relatively low sliding speeds.

Due to their design, figure 1, these bearings are used where alignment movements between shaft and housing must be accommodated, or for oscillating and tilting movements. Elastic deformation of the structure, bending of the shaft, or other misaligning factors sometimes cannot be avoided and cylindrical bushings suffer severe corner loads (figure 2) and heavy local wear.

For oscillating movements and small oscillating angles, use of rolling bearings often leads to brinelling and the bearings fail. Rolling bearings can be considered generally as bearings for rotation, whereas spherical plain bearings are for oscillation.

Spherical plain bearings are classified in two groups - "Maintenance-required" types are metal-to-metal contact requiring lubrication. "Maintenance-free" or self-lubricating types do not require lubrication.



Steel-on-steel spherical plain bearings
Figure 1

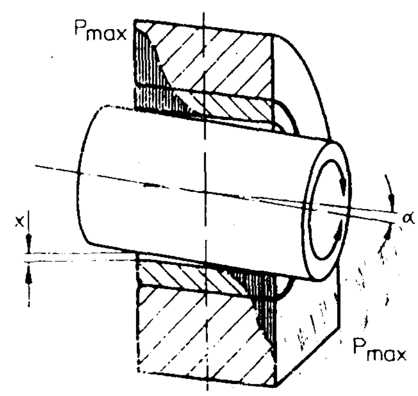


Figure 2

For metal-to-metal bearings, progress in the field of surface treatment has opened many new applications and there is no end in sight to this development.

Modern plastics, fabrics, or composite materials have enabled sliding layers to be developed permitting freedom from maintenance in operation. Many factors are of extreme importance to the designer and will be discussed.

SPHERICAL PLAIN BEARINGS : ADVANTAGES

Spherical plain bearings can be a technically better solution and can represent a cost savings if all factors are considered by the designer.

SPHERICAL PLAIN BEARINGS - Permit universal motion in all directions, figure 3. Cylindrical bushings only permit rotational movements and misalignment or tilting can cause corner loading, reduce bearing life due to excessive wear, and create high stresses in shafts and frame.

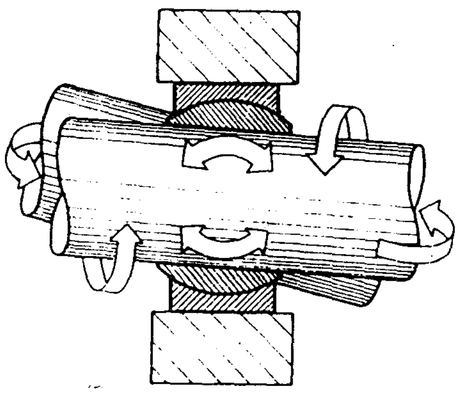


Figure 3

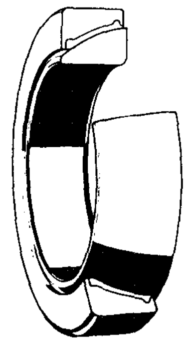
- Eliminate stress concentrations and premature wear.
- Allow freedom of deflection, such as shaft or frame bending, and compensate for errors of alignment. This allows more flexible designs and savings in weight.
- Simplify the assembly process by time-saving ease of assembly, eliminating the necessity for precision line boring, compensating for misalignment caused by tolerance build-up.

TYPES OF SPHERICAL PLAIN BEARINGS

Spherical plain bearings are manufactured in three basic designs; radial, angular contact, and thrust. Different combinations of sliding contact surfaces are possible with each. Each bearing design is suitable for certain applications. The choice of bearing, or combination of bearings, depends on a number of factors.

RADIAL BEARING (figure 1) - Radial types are primarily for heavy radial load but can take some thrust in both directions. The assembly forms are a non-separable unit. This type of bearing is not suitable for pure thrust loads. Combined loads are permissible if the thrust load does not exceed 30% of the radial load - special attention is recommended if thrust exceeds 10% of the radial load.

ANGULAR CONTACT BEARINGS (figure 4) - The sphered sliding contact surfaces of the angular contact bearings are inclined at an angle to the bearing axis. Consequently, the bearings are suitable for combined radial and axial loads.



Angular contact spherical plain bearings
Figure 4

The bearing unit is separable. Single mounted units can support thrust acting only in one direction. When two angular contact bearings are arranged, a radial bearing can be produced which is capable of carrying heavy radial loads and heavy thrust loads in both directions. In this case, special attention is required due to the very rigid design of the surrounding part, wear in the bearings may eventually cause loss of preload and instability in the system.

In special cases, bearings consist of a one piece inner ring and two separate outer rings (figure 5). The outer rings can be adjusted using spacer rings in order to achieve a defined axial clearance.

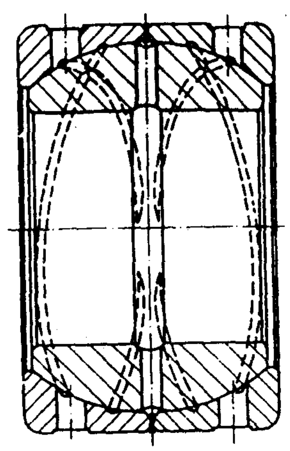


Figure 5