An Expert System for Designing Hydrodynamic Journal Bearings

by

Mohammad Affan Badar

A Thesis Presented to the

FACULTY OF THE COLLEGE OF GRADUATE STUDIES

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DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
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This thesis, written by

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Date 18/11/1993

بسم الله الرحمن الرحيم

To my family members

and

well wishers

ACKNOWLEDGEMENTS

"In the name of Allah (God), Most Gracious, Most Merciful. Read, In the name of thy Lord and Cherisher, Who created. Created man from a {leech-like} clot. Read, and thy Lord Is Most Bountiful. He Who taught {the use of} the pen. Taught man that Which he knew not. Nay, but man doth Transgress all bounds. In that he looketh Upon himself as self-sufficient. Verily, to thy Lord Is the return {of all}." (The Holy QURAN. Surah no. 96)

Above and first of all, I thank and pray to Allah, the Almighty, for His guidance and protection throughout my life including the years of this study. Praise be to Him, with Whose gracious help, it was possible to accomplish this task. I am happy to have had a chance to glorify His name in the sincerest way through this small accomplishment and I ask Him to accept my efforts.

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THESIS ABSTRACT

NAME OF STUDENT	***: MOHAMMAD AFFAN BADAR***
TITLE OF STUDY	***: An Expert System for Designing Hydrodynamic Journal Bearings***
MAJOR FIELD	***: Mechanical Engineering***
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Different methodologies for designing hydrodynamic journal bearings are reviewed and an integrated and dependable design methodology is developed. Various architectures used in design expert systems are reviewed and an architecture consisting of several stages of grouped rules is employed. The architecture uses a rule-based production system. Relevant objective knowledge is represented appropriately. In this Turbo Prolog program, initially, the user is asked to enter the required prescriptive and functional specifications and the decision-criterion. The design solutions of successive stages are examined for any violations of the design constraints. Utility value of each of the final solutions is calculated and the design solutions having utility values above a certain limit are printed in the output file. The expert system makes an exhaustive search for all the solutions. The results of this expert system are compared with those of the previous workers to illustrate its usefulness.

MASTER OF SCIENCE DEGREE

KING FAHD UNIVERSITY OF PETROLEUM AND MINERALS

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January, 1993

خلاصة الرسالة

إسم الطالب : محمد عفان بدر
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في هذا البحث تم مراجعة الطرق المختلفة لتصميم مساند الارتكاز الهيدروديناميكية لأعمدة نقل الحركة ، كما تم تطوير طريقة تصميم متكامله يمكن الاعتماد عليها . ولقد تم ايضاً مراجعة طرق البناء المختلفة التى تستخدم في تصميم الانظمة الماهره وبناء على ذلك تم اختيار واستخدام احد اساليب البناء والذي يتكون من عده مراحل من القواعد المجمعه .

واسلوب البناء هذا يستخدم نظام إنتاج قائم على اسس مقننه . وقد تم تمثيل المعلومات الموضوعية (الواقعية) ذات الصله بطريقة مناسبة .

ويتم العمل بهذا النظام (الذي يستخدم برنامج تربو – لوج) بالطريقة الآتيه . أولاً يطلب البرنامج من المستعمل ادخال الوظيفة المطلوبه والمواصفات الوظيفية بالاضافة الى الخاصية التى سيتخذ على اساسها القرار ، ثم يتم اختبار الحلول المقترحة للتصميم اثناء المراحل المتتابعة من حيث وجود اي اخلال بقيود التصميم الموضوعه . بعد ذلك يتم حساب معامل الاستفادة لكل حل من الحلول النهائية وأخيراً يتم طباعة تلك الحلول التى لها معامل استفادة اكبر من حد معين في ملف خاص بالنتائج .

ويقوم النظام الماهر المقترح في هذه الدراسة بعمل بحث شامل لكل الحلول . وقد تم مقارنة النتائج التى تم الحصول عليها من النظام الحالي بتلك النتائج التي سبق ان حصل عليها آخرون لتوضيح مدى فائدة النظام المقترح حالياً .

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NOMENCLATURE

C	:	radial clearance, m
C_H	:	lubricant specific heat, $J/kg\text{-}^\circ C$ (average value = 1760 $J/kg\text{-}^\circ C$)
D	:	journal diameter, m
e	:	journal displacement or eccentricity, m
F	:	friction force on journal, N
f	:	coefficient of friction (= F/W), dimensionless
F_{MB}	:	journal torque parameter = $(T/WR)\{P/\omega\mu\}^{1/2}$, dimensionless
$F.S.$	:	factor of safety, dimensionless
g	:	acceleration due to gravity = 9.81 m/s^2
H	:	power loss, W
h	:	variable film thickness, m
h_0	:	minimum film thickness, m
H_{MB}	:	minimum film thickness parameter = $(h_0/R)\{P/\omega\mu\}^{1/2}$, dimensionless
J	:	mechanical equivalent of heat, J
L	:	axial length of bearing, m
N	:	speed, rps
P	:	load per unit projected bearing area (= W/LD), Pa
p	:	pressure developed in lubricant film, Pa
p_l	:	long-bearing pressure, Pa
p_{max}	:	maximum pressure developed in lubricant film, Pa
p_s	:	short-bearing pressure, Pa

Q	:	flow of lubricant drawn into clearance space by journal, m^3/s
Q_s	:	flow of lubricant out both sides of bearing (side leakage), m^3/s
R	:	journal radius, m
r^2	:	coefficient of determination, dimensionless (Secs. 5.1 and 7.1)
S	:	bearing characteristic number or Sommerfeld number $\{=(R/C)^2\mu N/P\}$, dimensionless
T	:	torque required, N.m
T_{in}	:	lubricant inlet temperature, °C
T_{op}	:	lubricant operating or average temperature, °C
T_{out}	:	lubricant outlet temperature, °C
ΔT	:	temperature-rise, °C
ΔT_t	:	trial value of temperature-rise, °C
U	:	utility value of a solution, dimensionless
u	:	utility value of a decision variable, dimensionless
V	:	surface speed of the shaft, m/s
W	:	load, N
W_R	:	load component along line of centers, N
W_T	:	load component normal to line of centers, N
w	:	weighting factor of a decision variable, dimensionless
x	:	coordinate in the direction of motion
z	:	coordinate in the axial direction
α	:	leading angle extending from start of film to the line of action of the load, deg
β	:	angular extent of bearing arc, deg
ε	:	eccentricity ratio ($=e/C$), dimensionless

μ	:	absolute or dynamic viscosity, Pa-s
ω	:	speed, rad/s
ω_s	:	dimensionless speed $= \omega\sqrt{C/g}$
Φ	:	position of minimum film thickness or attitude angle, deg
ρ	:	lubricant density, kg/m^3 (average value $= 861\ kg/m^3$)
θ	:	angular coordinate measured in direction of rotation, deg
θ_A	:	angle from line of centers to start of film measured in direction of rotation, deg
θ_{cav}	:	angular position to cavitation boundary, deg (Appendix)
θ_{Pmax}	:	position of maximum film pressure, deg
θ_{PB}	:	position at which film terminates, deg
ζ	:	axial dimension $= z/(L/2)$, dimensionless (Appendix)

Note

The followings are used as either subscripts or abbreviations:

max.	:	maximum
min.	:	minimum

CHAPTER 1

INTRODUCTION

1.1 EXPERT SYSTEMS

Expert systems, also called knowledge-based systems, comprise one of the forefront areas of the exciting field of artificial intelligence (AI) [1]. AI is the study of how to make computers do things at which, currently, people are better[2]. This assertion implies that humans not only process information but they understand it as well because they are able to relate the information to others and come up with new ideas. Another way of defining AI is that its efforts focus on the development of computer-based systems (hardware and software) capable of accomplishing tasks and solving problems by using knowledge, reason, and intuition, capabilities normally thought of as belonging solely to humans [3]. AI is helping researchers in understanding problem solving abilities of humans and developing computer programs that use knowledge bases intelligently to solve practical problems [4-5].

Expert systems are sophisticated computer programs that manipulate domain specific knowledge to solve problems intelligently in a narrow problem area by using a search-inference framework [6-7]. The process of building an expert system is termed as knowledge engineering whereas the person who designs and builds the expert system is called a knowledge engineer [7]. Since an expert system (ES) uses domain specific knowledge bases, it is often called a knowledge based expert system (KBES). Since a KBES uses a search-inference

framework to solve a problem, it 'thinks' [8] like a domain specific expert. A KBES consists of domain specific knowledge bases, an inference engine, and a user interface [A1]. The user interface provides necessary explanation of the actions of KBES if asked by a non-expert user. If the program is designed to help an expert, it is called a design assistant (DA) where the user interface need not be as elaborate as that of a KBES [6].

An expert system[9-10] acquires knowledge through knowledge-acquisition software tools from a trained specialist called knowledge engineer, who, in turn, acquires the knowledge from one or more experts in the specific field called domain experts. Once the expert system has been developed, this participation of domain experts could be ended. Since the domain normally continues to develop, the domain expert's advice is occasionally sought. A prime advantage of expert systems is that they capture the knowledge of experts that may otherwise be lost through death or retirement. Moreover, they can contain the cumulative knowledge of several experts, they are available any time of day or night, and they can be distributed widely throughout an organization. In addition, they have the ability to save design engineers from having to carry out redundant tasks, improve the consistency of designs, search large databases for optimal design, reduce the cost of design while at the same time improving quality, and reduce manual errors and produce more reliable designs. However, it should be noted that expert systems are not a substitute for a human expert. Unless a problem is fully understood, which can come only from humans, the expert system project will fail.

The user of an expert system works through a keyboard interface in a formal language. As a natural-language processing ability becomes available, it

can make this interchange very friendly. Eventually, speech recognition and generation may replace the keyboard. The input consists of system facts and suppositions of varying degrees of validity. The user interface tends to be highly interactive, following the format of a question and answer session. The expert system returns answers, recommendations, or diagnoses.

In a design expert system a graphics interface may be required in order to visualize the object being designed. At present most expert systems utilize a specialized database, but they are moving rapidly to be able to draw upon generalized databases.

The elements of an expert system are shown in Fig. 1.1 [1]. The two major divisions are the knowledge base and the inference engine. The knowledge base is unique to a particular domain but the inference engine may be common to many domains of knowledge. A 'knowledge acquisition facility' is the component responsible for entering the knowledge into a database. At its simplest level this facility acts as an editor and knowledge is entered directly in a form acceptable by the language of the expert system. The 'assertions' component, also called the working memory or temporary data store, contains the knowledge about the particular problem being solved. Facts are represented by predicate logic, frames, or semantic networks. The 'knowledge relationships' component contains formulae showing the relationship among several pieces of information. The 'if-then' production rule is the most common relationship. This rule has the form:

IF condition THEN action

In any knowledge base, there is a balance between the factual or declarative knowledge and the rules for manipulating those facts (procedural knowledge).

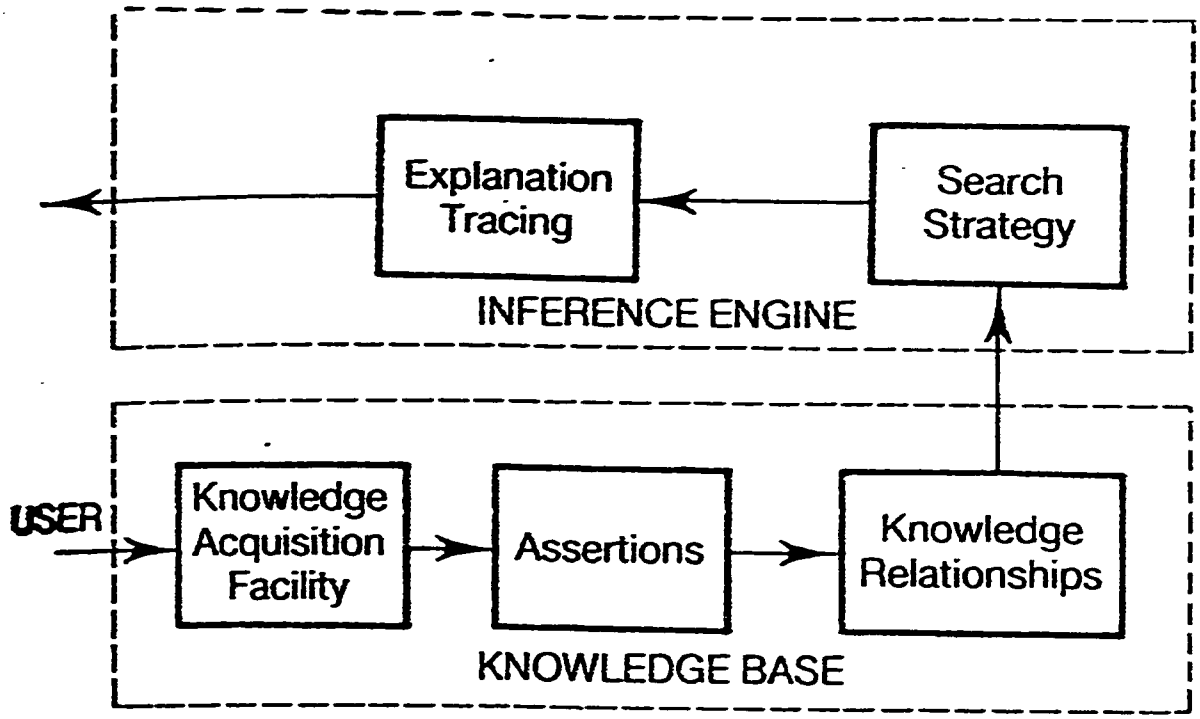


Fig. 1.1 General Structure of an Expert System [1].

The inference engine contains the control mechanisms for the expert system. In a production system the AI reasoning is responsible for choosing which rule to perform next. This constitutes the 'search strategy'. Since the probability of one rule following another is less than 100 percent in most cases, it has been necessary to incorporate uncertainty into the rules. With a small number of rules, it is practical to search in random order, but if the number of rules is large it is necessary to partition into sublists on some logical basis. 'Explanation tracing' is provided to retrace the chain of production rules that led to the development of the system. This greatly enhances the credibility of the expert system.

Special computer languages were developed initially for expert systems. LISP (a list processing language) and PROLOG (programming in logic) are most commonly used with the early expert systems, but more recently expert systems have employed more general purpose languages like FORTRAN, PASCAL, and C. Higher level programming languages have also been developed. The most common are OPS-5, KEE, and ART.

As work on expert systems evolved, researchers noticed that the knowledge base dealing with facts and if-then rules was specific to the problem domain, but the inference engines were similar in different problem domains of the same type. This led to development of expert system shells, or expert system building frameworks, which require little programming.

1.2 DESIGN

A mechanical-design problem can be stated as follows [11]: Given a design goal (need), a set of design data, and a set of design constraints, find a plan that

creates a component and/or a system that satisfies both the goal and constraints. This component or system constitutes the solution to the design problem at hand and requires optimization and improvements. Critical improvements are usually performed based on market or customer feedback.

Alternatively, one may define design as a goal-directed problem solving activity which results in creating or modifying or selecting system(s) or component(s) [6]. This definition of design incorporates the three classes of design problems: A creative or innovative solution is required for a class I design problem. A class II design problem is solved by improving an existing design. A class III design problem is a routine design problem which involves a direct selection of standard system(s) or component(s) [12]. A class II design problem which requires some creative thinking, is also known as an adaptive design problem [13].

In terms of problem-solving, the above three classes can be identified as follows [14]: Class III problem solving in which the way to decompose a design problem is already known and compiled 'design plans' are available for each major stage in design, class II in which the components of the object being designed are known but design plans are not available in a compiled form, and class I in which even the components are unknown.

The phases of the design process from start to finish are: recognition of a need, problem definition, synthesis, analysis and optimization, evaluation and documentation. The synthesis and analysis phases are the most important. Synthesis is the process by which the merits and quality of design solutions are studied and evaluated. Analysis is the process by which quantitative evaluations of the design solutions are performed. If the analysis reveals that a design

solution fails, the synthesis procedure must begin again.

An expert identifies a class III design problem very quickly as a familiar problem. Therefore, he defines or redefines the problem very clearly and develops a perceptual representation [6] of the solution to the problem. Thus he uses a top-down approach in solving a class III design problem. An expert cannot fully recognise a class II design problem as a familiar one. His perceptual image of both the problem and the solution is only partially intelligible. He mixes a large amount of top-down strategy and a small amount of bottom-up strategy in performing the adaptive design which is required by the class II design problem. An expert is generally disturbed by a class I design problem. He knows that he is looking at an ill-defined problem; he attempts to redefine the problem. He fails to construct a perceptual embodiment of a major part of the problem immediately. Therefore he pursues a bottom-up technique in solving this problem.

1.3 HYDRODYNAMIC JOURNAL BEARINGS

The design of journal bearings is of considerable importance to the development of rotating machinery. Journal bearings are essential machine components to compressors, pumps, turbines, internal-combustion engines, motors, generators, etc., [15-16]. In a journal bearing [15-17] a shaft or journal rotates or oscillates within a close-fitting cylindrical sleeve (the bearing) and the relative motion is sliding. The journal and bearing surfaces are separated by a film of lubricant (liquid or gas) that is supplied to the clearance space between the surfaces. The clearance space [15-16] permits assembly of the journal and bearing, provides space for the lubricant, accommodates unavoidable thermal expansions and tolerates any shaft misalignment or deflection. The basic

purpose of a journal bearing is to provide radial support to a rotating shaft. Under load, the centers of the journal and the bearing are separated by a distance called eccentricity. This eccentric arrangement establishes a converging-wedge geometry, which in conjunction with the relative motion of the journal and the bearing permits a pressure to be developed by viscous effects within the thin film of lubricant and thus produces a load-carrying capability.

Journal bearings are termed full bearings [15-16] when the bearing surface completely surrounds the journal (Fig. 1.2a). Because they are inexpensive and easy to make, full journal bearings are the most commonly used. Journal bearings are called partial bearings [15-16] when the bearing surface extends over only a segment of the circumference, generally 180°, 120°, and 60° (Fig. 1.2b - 1.2d). Partial bearings are used where the load is unidirectional. Partial journal bearings in which there is no radial clearance are termed fitted bearings, while those in which there is clearance are called clearance bearings [15-16]. They may also be classified as centrally or eccentrically loaded depending upon whether the load line bisects the bearing arc or divides it eccentrically [18].

A lubricant [17] is any substance which, when inserted between the moving surfaces, reduces friction, wear, and heating of machine parts which move relative to each other. Lubricating material has both a very low shear strength (hence it has a low coefficient of friction) and a sufficient film strength to hold the two surfaces apart when the film is thin [19].

Lubrication mechanisms [17] may be of five types: hydrodynamic, hydrostatic, elastohydrodynamic, boundary, and solid-film, on which classification of journal bearings depends. Hydrodynamic or full-film or fluid lubrication means that the load-carrying surfaces of the bearing are separated

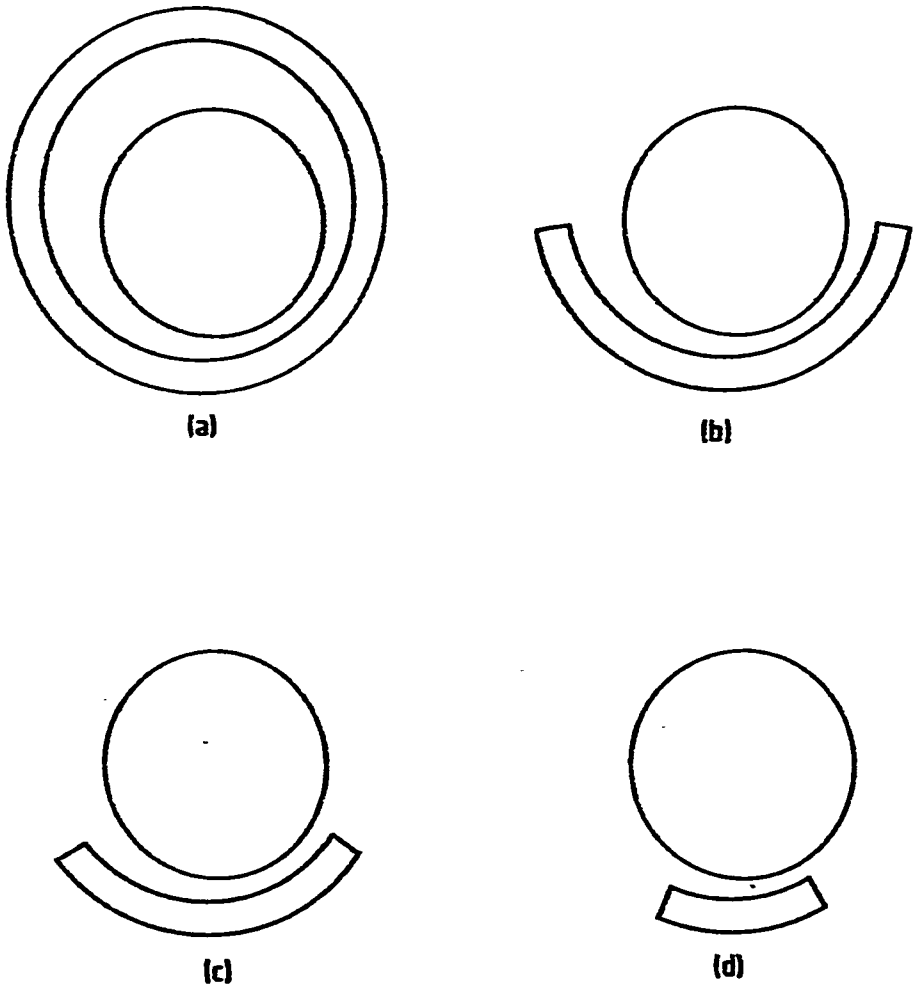


Fig. 1.2 Journal Bearing Geometries: (a) Full Bearing, (b) 180° Partial Bearing, (c) 120° Partial Bearing, and (d) 60° Partial Bearing.

by a relatively thick film of lubricant, so as to prevent metal-to-metal contact, and that the stability thus obtained can be explained by the laws of fluid mechanics. This lubrication does not depend upon the introduction of the lubricant under pressure, though that may occur; but it does require existence of an adequate supply of oil at all times. The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing. Hydrodynamic journal bearings are also called self-acting bearings [15-16].

The mathematical theory of hydrodynamic lubrication is based upon Reynolds work [17,20]. Therefore the differential equation governing the pressure in the lubricating film is called the Reynolds equation. The assumptions made in developing the equation are:

1. the lubricant obeys Newton's law of viscous flow,
2. the forces due to the inertia of the lubricant are neglected,
3. the lubricant is assumed to be incompressible,
4. the viscosity is assumed to be constant throughout the film,
5. fluid films are so thin in comparison with the bearing radius that the curvature can be neglected,
6. the flow is laminar, and
7. the motion of the fluid in a direction normal to the surfaces is neglected in comparison with motion parallel thereto.

The Reynolds equation for most machine design applications for a steadily running bearing is given by [19-21]

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right) = 6U\mu\frac{\partial h}{\partial x} \qquad (1.1)$$

Generally, three types of circumferential boundary conditions are applied in solving the Reynolds equation. They are Sommerfeld, Gumbel, and Swift-Stieber conditions [15-16]. The three types of boundary conditions are summarized in Table A.1 in the appendix.

In case of long bearings, where L/D > 2 , the pressure does not change in the axial direction (the z-axis), i.e. there is no side leakage. Therefore, neglecting the axial pressure flow term, Eq. (1.1) reduces to

$$\frac{d}{dx}\left(h^3\frac{dp}{dx}\right) = 6U\mu\frac{dh}{dx} \tag{1.2}$$

which is the classical Reynolds equation for one-dimensional flow. This equation has been solved by Sommerfeld and Gumbel. The results are presented in Table A.2 [15-16].

In case of short bearings (L/D < 1/4), all of the entering lubricant is diverted to the side leakage. Under this condition, the axial pressure flow in the z-direction will dominate over the circumferential flow in the x - direction. Also, h is usually not a function of z (only a function of x). Therefore the Reynolds equation may be written as

$$\frac{d^2p}{dz^2} = 6U\mu\frac{dh/dx}{h^3} \tag{1.3}$$

which is also known as Ocvirek equation. If the boundary conditions are taken as (i) at z = 0, dp/dz = 0 (symmetry about z = 0) and (ii) at z = $\pm L/2$, p = 0, then Eq. (1.3) may be expressed as

$$p = 3U\mu\frac{dh/dx}{h^3}(z^2 - L^2/4) \tag{1.4}$$

The results obtained using Gumbel circumferential boundary conditions are shown in Table A.3 [15-16].

The notations used in Eqs. (1.1) - (1.4) are:

p = film pressure

V = surface speed of the shaft

h = variable film thickness

x = co-ordinate in the direction of motion

z = co-ordinate in the axial direction

μ = absolute viscosity of the lubricant

L = axial bearing length.

For finite-length of bearings ($1/4 \leq L/D \leq 2$), Eq. (1.1) has been solved by many researchers. Reason and Narang [22] proposed an approximate technique to design steadily loaded journal bearings on a hand-held calculator that makes use of both long- and short- bearing theories. They consider the film pressure p as the harmonic mean of the short-bearing pressure p_s and the long-bearing pressure p_l, i.e.

$$1/p = 1/p_s + 1/p_l \tag{1.5}$$

Table A.4 contains the pressure and various performance parameters obtained with the help of this combined solution approximation [15-16]. These parameters are written in terms of I_s and I_e. Values of these quantities are presented in Table A.5 [22].

However, common practice is to use design charts for representing bearing performance data. The most commonly used set of design charts was constructed by Raimondi and Boyd [18,23,24]. The present work makes use of the charts and tables given by them [24] in developing the required knowledge for designing journal bearings. Film rupture has been considered in preparing these charts. An attempt is made to present the knowledge in a proper form to

be used as knowledge bases in the present expert system. This representation of knowledge has been performed in chapters 4 to 7.

1.4 PRESCRIPTIVE AND FUNCTIONAL SPECIFICATIONS

The basic form of a journal bearing is shown in Fig. 1.3. While designing journal bearings, one may distinguish between two groups of variables [17]. The first group consists of the load per unit of projected bearing area P (= W/LD), bearing industrial-application for selecting the allowable range of the clearance, lubricant inlet temperature, L/D ratio, SAE grade of the lubricant, viscosity μ, and bearing dimensions (diameter D, length L, clearance C, and bearing arc β). They are prescriptive variables and may be controlled by the designer.

The second group contains the dependent variables like the load W, speed N, Sommerfled number (or bearing characteristic number) S, coefficient of friction f, temperature-rise ΔT, minimum film thickness h_m, eccentricity ratio ε, attitude angle Φ, oil flow Q, side leakage Q_s, maximum pressure p_{max}, position of maximum film pressure $\theta_{p_{max}}$, position at which film terminates, θ_{p0}, angle from line of centers to start of film θ_A, and torque required or power lost. These variables may be regarded as functional specifications. The designer cannot control these except indirectly by changing one or more of the first group, but must impose certain limitations to assure satisfactory performance.

The fundamental problem in bearing design, therefore, is to define satisfactory limits for the second group of variables and then to decide upon values for the first group such that these limitations are not exceeded, then the design is complete.

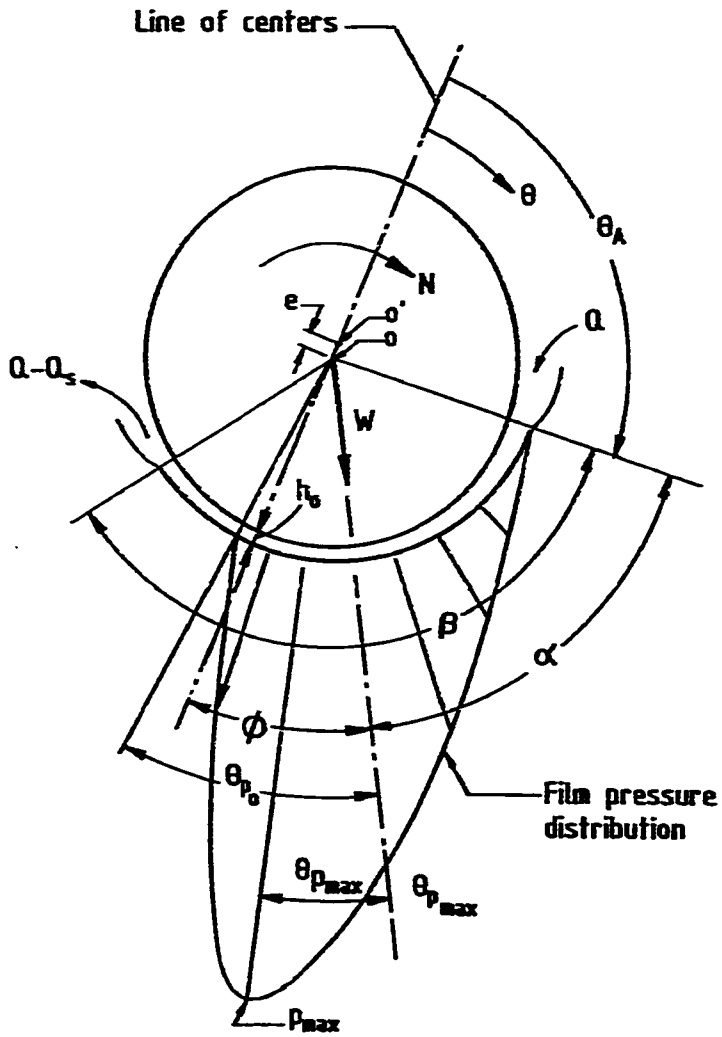


Fig 1.3 Journal Bearing Nomenclature [24].

1.5 SCOPE

The objective of the present work is to build an expert system 'ES' for designing full and partial (180°, 120°, and 60°) hydrodynamic journal bearings of finite-length (class III design problem). Partial journal bearings considered are centrally loaded and of the clearance type. Different methods available in the literature for designing hydrodynamic journal bearings are reviewed. On the basis of the facts and rules collected, a new integrated and dependable design procedure is established.

Various architectures used in design expert systems are reviewed and an architecture consisting of several stages is employed for development of the present 'ES'. This architecture uses the goal-driven rule-based production system (c.f. Sec. 2.1) in which the rules and the databases for each stage are grouped. At the first stage which is the input stage, the user is asked to enter the required functional and prescriptive specifications. The programming language chosen is Turbo Prolog version 2.0.

The 'ES' presumes the use of available standard bearing materials, becuase it is built to solve a class III design problem. This 'ES' solves a bearing design problem using the following: L/D ratios of 1/4, 1/2, 0.6, 1, 1.5, and 2; SAE 10, 20, 30, 40, 50, 60, and 70 oils, decision criterion as maximum load, mininimum friction, and optimal clearance; and full journal bearing or partial journal bearing having bearing arcs of 180°, 120°, and 60°.

The design data corresponding to different performance variables at L/D ratio of 1/4, 1/2, 1, and ∞ for both full and partial journal bearings are found in the literature. The performance data at L/D ratio of 0.6, 1.5, and 2 are

derived using well established relation. The best fit equations are obtained by curve fitting these data. The available objective knowledge (c.f. Sec. 2.2) on stiffness and damping coefficients of journal bearings is gathered. For utility function method to be employed for evaluating the design solutions, necessary data bases are devised as reasonably as possible. Thus all the relevant knowledge bases are obtained. They are then represented in a proper form so as to be used skillfully by the 'ES'.

Optimum bearing design may be based on one of the following criteria: the max. load, the min. friction, or the optimal clearance. One computer program has been developed for the first two criteria and another one for the third criterion. Similar but separate programs have been developed for full and partial journal bearings.

At the end of the input (first) stage, there will be a large number of design solutions whose feasibility is checked in the succeeding stages. At each stage, the solutions which do not satisfy that particular design constraint are discarded. At the second stage, the temperature-rise is checked to ensure that the lubricant outlet temperature should not go beyond 120 °C. At the next stage, the clearance obtained is examined in the light of its manufacturability. This order is followed in the program for the max. W and the min. f conditions. But, in the second program for the optimal clearance, temperature-rise is to be obtained iteratively, therefore the clearance-check is performed prior to the temperature-rise check. In the other stages, minimum film thickness and stability requirements are examined. For partial bearings, stability check is not performed. If at any stage all the solutions fail to satisfy a given design constraint, then that constraint may be modified in order to get at least one

solution.

The design solutions that satisfy all of the above mentioned constraints are evaluated in the decision making stage where the utility value is calculated. The design alternative having maximum utility value U_{max} and the alternatives having utility values above 95% of the U_{max} are identified. In the output stage, all the finally selected design solutions are printed in the 'results.out' file.

The present 'ES' makes an exhaustive search for all the solutions to a given design problem. The results are presented and compared with those of the earlier researchers to illustrate the usefulness of this expert system.

CHAPTER 2

LITERATURE REVIEW

A large body of material is available in the literature related to the design of journal bearings and the expert systems used in design automation. They are summarized here under separate sub-headings.

2.1 AN OVERVIEW OF EXPERT SYSTEMS

An expert system [25-26] is the one that handles real world, complex problems requiring an expert's interpretation and solves these problems using a computer model of expert human reasoning, reaching the same conclusions as those the human expert would reach if faced with a comparable problem. Experts[26] are those who are very good at solving specific types of problems. Their skills usually come from extensive experience and detailed specialized knowledge of the problems they have handled. A number of constraints on system design is placed by the fact that expert systems are typically used in those fields where both the experts who are essential to their development and, in most cases, those who will use them when developed are highly skilled and largely autonomous professionals.

A computer-based expert system[26] seeks to capture enough of the human specialist's knowledge so that it too will solve problems expertly. Over the past ten years various research groups in artificial intelligence (AI) have built specialized systems containing the expertise needed to solve the problems of medical diagnosis and treatment, chemical structural analysis, geological

exploration, computer configuration selection, and computer fault diagnosis, among others. Although these systems are very different and specific to each application it has gradually been discovered that computer-based technology for representing knowledge and reasoning with expertise can be quite general.

The computational method[25] used commonly in expert systems is the use of a knowledge base of rules of the form of situation and action pairs. The whole focus in a knowledge-based system is on knowledge, i.e., facts about the task domain and heuristics or rules of thumb that guide the use of knowledge to solve problems in the domain [27]. The distinction between expert systems and knowledge-based systems is not clear. When the domain of expert knowledge in a computer is emphasized, the system is called an expert system and when the knowledge representation formalism (such as a rule, frame etc.) is emphasized in a system design, it is called a knowledge-based system [28].

Negoita[29] defines knowledge engineering as a discipline devoted to integrating human knowledge in computer systems. The distinctive characteristic of any knowledge-based system is that its processes are state-driven. Decisions about how to process data are part of the knowledge of the system. In other words, an intelligent system may generate rule-bases. Knowledge is procedural in the sense that it tells how the data concerning a problem can be manipulated to solve it. By internalizing procedural knowledge as a model of the world, the machine becomes intelligent. An expert system is an information system that can pose and answer questions relating to information borrowed from human experts and stored in the system's knowledge base. The answers are automatically extracted from the data descriptions by a user-invisible inference procedure. The users not only present data in a high-level, human-oriented

manner, but also describe the operations used to retrieve those data.

Formalizing the experience of human experts and making it scientifically testable is one of the major goals of expert systems. Weiss and Kulikowski[26] have formalized their experience, which is best described as dealing with a well defined class of expert systems that solve classification problems using a rule-based approach. A classification problem is one where one must place a subject, object or phenomenon (signal) into one of several prespecified classes. Diagnostic decision-making in medicine or in a specific domain is one of the clearest examples of a classification problem. Other examples are the interpretation of geological signals in mineral prospecting and oil exploration and the selection of advice in repairing faulty equipment. Rule-based approaches are those that use a large store of rules for their reasoning. They have tried to show that representing expert knowledge need not be very complex; strategies of reasoning can be designed to work without excessively long or arcane chains of logic. A reliable expert, a knowledge engineer who will de-brief the expert, and a good sample of test cases of difficult problems are the elements required to design and then to challenge an evolving expert system. In approaching the design of an expert system, they have found that programming skills are helpful, but usually secondary to knowledge of how decisions are made in a problem domain.

An important way of encoding knowledge that can be incorporated into a working program is in the form of a production system [25,27]. A production system is a collection of rules of the form of condition and action pairs, where the conditions are statements about the contents of a global database of facts and the actions are procedures which may modify the contents of the database

[25]. Program execution consists solely of a continuous sequence of *cycles* [25] terminating when some halting action is executed. At each cycle, all rules with conditions that are satisfied by the contents of the database are determined. If there is more than one such rule, one is selected by means of some suitable 'conflict resolution' strategy (e.g., the first in some pre-determined priority sequence is taken). All the actions associated with the selected rule are then performed and the database changed accordingly. Such a production system is said to have a condition-driven, forward-chained, or pattern-driven architecture [25].

An alternative architecture is the action-driven, goal-driven, backward-chained, or consequent-driven production system [25]. In this case, rules can usefully be thought of as having the form of antecedent and consequent pairs. The system is given in effect, a goal to prove through deductive inference. The consequent part of rules are examined to find one which could make the goal true. When such a rule is found it is examined to see if all its conditions are true. If they are, the rule is fired, if not the process continues recursively in an attempt to show that each condition of the rule is true.

The second declarative form of representation is a restricted form of first-order predicate logic [25,27]. The programming language PROLOG permits such 'logic programs' to be executed directly. A PROLOG program consists of a collection of logic statements (of a restricted form) [25]. Execution corresponds effectively to performing deduction from the constituent clauses of the program. The following is a very simple example of a program which performs deductive reasoning over a database [25]:

```
parent_of(X,Y) ← mother_of(X,Y).
parent_of(X,Y) ← father_of (X,Y).
grandfather_of(X,Y) ← father_of(X,Z),parent_of(Z,Y).
mother_of(martha,henry).
mother_of(susan,ronald).
mother_of(jane,lousie).
father_of(john,henry).
father_of(mark,jane).
father_of(michael,ronald).
father_of(ronald,frances).
```

The above is the complete program. (The first three lines are rules, the others are facts.) The program has a declarative meaning (taking the 'natural' interpretation of the names used). Thus line 1 can be read: for all X and Y, X is the parent of Y if X is the mother of Y. Line 2 may be read: for all X and Y, X is the parent of Y if X is the father of Y. Line 3 denotes: for all X and Y, X is the grandfather of Y if, for some Z, X is the father of Z and Z is the parent of Y. Line 4 denotes: martha is the mother of henry. In addition to the declarative meaning, the program has a procedural interpretation, e.g., (line 3) "to prove that X is the grandfather of Y, find a Z such that X is the father of Z and Z is the parent of Y".

To satisfy a goal (left-hand-side) the system tries to satisfy the conditions (subgoals) on the right-hand-side, one at a time , in left-to-right order. If a goal cannot be satisfied the system backtracks. Thus program execution is effectively a left-to-right, depth-first search through an AND/OR tree of logical relations, chaining backwards from goals to be established to given facts.

Entering the following goal: grandfather_of(michael,Y) will produce the solution Y = frances. To find all grandfather/grandchild relationships it is only necessary to enter the goal: grandfather_of(X,Y). This will return the solutions X = mark, Y = louise and X = michael, Y = frances, respectively.

The potential value of the idea of 'logical programming' is considerable, offering an extremely compact means of representing complex logical relations and a database of relevant facts, together with a built-in means of performing deductions, which is firmly based on the theory of mathematical logic [25]. Unfortunately, it would seem that no non-trivial expert system has so far been constructed in this way, so no practical evaluation is possible. This is, however, an important area to pursue in the future. If expert systems are to be used as a medium for formulating, accumulating and transferring expert knowledge, perhaps over several generations, rather than as 'one-off' high performance systems, it is extremely important to establish a standard framework, atleast for all the systems in a particular field [25].

Another way of knowledge representation is frame systems, where things and events are represented by a collection of frames [27]. Each frame corresponds to an entity and contains a number of labelled slots for things pertinent to that entity. Slots in turn may be blank, or be specified by referring to other frames, so the collection of frames is linked together into a network.

Hayes-Roth [30] described the state of knowledge systems technology and its commercialization in the United States. He discussed weaknesses in the technology and emphasized a need to incorporate knowledge that is difficult to represent. Ishizuka and Moto-oka [28] outlined activities in Japan concerning the construction of expert systems.

Bramer [25] indicated that a problem associated with deductive reasoning systems is the need to make default assumptions when reasoning about incompletely specified domains. Thus deductions may need to be made which are consistent with the available information but are not probably correct and may, in fact, turn out to be incorrect in the light of information subsequently obtained. Another problem associated with expert system development is that of automatically acquiring new information or refining existing information on the basis of experience gained. Also, there is a need for well-designed explanatory capabilities and a sound model of inexact reasoning.

Kidd and Welbank [31] considered the important problem of acquiring knowledge from human experts and suggested three stages for this purpose:

1) Identifying the role of the system and the basic structure of the problem domain: There are two major objectives at this first stage of knowledge acquisition, the first is to identify the role which the proposed expert system should play in the problem environment and to outline what knowledge is needed for the system to fulfil this role. The second is to identify the basic structure of the problem domain in order to decide on the architecture of the proposed expert system or to select an appropriate shell.

2) Eliciting the detailed knowledge from the expert: The objective at this stage is to gain from the expert the richest possible collection of information about his actual working knowledge in solving problems within the domain. To satisfy this aim, it is recommended that knowledge engineers seek to employ a variety of knowledge acquisition techniques on any application.

3) Debugging and refining the knowledge base: The major objective of the

final stage of knowledge acquisition is to test rigorously the performance of the evolving system against the performance of the human expert in order to verify the robustness, consistency, and appropriateness of the knowledge base contents.

Because the knowledge base in an expert system is put there by human experts and because much human knowledge is vague, it is usually true that facts and rules are neither totally certain nor totally consistent. For this reason, a basic issue in the design of expert systems is how to equip them with a computational capability for evidence transmission [29]. To solve this problem, researchers have augmented the inference procedures with mechanisms that combine evidence degrees according to the rules of plausible reasoning. Plausible reasoning is simply drawing conclusions from facts that seem to be correct. In most systems, this mechanism is purely heuristic. Recently, however, some investigators have tried to make that mechanism mathematically sound.

Another approach[29] is based on the theory of fuzzy sets that uses approximate reasoning, which means drawing conclusions taking the consistency of the facts into account. The treatment of fuzziness is a critical issue in knowledge representation. To say that a word is fuzzy is to say that sometimes there is no definite answer as to whether or not the word applies to something. The indeterminacy is due to an aspect of the meaning of the word rather than to the state of our knowledge. On all expert systems based on semantic manipulation (explained in the next para) and approximate reasoning, the emphasis is on fuzziness viewed as an intrinsic property of natural language. An evident advantage of the fuzzy set approach is the possibility of representing numeric and linguistic variables in a uniform way and of using a sound formalism to handle them.

A semantic system [29] is software that uses fuzzy set technology to translate the meaning of a vocabulary. Once the knowledge engineer has developed the semantic system, the user can exploit it without any interface. With a semantic system, the user can encode knowledge in many forms: production rules, production systems, and verbal models. A verbal model can be viewed as a production system with mathematical operators.

If one represents facts as objects and rules as morphisms, the mathematical theory of categories [29] is a good language for describing the mechanisms of evidence combination. The difference between plausible and approximate reasoning becomes the difference between the categories on which one models the facts. In this way, an algebra of knowledge becomes available to the system designer and knowledge diagrams become models of both production systems or declarative systems used in logic programming.

Negoita [29] reported that the problem-solving power of an expert system based on plausible reasoning and symbolic manipulation is primarily a function of the domain-specific information in the knowledge base and is only secondarily a function of the system's inference method. The problem-solving power of an expert system based on approximate reasoning and semantic manipulation is primarily a function of the inference method it employs. No longer are knowledge engineers merely intermediaries between the human expert and the developing knowledge base. In their new role, they are independent of both.

Knowledge acquisition has been a long-standing bottleneck in artificial intelligence [29]. Certainly, the most powerful knowledge systems are those that contain the most knowledge. Symbolic systems deal with description, assertion, and encoding of decision rules. Emphasis is on recursion and list structures,

which can be treated by procedural languages. Knowledge systems based on approximate reasoning are oriented less toward list structures and more toward logic programming. The programmer must concentrate instead on the meaning of what is to be achieved and he must express that meaning in a declarative style. A major issue in this approach is the use of natural language and syntax. Making software user friendly will become progressively more critical in the next decade as more powerful machines become available to a wider range of individuals than ever before. Many computer users will have little computer science training or inclination to get the training. They will want to use natural language in any dialogue with any computer. Only the fuzzy system approach makes such communication possible.

Wolfgram et al., [32] explained that expert systems are appropriate where there are no established theories, where human expertise is scarce or in high demand, and where the information is cloudy or fuzzy. Ramsey [10] presented review of four prominent expert systems which may help a mechanical engineer to develop a powerful system for storing, organizing, and retrieving technical expertise. In practice he/she can become a knowledge engineer.

Fagan [33] reported the progress made on the development of an expert system 'ES', written in Micro-Prolog, for bearing selection and application. This 'ES' assists the designer in selecting the right combination of ball and roller bearings to support a shaft for a given set of operating conditions. The system also offers advice on the mounting requirements of the bearings.

Brown [34] discussed an approach to handling failures that occur during design problem-solving. His work referred to the class III design problems(c.f. Sec. 1.2), which require that at every stage of the design, the designer knows

both what sequence of design steps are appropriate and also what knowledge is required. The expert system 'AIR-CYL' [35] incorporates these failure handling facilities. Additional knowledge, in order to deal with failures is given to the system. Brown and Chandrasekaran [35] built 'AIR-CYL' using a hierarchical artchitecture for routine design work like design of an air cylinder. They considered routine design to be largely a top-down activity. Several types of agents(active problem-solver modules) in the hierarchy's decision-making structure included specialists, plans, steps, tasks, and constraints. The design activity was consisting of four phases: requirements, rough design, design, and redesign.

Dixon and Simmons [36] constructed an expert system to design standard V-belt drives. The Design-Evaluate-Redesign architecture(Fig. 2.1a), an iterative method of design was the basis of the expert system called VEXPERT, written in LISP using OPS-5 production rules. Utility-Decision theory models were used for decision making. Values of utilities and weighting factors were attached to the parameters used to evaluate design. Dixon et al., [37] described the first working version of a program called Dominic whose architecture is shown in Fig. 2.1b. Dominic performs design by iterative redesign in a domain-independent manner. The program strategy stresses the concept of dependencies to guide its redesign process. A dependency expresses a relationship between a performance parameter and a design variable, that is, dependencies describe how individual performance parameters depend on the design variables. Dominic has been successfully tested in four different domains. Its performance on two of these (V-belt drive design and design of extruded heat sinks) is presented in their paper. Dominic is a hill-climbing algorithm, similar in this respect to standard optimization methods. However, its problem formulation or

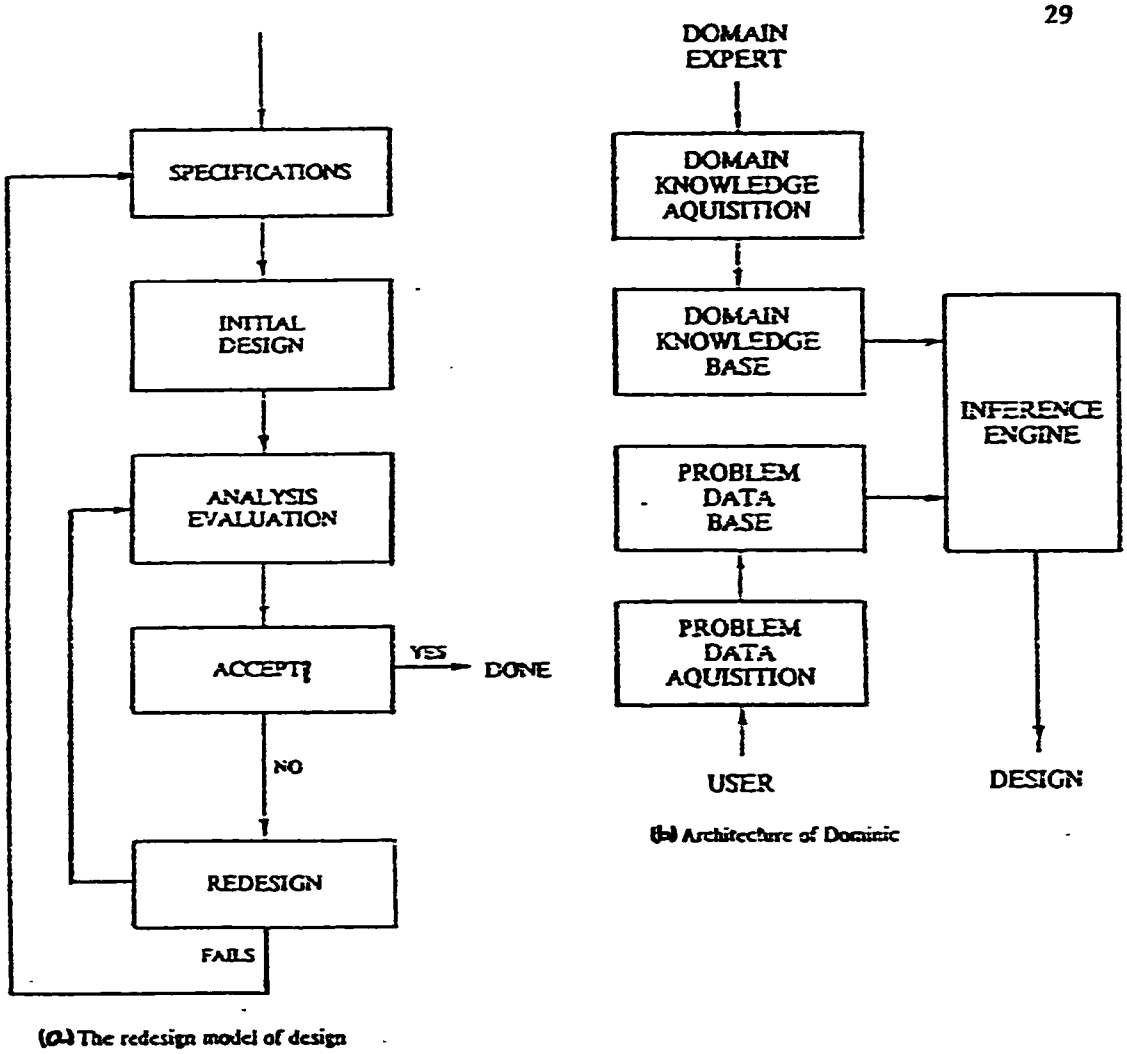


Fig. 2.1 Architecture of Expert System 'Dominic' Based on Iterative Redesign Method [37].

input language is more flexible.

Chandrasekaran [14] provided a critique of the abstraction level of the currently dominant approaches and proposed an alternative level of abstraction, i.e., high-level building blocks called shells. He found six generic tasks to be very useful as building blocks for the construction of knowledge-based systems. Because of their role as building blocks, they are called elementary generic tasks. The tasks include hierarchical classification, hypothesis matching or assessment, knowledge-directed information passing, abductive assembly, hierarchical design by plan selection and refinement, and state abstraction. They are used as building blocks for class III design problem solvers.

Chandrasekaran and Swartout [38] reported that explicit representations facilitate explanation in knowledge systems. Sheu et al., [39] presented a knowledge-based methodology for analog IC design assisted by an expert system for the iterative design process. A prototype expert system has been developed, which adopts the flexible architecture approach.

El-Kady et al., [40] developed an expert system for power cable design, which served two purposes, to assist experienced cable designers in arriving at the best and the most economical design for a particular application, and to educate novice cable engineers and inform them of the various aspects of cable component selection and design specifications. Lamirande and Roberge [41] used expert system in managing the water chemistry of the heat transport system of a CANDU nuclear reactor.

Andreychikov et al., [42] described comparative analysis of expert systems and automated banks of engineering knowledge (ABEK) designed for

automating the processes of making design decisions in the early stages of design of various engineering objects. A sequence of design procedures, formulation of design task, generation(synthesis) and then analysis of design variants, and selection of a rational variant for further design is followed in using ABEK. They reported that the ABEK must be reinforced with a well-developed and successfully applicable, in existing expert systems, intelligent user interface. According to them, a workable design procedure may consist of seven stages. These stages are described later in Sec. 3.1.

Johnson and Keravnou[43] describes two basic forms of interaction employed in expert systems: user initiated and computer initiated. In the user initiated mode of interaction the system is restricted to respond to user requests only, while, in the computer initiated mode of interaction the user is restricted to respond to system requests only. The desirable mode of interaction in expert system is a mixed-initiative one, whereby the initiative switches from one basic mode to the other. They have also discussed a generalized model of diagnostic behavior shared by most diagnostic tasks.

From the perspective of knowledge engineering there are three types of knowledge-based systems [44]. Type I (Type K1) system performs the following functions on its knowledge-base: browsing, retrieval, transfer, etc. Type II (Type K2) system manipulates its knowledge-base using the following operations: conversion, examination, extraction, diagnosis, problem solving, etc. Type III (Type K3) system resorts to the following operations: reasoning, abstracting, inference, innovation, etc. Class III design problems can be solved using Type II (Type K2) knowledge-based systems. A class I or a class II design problem may be tackled using a Type III (Type K3) knowledge-based system.

2.2 COMPUTER-AIDED DESIGN

Engineering design [45] is an iterative, decision-making activity whereby scientific and technological information is used to produce a system, device, or process which is different, in some degree, from what the designer knows to have done before and which is meant to meet human needs. Suh[46] defines design as the creation of synthesized solutions in the form of products, processes or systems that satisfy perceived needs through the mapping between the functional requirements (FRs) in the functional domain and the design parameters (DPs) of the physical domain through the proper selection of DPs that satisfy FRs. In addition to the FRs, designers often have to specify constraints.

Design involves four distinct aspects of engineering and scientific endeavor [46]: the *problem definition* from a "fuzzy" array of facts and myths into a coherent statement of the question; the *creative process* of devising a proposed physical embodiment of solutions (this is an 'ideation' process, which is highly subjective); the *analytical process* of determining whether the proposed solution is correct or rational (by making correct design decisions as well as evaluating the details of specific design features); and the *ultimate check of the fidelity of the design product to the original* perceived needs. The analytical process is deterministic and is based on a finite set of basic principles or axioms. Axioms [46] are formal statements either of what people already know, or of the knowledge imbedded in many things that people do or use routinely. The creative and analytical processes are interrelated. The design axioms help the creative process of the design activity.

Constraints [46] in the context of axiomatic design represent the bounds on an acceptable solution. They are of two kinds: input constraints, which are constraints in design specifications, and system constraints, which are constraints imposed by the system in which the design solution must function.

Much work has been done to understand the creative process and develop a 'design methodology' that can systematize the design process. Harrisberger [47] lists the following techniques which can be used to aid the design process: Trigger-work, Checklist, Morphological, Attribute-seeking, and Brainstorming. Many authors present *rules* for design in various situations as well as general methodologies [46]. Many software packages have also been created to assist the designer.

The advent of computers has a significant impact in the design field [46]. Various expert systems and other AI based programs have been developed according to ad-hoc design rules, where the design answers are arrived at through a series of queries. Such expert systems would result in a better design, improve quality, reduce time and cost, and they could be linked to other expert systems, which would further enhance the whole design process [33].

However, truly intelligent computers for design cannot be developed until the basic principles that have general applicability in all synthesis processes are incorporated as part of the computer software [46]. Thus computers can not yet make design decisions based on their own built-in intelligence because even the most ideal computer can not deal with the vast data base that will be needed to anticipate all different possibilities in a general design situation.

The basic assumption of an *axiomatic approach* to design is that there

exists a fundamental set of principles that determines good design practice. This approach differs philosophically from the current trend, which relies on large computers with vast data bases.

A fundamental limitation of computer-based technology given by Bremmerman's limit [48] makes it imperative that basic principles and the methodologies that stem from the principles, provide the conceptual framework and explicit tools for design, thus eliminating the need for an exhaustive search of all possibilities.

The design axioms may be expressed mathematically with the help of symbolic logic [49] in order to be able to utilize the power of digital computers in making design decisions. The programming language chosen for this purpose is PROLOG, which is a fifth-generation, declarative and very high-level programming language [50-52].

The ability to state the axioms as clauses in a logical programming language is utilized to develop an expert system for axiomatic design [46]. A PROLOG software driver for axiomatic design may be based on three layers or levels: primary level consisting of the design axioms encoded in PROLOG, secondary or intermediate level serving as the interface between the primary level and the data level, and data level (a software pertaining to the raw data specification). A potential system architecture that is designed to support such expert system is shown in Fig. 2.2 [46]. The heart of the system is an axiomatic expert. The user may interact with the system through a friendly interface as well as the graphics capabilities of a computer-aided design (CAD) package. The core axiomatic expert and the CAD systems communicate through the data transformer, a package that converts the data for alternate designs from CAD-

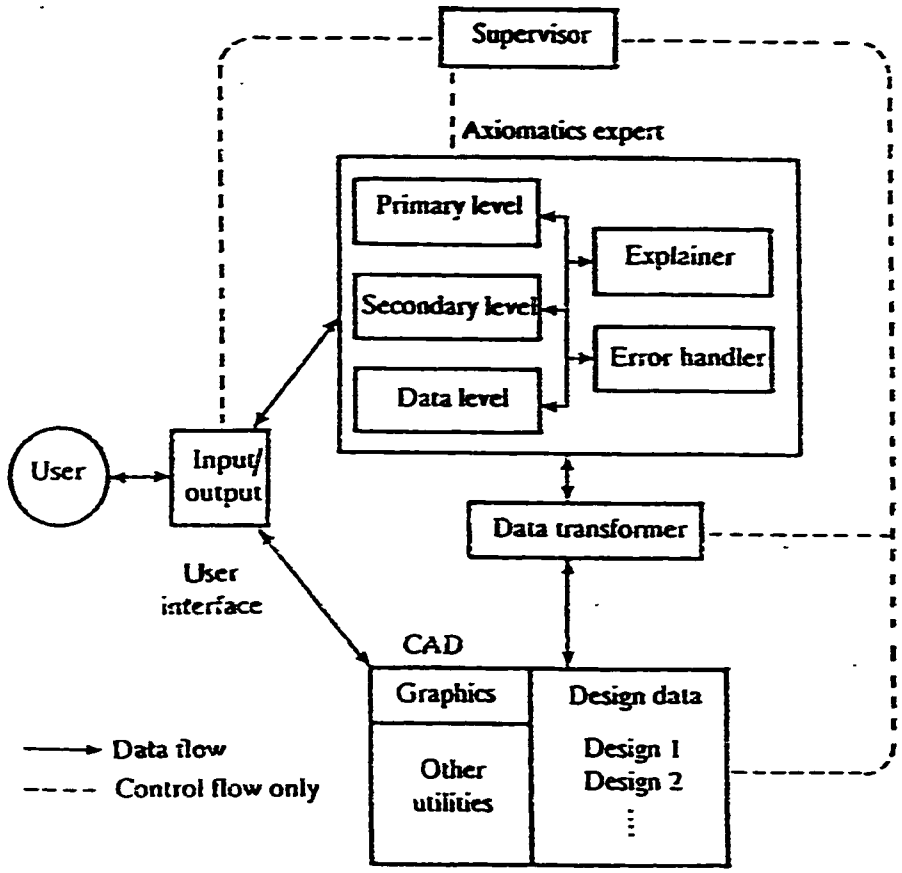


Fig. 2.2 Architecture of Axiomatic Expert System [46].

compatible to computerized axiomatic system CAS-compatible knowledge.

Bardasz and Zeid[11] proposed analogical problem solving 'APS' to mechanical design that has two characteristics: open end and iterative. An open-ended problem is one that has multiple solutions. The 'APS' is based on the fundamental principle that the problem solving can be assisted by the review of solutions to past problems that have been attempted. In general, this has seven steps: reminding, modifying, mapping, evaluation, repair, generalization, and storing. As an example, the design of a gear-and-shaft assembly has been considered [11].

Dixon[53] proposed new goals for design education, especially in mechanical engineering. He divided the mechanical design research field into six categories: descriptive models of design concerned with the study of the techniques and problem-solving strategies, prescriptive models of design concerned with how design should be done (two most relevant prescriptions are the German school and the axiomatic method, a term used by Suh[46]), computer-based models of design that expresses a method by which computer may solve a particular design problem resulting in a powerful system for mechanical design, languages, representations, & environments, analysis to support design, and design for manufacturing and the life cycle analysis.

As far as design variables are concerned Dixon[54] identifies input and output variables of the design process. Simon[55] suggests the use of input, output, and solution variables. Middendorf[45] and Guntur[6] use functional and prescriptive specifications as the design process variables.

Guntur[6] describes that in a class I design problem the unknown

functional specifications are too many as compared to the known functional specifications therefore the problem is very vaguely defined. In a class II design problem the unknown functional specifications are not as many as those in a class I design problem. In addition, some prescriptive specifications are also known. In this case though the complete plans for the solution can not be readily prepared, at least partial plans for the solution of the problem can be put together immediately while some perceptual blocks are being cleared. A class III design problem is well defined since sufficient functional and prescriptive specifications are available and therefore complete plans for the solution of the problem can be easily formulated by a domain expert.

Once most of the perceptual blocks [56] are cleared, a class I design problem is transformed into a class II design problem. When the remaining perceptual blocks in the transformed class II design problem are removed, it is reduced to a class III design problem. Thus perceptual blockbusting[56], which is an essential part of conceptualization and creative thinking is necessary for solving class I or class II design problems. For creative thinking there are two techniques: brainstorming and synectics [1].

In automating the design process, the first step is to represent the procedural, factual, and control knowledge in a suitable way. One of the most difficult tasks in automating the design process is the representation of inductive procedural knowledge which is basically conjectural knowledge. Recent research shows that it is possible to use analogical problem solving methods or messy genetic algorithms to solve class I or class II design problems [11,57], provided that very fast computers with parallel processing capabilities are available. The procedures used in solving a class III design problem are generally well

established. Thus, the procedural knowledge, factual knowledge, and control knowledge used in solving a class III design problem can be classified as objective knowledge [58], i.e., the knowledge contained in books, handbooks, and other publications.

In the final stage of the design process used in solving a class III problem, a decision making procedure has to be employed in order to choose the best solution from a set of alternative design solutions. At this stage subjective or inductive knowledge has to be used. This decision making amounts to solving a multiparameter multiobjective optimization problem. For this purpose, 'utility function' method may be employed [6,36,46,59,60]. Keeney and Raiffa[59] discuss the importance of utility or value analysis. If an appropriate utility is assigned to each possible consequence and the expected utility of each alternative is calculated, then the best course of action is the alternative with the highest expected utility. However, this method is either algorithmic, which apply design rules to a specific situation, or not generalizable and lacks fundamental principles that can be applied to all design situations [46].

2.3 JOURNAL BEARING DESIGN

For journal bearing design, Boyd and Raimondi [61-62] have obtained solutions to the Reynolds equation for bearings of infinite length ($L/D = \infty$). Raimondi and Boyd [18,23,24] have reported the results in three parts of a comprehensive program to obtain data not included in previous investigations. They have used digital computer techniques for solving the fundamental equations numerically. They have assumed that the viscosity remains constant as the lubricant passes through the load area where both the temperature and the pressure vary. The second assumption in Parts I and II of these papers is

that rupture does not take place in the trailing portion of the film, whereas, Part III assumes that the oil film is ruptured when the film pressure falls below atmospheric pressure.

The results [18,23,24] contain 45 detailed charts and 6 tables of numerical information for the design of bearings with slenderness ratios (L/D) of 1, 1/2 and 1/4 for both partial (60, 120 and 180 degrees) and full journal bearings. Tabular form of Part III [24] has data also for L/D = ∞ and for optimum conditions, namely, maximum load and minimum friction. Shigley [17] reproduced charts of the tabular data of Raimondi and Boyd [24] for full journal bearings for L/D ratios of 1/4, 1/2, 1, and ∞.

Each of the charts is entered using the bearing-characteristic number or the Sommerfeld number (S). This is defined as

$$S = (R/C)^2 \mu N/P \tag{2.1}$$

where R is the journal radius, C the radial clearance, μ the viscosity, N the speed, and P the bearing unit load. If the viscosity of the lubricant is known, S can be calculated, otherwise, viscosity must be found by an iterative process given in [17] or [19].

Raimondi and Boyd [23] suggest the use of the following formula for other L/D ratios within the interval 1/4 < L/D < ∞:

$$y_{L/D} = \frac{1}{(L/D)^3}\Big[-(1/8)(1-L/D)(1-2L/D)(1-4L/D)y_\infty + (1/3)(1-2L/D)(1-4L/D)y_1$$

$$- (1/4)(1-L/D)(1-4L/D)y_{1/2} + (1/24)(1-L/D)(1-2L/D)y_{1/4}\Big] \tag{2.2}$$

where y = desired performance variable and the subscript of y is the L/D ratio at which the variable is being evaluated. For partial bearings with other bearing arc angles they recommend the following formula:

$$r_\beta = \frac{1}{7200}[(\beta-120)(\beta-60)r_{180} - 2(\beta-180)(\beta-60)r_{120} + (\beta-180)(\beta-120)r_{60}] \quad (2.3)$$

Welsh [63] has given a chart for the minimum recommended diameter corresponding to a given load and speed for L/D = 0.6. For other L/D ratios, he has given correction factors to calculate the load value. Clayton and Taylor [64] presented design data for the steadily loaded full central circumferential grooved plain journal bearings with a consideration of film reformation to ensure satisfactory lubricant flow continuity.

Seireg and Dandage [65] obtained mathematical expressions for the operative characteristics of a 360 degree journal bearing for L/D ratio in the range of $1/4 \le L/D \le 1$, by curve-fitting the design data of Raimondi and Boyd [24], in which film rupture was taken into account. Similarly, Al-Dukhaiyel [66] developed equations for various performance variables by curve-fitting the design data of Shigley [17] for full journal bearings with L/D ratio of ∞, 1, 1/2 and 1/4.

General information on journal bearing material and its selection criteria has been outlined in [15-17]. Journal bearing materials may be divided into two groups: metallics and nonmetallics [15-16]. The metallic group includes aluminimum alloys, babbits (tin-, lead-, and aluminimum-based), copper alloys (brass and bronze), zinc, and iron. The nonmetallic group includes plastics, carbon graphites, cemented carbides, and other proprietary materials. The nonmetallics have been generally used in self-lubrication applications because they can provide low friction and wear without the aid of a lubricant. Physical properties, typical applications, and numerical ranking of the performance characteristics of a variety of journal bearing materials are presented in [15-16].

In this investigation (or program development), it is assumed that the bearing material is from the metallic group of materials.

Viscosity-temperature chart of typical SAE numbered oils is presented in [17,18,19,61,67]. Viscosity-temperature equations for various oils have been obtained by Al-Dukhaiyel [66] and Seireg and Dandage [65]. A table for selecting bearing unit loads depending on the application is given in [15-17,67-68]. Orthwein [19] has given a flow chart for the design of journal bearings with the Raimondi and Boyd procedure [24].

Values of minimum film thickness variable h_o/C for the maximum load and the minimum friction conditions for L/D ratios of 1/4, 1/2, 1, and ∞ for bearing arcs of 360°, 180°, 120°, and 60° are available from Raimondi and Boyd [24], which will be used for bearing design for the two conditions. Moes and Bosma [69] developed a design chart for the full journal bearings which enables the designer to select optimum bearing dimensions for the largest minimum film thickness (max. W) and the minimum frictional torque (min. f) conditions. This chart is constructed in terms of two dimensionless groups that include minimum film thickness and frictional torque.

Shigley [17] discussed how the design may be optimized by plotting curves of the performance factors as functions of the quantities over which the designer has control. He presented plots with clearance as the independent variable and suggests the optimum zone for clearance must be lower than that for maximum load condition because wear would increase the clearance. Juvinall [68] and Keith, Jr. [15-16] recommend that optimal clearance should lie in the range of maximum load and minimum friction conditions. This would result in a lower

cost of manufacturing. Also, Keith, Jr. [15-16] gave a plot of recommended minimum clearance for given shaft speed and diameter, which could be used as a helpful guide in this process.

Seireg and Dandage [65] described an empirical procedure for predicting the performance of journal bearings based on experimental thermohydrodynamic consideration. They developed relationships, which were utilized to construct design nomograms for evaluating a modified Sommerfeld number S^* to replace the isoviscous Sommerfeld number S for any given oil and inlet temperature.

Shigley [17] has explained how to choose the L/D ratio. A long bearing reduces the coefficient of friction and the side flow of oil and therefore is desirable where thin-film or boundary lubrication is present. On the other hand, where forced-feed or positive lubrication is present, the L/D ratio should be relatively small. The short bearing length results in a greater flow of oil out of the ends, thus keeping the bearing cooler. If shaft deflection is likely to be severe, a short bearing should be used to prevent metal-to-metal contact at the ends of the bearings. Welsh [63] argues that the optimum L/D ratio is 0.6 because any lower length tends to result in too great a loss of oil pressure because of the side-leakage flow, while any larger length tends to create difficulties with alignment and edge loading.

If high temperatures are a problem, a partial bearing should be used, because relieving the non-load-bearing area of a bearing can very substantially reduce the heat generated [17].

Vijayaraghavan and Keith, Jr. [70] discussed the effect of cavitation on the performance of a line-grooved misaligned journal bearing for both flooded and

starved inlet conditions. Almost all of the journal bearing design solutions are based on the assumption that the lubricant is supplied to the bearing at a specific rate, termed as classical rate. In practice, however, the supply rate may be less (the starved condition) or greater (the flooded condition) than the classical rate. To account this and thus provide more realistic design information, Connors [71] developed design charts which incorporate the influence of lubricant supply rate on the performance of a full journal bearing for L/D = 1 and permit the determination of the film thickness, friction and temperature rise of the film.

Welsh [63] has given an empirical relation as well as a chart to find minimum acceptable clearance for L/D = 0.6. A summary of clearances used in industrial applications in the form of a table is given by Spotts [67]. Juvinall [68] describes that for journals of 25-150 mm diameter the clearance ratio (C/R) is usually of the order of 0.001 for precision, 0.002 for general machinery and 0.004 for rough-service machinery bearings. Shigley [17] has also given the graph of recommended radial clearances for full bronze bearings with various degrees of finish.

For minimum oil film thickness h_0, Juvinall [68] suggests the relationship $h_0 \geq 0.005 + 0.00004D$ (h_0 and D *are in mm*), to be used with a safety factor of 2 for calculating S. This has been explained in detail in Sec. 3.3.3. Also, he recommends the maximum acceptable oil temperature should be below $93° - 121°C$.

Vance [72] has given equations and charts for dimensionless stiffness and damping coefficients for the short-bearing and for the 6 shoe tilting pad bearing

for L/D = 0.5 and mathematical expressions for dimensionless stiffness and damping coefficients for the fully cavitated ($\pi - film$) short bearing. Seireg and Dandage [65] obtained expressions for the dynamic properties (stiffness and damping coefficients) of a 360 degree journal bearing. Rao [73] has given charts for stiffness and damping coefficients of plain cylindrical and grooved bearings for L/D = 0.5 and 1. Lund and Saibel [74] obtained expressions for stiffness and damping coefficients for plain cylindrical bearings as functions of eccentricity ratio ε, Sommerfeld number S, and slenderness ratio L/D. Kirk and Gunter [75] treated the effect of support damping and flexibility on the response characteristics of a symmetric rotor mounted in non-linear fluid-film bearings. The bearings were regarded as short bearings. They presented charts for dimensionless stiffness and damping coefficients for a short journal bearing with a cavitated film as function of ε.

The stability of rotor-bearing system in the present context has been described in Sec. 3.3.4. Kirk and Gunter [75] studied the stability and transient motion of journal bearings on rigid supports as well as flexible supports and presented the stability maps. Badgley and Booker [76] investigated stability of full journal bearings with a fixed, rigid, and perfectly balanced rotor under the assumption of light initial impact and plane-motion for the short-bearing, long-bearing, and finite-length bearing approximations. In all the cases examined, instability was not observed above static eccentricity ratios of 0.83. They presented instability threshold curves for all three bearing assumptions (for small initial-velocity disturbances).

CHAPTER 3

DEVELOPMENT OF DESIGN EXPERT SYSTEMS

The need to develop design expert systems originates from a need to automate the design process; design automation reduces the time required for developing a specific product. Advantage of expert systems is that they capture the knowledge of experts that may otherwise be lost through death or retirement. Moreover, they can contain the cumulative knowledge of several experts; they are available any time of day or night and they can be distributed widely throughout an organization. In addition, expert systems have the ability to save design engineers from having to carry out redundant tasks, improve the consistency of designs, search large databases for optimal design, reduce the cost of design while at the same time improve quality, reduce manual errors, also produce very reliable designs. Also, they could be linked directly to other expert systems, which would further enhance the whole design process. However, it should be noted that expert systems are not a substitute for a human expert. Unless a problem is fully understood, which can come only from humans, the expert system project will fail.

In order to be successful a design expert system should have an access to objective knowledge [58], i.e., deductive procedural knowledge (or a set of rules based on analysis and experimentation), factual knowledge, and control knowledge, (or information available in handbooks, books, publications and manufacturers' manuals) and inductive procedural knowledge (or information acquired through experience by experts). In the case of an expert system built to

solve a class III design problem the inductive procedural knowledge may be used only at the time of picking the 'right' design solution.

3.1 DESIGN PROCESS

The schematic diagram of the design process shown in Fig. 3.1 is given by Guntur [6] and includes failure handlers [34]. In this figure, functional specifications (FS) and prescriptive specifications (PS) are used as the design process variables. The design process consists of the following phases [42]: definition of the need for the designed object (symbolized by the problem describer), formation of the tactical and technical requirements for the object (depicted as the prescriptive and functional specifications), formation of the functional structure of the object (represented by the concept builder and synthesizer), development of the principles of operation of the object (included in the knowledge base), development of the engineering solution (effected by the concept builder and synthesizer), development of the mathematical models for computing the parameters (executed by the analyst and evaluator), and computation and/or optimization of the object parameters (performed in the analyst and evaluator). This is the most general scheme. However, in the present work, concept builder and synthesizer are not required.

3.2 ARCHITECTURE

In the previous chapter, work done by the previous researchers on expert systems and design automation has been reviewed, which has presented the various architectures being employed in this field. The important architectures are summarized briefly here.

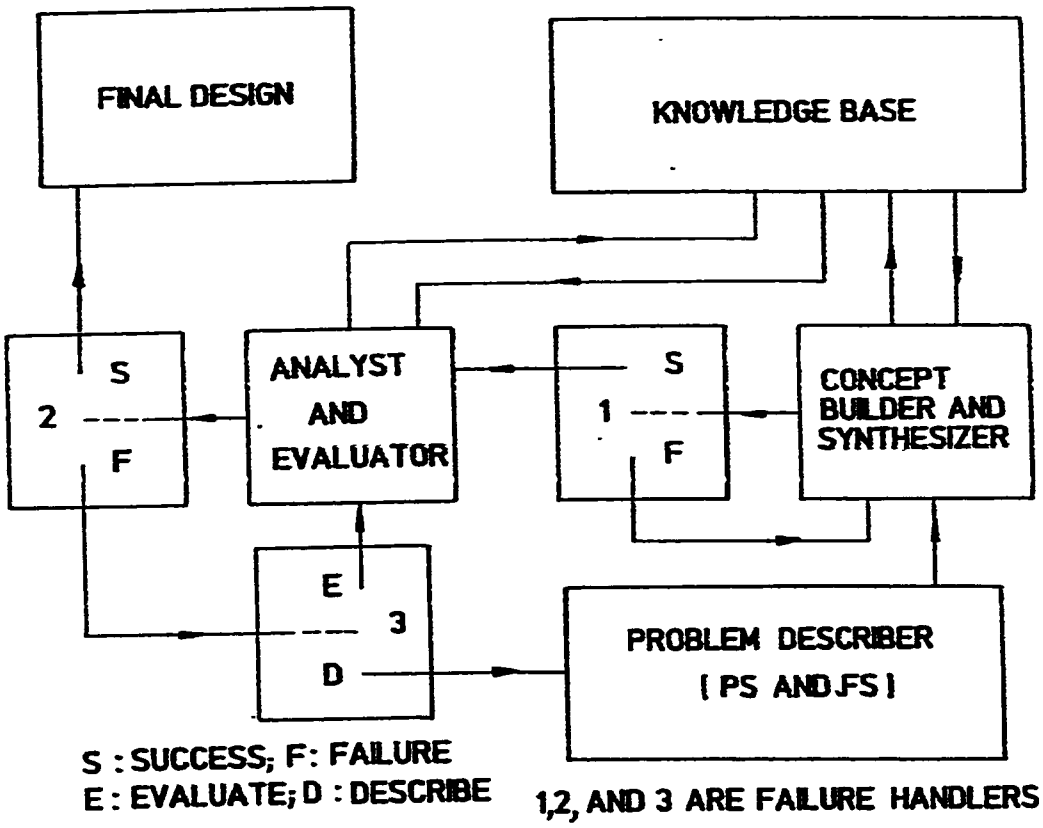


Fig. 3.1 A Schematic Diagram of the Design Process (PS: Prescriptive Specifications and FS: Functional Specifications) [6].

Brown [34] discussed an approach to handling failures that occur during design problem solving. The expert system 'AIR-CYL' built by Brown and Chandrasekaran [35] incorporates this failure theory. Brown and Chandrasekaran [35] use a hierarchical approach for routine design work (class III design problem) like the design of an air cylinder.

Dixon and Simmons [36] constructed an expert system to design standard V-belt drives (a class III design problem). The Design-Evaluate-Redesign architecture, an iterative method of design was the basis of this expert system called VEXPERT. Dixon et al., [37] described the first working version of a program called 'Dominic' which performs design by iterative redesign in a domain-independent manner. This has been successfully tested in four different domains of class III design problems. Dominic is a hill-climbing algorithm, similar to some standard optimization methods. However, its problem formulation or input language is more flexible. In the architecture based on iterative redesign, an initial design has to be created with the help of an inductive method which may not yield all feasible design solutions. Because of this drawback in creating the initial design an expert system based on the Design-Evaluate-Redesign architecture may not attempt to make an exhaustive search for all the feasible design solutions.

Bardasz and Zeid [11] proposed analogical problem solving to mechanical design, which is based on the fundamental principle that problem solving can be assisted by the review of solutions to past problems that have been attempted. Jacob and Froscher [77] suggested the idea of grouping rules and knowledge bases used at one time. Suh [46] uses axiomatic approach in developing axiomatic expert system.

The goal-driven or backward-chained (c.f. Sec. 2.1) architecture employing top-down search strategy in solving the present journal bearing design problem (class III design problem) is proposed by Guntur [6]. This architecture integrates the methodologies given by Brown [34], Bardasz and Zeid [11], and Jacob and Froscher [77]. The design process which starts with the definition of a problem and ends in a final design solution, if it is successful, can be modeled as sequence of states or stages. At the initial state (or stage 0) only a small number of functional and prescriptive specifications (or a short input symbol-structure) is available. This architecture is shown in Fig. 3.2. An operator is used to move from one state to another. For example to move from the initial state (stage 0) to state 1 (stage 1) operator 1 is used. This operator consists of a group of rules and a set of knowledge bases.

The database used by each operator is written onto a blackboard (or a part of the RAM of a computer) and it is retained on the blackboard as long as the operator is active. At the end of each state (or stage) an internal database consisting of all partial feasible solutions is created and is written onto the blackboard and all other data bases that are not required are erased from the blackboard. The communication between immediately past operator to the present operator is through the internal database of all partial feasible solutions created by the immediately past operator (see Fig. 3.2). Just before the final stage all feasible solutions are written onto an internal database. At the final stage all feasible solutions are evaluated.

One of the advantages of this architecture is that an exhaustive search is made to find the proper design solution(s). The other advantage is that by grouping knowledge bases and rules, and by limiting the communication

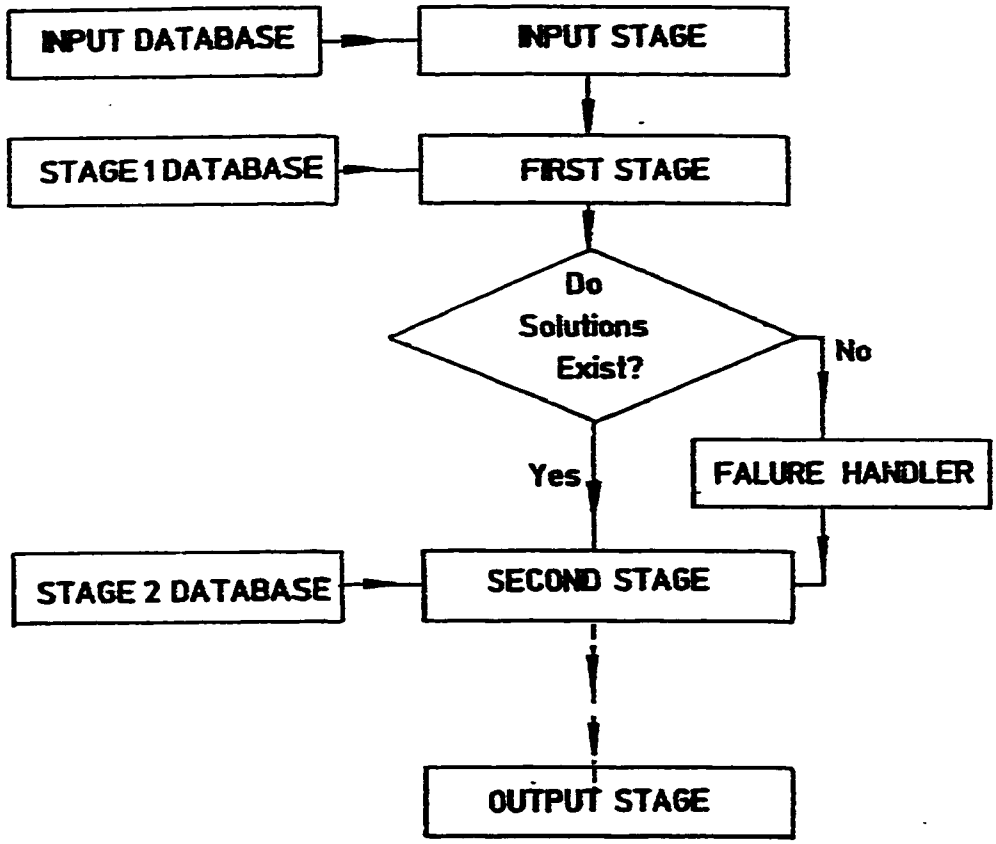


Fig. 3.2 Architecture of the Present Expert System (Some Stages Are Not Shown) [6].

between operators the memory of the computer is used efficiently. Of course, if the parametric constraints permit variations over wide ranges of these parameters there may be a combinatorial explosion of feasible solutions and memory problems may arise. This is a major disadvantage of the architecture used but the user may avoid this problem by properly altering the parametric ranges.

One more advantage of this architecture is that at the end of each stage the program reports the number of solutions available at that stage. If it is known that no solution is available, then the constraint of that stage can be modified suitably and the program can be run again. This way, a failure handler may be incorporated. A failure handler may also be built into the program. At each stage the discriminating and recombining skills of a designer may be employed to produce appropriate partial design solutions to a given problem.

3.3 CRITERIA FOR CHECKING FEASIBILITY OF THE SOLUTIONS

Different constraints are employed to discriminate between appropriate and inappropriate solutions. These are described below. This procedure of constraint-checking is illustrated with the help of numerical examples considered in the succeeding chapters 4 to 7.

3.3.1 Temperature-Rise Check

Juvinall [68] suggested that outlet temperature of the lubricant must be in the range of 93 to 121°C. In the present system it has been ensured that the outlet temperature should not be over 120°C. In case it is over 120°C in a solution, that solution is rejected. A failure handler may be used if there is not even a single solution at the end of this stage; i.e., this constraint may be

modified.

3.3.2 Clearance Check

The clearance depends to some extent on the desired quality. Small clearances can be maintained with high-grade workmanship but are expensive. Therefore when costs are to be reduced, generous clearances are used. Spotts [67] summarized the clearances used in industrial applications which has been presented in Table 3.1. He has given clearance limits for different diameter shafts upto 5.5 inch; this limit has been increased to 200 mm. Therefore the bearing diameter has been limited to 200 mm in the present work. The clearances of each partial solution coming to this stage must be checked in the light of its manufacturability with the help of Table 3.1. A failure handler may be used if there is not even a single solution at the end of this stage. That is, the criterion can be changed by selecting the industrial application other than the previous one, or this constraint can be modified, or the maximum limit of the diameter can be increased.

3.3.3 Minimum Film Thickness Check

From the graph of minimum oil film thickness variable h_0/C versus Sommerfeld number S (see Fig. 4.1 taken from [17]), it is clear that h_0/C reduces, if S decreases. The value of S may be lowered if the unit load P or the clearance C is increased during operation [c.f. Eq. (2.1)]. In order to ensure that h_0 remains large enough, Juvinall [68] suggests the relationship $h_0 \geq 0.005 + 0.00004D$ (h_0 and D *are in mm*), to be used with a safety factor of 2 taken for S. This means that even if the load P is twice of its original value during operation or clearance is $\sqrt{2}$ times its initial value (i.e., 40% increase)

Table 3.1. Bearing Clearances in Industrial Applications [67].

	Running clearance(mm) for shaft diameter under				
Application	12 mm	25 mm	50 mm	100 mm	200 mm*
Precision spindle practice: 1. P < 3.5 MPa and V < 2.54 m/s	0.00635 to 0.01905	0.01905 to 0.0381	0.0381 to 0.0635	0.0635 to 0.0889	0.0889 to 0.127
Precision spindle practice: 2. Otherwise	0.0127 to 0.0254	0.0254 to 0.0508	0.0508 to 0.0762	0.0762 to 0.1143	0.1143 to 0.1651
Electric motor and generator practice	0.0127 to 0.0381	0.0254 to 0.0508	0.0381 to 0.0889	0.0508 to 0.1016	0.0762 to 0.1524
General machine practice (continuous rotating motion)	0.0508 to 0.1016	0.0635 to 0.1143	0.0762 to 0.127	0.1016 to 0.1778	0.127 to 0.2032
General machine practice (oscillating motion)	0.0635 to 0.1143	0.0635 to 0.1143	0.0762 to 0.127	0.1016 to 0.1778	0.127 to 0.2032
Rough machine practice	0.0762 to 0.1524	0.127 to 0.2286	0.2032 to 0.3048	0.2794 to 0.4064	0.3556 to 0.508

*In the reference it is 5.5 inch, but here it is extended to 200 mm.

due to wear, the film thickness will be in the acceptable range. To apply this rule, the S is halved and then corresponding h_o is obtained to check the feasibility. A failure handler may be used if there is not even a single solution at the end of this stage; that is, this constraint may be modified.

3.3.4 Stability Check

Rotordynamic instability has been discussed by Vance [72], Rao [73], and Goodwin [78]. Majority of problems encountered in rotor-bearing systems involves synchronous whirl (vibration), when shaft whirling is synchronous with shaft speed, and it is produced by rotor imbalance. Amplitudes of synchronous whirl can be minimized by: a) balancing the rotor, b) changing the speed (so that it is not close to the critical speed), and c) adding damping to the rotor-bearing system.

The remaining minority of problems involving nonsynchronous whirl are:
i) supersynchronous vibrations due to shaft misalignment,
ii) subsynchronous and super synchronous vibrations due to cyclic variations of parameters caused by loose bearing housings or shaft rubs, and
iii) nonsynchronous (generally subsynchronous) rotor whirling that becomes unstable when a certain speed called the threshold speed of instability is reached.

Problems of the first and second types have solutions: align the shafts, tighten the bearing housing, or eliminate the rub. Problems of the third kind do not have such solutions and are referred to as rotordynamic instability problems. Since instability frequencies are subsynchronous, they almost always occur when shaft speeds are higher than the natural whirling frequency.

Hydrodynamic journal bearings are the most common source of rotordynamic instability where rotor has self-excited vibrations due to fluid film forces. There is a threshold speed, above which the rotor-bearing system becomes unstable. This is called 'oil whip' and characterized by subsynchronous whirling. The threshold speed of instability is generally about twice the first critical speed. It is the cross-coupled stiffness of journal bearings which destabilizes the system. Although the damping of such bearings is high, it is not enough to suppress the oil whip at high rotor speeds. When journal bearings are lightly loaded (i.e., operating at a very small eccentricity ratio ε and pressure developed in the film is insignificant), the shaft is often observed to whirl at a frequency just below one-half of the shaft speed (0.46 - 0.48 times the shaft speed), even at speeds below the threshold speed of oil whip instability. This is termed as 'oil whirl' or 'half-speed whirl'. This reduces the bearing load-carrying capacity to zero and makes the rotor unstable. Thus oil whirl may be a harbinger of unstable oil whip at a higher shaft speed and consequent destruction of the bearings is a possibility. Pressure-dam bearings are designed to supplement the actual supported load with hydrodynamic pressure in the oil film and thereby increase the eccentricity ratio at high speeds.

Badgley and Booker [76] investigated stability of full journal bearings with a fixed, rigid, and perfectly balanced rotor under the assumption of light initial impact and plane-motion for the short-bearing, long-bearing, and finite-length bearing approximations. In all cases examined, instability was not observed above a static eccentricity ratio ε of 0.83. They presented instability threshold curves for all three bearing assumptions after small initial-velocity disturbances, which has been shown in Fig. 3.3. This stability map shows the dimensionless threshold

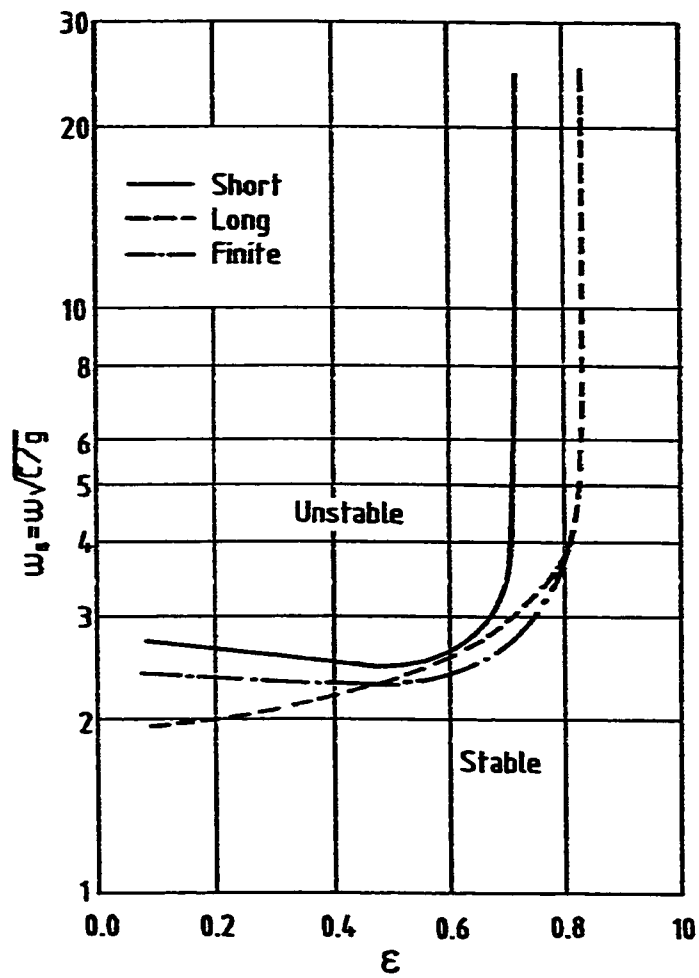


Fig. 3.3 Instability Threshold Curves for Full Journal Bearings for Indicated Assumptions After Small Initial-Velocity Disturbances [76].

threshold speed of instability ω_s versus the equilibrium eccentricity ratio ε. The dimensionless speed $\omega_s = \omega\sqrt{C/g}$, where $\sqrt{g/C}$ is the natural frequency of a rigid rotor supported on a spring with a static deflection of C (the radial clearance) and g is the acceleration of gravity. Typically, these curves are generated by computing the dimensionless speed at which the real part of the eigenvalue becomes positive for various values of static eccentricity ratio. It is obvious from the figure that for stability of finite-length full journal bearings either ε must be ≥ 0.83 or dimensionless speed ω_s should be ≤ 2.33.

A failure handler may be used if there is not even a single solution at the end of this stage; that is, this constraint may be modified.

For partial bearings there is no published work that contains a stability criterion, i.e., objective knowledge is lacking. Therefore the stability check is not performed in the case of partial bearings.

3.4 DECISION MAKING STAGE

In this stage, inductive procedures have to be used to make the final decision. A set of rules to evaluate each of the design solutions that satisfy all the constraints discussed in the previous section is used in this stage. Basically a multiparameter, multiobjective optimization problem has to be solved in this stage. This optimization problem is solved by using the utility function method [6,36,46,59,60]. Pertaining to judgments or decisions of preference, the concepts of interest are those of value or utility and weight.

A value function [81] represents a formalization of an individual's value or preference structure for decision problems under conditions of certainty; it serves

to compare various levels of different attributes indirectly. In particular, a value function u associates a real number $u(x)$ to each point in an evaluation space, provided that, if the decision maker prefers x_i to x_j, then he prefers $u(x_i)$ to $u(x_j)$. In essence, the value function rescales the values of an attribute X. Utility functions [81] are appropriate for the assessment of preferences under conditions of uncertainty. They are more complex to assess than value functions, since the decision maker's preferences must be taken into account. Utility function is a value function, although the converse is not necessarily true. In the present case, decision is made under conditions of certainty, therefore the utility functions and the value functions are identical.

Weight [81] is the importance of a piece of information to individual's judgments of preference about states of the world. Scaling constants [81] are used to weight the various unidimensional value or utility functions that enter a multidimensional utility function under conditions of certainty. For example, in the case of two attributes X and Y, if the decision maker is indifferent, then the two scaling constants (weighting factors) are equal, i.e., $w_x = w_y$; if the option with the highest X value is preferred, then $w_x > w_y$; and, if the option with the highest Y value is preferred, then $w_y > w_x$.

In the present work, the parameters used in the decision making process are the 'L/D ratio', 'temperature'(in °C), 'C' (the clearance in mm), ' h_0 ' (the minimum film thickness in mm), and 'T' (the torque required in N.m). In each solution of the previous stage all decision parameters are normalized except the L/D ratio, which itself is the decision variable. The ratio of $(T_{out} - T_{in})$ and $(120 - T_{in})$ is the normalized variable for the 'temperature' parameter. The

ratio of C and the minimum clearance limit corresponding to each solution of the previous stage C_{min} is the normalized decision variable for the 'clearance'. The ratio of h_0 and the first value of minimum film thickness for the solutions of the previous stage h_{0_1} is the normalized decision variable for the 'minimum film thickness'. The ratio of T and the first value of torque for the solutions of the previous stage T_1 is the normalized decision variable for the 'torque'.

Each utility function used in the present case is a nonlinear (step) function of a (normalized) decision variable and the functional relationship between a (normalized) decision variable and the corresponding utility function is stored in a database used in this stage. The final decision can be made on the basis of maximum load or minimum friction or optimal clearance. The database of this stage also contains the weighting factors which are to be used in the three cases for the five utility functions corresponding to the five decision variables.

The utility values and the weighting factors used in the present work are devised as reasonably as possible. They are presented in Tables 3.2 - 3.7. Welsh [63] suggests the optimum L/D ratio to be 0.6. Therefore this ratio has been assigned the maximum utility value of 1.0 and others have been penalized. The minimum temperature is the most preferable, therefore it corresponds to the highest utility value of 2.1 and higher temperatures have lower utility values. For the parameter 'clearance', the maximum utility value of 1.0 is assigned to the clearance C when it is equal to 1.2 times the minimum clearance limit C_{min}. Similarly, the largest value of h_0/h_{0_1} and the lowest value of T/T_1 have the highest utility values of 2.0 and 3.0, respectively. When decision criterion is maximum load, then h_0/h_{0_1} is the most important decision variable of all and

Table 3.2. Utility Value for the Decision Variable: 'L/D Ratio'.

Serial No.	Range	Utility Value
1	0.0 to 0.25	0.75
2	0.25 to 0.5	0.9
3	0.5 to 0.6	1.0
4	0.6 to 1.0	0.95
5	1.0 to 1.5	0.85
6	1.5 to 2.0	0.7

Table 3.3. Utility Value for the Decision Variable: 'Temperature' $\{(T_{out} - T_{in})/(120 - T_{in})\}$.

Serial No.	Range	Utility Value
1	0.0 to 0.1	2.1
2	0.1 to 0.2	2.05
3	0.2 to 0.3	1.98
4	0.3 to 0.4	1.9
5	0.4 to 0.5	1.8
6	0.5 to 0.6	1.68
7	0.6 to 0.7	1.54
8	0.7 to 0.8	1.38
9	0.8 to 0.9	1.2
10	0.9 to 1	1
11	1.0 to 20	0

Table 3.4. Utility Value for the Decision Variable: 'Clearance' (C/C_{min}).

Serial No.	Range	Utility Value
1	0.0 to 0.8	0.0
2	0.8 to 0.95	0.25
3	0.95 to 1	0.8
4	1.0 to 1.05	0.88
5	1.05 to 1.1	0.94
6	1.1 to 1.15	0.98
7	1.15 to 1.2	1.0
8	1.2 to 1.25	0.96
9	1.25 to 1.35	0.9
10	1.35 to 1.5	0.8
11	1.5 to 1.75	0.7
12	1.75 to 2.25	0.55
13	2.25 to 3.0	0.4
14	3.0 to 5.0	0.3
15	5.0 to 15	0.0

Table 3.5. Utility Value for the Decision Variable: 'Minimum Film Thickness' (h_o/h_{o_t}).

Serial No.	Range	Utility Value
1	0.0 to 0.1	0.05
2	0.1 to 0.2	0.12
3	0.2 to 0.3	0.2
4	0.3 to 0.4	0.3
5	0.4 to 0.5	0.4
6	0.5 to 0.6	0.5
7	0.6 to 0.7	0.6
8	0.7 to 0.8	0.7
9	0.8 to 0.9	0.8
10	0.9 to 1.0	0.85
11	1.0 to 1.2	0.9
12	1.2 to 1.4	0.95
13	1.4 to 1.6	1.0
14	1.6 to 1.8	1.1
15	1.8 to 2.0	1.2
16	2.0 to 2.25	1.35
17	2.25 to 2.5	1.5
18	2.5 to 2.75	1.63
19	2.75 to 3.0	1.75
20	3.0 to 3.5	1.85
21	3.5 to 5.0	1.92
22	5.0 to 10	1.98
21	10.0 to 50	2

Table 3.6. Utility Value for the Decision Variable: 'Torque' (T/T_1).

Serial No.	Range	Utility Value
1	0.0 to 0.1	3.0
2	0.1 to 0.3	2.9
3	0.3 to 0.4	2.8
4	0.4 to 0.6	2.7
5	0.6 to 0.8	2.5
6	0.8 to 1.0	2.25
7	1.0 to 1.25	2.0
8	1.25 to 1.5	1.7
9	1.5 to 1.75	1.4
10	1.75 to 2.0	1.2
11	2.0 to 2.5	1.0
12	2.5 to 3.5	0.8
13	3.5 to 5.0	0.6
14	5.0 to 7.0	0.4
15	7.0 to 10	0.2
16	10.0 to 50	0.0

Table 3.7. Weighting Factors.

	Decision Variables				
Basis	'L/D ratio'	'temperature' $\frac{T_{out}-T_{in}}{120-T_{in}}$	'clearance' C/C_{min}	'min. film thickness' h_0/h_{0_1}	'torque' T/T_1
1. Maximum Load	0.3	0.35	0.4	1.0	0.4
2. Minimum Friction	0.3	0.35	0.4	0.4	1.0
3. Optimal Clearance	0.3	0.35	1.0	0.4	0.4

therefore, it has the highest weighting factor of 1.0. Similarly, in case of minimum friction and optimal clearance, T/T_1 and C/C_{min} have the maximum weighting factors of 1.0, respectively. Since the utility functions have been used for the first time for journal bearing design, these knowledge bases are not tested thoroughly. In future, if an expert wants to modify them he can change these databases. However, there will be no change in the computer program.

The sum of the weighted values of all the five utility functions (U) for each design solution of the previous stage is determined with the help of the following relation

$$U = (wu)_{L/D} + (wu)_{(T_{max}/T_{min})/(T_{max}/T_{min})} + (wu)_{C/C_{min}} + (wu)_{k_a/k_{a_1}} + (wu)_{T/T_1} \qquad (3.1)$$

where, w is the weighting factor, u is the utility value, and subscript denotes the decision variable.

In this stage, the design solution of the previous stage for which the utility factor is the highest (U_{max}) is identified. In addition, all the design solutions of the earlier stage for which the utility factors are at least ninety five percent (95%) of the U_{max} are identified. In this stage only the discriminating skills of a designer are used. Therefore the number of solutions at the end of this stage is less than or equal to the number of solutions of the previous stage.

3.5. OUTPUT STAGE

The goal of this stage is to file all the chosen solutions. Each design solution for which the utility value is atleast ninety five percent (95%) of the highest utility value is filed in the 'results.out' file for the perusal of the user.

3.6. PROGRAMMING LANGUAGE

The computer programming languages used most commonly in the expert systems are briefed earlier (c.f. Sec. 1.1). The programming language used to write this goal-driven rule-based production system is Turbo Prolog, which is a fifth-generation, declarative, and very high-level programming language [50-52]. Turbo Prolog is used because of its database handling and windowing capabilities. In addition, at various stages of this production system the backtracking facility of Turbo Prolog is efficiently used.

On the basis of the architecture described in this chapter, expert systems have been developed for designing full and partial journal bearings for the maximum load, the minimum friction, and the optimal clearance conditions. Their working with numerical example is explained in the succeeding chapters 4 to 7.

CHAPTER 4

DESIGN OF FULL JOURNAL BEARINGS FOR MAXIMUM LOAD AND MINIMUM FRICTION

The objective knowledge required for designing full journal bearings for the maximum load(W) and the minimum friction(f) is collected from the following sources: Raimondi and Boyd [24], Keith, Jr. [15-16], Shigley [17], Orthwein [19], Juvinall [68], and Al-Dukhaiyel [66]. The values of h_0/C taking film rupture into account for full journal bearings at L/D = 0.25, 0.5, 1.0, and ∞ are given by Raimondi and Boyd [24] or can be obtained from Shigley's chart [17] shown in Fig. 4.1, which also gives the corresponding values of the Sommerfeld number S. Once S is known, all the performance variables are obtained from the charts of [17,24]. Then, the values of the desired performance variables at L/D = 0.6, 1.5, and 2.0 are found with the help of the following equation suggested by Raimondi and Boyd [23]:

$$y_{L/D} = \frac{1}{(L/D)^3}\Big[-(1/8)(1-L/D)(1-2L/D)(1-4L/D)y_{\infty} + (1/3)(1-2L/D)(1-4L/D)y_1$$

$$- (1/4)(1-L/D)(1-4L/D)y_{1/2} + (1/24)(1-L/D)(1-2L/D)y_{1/4}\Big] \qquad (4.1)$$

where, y = desired performance variable and the subscript of y is the L/D ratio at which the variable is being evaluated. For example, y = h_0/C and its value for the max. W condition at L/D = 0.6 is to be found. The values of h_0/C at L/D ratio of 1/4, 1/2, 1, and ∞, for the max. W condition (given in [24]) are 0.272, 0.427, 0.533, and 0.655. Then the value at L/D = 0.6 is obtained as 0.46 using the above equation.

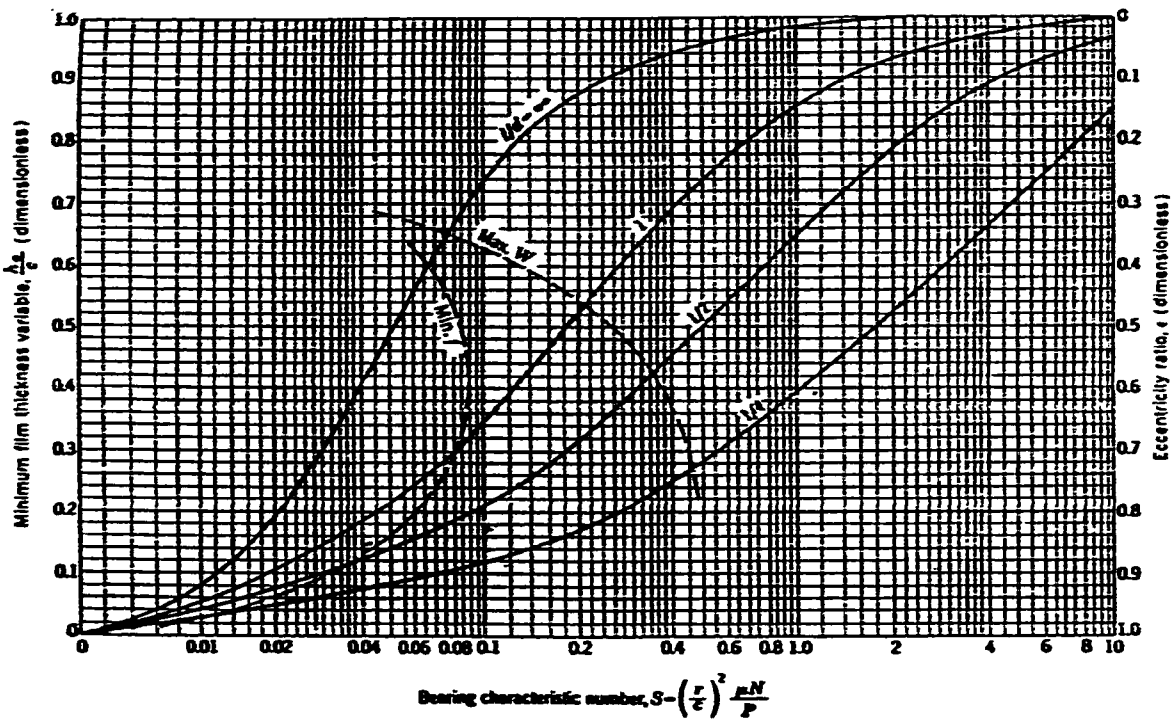


Fig. 4.1 Chart for Minimum Film Thickness Variable and Eccentricity Ratio [17].

An attempt was made to find bearing stiffness and damping coefficients too, but it was only partially successful. Rao [73] gave charts for only L/D ratio of 0.5 and 1. Seireg and Dandage [65] obtained mathematical expressions for the dynamic properties of a full journal bearing for L/D ratio of 0.25 to 1.0, but the values obtained did not match (specially for L/D = 0.5) with those obtained from Rao's figures [73]. It seemed that some of the coefficients in the expressions were misprinted. In order to avoid this descripancy Seireg and Dandage were contacted; their input was not received, yet. Lund and Saibel's equations [74] did not yeild any useful information. There is a definite need to expand the knowledge-base pertaining to stiffness and damping coefficients.

The present program uses the knowledge base created by Siereg and Dandage [65] when applicable. It is important to note here that the integrity of this knowledge-base is questionable. The creation of a dependable knowledge base for stiffness and damping coefficients will be one of the future research projects one can work on.

As discussed in the previous chapter, the language used to write this goal-directed rule-based production system is Turbo Prolog and the program is divided into stages from input to output.

4.1 VISCOSITY - TEMPERATURE RELATIONSHIPS

To obtain absolute viscosity μ (mPa-s) of different grades of the SAE (10, 20, 30, 40, 50 60, and 70) oils at various operating temperatures (°C), Al-Dukhaiyel [66] has given the following equations by curve fitting the data obtained from the chart given in Shigley [17], which has been presented in Fig. 4.2. The equations are checked and its accuracy is found to be within ± 5 %.

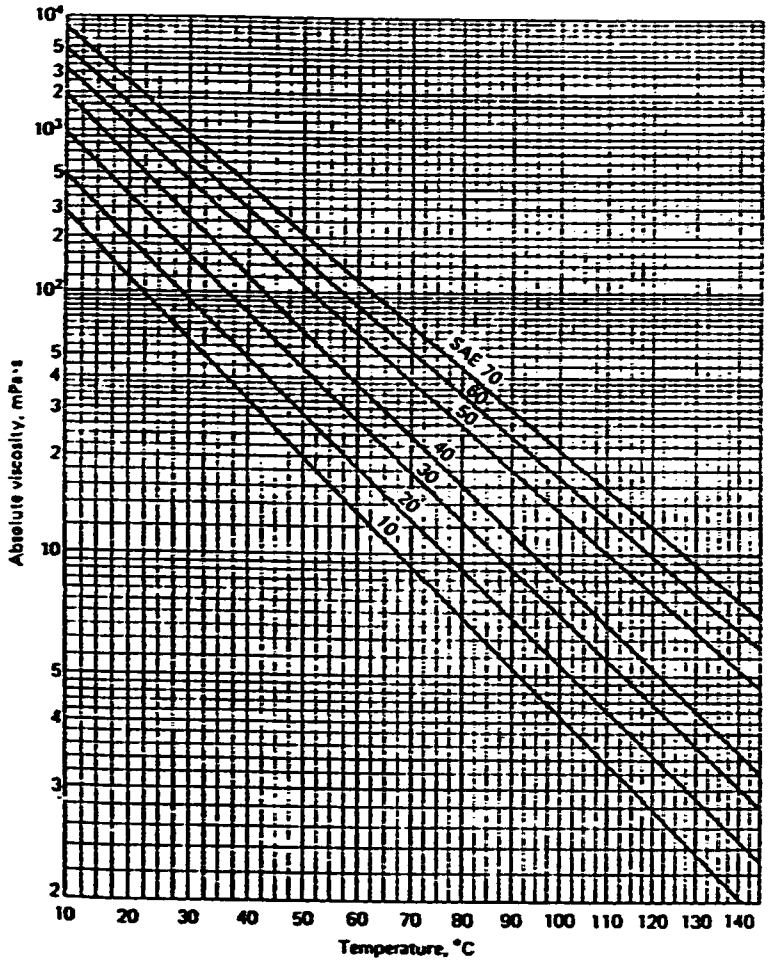


Fig. 4.2 Viscosity - Temperature Chart [17].

4.1.1 Temperature from 10 - 50 °C

$$\mu_{10} = 1020 - 765 \times \log_{10}(T_{cp}) - 100 \times [\log_{10}(T_{cp})]^2 + 120 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{20} = 2881 - 3740 \times \log_{10}(T_{cp}) + 1519 \times [\log_{10}(T_{cp})]^2 - 180.3 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{30} = 6776 - 9685 \times \log_{10}(T_{cp}) + 4562 \times [\log_{10}(T_{cp})]^2 - 702 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{40} = 16604 - 25767 \times \log_{10}(T_{cp}) + 13391 \times [\log_{10}(T_{cp})]^2 - 2327 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{50} = 23117 - 32564 \times \log_{10}(T_{cp}) + 14723 \times [\log_{10}(T_{cp})]^2 - 2076 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{60} = 44373 - 69067 \times \log_{10}(T_{cp}) + 35935 \times [\log_{10}(T_{cp})]^2 - 6241 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{70} = 54465 - 76243 \times \log_{10}(T_{cp}) + 33875 \times [\log_{10}(T_{cp})]^2 - 4579 \times [\log_{10}(T_{cp})]^3$$

4.1.2 Temperature from 50 - 90 °C

$$\mu_{10} = 2752.82 - 4118.6 \times \log_{10}(T_{cp}) + 2072 \times [\log_{10}(T_{cp})]^2 - 349.97 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{20} = 3685.05 - 5480.06 \times \log_{10}(T_{cp}) + 2740.02 \times [\log_{10}(T_{cp})]^2$$

$$- 460.14 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{30} = 10310.8 - 15857 \times \log_{10}(T_{cp}) + 8173.8 \times [\log_{10}(T_{cp})]^2$$

$$- 1410.85 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{40} = 14489 - 22107 \times \log_{10}(T_{cp}) + 11299 \times [\log_{10}(T_{cp})]^2$$

$$- 1933 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{50} = 24686.4 - 37622 \times \log_{10}(T_{cp}) + 19204 \times [\log_{10}(T_{cp})]^2$$

$$- 3281 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{60} = 27503.7 - 41138 \times \log_{10}(T_{cp}) + 20607 \times [\log_{10}(T_{cp})]^2$$

$$- 3455 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{70} = 30840.3 - 44723 \times \log_{10}(T_{cp}) + 21664.84 \times [\log_{10}(T_{cp})]^2$$

$$- 3503.77 \times [\log_{10}(T_{cp})]^3$$

4.1.3 Temperature from 90 - 140 °C

$$\mu_{10} = 1392.5 - 1915.2 \times \log_{10}(T_{cp}) + 884.3 \times [\log_{10}(T_{cp})]^2$$

$$- 136.9 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{20} = 6.7 + 174.4 \times \log_{10}(T_{cp}) - 161 \times [\log_{10}(T_{cp})]^2$$

$$+ 36.73 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{30} = 3064.49 - 4238.3 \times \log_{10}(T_{cp}) + 1964.9 \times [\log_{10}(T_{cp})]^2$$

$$- 305.08 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{40} = 3271.5 - 4395.5 \times \log_{10}(T_{cp}) + 1976.3 \times [\log_{10}(T_{cp})]^2$$

$$- 297.15 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{50} = 5354.64 - 7213 \times \log_{10}(T_{cp}) + 3251.7 \times [\log_{10}(T_{cp})]^2$$

$$- 490.28 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{60} = 6698.23 - 8984.6 \times \log_{10}(T_{cp}) + 4033.4 \times [\log_{10}(T_{cp})]^2$$

$$- 605.69 \times [\log_{10}(T_{cp})]^3$$

$$\mu_{70} = 9093 - 12210 \times \log_{10}(T_{cp}) + 5485.5 \times [\log_{10}(T_{cp})]^2$$

$$- 824.17 \times [\log_{10}(T_{cp})]^3$$

The above expressions are used in the present system as viscosity-temperature relationships. Different stages of the program for designing full journal bearings for the max. W and the min. f conditions, are described below.

4.2 INPUT STAGE (STAGE 0)

The functional and prescriptive specifications (c.f. Sec. 1.4) available at this stage (stage 0) in the 'Expert System' for designing full journal bearings for maximum load and minimum friction conditions are listed below:

1. the bearing radial load to be supported: W (kN),
2. the speed of the journal: N_j (*rpm*),
3. the bearing unit load: P (MPa), the user either enters the value of P or selects the application from the menu where the bearing would be used so that P can be obtained from the database of stage 0,
4. the bearing industrial application for selecting the allowable range of the clearance,
5. the lubricant inlet temperature: T_{in} (°C),

6. the decision criterion for utility value, and
7. the L/D ratio (0.25, 0.5, 0.6, 1.0, 1.5, 2.0, or all six).

The group of rules and databases used at this stage are:

1. a set of rules for menu-handling and screen-handling.
2. a set of rules for prompting and entering functional and prescriptive specifications,
3. if the user selects the application instead of entering the value of P, a knowledge base in the form of a table [15-17,67-68] as shown in Table 4.1 is used (the program selects the lower value),
4. a knowledge base containing all relevant L/D ratios (see Table 4.2),
5. a set of rules to find the length and corresponding diameter of the bearing using $P = W/(L.D)$.

Thus each partial solution of stage 0 consists of the L/D ratio, length, and diameter. In this stage, recombining skills of a designer are used. If the user selects all the six L/D ratios, then there will be 12 partial solutions (6 L/D ratios combined with 2 optimization conditions: max. W and min. f). At the end of this stage all partial solutions are written onto an internal database. All databases of stage 0 are then retracted except the newly written database of all partial solutions of stage 0.

The rules and databases used in designing full journal bearings for max. W and min. f are divided into six more stages as follows:

Table 4.1. Unit Load P in Current Use for Journal Bearings [15-17,67-68].

Application	Range of P MPa	P selected MPa
Diesel engines: 1. Main bearings	6 - 12	6
2. Crankpin	8 - 15	8
3. Wristpin	14 - 15	14
Electric motor bearings	0.8 - 1.5	0.8
Steam turbine bearings	0.8 - 1.5	0.8
Gear reducer bearings	0.8 - 1.5	0.8
Automative gasoline engines: 1. Main bearings	4 - 5	4
2. Crankpin	10 - 15	10
Air compressors: 1. Main bearings	1 - 2	1
2. Crankpin	2 - 4	2
Centrifugal pump bearings	0.6 - 1.2	0.6
Marine lineshaft bearings	0.17 - 0.25	0.17
Aircraft engine connecting rod bearings	4.8 - 13.8	4.8
Roll neck bearings	10 - 17.5	10
Railway axle bearings	2 - 2.4	2
Light lineshaft bearings	0.1 - 0.2	0.1
Heavy lineshaft bearings	0.7 - 1.0	0.7

Table 4.2. Slenderness Ratios Used in the Program.

Serial No.	L/D ratio
1	0.25
2	0.5
3	0.6
4	1.0
5	1.5
6	2.0

4.3 STAGE I

The goal of stage I is to obtain all partial feasible solutions which fulfil the outlet temperature condition that is $T_{out} \leq 120^{\circ}C$. Each partial feasible solution of this stage consists of the L/D ratio, length, diameter, all ten performance variables (corresponding to various functional specifications), namely, Sommerfeld or bearing characteristic number: S, minimum film thickness variable: h_0/C, temperature-rise variable: T_{var}, coefficient of friction variable: (R/C)f, attitude angle (position of the minimum film thickness): Φ (degree), flow variable: Q/RCNL, flow ratio: Q_s/Q, pressure ratio: P/p_{max}, position of maximum film pressure: θ_{Pmax} (degree), position at which film terminates: θ_{F0} (degree), and lubricant inlet and operating temperatures. To achieve this goal the following sets of rules and database are used:

1. a knowledge base consisting of all the performance variables: S, h_0/C, T_{var}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, θ_{Pmax}, and θ_{F0} in the form as presented in Table 4.3 (from [17,24] for L/D ratio of 1/4 1/2, & 1, and then obtained the rest using Eq. (4.1)),
2. a set of rules to determine the lubricant operating and outlet temperatures,
3. a set of rules to check whether or not the calculated outlet temperature in each case is below $120^{\circ}C$ (a condition discussed in Sec. 3.3.1).

The dimensionless temperature-rise variable is

$$T_{var} = (\rho C_H \Delta T)/P \tag{4.2}$$

where, ρ is the density, C_H the specific heat, and ΔT the temperature-rise from inlet to outlet of the lubricant. P is the bearing unit load. Average values of ρ and

Table 4.3. Performance Variables of Full Journal Bearings* [17,24].

L/D	S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}	$\theta_{p_{max}}$	θ_{p_0}
Corresponding to max. W condition										
1/4	0.46	0.272	40.0	12.4	36.4	5.37	0.848	0.272	13.7	45.3
1/2	0.345	0.427	31.0	9.0	48.8	4.8	0.718	0.374	17.5	66.0
0.6	0.301	0.46	27.553	7.565	52.413	4.639	0.694	0.391	17.768	71.98
1.0	0.208	0.533	20.0	4.9	59.0	4.1	0.56	0.47	17.4	86.0
1.5	0.161	0.572	16.117	3.438	61.536	3.685	0.431	0.543	16.632	94.016
2.0	0.139	0.592	14.198	2.885	62.519	3.434	0.346	0.591	16.073	98.27
Corresponding to min. f condition										
1/4	0.012	0.03	4.05	1.0	12.6	6.12	0.99	0.11	4.4	18.0
1/2	0.036	0.116	7.2	1.8	25.2	5.65	0.932	0.21	11.7	35.0
0.6	0.053	0.165	8.808	2.09	31.003	5.357	0.917	0.243	14.749	43.126
1.0	0.078	0.3	11.0	2.4	43.7	4.48	0.76	0.365	18.9	64.0
1.5	0.082	0.386	11.149	2.331	50.131	3.872	0.59	0.464	19.221	76.765
2.0	0.081	0.434	10.892	2.221	53.266	3.521	0.477	0.526	18.772	83.734

*Values at L/D = 0.6, 1.5, and 2 are obtained using Eq. (4.1) suggested by [23].

and C_H are 861 kg/m^3 and 1760 $J/kg-°C$ [17]. Thus from Eq. (4.2), ΔT the temperature-rise can be obtained. Then the operating and outlet temperatures, respectively are

$$T_{op} = T_{in} + (\Delta T)/2.$$

$$T_{out} = T_{in} + \Delta T.$$

If no partial solution within the desired outlet temperature limit is obtained this condition may be modified to act as a failure handler [34]. In this case each partial solution will have the L/D ratio, length, diameter, Sommerfeld or bearing characteristic number: S, minimum film thickness variable: h_0/C, temperature-rise variable: T_{op}, coefficient of friction variable: (R/C)f, attitude angle (position of the minimum film thickness): Φ, flow variable: Q/RCNL, flow ratio: Q_s/Q, pressure ratio: P/p_{max}, position of maximum film pressure: $\theta_{p_{max}}$, position at which film terminates: θ_{p0}, T_{op}, and T_{in}. At the end of this stage all partial solutions are written onto an internal database. All databases of stage 1 are then retracted except the newly written database of all partial solutions of stage 1. The database containing partial solutions of stage 0 is also erased.

In stage 1 discriminating skills of a designer are used to discriminate between appropriate and inappropriate solutions. At this stage recombining skills of a designer are not used. The number of partial solutions of stage 1 is N_1.

4.4 STAGE 2

The final goal of stage 2 is to get the oil-grade and determine the viscosity for each of the partial design solutions of stage 1. The group of rules and the

database used in this stage are given below:

1. a set of rules to get the oil-grade selected by the user (SAE 10, 20, 30, 40, 50, 60, 70, or all seven),
2. a set of rules to determine the viscosity of the oil using the equations of Sec. 4.1,
3. a database consisting of the coefficients of the equations for determining the viscosity (c.f. Sec. 4.1), which is in the form of a table.

At the end of stage 2 there will be the same number of solutions as that of stage 1 if only one oil-grade is tried or seven times the number of solutions of stage 1 if all seven oil-grades are tried. At this stage no failure handler is used. Each partial solution of stage 2 consists of the oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{max}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{p_{max}}$, θ_{cr}, T_{cr}, T_{in}, and μ.

At the end of stage 2 all partial solutions are written onto an internal database (on the blackboard). The database of stage 2 containing coefficients of the equations to determine viscosity and the database consisting of all partial solutions of stage 1 are erased. In stage 2, the recombining skills of a designer are used. The number of partial solutions of stage 2 is N_2.

4.5 STAGE 3

The final goal of this stage is to determine the clearance and check whether this is in the prescribed range for each partial solution of stage 2. The group of rules and the database used in this stage are given below:

1. a set of rules to determine the clearance using $S = (R/C)^2 \mu N/P$.
2. a set of rules to check whether the calculated clearance in each case is within

the prescribed range,

3. a database consisting of clearance limits depending on the industrial application in the form of a table [67] (c.f. Table 3.1).

If no partial solution within the desired clearance limit for manufacturability is obtained, this condition may be modified by a failure handler [34]. Each partial solution of stage 3 will have the load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{rw}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, θ_{Pmax}, θ_{P0}, T_{cp}, T_{in}, μ, C, and corresponding minimum limit of clearance C_{min}. At the end of stage 3 all partial solutions are written onto an internal database. The database consisting of all partial solutions of stage 2 is erased. The database of stage 3 is also erased.

For each partial solution of stage 2 there may be either no solution or one solution at stage 3. A failure handler is used if there is not even a single solution; that is, the industrial application is changed or acceptable clearance limits are modified or the maximum diameter limit of 200 mm is increased. In stage 3 only discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_3 will be less than or equal to the number of partial solutions in the previous stage N_2.

4.6 STAGE 4

The final goal of this stage is to determine whether or not each of the partial solutions of stage 3 satisfies the minimum film thickness criterion given by Juvinall [68] (c.f. Sec. 3.3.3); that is, for a factor of safety of 2, $h_0 \geq 0.005 + 0.00004D$ (h_0 and D *are in mm*). The group of rules and the database

used in this stage are given below:

1. a database consisting of S with safety factor of 2 and corresponding h_0/C which is in the form of Table 4.4 [17,24],
2. a set of rules to determine the value of h_0 and check whether it is in the acceptable range.

Each design solution of stage 4 will have the unit load P, speed N, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{rm}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{p_{max}}$, θ_{p0}, T_{cp}, T_{in}, μ, C, corresponding minimum limit of clearance C_{min}, and $h_{0_{F.S.=2}}$. A failure handler may be used if there is no design solution at the end of this stage; that is, this constraint may be modified.

At the end of stage 4 all partial solutions are written onto an internal database and at the same time the database consisting of all partial solutions of stage 3 is erased. The database of stage 4 is also retracted. In stage 4 only the discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_4 will be less than or equal to the number of partial solutions in the previous stage N_3.

4.7 STAGE 5

This stage has been divided into two parts:

4.7.1 Stage 5.1 or Stability Check

The final goal of this stage is to determine whether or not each of the partial solutions of stage 4 is stable (c.f. Sec. 3.3.4). The group of rules used in this stage are given below:

Table 4.4. Values* of h_o/C for Full Journal Bearings with a Factor of Safety of 2 [17,24].

L/D	S	h_o/C
Corresponding to max. W		
1/4	0.23	0.182
1/2	0.1725	0.29
0.6	0.1507	0.3154
1.0	0.104	0.36
1.5	0.0807	0.3763
2.0	0.0695	0.3824
Corresponding to min. f		
1/4	0.006	0.02
1/2	0.018	0.07
0.6	0.0264	0.1021
1.0	0.039	0.18
1.5	0.0411	0.2243
2.0	0.0406	0.2475

*Values at L/D = 0.6, 1.5, and 2 are obtained using Eq. (4.1) suggested by [23].

1. a set of rules to determine actual h_0, coefficient of friction f, torque T ($=fWR$), eccentricity ratio $\varepsilon(=1-h_0/C)$, and dimensionless speed ω_s ($=\omega\sqrt{C/g}, \omega=2\pi N$).
2. a set of rules to check whether the design solution lies in the stable range, i.e. either $\varepsilon \geq 0.83$ or $\omega_s \leq 2.33$.

Each design solution of this stage will have the unit load P, speed N, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{rw}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{p_{max}}$, θ_{p0}, T_{rp}, T_{ie}, μ, C, corresponding minimum limit of clearance C_{min}, $h_{0_{F.S.=2}}$, actual h_0, torque T, eccentricity ratio ε, and dimensionless speed ω_s. A failure handler may be used if there is no design solution at the end of this stage. That is, this constraint may be modified.

At the end of stage 5.1 all partial solutions are written onto an internal database and at the same time the database consisting of all partial solutions of stage 4 is erased. In stage 5.1 only the discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_5 will be less than or equal to the number of partial solutions in the previous stage N_4.

4.7.2 Stage 5.2 or Determination of Utility Value

The final goal of this stage is to determine the utility value of all the design solutions of stage 5.1 (c.f. Sec. 3.4). The group of rules and databases used in this stage are given below:

1. a set of rules to get the decision-criterion entered by the user in the input stage (i.e. maximum load or minimum friction).
2. a knowledge base consisting of the utility values corresponding to all the five

decision variables and the weighting factors for both the decision criteria (c.f. Tables 3.2 to 3.7),

3. a set of rules to determine the utility value for each design solution of stage 5.1, using the following equation

$$U = (wu)_{L/D} + (wu)_{(T_{max} - T_{in})/(T_{in})} + (wu)_{C/C_{min}} + (wu)_{h_0/h_{0_1}} + (wu)_{T/T_1} \qquad (4.3)$$

where, w is the weighting factor, u is the utility value, and subscript denotes the decision variable.

Each design solution of stage 5.2 will have the unit load P, speed N, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{max}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{p_{max}}$, θ_{p0}, T_{op}, T_{in}, μ, C, corresponding minimum limit of clearance C_{min}, $h_{0_{F.S.-2}}$, actual h_0, torque T, eccentricity ratio ε, dimensionless speed ω_s, and the utility value U. This stage is a simple procedural stage. Neither the discriminating nor the recombining skills of a designer are used at this stage. Therefore, the number of design solutions of this stage will be the same as that of stage 5.1 which is N_5.

At the end of stage 5.2 all partial solutions are written onto an internal database. The design solutions of stage 5.1 are erased. The databases of this stage are also erased. The user can modify the databases of Tables 3.2 to 3.7.

4.8 STAGE 6

This stage also consists of two parts:

4.8.1 Stage 6.1 or Decision Making Stage

The goal of this stage is to identify the design solution of the previous stage which has the highest utility factor U_{max}. In addition, all the design solutions of the previous stage for which the utility factors are at least ninety five percent (95%) of the U_{max} are identified.

The group of rules and databases used in this stage are given below:

1. a set of rules to identify the design solution which has the highest utility value U_{max},
2. a set of rules to identify the design solutions which have utility values of at least 95% of U_{max}.

In this stage only the discriminating skills of a designer are used. Therefore the number of solutions at the end of this stage N_6 is less than or equal to the number of solutions of the previous stage N_5.

Each design solution of stage 6.1 will have the unit load P, speed N, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, $T_{i,o}$, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{p_{max}}$, θ_{p0}, T_{op}, T_{in}, μ, C, corresponding minimum limit of clearance C_{min}, $h_{0_{F.S.=2}}$, actual h_0, torque T, eccentricity ratio ε, dimensionless speed ω_s, and the utility value U. At the end of stage 6.1 all partial solutions are written onto an internal database.

4.8.2 Stage 6.2 or Output Stage

The goal of this stage is to file all the design solutions that qualify the utility value criterion. Each design solution for which the utility value is at least ninety five percent (95%) of the highest utility value (U'_{max}) is printed in the 'results.out' file for the perusal of the user.

The group of rules and databases used in this stage are given below:

1. a set of rules and databases to find stiffness and damping coefficients using the relations given by Seireg and Dandage [65] in terms of S and L/D (for $1/4 \leq L/D \leq 1$),
2. a set of rules to calculate eccentricity e, coefficient of friction f, power loss H ($= 2\pi NT$), oil flow Q, side leakage Q_s, and maximum pressure p_{max}.
3. a knowledge base containing the L/D ratio, the optimization condition, i.e. maximum load or minimum friction, and the oil-grade,
4. a set of rules to print the selected design solutions in the 'results.out' file.

In stage 6.2, neither the discriminating nor the recombining skills of a designer are used. Therefore the number of solutions at the end of this stage is the same as that of stage 6.1, i.e. N_6. The design solutions of stage 5.2 are erased. At the end of this stage all the solutions are filed in the 'results.out' file. The design solutions of all the previous stages including stage 6.1 are erased and the databases used in this stage (stage 6.2) are also erased.

The 'results.out' file will have the functional and prescriptive specifications (c.f. Sec. 1.4) as load W, speed N, unit load P, bearing industrial application, lubricant inlet temperature, decision criterion, L/D ratio, optimization condition, oil-grade, length L, diameter D, T_{op}, μ, S, C, h_0, ε, eccentricity e, Φ, coefficient

of friction f, torque T, power loss H, total oil supplied Q, side leakage Q_s, maximum film pressure p_{max}, position of maximum film pressure $\theta_{p_{max}}$, terminating position of film θ_{p_0}, dimensionless velocity ω_s, stiffness coefficients (K_{XX}, K_{XY}, K_{YX} and K_{YY}), damping coefficients (C_{XX}, C_{XY}, C_{YX} and C_{YY}), and the utility value U.

4.9 EXAMPLE

To illustrate the use of the present 'Expert System' it is used to design full journal bearings for 1) maximum load and 2) minimum friction condition. This example is similar to the one given by Juvinall [68]. The input specifications for this example are as follows:

Bearing load = 17 kN

Speed of the journal = 1800 rpm

Bearing unit load = 1.511 MPa

Bearing industrial application = Electric motor and generator practice

Lubricant inlet temperature = 66.5 °C

L/D ratio = all six ratios (1/4, 1/2, 0.6, 1, 1.5, and 2)

Oil-grade = all seven oil-grades.

In this case the number of partial design solutions at the end of stage 1, N_1 is 12. The number of partial design solutions at the end of stage 2, N_2 is 84. At the end of stage 3 there are thirty one (N_3=31) solutions. The number of design solutions at the end of stage 4, N_4 is 31. At the end of stage 5 there are thirty one (N_5=31) solutions. The number of design solutions at the end of decision making stage N_6 will depend on the decision making criterion used.

4.9.1 Based on Maximum Load

On the basis of maximum load the 'Expert System' has found and filed nine design alternatives. The print-out of this file is attached on the succeeding pages. Out of these nine design alternatives, the solution with L/D = 2, optimization condition as maximum load, and SAE 50 oil has the maximum utility value = 3.58.

The load on the bearing, W = 17.00 kN

The speed of the journal, N = 30.000 rps

The bearing unit load, P = 1.511 MPa

The bearing-application for clearance is No. 2. Electric motor and generator practice

The inlet temperature, Tin = 66.50 deg C

The decision-criterion for utility is No. 1. Based on maximum load

Stable Design Alternatives Are

1. L/D = 1.5

Optimization condition: The Maximum Load

Design Based on SAE 30 Oil

Length, L = 130 mm

Diameter, D = 86.67 mm

Operating Temperature, Top = 74.54 deg. C

Viscosity of the Lubricant, Mu = 15.13 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.161

Radial Clearance, C = 0.059 mm

Minimum film thickness, hmin = 0.034 mm

Eccentricity Ratio, e/C = 0.428

Eccentricity, e = 0.025 mm

Position of minimum film thickness = 61.54 degree

Coefficient of friction, f = 0.0047

Torque required = 3.462 N.m

Power Lost = 652.76 W

Total Oil Supplied = 36863.1 cubic mm per sec

Side Leakage = 15888.0 cubic mm per sec

Maximum Film Pressure = 2.783 MPa

Position of Maximum Film Pressure = 16.63 degree

Terminating Position of Film = 94.02 degree

The dimensionless velocity, ws = 0.46

Stiffness and dammping coefficients are not found.

The Utility value = 3.42

2. L/D = 2.0

Optimization condition: The Maximum Load

Design Based on SAE 40 Oil

Length, L = 150 mm; Diameter, D = 75.00 mm

Operating Temperature, Top = 73.58 deg. C

Viscosity of the Lubricant, Mu = 20.52 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.139

Radial Clearance, C = 0.064 mm

Minimum film thickness, hmin = 0.038 mm

Eccentricity Ratio, e/C = 0.408

Eccentricity, e = 0.026 mm

Position of minimum film thickness = 62.52 degree

Coefficient of friction, f = 0.0049

Torque required = 3.149 N.m

Power Lost = 593.79 W

Total Oil Supplied = 37205.4 cubic mm per sec

Side Leakage = 12873.1 cubic mm per sec

Maximum Film Pressure = 2.557 MPa

Position of Maximum Film Pressure = 16.07 degree

Terminating Position of Film = 98.27 degree

The dimensionless velocity, ws = 0.48

Stiffness and dammping coefficients are not found.

The Utility value = 3.46

3. L/D = 1.0

Optimization condition: The Maximum Load

Design Based on SAE 50 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 76.47 deg. C

Viscosity of the Lubricant, Mu = 29.78 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.208

Clearance, C = 0.089 mm; Minimum film thickness, hmin = 0.048 mm

Eccentricity Ratio, e/C = 0.467; Eccentricity, e = 0.042 mm

Position of minimum film thickness = 59.00 degree

Coefficient of friction, f = 0.0083

Torque required = 7.444 N.m; Power Lost = 1403.72 W

Total Oil Supplied = 61751.3 cubic mm per sec

Side Leakage = 34580.7 cubic mm per sec

Maximum Film Pressure = 3.215 MPa

Position of Maximum Film Pressure = 17.40 degree

Terminating Position of Film = 86.00 degree

The dimensionless velocity, ws = 0.57

Stiffness coefficients: Kxx = 370.12 kN/mm; Kxy = 699.65 kN/mm

Kyx = 312.42 kN/mm; Kyy = 398.27 kN/mm

Damping coefficients: Cxx = 5.93 kN-s/mm; Cxy = 1.98 kN-s/mm

Cyx = 2.16 kN-s/mm; Cyy = 3.94 kN-s/mm

The Utility value = 3.41

4. L/D = 1.5

Optimization condition: The Maximum Load

Design Based on SAE 50 Oil

Length, L = 130 mm; Diameter, D = 86.67 mm

Operating Temperature, Top = 74.54 deg. C

Viscosity of the Lubricant, Mu = 32.09 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.161

Radial Clearance, C = 0.086 mm

Minimum film thickness, hmin = 0.049 mm

Eccentricity Ratio, e/C = 0.428

Eccentricity, e = 0.037 mm

Position of minimum film thickness = 61.54 degree

Coefficient of friction, f = 0.0068

Torque required = 5.041 N.m

Power Lost = 950.61 W

Total Oil Supplied = 53683.9 cubic mm per sec

Side Leakage = 23137.8 cubic mm per sec

Maximum Film Pressure = 2.783 MPa

Position of Maximum Film Pressure = 16.63 degree

Terminating Position of Film = 94.02 degree

The dimensionless velocity, ws = 0.56

Stiffness and dammping coefficients are not found.

The Utility value = 3.50

5. L/D = 2.0

Optimization condition: The Maximum Load

Design Based on SAE 50 Oil

Length, L = 150 mm; Diameter, D = 75.00 mm

Operating Temperature, Top = 73.58 deg. C

Viscosity of the Lubricant, Mu = 33.32 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.139

Radial Clearance, C = 0.082 mm

Minimum film thickness, hmin = 0.048 mm

Eccentricity Ratio, e/C = 0.408

Eccentricity, e = 0.033 mm

Position of minimum film thickness = 62.52 degree

Coefficient of friction, f = 0.0063

Torque required = 4.013 N.m

Power Lost = 756.65 W

Total Oil Supplied = 47409.6 cubic mm per sec

Side Leakage = 16403.7 cubic mm per sec

Maximum Film Pressure = 2.557 MPa

Position of Maximum Film Pressure = 16.07 degree

Terminating Position of Film = 98.27 degree

The dimensionless velocity, ws = 0.54

Stiffness and dammping coefficients are not found.

The Utility value = 3.58

6. L/D = 1.0

Optimization condition: The Maximum Load

Design Based on SAE 60 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 76.47 deg. C

Viscosity of the Lubricant, Mu = 39.30 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.208

Clearance, C = 0.103 mm; Minimum film thickness, hmin = 0.055 mm

Eccentricity Ratio, e/C = 0.467; Eccentricity, e = 0.048 mm

Position of minimum film thickness = 59.00 degree

Coefficient of friction, f = 0.0095

Torque required = 8.551 N.m; Power Lost = 1612.44 W

Total Oil Supplied = 70933.4 cubic mm per sec

Side Leakage = 39722.7 cubic mm per sec

Maximum Film Pressure = 3.215 MPa

Position of Maximum Film Pressure = 17.40 degree

Terminating Position of Film = 86.00 degree

The dimensionless velocity, ws = 0.61

Stiffness coefficients: Kxx = 322.21 kN/mm; Kxy = 609.08 kN/mm

Kyx = 271.98 kN/mm; Kyy = 346.72 kN/mm

Damping coefficients: Cxx = 5.16 kN-s/mm; Cxy = 1.72 kN-s/mm

Cyx = 1.88 kN-s/mm; Cyy = 3.43 kN-s/mm

The Utility value = 3.42

7. L/D = 1.5

Optimization condition: The Maximum Load

Design Based on SAE 60 Oil

Length, L = 130 mm; Diameter, D = 86.67 mm

Operating Temperature, Top = 74.54 deg. C

Viscosity of the Lubricant, Mu = 42.58 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.161

Radial Clearance, C = 0.099 mm

Minimum film thickness, hmin = 0.057 mm

Eccentricity Ratio, e/C = 0.428

Eccentricity, e = 0.042 mm

Position of minimum film thickness = 61.54 degree

Coefficient of friction, f = 0.0079

Torque required = 5.807 N.m

Power Lost = 1095.01 W

Total Oil Supplied = 61838.5 cubic mm per sec

Side Leakage = 26652.4 cubic mm per sec

Maximum Film Pressure = 2.783 MPa

Position of Maximum Film Pressure = 16.63 degree

Terminating Position of Film = 94.02 degree

The dimensionless velocity, ws = 0.60

Stiffness and dammping coefficients are not found.

The Utility value = 3.57

8. L/D = 2.0

Optimization condition: The Maximum Load

Design Based on SAE 60 Oil

Length, L = 150 mm; Diameter, D = 75.00 mm

Operating Temperature, Top = 73.58 deg. C

Viscosity of the Lubricant, Mu = 44.34 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.139

Radial Clearance, C = 0.094 mm

Minimum film thickness, hmin = 0.056 mm

Eccentricity Ratio, e/C = 0.408

Eccentricity, e = 0.039 mm

Position of minimum film thickness = 62.52 degree

Coefficient of friction, f = 0.0073

Torque required = 4.629 N.m

Power Lost = 872.84 W

Total Oil Supplied = 54689.8 cubic mm per sec

Side Leakage = 18922.7 cubic mm per sec

Maximum Film Pressure = 2.557 MPa

Position of Maximum Film Pressure = 16.07 degree

Terminating Position of Film = 98.27 degree

The dimensionless velocity, ws = 0.58

Stiffness and dammping coefficients are not found.

The utility value = 3.55

9. L/D = 1.0

Optimization condition: The Maximum Load

Design Based on SAE 70 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 76.47 deg. C

Viscosity of the Lubricant, Mu = 50.44 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.208

Clearance, C = 0.116 mm; Minimum film thickness, hmin = 0.062 mm

Eccentricity Ratio, e/C = 0.467; Eccentricity, e = 0.054 mm

Position of minimum film thickness = 59.00 degree

Coefficient of friction, f = 0.0108

Torque required = 9.688 N.m; Power Lost = 1826.79 W

Total Oil Supplied = 80362.7 cubic mm per sec

Side Leakage = 45003.1 cubic mm per sec

Maximum Film Pressure = 3.215 MPa

Position of Maximum Film Pressure = 17.40 degree

Terminating Position of Film = 86.00 degree

The dimensionless velocity, ws = 0.65

Stiffness coefficients: Kxx = 284.40 kN/mm; Kxy = 537.62 kN/mm

Kyx = 240.07 kN/mm; Kyy = 306.03 kN/mm

Damping coefficients: Cxx = 4.55 kN-s/mm; Cxy = 1.52 kN-s/mm

Cyx = 1.66 kN-s/mm; Cyy = 3.03 kN-s/mm

The Utility value = 3.48

4.9.2 Based on Minimum Friction

On the basis of minimum friction the 'Expert System' has found and filed seven design alternatives. The print-out of this file is attached on the succeeding pages. Out of these seven design alternatives, the solution with L/D = 1.5, optimization condition as minimum friction, and SAE 10 oil has the highest utility value equal to 4.39.

The load on the bearing, W = 17.00 kN

The speed of the journal, N = 30.000 rps

The bearing unit load, P = 1.511 MPa

The bearing-application for clearance is No. 2. Electric motor and generator practice

The inlet temperature, Tin = 66.50 deg C

The decision-criterion for utility is No. 2. Based on minimum friction

Stable Design Alternatives Are

1. L/D = 1.0

Optimization Condition: The Minimum Friction

Design Based on SAE 10 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 71.98 deg. C

Viscosity of the Lubricant, Mu = 8.63 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.078

Radial Clearance, C = 0.079 mm

Minimum film thickness, hmin = 0.024 mm

Eccentricity Ratio, e/C = 0.700

Eccentricity, e = 0.055 mm

Position of minimum film thickness = 43.70 degree

Coefficient of friction, f = 0.0036

Torque required = 3.205 N.m

Power Lost = 604.31 W

Total Oil Supplied = 59306.8 cubic mm per sec

Side Leakage = 45073.2 cubic mm per sec

Maximum Film Pressure = 4.140 MPa

Position of Maximum Film Pressure = 18.90 degree

Terminating Position of Film = 64.00 degree

The dimensionless velocity, ws = 0.53

Stiffness coefficients: Kxx = 759.27 kN/mm; Kxy = 785.13 kN/mm

Kyx = 25.15 kN/mm; Kyy = 428.04 kN/mm

Damping coefficients: Cxx = 6.96 kN-s/mm; Cxy = 2.09 kN-s/mm

Cyx = 1.93 kN-s/mm; Cyy = 1.88 kN-s/mm

The Utility value = 4.19

2. L/D = 1.5

Optimization Condition: The Minimum Friction

Design Based on SAE 10 Oil

Length, L = 130 mm; Diameter, D = 86.67 mm

Operating Temperature, Top = 72.06 deg. C

Viscosity of the Lubricant, Mu = 8.61 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.082

Radial Clearance, C = 0.063 mm

Minimum film thickness, hmin = 0.024 mm

Eccentricity Ratio, e/C = 0.614

Eccentricity, e = 0.038 mm

Position of minimum film thickness = 50.13 degree

Coefficient of friction, f = 0.0034

Torque required = 2.479 N.m

Power Lost = 467.50 W

Total Oil Supplied = 40938.9 cubic mm per sec

Side Leakage = 24154.0 cubic mm per sec

Maximum Film Pressure = 3.256 MPa

Position of Maximum Film Pressure = 19.22 degree

Terminating Position of Film = 76.76 degree

The dimensionless velocity, ws = 0.48

Stiffness and dammping coefficients are not found.

The Utility value = 4.39

3. L/D = 2.0

Optimization Condition: The Minimum Friction

Design Based on SAE 10 Oil

Length, L = 150 mm; Diameter, D = 75.00 mm

Operating Temperature, Top = 71.93 deg. C

Viscosity of the Lubricant, Mu = 8.64 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.081

Radial Clearance, C = 0.055 mm

Minimum film thickness, hmin = 0.024 mm

Eccentricity Ratio, e/C = 0.566

Eccentricity, e = 0.031 mm

Position of minimum film thickness = 53.27 degree

Coefficient of friction, f = 0.0032

Torque required = 2.061 N.m

Power Lost = 388.61 W

Total Oil Supplied = 32421.1 cubic mm per sec

Side Leakage = 15464.9 cubic mm per sec

Maximum Film Pressure = 2.873 MPa

Position of Maximum Film Pressure = 18.77 degree

Terminating Position of Film = 83.73 degree

The dimensionless velocity, ws = 0.44

Stiffness and dammping coefficients are not found.

The Utility value = 4.34

4. L/D = 2.0

Optimization Condition: The Minimum Friction

Design Based on SAE 20 Oil

Length, L = 150 mm; Diameter, D = 75.00 mm

Operating Temperature, Top = 71.93 deg. C

Viscosity of the Lubricant, Mu = 10.77 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.081

Radial Clearance, C = 0.061 mm

Minimum film thickness, hmin = 0.026 mm

Eccentricity Ratio, e/C = 0.566

Eccentricity, e = 0.034 mm

Position of minimum film thickness = 53.27 degree

Coefficient of friction, f = 0.0036

Torque required = 2.301 N.m

Power Lost = 433.87 W

Total Oil Supplied = 36196.6 cubic mm per sec

Side Leakage = 17265.8 cubic mm per sec

Maximum Film Pressure = 2.873 MPa

Position of Maximum Film Pressure = 18.77 degree

Terminating Position of Film = 83.73 degree

The dimensionless velocity, ws (used by Vance) = 0.47

Stiffness and dammping coefficients are not found.

The Utility value = 4.38

5. L/D = 1.5

Optimization condition: The Maximum Load

Design Based on SAE 30 Oil

Length, L = 130 mm; Diameter, D = 86.67 mm

Operating Temperature, Top = 74.54 deg. C

Viscosity of the Lubricant, Mu = 15.13 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.161

Radial Clearance, C = 0.059 mm

Minimum film thickness, hmin = 0.034 mm

Eccentricity Ratio, e/C = 0.428

Eccentricity, e = 0.025 mm

Position of minimum film thickness = 61.54 degree

Coefficient of friction, f = 0.0047

Torque required = 3.462 N.m

Power Lost = 652.76 W

Total Oil Supplied = 36863.1 cubic mm per sec

Side Leakage = 15888.0 cubic mm per sec

Maximum Film Pressure = 2.783 MPa

Position of Maximum Film Pressure = 16.63 degree

Terminating Position of Film = 94.02 degree

The dimensionless velocity, ws = 0.46

Stiffness and dammping coefficients are not found.

The Utility value = 4.26

6. L/D = 2.0

Optimization condition: The Maximum Load

Design Based on SAE 30 Oil

Length, L = 150 mm; Diameter, D = 75.00 mm

Operating Temperature, Top = 73.58 deg. C

Viscosity of the Lubricant, Mu = 15.62 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.139

Radial Clearance, C = 0.056 mm

Minimum film thickness, hmin = 0.033 mm

Eccentricity Ratio, e/C = 0.408

Eccentricity, e = 0.023 mm

Position of minimum film thickness = 62.52 degree

Coefficient of friction, f = 0.0043

Torque required = 2.748 N.m

Power Lost = 518.12 W

Total Oil Supplied = 32464.0 cubic mm per sec

Side Leakage = 11232.5 cubic mm per sec

Maximum Film Pressure = 2.557 MPa

Position of Maximum Film Pressure = 16.07 degree

Terminating Position of Film = 98.27 degree

The dimensionless velocity, ws = 0.45

Stiffness and dammping coefficients are not found.

The Utility value = 4.24

7. L/D = 2.0

Optimization condition: The Maximum Load

Design Based on SAE 40 Oil

Length, L = 150 mm; Diameter, D = 75.00 mm

Operating Temperature, Top = 73.58 deg. C

Viscosity of the Lubricant, Mu = 20.52 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.139

Radial Clearance, C = 0.064 mm

Minimum film thickness, hmin = 0.038 mm

Eccentricity Ratio, e/C = 0.408

Eccentricity, e = 0.026 mm

Position of minimum film thickness = 62.52 degree

Coefficient of friction, f = 0.0049

Torque required = 3.149 N.m

Power Lost = 593.79 W

Total Oil Supplied = 37205.4 cubic mm per sec

Side Leakage = 12873.1 cubic mm per sec

Maximum Film Pressure = 2.557 MPa

Position of Maximum Film Pressure = 16.07 degree

Terminating Position of Film = 98.27 degree

The dimensionless velocity, ws = 0.48

Stiffness and dammping coefficients are not found.

The Utility value = 4.24

In this example, all the design alternatives satisfy temperature-rise check, therefore at the end of stage 1 and 2, respectively, there are 12 and 84 solutions. For example for L/D = 0.5, maximum load, and SAE 10 oil, the lubricant outlet temperature T_{out} is found to be 97.5 °C, well within the prescribed limit. Out of these 84 solutions, 53 solutions are discarded because they do not fulfil the condition of industrial application for clearance. The above example has L = 75 mm, D = 150 mm, and C = 0.046 mm. For electric motor and generator practice the clearance limit for diameter 100 - 200 mm is 0.0762 to 0.1524 mm and obviously the clearance of 0.046 mm is not in the desired range. Therefore this solution is rejected at stage 3. At the end of stage 3 the number of solutions is 31.

All the thirty one solutions satisfy minimum film thickness and stability requirements. Therefore at the end of stage 4 and 5 there are 31 solutions. For example with L/D = 1.0, minimum friction condition, and SAE 10 oil, the solution is L = D = 106 mm, T_{out} = 77.5°C below 120°C, C = 0.079 mm (in the range of 0.0762 - 0.1524 mm), $h_{0_{F.S.=2}}$ =0.0141 mm which is greater than 0.005+0.00004D = 0.0092 mm (acceptable), and ε = 0.7 which is less than 0.83; but ω_s = 0.53 less than 2.33 (therefore stable).

The utility value U on the basis of maximum load for the above stable solution (L/D = 1.0, —) is 3.23. On the same basis U_{max} = 3.58 whose ninety five percent (95%) is 3.40. Therefore the solution with U = 3.23 is rejected in the decision making stage (stage 6.1) and could not be printed in the 'results.out' file. The number of solutions at the end of stage 6, which are filed is only 9; it is less than 31 at the end of stage 5.

The utility value U on the basis of minimum friction for the same solution (L/D = 1.0, —) is 4.19. On the same basis U_{max} = 4.39 whose ninety five percent (95%) is 4.17. Therefore the solution with U = 4.19 is accepted in the decision making stage (stage 6.1) and is printed in the 'results.out' file. The number of solutions at the end of stage 6, which are filed is only 7 in this case.

The present results are to be compared with the previous work in order to show the usefulness of this expert system 'ES'. For the values of W and N considered in the example, minimum recommended diameter obtained from Welsh's chart [63] at L/D = 0.6 is 110 mm. The 'ES' gives D = 136.67 mm at L/D = 0.6. Therefore it is in agreement with Welsh's recommendation.

Most of the researchers Shigley [17], Orthwein [19], and Juvinall [68] have used the charts supplied by Raimondi and Boyd [24] for the maximum load and the minimum friction conditions. Since the example [Sec. 4.9] with L/D = 0.5 and SAE 10 oil has been solved by Juvinall [68], the present solution is compared with his solution. Instead of giving T_{in} as input he has given T_{op} =82°C as input. To have the same operating temperature, inlet temperature is taken as 66.5 and 78.5 °C, respectively for the max. W and the min. f conditions when solving by the 'ES'. As mentioned earlier that the solution with L/D = 0.5 and SAE 10 oil is rejected by the 'ES' because of the violation of clearance constraint. If that constraint is removed, i.e. the clearance found is not checked then the 'ES' gives the solution.

For maximum load:

The solution obtained from the 'ES' is T_{op} = 81.96 °C, μ = 6.49 mPa.s, S = 0.345, C = 0.046 mm, h_0 = 0.02 mm, f = 0.0055, Q = 37,131.2 mm^3/s, and

side leakage Q_s = 26,660.2 mm^3/s. For the same condition, Juvinall [68] found μ = 6.3 mPa.s, S = 0.342, C = 0.0448 mm, h_0 = 0.019 mm, f = 0.0052, Q = 36,300 mm^3/s, and side leakage Q_s = 26,100 mm^3/s. Therefore it can be concluded that the present result is in agreement with that given by Juvinall.

For minimum friction:

The present system gives T_{op} = 82.1 °C, μ = 6.47 mPa.s, S = 0.036, C = 0.142 mm, h_0 = 0.016 mm, f = 0.0034, Q = 135,055.6 mm^3/s, and side leakage Q_s = 125,871.8 mm^3/s. For the same condition, Juvinall [68] found μ = 6.3 mPa.s, S = 0.036, C = 0.138 mm, h_0 = 0.0152 mm, f = 0.0032, Q = 131,600 mm^3/s, and side leakage Q_s = 122,400 mm^3/s. Therefore it can be concluded that the present result is in agreement with that given by Juvinall for the min. f condition also.

4.10 COMPARISON WITH MOES AND BOSMA CHART

Moes and Bosma [69] developed a design chart shown in Fig. 4.3 for the full journal bearing which enables the designer to select optimum bearing dimensions. Keith, Jr. [15-16] also suggests the use of the same chart. The chart is constructed in terms of two dimensionless groups H_{MB} and F_{MB}, which are

$$H_{MB} = (h_0/R)\{P/\omega\mu\}^{1/2} \text{ and } F_{MB} = (T/WR)\{P/\omega\mu\}^{1/2} \quad (4.4)$$

With the help of the above parameters, radial clearance is found as

$$C = \{H_{MB}R/(1-\varepsilon)\}(\omega\mu/P)^{1/2} \quad (4.5)$$

The chart contains two families of curves: curves of constant L/D ratio and

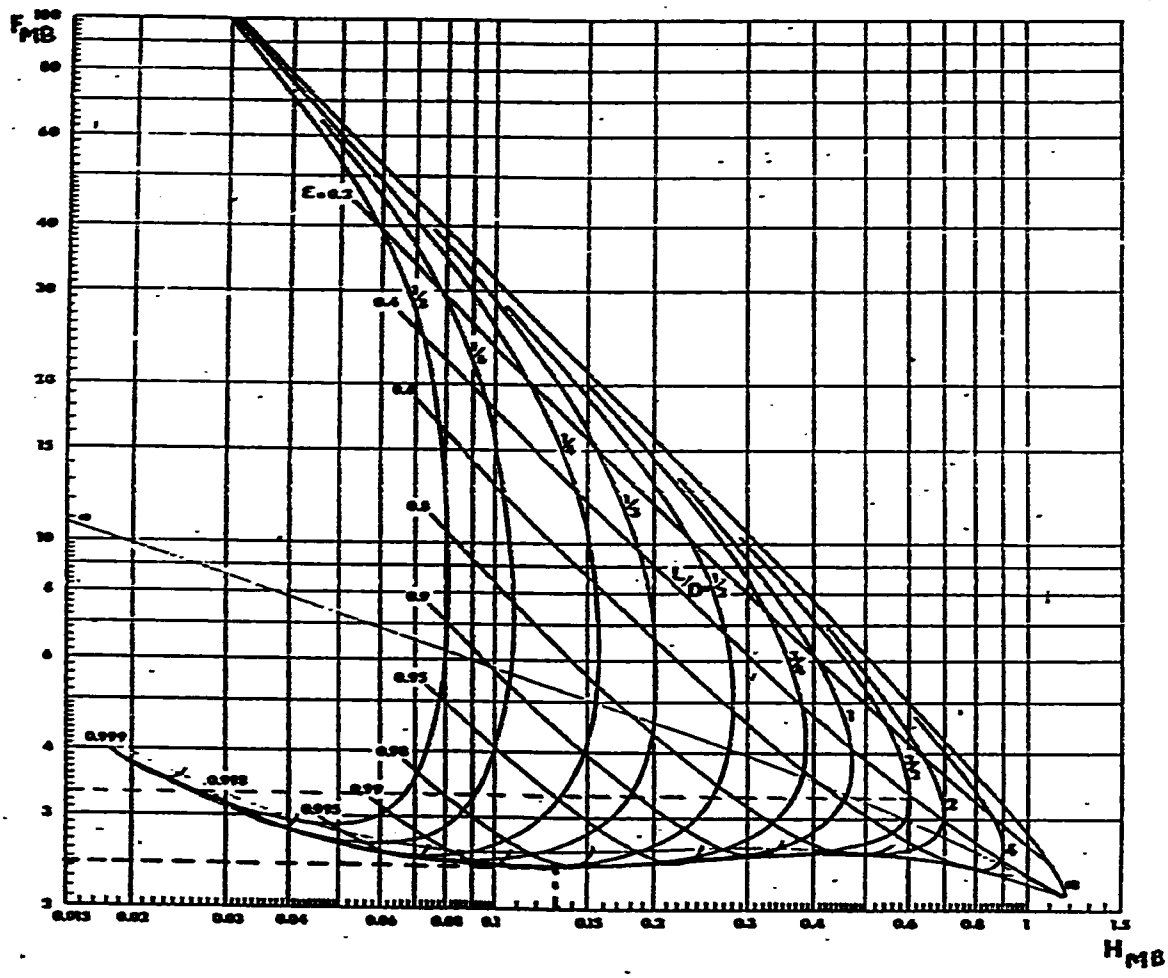


Fig. 4.3 Design Chart for Full Journal Bearings for Estimating the Optimum Bearing Clearance for a Given L/D Ratio [69].

curves of constant ε. Its usage is explained with an example below and the results from the chart are compared with that of 'ES'.

In the previous example [Sec. 4.9], consider a case with L/D = 1.5 and SAE 30 oil. Here W = 17 kN, N = 30 rps, P = 1.511 MPa, D = 86.67 mm, and R = 43.335 mm.

4.10.1 Maximum W (or Maximum h_0) Condition

The 'ES' gives the solution as μ = 15.13 mPa-s, C = 0.059 mm, T = 3.462 N.m, h_0 = 0.034 *mm*, and ε = 0.428.

For the largest h_0, the coordinates on Moes and Bosma chart (Fig. 4.3) corresponding to the maximum H_{MB} for L/D = 1.5 is located as H_{MB} = 0.6, F_{MB} = 3.3 and ε = 0.42. Then the solution is obtained as C = 0.0616 mm, T = 3.34 N.m, and h_0 = 0.0357 *mm*.

Thus it is obvious that the results of 'ES' are in good agreement with those obtained from the chart of Moes and Bosma for the above condition.

4.10.2 Minimum f (or Minimum T) Condition

The 'ES' gives the solution as μ = 16.46 mPa-s, C = 0.087 mm, T = 3.428 N.m, h_0 = 0.033 *mm*, and ε = 0.614.

For the minimum torque (minimum value of F_{MB} at L/D = 1.5, the coordinates on the chart (Fig. 4.3) are located as H_{MB} = 0.13, F_{MB} = 2.45 and ε = 0.981. Then the solution is obtained as C = 0.4036 mm, T = 2.64 N.m, and h_0 = 0.00807 *mm*.

In this case, the results of the 'ES' do not agree with those obtained from the chart of Moes and Bosma. In the latter case although the value of the torque is low, the clearance is very high, which is suitable only for the worst kind of industrial application (see Table 3.1), i.e., for rough machine practice only, where for diameter under 100 mm, the allowable range of C is 0.2794 to 0.4064 mm. Further, minimum film thickness requirement is not satisfied. According to Juvinall [68], $h_{0_{F.S.=2}} \geq 0.005+0.00004D(= 0.00846$ mm). But, even the actual h_0 (= 0.00807 mm), without any factor of safety, does not fulfil Juvinall's criterion [68]. Because of wear, clearance will increase and minimum film thickness will decrease further. In addition, very large clearance makes the bearing noisy [17]. Therefore the present results are practicable and those obtained from Moes and Bosma's chart are not. To support this conclusion one more example is considered.

Keith, Jr. [16] has given an example with L/D = 3/4, μ = 34.47 mPa-s, W = 8 kN, N = 30 rps, D = 100 mm, and L = 75 mm. For this example with minimum torque condition he found C = 0.7645 mm. For diameter under 100 mm, the maximum clearance limit even with the worst industrial application is 0.4064 mm (Table 3.1). Therefore this value of clearance does not comply with the general industrial practice. Hence the solution obtained using Moes and Bosma chart is not practicable.

CHAPTER 5

DESIGN OF FULL JOURNAL BEARINGS FOR OPTIMAL CLEARANCE

The sources of the objective knowledge and viscosity-temperature relationships used for designing journal bearings for optimal clearance condition are the same as those in chapter 4. As has been stated in chapter 4, the values of Sommerfeld number S were known for the maximum load and minimum friction conditions for different L/D ratios. However, in the present case, values of S are unknown and they are to be determined after using an iterative method for finding the temperature-rise, which will be discussed later in this chapter. Before applying the iterative method to find the temperature-rise, value of clearance has to be determined. Keith, Jr. [15-16] has given a plot of the recommended minimum clearance for a given shaft speed and diameter. The plot is shown in Fig. 5.1. The values are in inches which have been converted to mm and presented in Table 5.1. The clearance is selected from this table. Once S is known, all necessary performance variables can be found.

5.1 PERFORMANCE VARIABLES

Al-Dukhaiyel [66] obtained equations for performance variables such as T_{var}, h_0/C, coefficient of friction variable (R/C)f, flow variable Q/RCNL, flow ratio Q_s/Q, and pressure ratio P/p_{max} of full journal bearings as functions of S by curve fitting tabulated data obtained from the charts presented in Shigley [17] for L/D ratios of 0.25, 0.5, 1.0, and ∞. The accuracy of these equations is

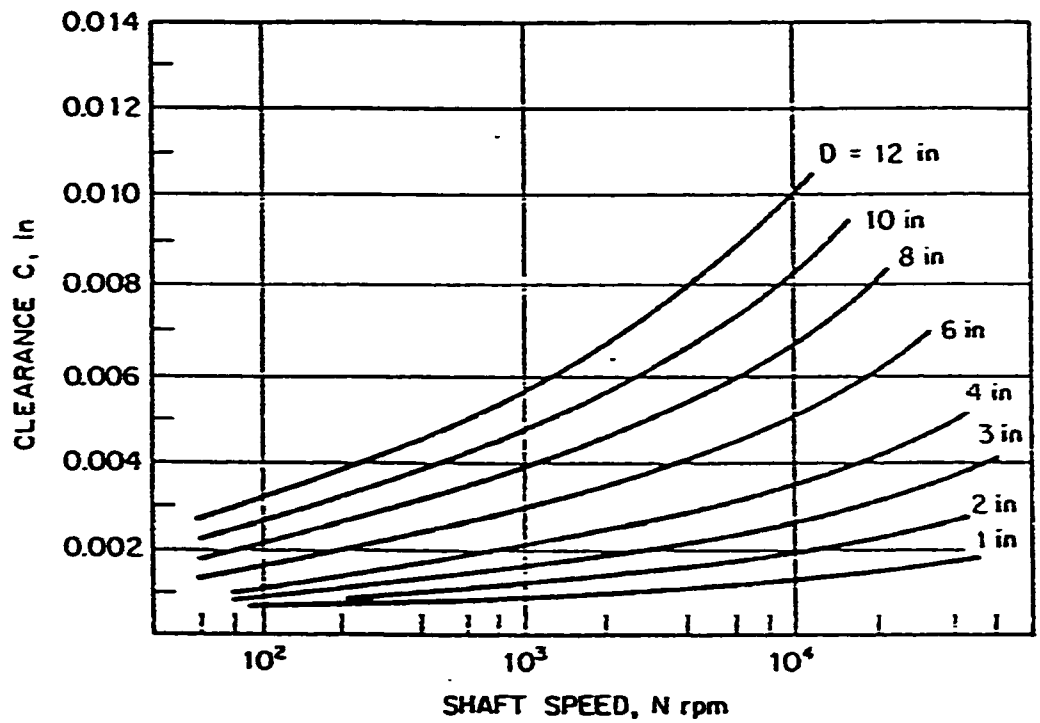


Fig. 5.1 Recommended Minimum Clearance Versus Journal Speed for a Given Journal Diameter [15-16].

Table 5.1. Recommended Minimum Clearance for Given Shaft Speed and Diameter [15-16].

Speed	Clearance (mm) for shaft diameter (mm) under							
(rpm)	25	50	75	100	150	200	250	300
60	0.0152	0.0165	0.0178	0.0229	0.0356	0.0457	0.057	0.07
80	0.016	0.018	0.02	0.0254	0.038	0.0508	0.0635	0.0762
100	0.0165	0.0193	0.0229	0.028	0.041	0.0546	0.0686	0.0813
200	0.0178	0.02	0.028	0.0356	0.0508	0.0673	0.0828	0.0991
400	0.0191	0.0254	0.0343	0.0447	0.061	0.0813	0.0991	0.1168
600	0.0198	0.028	0.0371	0.0488	0.0673	0.0889	0.108	0.127
800	0.02	0.0297	0.0406	0.0508	0.0711	0.0965	0.1168	0.1346
1,000	0.0218	0.03	0.0417	0.0538	0.0762	0.1	0.1219	0.1448
2,000	0.024	0.0356	0.0483	0.0625	0.0889	0.1168	0.1422	0.17
4,000	0.028	0.0406	0.0546	0.0726	0.1041	0.137	0.1676	0.2032
6,000	0.0299	0.0437	0.0597	0.0813	0.1143	0.1524	0.1854	0.2261
8,000	0.03	0.0475	0.0635	0.086	0.1239	0.1626	0.1980	0.2426
10,000	0.033	0.0508	0.0668	0.0914	0.1295	0.17	0.2083	0.254
20,000	0.038	0.0579	0.0813	0.1067	0.1562	0.2083	0.2591	0.3099
40,000	0.0457	0.0686	0.0965	0.127	0.1905	0.2591	0.3124	0.3632

within ±5 %. The error increases if S is less than 0.002, which occurs rarely. Therefore for L/D = 0.25, 0.5, and 1.0, the same equations are used in the present expert system. For other L/D ratios (0.6, 1.5, and 2.0) first the corresponding data are obtained with the help of Eq. (4.1) given by Raimondi and Boyd [23]. Then the best fit equations are obtained using 'SAS' package available at the K.F.U.P.M. Data Processing Center. Each of these equations is written with its coefficient of determination r^2 value that is a measure of the percentage of data explained with this equation [79]. In addition to the six variables considered by Al-Dukhaiyel [66], the attitude angle (the angular location of the minimum film thickness) Φ is also taken into account.

For all the six L/D ratios, the values of the performance variables have been presented in Tables 5.2 - 5.8 and the corresponding cubical equations have been summarized below. To get good correlation, the values of S have been divided into three ranges from 0.002 to 0.1, 0.1 to 1, and 1 to 10 and each range contains 5 to 7 data points. Therefore, the values of r^2 obtained are close to 1.0.

5.1.1 Temperature-rise Variable

The dimensionless temperature-rise variable is

$$T_{rw} = (\rho C_{ll} \Delta T)/P \tag{5.1}$$

where ρ is the density, C_{ll} the specific heat, and ΔT the temperature-rise from inlet to outlet of the lubricant. P is the bearing unit load. Average values of ρ and C_{ll} are 861 kg/m^3 and 1760 $J/kg-^\circ C$ [17].

Table 5.2. Temperature-Rise Variable for a Full Journal Bearing for Given S and L/D Ratio [17,24].

	L/D ratio						
S	0.25	0.5	0.6*	1.0	1.5*	2.0*	∞
0.01	3.5	3.5	3.5	3.5	8.22	11.867	29.0
0.02	5.0	5.0	5.0	5.0	7.04	8.609	16.0
0.04	7.3	7.3	7.3	7.3	7.62	7.858	9.0
0.06	9.0	9.0	9.0	9.0	8.91	8.836	8.5
0.08	10.5	10.5	10.5	10.5	10.19	9.942	8.8
0.1	13.5	13.5	13.5	13.5	12.7	12.089	9.2
0.2	20.	20.0	20.0	20.0	19.26	18.687	16.0
0.4	35.0	35.0	35.0	35.0	35.0	35.0	35.0
0.6	50.0	50.0	50.0	50.0	50.0	50.0	50.0
0.8	66.0	66.0	66.0	66.0	66.0	66.0	66.0
1.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0
2.0	155.0	155.0	155.0	155.0	155.0	155.0	155.0
4.0	330.0	330.0	330.0	330.0	330.0	330.0	330.0
6.0	490.0	490.0	490.0	490.0	490.0	490.0	490.0
8.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0

*Corresponding values are obtained using Eq. (4.1) given by [23].

Table 5.3. Minimum Film Thickness Variable for a Full Journal Bearing for Given S and L/D Ratio [17,24].

		L/D ratio						
S	$\log_{10}(S)$	0.25	0.5	0.6*	1.0	1.5*	2.0*	∞
0.002	-2.699	0.02	0.02	0.02	0.02	0.022	0.023	0.03
0.01	-2.0	0.03	0.04	0.042	0.05	0.057	0.062	0.08
0.02	-1.699	0.05	0.08	0.083	0.105	0.127	0.141	0.2
0.04	-1.398	0.08	0.12	0.128	0.18	0.231	0.264	0.4
0.06	-1.222	0.095	0.16	0.170	0.24	0.312	0.36	0.56
0.08	-1.097	0.105	0.185	0.204	0.3	0.387	0.444	0.67
0.1	-1.0	0.12	0.21	0.233	0.34	0.435	0.496	0.74
0.14	-0.854	0.14	0.26	0.299	0.43	0.53	0.591	0.82
0.2	-0.699	0.17	0.32	0.364	0.5	0.6	0.659	0.88
0.4	-0.398	0.25	0.46	0.532	0.69	0.773	0.815	0.94
0.6	-0.222	0.31	0.54	0.62	0.78	0.852	0.884	0.96
0.8	-0.097	0.36	0.6	0.674	0.82	0.884	0.912	0.976
1.0	0.0	0.4	0.65	0.723	0.86	0.915	0.938	0.98
1.4	0.146	0.46	0.72	0.784	0.9	0.945	0.962	0.99
2.0	0.301	0.52	0.79	0.841	0.93	0.963	0.976	0.996
4.0	0.602	0.66	0.89	0.921	0.97	0.987	0.992	1.0
6.0	0.778	0.74	0.93	0.95	0.98	0.99	0.994	1.0
8.0	0.903	0.8	0.95	0.966	0.99	0.997	0.999	1.0
10.0	1.0	0.845	0.96	0.976	1.0	1.0	1.0	1.0

*Corresponding values are obtained using Eq. (4.1) given by [23].

Table 5.4. Coefficient of Friction Variable for a Full Journal Bearing for Given S and L/D Ratio [17,24].

	L/D ratio						
S	0.25	0.5	0.6*	1.0	1.5*	2.0*	∞
0.002	0.0	0.0	0.0	0.0	0.0	0.0	0.0
0.01	0.9	0.8	0.801	0.78	0.748	0.725	0.620
0.02	1.4	1.25	1.206	1.1	1.039	1.006	0.9
0.04	2.2	1.85	1.779	1.6	1.487	1.423	1.2
0.06	2.9	2.4	2.281	2.0	1.835	1.746	1.45
0.08	3.7	3.0	2.814	2.4	2.175	2.059	1.7
0.1	4.3	3.5	3.282	2.8	2.541	2.408	2.0
0.14	5.5	4.5	4.169	3.5	3.191	3.05	2.7
0.2	6.5	5.7	5.335	4.6	4.272	4.127	3.8
0.4	12.0	9.7	9.181	8.25	7.889	7.745	7.5
0.6	16.0	13.5	12.936	12.0	11.698	11.603	11.6
0.8	20.0	17.0	16.586	16.0	15.864	15.844	16.0
1.0	24.0	21.0	20.586	20.0	20.0	20.0	20.0
1.4	32.0	28.0	28.0	28.0	28.0	28.0	28.0
2.0	45.0	40.0	40.0	40.0	40.0	40.0	40.0
4.0	85.0	80.0	80.0	80.0	80.0	80.0	80.0
6.0	120.0	120.0	120.0	120.0	120.0	120.0	120.0
8.0	160.0	160.0	160.0	160.0	160.0	160.0	160.000
10.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0

*Corresponding values are obtained using Eq. (4.1) given by [23].

Table 5.5. Attitude Angle for a Full Journal Bearing for Given S and L/D Ratio [17.24].

	L/D ratio						
S	0.25	0.5	0.6*	1.0	1.5*	2.0*	∞
0.002	6.0	8.0	7.901	8.0	8.346	8.625	10.0
0.01	12.0	16.0	17.747	22.0	24.47	25.781	30.0
0.02	14.0	21.0	22.804	28.0	31.623	33.742	41.5
0.04	18.0	26.0	27.765	34.0	38.988	42.062	54.0
0.06	20.0	29.0	31.818	40.0	45.68	48.984	61.0
0.08	22.0	32.0	35.265	44.0	49.654	52.844	64.0
0.1	24.0	34.0	38.13	48.0	53.63	56.594	66.0
0.14	26.0	37.0	42.506	54.0	59.395	61.875	68.0
0.2	29.0	42.0	47.384	58.0	62.586	64.558	68.7
0.4	35.0	51.0	57.374	68.0	71.074	71.794	69.8
0.6	40.0	57.0	63.333	73.0	75.0	75.0	70.0
0.8	43.0	62.0	67.824	76.0	77.074	76.578	70.0
1.0	46.0	64.0	69.938	78.0	78.716	77.937	70.0
1.4	50.0	70.0	74.63	80.0	79.63	78.437	70.0
2.0	56.0	74.0	77.994	82.0	80.938	79.406	70.0
4.0	66.0	81.0	83.241	84.0	81.741	79.781	70.
6.0	72.0	84.0	85.06	84.0	81.352	79.383	70.5
8.0	75.0	85.0	85.75	84.25	81.506	79.528	70.8
10.0	77.0	86.0	86.463	84.5	81.63	79.625	71.0

*Corresponding values are obtained using Eq. (4.1) given by [23].

Table 5.6. Flow Variable for a Full Journal Bearing for Given S and L/D Ratio [17,24].

	L/D ratio						
S	0.25	0.5	0.6*	1.0	1.5*	2.0*	∞
0.002	6.15	5.9	5.808	4.82	3.755	3.043	0.06
0.01	6.13	5.83	5.726	4.78	3.784	3.121	0.36
0.02	6.1	5.75	5.624	4.72	3.809	3.21	0.74
0.04	6.08	5.62	5.441	4.62	3.894	3.433	1.6
0.06	6.03	5.54	5.321	4.55	3.95	3.585	2.2
0.08	6.0	5.47	5.222	4.47	3.939	3.629	2.5
0.1	5.92	5.4	5.364	4.93	4.442	4.111	2.7
0.14	5.8	5.25	4.967	4.26	3.842	3.618	2.89
0.2	5.7	5.1	4.795	4.1	3.731	3.545	3.0
0.4	5.4	4.7	4.395	3.78	3.504	3.381	3.1
0.6	5.25	4.43	4.144	3.6	3.378	3.287	3.12
0.8	5.2	4.27	3.996	3.5	3.312	3.24	3.14
1.0	5.0	4.21	3.925	3.42	3.243	3.183	3.15
1.400	4.8	3.9	3.697	3.35	3.229	3.187	3.15
2.0	4.6	3.7	3.553	3.31	3.226	3.196	3.16
4.0	4.2	3.45	3.379	3.27	3.23	3.212	3.17
6.0	3.9	3.37	3.323	3.25	3.221	3.208	3.17
8.0	3.7	3.31	3.275	3.22	3.198	3.188	3.16
10.0	3.65	3.3	3.26	3.2	3.18	3.172	3.16

*Corresponding values are obtained using Eq. (4.1) given by [23].

Table 5.7. Flow Ratio for a Full Journal Bearing for Given S and L/D Ratio [17,24].

	L/D ratio						
S	0.25	0.5	0.6*	1.0	1.5*	2.0*	∞
0.002	1.0	1.0	1.0	0.99	0.805	0.663	0.0
0.01	0.99	0.98	0.98	0.95	0.769	0.632	0.0
0.02	0.98	0.96	0.959	0.915	0.738	0.606	0.0
0.04	0.97	0.93	0.928	0.86	0.689	0.564	0.0
0.06	0.96	0.905	0.902	0.8	0.634	0.517	0.0
0.08	0.945	0.88	0.876	0.76	0.599	0.487	0.0
0.1	0.94	0.87	0.86	0.72	0.562	0.454	0.0
0.14	0.92	0.84	0.811	0.65	0.498	0.399	0.0
0.2	0.9	0.80	0.747	0.56	0.416	0.329	0.0
0.4	0.86	0.7	0.613	0.4	0.276	0.21	0.0
0.6	0.82	0.61	0.508	0.29	0.184	0.133	0.0
0.8	0.79	0.54	0.437	0.23	0.137	0.095	0.0
1.0	0.76	0.5	0.399	0.2	0.114	0.077	0.0
1.4	0.7	0.42	0.322	0.14	0.069	0.042	0.0
2.0	0.64	0.34	0.254	0.1	0.044	0.023	0.0
4.0	0.48	0.18	0.131	0.050	0.022	0.012	0.0
6.0	0.38	0.12	0.087	0.035	0.017	0.01	0.0
8.0	0.32	0.08	0.056	0.022	0.011	0.007	0.0
10.0	0.275	0.06	0.043	0.02	0.012	0.009	0.0

*Corresponding values are obtained using Eq. (4.1) given by [23].

Table 5.8. Pressure Ratio for a Full Journal Bearing for Given S and L/D Ratio [17,24].

	L/D ratio						
S	0.25	0.5	0.6*	1.0	1.5*	2.0*	∞
0.002	0.04	0.07	0.073	0.08	0.083	0.085	0.09
0.01	0.1	0.14	0.154	0.2	0.234	0.254	0.33
0.02	0.125	0.18	0.197	0.265	0.321	0.355	0.49
0.04	0.15	0.22	0.23	0.31	0.393	0.449	0.68
0.06	0.17	0.24	0.256	0.350	0.442	0.502	0.750
0.08	0.180	0.26	0.281	0.385	0.481	0.542	0.790
0.1	0.19	0.28	0.297	0.395	0.491	0.553	0.81
0.14	0.205	0.3	0.32	0.42	0.514	0.574	0.82
0.2	0.22	0.33	0.355	0.46	0.55	0.606	0.83
0.4	0.26	0.385	0.41	0.505	0.585	0.636	0.835
0.6	0.29	0.42	0.44	0.525	0.599	0.646	0.836
0.8	0.31	0.44	0.457	0.535	0.606	0.652	0.838
1.0	0.32	0.46	0.472	0.54	0.608	0.652	0.84
1.4	0.35	0.48	0.485	0.542	0.606	0.65	0.84
2.0	0.38	0.505	0.502	0.544	0.604	0.647	0.84
4.0	0.44	0.52	0.512	0.547	0.605	0.648	0.842
6.0	0.475	0.523	0.514	0.548	0.607	0.650	0.846
8.0	0.49	0.527	0.516	0.549	0.608	0.651	0.848
10.0	0.5	0.53	0.518	0.55	0.609	0.652	0.85

*Corresponding values are obtained using Eq. (4.1) given by [23].

5.1.1.1 Sommerfeld number S from 0.002 - 0.1:

$$(T_{var})_{LJD=0.25-1} = 61 + 80 \times \log_{10}(S) + 39 \times [\log_{10}(S)]^2 + 6.64 \times [\log_{10}(S)]^3$$

$$(T_{var})_{LJD=1.5} = 72.31 + 107.265 \times \log_{10}(S) + 57.28 \times [\log_{10}(S)]^2 + 9.84 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.987$.

$$(T_{var})_{LJD=2} = 65.91 + 94.75 \times \log_{10}(S) + 47.69 \times [\log_{10}(S)]^2 + 6.91 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.991$.

5.1.1.2 Sommerfeld number S from 0.1 - 1:

$$(T_{var})_{LJD=0.25-1} = 80 + 164 \times \log_{10}(S) + 144 \times [\log_{10}(S)]^2 + 47 \times [\log_{10}(S)]^3$$

$$(T_{var})_{LJD=1.5} = 80.16 + 164.47 \times \log_{10}(S) + 149.02 \times [\log_{10}(S)]^2 + 54.84 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

$$(T_{var})_{LJD=2} = 80.16 + 164.84 \times \log_{10}(S) + 151.94 \times [\log_{10}(S)]^2 + 59.95 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

5.1.1.3 Sommerfeld number S from 1 - 10:

$$(T_{var})_{LJD=0.25-2} = 286.94 - 1182.78 \times \log_{10}(S) + 2862.96 \times [\log_{10}(S)]^2 - 1294.88 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 1.0$.

5.1.2 Minimum Film Thickness Variable

5.1.2.1 Sommerfeld number S from 0.002 - 0.1:

$$(h_0/C)_{L/D=0.25} = 0.191 + 0.024 \times \log_{10}(S) - 0.07 \times [\log_{10}(S)]^2 - 0.021 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=0.5} = 0.576 + 0.491 \times \log_{10}(S) + 0.138 \times [\log_{10}(S)]^2 + 0.013 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=0.6} = 0.6925 + 0.6305 \times \log_{10}(S) + 0.1875 \times [\log_{10}(S)]^2 + 0.0171 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

$$(h_0/C)_{L/D=1} = 1.152 + 1.18 \times \log_{10}(S) + 0.421 \times [\log_{10}(S)]^2 + 0.053 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=1.5} = 1.332 + 1.1966 \times \log_{10}(S) + 0.3253 \times [\log_{10}(S)]^2 + 0.0229 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

$$(h_0/C)_{L/D=2} = 1.477 + 1.2735 \times \log_{10}(S) + 0.3123 \times [\log_{10}(S)]^2 + 0.0148 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

5.1.2.2 Sommerfeld number S from 0.1 - 1:

$$(h_0/C)_{L/D=0.25} = 0.4 + 0.44 \times \log_{10}(S) + 0.15 \times [\log_{10}(S)]^2 - 0.008 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=0.5} = 0.648 + 0.476 \times \log_{10}(S) - 0.047 \times [\log_{10}(S)]^2 - 0.084 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=0.6} = 0.7198 + 0.3815 \times \log_{10}(S) - 0.3637 \times [\log_{10}(S)]^2 - 0.2708 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9994$.

$$(h_0/C)_{L/D=1} = 0.855 + 0.218 \times \log_{10}(S) - 0.671 \times [\log_{10}(S)]^2 - 0.373 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=1.5} = 0.9107 + 0.1232 \times \log_{10}(S) - 0.7705 \times [\log_{10}(S)]^2 - 0.4569 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

$$(h_0/C)_{L/D=2} = 0.9345 + 0.1033 \times \log_{10}(S) - 0.6644 \times [\log_{10}(S)]^2 - 0.3658 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

5.1.2.3 Sommerfeld number S from 1 - 10:

$$(h_0/C)_{L/D=0.25} = 0.4 + 0.36 \times \log_{10}(S) + 0.16 \times [\log_{10}(S)]^2 - 0.075 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=0.5} = 0.65 + 0.529 \times \log_{10}(S) - 0.213 \times [\log_{10}(S)]^2 - 0.006 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=0.6} = 0.7171 + 0.5015 \times \log_{10}(S) - 0.3167 \times [\log_{10}(S)]^2 + 0.0739 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99997$.

$$(h_0/C)_{L/D=1} = 0.86 + 0.319 \times \log_{10}(S) - 0.311 \times [\log_{10}(S)]^2 + 0.131 \times [\log_{10}(S)]^3$$

$$(h_0/C)_{L/D=1.5} = 0.9188 + 0.2055 \times \log_{10}(S) - 0.2096 \times [\log_{10}(S)]^2 + 0.0852 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.996$.

$$(h_0/C)_{L/D=2} = 0.942 + 0.158 \times \log_{10}(S) - 0.169 \times [\log_{10}(S)]^2 + 0.069 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9965$.

5.1.3 Coefficient of Friction Variable

5.1.3.1 Sommerfeld number S from 0.002 - 0.1:

$$[(R/C)f]_{L/D=0.25} = 20.44 + 26.97 \times \log_{10}(S) + 13.1 \times [\log_{10}(S)]^2 + 2.25 \times [\log_{10}(S)]^3$$

$$[(R/C)f]_{L/D=0.5} = 13.92 + 16.45 \times \log_{10}(S) + 7.08 \times [\log_{10}(S)]^2 + 1.063 \times [\log_{10}(S)]^3$$

$$[(R/C)f]_{L/D=0.6} = 13.135 + 15.776 \times \log_{10}(S) + 7 \times [\log_{10}(S)]^2 + 1.099 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

$$[(R/C)f]_{L/D=1} = 11.14 + 13.5 \times \log_{10}(S) + 6.2 \times [\log_{10}(S)]^2 + 1.02 \times [\log_{10}(S)]^3$$

$$[(R/C)f]_{L/D=1.5} = 9.488 + 11.2 \times \log_{10}(S) + 5.05 \times [\log_{10}(S)]^2 + 0.816 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

$$[(R/C)f]_{L/D=2} = 8.885 + 10.46 \times \log_{10}(S) + 4.73 \times [\log_{10}(S)]^2 + 0.7688 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

5.1.3.2 Sommerfeld number S from 0.1 - 1:

$$[(R/C)f]_{L/D=0.25} = 23.8 + 40.1 \times \log_{10}(S) + 26.2 \times [\log_{10}(S)]^2 + 5.64 \times [\log_{10}(S)]^3$$

$$[(R/C)f]_{L/D=0.5} = 20.9 + 41.2 \times \log_{10}(S) + 37.7 \times [\log_{10}(S)]^2 + 14 \times [\log_{10}(S)]^3$$

$$[(R/C)f]_{L/D=0.6} = 20.5 + 42.84 \times \log_{10}(S) + 42.5 \times [\log_{10}(S)]^2 + 17.3 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

$$[(R/C)f]_{L/D=1} = 20 + 44.6 \times \log_{10}(S) + 44.1 \times [\log_{10}(S)]^2 + 16.7 \times [\log_{10}(S)]^3$$

$$[(R/C)f]_{L/D=1.5} = 19.986 + 47.16 \times \log_{10}(S) + 49.96 \times [\log_{10}(S)]^2 + 20.84 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99995$.

$$[(R/C)f]_{L/D=2} = 19.996 + 47.72 \times \log_{10}(S) + 50.48 \times [\log_{10}(S)]^2$$

$$+ 20.91 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99996$.

5.1.3.3 Sommerfeld number S from 1 - 10:

$$[(R/C)f]_{L/D=0.25} = 23.8 + 78.3 \times \log_{10}(S) - 65.2 \times [\log_{10}(S)]^2 + 163 \times [\log_{10}(S)]^3$$

$$[(R/C)f]_{L/D=0.5-2} = 16.5 + 83.2 \times \log_{10}(S) - 61.62 \times [\log_{10}(S)]^2 + 161.65 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99997$.

5.1.4 Attitude Angle

5.1.4.1 Sommerfeld number S from 0.002 - 0.1:

$$(\Phi)_{L/D=0.25} = 57.23 + 50.11 \times \log_{10}(S) + 19.93 \times [\log_{10}(S)]^2 + 3.11 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$.

$$(\Phi)_{L/D=0.5} = 59.77 + 30.2 \times \log_{10}(S) + 4.538 \times [\log_{10}(S)]^2 + 0.17 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

$$(\Phi)_{L/D=0.6} = 87.69 + 72.84 \times \log_{10}(S) + 27.44 \times [\log_{10}(S)]^2 + 4.227 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

$$(\Phi)_{L/D=1} = 122.68 + 114.84 \times \log_{10}(S) + 47.99 \times [\log_{10}(S)]^2$$

$$+ 7.85 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

$$(\Phi)_{L/D = 1.5} = 119.88 + 95.85 \times \log_{10}(S) + 35.2 \times [\log_{10}(S)]^2$$

$$+ 5.56 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

$$(\Phi)_{L/D = 2} = 111.26 + 72.36 \times \log_{10}(S) + 20.77 \times [\log_{10}(S)]^2$$

$$+ 2.98 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99966$.

5.1.4.2 Sommerfeld number S from 0.1 - 1:

$$(\Phi)_{L/D = 0.25} = 45.99 + 30.38 \times \log_{10}(S) + 9.04 \times [\log_{10}(S)]^2$$

$$+ 0.939 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9993$.

$$(\Phi)_{L/D = 0.5} = 64.37 + 31.24 \times \log_{10}(S) - 6.42 \times [\log_{10}(S)]^2$$

$$- 6.64 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9984$.

$$(\Phi)_{L/D = 0.6} = 70.14 + 26.37 \times \log_{10}(S) - 19.25 \times [\log_{10}(S)]^2$$

$$- 14.44 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

$$(\Phi)_{L/D = 1} = 77.9 + 15.08 \times \log_{10}(S) - 35.32 \times [\log_{10}(S)]^2$$

$$- 23.55 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

$$(\Phi)_{L/D=1.5} = 78.55 + 9.26 \times \log_{10}(S) - 34.47 \times [\log_{10}(S)]^2 - 22.15 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9992$.

$$(\Phi)_{L/D=2} = 77.76 + 6.435 \times \log_{10}(S) - 30.91 \times [\log_{10}(S)]^2 - 19.36 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9989$.

5.1.4.3 Sommerfeld number S from 1 - 10:

$$(\Phi)_{L/D=0.25} = 44.82 + 35.28 \times \log_{10}(S) + 6.55 \times [\log_{10}(S)]^2 - 9.58 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99956$.

$$(\Phi)_{L/D=0.5} = 65.95 + 27.62 \times \log_{10}(S) + 0.378 \times [\log_{10}(S)]^2 - 8.06 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

$$(\Phi)_{L/D=0.6} = 70.69 + 28.64 \times \log_{10}(S) - 13.67 \times [\log_{10}(S)]^2 + 0.706 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9994$.

$$(\Phi)_{L/D=1} = 76.83 + 25.73 \times \log_{10}(S) - 31.35 \times [\log_{10}(S)]^2 + 13.26 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.997$.

$$(\Phi)_{L/D=1.5} = 77.24 + 20.32 \times \log_{10}(S) - 30.28 \times [\log_{10}(S)]^2 + 14.36 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9887$.

$$(\Phi)_{L/D=2} = 76.56 + 16.3 \times \log_{10}(S) - 26.24 \times [\log_{10}(S)]^2 + 13.02 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9799$.

5.1.5 Total Oil Flow Variable

5.1.5.1 Sommerfeld number S from 0.002 - 0.1:

$$(QjRCNL)_{L/D=0.25} = 4.214 - 3.052 \times \log_{10}(S) - 1.616 \times [\log_{10}(S)]^2 - 0.2784 \times [\log_{10}(S)]^3$$

$$(QjRCNL)_{L/D=0.5} = 4.474 - 1.225 \times \log_{10}(S) - 0.317 \times [\log_{10}(S)]^2 - 0.0218 \times [\log_{10}(S)]^3$$

$$(QjRCNL)_{L/D=0.6} = 6.5 + 2.82 \times \log_{10}(S) + 2.01 \times [\log_{10}(S)]^2 + 0.393 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9477$.

$$(QjRCNL)_{L/D=1} = 3.046 - 2.057 \times \log_{10}(S) - 0.818 \times [\log_{10}(S)]^2 - 0.1108 \times [\log_{10}(S)]^3$$

$$(QjRCNL)_{L/D=1.5} = 9.4 + 8.96 \times \log_{10}(S) + 4.64 \times [\log_{10}(S)]^2 + 0.777 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.8025$.

$$(QjRCNL)_{L/D=2} = 9 + 8.3 \times \log_{10}(S) + 3.92 \times [\log_{10}(S)]^2 + 0.617 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.945$.

5.1.5.2 Sommerfeld number S from 0.1 - 1:

$$(QjRCNL)_{I/D-0.25} = 5.033 - 1.125 \times \log_{10}(S) - 0.395 \times [\log_{10}(S)]^2 - 0.1616 \times [\log_{10}(S)]^3$$

$$(QjRCNL)_{I/D-0.5} = 4.209 - 0.4166 \times \log_{10}(S) + 2.95 \times [\log_{10}(S)]^2 + 2.176 \times [\log_{10}(S)]^3$$

$$(QjRCNL)_{I/D-0.6} = 3.92 - 0.719 \times \log_{10}(S) + 1.616 \times [\log_{10}(S)]^2 + 1.2 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99976$.

$$(QjRCNL)_{I/D-1} = 3.422 - 0.6877 \times \log_{10}(S) + 0.689 \times [\log_{10}(S)]^2 + 0.4087 \times [\log_{10}(S)]^3$$

$$(QjRCNL)_{I/D-1.5} = 3.247 - 0.5506 \times \log_{10}(S) + 0.3122 \times [\log_{10}(S)]^2 + 0.1634 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99956$.

$$(QjRCNL)_{I/D-2} = 3.187 - 0.4298 \times \log_{10}(S) + 0.2127 \times [\log_{10}(S)]^2 + 0.1447 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

5.1.5.3 Sommerfeld number S from 1 - 10:

$$(QjRCNL)_{I/D-0.25} = 5 - 1.125 \times \log_{10}(S) - 0.577 \times [\log_{10}(S)]^2 + 0.29 \times [\log_{10}(S)]^3$$

$$(Q/RCNL)_{L/D=0.5} = 4.21 - 2.257 \times \log_{10}(S) + 2.107 \times [\log_{10}(S)]^2 - 0.7597 \times [\log_{10}(S)]^3$$

$$(Q/RCNL)_{L/D=0.6} = 3.874 - 1.3808 \times \log_{10}(S) + 1.177 \times [\log_{10}(S)]^2 - 0.4139 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9993$.

$$(Q/RCNL)_{L/D=1} = 3.42 - 0.5628 \times \log_{10}(S) + 0.7906 \times [\log_{10}(S)]^2 - 0.4478 \times [\log_{10}(S)]^3$$

$$(Q/RCNL)_{L/D=1.5} = 3.2388 - 0.1119 \times \log_{10}(S) + 0.3229 \times [\log_{10}(S)]^2 - 0.2714 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99$.

$$(Q/RCNL)_{L/D=2} = 3.1817 + 0.0097 \times \log_{10}(S) + 0.199 \times [\log_{10}(S)]^2 - 0.2199 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.987$.

5.1.6 Ratio of Side Flow to Total Flow

5.1.6.1 Sommerfeld number S from 0.002 - 0.1:

$$(Q_s/Q)_{L/D=0.25} = 0.7658 - 0.2628 \times \log_{10}(S) - 0.1036 \times [\log_{10}(S)]^2 - 0.014 \times [\log_{10}(S)]^3$$

$$(Q_s/Q)_{L/D=0.5} = 0.4403 - 0.6626 \times \log_{10}(S) - 0.2734 \times [\log_{10}(S)]^2 - 0.0382 \times [\log_{10}(S)]^3$$

$$(Q_s/Q)_{L/D=0.6} = 0.5026 - 0.5207 \times \log_{10}(S) - 0.188 \times [\log_{10}(S)]^2$$

$$- 0.0235 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9994$.

$$(Q_s/Q)_{L/D = 1} = - 0.0686 - 1.196 \times \log_{10}(S) - 0.4736 \times [\log_{10}(S)]^2$$

$$- 0.0651 \times [\log_{10}(S)]^3$$

$$(Q_s/Q)_{L/D = 1.5} = - 0.1582 - 1.0966 \times \log_{10}(S) - 0.4377 \times [\log_{10}(S)]^2$$

$$- 0.0606 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

$$(Q_s/Q)_{L/D = 2} = - 0.181 - 0.9736 \times \log_{10}(S) - 0.3941 \times [\log_{10}(S)]^2$$

$$- 0.0553 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9992$.

5.1.6.2 Sommerfeld number S from 0.1 - 1:

$$(Q_s/Q)_{L/D = 0.25} = 0.7596 - 0.3382 \times \log_{10}(S) - 0.2792 \times [\log_{10}(S)]^2$$

$$- 0.121 \times [\log_{10}(S)]^3$$

$$(Q_s/Q)_{L/D = 0.5} = 0.4952 - 0.5511 \times \log_{10}(S) - 0.1097 \times [\log_{10}(S)]^2$$

$$+ 0.0675 \times [\log_{10}(S)]^3$$

$$(Q_s/Q)_{L/D = 0.6} = 0.3949 - 0.4805 \times \log_{10}(S) + 0.2264 \times [\log_{10}(S)]^2$$

$$+ 0.2598 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9989$.

$$(Q_s/Q)_{L/D = 1} = 0.1957 - 0.3733 \times \log_{10}(S) + 0.3858 \times [\log_{10}(S)]^2$$

$$+ 0.2358 \times [\log_{10}(S)]^3$$

$$(Q_f/Q)_{L/D=1.5} = 0.1112 - 0.2609 \times \log_{10}(S) + 0.4389 \times [\log_{10}(S)]^2 + 0.254 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

$$(Q_f/Q)_{L/D=2} = 0.0747 - 0.2 \times \log_{10}(S) + 0.4024 \times [\log_{10}(S)]^2 + 0.2276 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

5.1.6.3 Sommerfeld number S from 1 - 10:

$$(Q_f/Q)_{L/D=0.25} = 0.76 - 0.2888 \times \log_{10}(S) - 0.4372 \times [\log_{10}(S)]^2 + 0.237 \times [\log_{10}(S)]^3$$

$$(Q_f/Q)_{L/D=0.5} = 0.5005 - 0.5186 \times \log_{10}(S) - 0.1401 \times [\log_{10}(S)]^2 + 0.2195 \times [\log_{10}(S)]^3$$

$$(Q_f/Q)_{L/D=0.6} = 0.3936 - 0.493 \times \log_{10}(S) + 0.0466 \times [\log_{10}(S)]^2 + 0.096 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

$$(Q_f/Q)_{L/D=1} = 0.2 - 0.4417 \times \log_{10}(S) + 0.4078 \times [\log_{10}(S)]^2 - 0.1461 \times [\log_{10}(S)]^3$$

$$(Q_f/Q)_{L/D=1.5} = 0.1015 - 0.264 \times \log_{10}(S) + 0.2845 \times [\log_{10}(S)]^2 - 0.111 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9968$.

$$(Q_f/Q)_{L/D=2} = 0.0676 - 0.2152 \times \log_{10}(S) + 0.2696 \times [\log_{10}(S)]^2$$

$$- 0.114 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99$.

5.1.7 Maximum Film Pressure Variable

5.1.7.1 Sommerfeld number S from 0.002 - 0.1:

$$(P/p_{max})_{L/D = 0.25} = 0.3427 + 0.2055 \times \log_{10}(S) + 0.0637 \times [\log_{10}(S)]^2$$
$$+ 0.0108 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D = 0.5} = 0.4667 + 0.2237 \times \log_{10}(S) + 0.0363 \times [\log_{10}(S)]^2$$
$$+ 0.003 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D = 0.6} = 0.5665 + 0.374 \times \log_{10}(S) + 0.1238 \times [\log_{10}(S)]^2$$
$$+ 0.0196 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9988$.

$$(P/p_{max})_{L/D = 1} = 0.658 + 0.3018 \times \log_{10}(S) + 0.0487 \times [\log_{10}(S)]^2$$
$$+ 0.006 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D = 1.5} = 0.5893 - 0.0676 \times \log_{10}(S) - 0.2015 \times [\log_{10}(S)]^2$$
$$- 0.0396 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.999$.

$$(P/p_{max})_{L/D = 2} = 0.5138 - 0.3709 \times \log_{10}(S) - 0.4053 \times [\log_{10}(S)]^2$$
$$- 0.0774 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

5.1.7.2 Sommerfeld number S from 0.1 - 1:

$$(P/p_{max})_{L/D=0.25} = 0.3204 + 0.1331 \times \log_{10}(S) - 0.0613 \times [\log_{10}(S)]^2 - 0.0643 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D=0.5} = 0.4593 + 0.1792 \times \log_{10}(S) - 0.0273 \times [\log_{10}(S)]^2 - 0.0272 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D=0.6} = 0.4718 + 0.1454 \times \log_{10}(S) - 0.0052 \times [\log_{10}(S)]^2 + 0.0382 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.99987$.

$$(P/p_{max})_{L/D=1} = 0.5403 + 0.0512 \times \log_{10}(S) - 0.087 \times [\log_{10}(S)]^2 + 0.007 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D=1.5} = 0.609 + 0.0474 \times \log_{10}(S) + 0.0262 \times [\log_{10}(S)]^2 + 0.1172 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9987$.

$$(P/p_{max})_{L/D=2} = 0.653 + 0.0308 \times \log_{10}(S) + 0.0223 \times [\log_{10}(S)]^2 + 0.11 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.998$.

5.1.7.3 Sommerfeld number S from 1 - 10:

$$(P/p_{max})_{L/D=0.25} = 0.3203 + 0.17 \times \log_{10}(S) + 0.1192 \times [\log_{10}(S)]^2 - 0.109 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D=0.5} = 0.46 + 0.2229 \times \log_{10}(S) - 0.2831 \times [\log_{10}(S)]^2$$

$$+ 0.1302 \times [\log_{10}(S)]^3$$

$$(P/p_{max})_{L/D=0.6} = 0.4595 + 0.2157 \times \log_{10}(S) - 0.2949 \times [\log_{10}(S)]^2$$

$$+ 0.138 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$.

$$(P/p_{max})_{L/D=1} = 0.54 + 0.01 \times \log_{10}(S)$$

$$(P/p_{max})_{L/D=1.5} = 0.61 - 0.0369 \times \log_{10}(S) + 0.0646 \times [\log_{10}(S)]^2$$

$$- 0.0289 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9967$.

$$(P/p_{max})_{L/D=2} = 0.6556 - 0.0507 \times \log_{10}(S) + 0.0868 \times [\log_{10}(S)]^2$$

$$- 0.0398 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.992$.

5.2 ITERATIVE METHOD FOR TEMPERATURE-RISE DETERMINATION

In this program, the user will enter the value of load W, speed N, unit load P, lubricant inlet temperature T_{in}, etc., in the input stage. Then diameter of the bearing can be calculated. On the basis of the diameter and speed, the clearance can be selected from the database (c.f. Table 5.1). To find viscosity of the lubricant, operating (average) temperature is required. For this purpose, the initial trial temperature-rise of 20°C is used. This is denoted as ΔT_r. Thus the operating temperature is $T_{op} = T_{in} + \Delta T_r/2$. Then the viscosity can be obtained using viscosity-temperature relationship discussed in Sec. 4.1. Corresponding to this viscosity, the bearing characteristic number S is determined from the

relation $S = (R/C)^2 \mu N/P$. Once S is known temperature-rise variable is found with the help of equations described in Sec. 5.1.1 and that is used to calculate temperature-rise ΔT from Eq. (5.1).

Now the absolute value of the difference between ΔT and ΔT_r is found to check whether the initial guess was right, otherwise the whole process is repeated with a new value of the temperature-rise of $(\Delta T_r + \Delta T)/2$. If the absolute difference between ΔT and ΔT_r is less than $3^\circ C$, the iterative process is stopped and the temperature-rise is taken as the average of the latest values of ΔT and ΔT_r. A Prolog recursive routine is used to determine the temperature-rise iteratively. Pertaining to this value of temperature-rise, the operating temperature and corresponding viscosity and Sommerfeld number S are obtained.

The several stages of the program for designing full journal bearings for optimal clearance condition are described below. In this program, a database is used to show menus for the bearing-application to get value of unit load P, L/D ratio, decision-criterion to find utility value, and oil-grade. The first three are shown in the first stage while menu for oil is needed in second stage. This is referred to as the input-menu database.

5.3 STAGE I

The final goal of stage I is to get the functional and prescriptive specifications (c.f. Sec. 1.4) entered by the user, determine the bearing length and diameter, select the clearance from the database and check whether it satisfies the manufacturability criterion. This has been divided into two parts as follows:

5.3.1 Stage 1.1 or Input Stage

The functional and prescriptive specifications available at this stage in the 'Expert System' for designing full journal bearings for optimal clearance condition are listed below:

1. the bearing radial load to be supported: W (kN).
2. the speed of the journal: N_j *(rpm)*.
3. the bearing unit load: P (MPa). the user either enters the value or selects the application from the menu where the bearing would be used to get the value from the database.
4. the bearing industrial application for the allowable clearance limits.
5. the lubricant inlet temperature: T_{in} (°C).
6. the decision criterion for utility value.
7. the L/D ratio (0.25, 0.5, 0.6, 1.0, 1.5, 2.0, or all six).

The group of rules and databases used at this stage are:

1. a set of rules for menu-handling and screen-handling.
2. a set of rules for prompting and entering the functional and prescriptive specifications,
3. if the user selects the application instead of entering the value of P, a knowledge base in the form of a table (c.f. Table 4.1) is used.
4. a knowledge base containing Table 4.2 to get the value of L/D ratio,
5. a set of rules to find the length and corresponding diameter of the bearing using P = W/(L.D).

Thus each partial solution of this stage consists of the L/D ratio, length, and diameter. At the end of this stage all partial solutions are written onto an

internal database. In this stage, there may be six partial solutions, if the user selects all the six relevant L/D ratios. Therefore, the recombining skills of a designer are used in this stage.

5.3.2 Stage 1.2 or Selection and Checking of Clearance

The final goal of this stage is to select the clearance and check whether this is in the prescribed range for each partial solution of the above input stage. The group of rules and database used in this stage are given below:

1. a set of rules to select the clearance from the database depending on the diameter and speed.
2. a knowledge base consisting of the values of clearance corresponding to the diameter and speed, which is in the form of Table 5.1.
3. a set of rules to check whether the calculated clearance in each case is within the prescribed range,
4. a database consisting of clearance limits depending on the industrial application in the form of a table (c.f. Table 3.1).

If no partial solution within the desired clearance limit for manufacturability is obtained, this condition may be modified to act as a failure handler [34]. Each partial solution of this stage will have the unit load P, speed N, L/D ratio, length L, diameter D, clearance C, corresponding to the minimum limit of clearance C_{min}, and lubricant inlet temperature T_{in}. At the end of stage 1.2, all partial solutions are written onto an internal database. The database consisting of all partial solutions of the stage 1.1 is erased. All the databases of stage 1 are also erased.

For each partial solution of stage 1.1 there may be either no solution or one solution at stage 1.2. A failure handler is used if there is not even a single solution. That is the industrial application is changed or clearance check condition is modified or the maximum bearing diameter limit of 200 mm is increased. In stage 1.2 only discriminating skills of a designer are used. The number of partial solutions at the end of this stage (stage 1.2) is N_1, which will be less than or equal to the number of partial solutions available at the end of stage 1.1.

Thus the number of partial solutions available at the end of stage 1 is N_1.

5.4 STAGE 2

The final goal of stage 2 is to get the oil-grade and determine the temperature-rise using the iterative method discussed in Sec. 5.2, the viscosity of the oil at the operating temperature T_{op}, and the bearing characteristic number S. Then the lubricant outlet temperature T_{out} is calculated and checked w.r.t. the condition mentioned in Sec. 3.3.1. If the outlet temperature is in the prescribed limit, all the performance variables (related to respective functional specifications): h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, and P/p_{max} are determined with the help of equations written in terms of $\log_{10}(S)$ in Sec. 5.1. To achieve this goal the following sets of rules and databases are used:

1. a set of rules to get the oil-grade (SAE 10, 20, 30, 40, 50, 60, 70, or all seven) as selected by the user.
2. a set of rules to determine the temperature-rise from the initial trial value to the final value using an iterative method (c.f. Sec. 5.2).
3. a set of rules to determine the viscosity of the oil using the equations

described in Sec. 4.1,

4. a database consisting of the coefficients of the equations for determining the viscosity (c.f. Sec. 4.1), which is in the form of a table.
5. a set of rules to determine S using the relation $S = (R/C)^2 \mu N/P$.
6. a set of rules to find the outlet temperature and check whether it is below 120°C,
7. a set of rules to determine the performance variables of all the solutions satisfying the temperature-check condition using the equations described in Sec. 5.1,
8. a database consisting of the coefficients of the equations for determining the performance variables (c.f. Sec. 5.1), which is in the form of a table.

Each partial feasible solution of this stage consists of the viscosity μ, unit load P, speed N, oil-grade, L/D ratio, length L, diameter D, clearance C, corresponding to the minimum limit of clearance C_{min}, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, P/p_{max}, and $(h_{0_{F.S.\ 2}})/C$ needed in checking minimum film thickness requirement in the next stage and is obtained when S is halved. At the end of stage 2 all partial solutions are written onto an internal database (on the blackboard). The databases of stage 2 containing coefficients of the equations to determine viscosity and the performance variables are retracted. Also the input-menu database and the database containing all partial solutions of stage 1 are retracted.

In stage 2 both recombining and discriminating skills of a designer are used. Recombining in the sense if the user selects all the seven oil-grades, the solutions available at the end of first stage will increase and discriminating in the sense that only those solutions are retained which satisfy the outlet temperature condition. The number of partial solutions of stage 2 is N_2 and it is not directly related to the number of solutions of the previous stage N_1.

For each partial solution of stage 1 there may be either no solution or one or more solutions at stage 2. A failure handler is used if there is not even a single solution: that is, the temperature condition may be modified.

5.5 STAGE 3

The final goal of this stage is to determine whether or not each of the partial solutions of stage 2 satisfies the minimum film thickness criterion given by Juvinall [68] (c.f. Sec. 3.3.3); that is for a factor of safety of 2, $h_0 \geq 0.005 + 0.00004D$ (h_0 and D *are in mm*). A set of rules to determine the value of $h_{0_{F.S.=2}}$ and check whether it is in the acceptable range is used in this stage.

Each design solution of stage 3 will have the viscosity μ, unit load P, speed N, oil-grade, L/D ratio, length L, diameter D, clearance C, corresponding to the minimum limit of clearance C_{min}, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, and P/p_{max}. A failure handler may be used if there is no design solution at the end of this stage. That is, this constraint may be modified.

At the end of stage 3 all partial solutions are written onto an internal database and at the same time the database consisting of all partial solutions of stage 2 is erased. In stage 3 only discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_3 will be less than or equal to the number of partial solutions in the previous stage N_2.

5.6 STAGE 4

The final goal of this stage is to determine whether or not each of the partial solutions of stage 3 is stable (c.f. Sec. 3.3.4). The group of rules used in this stage is given below:

1. a set of rules to determine actual h_0, coefficient of friction f, torque T (=fWR), eccentricity ratio $\varepsilon\,(=1-h_0/C)$, and dimensionless speed ω_s $(=\omega\sqrt{C/g},\ \omega=2\pi N)$.
2. a set of rules to check whether the design solution is stable, i.e., either $\varepsilon \geq 0.83$ *or* $\omega_s \leq 2.33$.

Each design solution of stage 4 will have the viscosity μ, unit load P, speed N, load W, oil-grade, L/D ratio, length L, diameter D, clearance C, corresponding minimum limit of clearance C_{min}, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q, Q, P/p_{max}, actual h_0, torque T, eccentricity ratio ε, and dimensionless speed ω_s. A failure handler may be used if there is no design solution at the end of this stage. That is, this constraint may be modified.

At the end of stage 4 all partial solutions are written onto an internal

database and at the same time the database consisting of all partial solutions of stage 3 is erased. In stage 4 only discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_4 will be less than or equal to the number of partial solutions in the previous stage N_3.

5.7 STAGE 5

The final goal of this stage is to determine the utility value of all the design solutions of stage 4 (c.f. Sec. 3.3.5). The group of rules and databases used in this stage are given below:

1. a set of rules to get the decision criterion entered by the user in the input stage 1.1, i.e., optimal clearance,
2. a knowledge base consisting of the utility values and the weighting factors for this decision criterion corresponding to all the five decision variables (c.f. Tables 3.2 to 3.7),
3. a set of rules to determine the utility value for each design solution of stage 4 using Eq. (4.3).

Each design solution of stage 5 will have the viscosity μ, unit load P, speed N, load W, oil-grade, L/D ratio, length L, diameter D, clearance C, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, P/p_{max}, actual h_0, torque T, eccentricity ratio ε, dimensionless speed ω_s, and the utility value U. This stage is a simple procedural stage. Neither the discriminating nor the recombining skills of a designer are used at this stage. Therefore, the number of design solutions of this stage N_5 will be the same as that of stage 4 N_4.

At the end of stage 5 all partial solutions are written onto an internal database. The design solutions of stage 4 are erased. The database of the stage 5 is also erased. The user can modify the database in Tables 3.2 to 3.7.

5.8 STAGE 6

This stage consists of two parts:

5.8.1 Stage 6.1 or Decision Making Stage

The goal of this stage is to identify the design solution of the previous stage which has the highest utility factor U_{max}. In addition all the design solutions of the previous stage for which the utility factors are at least ninety five percent (95%) of the U_{max} are identified.

The group of rules and the databases used in this stage are given below:

1. a set of rules to identify the design solution which has the highest utility value U_{max},
2. a set of rules to identify the design solutions which have the utility values of at least 95% of the U_{max}.

In this stage only the discriminating skills of a designer are used. Therefore the number of solutions at the end of this stage N_6 is less than or equal to the number of solutions of the previous stage N_5.

Each design solution of stage 6.1 will have the viscosity μ, unit load P, speed N, load W, oil-grade, L/D ratio, length L, diameter D, clearance C, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL ,

Q_s/Q, P/p_{max}, actual h_o, torque T, eccentricity ratio ε, dimensionless speed ω_s, and the utility value U. At the end of stage 6.1 all design solutions are written onto an internal database and at the same time all partial solutions of stage 5 are erased.

5.8.2 Stage 6.2 or Output Stage

The goal of this stage is to file all the selected design solutions. Each design solution for which the utility value is at least ninety five percent (95%) of the highest utility value (U_{max}) is filed in the 'results.out' file for the perusal of the user.

The group of rules and the databases used in this stage are given below:

1. a set of rules and databases to find stiffness and damping coefficients using the relations given by Seireg and Dandage [65] for $1/4 \le L/D \le 1$.
2. a set of rules to calculate eccentricity e, coefficient of friction f, power loss H ($= 2\pi NT$), oil flow Q, side leakage Q_s, and maximum pressure p_{max}.
3. a knowledge base containing the L/D ratio and the oil-grade.
4. a set of rules to print the best design solutions in the 'results.out' file.

In this stage neither the discriminating nor the recombining skills of a designer are used. Therefore the number of solutions at the end of this stage is the same as N_6. At the end of this stage all the solutions are filed in the 'results.out' file. The design solutions of all the previous stages including stage 6.1 are erased and the database used in this stage are also erased.

The 'results.out' file will have the functional and prescriptive specifications such as the load W, speed N, unit load P, bearing industrial application,

lubricant inlet temperature, decision criterion, L/D ratio, oil-grade, length L, diameter D, T_{op}, μ, S, C, h_o, ε, eccentricity e, coefficient of friction f, torque T, power loss H, total oil supplied Q, side leakage Q_s, maximum film pressure p_{max}, position of minimum film thickness Φ, dimensionless velocity ω_c, utility value U, and stiffness and damping coefficients.

5.9 EXAMPLE

To illustrate the use of the present 'Expert System' it is used to design full journal bearings for optimal clearance condition. This example is similar to the one given by Juvinall [68] and solved in the previous chapter for the max. W and the min. f conditions. The input specifications for this example are as follows:

Bearing load = 17 kN

Speed of the journal = 1800 rpm

Bearing unit load = 1.511 MPa

Bearing industrial application = Electric motor and generator practice

Lubricant inlet temperature = 75 °C

L/D ratio: all six ratios

Oil-grade: all seven oil-grades.

In this case the number of partial design solutions at the end of stage 1 (N_1) is 4. Solutions corresponding to L/D ratio of 0.25 and 2.0 are discarded because D is 212 mm which is higher than acceptable limit of 200 mm and C is 0.0483 mm which is below the acepetable range of 0.0508 to 0.1016 mm for electric motor and generator practice for shaft diameter under 100 mm (D = 75 mm), respectively. The number of partial design solutions at the end of stage 2

(N_2) is 28. At the end of stage 3 there are twenty eight (N_3 = 28) solutions. The number of design solutions at the end of stage 4 (N_4) is 28. At the end of stage 5 there are twenty eight (N_5=28) solutions. The number of design solutions at the end of decision making stage (N_6) is 10 and all ten design alternatives are filed in the 'results.out' file. The print out of this file is attached herewith on the succeeding pages. Out of these ten design alternatives, the solution with L/D = 1.5 and SAE 30 oil has the highest utility value = 3.43 and solution with L/D = 1 and SAE 10 oil has the second highest utility value = 3.42.

The load on the bearing, W = 17.00 kN

The speed of the journal, N = 30.000 rps

The bearing unit load, P = 1.511 MPa

The bearing-application for clearance is No. 2. Electric motor and generator practice

The inlet temperature, Tin = 75.000 deg C

The decision-criterion for utility is No. 1. Based on optimal clearance

Stable Design Alternatives Are

1. L/D = 1.0

Design Based on SAE 10 Oil

Length, L = 106 mm

Diameter, D = 106.00 mm

Operating Temperature, Top = 79.71 deg. C

Viscosity of the Lubricant, Mu = 6.90 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.049

Radial Clearance, C = 0.089 mm

Minimum film thickness, hmin = 0.019 mm

Eccentricity Ratio, e/C = 0.791

Eccentricity, e = 0.070 mm

Coefficient of friction, f = 0.0030

Torque required = 2.714 N.m

Power Lost = 511.79 W

Total Oil Supplied = 68735.2 cubic mm per sec

Side Leakage = 57219.0 cubic mm per sec

Maximum Film Pressure = 4.548 MPa

Position of minimum film thickness = 36.88 degree

The dimensionless velocity, ws = 0.57

The Utility value = 3.42

Stiffness coefficient, Kxx = 929.91 kN/mm

Kxy = 770.73 kN/mm

Kyx = 98.22 kN/mm

Kyy = 361.69 kN/mm

Damping coefficient, Cxx = 5.87 kN-s/mm

Cxy = 1.64 kN-s/mm

Cyx = 1.45 kN-s/mm

Cyy = 1.23 kN-s/mm

2. L/D = 1.5

Design Based on SAE 10 Oil

Length, L = 130 mm

Diameter, D = 86.67 mm

Operating Temperature, Top = 80.18 deg. C

Viscosity of the Lubricant, Mu = 6.81 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.065

Radial Clearance, C = 0.063 mm

Minimum film thickness, hmin = 0.021 mm

Eccentricity Ratio, e/C = 0.668

Eccentricity, e = 0.042 mm

Coefficient of friction, f = 0.0028

Torque required = 2.066 N.m

Power Lost = 389.61 W

Total Oil Supplied = 42279.5 cubic mm per sec

Side Leakage = 26554.1 cubic mm per sec

Maximum Film Pressure = 3.344 MPa

Position of minimum film thickness = 46.41 degree

The dimensionless velocity, ws = 0.48

The Utility value = 3.35

Stiffness and dammping coefficients are not found.

3. L/D = 1.0

Design Based on SAE 20 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 79.71 deg. C

Viscosity of the Lubricant, Mu = 8.23 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.058

Radial Clearance, C = 0.089 mm

Minimum film thickness, hmin = 0.021 mm

Eccentricity Ratio, e/C = 0.763

Eccentricity, e = 0.068 mm

Coefficient of friction, f = 0.0034

Torque required = 3.022 N.m

Power Lost = 569.92 W

Total Oil Supplied = 68148.1 cubic mm per sec

Side Leakage = 55135.5 cubic mm per sec

Maximum Film Pressure = 4.341 MPa

Position of minimum film thickness = 39.23 degree

The dimensionless velocity, ws = 0.57

The Utility value = 3.34

Stiffness coefficients: Kxx = 823.08 kN/mm; Kxy = 740.98 kN/mm

Kyx = 73.49 kN/mm; Kyy = 367.77 kN/mm

Damping coefficients: Cxx = 6.35 kN-s/mm; Cxy = 1.85 kN-s/mm

Cyx = 1.64 kN-s/mm; Cyy = 1.35 kN-s/mm

4. L/D = 1.5

Design Based on SAE 20 Oil

Length, L = 130 mm

Diameter, D = 86.67 mm

Operating Temperature, Top = 80.18 deg. C

Viscosity of the Lubricant, Mu = 8.10 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.077

Radial Clearance, C = 0.063 mm

Minimum film thickness, hmin = 0.023 mm

Eccentricity Ratio, e/C = 0.628

Eccentricity, e = 0.039 mm

Coefficient of friction, f = 0.0031

Torque required = 2.292 N.m

Power Lost = 432.18 W

Total Oil Supplied = 43370.7 cubic mm per sec

Side Leakage = 26161.4 cubic mm per sec

Maximum Film Pressure = 3.216 MPa

Position of minimum film thickness = 49.19 degree

The dimensionless velocity, ws = 0.48

The Utility value = 3.37

Stiffness and dammping coefficients are not found.

5. L/D = 1.0

Design Based on SAE 30 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 79.71 deg. C

Viscosity of the Lubricant, Mu = 12.78 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.090

Radial Clearance, C = 0.089 mm

Minimum film thickness, hmin = 0.028 mm

Eccentricity Ratio, e/C = 0.682

Eccentricity, e = 0.061 mm

Coefficient of friction, f = 0.0044

Torque required = 3.990 N.m

Power Lost = 752.36 W

Total Oil Supplied = 66357.3 cubic mm per sec

Side Leakage = 48986.9 cubic mm per sec

Maximum Film Pressure = 3.884 MPa

Position of minimum film thickness = 46.13 degree

The dimensionless velocity, ws = 0.57

The Utility value = 3.38

Stiffness coefficients: Kxx = 606.49 kN/mm; Kxy = 671.48 kN/mm

Kyx = -8.32 kN/mm; Kyy = 383.43 kN/mm

Damping coefficients: Cxx = 6.05 kN-s/mm; Cxy = 1.84 kN-s/mm

Cyx = 1.75 kN-s/mm; Cyy = 1.86 kN-s/mm

6. L/D = 1.5

Design Based on SAE 30 Oil

Length, L = 130 mm

Diameter, D = 86.67 mm

Operating Temperature, Top = 80.18 deg. C

Viscosity of the Lubricant, Mu = 12.59 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.120

Radial Clearance, C = 0.063 mm

Minimum film thickness, hmin = 0.031 mm

Eccentricity Ratio, e/C = 0.499

Eccentricity, e = 0.031 mm

Coefficient of friction, f = 0.0038

Torque required = 2.821 N.m

Power Lost = 532.04 W

Total Oil Supplied = 41095.5 cubic mm per sec

Side Leakage = 21575.6 cubic mm per sec

Maximum Film Pressure = 3.045 MPa

Position of minimum film thickness = 58.10 degree

The dimensionless velocity, ws = 0.48

The Utility value = 3.43

Stiffness and dammping coefficients are not found.

7. L/D = 1.0

Design Based on SAE 40 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 79.71 deg. C

Viscosity of the Lubricant, Mu = 16.48 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.116

Radial Clearance, C = 0.089 mm

Minimum film thickness, hmin = 0.033 mm

Eccentricity Ratio, e/C = 0.630

Eccentricity, e = 0.056 mm

Coefficient of friction, f = 0.0054

Torque required = 4.845 N.m

Power Lost = 913.60 W

Total Oil Supplied = 65216.8 cubic mm per sec

Side Leakage = 44930.9 cubic mm per sec

Maximum Film Pressure = 3.678 MPa

Position of minimum film thickness = 52.19 degree

The dimensionless velocity, ws = 0.57

The Utility value = 3.36

Stiffness coefficients: Kxx = 508.42 kN/mm; Kxy = 634.35 kN/mm

Kyx = -71.43 kN/mm; Kyy = 392.77 kN/mm

Damping coefficients: Cxx = 5.89 kN-s/mm; Cxy = 1.85 kN-s/mm

Cyx = 1.84 kN-s/mm; Cyy = 2.29 kN-s/mm

8. L/D = 1.5

Design Based on SAE 40 Oil

Length, L = 130 mm

Diameter, D = 86.67 mm

Operating Temperature, Top = 80.18 deg. C

Viscosity of the Lubricant, Mu = 16.22 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.155

Radial Clearance, C = 0.063 mm

Minimum film thickness, hmin = 0.034 mm

Eccentricity Ratio, e/C = 0.452

Eccentricity, e = 0.028 mm

Coefficient of friction, f = 0.0050

Torque required = 3.706 N.m

Power Lost = 698.94 W

Total Oil Supplied = 40255.1 cubic mm per sec

Side Leakage = 19145.3 cubic mm per sec

Maximum Film Pressure = 2.876 MPa

Position of minimum film thickness = 60.20 degree

The dimensionless velocity, ws = 0.48

The Utility value = 3.39

Stiffness and dammping coefficients are not found.

9. L/D = 1.0

Design Based on SAE 50 Oil

Length, L = 106 mm; Diameter, D = 106.00 mm

Operating Temperature, Top = 79.71 deg. C

Viscosity of the Lubricant, Mu = 26.39 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.186

Radial Clearance, C = 0.089 mm

Minimum film thickness, hmin = 0.043 mm

Eccentricity Ratio, e/C = 0.517

Eccentricity, e = 0.046 mm

Coefficient of friction, f = 0.0075

Torque required = 6.720 N.m

Power Lost = 1267.21 W

Total Oil Supplied = 62212.7 cubic mm per sec

Side Leakage = 36211.0 cubic mm per sec

Maximum Film Pressure = 3.329 MPa

Position of minimum film thickness = 57.23 degree

The dimensionless velocity, ws = 0.57

The Utility value = 3.26

Stiffness coefficients: Kxx = 386.68 kN/mm; Kxy = 668.81 kN/mm

Kyx = 350.56 kN/mm; Kyy = 398.69 kN/mm

Damping coefficients: Cxx = 5.85 kN-s/mm; Cxy = 1.94 kN-s/mm

Cyx = 2.09 kN-s/mm; Cyy = 3.54 kN-s/mm

10. L/D = 1.5

Design Based on SAE 50 Oil

Length, L = 130 mm

Diameter, D = 86.67 mm

Operating Temperature, Top = 80.18 deg. C

Viscosity of the Lubricant, Mu = 25.94 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.248

Radial Clearance, C = 0.063 mm

Minimum film thickness, hmin = 0.041 mm

Eccentricity Ratio, e/C = 0.345

Eccentricity, e = 0.022 mm

Coefficient of friction, f = 0.0074

Torque required = 5.433 N.m

Power Lost = 1024.47 W

Total Oil Supplied = 38650.2 cubic mm per sec

Side Leakage = 14460.1 cubic mm per sec

Maximum Film Pressure = 2.680 MPa

Position of minimum film thickness = 65.20 degree

The dimensionless velocity, ws = 0.48

The Utility value = 3.31

Stiffness and dammping coefficients are not found.

In this example, all the design alternatives satisfy temperature, minimum film thickness, and stability constraints, therefore at the end of stage 2, 3, and 4 (with all 7 oil-grades tried), there are 28 solutions always. For example, for L/D = 0.5 and SAE 10 oil, the lubricant outlet temperature T_{out} is found to be 88.3 °C less than 120 °C, $h_{\theta_{FS-2}}$ =0.012 mm which is greater than 0.005+0.00004D = 0.011 mm (D = 150 mm), and ε = 0.8 which is less than 0.83; but ω_c = 0.57 less than 2.33.

Out of the twenty eight (28) solutions, eighteen (18) are discarded in the decision making stage. For example, the utility value U on the basis of optimal clearance for the above stable solution for L/D = 0.5 and SAE 10 oil is 3.2. On the same basis U_{max} = 3.43 whose ninety five percent (95%) is 3.26. Therefore the solution with U = 3.2 is rejected in the decision making stage (stage 6.1) and could not be printed in the 'results.out' file. The number of solution at the end of stage 6 which are filed is only 10 which is less than 28 at the end of stage 5.

5.10 COMPARISON WITH PREVIOUS WORK

The example in Sec. 5.9 is very general to make exhaustive search. One case of this example with L/D = 0.5 and SAE 10 oil has been solved by Juvinall [68]; therefore the present solution is compared with his solution. Instead of giving T_{in} as input he has given T_{sp} =82°C as input. To have the same operating temperature, inlet temperature is taken as 75 °C, for optimal clearance condition when solving by the 'Expert System'. As mentioned earlier that the solution with L/D = 0.5 and SAE 10 oil is rejected by the 'ES' because of low utility value. If the acceptable limit of the utility value, which a solution

may have, is lowered to 90% of the f'_{max}, then solution is obtained by the 'FS'. It is to be noted that for the max. W and the min. f conditions this problem has been solved in Chap. 4. Thus in order to show the solutions graphically, there are three points or there are solutions available for three clearance values. Changing the database from which the clearance is selected, two more solutions are obtained for two other clearance values. Variation of functional specifications: h_{cr}, f, Q, and T_{max} versus clearance for constant μ, N, L, D, and W is shown in Fig. 5.2. It may be observed that the optimal clearance = 0.089 mm lies in between the values found with the max. W (C = 0.046 mm) and the min. f conditions (C = 0.142 mm). This is in agreement with Juvinall's recommendation [68]. Thus the clearance value selected on the basis of Fig. 5.1 (given by Keith, Jr. [15-16]) is a good compromise between the max. W and min. f conditions. However, Shigley [17] argues that the optimal clearance should be lower than that for max. W because wear increases the clearance. But low clearance value increases the cost of manufacturing, therefore is not suitable. As far as wear is concerned, it can be noticed from the present solution that even if the clearance is increased by 40% it will be within the safe limit of the min. film thickness. Another advantage is that larger clearance, such as 0.08 to 0.11 mm, would result in lower friction losses and tend to make the bearing run cooler.

Comparing values of the present solution for optimal clearance with that given by Juvinall [68], it is found that Juvinall suggests SAE 20 oil instead of SAE 10 oil because of minimum film thickness requirement. The reason is that he is not only taking factor of safety of 2 in load, but also taking the value of C that corresponds to the min. f condition. This makes factor of safety higher than

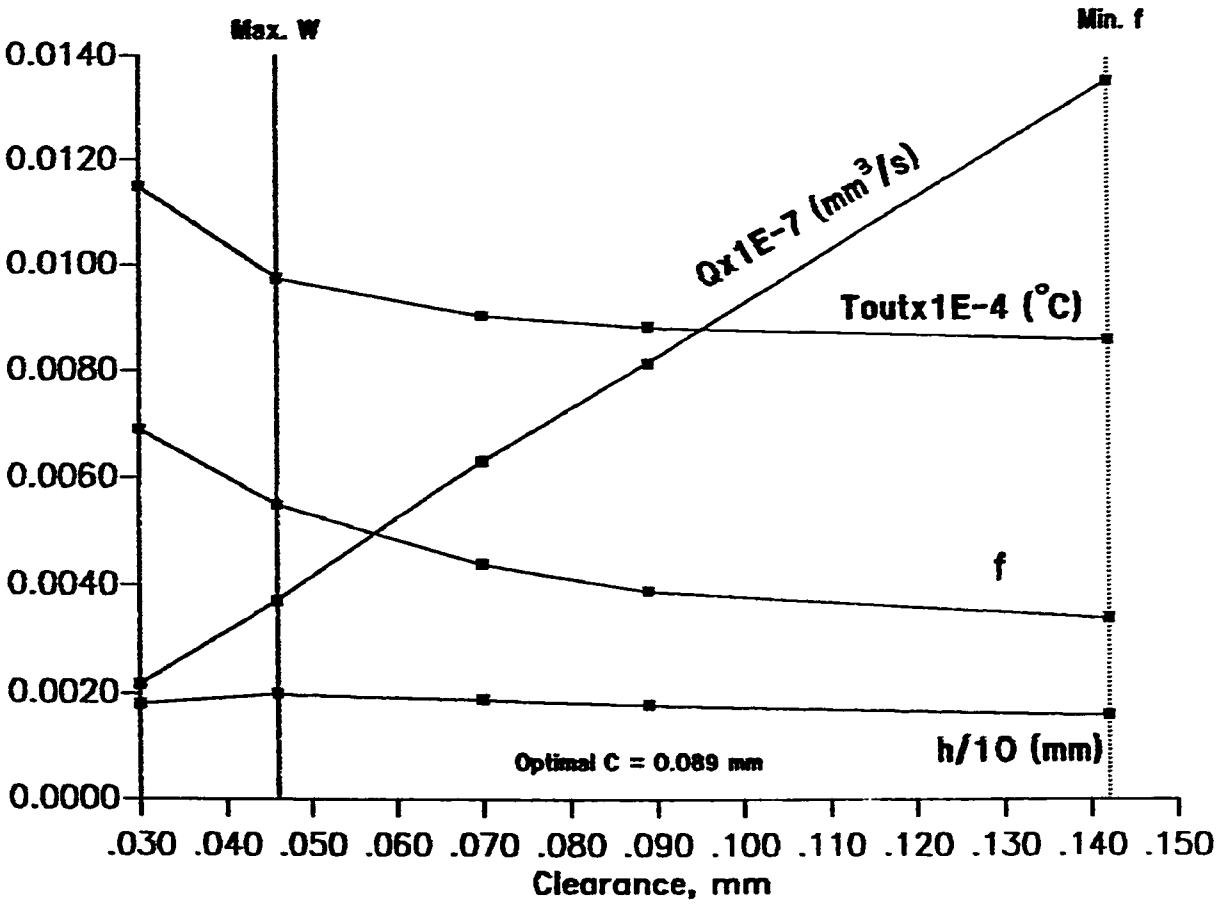


Fig. 5.2. Variation of h_0, f, Q, and T_{out} With Clearance When W, N, L, D, and μ Are Constant (Example of [68]).

2 in calculating S. For this value of S, he obtains h_0 which does not satisfy Juvinall's criterion. However, in the present system h_0 is determined with factor of safety 2 in calculating S, and therefore found to be acceptable in the case of SAE 10 oil.

For comparison purpose, the solution with SAE 20 oil obtained from the 'ES' is C = 0.089 mm, h_0 = 0.02 mm, f = 0.0044, Q = 81,049.5 mm^3/s, side leakage Q_s = 70,015.4 mm^3/s, power loss H = 1050.24 W, and temperature-rise ΔT = 13.3 °C. For the same condition Juvinall [68] suggests C = 0.05 to 0.07 mm (the 0.07 dimension may be increased owing to the cost consideration), side leakage Q_s = 31,700 to 52,100 mm^3/s, power loss H = 1180 to 990 W, and temperature-rise ΔT = 27.3 to 13.9 °C. It can be viewed that the differences in both results are because of the difference in clearance, otherwise the results are similar. Therefore it can be concluded that the program is working well.

Regarding the optimum range of clearance, another example is taken from Shigley and Mischke [80]; the corresponding functional and prescriptive specifications are: W = 2.224 kN, N = 1800 rpm, P = 1.532 MPa, T_{in} = 37.78 °C, L/D = 1, and SAE 20 oil.

The solution obtained from the 'ES' is presented in Fig. 5.3 in which variation of h_0, T_{out}, Q, and f is shown w.r.t. clearance. For comparison, for C = 0.0445 mm, the 'ES' gives the solution as h_0 = 0.018 mm, f = 0.0081, Q = 4148.5 mm^3/s, and T_{out} = 50.84°C, whereas Shigley and Mischke [80] found h_0 = 0.0193 mm, f = 0.0084, Q = 4129.5 mm^3/s, and T_{out} = 53.33°C. The values

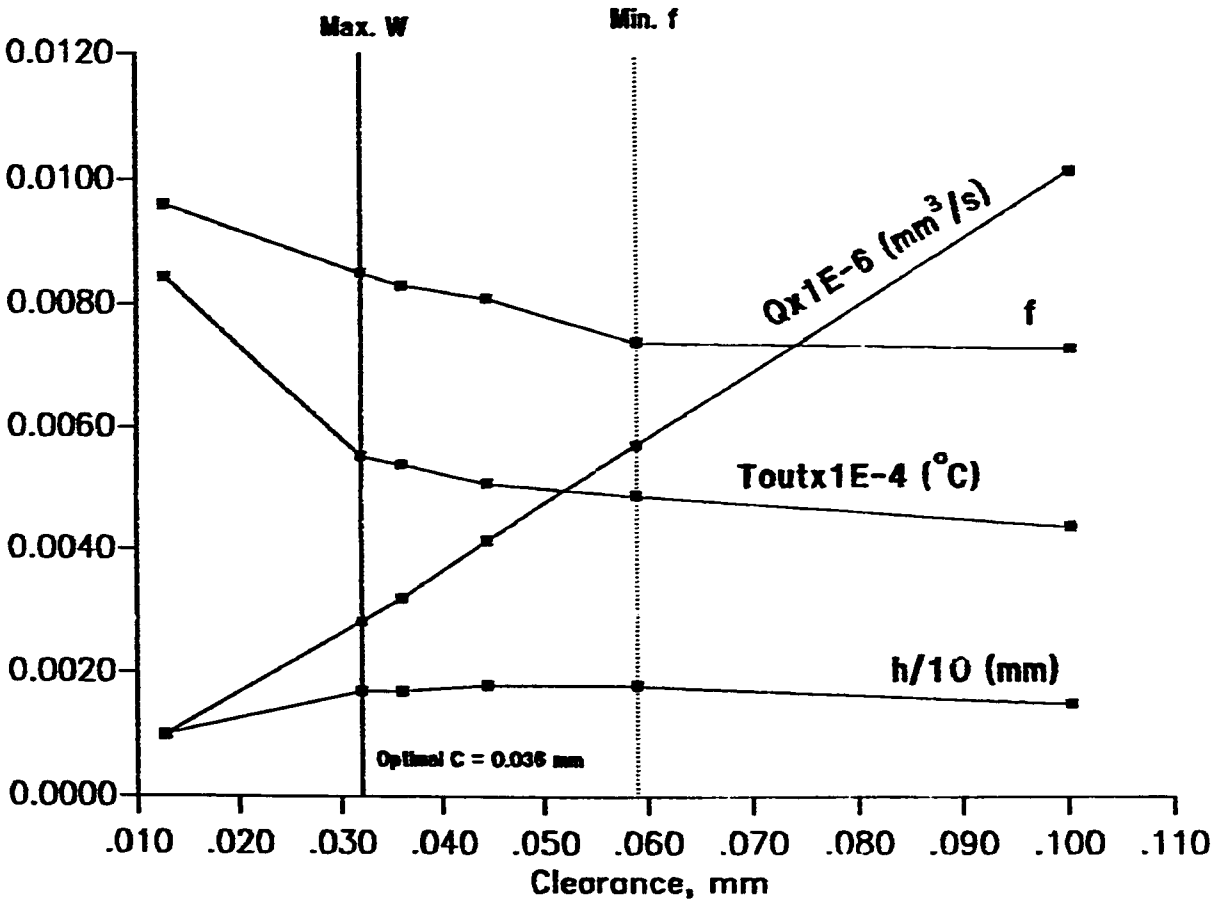


Fig. 5.3. Variation of h_0, f, Q, and T_{out} With Clearance When W, N, L, D, and μ Are Constant (Example of [80]).

are in good agreement except T_{out}. The difference in T_{out} values may be owing to the iterative method used for finding the temperature-rise.

The optimal clearance (= 0.036 mm) found, falls in the range of max. W (C = 0.032 mm) and min. f (C = 0.059 mm) whereas Shigley and Mischke [80] recommend an optimal clearance of 0.028 mm, below that corresponding to the max. W.

CHAPTER 6

DESIGN OF PARTIAL JOURNAL BEARINGS FOR MAXIMUM LOAD AND MINIMUM FRICTION

In the present work, partial bearings of only 180°, 120°, and 60° arcs are considered. The objective knowledge required for designing partial journal bearings for the maximum load(W) and the minimum friction(f) is collected from the following sources: Raimondi and Boyd [24], Keith, Jr. [15-16], Orthwein [19], and Spotts [67]. The values of minimum film thickness variable h_o/C for both the conditions: max. W and min. f, taking film rupture into account, for partial journal bearings at L/D = 0.25, 0.5, 1.0, and ∞ are given by Raimondi and Boyd [24] and corresponding values of the Sommerfeld number S and the relevant performance variables are obtained from the tabulated data of the same source (by interpolation, if necessary). Then, the values at L/D = 0.6, 1.5, and 2.0 are found with the help of Eq. (4.1).

The structure of the program developed for designing partial journal bearings for the max. W and the min. f conditions is similar to that discussed in chapter 4, since the Sommerfled number S is known. Also, viscosity-temperature relationships used here are the same as described in Sec. 4.1. However, in the case of partial bearings, stiffness and damping coefficients are not found due to the lack of an appropriate knowledge-base. Different stages of the program for designing partial journal bearings for the max. W and the min. f conditions, are described below.

6.1 INPUT STAGE (STAGE 0)

The functional and prescriptive specifications(c.f. Sec. 1.4) available at this stage (stage 0) in the 'Expert System' for designing partial journal bearings for the maximum load and the minimum friction conditions are listed below:

1. the bearing radial load to be supported: W (kN),
2. the speed of the journal: $N_j(rpm)$,
3. the bearing arc β (180°, 120°, 60°, or all three),
4. the bearing unit load: P (MPa), the user either enters or selects the application from the menu where the bearing would be used to get the value from the database,
5. the bearing industrial application to obtain the allowable range of clearance,
6. the lubricant inlet temperature: T_{in} (°C),
7. the decision-criterion (max. W or min. f),
8. the L/D ratio (0.25, 0.5, 0.6, 1.0, 1.5, 2.0, or all six).

The group of rules and databases used at this stage are:

1. a set of rules for menu-handling and screen-handling,
2. a set of rules for prompting and entering the functional and prescriptive specifications,
3. if the user selects the application instead of entering the value of P, a knowledge base in the form of a table(c.f. Table 4.1),
4. a knowledge base containing Table 4.2 to get the value of L/D ratio,
5. a set of rules to find the length and corresponding diameter of the bearing using P = W/(L.D).

Thus, each partial solution of stage 0 consists of the L/D ratio, length, and

diameter. In this stage, the recombining skills of a designer are used in combining the solutions pertaining to all the three bearing arcs and six L/D ratios. At the end of this stage all partial solutions are written onto an internal database. All databases of stage 0 are then retracted except the newly written database of all partial solutions of stage 0.

The rules and databases used in designing partial journal bearings for max. W and min. f are divided into six more stages as follows:

6.2 STAGE 1

The goal of stage 1 is to obtain all partial feasible solutions which fulfils the outlet temperature constraint, i.e., $T_{out} \leq 120°C$. To achieve this goal the following sets of rules and database are used:

1. a knowledge base consisting of all the performance variables: S, h_0/C, T_{var}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, θ_{Pmax}, θ_{P0}, and θ_A in the form as presented in Tables 6.1 - 6.3 ,
2. a set of rules to determine the lubricant's operating and outlet temperatures,
3. a set of rules to check whether or not the outlet temperature obtained in each case is below 120°C (a condition discussed in Sec. 3.3.1).

If no partial solution within the desired outlet temperature limit is obtained this condition may be modified to perform failure handling [34]. In this case each partial solution will have the bearing arc β, L/D ratio, length, diameter, Sommerfeld or bearing characteristic number: S, minimum film thickness variable: h_0/C, temperature-rise variable: T_{var}, coefficient of friction variable: (R/C)f, attitude angle (position of the minimum film thickness): Φ, flow variable:

Table 6.1. Performance Variables of 180° Partial Journal Bearings* [24].

L/D	S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}	$\theta_{p_{max}}$	θ_{p0}	θ_A
Corresponding to max. W condition											
1/4	0.574	0.276	52.95	10.95	35.95	4.14	0.784	0.277	12.69	44.5	53.87
1/2	0.364	0.418	27.41	5.82	46.18	3.93	0.651	0.372	14.06	60.25	43.83
0.6	0.306	0.45	22.5	4.81	48.03	3.81	0.622	0.381	13.83	64.51	41.77
1.0	0.2	0.52	14.0	3.0	51.0	3.41	0.49	0.44	12.4	74.0	38.5
1.5	0.157	0.558	10.82	2.28	51.94	3.12	0.373	0.5	11.07	79.14	37.56
2.0	0.14	0.578	9.6	1.99	52.23	2.95	0.299	0.542	10.22	81.8	37.32
Corresponding to min. f condition											
1/4	0.01	0.03	6.46	0.877	12.28	3.29	0.961	0.11	6.3	17.5	77.68
1/2	0.13	0.233	16.79	3.01	33.54	3.62	0.783	0.288	12.43	46.4	56.7
0.6	0.143	0.302	15.92	2.93	38.62	3.58	0.726	0.321	13.16	53.78	51.45
1.0	0.15	0.44	13.0	2.5	47	3.3	0.54	0.42	13.2	69.0	42.8
1.5	0.141	0.503	11.17	2.16	49.62	3.037	0.4	0.492	12.24	76.44	40.14
2.0	0.133	0.531	10.21	1.966	50.38	2.87	0.316	0.536	11.456	80.03	39.39

*Values at L/D = 0.6, 1.5, and 2.0 are obtained using Eq. (4.1) given in [23].

Table 6.2. Performance Variables of 120° Partial Journal Bearings* [24].

L/D	S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}	θ_{Pmax}	θ_{P0}	θ_A
Corresponding to max. W condition											
1/4	0.519	0.259	52.46	7.96	31.93	2.83	0.707	0.262	9.69	39.49	88.03
1/2	0.331	0.382	26.28	4.23	37.56	2.74	0.544	0.329	8.18	52.24	80.68
0.6	0.288	0.408	22.16	3.6	37.67	2.647	0.493	0.333	7.355	55.29	80.97
1.0	0.21	0.46	15.3	2.5	37.5	2.4	0.35	0.368	5.6	60.0	82.0
1.5	0.18	0.485	12.824	2.07	37.29	2.248	0.254	0.407	4.755	61.18	82.54
2.0	0.168	0.497	11.88	1.895	37.17	2.165	0.198	0.433	4.36	61.37	82.78
Corresponding to min. f condition											
1/4	0.038	0.06	17.73	1.603	15.73	1.913	0.895	0.14	6.486	21.47	104.3
1/2	0.199	0.279	22.16	3.063	32.16	2.425	0.623	0.291	8.64	45.59	87.85
0.6	0.189	0.32	19.77	2.79	33.62	2.39	0.556	0.307	8.11	51.25	86.39
1.0	0.16	0.4	14.7	2.14	35.5	2.23	0.384	0.36	6.6	60.0	84.5
1.5	0.146	0.437	12.56	1.843	35.94	2.11	0.275	0.402	5.693	62.19	84.06
2.0	0.141	0.454	11.705	1.718	36.04	2.04	0.214	0.429	5.217	62.55	83.96

*Values at L/D = 0.6, 1.5, and 2.0 are obtained using Eq. (4.1) given in [23].

Table 6.3. Performance Variables of 60° Partial Journal Bearings* [24].

L/D	S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}	$\theta_{p_{max}}$	θ_{p0}	θ_A
Corresponding to max. W condition											
1/4	0.209	0.15	47.95	3.14	19.28	1.09	0.556	0.177	3.75	26.85	130.7
1/2	0.149	0.2	29.4	2.02	19.57	1.05	0.35	0.201	2.2	30.0	130.4
0.6	0.141	0.21	26.97	1.86	19.37	1.03	0.3	0.204	2.01	30.05	130.5
1.0	0.13	0.23	23	1.6	19	0.98	0.19	0.222	1.9	30.0	130.7
1.5	0.129	0.238	21.53	1.51	18.87	0.952	0.13	0.239	1.77	29.96	130.8
2.0	0.1294	0.242	20.93	1.473	18.83	0.938	0.098	0.25	1.702	29.95	130.9
Corresponding to min. f condition											
1/4	0.084	0.1	41.0	2.01	16.87	0.894	0.62	0.159	4.3	23.7	133.1
1/2	0.106	0.16	28.25	1.644	18.11	0.896	0.396	0.19	2.84	28.16	131.9
0.6	0.112	0.182	26.26	1.62	18.37	0.915	0.33	0.196	2.54	28.905	131.5
1.0	0.12	0.22	22.8	1.55	18.7	0.93	0.193	0.219	2.0	30.0	131
1.5	0.121	0.231	21.41	1.49	18.71	0.916	0.125	0.237	1.785	30.26	131.0
2.0	0.12	0.235	20.822	1.453	18.66	0.902	0.092	0.248	1.697	30.3	131.1

*Values at L/D = 0.6, 1.5, and 2.0 are obtained using Eq. (4.1) given in [23].

variable: Q/RCNL, flow ratio: Q_s/Q, pressure ratio: P/p_{max}, position of maximum film pressure: θ_{Pmax}, position at which film terminates: θ_{P0}, position at which film starts: θ_A, T_{cp}, and T_{in}. At the end of this stage all partial solutions are written onto an internal database. All databases of stage 1 are then retracted except the newly written database of all partial solutions of stage 1.

In stage 1, the discriminating skills of a designer are used to discriminate between appropriate and inappropriate solutions. At this stage, recombining skills of a designer are not used. The number of partial solutions of stage 1 is N_1.

6.3 STAGE 2

The final goal of stage 2 is to get the oil-grade and determine the viscosity for each of the partial design solutions of stage 1. The group of rules and the database used in this stage are given below:

1. a set of rules to get the oil-grade selected by the user (SAE 10, 20, 30, 40, 50, 60, 70, or all seven),
2. a set of rules to determine the viscosity of the oil using the equations described in Sec. 4.1,
3. a database consisting of the coefficients of the equations for determining the viscosity (c.f. Sec. 4.1), which is in the form of a table.

At the end of stage 2 there will be the same number of solutions as that of stage 1 if only one oil-grade is tried or seven times the number of solutions of stage 1 if all seven oil-grades are tried. At this stage a failure handler is not used. Each partial solution of stage 2 consists of the arc β, SAE oil-grade, L/D

ratio, length, diameter, S, h_0/C, T_{wr}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{p_{max}}$, θ_{pc}, θ_A, T_{op}, T_{in}, and μ.

At the end of stage 2 all partial solutions are written onto an internal database (on the blackboard). The database of stage 2 containing coefficients of the equations to determine viscosity and the database consisting of all partial solutions of stage 1 are erased. In stage 2, the recombining skills of a designer are used. The number of partial solutions of stage 2 is N_T.

6.4 STAGE 3

The final goal of this stage is to determine the clearance and check whether this is in the prescribed range for each partial solution of stage 2. The group of rules and the database used in this stage are given below:

1. a set of rules to determine the clearance using $S = (R/C)^2 \mu N/P$,
2. a set of rules to check whether the calculated clearance in each case is within the prescribed range,
3. a database consisting of clearance limits depending on the industrial application in the form of a table (c.f. Table 3.1).

If no partial solution within the desired clearance limit for manufacturability is obtained, this condition may be modified to handle this failure [34]. Each partial solution of stage 3 will have the arc β, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{wr}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{p_{max}}$, θ_{pc}, θ_A, T_{op}, T_{in}, μ, C, and corresponding minimum limit of clearance C_{min}. At the end of stage 3 all partial solutions are written onto an internal database. The database consisting of all partial solutions of stage 2 is

erased. The database of stage 3 is also erased.

For each partial solution of stage 2 there may be either no solution or one solution at stage 3. A failure handler is used if there is not even a single solution. That is the industrial application is changed or clearance limits are modified or the maximum diameter limit of 200 mm is increased. In stage 3 only discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_3 will be less than or equal to the number of partial solutions in the previous stage N_2.

6.5 STAGE 4

The final goal of this stage is to determine whether or not each of the partial solutions of stage 3 satisfies the minimum film thickness criterion given by Juvinall [68] (c.f. Sec. 3.3.3). That is for a factor of safety of 2, $h_0 \geq 0.005 + 0.00004D$ (h_0 and D *are in mm*). The group of rules and the database used in this stage are given below:

1. a database consisting of S with a safety factor of 2 and corresponding h_0/C in the form of Table 6.4; the values of h_0/C for L/D ratio of 0.25, 0.5, and 1 are obtained from Raimondi and Boyd [24] (using interpolation, if necessary); the values of h_0/C for L/D ratio of 0.6, 1.5, and 2, are found using Eq. (4.1),
2. a set of rules to determine the value of $h_{0_{F.S.=2}}$, and check whether it is in the acceptable limit,
3. a set of rules to determine actual h_0, coefficient of friction f, and torque T (=fWR).

Table 6.4. Values* of h_0/C for Partial Journal Bearings with a Factor of Safety of 2 [24].

	$\beta = 180°$		$\beta = 120°$		$\beta = 60°$	
L/D	S	h_0/C	S	h_0/C	S	h_0/C
Corresponding to max. W						
1/4	0.287	0.2059	0.2595	0.1185	0.1045	0.108
1/2	0.182	0.2785	0.1655	0.253	0.0745	0.13
0.6	0.153	0.3012	0.144	0.2946	0.07	0.1829
1.0	0.1	0.348	0.105	0.36	0.065	0.26
1.5	0.0785	0.3704	0.09	0.3763	0.0644	0.2686
2.0	0.07	0.381	0.084	0.3786	0.065	0.2619
Corresponding to min. f						
1/4	0.0052	0.015	0.0189	0.0392	0.0422	0.05975
1/2	0.0648	0.155	0.09935	0.2016	0.053	0.11
0.6	0.0715	0.2032	0.0945	0.2383	0.056	0.1653
1.0	0.075	0.29	0.08	0.3	0.06	0.244
1.5	0.0705	0.3218	0.073	0.3206	0.0605	0.2514
2.0	0.0665	0.3333	0.0705	0.3273	0.06	0.2437

*Values at L/D = 0.6, 1.5, and 2.0 are obtained using Eq. (4.1) given in [23].

Each design solution of stage 4 will have the arc β, unit load P, speed N, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{var}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{P_{max}}$, θ_{P0}, θ_A, T_{op}, T_{in}, μ, C, corresponding minimum limit of clearance C_{min}, $h_{0_{F.S.=2}}$, actual h_0, and torque. A failure handler may be used if there is no design solution at the end of this stage. That is, this constraint may be modified.

At the end of stage 4 all partial solutions are written onto an internal database and at the same time the database consisting of all partial solutions of stage 3 is erased. The database of stage 4 is also retracted. In stage 4 only discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_4 will be less than or equal to the number of partial solutions in the previous stage N_3.

6.6 STAGE 5

The final goal of this stage is to determine the utility value of all the design solutions of stage 4 (c.f. Sec. 3.4). The group of rules and the databases used in this stage are given below:

1. a set of rules to get the decision criterion entered by the user in the input stage (maximum load or minimum friction),
2. a knowledge base consisting of the utility values and the weighting factors for both the decision criteria corresponding to all the five decision variables (c.f. Tables 3.2 to 3.7),
3. a set of rules to determine the utility value for each design solution of stage 4 using Eq. (4.3).

Each design solution of stage 5 will have the arc β, unit load P, speed N, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{nor}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{P_{max}}$, θ_{P0}, θ_A, T_{op}, T_{in}, μ, C, corresponding minimum limit of clearance C_{min}, $h_{0_{F.S.=2}}$, actual h_0, torque T , and the utility value U. This stage is a simple procedural stage. Neither the discriminating nor the recombining skills of a designer are used at this stage. Therefore, the number of design solutions of this stage will be the same as that of stage 4, i.e. $N_5 = N_4$.

At the end of stage 5 all partial solutions are written onto an internal database. The design solutions of stage 4 are erased. The databases of this stage are also erased. The user can modify the databases in Tables 3.2 to 3.7, if required.

6.7 STAGE 6

This stage consists of two parts:

6.7.1 Stage 6.1 or Decision Making Stage

The goal of this stage is to identify the design solution of the previous stage which has the highest utility factor U_{max}. In addition all the design solutions of the previous stage for which the utility factors are at least ninety five percent (95%) of the U_{max} are identified.

The group of rules and the databases used in this stage are given below:

1. a set of rules to identify the design solution which has the highest utility value U_{max},

2. a set of rules to identify the design solutions which have the utility values of at least 95% of the U_{max}.

In this stage only the discriminating skills of a designer are used. Therefore the number of solutions at the end of this stage N_6 is less than or equal to the number of solutions of the previous stage N_5.

Each design solution of stage 6.1 will have the arc β, unit load P, speed N, load W, oil-grade, L/D ratio, length, diameter, S, h_0/C, T_{var}, (R/C)f, Φ, Q/RCNL, Q_s/Q, P/p_{max}, $\theta_{P_{max}}$, θ_{P0}, θ_A, T_{op}, T_{in}, μ, C, corresponding minimum limit of clearance C_{min}, $h_{0_{F.S.=2}}$, actual h_0, torque T , and the utility value U. At the end of stage 6.1 all partial solutions are written onto an internal database.

6.7.2 Stage 6.2 or Output Stage

The goal of this stage is to file all the finally selected design solutions. Each design solution for which the utility value is at least ninety five percent (95%) of the highest utility value (U_{max}) is printed in the 'results.out' file for the perusal of the user.

The group of rules and the database used in this stage are given below:

1. a set of rules to calculate eccentricity ratio ε(= $1-h_0/C$), dimensionless speed ω_s, eccentricity e, coefficient of friction f, power loss H (= $2\pi NT$), oil flow Q, side leakage Q_s, and maximum pressure p_{max},
2. a knowledge base containing the L/D ratio, optimization condition: maximum load or minimum friction, oil-grade, and bearing arc β,
3. a set of rules to print the finally selected design solutions in the 'results.out'

file.

In this stage (stage 6.2) neither the discriminating nor the recombining skills of a designer are used. Therefore the number of solutions at the end of this stage is the same as that of stage 6.1, N_6. The design solutions of stage 5 are erased. At the end of this stage all the solutions are printed in the 'results.out' file. The design solutions of all the previous stages including stage 6.1 are erased and the database used in this stage (stage 6.2) is also erased.

The 'results.out' file will have the functional and prescriptive specifications (c.f. Sec. 1.4) as the load W, speed N, unit load P, bearing industrial application, lubricant inlet temperature, decision criterion, L/D ratio, optimization condition, oil-grade, bearing arc β, length L, diameter D, T_{op}, μ, S, C, h_0, ε, eccentricity e, Φ, coefficient of friction f, torque T, power loss H, total oil supplied Q, side leakage Q_s, maximum film pressure p_{max}, position of maximum film pressure $\theta_{p_{max}}$, terminating position of film θ_{p0}, starting position of film θ_A, dimensionless velocity ω_s, and the utility value U.

6.8 EXAMPLE

To illustrate the use of the present 'Expert System' it is used to design partial journal bearings for 1) maximum load and 2) minimum friction condition. This example is similar to the one used by Orthwein[19]. The input specifications for this example are as follows:

Bearing load = 32 kN

Speed of the journal = 3600 rpm

Bearing unit load = 1.38 MPa

Bearing arc: all three arcs

Bearing industrial application = General machine practice (rotating motion)

Lubricant inlet temperature = 43.3 °C

L/D ratio: all six ratios

Oil: all seven grades of oil.

In this case the number of partial design solutions at the end of stage 1, N_1 is 36. The number of partial design solutions at the end of stage 2, N_2 is 252. At the end of stage 3 there are thirty nine (N_3=39) solutions. The number of design solutions at the end of stage 4, N_4 is 39.

In the above case all the design alternatives satisfy temperature-rise check, therefore at the end of stage 1 and 2, respectively, there are 36 and 252 solutions. Out of these 252 solutions, 213 solutions are discarded because they do not fulfil the condition for clearance. Therefore, at the end of stage 3 the number of solutions are 39. All these satisfy minimum film thickness requirement. Therefore at the end of stage 4 there are 39 solutions and consequently at the end of stage 5, all 39 solutions have respective utility values based on the decision-criterion. Thus at the end of stage 5 there are thirty nine (N_5=39) solutions. The number of design solutions at the end of decision making stage N_6 will depend on the decision making criterion used.

6.8.1 Based on Maximum Load

On the basis of maximum load the 'Expert System' has found twenty solutions, having utility values above 95% of the U_{max}, i.e., $N_6 = 20$ and filed all the twenty design alternatives. Out of these 20 design alternatives the solution with L/D = 1.5, optimization condition as maximum load, SAE 10 oil, and $\beta = 180°$ has the highest utility value = 3.4 and 95% of this value is 3.23. It is noticed, that none of the solutions for a 60° partial bearing has a utility factor above 95% of the U_{max}. All the twenty design solutions are presented in Sec. B.1 and only the first two solutions are displayed on the following pages owing to the lack of space.

Partial Bearing Design Specifications

The load on the bearing, W = 32.00 kN

The speed of the journal, N = 60.000 rps

The bearing unit load, P = 1.380 MPa

The bearing-application for clearance is No. 3. General machine practice - rotating motion

The inlet temperature, Tin = 43.300 deg C

The decision-criterion for utility is No. 1. Based on maximum load

Design Alternatives Are

1. L/D = 1.0

Optimization Condition: The Maximum Load

Design Based on SAE 10 Oil

180 deg. Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 49.67 deg. C

Viscosity of the Lubricant, Mu = 20.32 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.200

Radial Clearance, C = 0.160 mm

Minimum film thickness, hmin = 0.083 mm

Eccentricity Ratio, e/C = 0.480

Eccentricity, e = 0.077 mm

Position of minimum film thickness = 51.00 degree

Coefficient of friction, f = 0.0063

Torque required = 15.333 N.m

Power Lost = 5782.89 W

Total Oil Supplied = 377511.7 cubic mm per sec

Side Leakage = 184980.7 cubic mm per sec

Maximum Film Pressure = 3.136 MPa

Position of Maximum Film Pressure = 12.40 degree

Terminating Position of Film = 74.00 degree

Starting Position of Film = 38.50 degree

The dimensionless velocity, ws = 1.52

The Utility value = 3.31

2. L/D = 1.5

Optimization Condition: The Maximum Load

Design Based on SAE 10 Oil

180 deg. Partial Bearing

Length, L = 187 mm; Diameter, D = 124.67 mm

Operating Temperature, Top = 48.23 deg. C

Viscosity of the Lubricant, Mu = 21.28 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.157

Radial Clearance, C = 0.151 mm

Minimum film thickness, hmin = 0.084 mm

Eccentricity Ratio, e/C = 0.442; Eccentricity, e = 0.067 mm

Position of minimum film thickness = 51.94 degree

Coefficient of friction, f = 0.0055

Torque required = 11.041 N.m; Power Lost = 4163.93 W

Total Oil Supplied = 330203.0 cubic mm per sec

Side Leakage = 123165.7 cubic mm per sec

Maximum Film Pressure = 2.760 MPa

Position of Maximum Film Pressure = 11.07 degree

Terminating Position of Film = 79.14 degree

Starting Position of Film = 37.56 degree

The dimensionless velocity, ws = 1.48

The Utility value = 3.40

6.8.2 Based on Minimum Friction

On the basis of minimum friction the 'Expert System' has found $N_6 = 28$ and filed all the twenty eight design alternatives. Out of these 28 design alternatives the solution with L/D = 1.5, optimization condition as minimum friction, SAE 10 oil, and $\beta = 120°$ has the best utility value of 4.5 and its 95% is 4.275. Here, all the 28 selected design solutions belong to all three kinds of partial bearings, i.e., 180°, 120°, and 60°. The print-out of the 'results.out' file is shown in Sec. B.2. Because of the space limitation, only the 10th and 11th design solutions are presented on the continuing pages.

The load on the bearing, W = 32.00 kN

The speed of the journal, N = 60.000 rps

The bearing unit load, P = 1.380 MPa

The bearing-application for clearance is No. 3. General machine practice - rotating motion

The inlet temperature, Tin = 43.300 deg C

The decision-criterion for utility is No. 2. Based on minimum friction

Design Alternatives Are

10. L/D = 1.0

Optimization Condition: The Minimum Friction

Design Based on SAE 10 Oil

120 deg. Partial Bearing

Length, L = 152 mm; Diameter, D = 152.00 mm

Operating Temperature, Top = 49.99 deg. C

Viscosity of the Lubricant, Mu = 20.13 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.160

Radial Clearance, C = 0.178 mm

Minimum film thickness, hmin = 0.071 mm

Eccentricity Ratio, e/C = 0.600; Eccentricity, e = 0.107 mm

Position of minimum film thickness = 35.50 degree

Coefficient of friction, f = 0.0050

Torque required = 12.173 N.m; Power Lost = 4590.82 W

Total Oil Supplied = 274747.4 cubic mm per sec

Side Leakage = 105503.0 cubic mm per sec

Maximum Film Pressure = 3.844 MPa

Position of Maximum Film Pressure = 6.60 degree

Terminating Position of Film = 60.00 degree

Starting Position of Film = 84.50 degree

The dimensionless velocity, ws = 1.61

The Utility value = 4.38

11. L/D = 1.5

Optimization Condition: The Minimum Friction

Design Based on SAE 10 Oil

120 deg. Partial Bearing

Length, L = 187 mm; Diameter, D = 124.67 mm

Operating Temperature, Top = 49.02 deg. C

Viscosity of the Lubricant, Mu = 20.73 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.146

Radial Clearance, C = 0.155 mm

Minimum film thickness, hmin = 0.068 mm

Eccentricity Ratio, e/C = 0.563; Eccentricity, e = 0.087 mm

Position of minimum film thickness = 35.94 degree

Coefficient of friction, f = 0.0046

Torque required = 9.134 N.m; Power Lost = 3444.76 W

Total Oil Supplied = 228546.1 cubic mm per sec

Side Leakage = 62850.2 cubic mm per sec

Maximum Film Pressure = 3.433 MPa

Position of Maximum Film Pressure = 5.69 degree

Terminating Position of Film = 62.19 degree

Starting Position of Film = 84.06 degree

The dimensionless velocity, ws = 1.50

The Utility value = 4.50

Partial journal bearings have limited use because they are difficult to manufacture and suitable only for unidirectional loading. Therefore previous researchers have paid little attention to them. Orthwein [19] has solved this example but for a given clearance. That is why the present results can not be compared with his results. In the next chapter, design on the basis of optimal clearance is discussed where the program starts with the selection of an appropriate clearance recommended by Keith, Jr. [15-16]. There by choosing the clearance used by Orthwein [19], the results are compared. In addition, working of this program is verified by selecting the clearance obtained for the max. W and the min. f conditions (c.f. Sec. 7.10).

CHAPTER 7

DESIGN OF PARTIAL JOURNAL BEARINGS FOR OPTIMAL CLEARANCE

The objective knowledge required for designing partial journal bearings of 180°, 120°, and 60° arcs for optimal clearance is collected from the same sources as detailed in the previous chapter. The viscosity-temperature relationships used are the same as discussed in Sec. 4.1. Since values of Sommerfeld number S are not known in the beginning, therefore the structure of the program and the iterative method of determining temperature-rise are similar to those used for designing full journal bearings for optimal clearance in chapter 5. Clearance is selected from Table 5.1. When the viscosity and the bearing characteristic number are found with the help of an iterative method, other performance variables may be obtained using the equations listed below.

7.1 PERFORMANCE VARIABLES

The equations for performance variables such as h_0/C, T_{var}, coefficient of friction variable (R/C)f, Φ, flow variable Q/RCNL, flow ratio Q_s/Q, and pressure ratio P/p_{max} of partial journal bearings of 60, 120, and 180 degree arcs are obtained using 'SAS' program by curve fitting tabulated data given in Raimondi and Boyd [24] for L/D ratios of 0.25, 0.5, and 1.0. Each data-set has seven points. The accuracy of these equations depend on the values of corresponding r^2 [79]. For the other L/D ratios of interest (0.6, 1.5, and 2.0), the corresponding data are obtained with the help of Eq. (4.1). Then the best

fit equations are obtained using 'SAS' package.

For all the six L/D ratios and three bearing arcs, the values of the performance variables have been presented in Tables 7.1 to 7.18. The equations used to obtain these performance variables have been summarized below.

7.1.1 60 Degree Partial Bearings

7.1.1.1 L/D = 0.25

$$(h_0/C) = 0.3464 + 0.322 \times \log_{10}(S) + 0.0586 \times [\log_{10}(S)]^2 - 0.0124 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9977$

$$(T_{max}) = 49.0801 + 40.4606 \times \log_{10}(S) + 90.4086 \times [\log_{10}(S)]^2 + 39.0326 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9828$

$$[(R/C)f] = 2.0498 + 9.6363 \times \log_{10}(S) + 24.6236 \times [\log_{10}(S)]^2 + 10.4277 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9850$

$$(\Phi) = 27.0561 + 14.6321 \times \log_{10}(S) + 6.60329 \times [\log_{10}(S)]^2 + 1.7128 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9991$

$$(Q/RCNL) = 1.8117 + 1.0383 \times \log_{10}(S) + 0.0352 \times [\log_{10}(S)]^2 - 0.0801 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9976$

Table 7.1. Performance Variables of 60° Partial Journal Bearings at L/D = 0.25 [24].

S	h_0/C	T_{max}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
35.8	0.925	499.0	121.0	71.55	3.16	0.067	0.251
16.0	0.824	260.0	58.7	58.51	3.04	0.131	0.249
5.2	0.607	136.0	24.5	41.01	2.57	0.236	0.242
1.65	0.4	86.1	11.2	30.14	1.98	0.346	0.228
0.333	0.2	54.9	4.27	21.7	1.3	0.496	0.195
0.084	0.1	41.0	2.01	16.87	0.894	0.620	0.159
0.011	0.03	29.1	0.713	10.81	0.507	0.786	0.107

Table 7.2. Performance Variables of 60° Partial Journal Bearings at L/D = 0.5 [24].

S	h_0/C	T_{max}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
14.2	0.9223	201.0	48.6	69.0	3.11	0.049	0.239
6.47	0.8152	109.0	24.2	52.6	2.91	0.088	0.239
2.14	0.6039	59.4	10.3	37.0	2.38	0.16	0.233
0.695	0.4	40.3	4.93	26.98	1.74	0.236	0.225
0.149	0.2	29.4	2.02	19.57	1.05	0.35	0.201
0.042	0.1	26.5	1.08	15.91	0.664	0.464	0.172
0.007	0.03	27.8	0.49	10.85	0.329	0.65	0.122

Table 7.3. Performance Variables of 60° Partial Journal Bearings at L/D = 0.6.

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
11.895	0.9223	169.53	40.93	68.694	3.098	0.042	0.238
5.468	0.8152	92.745	20.52	52.039	2.881	0.076	0.238
1.825	0.6039	51.455	9.16	35.964	2.328	0.137	0.233
0.599	0.4	35.583	4.289	26.13	1.681	0.202	0.225
0.13	0.2	26.686	1.786	19.143	0.995	0.306	0.204
0.038	0.1	25.11	0.942	15.707	0.617	0.415	0.176
0.007	0.03	28.11	0.465	10.862	0.296	0.61	0.127

Table 7.4. Performance Variables of 60° Partial Journal Bearings at L/D = 1.0 [24].

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
8.52	0.9212	121.0	29.1	67.92	3.07	0.027	0.252
3.92	0.8133	67.4	14.8	50.96	2.82	0.048	0.251
1.34	0.601	39.1	6.61	33.99	2.22	0.085	0.247
0.45	0.4	28.2	3.29	24.56	1.56	0.127	0.239
0.101	0.2	22.5	1.42	18.33	0.883	0.2	0.22
0.031	0.1	23.2	0.822	15.33	0.519	0.287	0.192
0.006	0.03	30.5	0.422	10.88	0.226	0.465	0.139

Table 7.5. Performance Variables of 60° Partial Journal Bearings at L/D = 1.5.

S	h_0/C	T_{ser}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
7.310	0.9212	103.68	24.883	67.393	3.053	0.018	0.27
3.364	0.8133	58.204	12.731	50.37	2.789	0.033	0.269
1.164	0.601	34.556	5.541	33.143	2.167	0.057	0.265
0.396	0.4	25.433	2.919	23.931	1.502	0.086	0.256
0.09	0.2	21.011	1.285	17.968	0.827	0.139	0.236
0.028	0.1	22.679	0.771	15.169	0.47	0.205	0.205
0.006	0.03	32.872	0.404	10.885	0.19	0.349	0.146

Table 7.6. Performance Variables of 60° Partial Journal Bearings at L/D = 2.0.

S	h_0/C	T_{ser}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
6.817	0.92	96.708	23.186	67.081	3.043	0.014	0.283
3.137	0.81	54.452	11.887	50.047	2.774	0.025	0.281
1.091	0.6	32.662	5.133	32.775	2.141	0.043	0.277
0.373	0.4	24.259	2.763	23.675	1.475	0.065	0.267
0.086	0.2	20.414	1.228	17.803	0.8	0.106	0.246
0.027	0.1	22.527	0.746	15.099	0.445	0.159	0.213
0.005	0.03	34.416	0.395	10.885	0.171	0.277	0.150

Table 7.7. Performance Variables of 120° Partial Journal Bearings at L/D = 0.25 [24].

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
18.4	0.9	502.0	124.0	76.97	3.34	0.143	0.456
8.45	0.8	254.0	60.4	65.97	3.44	0.26	0.438
3.04	0.6	125.0	26.6	51.23	3.42	0.442	0.389
1.12	0.4	75.8	13.5	40.42	3.2	0.599	0.321
0.268	0.2	42.7	5.65	28.38	2.67	0.753	0.237
0.074	0.1	25.9	2.63	20.55	2.21	0.846	0.178
0.01	0.03	11.6	0.832	12.11	1.69	0.931	0.112

Table 7.8. Performance Variables of 120° Partial Journal Bearings at L/D = 0.5 [24].

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
5.42	0.9	149.0	36.6	74.99	3.29	0.124	0.431
2.51	0.8	77.2	18.1	63.38	3.32	0.225	0.424
0.914	0.6	40.5	8.2	48.07	3.15	0.386	0.389
0.354	0.4	27.0	4.43	38.5	2.8	0.53	0.336
0.097	0.2	19.0	2.17	28.02	2.18	0.684	0.261
0.032	0.1	15.1	1.24	21.02	1.7	0.787	0.203
0.006	0.03	10.6	0.55	13.0	1.19	0.899	0.136

Table 7.9. Performance Variables of 120° Partial Journal Bearings at L/D = 0.6.

S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
4.097	0.9	112.91	27.684	74.391	3.268	0.116	0.417
1.905	0.8	59.238	13.803	61.822	3.264	0.21	0.411
0.7	0.6	31.85	6.344	46.84	3.039	0.361	0.381
0.271	0.4	22.144	3.518	37.617	2.638	0.5	0.334
0.08	0.2	16.748	1.813	27.834	2.001	0.663	0.265
0.028	0.1	14.242	1.09	21.112	1.522	0.781	0.209
0.006	0.03	9.993	0.517	13.17	1.037	0.919	0.144

Table 7.10. Performance Variables of 120° Partial Journal Bearings at L/D = 1.0 [24].

S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
2.14	0.9	59.5	14.5	72.43	3.2	0.088	0.427
1.01	0.8	32.6	7.44	58.25	3.11	0.157	0.42
0.385	0.6	19.0	3.6	43.98	2.75	0.272	0.396
0.162	0.4	15.0	2.16	35.65	2.24	0.384	0.356
0.053	0.2	13.9	1.27	27.42	1.57	0.535	0.29
0.021	0.1	14.4	0.855	21.29	1.11	0.657	0.233
0.005	0.03	14.0	0.461	13.49	0.694	0.812	0.162

Table 7.11. Performance Variables of 120° Partial Journal Bearings at L/D = 1.5.

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
1.499	0.9	42.018	10.189	70.949	3.152	0.065	0.461
0.717	0.8	23.796	5.346	56.349	3.009	0.117	0.453
0.282	0.6	14.753	2.695	42.394	2.569	0.203	0.429
0.126	0.4	12.658	1.704	34.629	2.006	0.289	0.388
0.044	0.2	13.348	1.08	27.206	1.32	0.411	0.316
0.018	0.1	15.653	0.768	21.372	0.881	0.514	0.252
0.005	0.03	19.859	0.438	13.632	0.508	0.647	0.173

Table 7.12. Performance Variables of 120° Partial Journal Bearings at L/D = 2.0.

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
1.262	0.9	35.548	8.594	70.057	3.123	0.052	0.487
0.608	0.8	20.495	4.564	55.394	2.951	0.092	0.478
0.244	0.6	13.164	2.356	41.571	2.47	0.16	0.453
0.112	0.4	11.783	1.529	34.123	1.88	0.229	0.41
0.04	0.2	13.3	1.005	27.1	1.188	0.33	0.333
0.017	0.1	16.625	0.732	21.41	0.761	0.416	0.264
0.004	0.03	24.025	0.427	13.697	0.412	0.528	0.179

Table 7.13. Performance Variables of 180° Partial Journal Bearings at L/D = 0.25 [24].

S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
16.3	0.9	653.0	163.0	81.4	3.44	0.176	0.513
7.6	0.8	320.0	79.4	73.7	3.71	0.32	0.489
2.84	0.6	146.0	35.1	58.99	4.11	0.534	0.417
1.08	0.4	79.8	17.6	44.96	4.25	0.698	0.336
0.263	0.2	36.5	6.88	30.43	4.07	0.837	0.241
0.074	0.1	18.4	2.99	21.43	3.72	0.905	0.18
0.01	0.03	6.46	0.877	12.28	3.29	0.961	0.11

Table 7.14. Performance Variables of 180° Partial Journal Bearings at L/D = 0.5 [24].

S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
4.38	0.9	177.0	44.0	79.97	3.41	0.167	0.518
2.06	0.8	87.8	21.6	72.14	3.64	0.302	0.499
0.794	0.6	42.7	9.96	58.01	3.93	0.506	0.438
0.321	0.4	25.9	5.41	45.01	3.93	0.665	0.365
0.092	0.2	15.0	2.54	31.29	3.56	0.806	0.273
0.031	0.1	9.8	1.38	22.8	3.17	0.886	0.208
0.006	0.03	5.3	0.581	13.63	2.62	0.951	0.132

Table 7.15. Performance Variables of 180° Partial Journal Bearings at L/D = 0.6.

S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
3.17	0.9	128.322	31.866	79.771	3.403	0.166	0.504
1.496	0.8	64.22	15.734	71.43	3.616	0.3	0.488
0.585	0.6	32.077	7.39	58.882	3.849	0.503	0.433
0.243	0.4	20.349	4.151	44.984	3.787	0.667	0.366
0.074	0.2	12.739	2.085	31.619	3.331	0.821	0.279
0.027	0.1	8.555	1.202	23.221	2.897	0.914	0.216
0.006	0.03	3.491	0.545	13.94	2.331	0.997	0.139

Table 7.16. Performance Variables of 180° Partial Journal Bearings at L/D = 1.0 [24].

S	h_0/C	T_{var}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
1.4	0.9	57.0	14.1	78.5	3.34	0.139	0.525
0.67	0.8	29.7	7.15	68.93	3.46	0.252	0.513
0.278	0.6	16.5	3.61	58.86	3.49	0.425	0.466
0.128	0.4	12.4	2.28	44.67	3.25	0.572	0.403
0.046	0.2	10.4	1.39	32.33	2.63	0.721	0.313
0.019	0.1	9.13	0.921	24.14	2.14	0.818	0.244
0.005	0.03	6.96	0.483	14.57	1.6	0.915	0.157

Table 7.17. Performance Variables of 180° Partial Journal Bearings at L/D = 1.5.

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
0.837	0.9	34.302	8.448	77.226	3.273	0.109	0.573
0.407	0.8	18.689	4.414	66.985	3.308	0.197	0.561
0.179	0.6	11.551	2.393	57.376	3.177	0.333	0.514
0.09	0.4	10.049	1.664	44.329	2.816	0.45	0.447
0.037	0.2	10.556	1.146	32.688	2.131	0.573	0.347
0.017	0.1	11.596	0.817	24.619	1.643	0.655	0.267
0.004	0.03	14.12	0.457	14.875	1.158	0.739	0.167

Table 7.18. Performance Variables of 180° Partial Journal Bearings at L/D = 2.0.

S	h_0/C	T_{wr}	(R/C)f	Φ	Q/RCNL	Q_s/Q	P/p_{max}
0.635	0.9	26.183	6.424	76.386	3.228	0.088	0.609
0.313	0.8	14.731	3.431	65.805	3.208	0.159	0.596
0.143	0.6	9.787	1.952	56.137	2.98	0.27	0.548
0.076	0.4	9.289	1.434	44.101	2.549	0.366	0.477
0.033	0.2	11.016	1.05	32.862	1.838	0.468	0.369
0.015	0.1	13.515	0.774	24.86	1.362	0.536	0.281
0.004	0.03	19.469	0.446	15.022	0.917	0.608	0.173

$$(Q_J/Q) = 0.3917 - 0.2192 \times \log_{10}(S) - 0.0008 \times [\log_{10}(S)]^2 + 0.0043 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.2176 + 0.0442 \times \log_{10}(S) - 0.0112 \times [\log_{10}(S)]^2 - 0.0024 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

7.1.1.2 L/D = 0.5

$$(h_0/C) = 0.4733 + 0.3712 \times \log_{10}(S) + 0.0436 \times [\log_{10}(S)]^2 - 0.0157 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9985$

$$(T_{max}) = 37.6091 + 47.6978 \times \log_{10}(S) + 56.1556 \times [\log_{10}(S)]^2 + 16.9990 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9845$

$$[(R/C)/f] = 4.4199 + 13.0280 \times \log_{10}(S) + 15.1360 \times [\log_{10}(S)]^2 + 4.6692 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9875$

$$(\Phi) = 29.5413 + 18.957 \times \log_{10}(S) + 10.1947 \times [\log_{10}(S)]^2 + 2.5239 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

$$(Q/RCNL) = 1.9788 + 1.1453 \times \log_{10}(S) - 0.0110 \times [\log_{10}(S)]^2 - 0.0884 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9983$

$$(Q_J/Q) = 0.2088 - 0.1570 \times \log_{10}(S) + 0.0172 \times [\log_{10}(S)]^2 - 0.0022 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.2279 + 0.0233 \times \log_{10}(S) - 0.0120 \times [\log_{10}(S)]^2 - 0.00001 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$

7.1.1.3 L/D = 0.60

$$(h_0/C) = 0.4985 + 0.3814 \times \log_{10}(S) + 0.0405 \times [\log_{10}(S)]^2 - 0.0169 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$

$$(T_{max}) = 36.0434 + 46.1228 \times \log_{10}(S) + 51.0671 \times [\log_{10}(S)]^2 + 14.7580 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9845$

$$[(R/C)/f] = 4.7983 + 12.8196 \times \log_{10}(S) + 13.6019 \times [\log_{10}(S)]^2 + 4.0335 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9880$

$$(\Phi) = 29.9274 + 20.2998 \times \log_{10}(S) + 11.4392 \times [\log_{10}(S)]^2 + 2.8503 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(Q/RCNL) = 2.0000 + 1.1721 \times \log_{10}(S) - 0.0114 \times [\log_{10}(S)]^2$$

$$- 0.08896 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9983$

$$(Q_J/Q) = 0.1697 - 0.1328 \times \log_{10}(S) + 0.0188 \times [\log_{10}(S)]^2$$

$$- 0.0066 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.2295 + 0.0190 \times \log_{10}(S) - 0.0114 \times [\log_{10}(S)]^2$$

$$+ 0.0008 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9997$.

7.1.1.4 L/D = 1.0

$$(h_0/C) = 0.5483 + 0.3985 \times \log_{10}(S) + 0.0336 \times [\log_{10}(S)]^2$$

$$- 0.0187 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$

$$(T_{var}) = 33.1910 + 42.5008 \times \log_{10}(S) + 41.1208 \times [\log_{10}(S)]^2$$

$$+ 10.2696 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9838$

$$[(R/C)f] = 4.9863 + 11.9027 \times \log_{10}(S) + 11.0654 \times [\log_{10}(S)]^2$$

$$+ 3.0159 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9895$

$$(\Phi) = 31.1042 + 23.7320 \times \log_{10}(S) + 13.9497 \times [\log_{10}(S)]^2$$

$$+ 3.3215 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(Q/RCNL) = 2.0456 + 1.2322 \times \log_{10}(S) - 0.0102 \times [\log_{10}(S)]^2 - 0.0897 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9983$

$$(Q_J/Q) = 0.0948 - 0.0803 \times \log_{10}(S) + 0.0159 \times [\log_{10}(S)]^2 - 0.0103 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.2455 + 0.0144 \times \log_{10}(S) - 0.0102 \times [\log_{10}(S)]^2 + 0.0022 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9997$.

7.1.1.5 L/D = 1.50

$$(h_0/C) = 0.5728 + 0.4065 \times \log_{10}(S) + 0.0296 \times [\log_{10}(S)]^2 - 0.0199 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$

$$(T_{wr}) = 31.8221 + 40.5831 \times \log_{10}(S) + 37.0158 \times [\log_{10}(S)]^2 + 8.4603 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9825$

$$[(R/C)f] = 4.9136 + 11.3423 \times \log_{10}(S) + 10.1802 \times [\log_{10}(S)]^2 + 2.7229 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9902$

$$(\Phi) = 31.8458 + 25.4938 \times \log_{10}(S) + 15.1694 \times [\log_{10}(S)]^2 + 3.5794 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(Q/RCNL) = 2.0687 + 1.2647 \times \log_{10}(S) - 0.0097 \times [\log_{10}(S)]^2 - 0.0908 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9982$

$$(Q_f/Q) = 0.0610 - 0.0511 \times \log_{10}(S) + 0.0096 \times [\log_{10}(S)]^2 - 0.0114 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(P/p_{max}) = 0.2640 + 0.0133 \times \log_{10}(S) - 0.0102 \times [\log_{10}(S)]^2 + 0.0034 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

7.1.1.6 L/D = 2.0

$$(h_0/C) = 0.5828 + 0.4085 \times \log_{10}(S) + 0.0283 \times [\log_{10}(S)]^2 - 0.0197 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9987$

$$(T_{var}) = 31.2601 + 39.8214 \times \log_{10}(S) + 35.0581 \times [\log_{10}(S)]^2 + 7.5437 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9816$

$$[(R/C)f] = 4.9172 + 11.1936 \times \log_{10}(S) + 9.6814 \times [\log_{10}(S)]^2 + 2.4856 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9899$

$$(\Phi) = 32.2740 + 26.4723 \times \log_{10}(S) + 15.5414 \times [\log_{10}(S)]^2$$

$$+ 3.5154 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9997$

$$(Q/RCNL) = 2.0791 + 1.2798 \times \log_{10}(S) - 0.0096 \times [\log_{10}(S)]^2$$

$$- 0.0905 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9982$

$$(Q_s/Q) = 0.0445 - 0.0399 \times \log_{10}(S) + 0.0093 \times [\log_{10}(S)]^2$$

$$- 0.0074 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.2763 + 0.0152 \times \log_{10}(S) - 0.0113 \times [\log_{10}(S)]^2$$

$$+ 0.0025 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

7.1.2 120 Degree Partial Bearing

7.1.2.1 L/D = 0.25

$$(h_0/C) = 0.4021 + 0.3702 \times \log_{10}(S) + 0.0534 \times [\log_{10}(S)]^2$$

$$- 0.0198 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9973$

$$(T_{nor}) = 48.2706 + 90.0589 \times \log_{10}(S) + 135.6586 \times [\log_{10}(S)]^2$$

$$+ 50.5504 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9853$

$$[(R/C)f] = 6.7604 + 22.7877 \times \log_{10}(S) + 35.6489 \times [\log_{10}(S)]^2$$

$$+ 13.0279 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9865$

$$(\Phi) = 39.4874 + 22.2979 \times \log_{10}(S) + 5.36989 \times [\log_{10}(S)]^2 + 0.5260 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9997$

$$(Q/RCNL) = 3.1534 + 0.6959 \times \log_{10}(S) - 0.2754 \times [\log_{10}(S)]^2 - 0.1289 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$

$$(Q_J/Q) = 0.6014 - 0.2961 \times \log_{10}(S) - 0.0614 \times [\log_{10}(S)]^2 + 0.0024 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9989$

$$(P/p_{max}) = 0.3203 + 0.1389 \times \log_{10}(S) - 0.0070 \times [\log_{10}(S)]^2 - 0.0124 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$.

7.1.2.2 L/D = 0.5

$$(h_0/C) = 0.6099 + 0.4311 \times \log_{10}(S) - 0.0002 \times [\log_{10}(S)]^2 - 0.0350 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9980$

$$(T_{max}) = 43.3949 + 76.6739 \times \log_{10}(S) + 68.7475 \times [\log_{10}(S)]^2 + 18.5388 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9881$

$$[(R/C)f] = 8.9985 + 20.4278 \times \log_{10}(S) + 17.7378 \times [\log_{10}(S)]^2$$

$$+ 4.6498 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9897$

$$(\Phi) = 49.8347 + 28.4504 \times \log_{10}(S) + 7.7752 \times [\log_{10}(S)]^2$$

$$+ 1.0926 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9995$.

$$(Q/RCNL) = 3.1662 + 0.6267 \times \log_{10}(S) - 0.4997 \times [\log_{10}(S)]^2$$

$$- 0.1719 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(Q_s/Q) = 0.3740 - 0.3382 \times \log_{10}(S) - 0.0227 \times [\log_{10}(S)]^2$$

$$+ 0.0105 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9994$

$$(P/p_{max}) = 0.3909 + 0.0985 \times \log_{10}(S) - 0.0447 \times [\log_{10}(S)]^2$$

$$- 0.0169 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$

7.1.2.3 L/D = 0.60

$$(h_0/C) = 0.6633 + 0.4363 \times \log_{10}(S) - 0.0270 \times [\log_{10}(S)]^2$$

$$- 0.0432 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9984$

$$(T_{wr}) = 41.0467 + 69.5497 \times \log_{10}(S) + 59.4693 \times [\log_{10}(S)]^2$$

$$+ 15.6123 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9895$

$$[(R/C)f] = 8.8656 + 18.6759 \times \log_{10}(S) + 15.1023 \times [\log_{10}(S)]^2 + 3.8015 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9908$

$$(\Phi) = 51.9800 + 30.5480 \times \log_{10}(S) + 9.3650 \times [\log_{10}(S)]^2 + 1.5674 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9997$

$$(Q/RCNL) = 3.1414 + 0.6290 \times \log_{10}(S) - 0.5725 \times [\log_{10}(S)]^2 - 0.1935 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(Q_J/Q) = 0.3087 - 0.3310 \times \log_{10}(S) + 0.0077 \times [\log_{10}(S)]^2 + 0.0150 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(P/p_{max}) = 0.3932 + 0.0782 \times \log_{10}(S) - 0.0531 \times [\log_{10}(S)]^2 - 0.0170 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$.

7.1.2.4 L/D = 1.00

$$(h_0/C) = 0.7827 + 0.4229 \times \log_{10}(S) - 0.1011 \times [\log_{10}(S)]^2 - 0.0624 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$

$$(T_{max}) = 34.9681 + 55.0469 \times \log_{10}(S) + 42.6367 \times [\log_{10}(S)]^2 + 9.8920 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9916$

$$[(R/C)/] = 8.0706 + 14.7393 \times \log_{10}(S) + 10.2367 \times [\log_{10}(S)]^2 + 2.3003 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9931$

$$(\Phi) = 57.9963 + 38.4480 \times \log_{10}(S) + 14.6766 \times [\log_{10}(S)]^2 + 2.7768 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(Q/RCNL) = 3.1021 + 0.6040 \times \log_{10}(S) - 0.7942 \times [\log_{10}(S)]^2 - 0.2619 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9997$

$$(Q_J/Q) = 0.1594 - 0.2432 \times \log_{10}(S) + 0.0693 \times [\log_{10}(S)]^2 + 0.0225 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.4224 + 0.0372 \times \log_{10}(S) - 0.0760 \times [\log_{10}(S)]^2 - 0.0186 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

7.1.2.5 L/D = 1.50

$$(h_0/C) = 0.8451 + 0.3933 \times \log_{10}(S) - 0.1616 \times [\log_{10}(S)]^2 - 0.0780 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9984$

$$(T_{nr}) = 31.6216 + 48.2393 \times \log_{10}(S) + 35.3703 \times [\log_{10}(S)]^2$$

$$+ 7.2520 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9916$

$$[(R/C)/f] = 7.4542 + 12.8040 \times \log_{10}(S) + 8.4017 \times [\log_{10}(S)]^2$$

$$+ 1.8162 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9945$

$$(\Phi) = 62.3868 + 45.1916 \times \log_{10}(S) + 19.1902 \times [\log_{10}(S)]^2$$

$$+ 3.8125 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$

$$(Q/RCNL) = 3.0952 + 0.5662 \times \log_{10}(S) - 0.9593 \times [\log_{10}(S)]^2$$

$$- 0.3123 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9994$

$$(Q_s/Q) = 0.0915 - 0.1651 \times \log_{10}(S) + 0.0785 \times [\log_{10}(S)]^2$$

$$+ 0.0197 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.4604 + 0.0109 \times \log_{10}(S) - 0.0997 \times [\log_{10}(S)]^2$$

$$- 0.0217 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

7.1.2.6 L/D = 2.00

$$(h_0/C) = 0.8733 + 0.3767 \times \log_{10}(S) - 0.1840 \times [\log_{10}(S)]^2$$

$$- 0.0814 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9983$

$$(T_{max}) = 30.0481 + 45.0170 \times \log_{10}(S) + 31.7613 \times [\log_{10}(S)]^2 + 5.8676 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9923$

$$[(R/C)f] = 7.1433 + 11.8151 \times \log_{10}(S) + 7.3953 \times [\log_{10}(S)]^2 + 1.5208 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9945$

$$(\Phi) = 64.9010 + 48.5140 \times \log_{10}(S) + 20.3838 \times [\log_{10}(S)]^2 + 3.7839 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$

$$(Q/RCNL) = 3.0936 + 0.5488 \times \log_{10}(S) - 1.0407 \times [\log_{10}(S)]^2 - 0.3356 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9992$

$$(Q_J/Q) = 0.0629 - 0.1100 \times \log_{10}(S) + 0.0918 \times [\log_{10}(S)]^2 + 0.0236 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9999$

$$(P/p_{max}) = 0.4867 - 0.0104 \times \log_{10}(S) - 0.1267 \times [\log_{10}(S)]^2 - 0.0286 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9992$.

7.1.3 180 Degree Partial Bearing

7.1.3.1 L/D = 0.25

$$(h_0/C) = 0.4090 + 0.3798 \times \log_{10}(S) + 0.0563 \times [\log_{10}(S)]^2$$

$$- 0.0199 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9970$

$$(T_{max}) = 44.9868 + 132.0005 \times \log_{10}(S) + 199.0649 \times [\log_{10}(S)]^2$$
$$+ 72.1824 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9857$

$$[(R/C)f] = 9.2120 + 33.6078 \times \log_{10}(S) + 50.5318 \times [\log_{10}(S)]^2$$
$$+ 18.1075 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9868$

$$(\Phi) = 45.4595 + 27.5202 \times \log_{10}(S) + 3.6480 \times [\log_{10}(S)]^2$$
$$- 0.9418 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9980$

$$(Q/RCNL) = 4.1952 + 0.0295 \times \log_{10}(S) - 0.4385 \times [\log_{10}(S)]^2$$
$$- 0.1121 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9781$

$$(Q_J/Q) = 0.6927 - 0.3001 \times \log_{10}(S) - 0.1019 \times [\log_{10}(S)]^2$$
$$- 0.0091 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9980$

$$(P/p_{max}) = 0.3393 + 0.1655 \times \log_{10}(S) + 0.0013 \times [\log_{10}(S)]^2$$
$$- 0.0122 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9974$.

7.1.3.2 L/D = 0.5

$$(h_0/C) = 0.6381 + 0.4487 \times \log_{10}(S) - 0.0054 \times [\log_{10}(S)]^2 - 0.0383 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9976$

$$(T_{wr}) = 53.3911 + 111.8834 \times \log_{10}(S) + 95.4607 \times [\log_{10}(S)]^2 + 24.8549 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9884$

$$[(R/C)f] = 12.6704 + 28.6679 \times \log_{10}(S) + 24.0069 \times [\log_{10}(S)]^2 + 6.1403 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9897$

$$(\Phi) = 60.9297 + 30.9128 \times \log_{10}(S) + 1.0408 \times [\log_{10}(S)]^2 - 1.5054 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$.

$$(Q/RCNL) = 3.8928 - 0.2862 \times \log_{10}(S) - 0.7317 \times [\log_{10}(S)]^2 - 0.1545 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9895$

$$(Q_s/Q) = 0.4637 - 0.4232 \times \log_{10}(S) - 0.0884 \times [\log_{10}(S)]^2 + 0.0017 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9984$

$$(P/p_{max}) = 0.4510 + 0.1460 \times \log_{10}(S) - 0.0422 \times [\log_{10}(S)]^2 - 0.0196 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9988$

7.1.3.3 L/D = 0.60

$$(h_0/C) = 0.7013 + 0.4531 \times \log_{10}(S) - 0.0400 \times [\log_{10}(S)]^2 - 0.049 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9979$

$$(T_{max}) = 51.3612 + 100.4663 \times \log_{10}(S) + 80.3894 \times [\log_{10}(S)]^2 + 20.3170 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9899$

$$[(R/C)/f] = 12.3491 + 25.7984 \times \log_{10}(S) + 20.0548 \times [\log_{10}(S)]^2 + 4.9056 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9910$

$$(\Phi) = 65.2930 + 30.9865 \times \log_{10}(S) - 0.9703 \times [\log_{10}(S)]^2 - 2.0567 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9987$

$$(Q/RCNL) = 3.7943 - 0.3683 \times \log_{10}(S) - 0.8935 \times [\log_{10}(S)]^2 - 0.1932 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9922$

$$(Q_s/Q) = 0.3995 - 0.4558 \times \log_{10}(S) - 0.0623 \times [\log_{10}(S)]^2 + 0.01 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9988$

$$(P/p_{max}) = 0.4630 + 0.1225 \times \log_{10}(S) - 0.0579 \times [\log_{10}(S)]^2 - 0.0215 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9991$.

7.1.3.4 L/D = 1.00

$$(h_0/C) = 0.8558 + 0.4019 \times \log_{10}(S) - 0.1717 \times [\log_{10}(S)]^2 - 0.0832 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9979$

$$(T_{max}) = 43.6607 + 74.7092 \times \log_{10}(S) + 52.8485 \times [\log_{10}(S)]^2 + 11.9117 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9922$

$$[(R/C)f] = 10.7286 + 19.0089 \times \log_{10}(S) + 12.5498 \times [\log_{10}(S)]^2 + 2.7150 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9935$

$$(\Phi) = 74.4870 + 28.9731 \times \log_{10}(S) - 5.7892 \times [\log_{10}(S)]^2 - 3.0891 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9980$

$$(Q/RCNL) = 3.4472 - 0.7436 \times \log_{10}(S) - 1.4491 \times [\log_{10}(S)]^2 - 0.3370 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9966$

$$(Q_J/Q) = 0.1923 - 0.4147 \times \log_{10}(S) + 0.0245 \times [\log_{10}(S)]^2 + 0.0298 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9991$

$$(P/p_{max}) = 0.5226 + 0.0437 \times \log_{10}(S) - 0.1290 \times [\log_{10}(S)]^2 - 0.0344 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9997$.

7.1.3.5 L/D = 1.50

$$(h_g/C) = 0.9337 + 0.2793 \times \log_{10}(S) - 0.3284 \times [\log_{10}(S)]^2 - 0.1203 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9977$

$$(T_{wr}) = 38.3151 + 62.3522 \times \log_{10}(S) + 41.7115 \times [\log_{10}(S)]^2 + 8.3204 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9925$

$$[(R/C)f] = 9.4145 + 15.2833 \times \log_{10}(S) + 9.2380 \times [\log_{10}(S)]^2 + 1.8482 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9944$

$$(\Phi) = 79.2086 + 26.2167 \times \log_{10}(S) - 10.0490 \times [\log_{10}(S)]^2 - 4.0919 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9983$

$$(Q/RCNL) = 3.1597 - 1.2952 \times \log_{10}(S) - 2.1354 \times [\log_{10}(S)]^2 - 0.5198 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9982$

$$(Q_s/Q) = 0.0821 - 0.2817 \times \log_{10}(S) + 0.1084 \times [\log_{10}(S)]^2 + 0.0466 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9993$

$$(P/p_{max}) = 0.5680 - 0.0738 \times \log_{10}(S) - 0.2442 \times [\log_{10}(S)]^2 - 0.0599 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9998$.

7.1.3.6 L/D = 2.00

$$(h_o/C) = 0.9626 + 0.1819 \times \log_{10}(S) - 0.4290 \times [\log_{10}(S)]^2 - 0.1433 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9968$

$$(T_{max}) = 35.7745 + 57.4850 \times \log_{10}(S) + 37.4337 \times [\log_{10}(S)]^2 + 6.8069 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9935$

$$[(R/C)/] = 8.7123 + 13.6733 \times \log_{10}(S) + 8.0385 \times [\log_{10}(S)]^2 + 1.5769 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9951$

$$(\Phi) = 82.0402 + 27.2613 \times \log_{10}(S) - 9.3887 \times [\log_{10}(S)]^2 - 3.8052 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9985$

$$(Q/RCNL) = 2.9875 - 1.6246 \times \log_{10}(S) - 2.5007 \times [\log_{10}(S)]^2 - 0.6103 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9986$

$$(Q_J/Q) = 0.0390 - 0.2032 \times \log_{10}(S) + 0.1247 \times [\log_{10}(S)]^2 + 0.0462 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9989$

$$(P/p_{max}) = 0.5875 - 0.1645 \times \log_{10}(S) - 0.3185 \times [\log_{10}(S)]^2 - 0.0742 \times [\log_{10}(S)]^3$$

and corresponding $r^2 = 0.9996$.

The different stages of the program for designing partial journal bearings for optimal clearance condition are described below. In this program, a database is used to show menus for the bearing-application to get value of unit load P, decision-criterion to calculate utility value, L/D ratio, oil-grade, and bearing arc. The first three are shown in the input stage (stage 0) while the last two are needed in stage 2. This is referred to as the input-menu database.

7.2 STAGE 0 (INPUT STAGE)

The final goal of stage 0 is to obtain the functional and prescriptive specifications (c.f. Sec. 1.4) entered by the user and determine the bearing length and diameter. The functional and prescriptive specifications available at this stage are listed below:

1. the bearing radial load to be supported: W (kN),
2. the speed of the journal: N_j (*rpm*),
3. the bearing unit load: P (MPa); the user either enters or selects the application from the menu where the bearing would be used to get the value from the database,
4. the bearing industrial-application,
5. the lubricant inlet temperature: T_{in} (°*C*),
6. the decision-criterion,
7. the L/D ratio (0.25, 0.5, 0.6, 1.0, 1.5, 2.0, or all six).

The group of rules and the databases used at this stage are:

1. a set of rules for menu-handling and screen-handling,
2. a set of rules for prompting and entering the functional and prescriptive specifications,

3. if the user selects the application instead of entering the value of P, a knowledge base in the form of a table (c.f. Table 4.1),
4. a knowledge base containing Table 4.2 to get the value of L/D ratio,
5. a set of rules to find the length and corresponding diameter of the bearing using P = W/(L.D).

Thus each partial solution of this stage consists of the L/D ratio, length, and diameter. At the end of this stage all partial solutions are written onto an internal database and the knowledge bases of this stage are erased.

7.3 STAGE 1

The final goal of this stage is to select the clearance and check whether this is in the prescribed range for each partial solution of the above input stage. The group of rules and the databases used in this stage are given below:

1. a set of rules to select the clearance from the database depending on the diameter and speed,
2. a knowledge base consisting of the values of clearance corresponding to the diameter and speed, which is in the form of Table 5.1,
3. a set of rules to check whether the calculated clearance in each case is within the prescribed range,
4. a database consisting of clearance limits depending on the industrial application which is in the form of a table (c.f. Table 3.1).

If no partial solution within the desired clearance limit for manufacturability is obtained, this condition may be modified to perform failure handling [34]. Each partial solution of this stage will have the unit load P, speed N, L/D ratio, length L, diameter D, clearance C, corresponding to the

minimum limit of clearance C_{min}, and lubricant inlet temperature T_{in}. At the end of stage 1, all partial solutions are written onto an internal database. The database consisting of all partial solutions of the stage 0 is erased. All the databases of stage 1 are also erased.

For each partial solution of stage 0 there may be either no solution or one solution at stage 1. A failure handler is used if there is not even a single solution. That is the industrial application is changed or clearance limits are modified or the maximum bearing diameter limit of 200 mm is increased. In stage 1, only discriminating skills of a designer are used. The number of partial design solutions at the end of this stage (stage 1) is N_1, which will be less than or equal to the number of partial solutions available at the end of stage 0.

Thus the number of partial solutions available at the end of stage 1 is N_1.

7.4 STAGE 2

The final goal of stage 2 is to get the oil-grade and bearing arc, and determine the temperature-rise using an iterative method discussed in Sec. 5.2, the viscosity of the oil at the operating temperature T_{op}, and the bearing characteristic number S. Then the lubricant outlet temperature T_{out} is calculated and checked w.r.t. the condition mentioned in Sec. 3.3.1. If the outlet temperature does not exceed the prescribed limit, all the performance variables: h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, and P/p_{max} are determined with the help of equations written in terms of $\log_{10}(S)$ in Sec. 7.1.

To achieve this goal the following sets of rules and the databases are used:

1. a set of rules to get the oil-grade (SAE 10, 20, 30, 40, 50, 60, 70, or all seven)

selected by the user,

2. a set of rules to get the bearing arc β (180°, 120°, 60°, or all three) chosen by the user,
3. a set of rules to determine the final value of the temperature-rise from an initial trial value using an iterative method (c.f. Sec. 5.2),
4. a set of rules to determine the viscosity of the oil using the equations described in Sec. 4.1,
5. a database consisting of the coefficients of the equations for determining the viscosity (c.f. Sec. 4.1) which is in the form of a table,
6. a set of rules to determine the Sommerfeld number $S = (R/C)^2 \mu N/P$,
7. a set of rules to find the outlet temperature and check whether it is below 120°C,
8. a set of rules to determine the performance variables of all the solutions satisfying the temperature constraint based on the equations described in Sec. 7.1,
9. a database consisting of the coefficients of the equations for determining the performance variables (c.f. Sec. 7.1), which is in the form of a table.

Each partial feasible solution of this stage consists of the bearing arc β, viscosity μ, unit load P, speed N, oil-grade, L/D ratio, length L, diameter D, clearance C, corresponding to the minimum limit of clearance C_{min}, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, P/p_{max}, and $(h_{0_{F.S.=2}})/C$ needed for checking the minimum film thickness requirement in the next stage and is obtained when S is halved. At the end of stage 2 all partial solutions are written onto an internal database (on the

blackboard). The databases of stage 2 containing coefficients of the equations to determine viscosity and the performance variables are retracted. Also the input-menu database and the database containing all partial solutions of stage 1 are retracted.

In stage 2 both recombining and discriminating skills of a designer are used. Recombining in the sense if the user selects all the seven oil-grades and three arcs, the solutions available at the end of stage 1 will increase and discriminating in the sense that only those solutions are retained which satisfy the outlet temperature condition. The number of partial solutions of stage 2 is N_2 and it is not directly related to the number of solutions of the previous stage N_1.

For each partial solution of stage 1 there may be either no solution or one (or more) solution at stage 2. A failure handler is used if there is not even a single solution; i.e., the temperature condition may be modified.

7.5 STAGE 3

The final goal of this stage is to determine whether each of the partial solutions of stage 2 satisfies the minimum film thickness criterion given by Juvinall [68] (c.f. Sec. 3.3.3); i.e., for a factor of safety of 2, $h_0 \geq 0.005 + 0.00004D$ (h_0 and D *are in mm*). A set of rules to determine the value of $h_{0_{F.S.=2}}$ and check whether it is in the acceptable range is used in this stage. In addition, a set of rules is used to determine actual h_0, coefficient of friction f, and torque T (=fWR).

Each design solution of stage 3 will have the arc β, viscosity μ, unit load P, speed N, load W, oil-grade, L/D ratio, length L, diameter D, clearance C, corresponding to the minimum limit of clearance C_{min}, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, P/p_{max}, actual h_0, and torque T. A failure handler may be used if there is no design solution at the end of this stage. That is, this constraint may be modified.

At the end of stage 3 all partial solutions are written onto an internal database and at the same time the database consisting of all partial solutions of stage 2 is erased. In stage 3 only discriminating skills of a designer are used. Therefore, the number of partial solutions at the end of this stage N_3 will be less than or equal to the number of partial solutions in the previous stage N_2.

7.6 STAGE 4

The final goal of this stage is to determine the utility value of all the design solutions of stage 3 (c.f. Sec. 3.4). The group of rules and the databases used in this stage are given below:

1. a set of rules to get the decision criterion entered by the user in the input stage (stage 0),
2. a set of rules to find eccentricity ratio ε and dimensionless speed ω_s,
3. a knowledge base consisting of the utility values and the weighting factors for the decision criterion based on optimal clearance corresponding to all the five decision variables (c.f. Tables 3.2 to 3.7),
4. a set of rules to determine the utility value for each design solution of stage 3

using Eq. (4.3).

Each design solution of stage 4 will have the arc β, viscosity μ, unit load P, speed N, load W, oil-grade, L/D ratio, length L, diameter D, clearance C, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, P/p_{max}, h_{op}, torque T, ε, ω_s, and utility value of the solution U. This stage is a simple procedural stage. Neither the discriminating nor the recombining skills of a designer are used at this stage. Therefore, the number of design solutions of this stage N_4 will be the same as that of stage 3 (N_3).

At the end of stage 4 all partial solutions are written onto an internal database. The design solutions of stage 3 are erased. The databases of the stage 4 are also erased. The user can modify the databases in Tables 3.2 to 3.7.

7.7 STAGE 5 OR DECISION MAKING STAGE

The goal of this stage is to identify the design solution of the previous stage which has the highest utility factor U_{max}. In addition all the design solutions of the previous stage for which the utility factors are at least ninety five percent (95%) of the U_{max} are identified.

The group of rules used in this stage are given below:

1. a set of rules to identify the design solution which has the highest utility value U_{max},
2. a set of rules to identify the design solutions which have the utility values of at least 95% of the U_{max}.

In this stage, only the discriminating skills of a designer are used. Therefore the number of solutions at the end of this stage N_5 is less than or equal to the number of solutions of the previous stage N_4.

Each design solution of stage 5 will have the arc β, viscosity μ, unit load P, speed N, load W, oil-grade, L/D ratio, length L, diameter D, clearance C, lubricant inlet temperature T_{in}, lubricant operating temperature T_{op}, lubricant outlet temperature T_{out}, Sommerfeld number S, h_0/C, (R/C)f, Φ, Q/RCNL , Q_s/Q, P/p_{max}, h_0, torque T, ε, ω_s, and utility value of the solution U. At the end of stage 5 all design solutions are written onto an internal database and at the same time all partial solutions of stage 4 are erased.

7.8 STAGE 6 OR OUTPUT STAGE

The goal of this stage is to file all the finally selected design solutions. Each design solution for which the utility value is at least ninety five percent (95%) of the highest utility value (U_{max}) is printed in the 'results.out' file for the perusal of the user.

The group of rules and the database used in this stage are given below:

1. a set of rules to calculate eccentricity e, coefficient of friction f, power loss H (= $2\pi NT$), oil flow Q, side leakage Q_s, and maximum pressure p_{max},
2. a knowledge base containing the L/D ratio, oil-grade, and bearing arc β,
3. a set of rules to print the chosen design solutions in the 'results.out' file.

In this stage neither the discriminating nor the recombining skills of a designer are used. Therefore the number of solutions at the end of this stage N_6 is equal to N_5. At the end of this stage all the design alternatives are printed in

the 'results.out' file. The design solutions of all the previous stages including stage 5 are erased and the database used in this stage is also erased.

The 'results.out' file will have the functional and prescriptive specifications (c.f. Sec. 1.4) as the load W, speed N, unit load P, bearing industrial-application, lubricant inlet temperature, decision-criterion, L/D ratio, oil-grade, bearing arc, length L, diameter D, T_{op}, μ, S, C, h_0, ε, eccentricity e, coefficient of friction f, torque T, power loss H, total oil supplied Q, side leakage Q_s, maximum film pressure p_{max}, position of minimum film thickness Φ, dimensionless velocity ω_s, and the utility value U.

7.9 EXAMPLE

To illustrate the use of the present 'Expert System' it is used to design a partial journal bearing for the optimal clearance. This example is similar to the one used by Orthwein[19]. The user enters the following specifications:

Bearing load = 32 kN

Speed of the journal = 3600 rpm

Bearing unit load = 1.38 MPa

Bearing industrial application = General machine practice (rotating motion)

Lubricant inlet temperature = 43.3 °*C*

L/D ratio: all six ratios

Oil: all seven grades of oil.

Bearing arc: all three arcs.

In this case the number of partial design solutions at the end of stage 1, N_1 is 2. For L/D = 0.25 and 0.5, the diameters (D) obtained are 304 mm and 216

mm, respectively. They are > 200 mm, therefore rejected. For L/D = 1.5 and 2, the values of D are 124.7 mm and 107.5 mm, respectively and the clearance selected in both the cases is 0.1041 mm. But acceptable limits of clearance when D is in the range of 100 to 200 mm and for general machine practice under rotating motion are 0.127 to 0.2032 mm. Therefore the value of C obained for L/D = 1.5 and 2 is not accepted. For L/D = 0.6 and 1, the diameters (D) are 196.7 mm and 152 mm, respectively and the clearance is 0.137 mm which is in the acceptable range. Consequently, the 'ES' gives N_1 as 2. In this stage, only discriminating skills of the designer are used.

The number of partial design solutions at the end of stage 2, N_2 is 42. In this stage, seven oil-grades and three bearing-arcs are tried and all the solutions satisfy T_{out} condition. At the end of stage 3 there are fourty two (N_3=42) solutions. All the solutions satisfy minimum film thickness requirement.

The number of design solutions at the end of stage 4, N_4 is 42 and utility value on the basis of optimal clearance of all the solutions is obtained. The number of design solutions at the end of stage 5 (decision making stage), N_5 = 12. All the twelve design alternatives are stored in the 'results.out' file in the output stage. The print-out of this file is attached herewith on the succeeding pages. Out of these 12 design alternatives, the solution with L/D = 1.0, SAE 10 oil, and β = 120° has the highest utility value (U_{max} = 3.38) and 95% of this U_{max} is 3.21.

Partial Bearing Design Specifications

The load on the bearing, W = 32.00 kN

The speed of the journal, N = 60.000 rps

The bearing unit load, P = 1.380 MPa

The bearing-application for clearance is No. 3. General machine practice - rotating motion

The inlet temperature, Tin = 43.300 deg C

The decision-criterion for utility is No. 1. Based on optimal clearance

Stable Design Alternatives Are

1. L/D = 1.0

Design Based on SAE 10 Oil

180 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 50.90 deg. C

Viscosity of the Lubricant, Mu = 19.17 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.256

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.079 mm

Eccentricity Ratio, e/C = 0.424

Eccentricity, e = 0.058 mm

Coefficient of friction, f = 0.0060

Torque required = 14.574 N.m

Power Lost = 5496.52 W

Total Oil Supplied = 327584.8 cubic mm per sec

Side Leakage = 144062.9 cubic mm per sec

Maximum Film Pressure = 3.008 MPa

Position of minimum film thickness = 55.98 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.30

2. L/D = 0.6

Design Based on SAE 10 Oil

120 Degree Partial Bearing

Length, L = 118 mm

Diameter, D = 196.67 mm

Operating Temperature, Top = 53.15 deg. C

Viscosity of the Lubricant, Mu = 17.30 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.387

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.066 mm

Eccentricity Ratio, e/C = 0.518

Eccentricity, e = 0.071 mm

Coefficient of friction, f = 0.0048

Torque required = 15.210 N.m

Power Lost = 5736.25 W

Total Oil Supplied = 266821.0 cubic mm per sec

Side Leakage = 118800.2 cubic mm per sec

Maximum Film Pressure = 3.907 MPa

Position of minimum film thickness = 40.88 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.27

3. L/D = 1.0

Design Based on SAE 10 Oil

120 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 50.90 deg. C

Viscosity of the Lubricant, Mu = 19.17 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.256

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.070 mm

Eccentricity Ratio, e/C = 0.490

Eccentricity, e = 0.067 mm

Coefficient of friction, f = 0.0044

Torque required = 10.787 N.m

Power Lost = 4068.26 W

Total Oil Supplied = 239456.1 cubic mm per sec

Side Leakage = 77269.4 cubic mm per sec

Maximum Film Pressure = 3.654 MPa

Position of minimum film thickness = 39.83 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.38

4. L/D = 0.6

Design Based on SAE 10 Oil

60 Degree Partial Bearing

Length, L = 118 mm

Diameter, D = 196.67 mm

Operating Temperature, Top = 53.91 deg. C

Viscosity of the Lubricant, Mu = 16.73 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.375

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.047 mm

Eccentricity Ratio, e/C = 0.655

Eccentricity, e = 0.090 mm

Coefficient of friction, f = 0.0021

Torque required = 6.550 N.m

Power Lost = 2470.44 W

Total Oil Supplied = 143573.1 cubic mm per sec

Side Leakage = 33054.7 cubic mm per sec

Maximum Film Pressure = 6.294 MPa

Position of minimum film thickness = 23.14 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.29

5. L/D = 1.0

Design Based on SAE 10 Oil

60 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 52.89 deg. C

Viscosity of the Lubricant, Mu = 17.51 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.234

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.043 mm

Eccentricity Ratio, e/C = 0.685

Eccentricity, e = 0.094 mm

Coefficient of friction, f = 0.0020

Torque required = 4.930 N.m

Power Lost = 1859.19 W

Total Oil Supplied = 122279.5 cubic mm per sec

Side Leakage = 18893.6 cubic mm per sec

Maximum Film Pressure = 5.953 MPa

Position of minimum film thickness = 20.85 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.32

6. L/D = 1.0

Design Based on SAE 20 Oil

180 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 50.90 deg. C

Viscosity of the Lubricant, Mu = 25.93 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.347

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.088 mm

Eccentricity Ratio, e/C = 0.357

Eccentricity, e = 0.049 mm

Coefficient of friction, f = 0.0079

Torque required = 19.219 N.m

Power Lost = 7248.26 W

Total Oil Supplied = 333808.0 cubic mm per sec

Side Leakage = 128592.6 cubic mm per sec

Maximum Film Pressure = 2.883 MPa

Position of minimum film thickness = 60.24 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.22

7. L/D = 1.0

Design Based on SAE 20 Oil

120 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 50.90 deg. C

Viscosity of the Lubricant, Mu = 25.93 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.347

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.078 mm

Eccentricity Ratio, e/C = 0.427

Eccentricity, e = 0.059 mm

Coefficient of friction, f = 0.0058

Torque required = 14.180 N.m

Power Lost = 5347.93 W

Total Oil Supplied = 254664.7 cubic mm per sec

Side Leakage = 72242.0 cubic mm per sec

Maximum Film Pressure = 3.529 MPa

Position of minimum film thickness = 43.16 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.30

8. L/D = 0.6

Design Based on SAE 20 Oil

60 Degree Partial Bearing

Length, L = 118 mm

Diameter, D = 196.67 mm

Operating Temperature, Top = 53.91 deg. C

Viscosity of the Lubricant, Mu = 22.46 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.503

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.053 mm

Eccentricity Ratio, e/C = 0.611

Eccentricity, e = 0.084 mm

Coefficient of friction, f = 0.0029

Torque required = 9.112 N.m

Power Lost = 3436.38 W

Total Oil Supplied = 157532.0 cubic mm per sec

Side Leakage = 33266.9 cubic mm per sec

Maximum Film Pressure = 6.194 MPa

Position of minimum film thickness = 24.82 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.29

9. L/D = 1.0

Design Based on SAE 20 Oil

60 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 52.89 deg. C

Viscosity of the Lubricant, Mu = 23.57 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.315

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.049 mm

Eccentricity Ratio, e/C = 0.641

Eccentricity, e = 0.088 mm

Coefficient of friction, f = 0.0026

Torque required = 6.226 N.m

Power Lost = 2348.04 W

Total Oil Supplied = 136460.9 cubic mm per sec

Side Leakage = 19178.8 cubic mm per sec

Maximum Film Pressure = 5.861 MPa

Position of minimum film thickness = 22.29 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.36

10. L/D = 0.6

Design Based on SAE 30 Oil

60 Degree Partial Bearing

Length, L = 118 mm

Diameter, D = 196.67 mm

Operating Temperature, Top = 53.91 deg. C

Viscosity of the Lubricant, Mu = 36.17 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.810

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.064 mm

Eccentricity Ratio, e/C = 0.536

Eccentricity, e = 0.073 mm

Coefficient of friction, f = 0.0052

Torque required = 16.389 N.m

Power Lost = 6180.95 W

Total Oil Supplied = 180534.3 cubic mm per sec

Side Leakage = 32858.3 cubic mm per sec

Maximum Film Pressure = 6.061 MPa

Position of minimum film thickness = 28.17 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.27

11. L/D = 1.0

Design Based on SAE 30 Oil

60 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 52.89 deg. C

Viscosity of the Lubricant, Mu = 38.23 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.512

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.060 mm

Eccentricity Ratio, e/C = 0.564

Eccentricity, e = 0.077 mm

Coefficient of friction, f = 0.0043

Torque required = 10.453 N.m

Power Lost = 3942.42 W

Total Oil Supplied = 160347.8 cubic mm per sec

Side Leakage = 19238.3 cubic mm per sec

Maximum Film Pressure = 5.741 MPa

Position of minimum film thickness = 25.29 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.32

12. L/D = 1.0

Design Based on SAE 40 Oil

60 Degree Partial Bearing

Length, L = 152 mm

Diameter, D = 152.00 mm

Operating Temperature, Top = 52.89 deg. C

Viscosity of the Lubricant, Mu = 54.65 mPa.s

Sommerfeld number (Bearing ch. no.), S = 0.731

Radial Clearance, C = 0.137 mm

Minimum film thickness, hmin = 0.068 mm

Eccentricity Ratio, e/C = 0.505

Eccentricity, e = 0.069 mm

Coefficient of friction, f = 0.0064

Torque required = 15.628 N.m

Power Lost = 5894.16 W

Total Oil Supplied = 178377.7 cubic mm per sec

Side Leakage = 18950.4 cubic mm per sec

Maximum Film Pressure = 5.671 MPa

Position of minimum film thickness = 28.12 degree

The dimensionless velocity, ws = 1.41

The Utility value = 3.26

7.10 DISCUSSIONS

The present results are compared with those given by Orthwein [19]. In the above example, for L/D = 1.0, SAE 20 oil, and 120° partial bearing, he found the following results. The operating temperature T_{op} = 52.27 °C, S = 0.33, C = 0.1524 mm, h_0 = 0.0856 mm, f = 0.0065, and H = 5980.5 W. For the same case the 'ES' gives the results as the operating temperature T_{op} = 50.9 °C, S = 0.347, C = 0.137 mm, h_0 = 0.078 mm, f = 0.0058, and H = 5347.9 W. It is noticed that both results are similar except for minor differences.

In order to see which solution is better, utility function method is employed. The solution obtained from the 'ES' with C = 0.137 mm has the utility value = 3.3. If C = 0.1524 mm is selected from the database, then the solution obtained by the 'ES' is: the operating temperature T_{op} = 50.15 °C, S = 0.291, C = 0.1524 mm, h_0 = 0.082 mm, f = 0.0055, H = 5067.4 W, and utility value U = 3.2. Since, the solution with C = 0.137 mm has higher utility value as compared to the solution with C = 0.1524 mm, therefore it can be inferred that the solution given by the 'ES' is slightly better are better than that given by Orthwein [19].

As mentioned in chapter 6, working of that program is examined with the help of this program. In this connection, firstly the results obtained for the max. W condition are verified. For the example used in chapter 6 and 7, the solution for L/D = 1, optimization condition as the max. load, SAE 10 oil, and 180° bearing, obtained in Chap. 6 is: L = D = 152 mm, operating temperature T_{op} = 49.67 °C, S = 0.2, C = 0.16 mm, h_0 = 0.083 mm, f = 0.0063, and H = 5782.9 W. To verify the above solution, C is taken as 0.16 mm, then this

program gives the results as L = D = 152 mm, operating temperature T_{op} = 49.64 °C, S = 0.2, h_0 = 0.083 mm, f = 0.0056, and H = 5113.4 W. Both the results are matching except for the respective differences in the values of f and H. Thus, it can be said that the program of the previous chapter for the max. W is working appropriately.

In case of minimum friction, the solution for L/D = 1, optimization condition as the min. friction SAE 10 oil, and 120° bearing, obtained in Chap. 6 is: L = D = 152 mm, operating temperature T_{op} = 49.99 °C, S = 0.16, C = 0.178 mm, h_0 = 0.071 mm, f = 0.005, and H = 4590.8 W. Therefore, if C is selected as 0.178 mm, this program gives L = D = 152 mm, operating temperature T_{op} = 49.75 °C, S = 0.161, h_0 = 0.074 mm, f = 0.0039, H = 3586.9 W. Both the results are almost similar except for the respective differences in f and H. Thus it is observed that the program of the previous chapter for the min. f is working well.

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS

An expert system 'ES' for designing hydrodynamic journal bearings both full and partial having arcs of 180°, 120°, and 60° (for solving a class III design problem), using an integrated and dependable design methodology, is successfully developed. For picking up the right solutions, utility function method has been used. The relevant knowledge bases are collected or derived using the established facts. The best fit equations are obtained by curve fitting various necessary data-sets. All the data bases obtained are represented in appropriate forms and found to be efficiently used by the 'ES'.

The 'ES' employs goal-directed rule-based production system and the architecture consists of several stages. The rules and the databases of each stage are properly grouped. At any stage, if there is no design solution, proper failure handler may be devised. The 'ES' makes an exhaustive search of all the alternative solutions and uses computer memory efficiently. With the help of this 'ES' time required for designing journal bearings is considerably reduced and now design engineer can spare this saved time in doing other tasks. Also, the risk of human error is reduced.

Usefulness of the 'ES' is illustrated by examples. The minimum diameter requirement suggested by Welsh [63] at L/D = 0.6 is satisfied (c.f. Example 4.9). The results obtained for full journal bearings for the maximum load are in agreement with the previous results [17,24,68,69]. However, design of full

journal bearings for the minimum friction differs from Moes and Bosma [69] because the latter suggest very large clearance. Although large clearance reduces the frictional torque, it hardly satisfies the manufacturing criterion given by Spotts [67] and particularly for precision practice it is not acceptable. Because of high clearance, minimum film thickness requirement suggested by Juvinall [68] is not fulfilled. In addition, bearing becomes noisy. Therefore the results of the 'ES' are more practicable than those obtained from Moes and Bosma [69].

In the case of designing full journal bearings for the optimal clearance, the 'ES' uses the clearance recommended by Keith, Jr. [15-16] which is a compromise between the clearances for the maximum load and the minimum friction conditions. The compromise is in agreement with Juvinall's recommendation [68]. However, Shigley [17] suggests a clearance lower than that for the max. load because eventual wear of the bearing will increase it. But low clearance will increase the cost of manufacturing. Therefore the present result is better than that suggested by Shigley [17] in terms of cost. Also, it is seen that it is still in the safe range if the clearance is increased by 40% because of wear.

In the case of partial bearings, the solution obtained by the 'ES' has higher utility value than that obtained by Orthwein [19]. Also, it is noted from the example discussed in chapter 6 that 60° partial bearings are less preferred when, load is the decision-criterion.

The following recommendations are made for future research work in this area. The knowledge base used for finding stiffness and damping coefficients is only for full journal bearings and for $0.25 \leq L/D \leq 1.0$. Also, it is not completely dependable. Therefore an adequate and dependable knowledge base is to be generated for both full and partial bearings for all the six L/D ratios used. In

the present system, stability check is not performed on partial bearings owing to the lack of a proper knowledge base. The appropriate knowledge base may be developed. The databases for getting utility values and weighting factors have been used for the first time. They need to be verified and modified accordingly.

The present work has separate programs for the max. load and the min. friction, and the optimal clearance. In future both the programs can be integrated. If an expert is available who has good knowledge of assigning utility values to bearing arcs when it is considered as an additional decision variable, then full bearings may be treated as partial bearings with an arc of 360° and the programs for full and partial bearings may be combined.

While using the utility function method, the decision criterion is max. load, min. friction, or optimal clearance. In future, cost and reliability (life) can also be added. For this purpose, a dependable knowledge base is to be devised. The present 'ES' presumes the use of standard bearing materials. If cost and reliability are regarded as decision criteria, then the material would also be considered as an additional decision variable.

From the manufacturing viewpoint, tolerances on the bearing diameter and length may also be specified. For the radial clearance, the allowable range is already taken from [67].

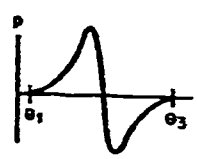
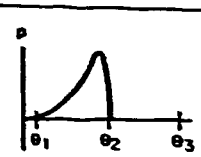
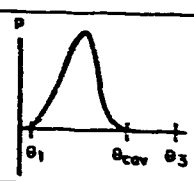
APPENDICES

A: ON THE REYNOLDS EQUATION

A.1 BOUNDARY CONDITIONS

The three types of circumferential boundary conditions applied in solving the Reynolds equation [c.f. Eq. (1.1)] are summarized in Table A.1.

Table A.1 Typical Boundary Conditions on the Reynolds Equation* [15-16].

Names associated with boundary conditions	Pressure profile	Mathematical expression
Sommerfeld (full Sommerfeld)		$p(\theta_1) = p(\theta_3) = 0$ (zero pressure means ambient or atmospheric pressure) For complete journal bearings: $\theta_1 = 0,\ \ \theta_3 = 2\pi$ For partial journal bearings: $\theta_1 = \beta_1,\ \ \theta_3 = \theta_1 + \beta$
Gumbel (half Sommerfeld)		$p(\theta_1) = p(\theta_2) = 0$ $p(\theta_2 \leq \theta \leq \theta_3) = 0$ For complete journal bearings: $\theta_1 = 0,\ \ \theta_2 = \pi,\ \ \theta_3 = 2\pi$
Swift-Stieber (Reynolds)		$p(\theta_2) = p_{cav}$ = atmospheric pressure $\frac{\partial p}{\partial \theta}(\theta_2) = 0$ $\theta_2 = \theta_{cav}$ which must be determined

*$\theta_1 = \beta_1 = \theta_A$, $\theta_3 = \theta_{PD}$, and θ_2 = angular position to zero pressure in film.

A.2 LONG-LENGTH BEARINGS

When L/D > 2, the Reynolds equation reduces to Eq. (1.2). The solutions of this equation, using both Sommerfeld and Gumbel boundary conditions are presented in Table A.2.

Table A.2 Long-Bearing Pressure and Performance Parameters [15-16].

Performance parameter	Sommerfeld conditions	Gumbel conditions
$\frac{p}{12\pi\mu N}\left(\frac{C}{R}\right)^2$	$\frac{(\varepsilon \sin\theta)(2+\varepsilon\cos\theta)}{(2+\varepsilon^2)(1+\varepsilon\cos\theta)^2}$	$\frac{(\varepsilon \sin\theta)(2+\varepsilon\cos\theta)}{(2+\varepsilon^2)(1+\varepsilon\cos\theta)^2}$ $0<\theta<\pi$, $0, \pi<\theta<2\pi$
$\frac{W_R}{3\mu VL}\left(\frac{C}{R}\right)^2$	0	$\frac{4\varepsilon^2}{(2+\varepsilon^2)(1-\varepsilon^2)}$
$\frac{W_T}{3\mu VL}\left(\frac{C}{R}\right)^2$	$\frac{4\pi\varepsilon}{(2+\varepsilon^2)\sqrt{1-\varepsilon^2}}$	$\frac{2\pi\varepsilon}{(2+\varepsilon^2)\sqrt{1-\varepsilon^2}}$
ϕ	$\frac{\pi}{2}$	$\tan^{-1}\left(\frac{\pi}{2}\frac{\sqrt{1-\varepsilon^2}}{\varepsilon}\right)$
S	$\frac{(2+\varepsilon^2)\sqrt{1-\varepsilon^2}}{12\pi^2\varepsilon}$	$\frac{(2+\varepsilon^2)(1-\varepsilon^2)}{6\pi\varepsilon\sqrt{4\varepsilon^2+\pi^2(1-\varepsilon^2)}}$
$\frac{F}{\mu VL}\frac{C}{R}$	$\frac{4\pi(1+2\varepsilon^2)}{(2+\varepsilon^2)\sqrt{1-\varepsilon^2}}$	$\frac{\pi(4+5\varepsilon^2)}{(2+\varepsilon^2)\sqrt{1-\varepsilon^2}}$
$\left(\frac{R}{C}\right)\Omega$	$\frac{1+2\varepsilon^2}{3\varepsilon}$	$\frac{4+5\varepsilon^2}{6\varepsilon}\sqrt{\frac{1-\varepsilon^2}{4\varepsilon^2+\pi(1-\varepsilon^2)}}$
$\frac{Q_s}{RCNL}$	0	0

A.3 SHORT-LENGTH BEARINGS

When L/D < 1/4, the Reynolds equation reduces to Eq. (1.3). The results obtained using Gumbel boundary conditions are shown in Table A.3.

Table A.3 Short-Bearing Pressure and Performance Parameters [15-16].

Performance parameter	Gumbel conditions
$\frac{p}{12\pi\mu N}\left(\frac{C}{R}\right)^2$	$-\frac{\varepsilon \sin\theta}{2(1+\varepsilon\cos\theta)^3}\left(\frac{L}{D}\right)^2(\zeta^2-1) \quad 0 \le \theta \le \pi,$ $0, \pi \le \theta \le 2\pi$
$\frac{W_R}{3\mu VL}\left(\frac{C}{R}\right)^2$	$\frac{4\varepsilon^2}{3(1-\varepsilon^2)^2}\left(\frac{L}{D}\right)^2$
$\frac{W_T}{3\mu VL}\left(\frac{C}{R}\right)^2$	$\frac{\pi\varepsilon^2}{3(1-\varepsilon^2)^3}\left(\frac{L}{D}\right)^2$
$\frac{W}{3\mu VL}\left(\frac{C}{R}\right)^2$	$\frac{\varepsilon\sqrt{\pi^2(1-\varepsilon^2)+16\varepsilon^2}}{3(1-\varepsilon^2)^2}\left(\frac{L}{D}\right)^2$
ϕ	$\tan^{-1}\frac{\pi(1-\varepsilon^2)^2}{4\varepsilon}$
S	$\frac{(1-\varepsilon^2)^2}{\pi\varepsilon\sqrt{\pi^2(1-\varepsilon^2)+16\varepsilon^2}}\left(\frac{D}{L}\right)^2$
$\frac{F}{\mu VL}\frac{C}{R}$	$\frac{2\pi}{\sqrt{1-\varepsilon^2}}$
$\left(\frac{R}{C}\right)\mathcal{U}$	$\frac{(2\pi)(1-\varepsilon^2)^{3/2}}{\varepsilon\sqrt{\pi^2(1-\varepsilon^2)+16\varepsilon^2}}$
$\frac{Q_s}{RCNL}$	$2\pi\varepsilon$

A.4 FINITE-LENGTH BEARINGS

When $1/4 \le L/D \le 2$, the solution of the of Reynolds equation has been obtained by Reason and Narang [22]. They made use of both long- and short-bearing theories. The pressure and various performance parameters are displayed in Table A.4. These parameters are expressed in terms of two quantities I_s and I_c. Values of these quantities are presented in Table A.5.

Table A.4 Pressure and Performance Parameters of the Combined Solution Approximation [15-16].

Performance parameter	Equation
$\frac{p}{12\pi\mu N}\left(\frac{C}{R}\right)^2$	$\dfrac{\frac{1}{2}\left(\frac{L}{D}\right)^2(1-\zeta^2)\frac{\varepsilon\sin\theta}{(1+\varepsilon\cos\theta)^3}}{1+\left(\frac{L}{D}\right)^2\frac{(2+\varepsilon^2)(1-\zeta^2)}{2(1+\varepsilon\cos\theta)(2+\varepsilon\cos\theta)}}$
$\frac{W_R}{3\mu VL}\left(\frac{C}{R}\right)^2$	$-2I_c$
$\frac{W_T}{3\mu VL}\left(\frac{C}{R}\right)^2$	$2I_s$
ϕ	$\tan^{-1}\left(\frac{-I_s}{I_c}\right)$
S	$\frac{1}{6\pi\sqrt{I_s^2+I_c^2}}$
$\frac{F}{\mu VL}\frac{C}{R}$	$3\varepsilon I_s+\frac{2\pi}{\sqrt{1-\varepsilon^2}}$
$\left(\frac{R}{C}\right)(f)$	$6\pi S\left(\frac{\varepsilon I_S}{2}+\frac{\pi}{3\sqrt{1-\varepsilon^2}}\right)$

$\dfrac{Q_{0,\pi}}{RCNL}$ †	$\pi\left[1+\varepsilon\mp\varepsilon E\left(1-\dfrac{2E}{\sqrt{1+2E}}\tanh^{-1}\dfrac{1}{\sqrt{1+2E}}\right)\left(\dfrac{L}{D}\right)^2\right]$ where $E=\dfrac{(1\pm\varepsilon)(2\pm\varepsilon)}{(2+\varepsilon^2)}\left(\dfrac{D}{L}\right)^2$
$\dfrac{Q_\pi}{Q_0}$	$1-\dfrac{Q_\pi}{Q_0}$
$\dfrac{J\pi C_{pl}\Delta T}{P}$	$\dfrac{1}{1-\frac{1}{2}Q_\pi/Q_0}\dfrac{4\pi(R/C)f}{Q_0/(RCNL)}$

†For Q_0 (flow through maximum film thickness at $\theta = 0$) use top signs; for Q_π (flow through minimum film thickness at $\theta = \pi$) use lower signs.

Table A.5 Values of I_s and I_c for Various values of L/D and ε [22].

Values of Integral $I_s\left(\frac{L}{D},\varepsilon\right)$

ε	$L/D=2$	$L/D=1\frac{1}{2}$	$L/D=1$	$L/D=\frac{3}{4}$	$L/D=\frac{1}{2}$	$L/D=\frac{1}{4}$	$L/D=\infty$
0.1	0.0839	0.0636	0.0380	0.0244	0.0120	0.0032	0.1570
0.2	0.1705	0.1300	0.0783	0.0505	0.0251	0.0067	0.3143
0.3	0.2628	0.2023	0.1236	0.0804	0.0404	0.0109	0.4727
0.4	0.3649	0.2847	0.1776	0.1172	0.0597	0.0164	0.6347
0.5	0.4831	0.3835	0.2462	0.1656	0.0862	0.0241	0.8061
0.6	0.6291	0.5102	0.3386	0.2345	0.1259	0.0363	0.9983
0.7	0.8266	0.6878	0.4793	0.3430	0.1927	0.0582	1.2366
0.8	1.1380	0.9771	0.7220	0.5425	0.3264	0.1071	1.5866
0.9	1.8137	1.6235	1.3002	1.0499	0.7079	0.2761	2.3083
0.95	2.7600	2.5455	2.1632	1.8467	1.3712	0.6429	3.2913
0.99	3.8092	3.5708	3.1293	2.7442	2.1291	1.0719	4.3763

Values of Integral $I_c\left(\frac{L}{D},\varepsilon\right)$

ε	$L/D=2$	$L/D=1\frac{1}{2}$	$L/D=1$	$L/D=\frac{3}{4}$	$L/D=\frac{1}{2}$	$L/D=\frac{1}{4}$	$L/D=\infty$
0.1	−0.0076	−0.0063	−0.0041	−0.0028	−0.0014	−0.0004	−0.0100
0.2	−0.0312	−0.0259	−0.0174	−0.0118	−0.0062	−0.0017	−0.0408
0.3	−0.0733	−0.0615	−0.0419	−0.0289	−0.0153	−0.0043	−0.0946
0.4	−0.1391	−0.1183	−0.0825	−0.0579	−0.0312	−0.0089	−0.1763
0.5	−0.2391	−0.2065	−0.1484	−0.1065	−0.0591	−0.0174	−0.2962
0.6	−0.3949	−0.3474	−0.2590	−0.1917	−0.1105	−0.0338	−0.4766
0.7	−0.6586	−0.5816	−0.4612	−0.3549	−0.2161	−0.0703	−0.7717
0.8	−1.1891	−1.0941	−0.8997	−0.7283	−0.4797	−0.1732	−1.3467
0.9	−2.7932	−2.6461	−2.3269	−2.0172	−1.4990	−0.6644	−3.0339
0.95	−6.0396	−5.8315	−5.3621	−4.8773	−3.9787	−2.1625	−6.3776
0.99	−9.8095	−9.5362	−3.9045	−8.2294	−6.9073	−3.9626	−10.2526

B: RESULTS OF PARTIAL JOURNAL BEARING PROGRAM FOR MAXIMUM LOAD AND MINIMUM FRICTION

B.1 BASED ON MAXIMUM LOAD

Partial Bearing Design Specifications

The load on the bearing, W = 32.00 kN

The speed of the journal, N = 60.000 rps

The bearing unit load, P = 1.380 MPa

The bearing-application for clearance is No. 3. General machine practice - rotating motion

The inlet temperature, Tin = 43.300 deg C

The decision-criterion for utility is No. 1. Based on maximum load

Design Alternatives Are

1. L/D = 1.0
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
180 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 49.67 deg. C
Viscosity of the Lubricant, Mu = 20.32 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.200
Radial Clearance, C = 0.160 mm
Minimum film thickness, hmin = 0.083 mm
Eccentricity Ratio, e/C = 0.480; Eccentricity, e = 0.077 mm
Position of minimum film thickness = 51.00 degree
Coefficient of friction, f = 0.0063
Torque required = 15.333 N.m; Power Lost = 5782.89 W
Total Oil Supplied = 377511.7 cubic mm per sec
Side Leakage = 184980.7 cubic mm per sec
Maximum Film Pressure = 3.136 MPa
Position of Maximum Film Pressure = 12.40 degree
Terminating Position of Film = 74.00 degree
Starting Position of Film = 38.50 degree
The dimensionless velocity, ws = 1.52
The Utility value = 3.31

2. L/D = 1.5
   Optimization Condition: The Maximum Load
   Design Based on SAE 10 Oil
   180 deg. Partial Bearing
   Length, L = 187 mm; Diameter, D = 124.67 mm
   Operating Temperature, Top = 48.23 deg. C
   Viscosity of the Lubricant, Mu = 21.28 mPa.s
   Sommerfeld number (Bearing ch. no.), S = 0.157
   Radial Clearance, C = 0.151 mm
   Minimum film thickness, hmin = 0.084 mm
   Eccentricity Ratio, e/C = 0.442; Eccentricity, e = 0.067 mm
   Position of minimum film thickness = 51.94 degree
   Coefficient of friction, f = 0.0055
   Torque required = 11.041 N.m; Power Lost = 4163.93 W
   Total Oil Supplied = 330203.0 cubic mm per sec
   Side Leakage = 123165.7 cubic mm per sec
   Maximum Film Pressure = 2.760 MPa
   Position of Maximum Film Pressure = 11.07 degree
   Terminating Position of Film = 79.14 degree
   Starting Position of Film = 37.56 degree
   The dimensionless velocity, ws = 1.48
   The Utility value = 3.40

3. L/D = 2.0
   Optimization Condition: The Maximum Load
   Design Based on SAE 10 Oil
   180 deg. Partial Bearing
   Length, L = 215 mm; Diameter, D = 107.50 mm
   Operating Temperature, Top = 47.67 deg. C
   Viscosity of the Lubricant, Mu = 21.71 mPa.s
   Sommerfeld number (Bearing ch. no.), S = 0.140
   Radial Clearance, C = 0.140 mm
   Minimum film thickness, hmin = 0.081 mm
   Eccentricity Ratio, e/C = 0.422; Eccentricity, e = 0.059 mm
   Position of minimum film thickness = 52.23 degree
   Coefficient of friction, f = 0.0052
   Torque required = 8.887 N.m; Power Lost = 3351.54 W
   Total Oil Supplied = 285446.9 cubic mm per sec
   Side Leakage = 85348.6 cubic mm per sec
   Maximum Film Pressure = 2.546 MPa
   Position of Maximum Film Pressure = 10.22 degree
   Terminating Position of Film = 81.80 degree
   Starting Position of Film = 37.32 degree
   The dimensionless velocity, ws = 1.42
   The Utility value = 3.32

4. L/D = 1.0
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 50.27 deg. C
Viscosity of the Lubricant, Mu = 19.74 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.210
Radial Clearance, C = 0.154 mm
Minimum film thickness, hmin = 0.071 mm
Eccentricity Ratio, e/C = 0.540; Eccentricity, e = 0.083 mm
Position of minimum film thickness = 37.50 degree
Coefficient of friction, f = 0.0051
Torque required = 12.291 N.m; Power Lost = 4635.59 W
Total Oil Supplied = 255580.9 cubic mm per sec
Side Leakage = 89453.3 cubic mm per sec
Maximum Film Pressure = 3.750 MPa
Position of Maximum Film Pressure = 5.60 degree
Terminating Position of Film = 60.00 degree
Starting Position of Film = 82.00 degree
The dimensionless velocity, ws = 1.49
The Utility value = 3.37

5. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.14 deg. C
Viscosity of the Lubricant, Mu = 20.65 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.180
Radial Clearance, C = 0.139 mm
Minimum film thickness, hmin = 0.068 mm
Eccentricity Ratio, e/C = 0.515; Eccentricity, e = 0.072 mm
Position of minimum film thickness = 37.29 degree
Coefficient of friction, f = 0.0046
Torque required = 9.226 N.m; Power Lost = 3479.60 W
Total Oil Supplied = 218878.4 cubic mm per sec
Side Leakage = 55595.1 cubic mm per sec
Maximum Film Pressure = 3.391 MPa
Position of Maximum Film Pressure = 4.75 degree
Terminating Position of Film = 61.18 degree
Starting Position of Film = 82.54 degree
The dimensionless velocity, ws = 1.42
The Utility value = 3.32

6. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
180 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 48.39 deg. C
Viscosity of the Lubricant, Mu = 21.17 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.159 mm
Minimum film thickness, hmin = 0.080 mm
Eccentricity Ratio, e/C = 0.497; Eccentricity, e = 0.079 mm
Position of minimum film thickness = 49.62 degree
Coefficient of friction, f = 0.0055
Torque required = 11.007 N.m; Power Lost = 4151.23 W
Total Oil Supplied = 338240.6 cubic mm per sec
Side Leakage = 135296.2 cubic mm per sec
Maximum Film Pressure = 2.805 MPa
Position of Maximum Film Pressure = 12.24 degree
Terminating Position of Film = 76.44 degree
Starting Position of Film = 40.14 degree
The dimensionless velocity, ws = 1.52
The Utility value = 3.31

7. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
180 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 47.95 deg. C
Viscosity of the Lubricant, Mu = 21.49 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.133
Radial Clearance, C = 0.142 mm
Minimum film thickness, hmin = 0.076 mm
Eccentricity Ratio, e/C = 0.469; Eccentricity, e = 0.067 mm
Position of minimum film thickness = 50.38 degree
Coefficient of friction, f = 0.0052
Torque required = 8.963 N.m; Power Lost = 3380.23 W
Total Oil Supplied = 283502.2 cubic mm per sec
Side Leakage = 89586.7 cubic mm per sec
Maximum Film Pressure = 2.575 MPa
Position of Maximum Film Pressure = 11.46 degree
Terminating Position of Film = 80.03 degree
Starting Position of Film = 39.39 degree
The dimensionless velocity, ws = 1.44
The Utility value = 3.34

8. L/D = 1.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 49.99 deg. C
Viscosity of the Lubricant, Mu = 20.13 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.160
Radial Clearance, C = 0.178 mm
Minimum film thickness, hmin = 0.071 mm
Eccentricity Ratio, e/C = 0.600; Eccentricity, e = 0.107 mm
Position of minimum film thickness = 35.50 degree
Coefficient of friction, f = 0.0050
Torque required = 12.173 N.m; Power Lost = 4590.82 W
Total Oil Supplied = 274747.4 cubic mm per sec
Side Leakage = 105503.0 cubic mm per sec
Maximum Film Pressure = 3.844 MPa
Position of Maximum Film Pressure = 6.60 degree
Terminating Position of Film = 60.00 degree
Starting Position of Film = 84.50 degree
The dimensionless velocity, ws = 1.61
The Utility value = 3.30

9. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.02 deg. C
Viscosity of the Lubricant, Mu = 20.73 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.146
Radial Clearance, C = 0.155 mm
Minimum film thickness, hmin = 0.068 mm
Eccentricity Ratio, e/C = 0.563; Eccentricity, e = 0.087 mm
Position of minimum film thickness = 35.94 degree
Coefficient of friction, f = 0.0046
Torque required = 9.134 N.m; Power Lost = 3444.76 W
Total Oil Supplied = 228546.1 cubic mm per sec
Side Leakage = 62850.2 cubic mm per sec
Maximum Film Pressure = 3.433 MPa
Position of Maximum Film Pressure = 5.69 degree
Terminating Position of Film = 62.19 degree
Starting Position of Film = 84.06 degree
The dimensionless velocity, ws = 1.50
The Utility value = 3.33

10. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 48.63 deg. C
Viscosity of the Lubricant, Mu = 20.99 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.137 mm
Minimum film thickness, hmin = 0.062 mm
Eccentricity Ratio, e/C = 0.546; Eccentricity, e = 0.075 mm
Position of minimum film thickness = 36.04 degree
Coefficient of friction, f = 0.0044
Torque required = 7.518 N.m; Power Lost = 2835.49 W
Total Oil Supplied = 193535.6 cubic mm per sec
Side Leakage = 41416.6 cubic mm per sec
Maximum Film Pressure = 3.217 MPa
Position of Maximum Film Pressure = 5.22 degree
Terminating Position of Film = 62.55 degree
Starting Position of Film = 83.96 degree
The dimensionless velocity, ws = 1.41
The Utility value = 3.27

11. L/D = 1.0
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 49.67 deg. C
Viscosity of the Lubricant, Mu = 27.64 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.200
Radial Clearance, C = 0.186 mm
Minimum film thickness, hmin = 0.097 mm
Eccentricity Ratio, e/C = 0.480; Eccentricity, e = 0.089 mm
Position of minimum film thickness = 51.00 degree
Coefficient of friction, f = 0.0074
Torque required = 17.885 N.m; Power Lost = 6745.07 W
Total Oil Supplied = 440323.6 cubic mm per sec
Side Leakage = 215758.6 cubic mm per sec
Maximum Film Pressure = 3.136 MPa
Position of Maximum Film Pressure = 12.40 degree
Terminating Position of Film = 74.00 degree
Starting Position of Film = 38.50 degree
The dimensionless velocity, ws = 1.64
The Utility value = 3.32

12. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 48.23 deg. C
Viscosity of the Lubricant, Mu = 29.58 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.157
Radial Clearance, C = 0.178 mm
Minimum film thickness, hmin = 0.100 mm
Eccentricity Ratio, e/C = 0.442; Eccentricity, e = 0.079 mm
Position of minimum film thickness = 51.94 degree
Coefficient of friction, f = 0.0065
Torque required = 13.017 N.m; Power Lost = 4909.09 W
Total Oil Supplied = 389294.6 cubic mm per sec
Side Leakage = 145206.9 cubic mm per sec
Maximum Film Pressure = 2.760 MPa
Position of Maximum Film Pressure = 11.07 degree
Terminating Position of Film = 79.14 degree
Starting Position of Film = 37.56 degree
The dimensionless velocity, ws = 1.61
The Utility value = 3.37

13. L/D = 2.0
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 47.67 deg. C
Viscosity of the Lubricant, Mu = 30.40 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.140
Radial Clearance, C = 0.165 mm
Minimum film thickness, hmin = 0.095 mm
Eccentricity Ratio, e/C = 0.422; Eccentricity, e = 0.070 mm
Position of minimum film thickness = 52.23 degree
Coefficient of friction, f = 0.0061
Torque required = 10.516 N.m; Power Lost = 3966.07 W
Total Oil Supplied = 337786.4 cubic mm per sec
Side Leakage = 100998.1 cubic mm per sec
Maximum Film Pressure = 2.546 MPa
Position of Maximum Film Pressure = 10.22 degree
Terminating Position of Film = 81.80 degree
Starting Position of Film = 37.32 degree
The dimensionless velocity, ws = 1.55
The Utility value = 3.32

14. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.14 deg. C
Viscosity of the Lubricant, Mu = 28.33 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.180
Radial Clearance, C = 0.163 mm
Minimum film thickness, hmin = 0.079 mm
Eccentricity Ratio, e/C = 0.515; Eccentricity, e = 0.084 mm
Position of minimum film thickness = 37.29 degree
Coefficient of friction, f = 0.0054
Torque required = 10.806 N.m; Power Lost = 4075.36 W
Total Oil Supplied = 256353.8 cubic mm per sec
Side Leakage = 65113.9 cubic mm per sec
Maximum Film Pressure = 3.391 MPa
Position of Maximum Film Pressure = 4.75 degree
Terminating Position of Film = 61.18 degree
Starting Position of Film = 82.54 degree
The dimensionless velocity, ws = 1.54
The Utility value = 3.31

15. L/D = 2.0
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 48.71 deg. C
Viscosity of the Lubricant, Mu = 28.90 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.168
Radial Clearance, C = 0.147 mm
Minimum film thickness, hmin = 0.073 mm
Eccentricity Ratio, e/C = 0.503; Eccentricity, e = 0.074 mm
Position of minimum film thickness = 37.17 degree
Coefficient of friction, f = 0.0052
Torque required = 8.915 N.m; Power Lost = 3362.09 W
Total Oil Supplied = 220683.6 cubic mm per sec
Side Leakage = 43695.3 cubic mm per sec
Maximum Film Pressure = 3.187 MPa
Position of Maximum Film Pressure = 4.36 degree
Terminating Position of Film = 61.37 degree
Starting Position of Film = 82.78 degree
The dimensionless velocity, ws = 1.46
The Utility value = 3.35

16. L/D = 1.5

Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 48.39 deg. C
Viscosity of the Lubricant, Mu = 29.35 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.188 mm
Minimum film thickness, hmin = 0.094 mm
Eccentricity Ratio, e/C = 0.497; Eccentricity, e = 0.093 mm
Position of minimum film thickness = 49.62 degree
Coefficient of friction, f = 0.0065
Torque required = 12.962 N.m; Power Lost = 4888.70 W
Total Oil Supplied = 398328.6 cubic mm per sec
Side Leakage = 159331.5 cubic mm per sec
Maximum Film Pressure = 2.805 MPa
Position of Maximum Film Pressure = 12.24 degree
Terminating Position of Film = 76.44 degree
Starting Position of Film = 40.14 degree
The dimensionless velocity, ws = 1.65
The Utility value = 3.32

17. L/D = 2.0

Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 47.95 deg. C
Viscosity of the Lubricant, Mu = 29.98 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.133
Radial Clearance, C = 0.168 mm
Minimum film thickness, hmin = 0.089 mm
Eccentricity Ratio, e/C = 0.469; Eccentricity, e = 0.079 mm
Position of minimum film thickness = 50.38 degree
Coefficient of friction, f = 0.0062
Torque required = 10.587 N.m; Power Lost = 3992.68 W
Total Oil Supplied = 334869.1 cubic mm per sec
Side Leakage = 105818.6 cubic mm per sec
Maximum Film Pressure = 2.575 MPa
Position of Maximum Film Pressure = 11.46 degree
Terminating Position of Film = 80.03 degree
Starting Position of Film = 39.39 degree
The dimensionless velocity, ws = 1.56
The Utility value = 3.32

18. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.02 deg. C
Viscosity of the Lubricant, Mu = 28.49 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.146
Radial Clearance, C = 0.182 mm
Minimum film thickness, hmin = 0.079 mm
Eccentricity Ratio, e/C = 0.563; Eccentricity, e = 0.102 mm
Position of minimum film thickness = 35.94 degree
Coefficient of friction, f = 0.0054
Torque required = 10.707 N.m; Power Lost = 4038.20 W
Total Oil Supplied = 267918.0 cubic mm per sec
Side Leakage = 73677.4 cubic mm per sec
Maximum Film Pressure = 3.433 MPa
Position of Maximum Film Pressure = 5.69 degree
Terminating Position of Film = 62.19 degree
Starting Position of Film = 84.06 degree
The dimensionless velocity, ws = 1.62
The Utility value = 3.27

19. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 48.63 deg. C
Viscosity of the Lubricant, Mu = 29.01 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.161 mm
Minimum film thickness, hmin = 0.073 mm
Eccentricity Ratio, e/C = 0.546; Eccentricity, e = 0.088 mm
Position of minimum film thickness = 36.04 degree
Coefficient of friction, f = 0.0051
Torque required = 8.839 N.m; Power Lost = 3333.45 W
Total Oil Supplied = 227523.3 cubic mm per sec
Side Leakage = 48690.0 cubic mm per sec
Maximum Film Pressure = 3.217 MPa
Position of Maximum Film Pressure = 5.22 degree
Terminating Position of Film = 62.55 degree
Starting Position of Film = 83.96 degree
The dimensionless velocity, ws = 1.53
The Utility value = 3.31

20. L/D = 2.0

Optimization Condition: The Maximum Load
Design Based on SAE 30 Oil
120 deg. Partial Bearing
Length, L = 215 mm
Diameter, D = 107.50 mm
Operating Temperature, Top = 48.71 deg. C
Viscosity of the Lubricant, Mu = 50.13 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.168
Radial Clearance, C = 0.194 mm
Minimum film thickness, hmin = 0.096 mm
Eccentricity Ratio, e/C = 0.503
Eccentricity, e = 0.097 mm
Position of minimum film thickness = 37.17 degree
Coefficient of friction, f = 0.0068
Torque required = 11.740 N.m
Power Lost = 4427.75 W
Total Oil Supplied = 290632.6 cubic mm per sec
Side Leakage = 57545.2 cubic mm per sec
Maximum Film Pressure = 3.187 MPa
Position of Maximum Film Pressure = 4.36 degree
Terminating Position of Film = 61.37 degree
Starting Position of Film = 82.78 degree
The dimensionless velocity, ws = 1.68
The Utility value = 3.24

B.2 BASED ON MINIMUM FRICTION

Partial Bearing Design Specifications

The load on the bearing, W = 32.00 kN

The speed of the journal, N = 60.000 rps

The bearing unit load, P = 1.380 MPa

The bearing-application for clearance is No. 3. General machine practice - rotating motion

The inlet temperature, Tin = 43.300 deg C

The decision-criterion for utility is No. 2. Based on minimum friction

Design Alternatives Are

1. L/D = 1.5
   Optimization Condition: The Maximum Load
   Design Based on SAE 10 Oil
   180 deg. Partial Bearing
   Length, L = 187 mm; Diameter, D = 124.67 mm
   Operating Temperature, Top = 48.23 deg. C
   Viscosity of the Lubricant, Mu = 21.28 mPa.s
   Sommerfeld number (Bearing ch. no.), S = 0.157
   Radial Clearance, C = 0.151 mm
   Minimum film thickness, hmin = 0.084 mm
   Eccentricity Ratio, e/C = 0.442; Eccentricity, e = 0.067 mm
   Position of minimum film thickness = 51.94 degree
   Coefficient of friction, f = 0.0055
   Torque required = 11.041 N.m; Power Lost = 4163.93 W
   Total Oil Supplied = 330203.0 cubic mm per sec
   Side Leakage = 123165.7 cubic mm per sec
   Maximum Film Pressure = 2.760 MPa
   Position of Maximum Film Pressure = 11.07 degree
   Terminating Position of Film = 79.14 degree
   Starting Position of Film = 37.56 degree
   The dimensionless velocity, ws = 1.48
   The Utility value = 4.45

2. L/D = 2.0
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
180 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 47.67 deg. C
Viscosity of the Lubricant, Mu = 21.71 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.140
Radial Clearance, C = 0.140 mm
Minimum film thickness, hmin = 0.081 mm
Eccentricity Ratio, e/C = 0.422
Eccentricity, e = 0.059 mm
Position of minimum film thickness = 52.23 degree
Coefficient of friction, f = 0.0052
Torque required = 8.887 N.m
Power Lost = 3351.54 W
Total Oil Supplied = 285446.9 cubic mm per sec
Side Leakage = 85348.6 cubic mm per sec
Maximum Film Pressure = 2.546 MPa
Position of Maximum Film Pressure = 10.22 degree
Terminating Position of Film = 81.80 degree
Starting Position of Film = 37.32 degree
The dimensionless velocity, ws = 1.42
The Utility value = 4.46

3. L/D = 1.0
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 50.27 deg. C
Viscosity of the Lubricant, Mu = 19.74 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.210
Radial Clearance, C = 0.154 mm; Min. film thickness, hmin = 0.071 mm
Eccentricity Ratio, e/C = 0.540; Eccentricity, e = 0.083 mm
Position of minimum film thickness = 37.50 degree
Coefficient of friction, f = 0.0051
Torque required = 12.291 N.m; Power Lost = 4635.59 W
Total Oil Supplied = 255580.9 cubic mm per sec
Side Leakage = 89453.3 cubic mm per sec
Maximum Film Pressure = 3.750 MPa
Position of Maximum Film Pressure = 5.60 degree
Terminating Position of Film = 60.00 degree
Starting Position of Film = 82.00 degree
The dimensionless velocity, ws (used by Vance) = 1.49
The Utility value = 4.45

4. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.14 deg. C
Viscosity of the Lubricant, Mu = 20.65 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.180
Radial Clearance, C = 0.139 mm
Minimum film thickness, hmin = 0.068 mm
Eccentricity Ratio, e/C = 0.515; Eccentricity, e = 0.072 mm
Position of minimum film thickness = 37.29 degree
Coefficient of friction, f = 0.0046
Torque required = 9.226 N.m; Power Lost = 3479.60 W
Total Oil Supplied = 218878.4 cubic mm per sec
Side Leakage = 55595.1 cubic mm per sec
Maximum Film Pressure = 3.391 MPa
Position of Maximum Film Pressure = 4.75 degree
Terminating Position of Film = 61.18 degree
Starting Position of Film = 82.54 degree
The dimensionless velocity, ws = 1.42
The Utility value = 4.49

5. L/D = 1.0
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
60 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 53.77 deg. C
Viscosity of the Lubricant, Mu = 16.83 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.130
Radial Clearance, C = 0.180 mm
Minimum film thickness, hmin = 0.041 mm
Eccentricity Ratio, e/C = 0.770; Eccentricity, e = 0.139 mm
Position of minimum film thickness = 19.00 degree
Coefficient of friction, f = 0.0038
Torque required = 9.233 N.m; Power Lost = 3482.12 W
Total Oil Supplied = 122490.3 cubic mm per sec
Side Leakage = 23273.2 cubic mm per sec
Maximum Film Pressure = 6.216 MPa
Position of Maximum Film Pressure = 1.90 degree
Terminating Position of Film = 30.00 degree
Starting Position of Film = 130.70 degree
The dimensionless velocity, ws (used by Vance) = 1.62
The Utility value = 4.34

6. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
60 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 53.10 deg. C
Viscosity of the Lubricant, Mu = 17.34 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.129
Radial Clearance, C = 0.151 mm
Minimum film thickness, hmin = 0.036 mm
Eccentricity Ratio, e/C = 0.762; Eccentricity, e = 0.115 mm
Position of minimum film thickness = 18.87 degree
Coefficient of friction, f = 0.0036
Torque required = 7.271 N.m; Power Lost = 2742.39 W
Total Oil Supplied = 100327.9 cubic mm per sec
Side Leakage = 13042.6 cubic mm per sec
Maximum Film Pressure = 5.774 MPa
Position of Maximum Film Pressure = 1.77 degree
Terminating Position of Film = 29.96 degree
Starting Position of Film = 130.85 degree
The dimensionless velocity, ws = 1.48
The Utility value = 4.35

7. L/D = 2.0
Optimization Condition: The Maximum Load
Design Based on SAE 10 Oil
60 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 52.83 deg. C
Viscosity of the Lubricant, Mu = 17.55 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.129
Radial Clearance, C = 0.131 mm
Minimum film thickness, hmin = 0.032 mm
Eccentricity Ratio, e/C = 0.758; Eccentricity, e = 0.099 mm
Position of minimum film thickness = 18.83 degree
Coefficient of friction, f = 0.0036
Torque required = 6.153 N.m; Power Lost = 2320.39 W
Total Oil Supplied = 84893.3 cubic mm per sec
Side Leakage = 8319.5 cubic mm per sec
Maximum Film Pressure = 5.520 MPa
Position of Maximum Film Pressure = 1.70 degree
Terminating Position of Film = 29.95 degree
Starting Position of Film = 130.93 degree
The dimensionless velocity, ws = 1.38
The Utility value = 4.31

8. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
180 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 48.39 deg. C
Viscosity of the Lubricant, Mu = 21.17 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.159 mm
Minimum film thickness, hmin = 0.080 mm
Eccentricity Ratio, e/C = 0.497; Eccentricity, e = 0.079 mm
Position of minimum film thickness = 49.62 degree
Coefficient of friction, f = 0.0055
Torque required = 11.007 N.m; Power Lost = 4151.23 W
Total Oil Supplied = 338240.6 cubic mm per sec
Side Leakage = 135296.2 cubic mm per sec
Maximum Film Pressure = 2.805 MPa
Position of Maximum Film Pressure = 12.24 degree
Terminating Position of Film = 76.44 degree
Starting Position of Film = 40.14 degree
The dimensionless velocity, ws = 1.52
The Utility value = 4.39

9. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
180 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 47.95 deg. C
Viscosity of the Lubricant, Mu = 21.49 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.133
Radial Clearance, C = 0.142 mm
Minimum film thickness, hmin = 0.076 mm
Eccentricity Ratio, e/C = 0.469; Eccentricity, e = 0.067 mm
Position of minimum film thickness = 50.38 degree
Coefficient of friction, f = 0.0052
Torque required = 8.963 N.m; Power Lost = 3380.23 W
Total Oil Supplied = 283502.2 cubic mm per sec
Side Leakage = 89586.7 cubic mm per sec
Maximum Film Pressure = 2.575 MPa
Position of Maximum Film Pressure = 11.46 degree
Terminating Position of Film = 80.03 degree
Starting Position of Film = 39.39 degree
The dimensionless velocity, ws = 1.44
The Utility value = 4.48

10. L/D = 1.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 49.99 deg. C
Viscosity of the Lubricant, Mu = 20.13 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.160
Radial Clearance, C = 0.178 mm
Minimum film thickness, hmin = 0.071 mm
Eccentricity Ratio, e/C = 0.600; Eccentricity, e = 0.107 mm
Position of minimum film thickness = 35.50 degree
Coefficient of friction, f = 0.0050
Torque required = 12.173 N.m; Power Lost = 4590.82 W
Total Oil Supplied = 274747.4 cubic mm per sec
Side Leakage = 105503.0 cubic mm per sec
Maximum Film Pressure = 3.844 MPa
Position of Maximum Film Pressure = 6.60 degree
Terminating Position of Film = 60.00 degree
Starting Position of Film = 84.50 degree
The dimensionless velocity, ws = 1.61
The Utility value = 4.38

11. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.02 deg. C
Viscosity of the Lubricant, Mu = 20.73 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.146
Radial Clearance, C = 0.155 mm
Minimum film thickness, hmin = 0.068 mm
Eccentricity Ratio, e/C = 0.563; Eccentricity, e = 0.087 mm
Position of minimum film thickness = 35.94 degree
Coefficient of friction, f = 0.0046
Torque required = 9.134 N.m; Power Lost = 3444.76 W
Total Oil Supplied = 228546.1 cubic mm per sec
Side Leakage = 62850.2 cubic mm per sec
Maximum Film Pressure = 3.433 MPa
Position of Maximum Film Pressure = 5.69 degree
Terminating Position of Film = 62.19 degree
Starting Position of Film = 84.06 degree
The dimensionless velocity, ws = 1.50
The Utility value = 4.50

12. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
120 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 48.63 deg. C
Viscosity of the Lubricant, Mu = 20.99 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.137 mm
Minimum film thickness, hmin = 0.062 mm
Eccentricity Ratio, e/C = 0.546; Eccentricity, e = 0.075 mm
Position of minimum film thickness = 36.04 degree
Coefficient of friction, f = 0.0044
Torque required = 7.518 N.m; Power Lost = 2835.49 W
Total Oil Supplied = 193535.6 cubic mm per sec
Side Leakage = 41416.6 cubic mm per sec
Maximum Film Pressure = 3.217 MPa
Position of Maximum Film Pressure = 5.22 degree
Terminating Position of Film = 62.55 degree
Starting Position of Film = 83.96 degree
The dimensionless velocity, ws = 1.41
The Utility value = 4.44

13. L/D = 1.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
60 deg. Partial Bearing
Length, L = 152 mm; Diameter, D = 152.00 mm
Operating Temperature, Top = 53.68 deg. C
Viscosity of the Lubricant, Mu = 16.90 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.120
Radial Clearance, C = 0.188 mm
Minimum film thickness, hmin = 0.041 mm
Eccentricity Ratio, e/C = 0.780; Eccentricity, e = 0.147 mm
Position of minimum film thickness = 18.70 degree
Coefficient of friction, f = 0.0038
Torque required = 9.328 N.m; Power Lost = 3518.07 W
Total Oil Supplied = 121229.5 cubic mm per sec
Side Leakage = 23397.3 cubic mm per sec
Maximum Film Pressure = 6.301 MPa
Position of Maximum Film Pressure = 2.00 degree
Terminating Position of Film = 30.00 degree
Starting Position of Film = 131.00 degree
The dimensionless velocity, ws = 1.65
The Utility value = 4.34

14. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
60 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 53.05 deg. C
Viscosity of the Lubricant, Mu = 17.38 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.121
Radial Clearance, C = 0.156 mm
Minimum film thickness, hmin = 0.036 mm
Eccentricity Ratio, e/C = 0.769; Eccentricity, e = 0.120 mm
Position of minimum film thickness = 18.70 degree
Coefficient of friction, f = 0.0037
Torque required = 7.427 N.m; Power Lost = 2801.20 W
Total Oil Supplied = 99795.4 cubic mm per sec
Side Leakage = 12474.4 cubic mm per sec
Maximum Film Pressure = 5.823 MPa
Position of Maximum Film Pressure = 1.78 degree
Terminating Position of Film = 30.26 degree
Starting Position of Film = 130.99 degree
The dimensionless velocity, ws = 1.50
The Utility value = 4.33

15. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 10 Oil
60 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 52.78 deg. C
Viscosity of the Lubricant, Mu = 17.59 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.120
Radial Clearance, C = 0.136 mm
Minimum film thickness, hmin = 0.032 mm
Eccentricity Ratio, e/C = 0.765; Eccentricity, e = 0.104 mm
Position of minimum film thickness = 18.66 degree
Coefficient of friction, f = 0.0037
Torque required = 6.309 N.m; Power Lost = 2379.46 W
Total Oil Supplied = 84865.6 cubic mm per sec
Side Leakage = 7807.6 cubic mm per sec
Maximum Film Pressure = 5.565 MPa
Position of Maximum Film Pressure = 1.70 degree
Terminating Position of Film = 30.30 degree
Starting Position of Film = 131.07 degree
The dimensionless velocity, ws = 1.40
The Utility value = 4.34

16. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 48.23 deg. C
Viscosity of the Lubricant, Mu = 29.58 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.157
Radial Clearance, C = 0.178 mm
Minimum film thickness, hmin = 0.100 mm
Eccentricity Ratio, e/C = 0.442; Eccentricity, e = 0.079 mm
Position of minimum film thickness = 51.94 degree
Coefficient of friction, f = 0.0065
Torque required = 13.017 N.m; Power Lost = 4909.09 W
Total Oil Supplied = 389294.6 cubic mm per sec
Side Leakage = 145206.9 cubic mm per sec
Maximum Film Pressure = 2.760 MPa
Position of Maximum Film Pressure = 11.07 degree
Terminating Position of Film = 79.14 degree
Starting Position of Film = 37.56 degree
The dimensionless velocity, ws = 1.61
The Utility value = 4.39

17. L/D = 2.0
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 47.67 deg. C
Viscosity of the Lubricant, Mu = 30.40 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.140
Radial Clearance, C = 0.165 mm
Minimum film thickness, hmin = 0.095 mm
Eccentricity Ratio, e/C = 0.422; Eccentricity, e = 0.070 mm
Position of minimum film thickness = 52.23 degree
Coefficient of friction, f = 0.0061
Torque required = 10.516 N.m; Power Lost = 3966.07 W
Total Oil Supplied = 337786.4 cubic mm per sec
Side Leakage = 100998.1 cubic mm per sec
Maximum Film Pressure = 2.546 MPa
Position of Maximum Film Pressure = 10.22 degree
Terminating Position of Film = 81.80 degree
Starting Position of Film = 37.32 degree
The dimensionless velocity, ws = 1.55
The Utility value = 4.37

18. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.14 deg. C
Viscosity of the Lubricant, Mu = 28.33 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.180
Radial Clearance, C = 0.163 mm
Minimum film thickness, hmin = 0.079 mm
Eccentricity Ratio, e/C = 0.515; Eccentricity, e = 0.084 mm
Position of minimum film thickness = 37.29 degree
Coefficient of friction, f = 0.0054
Torque required = 10.806 N.m; Power Lost = 4075.36 W
Total Oil Supplied = 256353.8 cubic mm per sec
Side Leakage = 65113.9 cubic mm per sec
Maximum Film Pressure = 3.391 MPa
Position of Maximum Film Pressure = 4.75 degree
Terminating Position of Film = 61.18 degree
Starting Position of Film = 82.54 degree
The dimensionless velocity, ws = 1.54
The Utility value = 4.39

19. L/D = 2.0
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 48.71 deg. C
Viscosity of the Lubricant, Mu = 28.90 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.168
Radial Clearance, C = 0.147 mm
Minimum film thickness, hmin = 0.073 mm
Eccentricity Ratio, e/C = 0.503; Eccentricity, e = 0.074 mm
Position of minimum film thickness = 37.17 degree
Coefficient of friction, f = 0.0052
Torque required = 8.915 N.m; Power Lost = 3362.09 W
Total Oil Supplied = 220683.6 cubic mm per sec
Side Leakage = 43695.3 cubic mm per sec
Maximum Film Pressure = 3.187 MPa
Position of Maximum Film Pressure = 4.36 degree
Terminating Position of Film = 61.37 degree
Starting Position of Film = 82.78 degree
The dimensionless velocity, ws = 1.46
The Utility value = 4.49

20. L/D = 1.5
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
60 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 53.10 deg. C
Viscosity of the Lubricant, Mu = 23.32 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.129
Radial Clearance, C = 0.175 mm
Minimum film thickness, hmin = 0.042 mm
Eccentricity Ratio, e/C = 0.762; Eccentricity, e = 0.133 mm
Position of minimum film thickness = 18.87 degree
Coefficient of friction, f = 0.0042
Torque required = 8.434 N.m; Power Lost = 3180.75 W
Total Oil Supplied = 116365.0 cubic mm per sec
Side Leakage = 15127.5 cubic mm per sec
Maximum Film Pressure = 5.774 MPa
Position of Maximum Film Pressure = 1.77 degree
Terminating Position of Film = 29.96 degree
Starting Position of Film = 130.85 degree
The dimensionless velocity, ws = 1.59
The Utility value = 4.31

21. L/D = 2.0
Optimization Condition: The Maximum Load
Design Based on SAE 20 Oil
60 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 52.83 deg. C
Viscosity of the Lubricant, Mu = 23.63 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.129
Radial Clearance, C = 0.151 mm
Minimum film thickness, hmin = 0.037 mm
Eccentricity Ratio, e/C = 0.758; Eccentricity, e = 0.115 mm
Position of minimum film thickness = 18.83 degree
Coefficient of friction, f = 0.0042
Torque required = 7.139 N.m; Power Lost = 2692.28 W
Total Oil Supplied = 98499.3 cubic mm per sec
Side Leakage = 9652.9 cubic mm per sec
Maximum Film Pressure = 5.520 MPa
Position of Maximum Film Pressure = 1.70 degree
Terminating Position of Film = 29.95 degree
Starting Position of Film = 130.93 degree
The dimensionless velocity, ws = 1.48
The Utility value = 4.30

22. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 48.39 deg. C
Viscosity of the Lubricant, Mu = 29.35 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.188 mm
Minimum film thickness, hmin = 0.094 mm
Eccentricity Ratio, e/C = 0.497; Eccentricity, e = 0.093 mm
Position of minimum film thickness = 49.62 degree
Coefficient of friction, f = 0.0065
Torque required = 12.962 N.m ; Power Lost = 4888.70 W
Total Oil Supplied = 398328.6 cubic mm per sec
Side Leakage = 159331.5 cubic mm per sec
Maximum Film Pressure = 2.805 MPa
Position of Maximum Film Pressure = 12.24 degree
Terminating Position of Film = 76.44 degree
Starting Position of Film = 40.14 degree
The dimensionless velocity, ws = 1.65
The Utility value = 4.37

23. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
180 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 47.95 deg. C
Viscosity of the Lubricant, Mu = 29.98 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.133
Radial Clearance, C = 0.168 mm
Minimum film thickness, hmin = 0.089 mm
Eccentricity Ratio, e/C = 0.469; Eccentricity, e = 0.079 mm
Position of minimum film thickness = 50.38 degree
Coefficient of friction, f = 0.0062
Torque required = 10.587 N.m; Power Lost = 3992.68 W
Total Oil Supplied = 334869.1 cubic mm per sec
Side Leakage = 105818.6 cubic mm per sec
Maximum Film Pressure = 2.575 MPa
Position of Maximum Film Pressure = 11.46 degree
Terminating Position of Film = 80.03 degree
Starting Position of Film = 39.39 degree
The dimensionless velocity, ws = 1.56
The Utility value = 4.37

24. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 49.02 deg. C
Viscosity of the Lubricant, Mu = 28.49 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.146
Radial Clearance, C = 0.182 mm
Minimum film thickness, hmin = 0.079 mm
Eccentricity Ratio, e/C = 0.563; Eccentricity, e = 0.102 mm
Position of minimum film thickness = 35.94 degree
Coefficient of friction, f = 0.0054
Torque required = 10.707 N.m; Power Lost = 4038.20 W
Total Oil Supplied = 267918.0 cubic mm per sec
Side Leakage = 73677.4 cubic mm per sec
Maximum Film Pressure = 3.433 MPa
Position of Maximum Film Pressure = 5.69 degree
Terminating Position of Film = 62.19 degree
Starting Position of Film = 84.06 degree
The dimensionless velocity, ws = 1.62
The Utility value = 4.35

25. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
120 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 48.63 deg. C
Viscosity of the Lubricant, Mu = 29.01 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.141
Radial Clearance, C = 0.161 mm
Minimum film thickness, hmin = 0.073 mm
Eccentricity Ratio, e/C = 0.546; Eccentricity, e = 0.088 mm
Position of minimum film thickness = 36.04 degree
Coefficient of friction, f = 0.0051
Torque required = 8.839 N.m; Power Lost = 3333.45 W
Total Oil Supplied = 227523.3 cubic mm per sec
Side Leakage = 48690.0 cubic mm per sec
Maximum Film Pressure = 3.217 MPa
Position of Maximum Film Pressure = 5.22 degree
Terminating Position of Film = 62.55 degree
Starting Position of Film = 83.96 degree
The dimensionless velocity, ws = 1.53
The Utility value = 4.45

26. L/D = 1.5
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
60 deg. Partial Bearing
Length, L = 187 mm; Diameter, D = 124.67 mm
Operating Temperature, Top = 53.05 deg. C
Viscosity of the Lubricant, Mu = 23.39 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.121
Radial Clearance, C = 0.181 mm
Minimum film thickness, hmin = 0.042 mm
Eccentricity Ratio, e/C = 0.769; Eccentricity, e = 0.139 mm
Position of minimum film thickness = 18.70 degree
Coefficient of friction, f = 0.0043
Torque required = 8.615 N.m; Power Lost = 3249.20 W
Total Oil Supplied = 115755.9 cubic mm per sec
Side Leakage = 14469.5 cubic mm per sec
Maximum Film Pressure = 5.823 MPa
Position of Maximum Film Pressure = 1.78 degree
Terminating Position of Film = 30.26 degree
Starting Position of Film = 130.99 degree
The dimensionless velocity, ws = 1.62
The Utility value = 4.31

27. L/D = 2.0
Optimization Condition: The Minimum Friction
Design Based on SAE 20 Oil
60 deg. Partial Bearing
Length, L = 215 mm; Diameter, D = 107.50 mm
Operating Temperature, Top = 52.78 deg. C
Viscosity of the Lubricant, Mu = 23.68 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.120
Radial Clearance, C = 0.157 mm
Minimum film thickness, hmin = 0.037 mm
Eccentricity Ratio, e/C = 0.765
Eccentricity, e = 0.120 mm
Position of minimum film thickness = 18.66 degree
Coefficient of friction, f = 0.0043
Torque required = 7.321 N.m; Power Lost = 2761.00 W
Total Oil Supplied = 98473.4 cubic mm per sec
Side Leakage = 9059.6 cubic mm per sec
Maximum Film Pressure = 5.565 MPa
Position of Maximum Film Pressure = 1.70 degree
Terminating Position of Film = 30.30 degree
Starting Position of Film = 131.07 degree
The dimensionless velocity, ws = 1.51
The Utility value = 4.29

28. L/D = 2.0

Optimization Condition: The Maximum Load
Design Based on SAE 30 Oil
120 deg. Partial Bearing
Length, L = 215 mm
Diameter, D = 107.50 mm
Operating Temperature, Top = 48.71 deg. C
Viscosity of the Lubricant, Mu = 50.13 mPa.s
Sommerfeld number (Bearing ch. no.), S = 0.168
Radial Clearance, C = 0.194 mm
Minimum film thickness, hmin = 0.096 mm
Eccentricity Ratio, e/C = 0.503
Eccentricity, e = 0.097 mm
Position of minimum film thickness = 37.17 degree
Coefficient of friction, f = 0.0068
Torque required = 11.740 N.m
Power Lost = 4427.75 W
Total Oil Supplied = 290632.6 cubic mm per sec
Side Leakage = 57545.2 cubic mm per sec
Maximum Film Pressure = 3.187 MPa
Position of Maximum Film Pressure = 4.36 degree
Terminating Position of Film = 61.37 degree
Starting Position of Film = 82.78 degree
The dimensionless velocity, ws (used by Vance) = 1.68
The Utility value = 4.29

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